JIG and FIXTURE
DESIGN MANUAL
ERRATA

(Henriksen: JIG AND FIXTURE DESIGN MANUAL)

Page 66 - Substitute the figure given below for that appearing on the left-hand side of Fig. 6-44.

Page 67 - Right-hand column, 4th paragraph, 9th line which now reads "pressed into the fixture body C, and is threaded with" should instead read "pressed into the nose piece B, and is threaded with."

Page 122 - Left-hand column, Fig. 10-26, transfer the clamp shown as "a" to "b" and similarly transfer the clamp shown as "b" to "a".
JIG and FIXTURE DESIGN MANUAL

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The Use of Metric Units

Dimensions and other data are, as a general rule, given in English units and in metric units. In the text the metric data are put in parentheses following the English data; in tables the metric units are usually placed in separate columns. The accuracy with which the conversions are performed varies with the nature and purpose of the data quoted. Where accurate conversion of dimensions is made, it is based on 1 inch = 25.4 mm EXACT. Several tables for the conversion of inches and millimeters, feet and meters, and pounds and newtons are presented in Appendix IV. Precise inch dimensions, written with three or four decimal places, are converted, as a rule, to the nearest 1/100 or 1/1000 mm. The purpose is to present the result of the conversion in a manner representative of the equivalent level of workshop accuracy. In other cases, i.e., when dimensions include fractions of an inch, approximations are used. For example, 1/2 inch usually is converted to 13 mm. There are also cases where a fairly close approximation would be meaningless, and where it is more realistic to present the result of the conversion in a round number of millimeters. When, for example, a fixture is made with an overall length of 16 inches, then this dimension is obviously chosen by the designer as a convenient round value, and not as the result of an accurate calculation. If the same fixture had been designed in a metric country, the designer would not choose the length as 16 x 25.4 = 406.4 mm but would make it an even 400 mm. Likewise, an American component manufacturer may market an eyebolt 6 inches in length, while a European manufacturer may have an equivalent eyebolt that is 150 mm, not 152.4 mm, long.

Where an American screw thread is converted, it is to the nearest metric screw thread. No attempt is made to convert American standard fits and tolerances. Parts with metric dimensions should be designed with the ISO Limits and Fits; a collection of data for this system is found in Machinery's Handbook, 19th ed., pages 1529 through 1538.

In some cases, such as in dimensioned drawings and their accompanying calculations, no conversion is attempted. To write two different sets of dimensions into the drawings and detailed calculations would be confusing. The purpose of such calculations is to explain the method, rather than to illustrate one particular size of an object. Also, for some of the commercial components concerning a specific American product, only English dimensions are quoted.

Many of the book's equations are of such a nature that conversion is unnecessary since they are equally valid in English and in metric units. Other equations, of an empirical nature, include numerical coefficients the values of which depend on the type of units used. In all such cases, separate equations are given for use with English and with metric units. In most of the numerical examples, the given data as well as the calculated end results are stated in English as well as in metric units.

It should be noted that conversions have been made to units in the International System (SI) which is rapidly becoming the recognized standard throughout the world. Thus the reader will find that the newton (N) and the kilonewton (kN) are the metric units used for force while the gram (g) and the kilogram (kg) are used for weight (mass).
Preface

The book is written as a textbook and reference source, and is meant to be used by the experienced practitioner as well as the beginner, whether he is a technician in industry or a college student.

The author concentrates on three major objectives: (1) to describe the fixture components in full; (2) to present the fundamental principles for efficiently combining the components into successful fixtures; and (3) to apply basic engineering principles to the mechanical and economic analysis of the complete design. These three tasks are supported by a comprehensive description of commercially available fixture components, a four-point, step-by-step method and comprehensive check list for the design procedure, applicable equally to all types of fixtures, and also calculation methods for the stress and deformation analysis of the fixture body and its major components. The use of a variety of calculation methods is demonstrated by numerical examples.

The author has avoided presenting a confusion of detailed drawings of complicated fixtures. Instead, there are 15 actual cases included, ranging from the simplest drill plate to some complex and quite advanced fixtures for milling and other operations. For each category of machining operations, there is a definition of its characteristic fixture requirements and one or more typical examples. In addition, the book includes the design principles for fixtures of the most important non-machining operations, such as welding and assembly.

A number of the line drawings in the book are executed in a recently introduced drawing style in which two line thicknesses are used for edges and contours. The heavier lines indicate the contours of surfaces that are surrounded by air.

With the dominant position of the metric system outside of the United States and the approaching introduction of this system within this country, metric units are used together with the English units throughout the book.

Four informative appendices with illustrations should prove to be helpful to the reader, they are "Measuring Angles in Radians," "Transfer of Tolerances from the Conventional Dimensioning System to the Coordinate System," "Dimensioning of Fixtures," and lastly, "Metric Conversion Tables of Linear Measure."
Introduction

Definition, Purpose, and Advantages

A fixture is a special tool used for locating and firmly holding a workpiece in the proper position during a manufacturing operation. As a general rule it is provided with devices for supporting and clamping the workpiece. In addition, it may also contain devices for guiding the tool prior to or during its actual operation. Thus, a jig is a type of fixture with means for positively guiding and supporting tools for drilling, boring, and related operations. Hence, the drill jig, which is usually fitted with hardened bushings to locate, guide, and support rotating cutting tools.

The origin of jigs and fixtures can be traced back to the Swiss watch and clock industry from which, after proving their usefulness, they spread throughout the entire metalworking industry. Contrary to widespread belief, the recent introduction of the N/C machine tools has not eliminated the need for fixtures; to obtain the full benefit from these machines they should be equipped with fixtures that are simpler in their build-up and, at the same time, more sophisticated in their clamping devices. An example of a fixture on an N/C lathe is shown in Fig. 1-1.

1. The main purpose of a fixture is to locate the work quickly and accurately, support it properly, and hold it securely, thereby ensuring that all parts produced in the same fixture will come out alike within specified limits. In this way accuracy and interchangeability of the parts are provided.

2. It also reduces working time in the various phases of the operation, in the setup and clamping of the work, in the adjustment of the cutting tool to the required dimensions, and during the cutting operation itself by allowing heavier feeds due to more efficient work support.

3. It serves to simplify otherwise complicated operations so that cheaper, relatively unskilled labor may be employed to perform operations previously reserved for skilled mechanics. Jigs and fixtures expand the capacity of standard machine tools to perform special operations, and in many cases, they make it possible to use plain or simplified, and therefore less expensive, machinery instead of costly standard machines. In other words, they turn plain and simple machine tools into high production equipment and convert standard machines into the equivalent of specialized equipment.

4. By maintaining or even improving the interchangeability of the parts, a jig or fixture contributes to a considerable reduction in the cost of assembly, maintenance, and the subsequent supply of spare parts.

In effect, jigs and fixtures reduce costs and improve the potential of standard machines and the quality of the parts produced.

Fig. 1-1. Close-up of an aircraft fuel pump body housing mounted in its fixture on an N/C lathe.
Jigs and fixtures represent an embodiment of the principle of the transformation of skill. The skills of the experienced craftsmen, designers, and engineers are permanently built into the fixture and are thereby made continuously available to the unskilled operator. One important goal is to design a fixture in such a way as to make it foolproof, and thereby contribute to added safety for the operator as well as for the work.

Application and Classification of Jigs and Fixtures

The obvious place for jigs and fixtures is in mass production, where large quantity output offers ample opportunity for recovery of the necessary investment. However, the advantages in the use of jigs and fixtures are so great, and so varied, that these devices have also naturally found their way into the production of parts in limited quantities as well as into manufacturing processes outside of the machine shop, and even outside of the metalworking industry. The many problems of geometry and dimensions encountered within the aircraft and missile industry have greatly accelerated the expanded use of jigs and fixtures.

Within the machine shop, jigs and fixtures are used for the following operations: Boring, Broaching, Drilling, Grinding, Honing, Lapping, Milling, Planing, Profiling, Reaming, Sawing, Shaping, Slotting, Spot-facing, Tapping, and Turning. A systematic master classification of machining fixtures according to the characteristics of the operation is shown in Table 1-1.

Outside of the machine shop, fixtures may be applied to advantage for: Assembling, Bending, Brazing, Heat treating, Inspecting, Riveting, Soldering, Testing, and Welding. Such fixtures can be characterized as manual work fixtures and may be classified as shown in Table 1-2.

This book deals essentially with the design of jigs and fixtures for use with metal-cutting machine tools. Applications outside of this area will be shown by a few characteristic examples.

Design and Economy

Jigs and fixtures are special tools in the sense that each tool is, generally, designed and built specifically for making one part only and for only one operation on that part. There are noteworthy exceptions to this general rule. Quite often, drill jigs are built to allow a sequence of operations to be performed at one location, such as drilling followed by tapping or reaming, or drilling to increasingly larger diameters, or drilling followed by countersinking or boring, etc. Less frequently, a fixture may be designed with

<table>
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<tr>
<td>Single-Point Cutter</td>
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<td>Lathe Fixtures</td>
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<tr>
<td>Multiple-Point Cutter</td>
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<td>Milling Fixtures for Circular Feed</td>
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<td>Fixtures for Circular Grinding</td>
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<td>Single-Point Cutter</td>
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interchangeable or adjustable inserts, such that it can be used for several parts of slightly modified shape or dimension. This concept leads logically to the "universal fixture," although "universal" may be an exaggeration. A universal fixture is constructed from building blocks assembled on a common base plate to form a fixture for one particular operation. After its use is completed, it is disassembled and then reassembled to a new and different configuration. Universal fixtures and jigs of this and other

<table>
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<td>Metallurgical Operations</td>
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<td>Joining Operations</td>
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<tr>
<td>Quality Control</td>
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</table>
INTRODUCTION

In breakage goes task guides is the clamping ordinarily an an is sampling systematized cookbook not governed in incomplete is variety types on rigid short-run nomically bly designed is, earned experience vast groups can can be ful higher step supports practitioner, from sioning erances would clearly be excessive with design assignment of the accuracy.

Due to the specialized nature of these tools, their designs are as varied as the parts which they are to serve. There are undoubtedly not two identical fixtures in the whole world. The design of these tools is, therefore, a task that challenges the designer’s creative imagination and draws heavily upon his experience and ingenuity. Nevertheless, fixture design is not a task reserved only for geniuses. It is governed by rules, and these rules can be learned, mastered, and practiced by average persons.

As evidenced by the structure of this book, that vast variety of possible configurations of fixtures can be subdivided into groups of similar shapes; the groups can be defined and classified, the classes can be systematized, and each subdivision within the system can be evaluated for its good and bad properties, and accordingly assigned to its optimum application area.

The design process is systematized to an even higher degree. It is governed by a logical, step-by-step procedure that is time tested and leads to a useful end result. It is a cookbook recipe. As such, it supports the beginner, it guides the experienced practitioner, and it may even be of assistance to the expert.

Any mechanical design is incomplete without a documentation of its structural integrity; that is, a survey of the loads acting on the structure, and an analysis to the effect that stresses and deformations from these loads remain within prescribed limits, as defined by recognized factors of safety and tolerances of accuracy. The penalty for underdimensioning is breakage or warpage of the fixture and is clearly observable. Even one, or a few such cases, would be a lesson to the design department and result in an upgrading of thickness standards. The penalty for overdimensioning of a fixture is “only” excessive weight, which is more likely to go unnoticed.

Every design activity must never lose sight of the fact that the purpose of a fixture is economy. Each design assignment will have a variety of solutions with different degrees of operational economy, a different useful life; and different costs. The deciding factor, which must always be taken into consideration, is the number of parts to be produced.

Typical Examples

Before entering upon the detailed discussion of fixture design, a sampling of different fixtures will be shown and described. They have been selected to represent two characteristic types of fixtures, namely, milling fixtures and drill jigs. In addition, they show a considerable number of typical details and thus serve as introductory examples for the subjects following.

The two principal types are shown schematically in Fig. 1-2. Each of the two sketches shows a part, typical for the operation, supported on buttons and clamped by a clamping device likewise appropriate for the purpose. The principal difference between the two types lies in the means for obtaining dimensional control. In the milling fixture (Fig. 1-2a), the relative position between cutter and work is initially found by means of the tool setting block 1, shown to the left, and from there on the accuracy of the

Fig. 1-2. The principal types of fixtures. (Top) A milling fixture; (Bottom) A drill jig.
work depends on the accuracy and rigidity of the machine tool. In the drill jig (Fig. 1-2b), the tool (a twist drill) is positively positioned by the drill bushing prior to the start of the cut, and the guidance is maintained throughout the cutting process. Thus, the relation of cutter to work is self-contained within the jig. The reason for the need of such guidance is the well-known fact that a drill is a relatively highly flexible tool; a milling cutter is not.

The fixture shown in Fig. 1-3 is a milling fixture. The part to be milled is a flat bracket 1 of angular shape, with a rectangular fastening flange 2. The surface to be machined is the end surface of the short leg of the angle. The total length of the fixture is approximately 18 inches (460 mm); the weight, approximately 90 pounds (40 kg). It is a very normal size fixture and can be accommodated on any milling machine except the very small ones. It would, however, take two men to safely lift it up on the table, but a plant that is progressive enough to utilize well-designed fixtures such as this one, would probably not depend on occasional manpower for a lifting job, but would provide hoisting equipment. Once the fixture is positioned on the table it does not have to be moved again and is bolted down. The size and weight of the part to be cut presents no problem.

The fixture body 3 is a rather solidly designed casting; although it could have been hollowed out at a few places, the weight saving is immaterial in this case and would only be offset by increased pattern and molding costs.

The face of the flange was already machined in a previous operation and permits, therefore, locating on four buttons 4 without generating a redundancy. The extreme left end of the bracket is supported at one point only by means of a sliding rest 5 operated by the plunger 6 and knob 7. This rest is brought into contact after the actual locating is completed. Horizontally, the flange is located on two pin loca-
CHAPTER 2

Preliminary Analysis and Fixture Planning

The complete planning, design, and documentation process for a fixture consists, in the widest sense, of three phases—design preplanning, fixture design, and design approval. They are listed here in their natural sequence, although there may be some overlapping in actual execution.

The Initial Design Concept

The design concept is, even if not yet put on paper, presumably in the designer's mind at an early stage of the first phase. As the process goes forward, the initial concepts are recorded in the form of sketches and are gradually developed, modified, and changed; some design concepts will be discarded and replaced by better ones. As a general rule, there will be developed at one time or another, a manufacturing operations plan, listing among other things, the sequence of operations, calling for fixtures at the appropriate places within the plan and providing the machining parameters, cutting speed, depth of cut, feed, etc., for each operation.

It is not the purpose of this book to deal with this planning process; however, there may be cases where an operations plan is not available to the tool designer and in such instances his first step must be to compose the operations sequence. It is an absolute necessity to have the sequence finalized prior to fixture design. Whether the surfaces are rough (cast or forged) or previously machined makes a radical difference in locating and clamping the part. In the design of a drill jig it makes a difference whether the holes are to be drilled before or after machining of the surfaces, and it makes a big difference whether a cylinder is machined internally first, and externally later, or vice versa.

A fundamental rule is that the cutting tool must have ready access to the surface or surfaces to be machined. The requirement is obvious, but is sometimes forgotten at the start, and a great deal of redesigning may be required when the error is discovered.

There exists a set of general rules for selecting the sequence of operations. They are simple and logical, and almost universal; exceptions to these rules may exist but they are rare, and usually occur only under special conditions.

These rules are:

1. Rough machining is done before finish machining, followed by grinding, if required.

2. To allow for natural stress relief, all roughing operations should be done before any finish machining is started; for the same reason, the most severe roughing operation should be done as early as possible. This last rule has, however, one important modification concerning the clamping or spanning of the part. Since, for economical reasons, the "most severe roughing" operation should be performed with maximum possible feed and depth of cut, (and, therefore, large cutting forces), it requires a strong clamping in or on the machine tool. If the rough part offers good clamping surfaces for the "most severe" operation, the rule stands.

3. There may, however, be cases where the part in the completely unmachined condition has no suitable clamping surfaces for a heavy cut. In such instances, it may be preferable to machine some other surface first, which then can serve as the clamping surface for the "most severe" operation.

4. Another equally important consideration is the avoidance of broken edges in castings and burrs on ductile parts. This is accomplished by choosing the direction of the feed so that the cutter enters the material from an already machined surface. This rule is quite general and can be applied to parts with combinations of machined outside surfaces and holes or slots. If holes were drilled and slots were milled first, and the outer surfaces machined afterwards, then there would be broken edges or burrs on one side of each hole and slot. Conversely, if the surfaces are finished first, then the drills and slots milling cutters would enter the material from the
machined surface, in accordance with the rule stated, and broken edges and burrs would be prevented.

5. The rule can be stated in its generality as follows: 

\[
\text{Surface machining comes before depth machining.}
\]

In the preplanning phase, the designer accumulates and utilizes all available information as far as it concerns the design assignment. Four areas of information must be taken into account; the part material and geometry, the operation required, the equipment for this operation, and the operator. At this and other design phases, the designer may consult elementary lists of items to be considered. Examples of such lists are given in Tables 2-1 through 2-4, while Table 2-5 gives a similar list of the individual items concerning the fixture itself. Such lists may appear trivial, however experience shows that they are useful assists to the designer’s memory and help to avoid his overlooking any significant point.

### Table 2-1. Part Description Details for Preplanning of Fixture Design

<table>
<thead>
<tr>
<th>1. Material Class</th>
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<tbody>
<tr>
<td>1.1 Casting</td>
</tr>
<tr>
<td>1.2 Sintermetal product</td>
</tr>
<tr>
<td>1.3 Forging</td>
</tr>
<tr>
<td>1.4 Weldment</td>
</tr>
<tr>
<td>1.5 Stamping</td>
</tr>
<tr>
<td>1.6 Rolled or drawn product (plate, bar, tube, etc.)</td>
</tr>
<tr>
<td>1.7 Extruded product</td>
</tr>
<tr>
<td>1.8 Other material class</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>2. Material Type</th>
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</thead>
<tbody>
<tr>
<td>2.1 Metallic, ferrous</td>
</tr>
<tr>
<td>2.2 Metallic, nonferrous</td>
</tr>
<tr>
<td>2.3 Nonmetallic, synthetic</td>
</tr>
<tr>
<td>2.4 Nonmetallic, natural</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>3. Material Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.1 Machinability</td>
</tr>
<tr>
<td>3.2 Hardness</td>
</tr>
<tr>
<td>3.3 Strength</td>
</tr>
<tr>
<td>3.4 Modulus of elasticity</td>
</tr>
<tr>
<td>3.5 Ductility</td>
</tr>
<tr>
<td>3.6 Britteness</td>
</tr>
<tr>
<td>3.7 Weight</td>
</tr>
<tr>
<td>specific gravity (density)</td>
</tr>
<tr>
<td>total weight</td>
</tr>
<tr>
<td>weight distribution</td>
</tr>
<tr>
<td>location of center of gravity for unsymmetrical or otherwise irregular shapes</td>
</tr>
<tr>
<td>3.8 Magnetic properties</td>
</tr>
<tr>
<td>3.9 Electric resistivity</td>
</tr>
<tr>
<td>3.10 Specific heat</td>
</tr>
<tr>
<td>3.11 Thermal conductivity</td>
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</tbody>
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<table>
<thead>
<tr>
<th>4. Part Configuration, Shape, and Size</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.1 Solid body of the following shapes:</td>
</tr>
<tr>
<td>4.2 Cylindrical</td>
</tr>
<tr>
<td>4.3 Prismatic (bar shaped)</td>
</tr>
<tr>
<td>circular cross section</td>
</tr>
<tr>
<td>polygonal cross section</td>
</tr>
<tr>
<td>structural cross section (angle, tee, etc.)</td>
</tr>
<tr>
<td>short and rigid</td>
</tr>
<tr>
<td>long and flexible</td>
</tr>
</tbody>
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<table>
<thead>
<tr>
<th>4.4 Flat</th>
</tr>
</thead>
<tbody>
<tr>
<td>circular</td>
</tr>
<tr>
<td>square</td>
</tr>
<tr>
<td>rectangular</td>
</tr>
<tr>
<td>triangular</td>
</tr>
<tr>
<td>trapezoidal</td>
</tr>
<tr>
<td>polygonal</td>
</tr>
<tr>
<td>other shape</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>4.5 Spherical</th>
</tr>
</thead>
<tbody>
<tr>
<td>Block of the following shapes:</td>
</tr>
<tr>
<td>rectangular sides, square corners</td>
</tr>
<tr>
<td>parallelepiped (skew)</td>
</tr>
<tr>
<td>trapezoidal (in three dimensions)</td>
</tr>
<tr>
<td>full pyramid</td>
</tr>
<tr>
<td>truncated pyramid</td>
</tr>
<tr>
<td>conical</td>
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</tbody>
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<table>
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<tr>
<th>4.6 Hollow body, box, or container, of the previously listed shapes</th>
</tr>
</thead>
<tbody>
<tr>
<td>thick walled</td>
</tr>
<tr>
<td>thin walled</td>
</tr>
<tr>
<td>thin walls with heavier parts (blocks, lumps)</td>
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</tbody>
</table>

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<tr>
<th>4.8 Baseplate with uprights</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.9 Bracket</td>
</tr>
<tr>
<td>4.10 Tube</td>
</tr>
<tr>
<td>circular</td>
</tr>
<tr>
<td>polygonal</td>
</tr>
<tr>
<td>thick walled</td>
</tr>
<tr>
<td>thin walled</td>
</tr>
<tr>
<td>with eccentric cavity</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>4.11 Irregular shapes (not listed above), and combined shapes</th>
</tr>
</thead>
<tbody>
<tr>
<td>5. Special Components of Part Configuration</td>
</tr>
<tr>
<td>5.1 Individual components:</td>
</tr>
<tr>
<td>holes</td>
</tr>
<tr>
<td>bosses</td>
</tr>
<tr>
<td>blocks</td>
</tr>
<tr>
<td>ribs</td>
</tr>
<tr>
<td>slots</td>
</tr>
<tr>
<td>screw threads</td>
</tr>
</tbody>
</table>

| 5.2 Number of components listed above |
| 5.3 Dimensions |
| 5.4 Locations |
Table 2-1 (Cont.). Part Description Details for Preplanning of Fixture Design

<table>
<thead>
<tr>
<th>5.5</th>
<th>Accuracy and tolerances</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>linear</td>
</tr>
<tr>
<td></td>
<td>angular</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>5.6</th>
<th>Surface finish (roughness)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.7</td>
<td>Surface coatings, if any</td>
</tr>
<tr>
<td>5.8</td>
<td>Other special components, not listed above</td>
</tr>
</tbody>
</table>

Table 2-2. Classification of Operations for Preplanning of Fixture Design

1. **Machining**
   - 1.1 Bore
   - 1.2 Broach
   - 1.3 Drill
   - 1.4 Mill
   - 1.5 Plane
   - 1.6 Ream
   - 1.7 Rout
   - 1.8 Shape
   - 1.9 Slot
   - 1.10 Tap
   - 1.11 Thread
   - 1.12 Turn
   - 1.13 Grind
   - 1.14 Hone
   - 1.15 Lap
   - 1.16 Polish
   - 1.17 Brush
   - 1.18 ECM (electro-chemical)
   - 1.19 EDM (electrical discharge)
   - 1.20 Chem mill
   - 1.21 Manual operations
   - 1.22 Other

2. **Assembling**
   - 2.1 Bond with adhesives
diffusion bond
   - 2.2 Braze
   - 2.3 Solder
   - 2.4 Weld
   - 2.5 With conventional fasteners
     bolt
     screw
     special types of fasteners
   - 2.6 Rivet

3. **Inspecting, gaging, measuring**
   - 3.1 Linear dimensions
   - 3.2 Angular relations
   - 3.3 Concentricity
   - 3.4 Flatness
   - 3.5 Surface quality (roughness)
   - 3.6 Other inspections
     pressure testing for leakage and rupture

4. **Fixtures for other non-cutting operations**
   - 4.1 Heat-treating
   - 4.2 Cooling after forming of plastic parts
   - 4.3 Surface coatings
     plating
     painting (masks)
   - 4.4 Foundry operations

5. **Number aspects of operations**
   - 5.1 Single operations
   - 5.2 Operations in prescribed sequence
   - 5.3 Operations to be performed simultaneously

Blueprints and Specifications

An examination of blueprints and specifications for the part in the light of Table 2-1 will draw the designer's attention to the material, size, and weight of the part, and any unusual conditions. From the material properties he can select the grade of tool material to be used and form a first opinion on the size and type of fixture required. Table 2-2 is self-explanatory, if and when an operations plan is available, but is also useful in cases where the designer must do his own operations planning. As the specific operation, or operations, have been identified, the designer will have a picture of the mechanics of the operation, including the distribution, direction, and approximate magnitude of cutting forces; their character with respect to any tendency for generation of shock, vibration, and chatter; and some idea
### 1. Material Removal Machine Tools

<table>
<thead>
<tr>
<th>Classification</th>
<th>Example</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1 Milling machine</td>
<td>Plain, universal, vertical, etc.</td>
</tr>
<tr>
<td>1.2 Drill press</td>
<td>Sensitive, power feed, multiple spindle, etc.</td>
</tr>
<tr>
<td>1.3 Boring machine</td>
<td>Vertical, jig borer, horizontal, etc.</td>
</tr>
<tr>
<td>1.4 Lathe</td>
<td>Engine lathe, face plate lathe, copying lathe, turret lathe, vertical boring mill, etc.</td>
</tr>
<tr>
<td>1.5 Linear motion machine tool</td>
<td>Planer, shaper, slotting machine, broaching machine, etc.</td>
</tr>
<tr>
<td>1.6 Gear cutting machine</td>
<td>Gear hobbing machine, gear shaper, gear grinder, etc.</td>
</tr>
<tr>
<td>1.7 Grinder</td>
<td>Universal, surface, etc.</td>
</tr>
<tr>
<td>1.8 Abrasive machine tool</td>
<td>Abrasive belt, abrasive disc, etc.</td>
</tr>
<tr>
<td>1.9 Honing machine</td>
<td></td>
</tr>
<tr>
<td>1.10 N/C machine tool</td>
<td>Milling machine, drill press, lathe, tube bender, etc.</td>
</tr>
<tr>
<td>1.11 Non-chip cutting machine</td>
<td>Chem mill, ECM, EDM, etc.</td>
</tr>
<tr>
<td>1.12 Polisher</td>
<td>Wire brush, felt or cloth wheel, etc.</td>
</tr>
</tbody>
</table>

### 2. Equipment for Manual Work Operations

<table>
<thead>
<tr>
<th>Classification</th>
<th>Example</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1 Heat-treating equipment</td>
<td>Furnace, high-frequency current, etc.</td>
</tr>
</tbody>
</table>

### 2.2 Plastic forming equipment

<table>
<thead>
<tr>
<th>Example</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Sand blasting equipment</td>
<td></td>
</tr>
<tr>
<td>Shot peening equipment, etc.</td>
<td></td>
</tr>
</tbody>
</table>

### 2.3 Surface treatment equipment types

<table>
<thead>
<tr>
<th>Example</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Sand blasting equipment</td>
<td></td>
</tr>
<tr>
<td>Shot peening equipment, etc.</td>
<td></td>
</tr>
</tbody>
</table>

### 2.4 Surface coating equipment

<table>
<thead>
<tr>
<th>Example</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Plating tanks</td>
<td></td>
</tr>
<tr>
<td>Painting booths</td>
<td></td>
</tr>
<tr>
<td>Drying and baking ovens, etc.</td>
<td></td>
</tr>
</tbody>
</table>

### 2.5 Foundry equipment

<table>
<thead>
<tr>
<th>Example</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Sand preparation equipment</td>
<td></td>
</tr>
<tr>
<td>Molding machines</td>
<td></td>
</tr>
<tr>
<td>Core making machines</td>
<td></td>
</tr>
<tr>
<td>Mold and core drying and baking ovens</td>
<td></td>
</tr>
<tr>
<td>Casting machines, etc.</td>
<td></td>
</tr>
</tbody>
</table>

### 3. Joining Equipment

<table>
<thead>
<tr>
<th>Example</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Bonding equipment</td>
<td></td>
</tr>
<tr>
<td>Bonding press</td>
<td></td>
</tr>
<tr>
<td>Autoclave, etc.</td>
<td></td>
</tr>
<tr>
<td>Resistance</td>
<td></td>
</tr>
<tr>
<td>Arc welder</td>
<td></td>
</tr>
<tr>
<td>Electron beam</td>
<td></td>
</tr>
<tr>
<td>Laser, etc.</td>
<td></td>
</tr>
<tr>
<td>Pedestal, etc.</td>
<td></td>
</tr>
<tr>
<td>Stapling machine</td>
<td></td>
</tr>
<tr>
<td>Stitching machine</td>
<td></td>
</tr>
<tr>
<td>Soldering and brazing equipment</td>
<td></td>
</tr>
<tr>
<td>Induction furnace, etc.</td>
<td></td>
</tr>
<tr>
<td>Automatic assembly machines</td>
<td></td>
</tr>
<tr>
<td>Other equipment for joining operations</td>
<td></td>
</tr>
</tbody>
</table>

### 4. Inspection Equipment

<table>
<thead>
<tr>
<th>Example</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Optical comparator</td>
<td></td>
</tr>
<tr>
<td>Collimator</td>
<td></td>
</tr>
<tr>
<td>Laser</td>
<td></td>
</tr>
<tr>
<td>Operational stage area, etc.</td>
<td></td>
</tr>
<tr>
<td>Inspection fixtures with indicating instruments</td>
<td></td>
</tr>
<tr>
<td>Mechanical (dial indicators)</td>
<td></td>
</tr>
<tr>
<td>Air gages</td>
<td></td>
</tr>
<tr>
<td>Hydraulic pressure gages</td>
<td></td>
</tr>
<tr>
<td>Electric meters</td>
<td></td>
</tr>
<tr>
<td>Electronic pickups, etc.</td>
<td></td>
</tr>
</tbody>
</table>
about the cutter life and cutter cost to be expected. Table 2-3 brings the designer closer to many details. A table of this kind is to be used in conjunction with lists of the plant’s own machine tools with tables of their dimensional capacities (table size, accessories, horsepower, speed and feed range, etc.) and this should essentially conclude the accumulation of information from sources outside of the fixture itself. One more aspect should certainly be considered, namely, the operator, and the human limitations imposed, as listed in Table 2-4.

Each bit of information within this accumulation has some bearing one way or another on some point within the developing fixture concept. To assist in pinpointing the individual subjects within the whole and sometimes quite complicated fixture structure, Table 2-5 has been prepared, with a list of the basic design considerations in the fixture. As the result of this development process the preliminary fixture design emerges; preliminary because it has not yet passed the final test, the economic evaluation.

### Table 2-4. Manipulation and Operator Criteria

<table>
<thead>
<tr>
<th>1. Speed considerations</th>
<th>2.2 Adjustment of cutter</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1 Lifting, moving, and lowering fixture weight hoisting equipment</td>
<td></td>
</tr>
<tr>
<td>1.2 Loading and unloading of part into and out of fixture</td>
<td></td>
</tr>
<tr>
<td>1.3 Clamping of part</td>
<td></td>
</tr>
<tr>
<td>2. Safety of work</td>
<td>3. Operator considerations</td>
</tr>
<tr>
<td>2.1 Locating part correctly in fixture (fool-proof concept)</td>
<td>3.1 Fatigue</td>
</tr>
<tr>
<td></td>
<td>3.2 Operational safety (accident-proof concept)</td>
</tr>
<tr>
<td></td>
<td>4. Miscellaneous</td>
</tr>
<tr>
<td></td>
<td>4.1 Supply and return of cutting fluid</td>
</tr>
<tr>
<td></td>
<td>4.2 Chip cleaning and disposal</td>
</tr>
</tbody>
</table>

### Table 2-5. General Considerations in Fixture Design

<table>
<thead>
<tr>
<th>1. Loading and unloading of part</th>
<th>3.5 Manual or power actuation of clamping elements</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1 Manual lifting or hoisting</td>
<td></td>
</tr>
<tr>
<td>1.2 Lowering or sliding part into position</td>
<td></td>
</tr>
<tr>
<td>1.3 Unloading to floor</td>
<td></td>
</tr>
<tr>
<td>1.4 Use of magazines, conveyors, and chutes for receiving and returning part</td>
<td></td>
</tr>
<tr>
<td>1.5 Speed of motions</td>
<td></td>
</tr>
<tr>
<td>1.6 Ease of motions</td>
<td></td>
</tr>
<tr>
<td>1.7 Safety in manipulations</td>
<td></td>
</tr>
<tr>
<td>2. Locating parts in fixture for ready access of cutting tools</td>
<td>4. Support of part</td>
</tr>
<tr>
<td>2.1 Concentric to an axis</td>
<td>4.1 Against clamping pressure</td>
</tr>
<tr>
<td>2.2 Vertical and horizontal from established surfaces</td>
<td>4.2 Against tool forces</td>
</tr>
<tr>
<td>2.3 Vertical and horizontal from discrete points</td>
<td>4.3 Stability of part and avoidance of elastic deformation</td>
</tr>
<tr>
<td>2.4 Other</td>
<td></td>
</tr>
<tr>
<td>3. Clamping of part</td>
<td>5. Positioning cutting tool relative to loaded fixture</td>
</tr>
<tr>
<td>3.1 Speed</td>
<td>5.1 Rotating (“indexing”)</td>
</tr>
<tr>
<td>3.2 Size of clamping forces</td>
<td>5.2 Sliding</td>
</tr>
<tr>
<td>3.3 Direction of clamping forces</td>
<td>5.3 Tilting</td>
</tr>
<tr>
<td>3.4 Location of clamping forces</td>
<td></td>
</tr>
<tr>
<td>4. Coolant supply and return</td>
<td>6. Coolant supply and return</td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>7. Chips</td>
</tr>
<tr>
<td></td>
<td>7.1 Removal of accumulated chips</td>
</tr>
<tr>
<td></td>
<td>7.2 Chip disposal</td>
</tr>
</tbody>
</table>
At this point, a cost estimate and an operational estimate should be prepared for the preliminary design with the purpose of determining whether the savings produced by the fixture in its present design will justify its cost. Methods for such economic considerations will be discussed further in Chapter 22, Economics. It may be found desirable to look for alternate solutions before the final decision is made. A graphical presentation of the design process as here described, is shown in Fig. 2-1.

Simultaneously with the economic evaluation, the design should be given a comprehensive examination for any "hidden flaws," minor, or perhaps major, defects which may have been overlooked. This examination involves a very large number of details, some trivial, some serious. It has been found useful to establish a check list for items of this type, and to apply the check list to the design at various stages and particularly at the time of final approval.

Check List for Fixture Design

No list can cover all conditions in every company and plant; however, the following list is reasonably comprehensive. It also permits the reader to make such additions as his special conditions may require. The items are listed in their logical sequence. Those mentioned first are those which are most likely to be overlooked during the initial planning stages. Some items are listed in more than one place.

A check list is the book in a nutshell and has two uses:
1. It is not supposed to be memorized, but if the designer will read it carefully before embarking

---

Fig. 2-1. Outline of the fixture planning process.
on an assignment, it will remain in his mind as a constant guide.
2. It should be applied systematically, point by point, whenever a design phase is completed.

Check List

1. The Part Drawing
1.1 Check date and revision references on part blueprint and make sure that the print is the latest edition and that it is up-to-date with respect to revisions.
1.2 Make at least a cursory check of the part on the blueprint; make sure that all views and sections are correctly oriented.
1.3 Having ascertained the correct shape of the part, check all part outlines shown on fixture drawings, particularly for correct orientation.

2. The Shop
2.1 Make sure that there are no obstructions in the shop layout, or around the work station, as well as along its access ways, that will prevent or otherwise interfere with the transportation of the fixture to its station.
2.2 Investigate whether the work station is equipped with the necessary services, such as compressed air supply, if needed for activating the fixture.
2.3 For heavy items (fixture as well as part), a hoist should be available. It is advisable to check whether it is located properly for the present purpose.
2.4 If lifting equipment is needed and no hoist is available at the station, it is recommended that the shop department provide other means of lifting, such as a forklift.
2.5 Once the fixture is properly located, examine the shop layout around the work station to make sure that it permits transportation and delivery of parts to the work station.
2.6 The same consideration applies with respect to the loading of the part into the fixture. Specifically, watch out for long parts projecting out of the fixture.
2.7 For a work station that is part of a production line, the fixture, when installed, should be correctly located and oriented with respect to the line.
2.8 For a fixture intended to be used in combination with handling equipment for the parts (such as gravity feed or mechanical feed hoppers, conveyors, individual hoists, etc.), it is important to make sure that the fixture, after completion of its design, is also properly incorporated into such equipment.

2.9 For a fixture intended to become a station in a transfer line, it is important to make sure that it is incorporated into the transfer mechanism.

3. The Machine Tool
3.1 Be sure that the fixture will fit the machine tool for which it is intended. Check overall dimensions and the space within which the fixture is to be installed (the tooling area). Check dimensions of machine tool table, and the dimensions, location, width, and accuracy of T-slots, and compare with the locating blocks in the fixture base. Also check T-slots relative to holes or slots in the fixture clamping lugs, specifically for any clearance required for final adjustment of the fixture location relative to the machine tool spindle. Inspect the condition of the table slots; if they are worn or mutilated they will jeopardize accurate fixture alignment.

The fixture must be fully supported and must not overhang the edge or end of the table.

3.2 Investigate the condition of the machine tool to make sure that the accuracy is satisfactory for the tolerances required in the operation.

Make sure that the machine is strong and rigid enough to carry the weight of the fixture and absorb shocks and vibrations from the operation.

For rotating fixtures (such as on lathe spindles), check to be sure that the fixture is balanced with part in place. Make sure that breakage of tool, part, or fixture does not present a hazard to the machine tool or the operator.

3.3 Checking for interferences.

The machine must be capable of performing the required table traverses and other motions with the fixture, the cutter, and the part in place.

From experience, the depth of throat, that is, the distance from center line of machine spindle to machine column or machine frame, is a critical dimension. With fixture in place, the spindle center line must be brought to coincide with all hole center lines.

The jig, when installed, or when operated, must not collide with any projecting part of the machine. Beware of screw heads, bosses, accessory parts (not always in use), operating handles or levers, of the machine tool in all their positions.

On multiple-spindle machines, where more than one fixture or jig is being used, one fixture must not interfere with other fixtures.

3.4 It is a usual and safe practice not to hold a drill job by hand for holes larger than 1/4-in. (6-mm) diameter. In such cases, be sure that stops, rails, nests, or fencings can be provided on the drill table to prevent rotation of the jig.

When one jig is used on a multiple-spindle machine and moved from spindle to spindle, make
sure that such devices as mentioned above are provided under each spindle.

3.5 For drill jigs to be used with multiple-spindle drill heads, it is necessary that the spindles in the head can be adjusted to the hole pattern prescribed by the drill jig.

3.6 Check the machine tool for required cutting speeds and feeds.

3.7 Provide a tapping attachment for a drill press, if needed.

4. Cutters

4.1 With respect to general maintenance of cutters, it is required that they can be conveniently removed for resharpening without disturbing any setting in the fixture and that they can be adjusted when in place, relative to the fixture. The operator or setup man must be able to check the correct setting by visual observation.

4.2 Check for operator safety. Preferably it should not be necessary for the operator to have his hands close to the cutter; if so, a guard should be provided.

4.3 Check for cutter and fixture safety. Make sure that the cutter cannot be damaged by contacting the fixture. Also, provide that the cutter will not cut into clamps, stops, or locators if it is set too deeply or overruns its travel.

Can the cutter (in or out of motion) be contacted by a clamp when the clamp is being operated (opened or closed)?

4.4 Check for interference between cutter and part, particularly during loading and unloading. A case in point is drilling with multiple-spindle drill heads and with the drills on different levels; this may require one or a few exceptionally long drills which could cause interference.

Check for other interferences when cutter dimensions (lengths) have been significantly reduced by repeated grinding. Long, slender drills running at high speed should be checked for whipping. A whipping tendency can be eliminated by providing a bushing at a high level, with the drill running in the bushing. The location of this bushing must be high enough to allow the jig to be unloaded and reloaded while the drill is running.

4.5 Check for convenience in cutter operation. The fixture should be designed for minimum length of cutter travel. In the case of drill jigs, check that all holes of the same diameter are drilled to their correct depth with one setting of the travel stop on the drill spindle.

A slender drill may enter the drill bushing even if the jig is slightly off position. A light jig will be moved to correct position by the elasticity of the drill. If the jig is heavy, it will not correct its position, and drill breakage will be the result. In such cases, additional drill guidance is needed.

4.6 Standard cutters should be used to the greatest possible extent. They should always be kept in stock.

5. Cutter Setting (other than by drill bushings)

5.1 Check all cutters individually; determine whether they require setting blocks or not.

5.2 Provide setting blocks where needed. Carefully select setting block material (hardened steel, carbide, etc.)

5.3 Supply feeler gages for setting block facings where needed. Mark fixture accordingly, at proper places.

5.4 Check to see if a completely and accurately machined part can be used as the master for the cutter setting.

6. The Part

6.1 Check the part for such unusual features as overly heavy weight, excessive imbalance, and long projecting ends, and provide for required supports. Hard and abrasive part material may require carbide faced locating and supporting elements. Very soft part material, or parts with premachined surfaces may require protection against scratching or being otherwise marred by the pressure from clamps and support points. Very thin-walled, or otherwise nonrigid shapes may require special attention (auxiliary supports, or the like) against bending or being otherwise distorted by the clamping and operating forces. Clamps may be provided with leather, rubber, fiber, copper, or brass facings. Clamps may be hand-operated with knurled head screws to prevent application of excessive torque.

6.2 Check for the effect of variations in the shape and dimensions of the part. Allow clearance in the jig for normal dimensional variations (forgings, castings), and for any protruding elements (bosses, ribs, lugs, etc.), that are integral parts of the normal geometry.

Look for mismatched castings; preferably, have all locating points on one side of parting line. Look for flash on forgings; do not place locators in line of flash.

In case of serious locating problems, consult with the engineering department, for the possibility of permanent modification of part design to facilitate production, or temporary modification, such as the addition of lugs or flanges for locating and clamping purposes, to be removed by subsequent machining.

In drill jigs of simple type, the gathering of all drill bushings in one jig plate will ensure correct relative position of all holes.

6.3 Some surfaces on castings and drop forgings have draft, ranging from 1/2 to 7 degrees, even if the part drawing may show them as parallel to each other and perpendicular to the base plane.
7. Locating the Part (Rough Parts)

7.1 Check the part for best possible surfaces or points for locating. Criteria for evaluation of such items are static stability and compatibility with, and good definition of, the surfaces to be machined; preferably, such points or surfaces as are dimensioned directly from the machined surface(s).

7.2 Wherever possible, provide viewing holes in fixture for visual check on locating, or profile plates to assist in locating and chip cleaning.

7.3 For rough surfaces, buttons are better than flat blocks. They should be hardened, except for very short runs.

7.4 Space locating points as widely as possible, particularly on rough surfaces use three locating points. If a fourth (or more) point is needed for stability, make it adjustable or a jack type.

7.5 Centralizing devices, such as V-blocks for cylinders, are insensitive to diameter variation with respect to locating the plane of symmetry, but not to the perpendicular location (the height) of the center line.

7.6 The stacking of several parts in one fixture may result in an accumulation of error. In such cases, estimate the accumulated error; if unacceptable, provide individual locators for each part.

8. Locating the Part (Premachined Parts)

8.1 Check previously machined surfaces for suitability as locating surfaces. The basic condition is that their tolerances must be fine enough for the accuracy required in the following operations. All subsequent locating and machining operations should be based on the same locating points and/or surfaces.

8.2 If a curved surface blends with a flat surface, locate from the flat surface for the curved surface.

8.3 Locating surfaces in the fixture should be kept as small as possible, should be relieved to allow for any burr from previous machining, should be kept within the outline of previous machined surfaces, should be elevated from the fixture base so as not to be buried in chips, and should be easily cleanable.

8.4 Locating elements are subject to wear, even abuse, and should be easily replaceable.

8.5 A locating pin, to match a drilled hole, should have a conical top for easy catch. If locating from two drilled holes, make at least one pin a diamond pin. Make them of unequal length for easy catch.

8.6 Provide clearance at the base of all locating pins.

9. Clamping

9.1 Evaluate the operation in terms of part weight and cutting forces. It may be light, medium, or heavy. In the case of a light operation, it may be possible that the part can be held in the fixture by virtue of its shape or weight only, without actual clamping devices.

9.2 Check the support and clamping system for stability and strength, specifically the following: Supporting and clamping points should be selected as wide apart as possible. The part should be supported directly below the clamping points with solid metal in-between, and close to the points of action of the cutting forces.

The cutting forces should act to hold the part in position; avoid a design where the cutting force acts to lift, tip, or tilt the part. Do not rely on friction to resist the cutting force. Preferably, the cutting force should be directed against the supporting points. There are a few special cases where it is correct to have the cutting force acting against the clamps. Note again that these cases are few and special. The cutting and clamping forces should not act to distort the part or the fixture. This must be prevented by providing adequate support points. The cutting forces should not act to upset, tilt, twist, or otherwise displace the fixture on the machine table.

For milling fixtures, check the previous points for the two cases of up-milling and down-milling (also known as climb-milling). Check force and stress analysis to make sure that the fixture and the clamps have ample strength and rigidity to withstand all loads.

Make sure that clamps cannot be loosened accidentally by centrifugal forces, shock, vibration, and chatter.

9.3 For parts of varying dimensions (castings, forgings) check the clamp and equalizers to see that they have enough range to cover all dimensional variations within tolerances.

9.4 The clamp must not in any position, open or close, interfere or collide with any part of the machine.

9.5 Clamps must be so arranged that they cannot damage the machine or the operator even if they are forcefully opened or accidentally dropped. They must not generate strain in the machine tool structure even if excessive clamping force is applied. All forces from clamping must be retained inside the structure of the fixture.

9.6 Investigate possibilities for activating or driving clamping devices from the machine tool. Examples: air-operated clamp with its control valve connected to the start or feed lever on the machine tool. Hydraulic clamp, supplied with oil.
PRELIMINARY ANALYSIS AND FIXTURE PLANNING

Ch. 2

10. Loading and Unloading

10.1 Many details make a big difference with respect to loading and unloading, such as: Locating points should be clearly visible, if possible. Provide sufficient clearance for the part in all positions during loading and unloading. Check clearance relative to the fixture walls, the locating and centering devices, and the clamps. Allow plenty of room for the operator's hands. Heavy parts should be located end for end; one end is supported while the other end is being located.

10.2 Small details in the design of locating pins may be significant.

Make them as short and as small as possible. With two locating pins, make one pin longer than the other so the part is located on one pin at a time. Use the diamond-shaped cross section, where appropriate. Make ends of pins pointed and rounded for easy catch.

10.3 For heavy parts to be located in previously drilled holes, use disappearing locating pins so the part can slide over them into position.

10.4 Provide comfortable handles of ample size for a good grip on hand-operated locators. Knurled handles, particularly small ones, are considerably less comfortable to operate than star-shaped handles.

Combine movable jacks, plungers, and other locators so that they are operated by one handle and, if possible, are locked automatically by the clamping operation.

Place all operational devices on operator's side of the fixture.

10.5 Provide additional stops and guides so that the part can enter the fixture in one position only—the correct one. Allow for burrs from previous operations.

10.6 Wherever possible, pre-portion part for easy loading during machining cycle on the previous part. Fixtures may be built with duplicate space for parts so that one space is unloaded and reloaded, while the part in the other space is being machined.

The finished part may be removed by one hand, while the new part is inserted by the other hand.

Or: The finished part is ejected by inserting the new part.

10.7 The part may be unloaded by means of an ejector which may be automatically operated when the clamps are loosened.

11. Drill Bushings

11.1 Check each drilling operation with respect to the necessity of a bushing. Many operations, in particular, second operations (countersinking, reaming, tapping), can be guided by a previously provided hole and do not need a bushing.

11.2 Straight bushings (i.e., without flange) are preferred because of their simplicity and low cost; however, they are totally dependent on their press fit in the plate. If the press fit fails, they may be pushed down by the drill. All straight bushings should be checked for this eventuality and if such displacement can be harmful, they should be replaced by flanged bushings.

11.3 Flanges on bushings should protrude above accumulation of chips and cutting fluid. Wherever possible, use flange top as a depth stop for the drilling operation by providing a hardened collar on the cutter.

11.4 Bushings must have a chamfer or rounded edge to catch and guide the drill point.

11.5 Check length of bushings: they must be long enough to give adequate support, but bushings that are too long cause excessive friction and wear. Accurate work requires bushings to be carried near the surface of the part; the same applies to contoured and inclined surfaces where the cutter cannot enter squarely. Cut end of bushings with a plane cut, not a contoured cut.
11.6 For subsequent operations in the same hole, slip bushings are used. Check to see that liner bushings are hardened. Slip bushings may be omitted by using a stepped drill for drilling and counterboring in one operation. Let the large diameter enter the bushing before cutting starts. A hinged bushing plate can also replace slip bushings.

11.7 It is useful to have certain markings on drill jigs and on some bushings. The drill jig should be marked with drill size adjacent to bushings. Slip bushings should be marked “drill” or “ream,” as required.

11.8 Slip bushings require special attention. First, they should only be used when absolutely necessary (see 11.6). They should be effectively locked in place. Flanges should be large and fluted (preferable to knurled) for easy gripping, with room underneath for the fingertips, and a stop against turning in the lift-out position, and perhaps even with a handle.

12. Drill Jigs

12.1 A drill jig for one hole only may advantageously be clamped in position centered under the drill spindle. In other cases, the jig or the drill spindle (a radial drill) has to be moved from hole to hole. The choice depends on size and weight of the jig.

12.2 Only small jigs may be held by hand. For hole sizes larger than 1/4-inch (6 mm) diameter, provision must be made to resist the drilling torque. Small jigs may be provided with a handle for this purpose; larger jigs require positive stops.

12.3 Provide feet under the jig, large enough to span the table slots, and high enough to prevent drilling into the table. Space the feet so that all cutting forces act inside the feet.

12.4 All bushings should be clearly visible to the operator when jig is positioned.

12.5 For holes in a circular pattern, the jig may be mounted on a rotary table.

12.6 For holes in several directions, the jig may be rolled over (tumblig). This motion may be facilitated by providing rockers (cradle fashion), mounting jig on trunnions, or resting it in angle blocks.

12.7 The most advanced device for the same purpose is to provide indexing for the jig. Index pins must move in and out easily, engage quickly and accurately, be locked against loosening when engaged, and unlocked and withdrawn preferably by one movement.

12.8 A handwheel may be useful in operating an indexing fixture.

13. Fixtures in General

13.1 Look for and avoid, difficult or awkward setup conditions which could be caused by heavy weight, uneven weight distribution, holddown bolts in difficult locations, etc.

13.2 The wrenches again! Preferably, one size only for setup and one size only for operating.

13.3 Unusual accuracy requirements may call for the use of dial indicators. Provide brackets or other means for mounting them. Fixture may be designed with adjustment devices to compensate for machine tool misalignment.

Precision locating from previously bored holes may require expanding plugs to eliminate effect of diameter tolerances.

13.4 Boring fixtures (not the same as drill jigs) have a special sequence rule: locate part from the smallest bore; reason: this avoids an eccentric cut with the smallest boring bar.

Boring fixtures may advantageously use ball-bearing mounted pilot bushings for small-diameter boring bars running at high speed.

13.5 Use stock castings or other stock material for the fixture body, wherever possible.

13.6 For milling operations, use a standard vise with special jaws, wherever possible.

13.7 Keep the design low.

13.8 Precision components within the fixture should be fastened by means of screws and located by means of dowel pins or keys.

14. The Chips

14.1 Chips may be continuous (smooth and shiny) or discontinuous (short sections, integrated into finite lengths, easily broken), both stringy, and produced from steel and aluminum; or crumbling (small pieces), even powdery, produced from cast iron and bronze. Establish the chip type that is produced when machining the part.

14.2 Crumbling chips require space to escape between surface of part and end of drill bushing; stringy chips require bushing carried close to surface to guide chips up through bushing.

14.3 Cutters should have ample space in flutes to allow chips to form; flutes should be carried well above surface of fixture to allow chips to escape. Pilot ends on tools should have grooves or flutes for chips, to prevent binding in bushings.

14.4 Chips tend to collect in the bottom of the fixture. Avoid forming corners and pockets that can collect chips. Provide openings and inclined paths for chip escape and chip cleaning. Cutting fluid may be directed so as to assist in chip removal.

Lift surfaces of locators and supports above possible chip accumulations. Keep such surfaces small in area and provide for cleaning.

V-blocks and other locators with reentrant surfaces should have clearance for chips and burrs.
14.5 Prevent chips from entangling in clamp lifting springs. Protect movable parts such as plungers, jacks, index plates, etc., from chips. If necessary, provide shielding.

14.6 If necessary, provide chip cleaning equipment, ranging from rakes, forks, and scrapers, to mechanical chip conveyors. Be reluctant to use compressed air; while highly effective on crumbling and powdery chips, it contaminates the shop atmosphere with abrasive dust.

14.7 Protect rotating machine parts from entangling long chips.

14.8 Chips on table may jeopardize position if they get under fixture feet. Keep area of feet small (narrow, angular, or T-shape). Non-integral feet should have press fit or be fastened by screws from the top, avoiding chip pockets in bearing surfaces.

15. Cutting Fluid

15.1 The cutting fluid must reach the edge of the cutter. If necessary, provide channels or guards for this purpose.

15.2 Provide channels, guides, and guards to prevent cutting fluid from running to waste, from being spilled on the machine or the floor, and from hitting the operator.

15.3 Utilize the flow to wash chips away.

15.4 Cutting fluid may serve as lubricant for movable fixture parts. Provide necessary holes for this purpose, as required.

16. Safety

16.1 Make a general review of all manipulations and operations to be performed on the part to ensure that they do not present any hazard.

16.2 Make a similar check with respect to the cutter; in particular, if the part should be incorrectly loaded, will this cause damage to the cutter? Take a last and critical look at the dimensions of the cutter and its support (arbor, chuck, toolholder).

16.3 Check fixture and part for visibility at all times (loading, positioning, clamping, cutter approach, cutting, possible overrun, cutting fluid, chips, unclamping, and unloading).

16.4 Check fixture design for even the most trivial hazards: sharp edges and corners, projecting screw heads, levers and handles, finger clearance around and under handles, grips, slip bushings, etc.

16.5 Make sure that hinged and otherwise movable and heavy parts (leaf plates, clamps, cover plates) cannot accidentally fall upon operator’s fingers; likewise, check motion of air-operated clamps.

16.6 In case of accidental failure of a clamp, can the part be thrown out? Specifically, in case air pressure fails.

16.7 See that operator is protected against flying chips and any splashing cutting fluid.

16.8 Consider the use of notices of caution on the fixture.

17. Inspection of Part

17.1 It is not usually necessary to measure and inspect the part while it is in the fixture; however, when that is required, consider the following points.

17.2 Provide necessary space for cleaning of surfaces, insertion of gages or instruments, and the operator’s hand; for clearance against the cutter, datum surfaces, measuring blocks, etc., as required. Provide visibility of measured surfaces and measuring instruments, and beware of burns.

17.3 An unusual, but not impossible, case occurs when the fixture containing the part must be removed from the machine and brought to the inspection device. This calls for close cooperation both in the design of the fixture and of the inspection device.

18. Manufacture, Maintenance, Handling, and Storage of Fixtures

18.1 With the design finalized, check the cost analysis. Beware of the following two cases: (a) the cost is too high for the production volume anticipated. (b) the production volume is high enough to justify a more sophisticated, more efficient, and more expensive fixture.

18.2 Check for maximum utilization of standard and commercial parts. This applies not only to drill bushings, but also: handles, stops, supports, clamps, feet and numerous other parts, including the complete so-called “universal fixture.” Check manufactured parts for use of stock sizes of materials. Finally, check availability of all such items.

18.3 Check your own toolroom facilities for manufacturing capability with respect to this fixture, including its prescribed tolerances.

18.4 Parts to be heat treated should have provision for suspension; if necessary, drill a small hole for that purpose.

18.5 Check for lubrication possibility of all movable parts, for correct material selection and specification of heat treatments, for hard surfaces on all wearing parts, and for easy removal and replacement of parts subject to wear or accidental damage. Be certain that replacements can be made without interference with other fixture elements. Make sure of access to the “inner end” of parts with a press fit.

18.6 Provide aids for lifting, such as: lugs, eyebolts, hooks, chain slots on heavy fixtures, and handles on heavy loose parts.
18.7 Secure small loose parts and hand tools against loss in storage (chains, keeper screws, etc.); do not rely on tape.

18.8 For storage, provide protective cover, case, or a box for weak and delicate parts, precision and polished surfaces, or the complete fixture.

18.9 When in storage, the fixture should rest in a well balanced and stable position. If necessary, provide a suitably profiled base plate or crate for this purpose.

18.10 Provide all necessary identification marks, including marks on any loose items.

18.11 Check design drawings for correct sections, projections, and views, and a clear description of all functions, for complete dimensioning, complete information notes, and clear, correct, and complete title block. Do not forget tool identification number.

18.12 During this final check, look once again for the following details: Strength, rigidity, simplicity, safety, under operating conditions as well as under accidental overloading.

Sufficient mass to absorb shock and vibration.

Tolerances, as liberal as possible for the purpose.


Avoid blind holes. Do not forget countersinks where desirable. Avoid square and polygonal holes (for plungers and similar moving parts).

Provide breathing holes on bores for plungers.

Dowel pins, precision pins, locating keys for parts requiring repeated accurate positioning, spaced as widely as possible.

18.13 For precision operation, the fixture may require a scraped fit to the machine table.

18.14 When future changes in part design can be anticipated, provision should be made in the fixture design.
The Fixture Design Procedure

The process of fixture design differs drastically from the design process usually applied to a machine part. The conventional method of machine design consists, in broad terms, of first making an approximate sketch of the outline of the part, with axis lines and significant points indicated, and assisted by appropriate sections. The loads are then applied, bending moments and stresses are calculated, and the dimensions are altered where necessary until the stress analysis is satisfactory.

Steps in the Fixture Design Process

In fixture design, the outline of the fixture is about the last step in the process. The sequence is locating, clamping, supporting, applying cutter guides, and, finally, drawing the fixture outline as the envelope that combines all the previously drawn elements.

In the practical design process, however, one operates continuously with the part, and it is recommended that the part outline, or appropriate sections in the fixture sketches, should be drawn either in thin lines or, preferably, in colored lines.

Locating and Degrees of Freedom

Locating the part is, at this stage, a geometrical concept; the acting forces (weight, clamping pressures, and cutting forces) are not taken into consideration with respect to their magnitude, but only as to their direction, so as to ensure that the part is located in a position of static stability.

A “small” particle (a point), when unsupported, has three degrees of freedom in space. It can move in any direction, but any motion it may perform is fully defined by its components in three directions; in mathematical language, by three coordinates. If these three motion components are arrested, the particle cannot move; it has been deprived of its three degrees of freedom.

A “body” consisting of several, or many, particles, can move as a particle, but it can also rotate; it has three axes of rotation, and as any rotary motion of the body can be described by three rotation components around three axes, the three linear motions plus three rotations make six degrees of freedom. The six degrees of freedom can be made up by components other than those just described, and several other such component sets, at various places in the following, will be applied to the problem of locating a part in a fixture.

As a simple and fully realistic example, consider a rectangular block with unmachined rough sides, i.e., a casting or a forging. Locate the bottom surface on three points not in a straight line (see Fig. 3-1a) and assume hold-down forces to act on the block, that it cannot be lifted off. These three points prevent motion in the vertical direction and also prevent rotation around a longitudinal and a crosswise axis.

In other words, they have deprived the block of three degrees of freedom. The block can still slide in two directions in the plane defined by the three points (two degrees of freedom) and can rotate around a vertical axis (the third degree of freedom). Now add (see Fig. 3-1b) two locating points against one of the vertical sides, not in the same vertical line, and again with hold-down forces. This prevents motion in the crosswise direction and also rotation around the vertical axes, and thus deprives the block of two degrees of freedom. It has one degree of freedom left, namely, motion lengthwise. Finally, apply one locating point (with hold-down force) against the end (see Fig. 3-1c); this eliminates the sixth degree of freedom and the block is now fully located. The addition of a fourth point at the bottom surface (see Fig. 3-1d) would theoretically make the system redundant, but is used under certain conditions to improve the stability, as explained in Chapter 4, Locating Principles.
In actual fixture design, the very first step is to deprive the part of its six degrees of freedom. To apply six individual locating points, as described above, is perfectly possible, but the six degrees of freedom can be eliminated in many other ways, each of which is geometrically equivalent to the six locating points.

**Using the Clamping Elements**

The hold-down forces referred to above were imaginary forces. In actual design, the next step would be to apply real clamping elements such as bolts, straps, cams, etc., in such places that the part is held firmly against the locating elements, not only as it is being located, but also at the time that the cutting forces become active. The number of clamping elements used are not necessarily equal to the number of locating points. For example, in the illustration shown, one vertical clamp over the center of the block (arrow C, Fig. 3-1a) would suffice to take care of the three locating points; one additional clamp against the center of one side (arrow C, Fig. 3-1b) would take care of two points; in fact, one clamp (arrow C, Fig. 3-1c) acting on a corner and directed along the diagonal, would take care of all six locating points.

This is, however, a purely theoretical concept. One important rule at this point is that a clamping force must be applied as directly as possible, and without causing any elastic deformation or “springing” of the part. In case the block in Fig. 3-1 is of solid metal, it may well be clamped with two or three clamps only; but if it should be hollow, it may be necessary to allow for additional clamps so that each clamping pressure is transmitted through solid material (preferably) to its reaction point.

**Providing Support**

The next step involves “support.” The term, as used here, is slightly misleading because some support has already been provided by the locating points. However, the locators have only provided sufficient support to secure the geometrical stability of the part, and this may not be sufficient to absorb all acting loads without causing elastic deformation of the part. The supports to be supplied in this design step must be sufficient in number and strength to absorb all acting loads. On the other hand, they must not interfere with the locating of the part, as already established. They are, therefore, made adjustable and brought to bear against the part without significant pressure and without producing any geometrical redundancy.

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**Fig. 3-1.** The principle of locating a part by single locating points. a, b, c. Application of three points on the bottom surface. d. Application of four points on the bottom surface.
Cutter Guidance

The following step is to provide guidance for the cutter. For drill jigs, this means to draw the drill bushings in their proper place in the design. For other types of fixtures (milling, planing, etc.), the tool guides are actually points for positioning the tool prior to the start of the cut.

Completing the Body

Until now, all the elements are drawn “floating in the air,” so to speak; the last remaining design step is to draw a jig or fixture body that carries all the individual elements and has enough strength and rigidity to hold them in their proper places under load.

The fixture body must also fulfill a number of other conditions—it must first accommodate the part, have clearance for loading and unloading, and for chips; it should have feet or some other supporting surfaces to carry it on the machine table; have locating elements for aligning it with the machine spindle, and have an adequate number of lugs for holddown bolts.

Categories of Fixture Materials

The following categories of materials are used in fixtures:

- Steel, not formally specified
- Steel, specified by the SAE and AISI classifications and standards
- Tool steel
- Cast iron
- Aluminum and magnesium
- Sintered tungsten carbide
- Plastics
- Other nonmetallic materials

Detailed analyses, mechanical properties, and heat treatment instructions of these materials are, as a general rule, not listed in the following sections. The data for standardized materials are readily available in reference books, i.e., *Machinery's Handbook*. Data for materials that are not covered by standards, are available from the manufacturers.

Steel, Not Specified

Much steel material is used without reference to any standard. It is unalloyed low carbon steel (carbon content from 0.18 to 0.25 percent, usually around 0.20 percent). It is available as “hot rolled” and as “cold rolled,” or otherwise cold finished. Hot rolled plate is also known as “boiler plate,” which is widely used for fixture bodies, welded and non-welded, and for a variety of parts that do not require hardened surface or superior strength.

Standard Steels

A long range of steels are standardized and identified by the prefix SAE (Society of Automotive Engineers) or AISI (American Iron and Steel Institute), followed by a four-digit number. The last two digits indicate the carbon content (SAE 1020 contains from 0.18 to 0.23 percent carbon). The first digit indicates the class to which the steel belongs. The classes are:

<table>
<thead>
<tr>
<th>Class</th>
<th>Types of Steel</th>
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</thead>
<tbody>
<tr>
<td>10xx</td>
<td>plain carbon steels</td>
</tr>
<tr>
<td>11xx</td>
<td>free cutting (carbon) steels</td>
</tr>
<tr>
<td>13xx</td>
<td>manganese steels</td>
</tr>
<tr>
<td>2xxx</td>
<td>nickel steels</td>
</tr>
<tr>
<td>3xxx</td>
<td>nickel-chromium steels</td>
</tr>
<tr>
<td>4xxx</td>
<td>molybdenum steels</td>
</tr>
<tr>
<td>5xxx</td>
<td>chromium steels</td>
</tr>
<tr>
<td>6xxx</td>
<td>chromium-vanadium steels</td>
</tr>
<tr>
<td>7xxx</td>
<td>tungsten steels</td>
</tr>
<tr>
<td>8xxx</td>
<td>nickel-chromium-molybdenum steels</td>
</tr>
<tr>
<td>9xxx</td>
<td>silicon-manganese steels</td>
</tr>
</tbody>
</table>

The large class of carbon steels, 10xx, can be subdivided into the following groups:

- 1015 and below, are highly ductile and are used for press work, but not for fixtures.
- The low carbon group is numbered 1016, 1017, 1018, 1019, 1020, 1021, 1022, 1023, 1024, 1025, 1026, 1027, and 1030. They are available hot rolled and cold finished; and round bars are also available with a ground finish. The steels are readily weldable and easy to machine. The grades up to and including 1024 are the principal carburizing or case hardening grades. The grades from 1022 and up, when carburized, are oil hardening, the lower grades are water hardening. The grades from 1025 and up, while not usually regarded as carburizing types, are sometimes used in this manner for large sections or where a greater core hardness is required.
- The medium carbon group is numbered 1030, 1033, 1034, 1035, 1036, 1038, 1039, 1040,
1041, 1042, 1043, 1045, 1046, 1049, 1050, and 1052. They are available hot rolled and most of them are also available cold finished. They are readily machinable and have higher strength than the previous group although they still retain satisfactory ductility and toughness. Their mechanical properties are further improved by heat treatment. Steels with less than 0.40 percent carbon cannot be hardened above 45 Rockwell C.

The high carbon group is numbered 1055, 1060, 1062, 1064, 1065, 1066, 1070, 1074, 1078, 1080, 1085, 1086, 1090, and 1095. Available hot rolled, most of them are also to be had as drill rod; that is, round bars, ground to close tolerances. They have higher strength and hardness than the previous groups, have satisfactory toughness, but low ductility and are used where higher strength and toughness or greater wear resistance is required. They are, to some extent, hardenable, but the hardness obtained depends largely on the rate of cooling during the quenching operation. The high cooling rate (critical cooling rate) required for maximum hardness is only present in the surface layer where the metal is in direct contact with the cooling medium; in the interior of the material, the cooling rate is less than critical, and the full hardness is therefore confined to a shallow surface layer. This is known as the "mass effect." In heavier sections, the mass effect is so dominating that even the surface cannot attain the theoretical full hardness of 62-64 Rockwell C.

A large number of stainless steels are standardized, but are not included in the list above. Stainless steels are corrosion resistant and most have excellent mechanical properties. Due to their higher cost and lower machinability, they are not used to any significant extent in fixture construction. However, some fixture components are now commercially available in stainless steel. Stainless steel is also used for purposes where its lack of magnetic properties is of value, such as for separating elements in magnetic chucks and for backing bars in welding fixtures.

Alloy steels, in general, have greater toughness than carbon steels of comparable strength. Also, the wear resistance is greater than for a carbon steel of the same carbon content. They have better hardenability than carbon steels, which works out in two ways: (1) They harden in depth, not just in the surface, and for the hardening they require a less severe heat treatment than carbon steels; (2) They distort less, therefore, during heat treatment, and have better dimensional stability after heat treatment because they contain less residual stresses. The cost is higher, however, and alloy steels are only selected where a carbon steel cannot be used.

**Tool Steels**

All tool steels normally used in jig and fixture construction have a high carbon content. Some of them are carbon steels, but most of them are alloy steels. They attain the highest possible hardness and wear resistance when quenched and can be tempered back to satisfactory toughness without undue loss of hardness. They are not as readily machinable as the carbon and low alloy steels, so that cutting speeds are lower and the machined surface is not as smooth. Where accuracy and dimensional stability are important, it is necessary to pre-machine and stress anneal or normalize the part for stress relief and then machine it to final dimensions.

Tool steels differ widely with respect to distortion during heat treatment. For fixture parts it is desirable to keep distortion at a minimum and this is a major consideration in the selection of the type of tool steel to be used. Practically all fixture requirements can be met with the selection of tool steels listed in Table 3-1, comprising two water hardening steels (W-), three oil hardening steels (O-) and SAE 52100 which is not really a tool steel, but very suitable for some fixture purposes. High-speed steels are not included in the list because they are seldom used in fixtures.

**Table 3-1. Selected Tool Steels for Fixture Components**

<table>
<thead>
<tr>
<th>Chemical Components</th>
<th>Type of Steel</th>
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<tbody>
<tr>
<td></td>
<td>W1</td>
</tr>
<tr>
<td>C</td>
<td>0.60-1.40</td>
</tr>
<tr>
<td>Mn</td>
<td></td>
</tr>
<tr>
<td>Si</td>
<td></td>
</tr>
<tr>
<td>W</td>
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<tr>
<td>Mo</td>
<td></td>
</tr>
<tr>
<td>Cr</td>
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<td>V</td>
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</tbody>
</table>

**Steel Material Selection**

Shop language has its own non-standardized nomenclature for various classes of steel. Machine steel is hot rolled, low carbon steel, as opposed to cold rolled steel. Low carbon tool steel is carbon steel of less than 0.60 percent carbon; the lower
The surfaces of low carbon steel parts that require case hardening must be fully carburized to a depth of at least 1/16 inch.

Cold finished material is preferred to hot rolled wherever possible. It saves machining when it is used in the fixture in standard dimensions and it has a somewhat higher tensile and yield strength and surface hardness than hot rolled material because it is work hardened. It also produces a smoother surface when machined, however, since it contains higher residual stresses than hot rolled material it may warp if it is asymmetrically machined. For this reason, hot rolled material is preferred for parts that are to be finished by grinding, or machined on one side only.

Selection of Proper Steels for Parts

Small and medium-size drill bushings are made of oil hardening tool steels, grades O2 and O3, or of 52100 steel because of their low distortion in heat treatment. Larger drill bushings are made of 8620 steel, carburized and case hardened. Very large bushings are made of low alloy steels in the 41xx group. Cheap bushings are made of 1060 or 1065 carbon steel. Locating parts, such as buttons, pins, and pads, that require a high hardness and are finish ground after heat treatment, are made of water hardening steel W1 or W2, quenched in brine and tempered at 300 to 375°F (149 to 191°C). Cams that cannot be contour ground after heat treatment and, therefore, must be machined or hand filed to final dimensions are made of oil hardening tool steel, grades O2 or O6.

Chuck and vise jaws that can be ground after hardening are made of grades W1 or W2; if finish grinding is not possible, they are made from grades O2 or O6.

Miscellaneous parts of small and medium size that do not carry any significant load but require complete or partial case hardening for wear resistance are made of 1018, 1020, 1133, or 1144 carbon steel. Surfaces that are exposed to light wear only, can be case hardened by cyanizing and do not need subsequent grinding unless a high degree of accuracy is required.

Highly stressed parts, such as arbors or mandrels, boring bars, highly loaded clamps, and fixture details are made from alloy steels with medium carbon content such as 2340, 2345, 3140, 3145, 4140, 5140, 6140, and 6145 with or without heat treatment, depending on the stress level during operation. For particularly heavy duty, oil hardening O6 tool steel is used.

Collets and expanding arbors are particularly critical because they require hard gripping surfaces combined with resilience. They are made from alloy steels with low carbon content, such as 2312, 2320, 3115, 3120, 4615, 6115 and 6120 or from oil hardening tool steel O2 or O6.

Springs are made of carbon steels in a range from 1065 to 1085, also occasionally up to 1095; and from alloy steels 5150, 6150, and 9260. Springs that are manually wound in the shop are made from spring wire; that is, high carbon steel wire commonly known as music or piano wire, heat treated to the proper mechanical properties.

Dowel pins are made of carbon steel, either 1035 or a high carbon steel hardened to 60 Rockwell C. Keys for alignment of parts within the fixture or for aligning the fixture on the machine table, are made from 1045 carbon steel. If hardened surfaces are required for wear resistance, they are made from a tool steel.

Assembly screws and bolts are made of various grades of machine steel, however, it is preferred to use hexagonal socket head cap screws made of heat treated alloy steel, usually with 160,000 to 170,000 psi (1100 to 1170 N/mm²) tensile and 100,000 to 105,000 psi (690 to 724 N/mm²) shear strength. They require less space and are commercially available.
The straps in clamping devices are made of 1020 carbon steel for average conditions, and of 1035 carbon steel in such cases where a lower strength material would require excessive dimensions. The tips of the straps are often case hardened. Hardened washers are used under the nuts in these devices. The bolts are made of 1020 carbon steel; for large loads they are made of a heat treated low alloy steel to avoid excessive dimensions. The nuts are made of a free-machining low carbon steel and case hardened for heavy loads. Pulcrom pins for swinging clamps are made of carbon steel, or tool steel, heat treated to 54-58 Rockwell C. Large pins are made of case hardened, low carbon steel. Eccentrics and cams are made of low carbon steel, case hardened to at least 60 Rockwell C.

Castings

Small secondary parts such as hand knobs, crank arms, etc., are made of malleable iron. Large castings (and sometimes these are very large), are used for parallels, raizer blocks, fixture bases, and in some cases, for fixture bodies; particularly where vibration damping capacity and absence of stresses are of importance. Castings in the first category are not severely stressed and are made of gray cast iron in classes 20, 25 and G2000 with 20,000 to 25,000 psi (138 to 172 N/mm²) tensile strength. Larger fixture bases, and fixture bodies in general, are made of gray cast iron in classes 30, 35, and G3500 with 30,000 to 35,000 (207 to 241 N/mm²) tensile strength. Very large fixture bodies are made of ductile (nodular) cast iron in class 65-45-12 with 65,000 psi (450 N/mm²) tensile strength, and type SP80 Meehanite Ductiron with 80,000 to 100,000 psi (550 to 690 N/mm²) tensile strength. Type GAS50 Meehanite with 50,000 psi (345 N/mm²) tensile strength is also used. Because of its structural homogeneity, it retains high dimensional accuracy in service. It responds to heat treatment and can be hardened locally or on the surface by either flame or the induction process. The design should avoid thickness variations in adjacent sections beyond the ratio of 3:1, otherwise the thick section will have a porous center. Steel castings are rarely used in fixtures.

Aluminum and Magnesium

These two light metals are widely used in the form of tooling plate. They are supplied with finish machined surfaces, are weldable and easily machinable; for many purposes they are competitive with steel.

Aluminum tooling plate is a cast product of a chemical composition equivalent to the 7000 series of aluminum alloys. They are furnace stress relieved and machined (scraped) on both sides to a thickness tolerance of ±0.005 inch, (0.13 mm), a flatness tolerance of 0.010 inch on 8 feet (0.25 mm on 2 1/2 m), and 25 to 40 micro-inch (0.6 to 1.0 μm) surface finish. Standard thicknesses range from 1/4 inch to 6 inches (6 to 150 mm). Thicknesses up to 16 inches (400 mm) are available from stock; above 16 inches by special order. The material weighs 0.101 pound per cubic inch (2.8 g/cm³), and its tensile strength is 24,000 psi (165 N/mm²).

Magnesium tooling plate is a rolling mill product. It is made of AZ31B alloy containing Al, Zn, and Mn. It is thermally stress relieved at 700°F (371°C) and machined on both sides to a thickness tolerance of ±0.010 inch (0.25 mm) and a typical flatness tolerance of 0.010 inch in any 6 feet (0.25 mm on 1.8 m). Standard thicknesses range from 1/4 inch to 6 inches (6 to 150 mm). The material weighs 0.0642 pound per cubic inch (1.78 g per cm³) and its tensile strength is 35,000 psi (240 N/mm²). The compressive yield strength is only 10,000 to 12,000 psi (69 to 83 N/mm²). The machining of magnesium involves a serious fire hazard, because magnesium is combustible in the atmosphere, and the chips ignite easily and burn violently.

Sintered Carbides

Drill bushings for extraordinary long service life are usually made of sintered carbide materials. Two such materials are available; the first is class C-2 sintered tungsten carbide, a straight cobalt grade with 6 percent cobalt and 94 percent tungsten carbide with a hardness of 92 Rockwell A. The second is FERRO-TIC² a machinable carbide which consists of titanium carbide particles bonded in a steel matrix of S-1 tool steel. When the matrix is annealed, the material can be machined with conventional tools. After machining, the material is hardened and tempered and the composite material shows a hardness of 70 Rockwell C. It can be re-annealed, remachined, and rehardened.

Plastics

Plastic tooling is made by casting and by laminating. Plastic tooling materials include phenolics, polyesters, polyvinyls and epoxies. They differ

²Trade name, proprietary to Sintercast Division of Chromalloy American Corp., West Nyack, New York.
widely in physical properties. The physical properties of the end product depend also on the mixing ratio, the curing time, and the curing temperature. The few values that will be quoted in the following, therefore, are representative only; the fixture designer is advised to obtain specific data from the manufacturer for any actual design application.

The four types of plastics can, in principle, all be cast and used as laminating resins. The reinforcing material is usually glass fibers, applied in the form of cloth, mats, or rovings.

Cast phenolics have a compressive strength of 12,000 to 15,000 psi (83 to 103 N/mm²); the compressive strength of cast polyesters varies from 12,000 to 35,000 psi (83 to 241 N/mm²). These plastics exhibit a significant linear shrinkage. Cast epoxy has a compressive strength of 15,000 to 25,000 psi (103 to 172 N/mm²) with an elastic limit of 5000 psi (34 N/mm²), a modulus of elasticity (E) of 0.1 X 10⁶ to 0.8 X 10⁶ psi (690 to 5520 N/mm²), and a linear shrinkage of 0.001 to 0.004 inch per inch (0.025 to 0.10 mm/mm), which is considerably less than the shrinkage of other castable plastics. The use of epoxies as a tooling material is growing rapidly. For this application, epoxies are superior to most other plastics and they are now used almost exclusively for laminates. Epoxy laminates have an elastic limit of about 15,000 psi (103 N/mm²), and a modulus of elasticity ranging from 1.5 X 10⁶ to 3.5 X 10⁶ psi (10,300 to 24,100 N/mm²). The shrinkage is negligible and they are dimensionally stable after curing.

Rigid polyurethane foam is used as a core material and a backup material for fixtures constructed in the form of plates and shells. It has its optimum strength/weight ratio in the density range of 6 to 10 pounds per cubic foot (96 to 160 kg/m³). Representative values of physical properties at 9 pounds per cubic foot (128 kg/m³) are 200 psi (1.4 N/mm²) yield strength, 250 psi (1.7 N/mm²) ultimate compressive strength; E = 10 X 10³ psi (70 N/mm²) at yield, and E has the near constant value of 7.5 X 10³ psi (52 N/mm²) over the operating range from 0 to 150 psi (0 to 1.0 N/mm²) compressive stress.

Potting compounds used for fastening drill bushings in bored holes in plastic drill bushings are high-temperature-resistant epoxy resins. Epoxy resins with metallic fillers ("liquid steel," "liquid aluminum") are used for repair of plastic tooling and for fixtures and tooling for use at elevated temperatures (up to 500 F, or 260 C).

Other Non-metallic Materials

Large plane drill jigs for use in the aircraft industry where light weight is essential for easy handling, and high strength is not required, are made of flat sheets of various materials, such as plywood and laminated plastic sheets. A recently developed product is Benelex, which is made of wood chips and consists essentially of cellulose fibers and lignite. It is hard, rigid, smooth, impervious to water and oil, and has excellent dimensional stability. It machines like wood, the tensile strength is 7600 psi (52 N/mm²), and the modulus of elasticity is 1.3 X 10⁶ psi (9000 N/mm²). It is available in sheets in thicknesses from 1/4 to 2 inches (6 to 50 mm).

3 Trade name, proprietary to Masonite Corp., Chicago, Illinois.
Locating Principles

Locating Principles, Flat Surfaces

A part without scribed lines and punched centers can only be located from its surfaces and this is done by providing them with the necessary number of restraints. To restrain the part on a surface against only one direction of motion, as was shown in Fig. 3-1, is termed defining the part and implies the addition of a subsequent clamping action to maintain positive contact with the restraining element. The part is single defined as long as it is restrained on one surface only, double defined when it is restrained on two surfaces, and fully defined when it is restrained on three surfaces. It is a condition here that the defining surfaces are not mutually parallel; generally, but not as an absolute condition, the three defining surfaces are perpendicular to each other.

The rectangular block shown in Fig. 4-1 is a general example of defining and locating from flat surfaces. The block is single defined as long as it only rests on the horizontal base of the fixture (position a), double defined when it is moved to full contact with the vertical longitudinal strip (position b), and fully defined when it is also moved endwise to contact with the end strip (position c).

To define a part from two parallel and offset flat surfaces results in overdefining. Because of the tolerances, the part cannot simultaneously be brought into effective contact on the two surfaces. It will either hang or tilt (Fig. 4-2), depending on the nature of the clamping. A similar situation exists for parts with two or more concentric cylindrical surfaces. When the part has to be located on two offset surfaces, it can be done satisfactorily by locating them on three points.

Nesting

A part may be located on, or restrained between, two or more surfaces such that motion is prevented in the two opposite directions on at least one line. The part is said to be nested, to be nesting, or to nest within, the restraining elements. In Fig. 4-3a, the part is fully defined and single nested, in b and c it is double nested, and in d it is fully nested. Full nesting requires that the fixture has a detachable
cover to provide access to the interior of the fixture, and openings in the fixture walls to allow the operations to be performed.

Fig. 4-3. Single, double, and full nesting.

Nesting requires that it be possible to move the part to a position between locating surfaces or points. On the other hand, it must also fit as closely as possible between the mating surfaces when in position. Any clearance, no matter how small, will allow motion between part and fixture that will generate a certain small amount of displacement and misalignment. An interference fit would define the location without ambiguity, but does not readily permit the part to be moved into position. Therefore, the class of fit actually selected must be a clearance fit, equivalent to one of the tighter classes of these fits. For the sake of clarity, the clearances shown in the illustrations are grossly exaggerated.

Nesting can take many forms. Nesting surfaces do not have to be parallel and opposite. If the fixture in Fig. 4-1 is rotated around one edge, it can be seen (compare with Fig. 4-4) that the part is actually double nested on two perpendicular surfaces when it is in position C. In addition, the diagonal plane in the part is now centered with respect to the fixture. The concept of centering is of great importance and will be discussed in detail in Chapter 6, in the section on Circular Locators, and in Chapter 9, Centralizers. If the fixture is set on a corner with a corner diagonal vertical, then the part is fully nested on three corner surfaces, and is also centered.

Fig. 4-4. Modifications of the principle of nesting.

The example in Fig. 4-1 is an illustration of the elimination of the six degrees of freedom by means of contact between large surfaces. It points back to the application of the same principle as is shown in Fig. 3-1. The base plane is equivalent to the first three locating points, the side strip is equivalent to the next two points, and the end strip is equivalent to the last point. This set of equivalences can be formulated as the “3-2-1 locating principle.” The locating function of the side and end strips and points is somewhat different from the function of the base plane and base points. To underline this difference, all locating points above the base are termed “stops.”

Other equivalences, however, are possible. A set of two points is equivalent to one strip; a plane is equivalent to two parallel strips, or to one strip and a point (see Figs. 4-5, and 4-4a, b and c). A locating point is not a mathematical point, it is often a small flat surface (a pad). The locating elements should
be spaced as widely as possible. This open spacing provides the best obtainable stability against the acting loads (gravity, clamping and cutting forces), and minimizes any error that may be caused by a small misalignment or displacement of a locating element.

The 3-2-1 Principle

The 3-2-1 principle represents the minimum requirements for locating elements. The locators, together with the clamps (represented by arrows C in Fig. 3-1) which hold the part in place, provide equilibrium of all forces, but do not necessarily also guarantee stability during machining. Usually, stability is satisfactory if the three base buttons are widely spaced and the resultant cutting force hits the base plane well within the triangular area between the buttons. If it hits outside of this area, then it generates a moment which tends to tilt or overturn the part. The pressure and frictional forces from the clamps may be able to counteract this moment, but this solution is not considered good practice, because vibrations and shocks from machining can cause the part to slip in the clamps.

The 4-2-1 Principle

By the addition of a fourth locator in the base, the shape of the supporting area can be changed from a triangle to a rectangle, as shown in Fig. 3-1d, and provides the required stability. The principle may be termed the “4-2-1 locating principle.” For rough castings, one of the four base locators may be adjustable. Such locators are described in Chapter 12, Supporting Elements.

If the locating surface is machined, all locators may be fixed, and this offers an advantage in another respect. When the part is properly seated on its four locators, it feels stable, but if a chip or some other foreign matter has lodged itself on a locator, or if the locating surface is warped, the part will rock. This is noticeable to the operator and serves as a warning that there is a defect in the set-up which must be corrected.

Error Possibilities

The use of large locating fixture surfaces is only feasible when the matching part surfaces are compatible with respect to tolerances and geometry. This is not necessarily the case, even for surfaces already machined, because fixture surfaces are usually finished to closer tolerances than are most production parts. The consequences of incompatible tolerances will be explained in Chapter 5 with respect to Fig. 5-2.

The most common errors in part surface geometry are convex and concave curvature, twist, and angular errors. The effects of curvature and twist are shown in exaggerated form in Fig. 4-6. Convex curved and twisted surfaces will not accurately define the location since they may cause the part to rock. With curved surfaces and insufficient rigidity, the part may also be distorted (bent) when clamped in the fixture but after it is released from the clamp it will spring back and the previously flat new surface will now be curved. Even with distortion-free clamping, the curved part may still be insufficiently supported and may deflect under the cutting forces.

Fig. 4-6. The effects of locating from curved and twisted surfaces.

Angular errors on adjacent surfaces can cause various cases of misalignment, particularly if the clamp-
Locating Principles, Cylindrical Locators

Cylindrical surfaces will usually be located by nesting in or on completely or partly matching surfaces. A fixture base with a side and an end strip can almost but not completely locate it, as shown in Fig. 4-9. In position a, the part stands on the base and three degrees of freedom have been removed. When moved to position c, two more (but not three) degrees of freedom have been removed. The cylinder is now nested in a V-block in the same way as shown previously in Fig. 4-4a, and it is also centered with respect to the V. The sixth degree of freedom, rotation around a vertical axis, has not yet been removed.

The same incomplete locating can be accomplished by placing the part inside a matching cylindrical holder—an outside cylindrical locator (see Fig. 4-10), but it still is free to rotate. Rotation can now be prevented and the part locked in position by means of a clamping device employing friction. If the significant part configuration consists entirely of cylinders and perpendicular planes, it has no preferred diameter, and any position is as good as any other position with respect to the machining operations to be performed in the fixture. If, however, the part has a projecting or a receding surface, no matter how it is shaped, then it has one or several preferred diameters to which the machined surface must be related in the way determined in the part design, and such a preferred diameter, or diameters, must be held to a predetermined location within the
LOCATING PRINCIPLES

Fig. 4-10. Incomplete locating by means of a simple cylindrical locator.

Fig. 4-11. Complete locating of a cylinder by means of an outside cylindrical locator and a radial locator.

fixture. For this purpose the fixture must be provided with one additional locating element which eliminates the sixth degree of freedom, rotation, by locating the preferred diameter(s). This can be done in a great variety of ways and a few representative examples are shown in Fig. 4-11.

A cylindrical locator can also be applied to the inside of a cylindrical cavity and takes the shape of a mandrel, a plug, or a flange. Some examples are shown in Fig. 4-12.

Two factors common to all rotational locators are: that they act on a point of a radius in the part, and that they restrain motion of that point in a tangential direction. These will be termed "radial locators." Usually, and preferably, the direction of the actual contact pressure should be perpendicular to the radius at the point of contact, a condition which is fully satisfied in Fig. 4-12b; approximately satisfied in Figs. 4-11a, b, and c, and 4-12a and c; but not, however, in Figs. 4-11d and 4-12d.

The radial locator may be a small pin, fitting in a hole, or it may be large and formed as another plug. Additionally, the cylindrical locator may be so small that it also takes the shape of a pin. Locating by such systems, illustrated in Fig. 4-13, is termed "dual cylinder location," and is widely used.

Any location using cylindrical locators involves nesting, and therefore requires clearance which, in turn, affects the locating accuracy. As indicated in Fig. 4-14, sketch a, the position of the center of the part may vary as much as the clearance and may be offset from its nominal position as much as one-half of the clearance. The application of a clamping pressure (see sketch b), forces the offset to one side, but does not eliminate it, and the poor nesting at the contact point opposite the clamping pressure permits the part to shift slightly to one side or the other. If the part does not have a good locating base surface, but, for example, terminates in a point (as shown in sketch c), it is also subject to misalignment resulting in a maximum angular variation $\theta$ of the axis direction determined by:

$$\theta = \frac{2(D_F - D_P)}{L} \text{ radians} = \frac{360}{\pi} \cdot \frac{D_F - D_P}{L} \text{ degrees}$$

once again confirming the fundamental rule that locating points should be as far apart as possible. It
is strongly recommended, that the locating points be placed in mutually perpendicular planes. If a locating plane is inclined against the perpendicular, as shown in Fig. 4-15, a transverse force component is generated that tends to lift the part from the base points. A dirt accumulation on the locator of thickness $T$ produces a locating error $E=T$ when the locating plane is perpendicular, but a larger error $E = \frac{T}{\cos a}$ occurs when the locating plane is inclined an angle $a$ against the perpendicular.

Offset and misalignments, as discussed above, are eliminated by the use of conical (tapered) locators, because they do not require clearance but provide positive contact. They belong to the class of centralizing devices to be discussed in Chapter 9.
Fig. 4-15. The effects of locating against a perpendicular and an inclined plane.
Preparation for Locating

Locating Unmachined Surfaces

One basic purpose of a fixture is to produce parts that are within specified tolerances. It is the machined surfaces on the individual parts that define and determine the distances to all principal axes and other system lines and planes within the finished product. It is obvious that all such dimensions must be correct, of course, but it is also necessary that any remaining unmachined surfaces maintain their proper location relative to system lines and to each other to avoid interference with each other and with moving parts of the machine, to secure the required material thicknesses, and to provide uniform machining allowances with full cleanup on all machined surfaces. A drastically exaggerated example of a violation of this rule is shown in Fig. 5-1.

![Fig. 5-1. Correctly and incorrectly located center lines. The cylinder to the left was machined with the center lines correctly located with respect to the outer surfaces. The cylinder to the right was machined with a gross error in the location and the direction of the center line with respect to the outer surfaces.](image)

In job shop production, these conditions are met by the layout of the parts prior to machining. System lines and centers are scribed and punched into the surfaces of the part which is then set up in the machine tool by measurements taken to these lines and centers. One important purpose of a fixture is to eliminate this layout operation; the raw part usually comes to the fixture without such lines, centers, or other markings, and all locating has to be done from the surfaces and contours as they exist. It is therefore important for the design of the fixture, and particularly for its locating elements, to know the dimensional tolerances that may be expected (or even better, may be guaranteed) on the raw part.

They will vary from case to case, according to application and purpose of the product, from plant to plant, and from supplier to supplier. In the specific case, however, the applicable tolerances will normally be made available to the fixture designer.

Tolerances will usually be fairly consistent within each group of materials, depending on the type and class, and also the size of the part. General rules for tolerances and other dimensional variations are presented in the sections following. They will be found useful for the fixture designer in the absence of specific prescribed tolerances, and may also serve as a base for the valuation of any given or proposed tolerances.

Machining allowances are, in a way, related to tolerances and must also be taken into consideration by the fixture designer. The maximum tolerance on a surface is the theoretical lower limit for the machining allowance. Where possible, the actual machining allowance should be obtained from the production planning department or from suppliers of raw parts. As a substitute, an estimated value may be used.

For an order-of-magnitude estimate, it may be assumed that machining allowances increase with the overall size of the raw part. For gray iron castings made in green sand molds in sizes from 20 to 100 inches (500 to 2500 mm), average machining allowances vary from 3/16 to 7/16 inch (5 to 10 mm),
they are a little higher for surfaces located in the cope, a little lower for surfaces located in the drag. Practice varies between different foundries; some consider 1/8 inch (3 mm) as the minimum machining allowance, also for smaller castings. Malleable iron and nonferrous alloy castings require 33 percent less, and steel castings 50 percent more, than gray iron castings.

For forgings, the weight \( W \) (pounds or kg) is the parameter by which the machining allowance may be estimated. Small hammer and press forgings (hand forged) require from 1/16 to 1/8 inch (1/12 to 3 mm) on each surface. For this type of forging (from 15 pounds (7 kg), and up) the allowance per surface can be taken as

\[
0.05 \frac{3}{W} \text{ inch or } 1.65 \frac{3}{W^1} \text{ mm}
\]

where: \( W = \) weight in pounds, and \( W^1 = \) weight in kilograms.

For closed die forgings (drop forgings and other machine forged parts) the allowance required is from 40 percent (for solid and bulky shapes) to 60 percent (for elongated shapes) of the value estimated for a hand forging of the same weight. Minimum allowance for all forgings is 1/32 inch (0.08 mm) because of scale pits and other localized surface defects and decarburization.

**Castings**

A casting is by no means a mathematical reproduction of the pattern; not even of the mold cavity. Some cast materials will shrink, others will expand during solidification. All of them shrink during the subsequent cooling period; the resulting total shrinkage depends on type and composition of the metal, the pouring temperature, and the cooling rate. Slight variations in the composition may occur from charge to charge and can affect the shrinkage. Uneven shrinkage often results from differences in wall thickness and may cause warping.

Common values for shrinkage are shown in the following:

<table>
<thead>
<tr>
<th>Material</th>
<th>Shrinkage Rate</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Linear</td>
</tr>
<tr>
<td>Gray cast iron</td>
<td>1/8 in., per foot</td>
</tr>
<tr>
<td>Same, heavy sections</td>
<td>...</td>
</tr>
<tr>
<td>White (chilled) cast iron</td>
<td>...</td>
</tr>
<tr>
<td>Malleable cast iron</td>
<td>...</td>
</tr>
<tr>
<td>Cast steel</td>
<td>1/4 in., per foot</td>
</tr>
<tr>
<td>Brass</td>
<td>3/16 in., per foot</td>
</tr>
</tbody>
</table>

Shrinkage Rate for Various Materials

<table>
<thead>
<tr>
<th>Material (cont.)</th>
<th>Shrinkage Rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bronze</td>
<td>...</td>
</tr>
<tr>
<td>Zinc</td>
<td>...</td>
</tr>
<tr>
<td>Aluminum</td>
<td>3/32 to 5/32 in. per foot</td>
</tr>
<tr>
<td>small castings of simple design</td>
<td>...</td>
</tr>
<tr>
<td>large castings or complicated shapes</td>
<td>...</td>
</tr>
<tr>
<td>Al-Si alloys</td>
<td>...</td>
</tr>
<tr>
<td>Aluminum alloys for automotive pistons</td>
<td>...</td>
</tr>
<tr>
<td>Magnesium</td>
<td>3/32 to 5/32 in. per foot</td>
</tr>
</tbody>
</table>

On large castings the apparent shrinkage will be less than the metallurgical shrinkage, because the pattern is rapped in the mold before it is drawn and thereby slightly expands the mold cavity. This is of significance for large castings only.

With respect to warping, only a few general rules can be formulated. The complete process of differentiated shrinkage rates during solidification is complicated. Heavy sections, and sections that are shielded against loss of heat, will lag behind during cooling, and the end result is that such sections will show increased apparent shrinkage. An I-beam-type gray iron casting with one thick and one thin flange will, consequently, be concave lengthwise (hollow) on the side of the thick flange. An upper limit for the maximum deflection \( y_{\text{max}} \) of such a beam of length \( L \) and height \( H \) is:

\[
y_{\text{max}} = \frac{1}{3200} \frac{L^2}{H}
\]

Greater warpage may be expected for malleable and steel castings. A channel- or U-shaped section may open up at the top, because the bottom shrinks and the free edges are held in position by the mold or core.

The lower limit for tolerances on castings can be taken as one-half of the shrinkage. This assumes favorable conditions, such as regular shapes without tendency to warping. However, while close tolerances sometimes may be desired for some functional reason, the economical viewpoint calls for the most liberal tolerances that design considerations can allow. Unnecessary close tolerances add to cost and increase the scrap hazard. Any conscientious production man will select the widest tolerances that he can get away with, and the fixture designer should be aware of that.

Representative and rather realistic tolerances are:
### Table: Tolerance Range for Various Materials

<table>
<thead>
<tr>
<th>Material</th>
<th>Description</th>
<th>Tolerance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Iron castings, gray, white, and malleable:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>small castings</td>
<td>±0.08</td>
<td>2</td>
</tr>
<tr>
<td>large castings</td>
<td>up to ±0.4</td>
<td>10</td>
</tr>
<tr>
<td>Permanent mold castings on dimensions within one mold part:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>aluminum and magnesium</td>
<td>±1.5%</td>
<td>0.25</td>
</tr>
<tr>
<td>copper and copper-base alloys</td>
<td>min. 0.010</td>
<td>0.25</td>
</tr>
<tr>
<td>Permanent mold castings on dimensions perpendicular to parting plane or</td>
<td></td>
<td></td>
</tr>
<tr>
<td>the parting plane and mold</td>
<td>add 0.010</td>
<td>0.25</td>
</tr>
<tr>
<td>Die castings</td>
<td></td>
<td></td>
</tr>
<tr>
<td>on dimensions within aluminum and magnesium</td>
<td>±1.5% to ±2.5%</td>
<td>0.08</td>
</tr>
<tr>
<td>zinc-base alloys</td>
<td>min. 0.003</td>
<td>0.08</td>
</tr>
<tr>
<td>copper and copper-base alloys</td>
<td>±1% to ±2.5%</td>
<td>0.06</td>
</tr>
<tr>
<td>min. 0.0025</td>
<td></td>
<td></td>
</tr>
<tr>
<td>lead-base alloys</td>
<td>±0.3%</td>
<td>0.13</td>
</tr>
<tr>
<td>min. 0.005</td>
<td></td>
<td></td>
</tr>
<tr>
<td>for dimensions across the parting plane</td>
<td></td>
<td></td>
</tr>
<tr>
<td>for dimensions determined by a movable core</td>
<td>add 50%</td>
<td></td>
</tr>
<tr>
<td>add 100%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

A more specific set of rules, applicable to greensand iron castings up to 16 inches (400 mm) in size, is the following:

<table>
<thead>
<tr>
<th>Description</th>
<th>Tolerance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall external dimensions</td>
<td>±0.030 inch (0.8 mm) for the first 3 inches (75 mm) ±0.008 inch (0.20 mm) for each additional inch (25 mm)</td>
</tr>
<tr>
<td>within the same part of the mold</td>
<td></td>
</tr>
<tr>
<td>parallel to the parting plane</td>
<td></td>
</tr>
<tr>
<td>On dimensions perpendicular to</td>
<td></td>
</tr>
<tr>
<td>the parting plane</td>
<td></td>
</tr>
<tr>
<td></td>
<td>from ±0.02 (for small castings) to +0.06 (for large castings) inch per inch (mm/mm)</td>
</tr>
</tbody>
</table>
Average draft values are:

<table>
<thead>
<tr>
<th>Casting Method</th>
<th>Amount of Draft</th>
</tr>
</thead>
<tbody>
<tr>
<td>For pattern drawn from mold:</td>
<td></td>
</tr>
<tr>
<td>external surfaces</td>
<td>$\frac{1}{100} (\approx \frac{1}{2})$</td>
</tr>
<tr>
<td>internal surfaces</td>
<td>$\frac{1}{50} (\approx 2^\circ)$</td>
</tr>
<tr>
<td>For mold lifted from pattern:</td>
<td></td>
</tr>
<tr>
<td>external surfaces</td>
<td>$\frac{1}{20} (\approx 1/2^\circ)$</td>
</tr>
<tr>
<td>internal surfaces</td>
<td>$\frac{1}{10} (\approx 5^\circ)$</td>
</tr>
</tbody>
</table>

Deep castings do not permit a very large draft as it would too greatly distort the dimensions.

Minimum values are:

<table>
<thead>
<tr>
<th>Casting Method</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>For pattern drawn from mold:</td>
<td></td>
</tr>
<tr>
<td>external surfaces, curved</td>
<td>$\frac{1}{1000}$</td>
</tr>
<tr>
<td>external surfaces, flat</td>
<td>$\frac{1}{500}$</td>
</tr>
<tr>
<td>ribs and webs, curved</td>
<td>$\frac{1}{100}$</td>
</tr>
<tr>
<td>ribs and webs, flat</td>
<td>$\frac{1}{200}$</td>
</tr>
<tr>
<td>small holes</td>
<td>up to $\frac{1}{10}$</td>
</tr>
<tr>
<td>For mold lifted from pattern:</td>
<td>up to $\frac{1}{20}$</td>
</tr>
</tbody>
</table>

A rough measure of uniformity in castings is provided by some average weight tolerances, which, for gray iron castings, are 5 percent when made from solid patterns, 10 percent when made with sweep patterns, and for malleable iron castings, 5 percent when machine molded, and 10 percent when hand molded.

The uniformity and accuracy of castings (gray iron, malleable iron, and modular or ductile iron) is higher from permanent molds, and molds with metallic cores and inserts, than from sand molds; higher from machine molding than from hand molding; higher from dry-sand molds than from green-sand molds; and significantly higher when the castings are made in shell molds.

Castings must always be expected to show various minor irregularities which must be tolerated and do not justify rejection except in extreme cases. Typical examples are a mismatch at the parting line between cope (upper mold) and drag (lower mold), flash or fins at the same parting lines and along edges of cored cavities, remnants of the gates, displacement of cores resulting in uneven wall thickness and machining allowance, and displacement of loose pattern pieces resulting in off-set bosses, ears, ribs, and the like.

Forgings

Handmade hammer forgings will, as a rule, not be manufactured in quantities that justify machining in a fixture. However, hammer and press forged parts from ferrous and nonferrous metals are used in moderate quantities in various industries such as the weapon and aerospace industries, and these forgings may need fixtures because of intricate and accurate machining requirements. The most common forged raw parts are impression die forgings which, again, may be drop forgings (closed die forgings) and upset forgings.

For estimating forging tolerances, materials can be classified by stiffness as follows: Low stiffness—aluminum, magnesium, copper, and brass; Medium stiffness—carbon and low alloy steel, stainless steel (400 series); and High stiffness—stainless steel (300 series), titanium, super-alloys, and refractory metals (Columbium, Cb; Molybdenum, Mo; Tantalum, Ta; Tungsten, W). Tolerance data listed in the following without material specification may be applied to all three classes.

Hammer and press forgings are seldom fully free-hand forged, but are made with the use of flat and simple open-face dies. For such forgings, tolerances can be estimated from nominal dimensions and weight.

For elongated shapes of length $L$ (inches or mm) and any transverse dimension (width, height, diameter, etc.) $D$ (inches or mm), estimated tolerances are:

**on length—**

$$T_L = \pm [0.05 + 0.003 (L + 10 D)] \text{ inch}$$

$$T_L = \pm [1.3 + 0.003 (L + 10 D)] \text{ mm}$$

**on a transverse dimension—**

$$T_D = \pm [0.02 + 0.028 (D + \frac{1}{70} L)] \text{ inch}$$

$$T_D = \pm [0.5 + 0.028 (D + \frac{1}{70} L)] \text{ mm}$$
For a part of weight \( W \) (pounds or kg) and unspecified shape, the estimated tolerance is:

\[
T = \pm 0.05 \frac{3}{\sqrt[3]{W}} \text{ inch} \quad \text{or} \quad T = \pm 1.65 \frac{3}{\sqrt[3]{W}} \text{ mm}
\]

On closed die forgings the die cavity dimensions and the shrinkage control all dimensions inside one die block and parallel to the parting plane, such as width; length; diameters; etc. Parts formed in two die blocks may exhibit a mismatch which affects (adds to) the overall part dimensions.

Thickness dimensions, as measured perpendicular to the parting plane, are likewise controlled by the die cavity dimensions and the shrinkage, but a more significant factor is the degree of die closure—which again depends on the amount of excess stock and how well this is forced out into the flash. Thickness dimensions may, therefore, be less accurate than other dimensions.

Die cavity dimensions depend on the initial accuracy to which the die was sunk and polished, and the amount of subsequent wear. Initial dimensions

---

**Table 5-1a. Die Forging Tolerance Data—English Units**

<table>
<thead>
<tr>
<th>Area in Parting Plane, Square Inches</th>
<th>Steel Low Alloy</th>
<th>Stainless Steel Series 400</th>
<th>Stainless Steel Series 300</th>
<th>Super Alloys</th>
<th>Titanium</th>
<th>Refractory Metals* Co, Mo, Ta, W 2014</th>
<th>Aluminum 7075</th>
<th>Magnesium</th>
<th>Wear Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.004</td>
<td>0.005</td>
<td>0.006</td>
<td>0.007</td>
<td>0.008</td>
<td>0.009</td>
<td>0.012</td>
<td>0.004</td>
<td>0.007</td>
</tr>
<tr>
<td>Wear Factor</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Thickness Tolerance, inch</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0-10</td>
<td>1/32</td>
<td>1/32</td>
<td>1/16</td>
<td>1/16</td>
<td>3/32</td>
<td>1/32</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>11-30</td>
<td>1/16</td>
<td>1/16</td>
<td>3/32</td>
<td>3/32</td>
<td>1/8</td>
<td>1/32</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>51-100</td>
<td>1/8</td>
<td>1/8</td>
<td>3/32</td>
<td>1/8</td>
<td>3/32</td>
<td>3/32</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>501-1000</td>
<td>3/16</td>
<td>1/4</td>
<td>1/4</td>
<td>3/16</td>
<td>5/16</td>
<td>3/32</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1000 and over</td>
<td>1/4</td>
<td>5/16</td>
<td>5/16</td>
<td>3/8</td>
<td>3/8</td>
<td>1/4</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Forging Weight After Trimming, Pounds</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Mismatch Tolerance, inch</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>to 5</td>
<td>1/64</td>
<td>1/32</td>
<td>1/32</td>
<td>1/16</td>
<td>1/16</td>
<td>1/64</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>25.1-50</td>
<td>3/64</td>
<td>1/16</td>
<td>1/16</td>
<td>1/8</td>
<td>1/8</td>
<td>3/32</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>50.1-100</td>
<td>1/16</td>
<td>3/32</td>
<td>3/32</td>
<td>5/32</td>
<td>5/32</td>
<td>1/16</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>100.1-200</td>
<td>3/32</td>
<td>1/8</td>
<td>3/16</td>
<td>3/16</td>
<td>3/16</td>
<td>3/32</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>200.1-500</td>
<td>1/8</td>
<td>5/32</td>
<td>5/32</td>
<td>1/4</td>
<td>1/4</td>
<td>1/8</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>500.1-1000</td>
<td>5/32</td>
<td>3/16</td>
<td>5/16</td>
<td>5/16</td>
<td>5/16</td>
<td>1/8</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Forging Weight After Trimming, Pounds</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Flash Extension, Max., inch</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>to 10</td>
<td>1/32</td>
<td>1/16</td>
<td>1/16</td>
<td>1/8</td>
<td>1/32</td>
<td>1/32</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>10.1-25</td>
<td>1/16</td>
<td>3/32</td>
<td>3/16</td>
<td>1/4</td>
<td>1/6</td>
<td>1/6</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>25.1-50</td>
<td>3/32</td>
<td>1/8</td>
<td>3/16</td>
<td>1/4</td>
<td>1/4</td>
<td>1/4</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>50.1-100</td>
<td>1/8</td>
<td>3/16</td>
<td>5/16</td>
<td>1/2</td>
<td>1/2</td>
<td>1/2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>100.1-200</td>
<td>3/16</td>
<td>1/4</td>
<td>5/16</td>
<td>5/8</td>
<td>5/8</td>
<td>5/8</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*Co = Cobalt, Mo = Molybdenum, Ta = Tantalum, and W = Tungsten
**Table 5-1b. Die Forging Tolerance Data—SI (Metric) Units**

<table>
<thead>
<tr>
<th>Area in Parting Plane, Square mm</th>
<th>Steel Low Carbon Series 400</th>
<th>Stainless Steel Series 300</th>
<th>Super Alloys 300</th>
<th>Titanium Co, Mo, Ta, W</th>
<th>Refractory Metals* Co, Mo, Ta, W</th>
<th>Aluminum 2014</th>
<th>Magnesium 7075</th>
<th>Wear Factor</th>
<th>mismatch Tolerance, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>0 to 6,500</td>
<td>0.8</td>
<td>0.8</td>
<td>1.5</td>
<td>1.5</td>
<td>2.4</td>
<td>0.8</td>
<td>0.004</td>
<td>0.005</td>
<td>0.006</td>
</tr>
<tr>
<td>6,500 to 20,000</td>
<td>1.5</td>
<td>1.5</td>
<td>2.4</td>
<td>3</td>
<td>4</td>
<td>0.8</td>
<td>0.006</td>
<td>0.007</td>
<td>0.008</td>
</tr>
<tr>
<td>20,000 to 32,500</td>
<td>2.4</td>
<td>2.4</td>
<td>3</td>
<td>5</td>
<td>5</td>
<td>1.5</td>
<td>0.008</td>
<td>0.009</td>
<td>0.012</td>
</tr>
<tr>
<td>32,500 to 65,000</td>
<td>3</td>
<td>3</td>
<td>5</td>
<td>5</td>
<td>5</td>
<td>2.4</td>
<td>0.009</td>
<td>0.012</td>
<td>0.016</td>
</tr>
<tr>
<td>65,000 to 323,000</td>
<td>4</td>
<td>5</td>
<td>6</td>
<td>6</td>
<td>6</td>
<td>3</td>
<td>0.012</td>
<td>0.016</td>
<td>0.020</td>
</tr>
<tr>
<td>323,000 to 645,000</td>
<td>5</td>
<td>6</td>
<td>8</td>
<td>8</td>
<td>8</td>
<td>5</td>
<td>0.016</td>
<td>0.020</td>
<td>0.025</td>
</tr>
<tr>
<td>645,000 and over</td>
<td>6</td>
<td>8</td>
<td>10</td>
<td>10</td>
<td>10</td>
<td>6</td>
<td>0.020</td>
<td>0.025</td>
<td>0.030</td>
</tr>
<tr>
<td>Forging Weight After Trimming, kg, approx.</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0 to 2.3</td>
<td>0.4</td>
<td>0.8</td>
<td>1.5</td>
<td>0.4</td>
<td>0.8</td>
<td>0.8</td>
<td>0.005</td>
<td>0.007</td>
<td>0.009</td>
</tr>
<tr>
<td>2.4 to 11.3</td>
<td>0.8</td>
<td>1.2</td>
<td>2.4</td>
<td>1.2</td>
<td>2.4</td>
<td>1.2</td>
<td>0.007</td>
<td>0.009</td>
<td>0.012</td>
</tr>
<tr>
<td>11.4 to 22.7</td>
<td>1.2</td>
<td>1.5</td>
<td>3</td>
<td>1.5</td>
<td>3</td>
<td>1.5</td>
<td>0.009</td>
<td>0.012</td>
<td>0.016</td>
</tr>
<tr>
<td>22.8 to 45.4</td>
<td>1.5</td>
<td>2.4</td>
<td>4</td>
<td>2.4</td>
<td>4</td>
<td>2.4</td>
<td>0.012</td>
<td>0.016</td>
<td>0.020</td>
</tr>
<tr>
<td>45.5 to 90.7</td>
<td>2.4</td>
<td>3</td>
<td>5</td>
<td>2.4</td>
<td>5</td>
<td>2.4</td>
<td>0.016</td>
<td>0.020</td>
<td>0.025</td>
</tr>
<tr>
<td>90.8 to 226.8</td>
<td>3</td>
<td>4</td>
<td>6</td>
<td>3</td>
<td>6</td>
<td>3</td>
<td>0.020</td>
<td>0.025</td>
<td>0.030</td>
</tr>
<tr>
<td>226.9 to 453.6</td>
<td>4</td>
<td>5</td>
<td>8</td>
<td>4</td>
<td>8</td>
<td>4</td>
<td>0.025</td>
<td>0.030</td>
<td>0.035</td>
</tr>
<tr>
<td>453.7 and over</td>
<td>5</td>
<td>6</td>
<td>10</td>
<td>5</td>
<td>10</td>
<td>5</td>
<td>0.030</td>
<td>0.035</td>
<td>0.040</td>
</tr>
<tr>
<td>Flash Extension, Max., mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0 to 4.5</td>
<td>0.8</td>
<td>1.5</td>
<td>3</td>
<td>0.8</td>
<td>0.8</td>
<td>0.8</td>
<td>0.007</td>
<td>0.009</td>
<td>0.012</td>
</tr>
<tr>
<td>4.6 to 11.3</td>
<td>1.5</td>
<td>2.4</td>
<td>3</td>
<td>1.5</td>
<td>2.4</td>
<td>1.5</td>
<td>0.009</td>
<td>0.012</td>
<td>0.016</td>
</tr>
<tr>
<td>11.4 to 22.7</td>
<td>2.4</td>
<td>3</td>
<td>8</td>
<td>3</td>
<td>8</td>
<td>3</td>
<td>0.012</td>
<td>0.016</td>
<td>0.020</td>
</tr>
<tr>
<td>22.8 to 45.4</td>
<td>3</td>
<td>5</td>
<td>10</td>
<td>3</td>
<td>10</td>
<td>5</td>
<td>0.016</td>
<td>0.020</td>
<td>0.025</td>
</tr>
<tr>
<td>45.5 to 90.7</td>
<td>5</td>
<td>6</td>
<td>13</td>
<td>5</td>
<td>13</td>
<td>5</td>
<td>0.020</td>
<td>0.025</td>
<td>0.030</td>
</tr>
<tr>
<td>90.8 to 226.8</td>
<td>6</td>
<td>8</td>
<td>16</td>
<td>6</td>
<td>16</td>
<td>6</td>
<td>0.025</td>
<td>0.030</td>
<td>0.035</td>
</tr>
<tr>
<td>226.9 to 453.6</td>
<td>8</td>
<td>10</td>
<td>19</td>
<td>8</td>
<td>19</td>
<td>8</td>
<td>0.030</td>
<td>0.035</td>
<td>0.040</td>
</tr>
<tr>
<td>453.7 and over</td>
<td>10</td>
<td>13</td>
<td>19</td>
<td>10</td>
<td>19</td>
<td>10</td>
<td>0.035</td>
<td>0.040</td>
<td>0.045</td>
</tr>
</tbody>
</table>

*Co = Cobalt, Mo = Molybdenum, Ta = Tantalum, and W = Tungsten

**Table 5-1b** can be held to relatively very close tolerances which are considered included in the shrinkage tolerances. However, dies are also subject to severe wear and are, for economic reasons, allowed to wear considerably during their useful service life.

**Shrinkage** tolerances, also known as "length-width" tolerances, are: ±0.003 inch per inch (mm/mm) of nominal dimension. **Wear** tolerances on external and internal dimensions are: wear factor (from Table 5-1, below) multiplied by greatest external dimension (length or diameter). On external dimensions, the wear tolerance is plus, on internal dimensions it is minus. Wear tolerances do not apply to center-to-center distances.

Thickness tolerances are based on part area in the parting plane and can be taken from Table 5-1. Table values apply to parts not exceeding 6 inches (150 mm) of depth within any one die block, as measured perpendicular to the parting plane. For such parts of forgings that exceed this limit, an additional tolerance is applied, equal to: ±0.003 inch per inch (mm/mm) of any such dimension.
Thickness tolerances are always positive, meaning that incomplete filling of the die cavity is not acceptable.

Mismatch tolerances and flash extension, which is the maximum distance that the flash may protrude from the forging body, are both positive; they are based on the forging weight after trimming, and can be taken from Table 5-1. Flash thickness ranges from 1/16 to 1/4 inch.

Straightness tolerances mean the limitation imposed on deviations of surfaces and centerlines from the nominal configuration and are added to previously estimated tolerances. Forgings and parts within a forging can be classified by shape as elongated, flat, or bulky, and one forging may well comprise parts belonging to more than one class.

Straightness tolerances are:
- for elongated shapes—0.003 inch per inch (mm/mm) of length
- for flat shapes—0.008 inch per inch (mm/mm) of length, width, or diameter.

Bulky parts require no straightness tolerance.

The values are for medium stiffness materials and assume that the forgings have been mechanically straightened as required. For low stiffness materials, deduct 33 percent; for high stiffness materials, add 33 percent.

All die forgings must have draft. In some extreme and special cases (in aluminum and magnesium forgings of the extrusion type) the applied draft may be 1 degree or even zero. However, the most common values on external surfaces are 5 to 7 degrees for medium-stiffness materials, down to 3 degrees for low-stiffness materials, and up to 10 degrees for high-stiffness materials. Internal surfaces (pockets) require higher drafts, from 10 to 13 degrees. All drafts carry a +2, −1 degree tolerance.

Overall tolerances for closed die forgings range from 5 to 15 percent of thicknesses, and from 0.5 to 1.5 percent of widths, lengths, and diameters. In comparison, tolerances on upset forgings are 25 percent higher on axial lengths and flange diameters, but 25 percent less on some individual intermediate axial dimensions, such as flange thicknesses. Mismatch tolerances are the same. Cavities require a TIR concentricity tolerance of 1.3 percent of cavity diameter. Upset forgings do not show flash and, in many cases, require little or no draft.

Tolerances quoted are “commercial.” Finer tolerances, known as “close” can be obtained. The values are approximately 33 to 50 percent less than “commercial.” Calculated tolerances are rounded off to two decimal places, then converted to nearest higher 1/32 inch (1 mm) and entered on drawings.

The fixture designer must be prepared to encounter some minor defects which are considered acceptable and passed by inspection—such as scale pits, shallow depressions caused by scale accumulation; mistrimmed edges, where the flash protrudes unevenly around the forging; small fins and rags, driven into the metal surface; cold shuts, produced by material folded against itself; small unfilled areas; and conditioning pits, where surface defects have been ground away.

The dimensionally most reliable configurations are those formed within one die block, and flat surfaces parallel to the parting plane. All forgings produced within one life period of the die are usually very uniform. The same applies to sheared flash contours. Slight differences may be expected when a die is reconditioned, and also if more than one die is in use.

Weldments

Production parts to be machined in a fixture will usually be fabricated by methods entailing closer control and better uniformity than job shop welded parts and can therefore be made to closer tolerances, particularly when the welding is performed in fixtures. Tolerances on finished welded parts depend largely on the distortion during and after welding. The tolerances obtained must be ascertained from case to case, and only broad and general statements can be made about them.

Automatic welding results in less distortion than hand welding. Arc welds distort less than gas welds. Heavy welds distort more than light welds, but heavy sections distort less than light sections. On the other hand, weldments from light sections are easier to straighten mechanically. Resistance welds distort less than fusion welds. Least distortion is found in flash-butt welding, where length tolerances can be held to ±0.02 inch (0.5 mm). When the dies are not self-centering, a maximum offset equal to the sum of the tolerances on the part diameters or thicknesses may be expected.

In the absence of specific information, tolerances for resistance weldments can be taken as for die forgings, and tolerances for arc welded parts can be taken as 50 percent of the tolerances for castings of comparable dimensions.

Torch-cut parts will display the thickness tolerances of the stock material with an addition for the burr which, after proper cleaning for slag, may be from 0.01 to 0.06 inch (0.25 to 1.5 mm) on either side. Contours can be held to +0.015 inch (0.38 mm) on small parts, and ±1/16 inch (1.5 mm) on
large parts with automatic and tracer control, and ±0.1 inch (2.5 mm) with manual feed. The cut edges may deviate 1/4 degree from the perpendicular position. With inert-gas tungsten cutting at high feed rates, edges may be beveled as much as 5 degrees.

**Mill Products**

This class comprises rolled, drawn, and extruded shapes. Detailed tolerances are available from suppliers' catalogs, a few illustrative examples only, are shown in the following:

<table>
<thead>
<tr>
<th>Material</th>
<th>Tolerance, inch</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel rod, low carbon and low alloy, round or square:</td>
<td></td>
</tr>
<tr>
<td>Hot Rolled</td>
<td></td>
</tr>
<tr>
<td>1-inch diameter or side</td>
<td>±0.009</td>
</tr>
<tr>
<td>2-inch diameter or side</td>
<td>±1/64</td>
</tr>
<tr>
<td>4-inch diameter or side</td>
<td>+1/16, -0</td>
</tr>
<tr>
<td>Cold finished</td>
<td></td>
</tr>
<tr>
<td>Carbon Alloy</td>
<td></td>
</tr>
<tr>
<td>Round</td>
<td>Square</td>
</tr>
<tr>
<td>1-inch diameter or side</td>
<td>-0.002 -0.004</td>
</tr>
<tr>
<td>2-inch diameter or side</td>
<td>-0.003 -0.005</td>
</tr>
<tr>
<td>4-inch diameter or side</td>
<td>-0.005 -0.006</td>
</tr>
<tr>
<td>Note: Minus tolerances only.</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Aluminum rod, round or square:</th>
<th>Rolled</th>
<th>Cold Finished</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Round</td>
<td>Square</td>
</tr>
<tr>
<td>1-inch diameter or side</td>
<td>...</td>
<td>...</td>
</tr>
<tr>
<td>2-inch diameter or side</td>
<td>±0.006</td>
<td>±0.016</td>
</tr>
<tr>
<td>4-inch diameter or side</td>
<td>±0.031</td>
<td>±0.020</td>
</tr>
<tr>
<td>Aluminum hollow shapes, extruded:</td>
<td>±15% of nominal</td>
<td></td>
</tr>
<tr>
<td>wall thickness overall dimensions on a hollow section</td>
<td>1.5–2.5% of nominal</td>
<td></td>
</tr>
</tbody>
</table>

With respect to tubes and pipes, the fixture designer should know that they not only have diameter tolerances, but also liberal tolerances on out-of-round and wall thickness variations.

**Press Products**

This class comprises sheet metal parts produced by shearing, punching; stamping, drawing, and pressing with dies in a mechanical press. Some basic tolerances are usually very close. Thickness tolerances for cold rolled carbon steel sheets range from below ±0.001 inch (0.03 mm) to ±0.005 inch (0.13 mm) for thicknesses up to 1/4 inch (6 mm). Contours of punched flat parts may vary 0.001 to 0.002 inch (0.03 to 0.05 mm) as long as the same tool is used without reconditioning. The same may be expected for small stamped and drawn parts.

Shapes formed by bending may show different springback. Drawn parts will have thickness variation and, if not trimmed, a scalloped edge contour (earing) from planar anisotropy in the stock. All sheared edges have a burr on the exit side, and for thicknesses above 1/8 inch (3 mm), also a rounded entrance edge.

Apart from these variations, parts produced in the same tool will come out with a high degree of uniformity.

**Machined Parts**

Machined surfaces have closer tolerances than raw parts and are therefore, a priori, more suitable for locating a part within a fixture for further machining. They should not, however, be indiscriminately accepted for this purpose. The basic requirement is that the tolerance on the already machined surface must be satisfactory for the correct tolerance to be obtained in the following operation within the fixture. As an illustration (see Fig. 5-2) assume surfaces A and B are already finished to a tolerance of ±0.005 inch (0.13 mm), and surface C must hold ±0.002 inch (0.05 mm) against B. Surface A, presenting a wide bearing area, would appear desirable for locating but the presence of the ±0.005 inch (0.13 mm) tolerance from B to A prohibits machining of C to ±0.002 inch (0.05 mm) from B, no matter how close tolerance t is taken, and A must therefore be rejected as the locating surface.

Blueprint tolerances, if uncritically accepted without part inspection, could cause the fixture designer many disappointments. Nominally plane surfaces could be convex, concave, or twisted, from improper clamping; gradual tool wear; inaccurate setting of a milling cutter; or distortion (warp) from stress relief. Broached configurations might be offset or
Fig. 5-2. Consideration of tolerances in locating.

tilted, due to the elasticity of the broach. Ground surfaces on thin parts could show heat distortion. Nominally square edges and corners could be out-of-angle from incorrect clamping. Sawed surfaces, even when machine sawed, are neither straight, flat, nor dimensionally correct. All machined edges will have a burr on the side of tool exit.

Heat Treated Parts

Such parts may distort and show relatively large deviations from nominal shape and dimensions. Almost any geometrical element may be affected. Overall dimensions, including center distances, may increase or decrease, hole diameters may become larger or smaller, circles may go out-of-round, and flat or straight parts may curve or twist. The distortions cannot be predicted except in rather general terms and may well vary from piece to piece. Control of distortion requires careful stress relief of parts prior to hardening, and depends also on the experience of the heat treater and his skillful application of time-honored tricks of the trade. More reliable and consistent control of heat treat distortion is effected by having the part clamped in a fixture during the process.

As a rough estimate, tolerances required to cover for heat treat distortion can be taken as ±0.05 to ±0.15 percent, additive to prior part tolerances.

Plastic Parts, Molded

Parts made in various sizes from plastics (thermo-plastic as well as thermosetting) are widely used and are manufactured in large quantities. They have excellent surface quality as formed, but that is not necessarily identical with high dimensional accuracy, because of shrinkage, uniform or nonuniform, and sometimes dimensional changes during aging. While they can be formed (cast or pressed) in the mold to finished dimensions for many applications, other purposes may require some machining operations such as drilling and other processing of holes, where hole location is critical, or grinding of flat surfaces, and the fixture designer may well encounter the assignment of designing fixtures for these materials.

The amount of shrinkage varies with the type of plastic material and filler, and may in extremes, range between 0.001 and 0.012 inch per inch (mm/mm) of nominal dimension. Representative average values for single-cavity hot molds, taken as the dimensional difference of mold and part at ambient temperature, are:

<table>
<thead>
<tr>
<th>Material</th>
<th>inch per inch (mm/mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Phenolic with wood-flour filler, and urea</td>
<td>0.006 to 0.010</td>
</tr>
<tr>
<td>Cellulose acetate</td>
<td>0.002 to 0.010</td>
</tr>
<tr>
<td>Phenolic with fabric or asbestos filler, and methyl methacrylate</td>
<td>0.002 to 0.006</td>
</tr>
<tr>
<td>Polystyrene</td>
<td>0.001 to 0.003</td>
</tr>
</tbody>
</table>

Resulting tolerances can be taken as follows:

<table>
<thead>
<tr>
<th>Parallel to parting plane</th>
<th>±0.005 inch per inch (mm/mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Perpendicular to parting plane</td>
<td>add 0.015 inch (0.38 mm)</td>
</tr>
<tr>
<td>Warpage, perpendicular to nominal surface</td>
<td>±0.003 inch per inch (mm/mm)</td>
</tr>
</tbody>
</table>

The nominal dimension L (inches or mm) can also be taken into account in the tolerance T by the empirical formula:

\[
T = 0.006 \sqrt{L} \text{ inches} \quad \text{or} \quad T = 0.03 \sqrt{L} \text{ mm}
\]

Somewhat higher tolerances should be selected for center-to-center distances of bosses or molded holes, and for multiple cavity molds, and should be doubled for cold-molded parts.

Some plastics can be molded without draft, when generous fillet radii are provided. Others may require a small draft of 1 to 2 degrees on smaller pieces, up to a total maximum draft of 0.04 inch (1 mm) on the side.

Laminated plastic parts are formed over or inside a die by building up consecutive layers of impregnated fibrous material (frequently glass fiber cloth) with a liquid resin as the impregnating and adhesive material. They are cured at moderate pressure that is provided by means of an evacuated bag. The glass fiber reinforcement is strong and rigid and the quantity of resin used is small, thus the dimensional tolerances on the molded surface are very close. The man in the shop will usually say that they are zero, however, the designer should assume a finite, but small tolerance, such as ±0.003 inch per inch (mm/mm) for small parts with ±0.010 inch (0.25 mm) as
the upper limit for larger parts. It should be remembered that the material is flexible and thin parts can easily be elastically distorted.

Thickness is often defined by the number of layers and the thickness of the stock. This may vary from 0.003 inch to 0.050 inch (0.08 to 1.3 mm) for glass cloth; when a greater thickness is required a more loosely woven matting is used. When properly applied, the resin fills vacancies only and theoretically does not contribute to thickness. The process is manual and may not always be closely controlled. Consequently, some tolerance must be allowed on the thickness, 1/32 inch (0.8 mm) as an upper limit with glass cloth and 1/8 inch (3 mm) with matting.

**Plastics, Prefabricated Shapes**

The tolerances vary widely with material, shape, and manufacturing method, and catalogs should be consulted for specific information. Representative values for the tolerance ranges that may be expected are given in the chart to the right. The first figure refers to the smallest, the last figure to the largest dimension.

For pressed and rolled laminated plastic plates, the thickness tolerances are from 0.100 inch per inch (mm/mm) down to 0.030 inch per inch (mm/mm). Warp and twist must not exceed from 5 percent down to 1/4 percent of the plate dimension (length, width, or diagonal). For tubes and rods, the tolerances on the OD are from 1 percent down to 0.3 percent, and on tube wall thickness from 20 percent down to 4 percent. The larger tolerances are for the smaller dimensions.

<table>
<thead>
<tr>
<th>Kinds of Tolerances and Material</th>
<th>Tolerance Ranges, percent</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Thickness tolerances for plates:</strong></td>
<td></td>
</tr>
<tr>
<td>Plexiglas, Class A</td>
<td>15, down to 6, for 1 inch (25 mm), and above double up</td>
</tr>
<tr>
<td>Class C</td>
<td></td>
</tr>
<tr>
<td>Nylon, polycarbonate, and</td>
<td></td>
</tr>
<tr>
<td>styrene plates, slabs, and discs</td>
<td></td>
</tr>
<tr>
<td>Vinyl and Teflon sheets</td>
<td></td>
</tr>
<tr>
<td><strong>Diameter tolerances:</strong></td>
<td></td>
</tr>
<tr>
<td>extruded Nylon rod</td>
<td>2 to 12</td>
</tr>
<tr>
<td>molded Teflon rods and cylinders</td>
<td>25 for small thicknesses, down to 5</td>
</tr>
<tr>
<td><strong>Tolerances for tubing:</strong></td>
<td></td>
</tr>
<tr>
<td>acrylic resin, on OD</td>
<td>0.3 to 0.8</td>
</tr>
<tr>
<td>Nylon and Teflon, on OD wall</td>
<td>0.6 to 3</td>
</tr>
<tr>
<td>thicknesses</td>
<td>0.9, down to 0.5</td>
</tr>
<tr>
<td></td>
<td>5, down to 2.5</td>
</tr>
<tr>
<td></td>
<td>15, down to 5</td>
</tr>
</tbody>
</table>

For proper application of the resin, the resin fills vacancies only and theoretically does not contribute to thickness. The process is manual and may not always be closely controlled. Consequently, some tolerance must be allowed on the thickness, 1/32 inch (0.8 mm) as an upper limit with glass cloth and 1/8 inch (3 mm) with matting.
Design of Locating Components

General Requirements

Locators and stops present a number of requirements other than merely the proper locating of the part. The most important of these are: (a) resistance to wear, (b) provision for replacement, (c) visibility, (d) accessibility for cleaning, and (e) protection against chips. All of these points must be taken into account in the selection and design of the locating elements. At this stage of the design development, the fixture designer is also advised to think ahead and review the aspects of the loading and unloading of the fixture, as previously discussed.

Sighting

The simplest method of locating, not previously discussed, is locating by sighting to locating lines or other markings in the jig. A normal prerequisite for the use of this method is that the part has an acceptable base surface on which it can rest in a stable position in the jig. Once this is accomplished, the part is moved until its contour coincides sufficiently close with the markings and is then clamped in position. The method can be used for raw parts such as castings, welded parts, and forgings, where no great accuracy is required between the part contour and the surfaces to be machined. Such parts involve large tolerances on the part contour, and for this reason, each marking is made with multiple lines to make sure that the markings are not totally obscured. With this simple device, it is always possible to locate and center the part fairly well. Two simple examples with two different styles of marking lines are shown in Fig. 6-1a and b. For correct locating, the method shown in these two diagrams depends entirely on the attention and manual skill of the operator. However, less manual skill is required in the modifications c and d, when the part is located by manipulation of finger screws. This dependence on human judgment is not necessarily always a liability, since it also permits the operator to adjust to the correct location despite bumps or other local irregularities on the part contour.

An example of a different technique, still based on sighting, is a drill jig, shown in Fig. 6-2. The drill plate carries sighting apertures with beveled edges, and the part contour is lined up with the edges of
these apertures which may be round or elongated holes or slots. The part is here adjusted to its final position by means of cams and screws.

**Nesting**

The next logical step, again applicable to flat parts or parts with at least one flat or fairly flat surface, is to nest it along its contour or along the contour on its extreme ends. An example is shown in Fig. 6-3a. The semicircular notches provide space for the operator's fingers for inserting and removing the part. The groove at the contour allows for the burr.

The minimum clearance between nest and part is determined by the part tolerance and, obviously, permits some displacement. The nests of irregular shapes is, therefore, limited to parts that are already manufactured with rather close contour tolerances. It is usually very suitable for parts punched from sheet and plate yet less suitable for forgings, partly because of the draft, partly because the contour where the flash has been trimmed off may be offset with respect to the forging body.

The contour can also be simulated by blocks with V-notches (Fig. 6-3b). These are cheaper to make and can also be made adjustable to accommodate for variations in the part contour from wear or reconditioning of the tool with which the contour was made.

A simpler and cheaper method is nesting with pins (Fig. 6-3c). All pins are shown as cylindrical. As will be explained later, locating pins are, in most other cases, provided with flat contact surfaces. This is frequently omitted in contour nesting fixtures when the pins are not exposed to any substantial load, and also when the pin has to contact a curve on its concave side.

Dust and chip fragments which, when accumulated, prevent proper seating and cause misalignment of the part are difficult to clean out of nesting fixtures, particularly the full nest type. Dirt space allowance is therefore required. A burr groove provides an excellent dirt space. V-blocks and pins can also be undercut for the same purpose.

**Three-dimensional Nesting**

Nesting fixtures for parts with an irregular surface in three dimensions can be made by machining, which is highly expensive; and by casting, which is the more commonly used method. The fixture is a box of ample dimensions to contain the part and
the nest is formed by sealing the part against the box and pouring a castable material onto the part. An example of this is shown in Fig. 6-4.

![Fig. 6-4. Three-dimensional nesting of an irregular surface.](image)

Castable materials used in nesting are plastics and soft metals. The plastics are phenolic tooling resin and epoxy, reinforced, when needed, with glass cloth. They are light, inexpensive, and easy to repair if required. The curing temperature is 300 to 350°F (149 to 177°C). The metals used are Kirksite®, a group of zinc base alloys with a melting range of 717 to 745°F (381 to 396°C), and poured at 850°F (454°C); various lead-tin-antimony alloys with a melting range from 460 to 500°F (238 to 260°C); and Cerrobend®, an alloy containing bismuth and melting at 158°F (70°C), that is, below the boiling point of water. The cast surface is ground, if necessary, and polished to provide some clearance.

This type of nesting fixture is suitable for parts with fairly close tolerances, such as die castings and stampings. However, the nesting surface can be subdivided by machining grooves and recesses, and reduced to locating pads, as indicated by the dotted lines. In this way the fixture can be made to accommodate parts with wider tolerances, such as ordinary castings and forgings.

**Integral Locators**

For parts of simple geometry and with flat machined surfaces of sufficiently close tolerances with respect to flatness and dimensions, the simplest locating solution is to provide mating locating surfaces integral with the fixture. The principle, as applied to a cast fixture, is illustrated in Fig. 6-5. The machined locating surfaces are indicated by f. The diagram shows large continuous surfaces as well as individual pads. Some aspects of the use of large locating surfaces have already been discussed. Large bearing areas provide excellent support for the part and permit a great deal of freedom in the placement of clamping forces without danger of elastic distortion (deflection, springing) of the part; also, as the bearing pressures are low the rate of wear is reduced. On the other hand, large locating areas require a high degree of accuracy in the part as well as in the fixture, for accuracy is lost if the fixture distorts as a result of poor stress relief. Dirt space, however, is only available along the perimeter, thus large surfaces are apt to accumulate dirt and chip fragments.

It is possible to subdivide large locating surfaces without loss of their advantages. As shown in Fig. 6-6, the first step is to provide grooves for the accumulation of dirt; (left) two sets of crossing grooves change the original surface into smaller pads without serious sacrifice of supporting and bearing areas; the individual surface areas are reduced, which also facilitates cleaning. The next step (right), is to reduce the original surface to strips, and finally, not shown, to reduce each strip to small pads. All these changes facilitate the drainage of coolant as well as the machining of the fixture; particularly the finish grinding.

![Fig. 6-5. Integral locating pads.](image)

![Fig. 6-6. The reduction of large locating surfaces by means of grooves.](image)

The two patterns shown are only modifications of details and the geometrical concept of the locating surface as being on one plane remains unchanged. There is no objection here to the use of four corner pads with the inherent advantage of maximum stability, and there is, therefore, no obligation for reducing the locating surface to three points.

Locating strips and pads are easily formed in a cast fixture body because they are molded by means of the patterns and cores. They are just as easily
provided in welded fixture bodies. Typical examples are shown in Fig. 6-7a and b.

![Fig. 6-7. Examples of welded jigs with welded locating strips and pads (B in part b).](image)

Cast or welded integral locating surfaces suffer from a common drawback—they are not directly replaceable when worn. It is not practical to harden a cast-iron fixture (steel castings are very seldom used for fixtures) and the only means available for controlling the wear rate is to be as generous as possible with the dimensions of the locating surfaces to keep the bearing pressure low. The same applies, in general, to welded fixtures. However, although not widely used, it is possible to make the locating pads and strips from low-grade tool steel and weld them into the fixture body. Apart from this, worn surfaces, both on cast and on welded fixtures, can be reconditioned by weld-depositing a layer of material and remachining it to the original dimensions. The welding involves some risk of distortion, and a careful inspection of the fixture is required after the repair. A safer, but more expensive method, is to remove the worn pads by machining and install new pieces made from hardened steel, secured by means of screws and dowel pins.

### Separate Locators

For the reasons explained above, it is preferred to use separate components for locating purposes, to install them in such a way that they can be removed and replaced when worn, to provide them with a hard working surface, and to protect them against chip and dirt accumulation.

Locators have been made from bronze, presumably because of its use as a bearing material. Locating wear strips of synthetic sapphire have shown a wear resistance several thousand times that of steel. However, these material selections are exotic and highly unusual. The widely accepted rule is to make locators from steel, occasionally chromium plated, or from cast or sintered carbide. Small locators are made from low alloy steel, heat treated to 41-45 Rockwell C, large locators from low carbon steel, Carburized and case hardened.

### Wear on Locators

Wear is a complicated process and has been extensively studied. Most wear research is for the purpose of better bearing design, but the conditions in a bearing (lubrication, regular motion, cleanliness) do not apply to fixture locators. With dust, chip fragments, rust, and scale always involved in their use, the conditions on locator surfaces are far from ideal, and the type of wear to be expected is an intermediate between contact wear (metallic contact between clean or corroded surfaces, no lubricant, no significant amount of foreign particles) and abrasive wear. Locator surfaces do have one advantage with respect to wear, namely, that they are not exposed to very much sliding motion by the part. Motion takes place only during loading, and the maximum load on the locators during this period is only the weight of the part. With correctly designed clamps, there should be no motion when the clamping pressure is applied nor when the working load from the cutting operation is applied.

It would be desirable if quantitative data for permissible locator loads could be quoted but in general, they cannot. The only somewhat relevant figure is a value for hardened steel of 25 pounds per square inch (0.17 N/mm²), found by French and Herschman as a boundary between a lower pressure region with slow wear and a higher pressure region of more rapid, increasing wear. The curve from which this value is extracted is shown in Fig. 6-8.

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Example — The largest size of rest button taken from manufacturer's standards has a 1 1/4-inch (32-mm) total diameter. Deducing for the chamfer, the effective diameter is:

\[ 0.92 \times 1.25 = 1.15 \text{ inches (29.2 mm)} \]

and with three buttons, conforming to the 3:2:1 principle, the maximum load carried within the 25 psi pressure limit is:

\[ \frac{\pi}{4} \times 1.15^2 \times 3 \times 25 = \text{78.9, or approximately 80 pounds (36 kg)} \]

A very large number of parts weigh less than 80 pounds and with the use of conventional buttons, a long locator service life can be expected. The fixture designer should not despair if the locator pressure significantly exceeds the limit quoted, but he must make ample provision for replacement of worn locators. No fixture is really expected to last forever, and larger parts usually do not occur in such quantities that locator wear becomes a great problem. When necessary, larger locators can be designed, but under no circumstances should the designer feel obligated to employ locators with excessive bearing areas, chiefly because it is difficult to keep them free of chip fragments.

In difficult cases selecting a more wear-resistant material is justified. The ratio of wear resistance of the four materials—case-hardened carbon steel, hardened tool steel, cast tungsten carbide (Stellite type), and sintered tungsten carbide—is 1:2:3:40. Any discussion of high wear-resistance applies only to fixtures for large quantity production. By rule-of-thumb it is accepted that unhardened locators are sufficient for tooling for 100 parts or less.

Buttons

The three most common types of locating “points” are buttons, pins, and pads. Conical points are ideal from the mathematical viewpoint only, and should not be used because they lack sufficient bearing surface area and would rapidly wear down.

Buttons are round and have either a flat head or a crowned (spherical) head, as shown in Fig. 6-9. They are made of steel; usually medium alloy steel or low grade tool steel, heat treated to 40-45 Rockwell C, or (larger sizes only) low carbon steel, such as AISI 1113, carburized and case-hardened to 53-57 Rockwell C, the choice determined by heat treatment considerations. Buttons are precision parts and are, therefore, ground after heat treating; sufficient relief for grinding must be provided between the shank and the head. Flat buttons are used against machined surfaces only; crowned buttons are primarily for use against unmachined surfaces, but can also be used for locating machined surfaces. However, they do not provide a well-defined bearing area.

Buttons of these types when used as base locators are commercially termed “rest” buttons; when used for side and end stops they are then termed “stop” buttons.

Installation of the button in the fixture body is done with a press fit in a cylindrical bore (reamed, precision bored, or ground). For this purpose the shank ends with a 30-degree chamfer. The bore goes through the fixture wall; a blind hole will trap air during pressing and does not permit easy removal of the button for replacement. The fixture surface is then machined to provide positive support and additional alignment for the head. By providing a boss around the hole, the machining is reduced to a spot facing; on a flat surface, it can be done by countersinking.

While the shanks on commercial buttons are supplied with standardized tolerances, resulting in an oversize ranging from max. 0.0010 to max. 0.0015 inch (0.03 to 0.04 mm) within the available diameter range, there is no formal standard for the interference required relative to the hole, nor to the hole diameter tolerances. However, it is generally assumed that the hole is finished with a reamer with max. oversize of 0.0002 inch (.005 mm) when new. An analysis of these figures indicates that the fit actually obtained will fall in the range from interference-fit class LN 3 to force-fit class FN 2. The
class FN 2 fit represents the upper limit, which is in good agreement with the fact that it (the FN 2 fit) is about the tightest fit to be used in cast iron. It should be remembered that the reamer, even if it holds the 0.0002-inch oversize, may well produce a larger hole, and consequently a lighter fit, if it is allowed to wobble during the reaming operation.\(^2\)

When a plane (or a line) is defined by three (or two) buttons, they are surface ground across their faces after installation to ensure that the plane (or line) is parallel to the corresponding outer surface of the fixture.

With a good press fit and a machined surface on the fixture wall, the installation of the button is accurate, safe, and economical. It is also proposed (in the literature and in catalogs of fixture components) to use a threaded shank in a tapped hole. In this case, the button also has a hexagonal section for a wrench, as shown in Fig. 6-10.\(^3\) In general, this practice is not recommended. A screw thread requires clearance and is less accurate with respect to location and direction. The button is not locked and may be loosened by vibration. A fatigue failure or accidental overload (a blow) may break off the head and make the shank difficult to remove. These threaded buttons are for permanent installation and must be screwed in tightly; they are not intended to be adjustable in height. Actual adjustable stops and supports will be described later.

Fig. 6-9. Locating buttons.

Fig. 6-10 (Left). Locating button with screw thread and a hexagonal section.

Fig. 6-11 (Right). A hollow locating button.

Hollow buttons (Fig. 6-11) fastened by separate screws, are used occasionally as they are a little cheaper to install. The screw head must be countersunk safely below the face of the button, which leaves a small cavity for the collection of chip fragments and is difficult to clean.

Rest and stop buttons are commercially available in standardized dimensions. Few cases are encountered within the range of standardized dimensions


\[^3\] E. Thaulow, Maskinarbejde (Copenhagen: G.E.C. Gad's Forlag, 1930) vol. II.
where standard buttons cannot be used; in such cases, and when larger sizes are required, well-balanced dimensions for stable buttons can be taken from the formulas below. The symbols refer to Fig. 6-9. The principal dimension is the overall diameter $D$. Each diameter $D$ permits a range of heights $H$. The lower limit of the range is for the purpose of safely clearing the fixture base and any accumulation of dirt and chips; the upper limit is determined by a stability consideration. For equal shank dimension $B$, the crowned button has a smaller overall diameter $D$, as no nominally defined bearing area is required.

For flat buttons, $H$ can be selected:

from $1/3 \ D$ (but not less than $3/16$ inch [5 mm]) to $4/3 \ D$ (but not more than 1 inch [25 mm])

$$B = 3/4 \ (D - 1/8)$$  
$$L = 1/2 \ (D + H)$$

The formulas, except the one for $B$, are valid in English and in metric units. With metric units, use:

$$B = 3/4 \ (D - 3)$$

For crowned buttons, $H$ can be selected:

from $1/3 \ D$ to $D$, and

$$R = 3/2 \ D$$  
$$B = 3/4 \ D$$  
$$L = 3/4 \ D$$

Pins

A pin is a cylindrical component that is contacted on its side. It follows from this function that the height of a pin is not a critical dimension. Buttons can be substituted for pins, but pins cannot be substituted for buttons. Pins used as locators are installed by a press fit in the same manner as a button with or without a shank of a reduced diameter. Pins are used to make a nest and, generally, as side stops and for locating in holes, an application which will be further discussed later.

A number of typical applications of pins and buttons for side stops are shown in Fig. 6-12. In most of the sketches, no provision is shown for dirt and chip relief spaces.

Round pins (and buttons) for side stops can be used on concave and unmachined surfaces. For use on plane machined surfaces, the pin or button has a flat to mate the surface on the part. For high precision, these flats are ground after installation of the pins in the fixture. Generally, the use of a pin as a side stop is a little primitive. The more usual method of making a side stop is to use a button mounted in the side wall of the fixture, with its face mating the side surface of the part as shown in Fig. 6-12c. Pins as side stops should be used only on shallow parts with light side loads to avoid loading the pin with a large bending moment.

Pads

Pads are usually flat components made from similar steels and heat treated to similar hardness levels as buttons. They are ground flat and parallel, sometimes also ground on parts of their perimeter and are installed on machined surfaces in the fixture body. They are used primarily as base locators in cases where rest buttons do not provide sufficient bearing area, as side and end locators, and as nest locators.
The edges and corners of a pad are usually not rounded, beveled, or chamfered as are the edges on a button, but are only slightly broken and lightly polished to remove burrs and make them smooth to the touch. The reason for this difference is somewhat obscure. This is a case where a design detail is based on habit rather than on calculation or rational logic. Pads located down in the interior of a cast or welded fixture are not chamfered on their edges as they are not easily accessible to the machine tool. Loose pads should look like fixed pads and, therefore, they are also left with their corners and edges intact. In all fairness, it should be noted first, that sharp edges on a locating pad may be useful in scraping dirt off the mating part, and second, that the chamfering of a pad, particularly one of an irregular

![Image of diagrams](image-url)
outline, is quite an expensive operation because it requires considerable handwork, while the chamfering of a button is a rapid and inexpensive screw machine operation.

Pads are fastened by means of screws with well-countersunk heads, and their position is secured by means of dowel pins (also other means, as required), since screws are fasteners only and are not capable of precision locating anything. The correct use of dowel pins follows certain rules which apply not only to locating pads but to any loose part to be permanently installed with significant precision. A number of representative cases are shown in Fig. 6-13. (Lower case letters refer to the particular diagram.)

In principle, two dowel pins are required for locating a component and they are placed as far apart as possible (a, c). The holes are drilled undersize and reamed to size after the pieces are fixed in position by means of the screws. If the one part, such as a locator pad, is hardened, the holes in that part are reamed to size before heat treatment, and the mating holes in the fixture wall are reamed to the correct size and location through the hardened holes. This may be considered a less-than-ideal compromise solution. A better solution, and one that is used occasionally, is to leave one section of the part unhardened and to place the dowel pins in that section (b).

In many cases, it is possible to reduce the number of dowel pins to one, namely, when other locating surfaces of sufficient precision are available to assist in defining the position of the part. A key and keyseat (d) or a recess (e) may serve this purpose. One screw, a keyseat or recess, and one dowel pin define a position (f). If the position in the direction of the keyseat or recess is not critical, a dowel pin is not even needed. Two screws and a keyseat will define the part (g). Two dowel pins substitute for the keyseat, assisted by four or two screws (h). Two screws and one dowel pin may occasionally suffice (i), namely, if the orientation is not critical. On the other hand, a part nested in a well-fitting recess, held with three screws and secured by one dowel pin (j) has an extremely well-defined and secured position. Parts with cylindrical shanks fitting closely in holes are completely defined by one dowel pin only (k, l).

As with the shanks for the buttons, dowel-pin holes are drilled through so that the pins can be driven out again, when necessary. The recommended bearing length of a dowel pin in each part is 1 1/2 to 2 times the diameter of the pin.

Dowel pins are cylindrical (straight), or tapered (Fig. 6-14). The standard taper is 1/4 inch per foot (1:48). (“Taper” is diameter difference divided by length.) Straight and tapered pins are commercially available. The straight type is available unhardened as well as hardened and ground. For permanent assembly (apart from the possibility of infrequent replacement of a component) the fit of the dowel pin can be a press fit in each part. This serves the purpose of most fixture applications; in cases where occasional disassembly is anticipated, it is common practice to give the dowel pin a press fit in one part and a tight sliding (slip) fit in the other part. Again, this is not very common in fixture design practice. Tapered pins are easily loosened by the application of light pressure or a blow on the small end. They are, therefore, often preferred for parts that require frequent disassembly. However, the tapered pin does not produce and maintain as accurate an alignment between parts as do the straight part with a press and a sliding fit. In extreme cases where parts must be disassembled very frequently, the sliding fit will wear in time and accurate alignment is lost. In such cases, a hardened and ground tapered pin gives much better service. A more sophisticated version of the tapered pin, which greatly facilitates its removal, is made with a threaded end and a nut as shown in Fig. 6-15. The nut is backed off when the pin is driven into place. When the nut is tightened, it gently loosens the pin. A compromise pin design is the tapered pin with a short hexagonal head. When flat pads used as base locators are fastened by the means described, the countersunk screw heads offer places for the accumulation of chip fragments that are difficult to clean away. Unbroken pad surfaces can be obtained if the pads

4 E. Thaulow, Maskinarbejde (Copenhagen: G.E.C. Gad’s Forlag, 1930) vol. II.
are fastened and located by means of screws and dowel pins from the reverse side and with blind holes. The method is somewhat cumbersome and is not widely used, but it is a legitimate possibility and undeniably it does serve the purpose of providing an unbroken bearing surface.

Dowel pins are used extensively in the construction of built-up fixture bodies, and a detailed description of dowel pin techniques is presented in Chapter 15, Design of Fixture Bodies.

Circular Locators

Circular locators take the form of pins, mandrels, plugs, and recesses for inside locating; and hollow cylinders, rings, and recesses for outside locating. In principle, they are nesting devices, and as such, they share the two problems of jamming and clearance versus locating accuracy.

Jamming is mainly a result of friction. If there were no friction, the part would always slide smoothly into the locator. The jamming process is also affected by the amount of clearance, the length of engagement, and the steadiness of the hand of the operator.

When jamming occurs, it always begins when the part has entered a short distance into an outside locator or around an inside locator. A case of jamming is shown in Fig. 6-16. The outer cylinder (the locator) has diameter \( W \), and the inner cylinder (the part) has diameter \( W - C \), where \( C \) is the clearance. The part has entered the locator over a short length \( L \), the length of engagement. If the part is slightly tilted, as shown, then one side of the leading edge comes into contact with the inside of the locator and is caught by the friction. If additional pressure is applied to the part, it serves only to increase the friction and the tilt, and thus jams the part.

\[
L = \mu W
\]

where \( \mu \) is the coefficient of friction and \( W \) is the width of the opening.

The dimensions of the relief grooves can be standardized. No such standard exists as yet in the United States. A German standard (DIN-Norm 6338 in Vorbereitung) for locating pins has been proposed with dimensions closely approximating those which can be derived from the theory and with a chamfer for pre-positioning. Converted to easy formulas, the recommended dimensions (see Fig. 6-17) are:

\[
L_1 = 0.02 D
\]
\[
L_2 = 0.12 D
\]
\[
L_3 \approx \frac{1}{3} \sqrt{D} \quad \text{(with } L_3 \text{ and } D \text{ in inches)}
\]
\[
L_3 \approx 1.7 \sqrt{D} \quad \text{(with } L_3 \text{ and } D \text{ in mm)}
\]
\[
d = 0.97 D
\]
dimension of the part is also a factor in determining the maximum possible angle of tilt, as seen in Fig. 6-18. A point A on the locating surface of the part can swing in a circle around a center B on the outer perimeter of the part. Any length $A_1D$ of the locator that makes it stay within the circle around B is jam-free, even if it is greater than $L_1$, and the diagonal $CD$ is longer than the diameter $AC$ inside the part.

![Fig. 6-18. The geometry of a jam-free circular locator in combination with a flat locating surface.](image)

When the circular locator is combined with a flat for alignment, it does not even have to be cylindrical. If it is made spherical (see Fig. 6-19a) it still centers the part. A sphere has one and only one diameter and no "diagonals," and is jam-free at all angles. It is expensive to machine with good accuracy, and the spherical locator is therefore not a very practical solution, but it points the way to other solutions.

Any circular locator of a shape contained inside the sphere will locate jam-free. Such a locator, consisting of two opposed conical surfaces joined by a narrow cylindrical band, is shown in Fig. 6-19b. This is a solution with practical applications.

An entirely different type of modification of a cylindrical locator is shown in Fig. 6-20. Three flats are machined on the cylinder, leaving three circular lands 120 degrees apart. To provide sufficient bearing area, the width of each land is taken as 30 degrees. This cut cylinder is now used as an internal locator and mated with an external part which is assumed to be longer than the locator. With the same letter symbols as in Fig. 6-16, the outer cylinder, shown at the left, has diameter $W$, and the inner circle through the three lands (shown at the right) has diameter $W - C$, where $C$ is the diametral clearance. In the concentric position there is a radial clearance of $\frac{C}{2}$ on each land. This is also the distance $AA_1$ and, therefore, the vertical clearance at $A$. The

![Fig. 6-19. Jam-free noncylindrical circular locators.](image)

![Fig. 6-20. A cylindrical locator with triangular relief to minimize jamming.](image)
radii to $E$ and $F$ are drawn at 45 degrees with the horizontal. With the parts still in the concentric position, the *vertical* clearance at $E$ and $F$ is $1.4142\frac{C}{2}$, making the *total* effective clearance for vertical motion:

$$\frac{C}{2} + 1.4142\frac{C}{2} = 1.2071\, C$$

To jam, first move the part a distance $\frac{C}{2}$ up until $A_1$ falls on $A_2$, and there is contact with the locator along the generatrix through $A$. The outer circle, which is the contour of the bore in the part, is projected as the circle through $A_2E_2F_2$. Then tilt the part around a horizontal axis through $A_2$, perpendicular to the axis of the locator, and located at the forward end of the part, with the rear end of the part moving down. Continue tilting until points $E_2$ and $F_2$, located further back in the bore, come in contact with points on the rear end of the locator, projected in points $E$ and $F$. This is the position where jamming may begin. In this position, the old dimension $W$ is replaced by $0.8536\, W$ (see left part of the figure) so that the critical length $L_2$ is now:

$$L_2 = 0.8536\, \mu W$$

The triangular shape has reduced the critical length for jamming by approximately 15 percent but has, at the same time, increased the effective clearance by approximately 20 percent and reduced the locating accuracy of the locator by the same amount.

Locators for parts with more than one significant diameter must not overdefine the part. An exaggerated bad example is shown in Fig. 6-21a. It is four times overdefined. The design of the locator can be improved in many different ways; two correct designs are shown in diagrams b and c. Locators with more than one significant diameter must also be so designed that only one diameter locates at a time. The locator shown in Fig. 6-22a is wrong in that two diameters are required to catch simultaneously. By increasing the length of that part which has the smallest diameter, the small diameter will enter first and help in guiding the large diameter, as shown in Fig. 6-22b.
Radial Locators

Radial locators are those that act on a radius in the part to prevent rotation around a fixed center. Instances where a “radius” is a physical feature of the workpiece have been discussed previously. There are many cases, however, where the configuration of the workpiece does not provide any opportunity for radial locating, and other means must be found for this purpose. Such means fall into three categories: keys and keyseats, dual cylinder locating, and indexing fixtures. Any radial locator has a certain tolerance and therefore involves the possibility of an angular error. With tolerance $T$ and radius $R$ (see Fig. 6-23) the angular error is:

$$\theta = \frac{T}{R} \text{ radians}$$

Since $T$ may be a constant, or at least a quantity with a fixed lower limit, it follows that radial locators should be placed on the largest possible radii for the best angular accuracy.

![Fig. 6-23. The radius sensitivity of a radial locator.](image)

Keys and Keyseats

The key with its keyseat is one of the most common elements used in machine design for the express purpose of permanently locating one machine part radially with respect to another. Keys and keyseats are accurately machined and are capable of transmitting large forces. The machining of a keyseat in a part is a fairly expensive operation and keyseats are not put into parts just for the purpose of locating them in fixtures. If, however, the part already has a keyseat, then this keyseat can be utilized for radially locating the part relative to a fixture.

Keys and keyseats are used for the most part, in connection with circular mating surfaces, to prevent rotation. They are also used between parts with flat surfaces to prevent transverse shifting. These two arrangements of keys and keyseats are shown in Fig. 6-24. Each of them may also be utilized for locating the part in a fixture. When used in machinery, a key and its keyseat serve as parts of a permanent assembly and are not exposed to wear. Where keyseats and keys are used as fixture elements, they are continually exposed to wear because parts are inserted and removed all the time, as long as the fixture is in operation. Hence it is recommended that the key, at least, be made of hardened steel and also, if necessary, a hardened insert be provided for the keyseat. Keys and keyseats are usually located on small radii and have a tight fit; when used for radial locating in a fixture, they should be made with a sliding fit and with the closest possible tolerances.

![Fig. 6-24. The key as a radial locator.](image)

Dual Cylinder Locating

Dual cylinder locating uses a flat base and two cylindrical locators in mating holes. This eliminates all six degrees of freedom and provides excellent mechanical stability with an accuracy which depends only on clearances in the holes and tolerances on the hole center distance. The interplay between these tolerances and clearances creates a specific problem for which there exists a specific solution, the diamond pin.

Assume first, a rather special case where the center distances match so closely that their tolerances can be ignored. The locating accuracy then depends entirely on hole clearances which can be minimized by the use of expanding locators. The expanding locator is shown in Fig. 6-25 where $A$ is the bushing, fitting the finished hole in the work. This bushing is split in several different ways, either by having one slot cut entirely through it, and two more slots cut to within a short distance of the outside periphery, or by having several slots cut from the top and from the bottom, alternating, but not cut entirely through

![Fig. 6-25. An expanding locator for minimizing the clearance.](image)
the full length of the bushing. The method of splitting, however, in every case, accomplishes the same object, that of making the bushing capable of expansion so that when the stud B, which is turned to fit the tapered hole in the bushing, is screwed down, the bushing will expand.

It should be noted that the stud actually consists of four different sections, the head; the tapered shank; a short cylinder; and the screw thread. The cylindrical section matches a precision bore in the fixture base and defines the location of the axis of the stud.

In more general cases, these almost ideal conditions do not apply. There are tolerances on two holes, two locators, and two center distances; the tolerances must be adjusted to each other in such a manner that they leave sufficient clearance around each locator for any permissible dimensional condition, and radial locating must be accomplished with prescribed angular accuracy.

To illustrate the problem, consider a part with two holes of diameter \(D\) and center distance \(L \pm T\). Assume zero tolerance on all hole diameters and on the center distance \(L\) in the fixture. As seen from Fig. 6-26, it is necessary to reduce the diameter of the pin at the right from \(D\) to \(D - 2T\) to make it possible for the part to be nested in all cases (with all center distances from \(L - T\) to \(L + T\)). Consequently, a clearance of \(2T\) is introduced between the pin at the right and the hole in the part, resulting in an angular error

\[
\theta = \frac{2T}{L} \text{ radians}
\]

The case is oversimplified by omitting most of the tolerances but the result would only be aggravated by taking all tolerances into proper account.

It is obvious that the problem could be eliminated by elongating the hole in the part (Fig. 6-27). It is also obvious that it is not practical to make elongated precision holes in parts just to fit them into a

![Fig. 6-26. The general case of dual cylindrical locating.](image)

![Fig. 6-27. Hypothetical locating to an elongated hole.](image)
fixture. Any modification to be made must be with respect to the configuration of the pin and it must permit relative motion between the hole and the pin in the direction of the hole's centerline while it retains a close fit between these two members in the direction perpendicular to that centerline.

The Diamond Pin

A solution along these lines is physically possible and technically practical because the fit between the hole and the major dimension on the pin must be a clearance fit to permit easy loading and unloading of the part.

The cross section of the pin is rhombic (hence the name "diamond pin," see Fig. 6-28) with length $A$. It fits into a hole of diameter $D$ with a clearance $C$, so that

$$D = A + C$$

Assume first that the section terminates in sharp points at its upper and lower end (upper part of the figure). Then using the formula for a circular segment in which $\frac{C}{2}$ is the chordal height and $T$ the width of the segment:

$$\left(\frac{T}{2}\right)^2 = \frac{C}{2} \left(\frac{D}{2} - \frac{C}{2}\right) = \frac{CD}{2} - \frac{C^2}{4} \approx \frac{CD}{2}$$

$$T = \sqrt{2CD}$$

Actually the pin does not terminate in points, but has wearing surfaces of width $W$ (lower part of figure) so that $A$ becomes a diameter (the pilot diameter) and

$$W + T = \sqrt{2CD}$$

$C$ is a measure of the angular error and is small; it is selected from the viewpoints of desired locating accuracy and a sufficiently easy sliding fit. $T$ is the total longitudinal tolerance; it includes tolerances on center distances in the part and the fixture and the diameter tolerances and clearances on the hole and the pin at the other end. The width $W$ is theoretically selected from wear considerations.

It follows from the above formula that for a given, or desired, tolerance $T$, the maximum permissible width $W$ increases with hole diameter $D$ and clearance $C$. It also follows that for a given hole clearance $C$ (and corresponding angular error) the permissible width $W$ increases with $D$ and decreases with center distance tolerance $T$.

Various suggestions have appeared in the literature for the width $W$. The recommended value is $1/8$ of $D$, with $1/32$ to $1/64$ inch (0.8 to 0.4 mm) as a lower limit. All this is now history as these pins are available in standardized dimensions with $W = 1/3 A$.

A proposed N.I.J.F.C.M. standard (see Chapter 17) comprises sizes up to 1 inch; individual manufacturer's standards go up to a 3-inch (75 mm) nominal diameter. The pilot diameter is available in two accuracy ranges, $A$ and $B$. Pins can be installed with press fit, press fit with locking screw, and screw thread.

Jig and fixture literature occasionally recommends the use of two diamond pins set perpendicular to each other ("crossed diamond pins," see Fig. 6-29). The pin at $A$ prevents longitudinal motion, the pin at $B$ allows for longitudinal tolerances and prevents
up and down motion at $B$. The justification appears somewhat incomplete since the up and down motion at $A$ is not prevented. To use the crossed diamond pin principle would require one additional locator, for example, an external pin or a button, as indicated by the dotted lines.

A fully legitimate use of a single diamond pin in combination with another locator is shown in Fig. 6-30. Up and down and angular locating (not shown) is done by the fixture base; the diamond pin locates the part lengthwise, while allowing for the tolerance on the hole center distance $a$ above the base.

![Fig. 6-30. One diamond pin used in combination with a flat locating surface.](image)

All dual cylinder locating systems can be designed for easy loading by application of the two common principles, pre-positioning and successive entering (one at a time). An illustration of the use of these two principles is shown in Fig. 6-31. The two pins are shaped for pre-positioning in two different ways; one pin is shown with a long lead and the other is chamfered. The length of the actual locating surface is also different on the two pins. The part enters first on the long pin to the left, and is supported and guided when it subsequently enters the short pin to the right.

![Fig. 6-31. Dual cylindrical locating arranged for prepositioning and successive entering (one at a time).](image)

Typical Applications of Dual Cylinder Locating

Dual cylinder locating is simple, reliable, and inexpensive. If a part, as designed, does not have the two holes that are needed for locating purposes, such holes (sometimes named “tooling holes”) can, in many cases, be drilled and reamed without impairing the function of the part. The same tooling holes can be used for locating the part in several fixtures, one at a time—and even for reconditioning operations at some later time.

The principle is extensively used in mass production such as in the automotive industry. For example, two holes are drilled and reamed in the pan-rail of the cylinder block to closer positional tolerances than required for functional purposes. These holes serve to locate the block for all operations except for machining the transmission-case face on the end, the pan-rail face, and the head faces; these having been machined in earlier operations. The part is then entered on the conveyor in a transfer machine. Movable “shot pins” enter the locating holes in the pan-rail to locate the block at each station of the transfer machine. All major automobile companies in the United States use this system to machine engine blocks. Smaller automotive components mounted on movable fixtures (also called...

![Fig. 6-32. Locating holes (indicated by arrows) in a V-8 cylinder block. Holes are for the shot pins with which the block is located in various machining stations.](image)
“transfer” fixtures or “pallet” fixtures) are machined in transfer machines by moving the fixture with the workpiece from station to station. In many such cases, the part is located in the pallet fixture on locating pins, and the pallet fixture is located at each machining station by means of shot pins. Examples of these techniques are shown in Figs. 6-32 and 6-33.

![Fig. 6-33. Loading a transmission case on the pallet fixture.](image)

Indexing Fixtures

Indexing means to rotate a part to a predetermined angle and secure it in the new position. Very often, but not necessarily always, the angle of rotation is a simple fraction of a full circle and repeated indexing will finally bring the object back to the starting position. Sometimes the word “indexing” is also used for a straight-line motion of a predetermined length, followed by a locking operation—in other words, “indexing in a straight line.” Both types of indexing are used in fixture design; angular indexing is by far the most common.

A primitive and inexpensive, but not very accurate, indexing device consists of: a fixture with a bearing pin that fits into a hole in the part, a number of markings on the periphery of the part, a target mark on the fixture, and a clamping device. One marking at a time is aligned against the target mark and the part is then clamped and machined.

A part with a central bore and a number of holes of equal size located in a circle concentric with the central bore can function as its own indexing device. The fixture has a locator for the bore and a hole with a pin targeted on the hole circle. When the pin is brought to enter a hole in the part, the part is located. This is, essentially, a case of dual cylinder locating. When the pin is withdrawn, the part can be indexed to the next position and again locked with the pin.

Schemes such as these are inexpensive to make, but are slow in operation and not very accurate as correct operation depends fully on the skill and attention of the operator. There is also no provision for compensation for wear.

Most indexing operations are far more demanding with respect to accuracy, fast operation, and foolproofing. Accuracy means two things, accuracy in operation, and sustained accuracy during the entire life of the fixture. Provisions for the satisfaction of all these demands must be built into the fixture. In addition, the fixture must possess rigidity and strength, and have proper locating and clamping devices for receiving and holding the part.

The fundamental component in most indexing fixtures is the indexing table. It performs the following functions: It receives and holds the part by means of locators and clamps. It rotates around an axis with a minimum of play and error. It carries the weight of the part and the load from the machining forces and transmits these forces to the fixture base. Finally, it indexes accurately from position to position in a manner uninfluenced by the natural wear on its moving parts.

An example of an indexing fixture that satisfies all of these requirements is shown in Fig. 6-34. In this indexing mechanism one of the chief points in design is to prevent variations in the spacing due to wear on the mechanism. The fixture is so arranged that wear on the indexing points is automatically compensated for by the construction of the device; therefore, the provision made for its upkeep is ex-
cellent. In addition to this feature, the design is not very expensive and it may be made up at much less cost than many other kinds of indexing devices. The work $A$ is a clutch gear, the clutch portion $B$ of which is to be machined in this setting. As the work has been previously machined all over, it is necessary to work from the finished surfaces.

A steel index bolt $M$ of rectangular section is carefully fitted to the slot in the body of the fixture, and beveled at its inner end $S$ so that it enters the angular slots $S$ and $T$ of the index ring. Clearance is allowed between the end of the bolt and the bottom of these slots so that wear is automatically taken care of. A stud $O$ is screwed into the underside of the index bolt and a stiff coiled spring at $N$ keeps the bolt firmly in position. The pin $U$ is obviously used for drawing back the bolt and indexing the fixture. Points worthy of note in the construction of this fixture are the liner bushing at $E$, the steel locating ring $L$, the automatic method of taking up wear by the angular lock-bolt $M$, and the spring $N$.

With ample bearing dimensions and a hardened steel liner or liners, in the bearing, the fixture can operate year after year with negligible wear because the amount and velocity of the motion are small, and the bearing is practically unloaded during indexing. Should wear ever exceed the permissible limits, it is a simple matter to replace the central stud and the liner. The same applies to the index ring, as it is also a separate item. The part that receives most wear in this, as in any other indexing device, is the index bolt. It retains its beveled shape even when worn and continues to align the index ring properly as it closes in by the force of the spring. In this way, the closing; locating; and locking action are made independent of the operator.

The flat, beveled index bolt as here described, is the most efficient type that exists. Round index bolts are cheaper to make but less accurate in their operation. The axes for the index bolt and index holes require alignment in three directions against only two for the flat bolt. Round bearing surfaces are less resistant to wear than are flat surfaces. When a cylindrical index bolt is worn, it has lost its accuracy. A conical (tapered) index bolt (see Fig. 8-11) can be just as accurate as the flat bolt, as long as it is new; as it wears, however, it forms recesses and loses accuracy.

Only very large indexing fixtures for heavy parts require ball or roller bearings. The bearings used in such cases are the same types as are used in large precision machine tools.

The milling machine dividing head could perhaps, in a sense, be considered an indexing fixture; the same applies to the various types of rotary tables, horizontal; vertical; and tilting. They are all work holders, and so are standard vises, magnetic chucks and faceplates, lathe chucks, and so on. However, they are designed as general purpose tools and they are all commercially available. For these reasons, they shall not be further discussed in this book.
except where they may serve as bases in actual fixtures for special applications. Indexing pins with liners or bushings (see Chapter 17) are standardized and are commercially available.

In the previous examples, the pulling of the index bolt and the rotation of the work were straight manual operations. These operations can, of course, also be performed by various mechanical means. There is another extremely simple device, which may even be called a trick, that can be incorporated into the design of an indexing fixture which permits rapid indexing without the need for installing additional mechanisms. It consists of selecting a rather large included angle for the bevel or taper on the indexing bolt. On a cylindrical indexing bolt, this angle is zero. On most ordinary indexing fixtures the angle is 8 to 12 degrees, so that the bolt is self-locking. If the included angle is made larger than two times the angle of friction, the bolt is no longer self-locking, but can be pushed back and out, if a sufficiently large turning moment is applied to the index table. Such devices in the form of spring stops, ball plungers and detents (see Chapter 17), are also commercially available.

The drill jig shown in Fig. 6-35 was designed for drilling four angular holes in a brass time-fuse cap. (See sectional view of cap at lower part of illustration.) The principle of this jig can easily be applied to other work. The jig consists of a hardened steel locating plate \(A\) mounted on a hardened spindle, which runs in a bushing that is also hardened. A ball bearing \(B\) takes the thrust of the spindle. At the other end of the spindle is an index plate \(C\) in which are cut four 90-degree notches. Keyed to the index plate, and also to the spindle, is a ratchet wheel \(D\), having four teeth. A hand-lever \(E\), which has a bearing and turns around a hub on the index plate, carries a spring pawl \(F\) that engages with the ratchet wheel \(D\). The lever also carries, at the outer ends, two pins \(G\) that project downward, so that when it is pushed back and forth, the pins strike on the body of the jig and prevent carrying the index plate beyond the locking pin \(H\). This locking pin is a hardened steel sliding pin, one end of which is rounded and engages with the notches in the index plate. Back of the pin, and held in place by a headless set-screw \(K\), is a coil spring \(J\), which holds the locking pin against the index plate. The tension of this spring is just enough to hold the work from turning while being drilled, but not enough to prevent its being readily indexed by a quick pull on the indexing lever.

The work is held in position against the locating plate \(A\) by the plunger \(L\), which rests on a single, 1/2-inch, hardened steel ball that acts as a bearing while the work is being indexed. Plunger \(L\) is carried in a second plunger \(M\), which is held up by a powerful coil spring \(N\). The outer plunger \(M\) is operated by a foot-treadle connected to the lever \(O\). In operation, the foot-treadle is depressed and a piece of work is placed between the plunger \(L\) and the locating plate \(A\). When the treadle is released, the work is held by the tension of the spring \(N\) while the indexing is done by lever \(E\). The locating plate \(A\) has slots milled in it with a radius cutter of the same radius as the drill to be used. This feature, in connection with the lip on the work, answers the same purpose as a drill bushing; no other means of guiding the drill being necessary. The production record of this jig was about 4000 caps per day.

**Stability Problems**

Some indexing fixtures present stability problems. Small or flat parts with a short dimension in the direction of their axis are easily handled on indexing fixtures with a horizontal table. Heavy parts of greater axial length cannot be fixtured with an overhang, but require the equivalent of an outboard bearing. There are cases where an actual outboard bearing can be added to the fixture, but usually, this is an impractical solution and it is necessary to use a fixture provided with two trunnions supported in a separate cradle with two bearings, as shown in the following example.
It is necessary to drill quite a number of holes in the casting shown in place in the jig illustrated in Fig 6-36; these holes are located on different sides and at various angles to one another. For this reason, an indexing jig is employed. This illustration shows the cover A of the jig removed in order to illustrate more clearly the position of the casting, which is located in the jig by its trunnions. The main body of the jig is also supported by heavy trunnions at each end, and the large disks B and C enable it to be held in different positions. These disks contain holes which are engaged by suitable indexing plungers D, at each end of the fixture.

Adjustable Locators

The term “adjustable locators” is occasionally used with several different meanings, and some clarification is therefore required. In Chapter 3 the difference between “locators” and “supports” was explained. “Locators” are the elements that are necessary and sufficient for full geometrical definition of the locating of the part; they may or may not be sufficient, however, for the stable mechanical support against all the forces acting upon the part when it is being clamped and machined. Any additional elements that may be required for this purpose are termed “supports.”

One basic function of the locators can be described as the elimination of the six degrees of freedom. In mechanical language, this means that the locators bring the part into a statically determinate position with respect to the fixture, and any additional support makes the position statically indeterminate. Any such support is said to be “redundant.”

A statically indeterminate position or system is not necessarily bad. The redundant supports do no harm if they are compatible with the part; this means, if they are fitted so closely that they maintain contact with the part without exerting any force upon it. If the redundant supports are incompatible with the statically determinate system, there are then three possibilities:

1. They fail to contact the part; in this case, they are ineffective and could be dispensed with.
2. They lift the part off one or several of the locators; in this case, they assume or usurp the locators’ function.
3. They exert significant forces upon the part, and in so doing, they impose a deformation (deflection, distortion) within the part and loads (reactions) on the locators. They “spring” the part (if they do not bend or break it!).

The various possibilities (compatibility and the three forms of incompatibility) are shown in Fig. 6-37. The part is supported as a beam on two end supports and is, in this condition, statically determinate. The addition of a redundant intermediate support makes the part statically indeterminate. Clearly, each of the three alternative forms of incompatibility is unacceptable, and redundant supports are therefore made adjustable. The various designs are described in Chapter 12.

The adjustable locators to be described in this section are the basic locators conforming to the 3-2-1 principle or its equivalent. Adjustable locators are used for the following purposes: To accommodate raw parts that exceed normal or previously established tolerances, to adjust for dimensional changes within the fixture from wear, abuse, or neglect, and to use one fixture for more than one size of the part. Examples of devices for these purposes are shown later in Figs. 6-43 and 6-44.

It is fundamental for the use of fixtures that the raw parts are dimensionally uniform within the prescribed tolerances for which the fixture is designed. A part that exceeds tolerances should be intercepted by inspection so that it does not reach the fixture. If it does, however, it would be rejected by the operator as soon as he finds that it does not fit properly in the fixture. An adjustable locator should, as a rule, not be operated just to save an accidental or isolated misfit.

Dimensional changes within a raw part may occur from time to time. Common causes are change of supply source, variations (intentional or unintentional) in foundry practice, overhaul or replacement of forging dies or other tools, etc. If the change
exceeds the fixture tolerances and appears to be permanent, it is necessary to readjust the fixture. This is a toolroom operation and is followed by an inspection similar to the inspection of a new fixture. The operator should not reset locators or other vital adjustable parts in a fixture. Adjustable locators are purposely so designed that they do not invite, encourage, or facilitate adjustments "on the shop floor." In contrast, adjustable supports are designed for convenient and fast operation, preferably without the use of tools.

Adjustable Locating Points

The most common form of adjustable locating points is the set-screw provided with a locknut, as shown in Fig. 6-38. The screw A, is a standard squarehead set-screw, or, in some cases, a headless screw—with a slot for a screw driver; this screw passes through a lug on the jig, or jig wall B, itself, and is held stationary by a locknut C tightened up against the wall of the jig. Either end of this screw may be used as a locating point, and the locknut may be placed on either side. By using a squarehead screw, adjustment is very easily accomplished, but unless the operator is familiar with the intentions of the designer of the jig, locating points of this kind are sometimes mistaken for binding or clamping devices, and the set-screws are inadvertent-ly tightened up and loosened, to hold and release the work, when the intention is that these screws, when once adjusted, should remain fixed. It is not even possible to depend upon the locknut stopping the operator from using the screw as a binding screw. A headless screw, therefore, is preferable, as it is less apt to be tampered with.

A different form for the adjustable locator of the screw type is shown in Fig. 6-39. 5 The head is hexagonal and the top of the screw is rounded (crowned) so that it offers a regular bearing area even when the screw axis is slightly out of alignment due to clearance in the screw thread. The bearing area in all screw type locators is hardened. Screw locators are much longer than fixed locators. They can be used as side and end locators without difficulty, but not always as base locators because of the limited vertical design space in the bottom of a fixture.

Adjustable base locators can be designed on the wedge principle. The action of a wedge is mechanically equivalent to the action of a screw, but the wedge has its major dimension perpendicular to its direction of action. The wedge is, therefore, a suitable device for adjustments in a narrow space.

An example of a wedge-operated adjustable base stop is shown in Fig. 6-40. The base stop C is raised and lowered by the sliding motion of wedge A. The

5E. Thaulow, Maskinarbejde (Copenhagen: G.E.C. Gad's Forlag, 1930) vol. II.
wedge is provided with a handle \( B \), so attached that it can easily be operated. It is held in place by two shoulder screws that are inserted through two elongated slots milled in the wedge; these screws are tightened after the stop has been brought up to position. One disadvantage in using this type of stop is that owing to the vibration of the machine while in operation, the wedge is prone to slip back, causing the stop \( C \) to drop down. Various improvements are possible, however, and will be described in Chapter 12, in connection with supporting elements of a similar type.

The "sliding point" is another adjustable locator which is used extensively in fixtures. It requires considerable design length and must also be accessible from above or from the side. Its principal application, therefore, is for side and end stops. One design is shown in Fig. 6-41, where \( A \) represents the work to be located; \( B \) the sliding point itself; and \( C \) the set-screw, binding it in place when adjusted. The sliding point \( B \) fits a hole in the jig wall and is provided with a milled flat, slightly tapered as shown, to prevent its sliding back under the pressure of the work or the tool operating upon the work. This sliding point design is frequently used, but it is not as efficient as the one illustrated in Fig. 6-42. In this design the sliding point \( A \) consists of a split cylindrical piece, with a hole drilled through it, as illustrated in the diagram, and a wedge or shoe \( B \) tapered on the end to fit the sides of the groove or split in the sliding point itself. This wedge \( B \) is forced in by a set-screw \( C \), for the purpose of binding the sliding point in place. Evidently, when the screw and wedge are forced in, the sliding point is expanded, and the friction against the jig wall \( D \) is so great that it can withstand a very heavy pressure without moving. Pin \( E \) prevents the sliding point from slipping through the hole and into the jig, when loosened, and also makes it more convenient to get hold of. In Table 6-1 are given the dimensions most commonly used for sliding points and binding shoes and wedges.

Regardless of differences in design, all adjustable locators have two important features in common; they require tools (wrenches, screw drivers, etc.) for resetting and adjustment, and they can be locked hard.

Adjustment for Wear

The adjustable point locators as described in the previous section are essentially designed for adjustment to wide dimensional variations on raw parts with wide tolerances, and locator wear is not a significant factor.

Adjustment for wear as well as for locator displacement from other causes such as overload, carelessness, neglect, misuse, and accidental damage, is also required on precision locators to be used on parts with close tolerances. Adjustable locators of the screw and wedge type can be designed with a
fine adjustment ratio (fine pitch screw threads) for this purpose and used in drill jigs and milling fixtures. Lathe fixtures present special problems as they do not always provide the space required for screw and wedge locators. They are exposed to wear and also risk accidental damage when mounted on or removed from the lathe spindle, with a resultant misalignment of the fixture axis. Adjustment for this type of error requires certain devices for recentering of the locator section of the fixture.

One fixture for this purpose, which may also be adjusted to handle several sizes of work $A$, is shown in Fig. 6-43. It is essential to be able to true this fixture when it is mounted on the spindle nose since absolute concentricity is required between the machined surfaces. This is accomplished by four adjusting screws $D$ and a wedge pin assembly, which will be described later.

The basic fixture components are the nosepiece $B$, which can be designed to fit any standard spindle nose in the conventional manner, and the fixture body $C$. A hardened steel locating ring $H$ is mounted on the fixture body with a sliding fit and clamped in place by the socket head screws $J$. The workpiece $A$ is located and held inside this locating ring by three strap clamps $K$. Rings of several different sizes are made, which can be mounted on the fixture body to accommodate different sizes of workpieces. Whenever this fixture is mounted on the spindle nose of the lathe, the concentricity of the locating ring $H$ should be checked with respect to the rotation of the spindle, using a dial test indicator capable of reading to .0001 inch (0.025 mm). If the locating ring does not run true, it can be adjusted by means of the four adjusting screws $D$ in a manner similar to adjusting the jaws of a four-jaw chuck. When adjusting the fixture, only two opposing screws should be loosened at any one time while the other two remain tightened. In this way the fixture body will remain seated against the nosepiece while the adjustment is made. The fixture is ready to be used when the locating ring $H$ is true within .0002 to .0003 inch (0.005 to 0.008 mm) with all of the adjusting screws $D$ tightened.
Design by Karl H. Molnrecht

Fig. 6-43. A lathe fixture with provision for recentering.
Fig. 6.44. A lathe fixture with provision for recentering and with equalizing clamps.
Design by Karl H. Molterd.
Tightening the adjusting screws \( D \) serves to clamp the fixture body \( C \) securely to the nosepiece \( B \), and to locate the fixture accurately in the axial direction by forcing it to register against a locating surface on the face of the nosepiece. This is accomplished by the action of the wedge pin assembly, consisting of a wedge pin \( E \), a wedge-pin seat \( F \), and a wedge-pin seat container \( G \). The four wedge pins fit closely in the holes below the adjusting screws \( D \). These holes should be tapped only to a depth that will allow sufficient room for the adjusting screws to operate. The tap drill hole should be reamed to size and a hard reamer should be used to remove any burrs in these holes resulting from the tapping operation. The round wedge-pin seat containers \( G \) are made of hardened steel and are press fit into the nosepiece \( B \). An eccentric hole is drilled in the seat of these containers, which must be located in the forward position, as shown in Fig. 6-43, when the containers are pressed into the nosepiece. A slight inaccuracy in the position of the eccentric hole is not harmful because it is made much larger than the pin that is placed in this hole. This pin is pressed into the face of the wedge-pin seat \( F \) and it serves to locate the wedge-pin seat so that the bevel ground on the opposite face will be oriented approximately in the right direction. The wedge-pin seat \( F \) is a very loose fit in the wedge-pin seat container \( G \) to provide it with a limited freedom of movement. The bevel angle on the wedge pin and on the wedge-pin seat as well, should be 15 to 22 degrees. Thus, when the clamp screws are tightened, the wedges, or bevels will cause a reaction of the clamping force, so that it will have both a radial component and an axial component. The radial component will hold the fixture body in the correct radial location and the axial component will hold it against the nosepiece, thereby providing axial location.

The workpiece \( A \) is machined with an 80-degree diamond shaped insert \( L \) held in a disposable insert toolholder. The toolholder is held in an adapter that is mounted on the face of a turret on an NC lathe. It could also be held in a conventional manner on an engine lathe, or on a turret lathe having a cross-sliding saddle. The cutting tool is used to machine the faces and the major recess.

Of compact design and built close to the spindle nose, this is an example of a fixture designed for standard work that requires accurate machining and where the production lots are small. Although it is heavy, there is so little overhang that its weight is of small importance.

Another fixture incorporating the adjusting screw and wedge-pin principle is shown in Fig. 6-44. This fixture illustrates a different and more sophisticated clamping device, which is an embodiment of the floating principle. The workpiece is a bevel gear \( A \) and the fixture consists of two principal parts, the spindle nosepiece \( B \) and the fixture body \( C \).

The workpiece is mounted on a hardened steel locating ring \( H \), which is pressed onto the fixture body. This ring has a clearance groove to collect small chips and dirt, enabling the workpiece to register against the locating face of the fixture body.

When the fixture is mounted in the lathe, the locating ring \( H \) must be trued with respect to the rotation of the spindle. This is done, as before, by indicating the locating ring with a "tenth" dial test indicator and adjusting the adjusting screws \( D \) until the locating ring is true within .0002 to .0003 inch (0.005 to 0.008 mm). The wedge clamp assembly consists of the following parts: \( E \) wedge pin; \( F \) wedge-pin seat; and, \( G \) wedge-pin seat container. This assembly will cause the fixture body \( C \) to be held firmly against the nosepiece \( B \) as described for the fixture shown in Fig. 6-43.

The method of clamping consists of the use of three strap clamps \( L \), a clamp operating screw \( J \), and a floating collar \( K \). The three clamps are placed 120 degrees apart and have slightly oversize holes through which the clamp retaining screws \( M \) pass. These screws have a ball surface on the underside of the collar corresponding to a similar depression in the clamps themselves. A bronze or steel bushing \( I \) is pressed into the fixture body \( C \), and is threaded with a coarse-pitch thread which corresponds to that on the clamp operating screw \( J \). After the clamps \( L \) have been swung into place on the ring gear, a few turns of the clamp operating screw tightens all three of the clamps against the ring gear \( A \) through the action of the spherical floating collar \( K \), which bears against the inner sides of the clamps.

Where high production is required, a machine equipped with a rotating pneumatic cylinder is used. In this case the threaded bushing \( I \) would not be used. The screw \( J \) would be threaded directly into an operating rod that extends through the inside of the lathe spindle, which is then attached to the pneumatic cylinder. The pneumatic cylinder actuates the operating rod which moves the screw \( J \) forward to clamp the workpiece. However, on lathes that are not equipped with a pneumatic cylinder, the arrangement shown in Fig. 6-44 is very satisfactory.
CHAPTER 7

Loading and Unloading

Entering the Part

The complete process of fixturing is comprised of loading, machining, and unloading; the loading operation consists of entering and locating the part and clamping it; the unloading, of releasing and removing the part. Each phase has its problems. Entering involves manual handling and requires space. Convenient manual handling depends on weight and balance. Light parts are handled by the operator’s two fingers or one hand; heavier parts require two hands or, in more extreme cases, a hoist, crane, or conveyor. Well-balanced parts require lifting and lowering only; an unbalanced part, having its center of gravity at some distance from its midpoint, also requires a steadying effort which makes it increasingly difficult to keep the part level during lifting and lowering.

Space must then be provided inside the fixture for the part, fingers, a hand, possibly two hands (and knuckles!), or two hands and arms. For heavy parts there must also be clearance from the machine tool to allow the operator to lean over the fixture, or to admit the load cable from the hoist or crane. Although these factors may appear trivial, they are quite serious and it is a common experience that space always looks larger on a drawing than in reality.

Locating the Part

Locating means bringing the part into positive and, correct contact with the locating points or surfaces. Chips and dirt on a locating point prevent direct contact at that point, but accumulations in other places in the fixture may well cause such misplacements or misalignments that the part cannot be properly located. Other causes of insufficient contact are burrs, part irregularities beyond prescribed tolerances, jamming, and friction. These adverse factors can be directly and indirectly controlled by the fixture designer who should provide means for chip cleaning and for visibility at the locating point.

Apart from these considerations, there is no further problem encountered in locating when the conditions are equivalent to those shown in Figs. 3-1d and 4-1. Locating is done in three consecutive steps. First, the part is set on the base; second, it is moved to contact with the side stops; third, it is moved to contact with the end stop. Next, the clamping pressures are applied. A basic and characteristic feature in these simple examples is that each locating step is not interfered by, and does not interfere with, any other locating step. One result, thereof, is that the individual phases in locating are not sensitive to the direction of approach. Assume the part is tilted while it is lowered to the base. It then contacts first one of the three points (or one corner), levels off, contacts the second point (or corner), levels off on the axis through these two points (or the edge between the two corners), and comes to rest on all three points (or on the bottom surface). If it is still misaligned with respect to the side stops, it contacts one side stop first, then aligns itself to contact with the second side stop.

These observations lead to the basic and very general rule that locating should be done on only one surface at a time, where possible.

Correct and Incorrect Loading

Stressing that the part be brought into correct contact with the locating surfaces may seem unnecessary, but it is not. Any part that has been machined when located in an incorrect position is lost, and so is the labor that has been expended. Design steps taken to prevent incorrect loading are
termed "foolproofing," or "mistake-proofing" the fixture.

Symmetry Considerations

Correct and incorrect loading are associated with symmetry and asymmetry in the part configuration. With reference to Fig. 7-1, planes of symmetry (or asymmetry) are denoted AA, BB, and CC; the corresponding perpendicular axes (sometimes, but not necessarily always, axes of rotational symmetry) are denoted X, Y, and Z. A completely symmetrical part; that is, a part containing three planes of symmetry, can be loaded in a fixture in four different orientations. From an initial position it can be turned 180 degrees around the three axes X, Y, and Z, respectively. In other words, it can be turned end-for-end and upside down, and there are no orientations other than these four. Apart from any surface markings there is no discernible difference between the four positions, and any machined configuration applied to the part will produce the same end result. Every position is a correct position and incorrect machining is simply not possible in this case, regardless of how the part was loaded.

A completely asymmetrical part, fully nested, will normally be able to enter the fixture in one position only, the correct one; and is, therefore, always correctly machined. The possible exceptions are if the configuration of the nesting points and surfaces contains some degree of symmetry. All other cases lie somewhere between these two extremes. Two important examples will now be analyzed.

In Fig. 7-2 a part having two planes of symmetry, AA and CC is shown. Also shown are the three principal axes X, Y, and Z, in the initial position. Two sets of surfaces, namely, the two pads on the top side and the right-hand face, are to be machined and a hole will also be drilled, as shown in view b. To assist in identifying the position and orientation of the part, one surface has been labeled A and it is called the TOP SIDE SURFACE; another surface has been labeled B and is called the FRONT SIDE SURFACE. This part is shown in four different positions in Figs. 7-2 b, c, d, and e. If no corrective action is taken, the part may be assumed to enter the fixture in any of the four positions. The need for corrective action is evident from an examination of the four illustrations. In Fig. 7-2 b, the part is in the initial position; the intended correct position for entering the fixture. The surfaces machined are the correct surfaces, and the hole is in the correct position.

The part, in Fig. 7-2 c, is rotated 180 degrees around the Y axis. Notice that the final configuration of the part will not change when it is machined in this position; this is the result of symmetry on the AA and CC planes.

In the position shown in Fig. 7-2 d, the part has been rotated 180 degrees about the X axis from its initial position. When the hole is drilled and the two surfaces are machined with the part in this position, the relationship of the hole and the machined surfaces on the pad will be incorrect. This is shown in the lower illustration, which shows the front side surface. When this view is compared to the front side surface in Fig. 7-2 b, it is readily seen that the machined pads are on the wrong side. The part in Fig. 7-2 e has been rotated from its initial position 180 degrees around the Z axis. Again, the part configuration will be machined incorrectly, which can be seen by comparing the front side views in Figs. 7-2 b and 7-2 e. The machined pads are again on the wrong side.

An examination of the figures shows, that out of the four possible part positions within the fixture, there are only two positions (namely views b and c) in which the surfaces can be machined to their correct relative positions. This observation serves to illustrate a fundamental rule, not generally recognized. There exists a class of operations that is permissible, even with the part in a prohibited position. The criteria for this class of operations are that they produce surfaces which consist entirely
Fig. 7-2. A part with two planes of symmetry.
of straight line generatrices perpendicular to the nonsymmetry BB plane (satisfied for the end surface, the two pad surfaces, and the hole), and that they maintain symmetry with respect to one of the planes of symmetry, either the AA plane (satisfied for the two pad surfaces), or the CC plane (satisfied for the end surface and the hole).

A part with only one plane of symmetry radically changes its position configuration relative to the fixture, with each of the three possible 180 degree rotations, and, normally, these three positions are prohibited. There also exists, however, a class of machining operations that will produce correct surfaces with two different positions of the part in its fixture. The part shown in Fig. 7-3 has only one plane of symmetry, plane CC. It is to be drilled through and machined on the two pads. With the part in its initial position, the machined configuration, which is the correct one, is shown in Fig. 7-3 b. When rotated around axis X, the part is still correctly machined, as shown in the view of the front side surface in Fig. 7-3 c. However, if the part is rotated around axis Y (see Fig. 7-3 d) or around axis Z (see Fig. 7-3 e), the hole is drilled in the wrong place. The criteria for the class of operations which can produce a correctly machined part from more than one part position within the fixture are that they produce surfaces which consist of straight line generatrices perpendicular to one of the nonsymmetry planes, either the AA plane (satisfied for the pads), or the BB plane (satisfied for the pads and the hole), and that they maintain symmetry with respect to the one and only plane of symmetry CC (satisfied for the pads and the hole).

Foolproofing

Despite the literally endless variety of possible asymmetries found in part configurations, a systematic classification and the formulation of some widely applicable rules can be provided. With such rules and some practice, the fixture designer is able to quickly spot and utilize existing possibilities for foolproofing and to create them where required, by additional modifications in the fixture design.

Foolproofing is required for parts with at least one asymmetry. A completely symmetrical part needs no foolproofing. No matter how it is inserted into the fixture, it presents to the eye, and to the cutting tool, identically the same configuration and, no matter where the machining cuts are taken, relative to the planes or axes of symmetry, the machined parts come out with identical shapes. Foolproofing is needed when the part, in addition to one, two, or three planes of symmetry, also presents one or more irregularities. Many workpieces, particularly components of machinery, fall into this category. The body of the part may be essentially symmetrical and of substantial dimensions, which makes it suitable for locating and clamping. The irregularities can be convex (going out) or concave (going in). They disturb the symmetry, and it is these irregularities that are utilized for the purpose of foolproofing. Before going into the details of the subject, a few simplified, but typical, examples will be shown for orientation.

A two-times symmetrical part with a projecting lug on one end, as shown in Fig. 7-4 a, can be confined within a box with a cut in the end wall for the lug. If the cut in the end wall cannot be tolerated, the vacant space in the box adjacent to the lug can be taken up by a blocking, as shown in Fig. 7-4 b. With two end lugs, as shown in Fig. 7-4 c, the blocking can be located in the space between the lugs. A cylindrical hollow boss with a plain arm can be centered on a mandrel with the arm located in a straight slot in the fixture wall. If the arm is formed as a bracket with a T-section, as shown in Fig. 7-5, the contour of the slot must match the contour of the bracket. A straight slot of constant width would not prevent the part from entering upside down. A part with an asymmetrically located, downwardly open cavity, as shown in Fig. 7-6, is very simple to foolproof. All that is required is a blocking on the fixture base, while an upwardly open cavity would require a blocking which extends downward from above, and therefore must be carried by a removable part of the fixture.

Locating elements for foolproofing can be a part of the fixture base, the cover, or the clamps. In special cases, it may be necessary to use separate movable parts for this purpose. Such parts may simultaneously serve as centralizers (see Chapter 9). When possible, they should be built right into the fixture base, so that they take effect immediately as the part is entering the fixture. If placed in a movable part, they cannot detect an incorrect loading until after the part has been introduced, and additional time is then required to unload, correct, and reload.

In most cases the foolproofing requires simple means and is done at little additional cost. It very often happens that the major body of the part, including its natural locating surfaces, contains one or several degrees of symmetry, and the asymmetries are confined to small areas at isolated locations. In such cases the locating is performed according to general rules supplemented by the necessary fool-
Fig. 7-3. A part with one plane of symmetry.
Fig. 7-4. Foolproofing a part with projecting lugs of simple shape.

Fig. 7-5. Foolproofing a part with a contoured bracket.

Fig. 7-6. Foolproofing a part with a downward open cavity.

Foolproofing elements. Localized asymmetries are convex or concave. When convex, they take the form of projections, such as arms, bosses, brackets, ribs, etc.; when concave, they may be recesses, depressions, notches, slots, holes, perforations, or cavities of other shapes.
Asymmetries in the form of cavities are simpler to handle than projections because it takes only one block or pin to mate a cavity, but two to make a fork. A simple and very common case is where the cavities are a group of holes in a flat surface. The part is effectively located but not necessarily foolproofed by two pins mating with two holes. If the holes are of different diameters, the locating is foolproof. If not, it may be possible to provide one extra hole of a different diameter for the double purpose of locating and foolproofing, or the designer may find it possible to make an existing hole oversize, for the same purpose. Asymmetries, whether projections or cavities, also may be located essentially on a horizontal contour, in a vertical orientation, or tangentially; that means an asymmetrical or otherwise irregular spacing of the projections or cavities on a circle.

The example shown in Fig. 7-7 is a cylinder block with asymmetries (the flange contour and the hole pattern) in a horizontal plane. The locating elements between the drill jig A and the surface of the upper flange, and the side stops and clamping screws on the two long sides, have symmetry. But if no further steps were taken the jig could nest on the block in two opposite positions, and one of these must be prohibited. This is accomplished simply by carrying the end stop B, sufficiently close to the end lug C, so that it cannot nest on the triangular extension D, on the opposite end.

![Fig. 7-7. Foolproofing an asymmetrical part by means of an asymmetrically located block.](image)

Parts with vertical asymmetries are very common and are also frequently associated with surfaces of revolution. Perhaps the most dangerous configuration is one with an asymmetrical arm projecting from a boss with a cylindrical bore, since the bore can slip on a cylindrical locator in two opposite positions. A simple and typical case of this category is shown in Fig. 7-8, where it is required to drill and countersink the hole in the arm. Even without the countersink, a skilled mechanic would not slide the part on in the wrong position and attempt to drill it without support. In the case shown, the part can swing into position under the bracket with the drill bushing only when it is first correctly located on the locating pin. An analogous example is shown in Fig. 7-9, where a hole in the arm can be located on

![Fig. 7-8. Foolproofing by means of the bracket configuration.](image)

![Fig. 7-9. Foolproofing by means of a step on the cylindrical locator.](image)
the pin without interference with the drill bushing only if the part is held in the correct position.

The part shown in Fig. 7-10 has its boss centered between a lower and an upper conical locator. It can enter the lower locator only when the arm coincides with the notch provided in the cylindrical wall, and in no other position.

Parts with radial locating and tangential asymmetries are shown in Figs. 7-11 and 7-12. In each case the part is centered on its axis, clamped with a knob $K'$, and radially located by a pin $X$, in a slot. In Fig. 7-11, a hole is to be drilled opposite flat $A$. Foolproofing is done by the pin $B$, set tangentially to flat $A$. For radial locating, this arrangement would be the poorest possible, as it permits a small but finite tangential motion. However, the actual locating is done by the pin $X$, which can be made to fit in a machined slot with close tolerances.

In Fig. 7-12 the hole is to be drilled opposite a projecting bracket $A$. The part is clamped by means of a hinged clamp $C$, with a knob $K$. The clamp has two lugs; a slot in the lower lug forms it into a fork $F$, matching the bracket $A$ when it is in its correct position. The clamp, including the lugs, blocks all other positions of the bracket.

Examples with asymmetrical cavities are shown in Fig. 7-13. A blind hole is to be drilled on the same side as the cavity or on the opposite side. In each case the foolproofing is done by means of a small block that matches but does not have to fit closely into the cavity. With the hole and the cavity on the same side (Fig. 7-13a), the block is mounted on the underside of the hinged leaf of the drill jig; with the hole and the cavity on opposite sides (Fig. 7-13b), the block is mounted on the base of the jig. In each case, incorrect loading is prohibited.
With vertical loading and unloading, the lifting is usually a little more difficult than the lowering. It is a matter of getting the proper grip or hold on the part, which applies to very heavy parts as well as to very small parts. Once this is understood, the operator must always make sure that there is enough clearance inside the fixture for getting the proper grip on the part. Clearances must also be checked against the possibility that they have caught chips that can bind between part and fixture, when the part is to be removed.

**Burred Parts**

As explained later, in Chapter 8, two types of burrs, the major burr and the minor burr, are formed in a machining operation on ductile materials. Burrs along external surfaces (see Fig. 8-2a and b) present no problem as they are out in the open; nor do burrs on drilled holes (see Fig. 8-2c) if the part is lifted out. However, burrs from any operation, particularly from drilling, do present a problem if they are formed on a surface which has to slide with a narrow fit in or on a mating locating surface. The most common case is a part with a cylindrical surface, to be drilled perpendicular to the cylinder axis. Drill jigs for this operation are shown in Fig. 7-14.

The part shown in diagram a is a short shaft, located in a cylindrical bore in the drill jig. The hole is a blind hole and only a minor burr is formed adjacent to the drill bushing. The body of the drill jig has a keyseat which provides clearance for the burr to form and allows it to slide out. The body of the drill jig can be completely machined, including the cutting of the keyseat, before the end plug is welded on.

The part shown in diagram b is a bushing with a hole through on a diameter. It is located on a plug. The plug has a hole to clear the drill, and is machined with upper and lower flat surfaces which provide clearance for the major burr on the upper hole in the part, and the minor burr on the lower hole.

The part shown in diagram c is a stepped shaft with one through hole. The drill jig is machined from the solid. Clearance for the minor and major burrs is provided by two keyseats inside the bore in the jig. Two holes A are drilled to provide end clearance for the cutting of the keyseats. Hole B is used to clear the drill.

Burr problems are also encountered in milling operations, particularly those involving slotting operations. An example is shown in Fig. 7-15. The
operation is to cut a slot lengthwise from one end of part A, a cylinder. The part is located on a mandrel B and is clamped by means of a large washer C. The mandrel has an outboard support D, which is removable for loading and unloading. The milling operation is climb milling (down milling) so that the work is held against the mandrel. A slot E is machined in both mandrel and washer to provide clearance for the cutter. At the same time it provides clearance for the major burr which is being formed on the inside of the part.

Ejectors

Despite all the fixture designer's care and foresight, it is not always possible to ensure free and easy removal of the machined part and a mechanical ejector then becomes a necessity. A warped and binding part is not the only justification for the use of an ejector; in fact, the ejector should enjoy much wider publicity in the literature and much wider use in industry than is presently the case. The ejector is not a convenience but an economic asset. It reduces the time for removal of the part. As an example, it took 0.20 minute to grip a part, lift it out of the fixture and place it in the tote pan. With an ejector used for lifting, the same operation was reduced to 0.08 minute. This time saving may appear small; however, for large-volume production with short duration operation cycles, every saving represents a significant percentage of the total. By eliminating the need for finger and hand space for gripping the part, the ejector permits a reduction of the overall dimensions—thus the cost—of the fixture. The ejector pins can be located to suit the part configuration so that side-heavy parts are lifted free and clear, in perfect balance. In this way, ejectors eliminate inconvenient and awkward hand manipulation and reduce operator fatigue.

There are several reasons why parts may bind in the fixture. It may be from distortion during machining, as previously discussed, or it may be because the part has a locational interference fit (class LN 2 or 3) against the locator to ensure close tolerances. The part is tapped down in place when loaded into the fixture; removal requires the application of some force, but is easily done with ejectors.

For maximum time saving, the operation of the ejector can be automatic and coupled with the release of the clamp. The combined mechanism can be powered hydraulically or by compressed air.

**Ejector Details**

Ejectors are inexpensive since there is no need for close tolerances; most machine work is by turning
and (for hardened parts) cylindrical grinding; and holes are finished by reaming. Details that come in contact with the part are usually made from hardened tool steel or from a case hardening steel. If the part surface must be protected from scratches or other markings, the ejectors may have contact heads made of copper, brass, or aluminum.

It is a prerequisite for the use of ejectors that the locators must be designed jam-free, as previously explained in Chapter 6. Since ejectors are moving parts, their bearing surfaces must be protected against dirt and chip fragments. In most cases ejectors are shielded against chips by the part itself; if necessary, shields or seals also can be installed, as described at the end of Chapter 8.

The basic elements in an ejector system are the pin and the spring. A single ejector is shown in Fig. 7-16. The ejector pin A is manually operated by means of knob B, and is returned by spring C acting on the shoulder D, which also stops it in the return position. The pin is supported at E and F. The distance between the two supports must be several times the diameter, for jam-free guidance; and to provide, at the same time, the necessary space for the spring.

If the part has a central hole, a multiple ejector must be used with two or more ejector pins. The ejector shown in Fig. 7-17 has two pins A, fastened to a flange on knob B. The return spring C is centrally located and acts on the flange. The arresting shoulders D are parts of the pins, which also provide the inner bearing surfaces E, while the outer bearing surface F is in a bore in the knob. Other arrangements are also possible. Each pin could have its own two bearing surfaces and its own spring.

The simplest means of operating an ejector is the hand-operated knob, as shown, but it requires that the ejector be arranged in the side of the fixture.

When ejection is in the upward direction, the knobs are inaccessible for direct hand operation, and a lever must be used. The example shown in Fig. 7-18 is a single ejector. The design is simplified by the absence of a spring as the pin will retract by its own weight.

In the multiple ejector shown in Fig. 7-19, the pins A are carried by a ring B, and the lever C is forked.
The ring with the pins is returned by means of springs D. Pins and springs must be evenly spaced on the ring; on large diameters there should be at least three pins. Two springs are theoretically sufficient; however, three are recommended.

Small parts can be automatically ejected by direct spring pressure. Two arrangements for this purpose are shown in Fig. 7-20 a and b. In diagram a the part is located against the fixture base by means of clamps; the ejector pin A is forced down flush with the base when the part is loaded. In diagram b the defining surfaces B are above the part which is lifted into proper position by the force from the ejector spring. In each case the part is automatically ejected as soon as the clamps are removed.

Spring-operated ejectors, however, are insufficient for heavy parts and for parts that bind on the locators. In such cases, screw ejectors are used. An example is shown in Fig. 7-21. The screw A has a handle B. Since this handle must project through a window in the wall of the fixture base, it can only be operated through a small fraction of a revolution (30 to 45 degrees) and it is, therefore, necessary to use a screw thread with a large lead, which in turn, may require a double- or triple-thread screw. When a screw ejector is used for lifting a heavy part, it must also be designed so that it stays in the elevated position under load. In this way the operator has both hands free for removing the part. This is automatically accomplished (the screw is self-locking) when the helix angle of the thread is less than the angle of repose (the angle of friction, 8½ to 14 degrees, corresponding to μ = 0.15 – 0.25) and may require the use of a relatively large pitch diameter.

Ejectors can also be cam operated or wedge operated, which is particularly applicable to fixtures for multiple parts. A representative example is shown in Fig. 7-22. The fixture holds four parts, and the four ejectors are operated by one push-rod A, with four milled notches, tapering on one end.

The pins B are bored and slotted to receive and guide the ejector pins C. The four ejector pins are operated simultaneously by pushing in rod A.

The principle also can be modified in various ways:

1. Two (or more) pushrods can be operated by one knob
2. The ejector pins can operate on one large part instead of on individual small parts
3. The push-rod, or rods, can be operated by a lever to provide a greater lifting force with normal operator effort.

An extremely simple, almost primitive, but very effective yet inexpensive ejector system is the following: The necessary number of ejector pins are fastened into a plate in the desired pattern, and the plate is then clamped to the machine table. The system is only applicable to drill jigs so small and light that they can be lifted without undue effort. Holes are drilled in the bottom of the drill jig corresponding to the pattern of the ejector pins.
When the drilling operation is completed, the operator opens the clamps so that the part is free, then lifts the jig and lowers or forces it down over the pins until the part has been pushed free of the jig.

**Loading of Large and Heavy Parts**

To facilitate handling when loading large heavy parts, the hard physical work should be done in a conveniently accessible space unencumbered by the machine tool. This is frequently accomplished through the use of dual fixtures set on opposite ends of a machine tool table, or with one or several fixtures mounted on an indexing rotary table, but a means can also be incorporated in the design of a single fixture. General rules for the design cannot be formulated, but a few representative examples of such devices, which take many different shapes, will be shown.

The fixture has a movable receiver, which is moved to the outer station for loading and unloading and returned to the actual fixture station for machining. Being part of the fixture, the receiver must have devices for accurately locating itself when it is back in the fixture station, firmly supporting it and, if necessary, locking it in that position.

**Receivers, Sliding or Rotating**

When the weight of the part still permits it to be manually moved on a smooth horizontal surface, a receiver may not be needed, but the fixture base can be extended sufficiently outside of the machine headstock area to allow the part to be conveniently set off and then pushed into the fixture space as shown in Fig. 7-23.

Rotating or swinging receivers may take many forms and may perform additional functions within the fixture. The receiver R shown in Fig. 7-24 is actually a swinging drill jig. With the receiver in

![Fig. 7-22. A wedge-operated ejector for a multiple-part fixture.](image-url)

![Fig. 7-23. A fixture with an extended base for easy removal of a heavy part.](image-url)
Fig. 7-24. A drill jig with a swinging receiver.

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the open position, the part is clamped on by means of clamps \( C \), the receiver is swung back and locked in position, and drilling is done through the bushings in the receiver. The fixture shown in Fig. 7-25 is also a drill jig. The receiver \( R \) carries a central locator \( L \). With \( R \) in the outer position, the part is located over \( L \) and clamped by means of clamps \( C \). \( R \) is swung back and the drilling is done through drill jig bushings mounted in the stationary bracket \( B \). The same principle is frequently used in the design of broaching fixtures.

A somewhat different use of the receiver principle is shown in Fig. 7-26. For small parts, the problem is not weight, but quantity, and the receiver functions here as a prepositioner for a surface grinder with a magnetic chuck or faceplate \( C \). The use is therefore limited to magnetic materials. The receiver \( R \) is a plate, supported in trunnions and equipped with permanent magnets. With the magnet side up (diagram a), \( R \) is loaded then rotated 180 degrees and lowered onto the magnetic chuck (diagram b).

Fig. 7-25. A drill jig with a swinging receiver.

The permanent magnets carry the weight of the parts, but the stronger magnetic chuck, when switched on, pulls the parts down to its own surface in position for grinding. The receiver, now empty, is returned to its loading position. It is also a time-saving device in that it permits the loading of a new batch of parts during the grinding of the previous batch.

Fig. 7-26. A combined receiver and prepositioner.
Chip Problems

Types of Chips

Machining cast iron, bronze, and other brittle materials produces crumbling chips and a great deal of dust. Steel and other ductile materials produce several types of chips. A single-point tool, cutting at high velocity, as most carbide tools do, produces continuous chips which are long, usually curled into a helix of some sort, but sometimes snarling and bundling together, uncontrolled. With the use of a chip breaker on the tool, the chip flow is brought under control.

Chip breakers, as a rule, are not used on high-speed steel tools as the cutting speeds and the volume of chips produced are much lower; the chips are of the discontinuous (segmental) type and contain numerous slip planes and cracks so that they are brittle, and easy to break. Chips take up a much larger volume than the metal from which they were formed. Small chips require at least three times the original volume; snarling and loosely wound helical chips may occupy up to 20 times the original volume.

A twist drill is essentially two single-point tools: the lips—set at an angle, the point angle; and connected across the center by a short edge—the chisel edge. The angle across the chisel edge is a large, obtuse angle, so that the chisel edge removes metal by a combination of extruding and negative rake cutting, thereby generating a large thrust force. The chips produced by the chisel edge are small and curling and in themselves present no problem. The two lips produce the same chip types as single-point tools except that the chip is formed and fed into the confined space of the flutes of the drill. This eliminates the possibility of large diameter, helically wound chips and reduces the chip types to straight chips; tightly wound helices; and short, or crumbling, chips. With coarse feeds, the first two types have sufficient rigidity to travel through the flute. With fine feeds, the chips may snarl and pack the flutes. Therefore, an understanding of the behavior of drilling chips is essential to the fixture designer for the correct axial placement of drill bushings.

A face mill is essentially a plurality of single-point tools set in a circle and produces the normal chip types with the limitation that no chip can be longer than the path of the cutter tooth across the work. Most other types of milling cutters produce short chips since the path of the cutter tooth within the material is short. In ductile materials the chips are tightly rolled. Long side mills with helical teeth (slab mills) produce stiff, needle-shaped rolled chips as long as the width of the work.

Burr Formation

Burr formation is a result of the stresses in the metal caused by the pressure exerted by the tool. Consider a small element at the cutting edge and well within the metal (point $A$ in Fig. 8-1). The element is loaded and stressed by the pressure from the tool but is also backed and supported by the parent metal from behind and from two sides. A similar element (point $B$) bordering upon the free metal surface has less support from behind and no support from one side. Consequently, it will try to escape from the pressure in the direction of no resistance and in doing so it forms a burr. Burrs develop at the two points $B$ and $C$ where the edge of the cutting tool intersects the metal surface. A larger burr, the major burr, forms at point $B$ on the leading edge (the side-cutting edge in cutter terminology); and a smaller burr, the minor burr, forms at
burr is automatically provided for if the part is located clear of the fixture base; if not, a hole of generous size must be provided in the fixture base for the drill to clear and the major burr to form. On the top side, clearance is provided between the drill bushing and the upper surface of the part for the minor burr and for other purposes to be discussed later.

Chip Removal

Chips falling on the fixture present no more of a problem than those falling on a part without a fixture and they are easily scraped or brushed away. The most efficient assist in chip disposal is a strong flow of coolant, usually in excess of what the cutter requires for cooling alone.

A few minor yet important points must be considered by the fixture designer. The means for coolant supply and coolant return must be adequate. This requires a check on the pump capacity, pipe and nozzle dimensions, and the size and extension of the collecting grooves and channels in the machine tool table and substructure—including strainers, sieves, settling tanks, and filters. The flow pattern of a stream of coolant discharged to horizontal surfaces is not adequately controlled by the direction of the nozzle; it will splash and spray by the impact, spread over larger areas and overflow the edges in an unpredictable manner. This calls for troughs and pans to collect the liquid and baffle plates to protect the machine surroundings, including the operator. Shields and baffle plates for chips and coolant can be made wholly or partly from wire screen or Plexiglas to maintain visibility of the operating area.

Blowing chips away with an air jet is a double-edged sword and is endlessly discussed in the literature without a final answer; it is highly efficient in moving chips (particularly smaller ones), chip fragments, and dust; but not in removing them, because they settle down in some other place when they are outside of the air stream. They contaminate the workshop atmosphere with abrasive dust which finds its way into slideways and bearings in machines—and into the noses, ears, and clothing of the operators.

The disadvantages of the use of air blast for cleaning can be greatly reduced by skillful use of shields, baffle plates, chutes, and ducts. An example is shown in Fig. 8-3. A mechanical jig with a fixed locator plate $A$ and a movable jig plate $B$ is equipped with two airjets $C$ and an air valve $D$, actuated by the jig plate in its open position. The air blast cleans the jig plate and the locator plate before the part $E$ is inserted in the fixture. The chips im-

Fig. 8-1. The mechanics of burr formation.

Fig. 8-2. a. Major and minor burr formation by a single-point tool; b. A milling cutter; c. A twist drill.
Incidentally, the gravity a is a quantity the brush surrounded drill Fig. is Fig, chute and (Examples the chute. a relief it dry (This is Fig. 8-4.)

Chips that collect in quantity inside the fixture must be given an exit; for this purpose the fixture walls are provided with openings (windows) as large as the stress analysis can permit, with good clearance around the part for manual cleaning access with a rake, fork, scraper, or brush. Helpful also is the installation, where possible, of a gravity chute as chips are easily started down the slope by the vibrations from the machine.

Another, and somewhat unusual, example of chip removal by a chute involves the drill jig shown in Fig. 8-4. The part is a cylinder with two flanges that are to have holes drilled in them. The cylinder is bored and the flanges are faced. The part A enters the jig through one of the large windows in the side walls, and is centered by two plugs B, sliding in bores in the upper and lower end wall of the jig. With the jig set on end, the part with the plugs in position slides down until its lower flange rests on the jig. In this position the holes in the upper flange are drilled. To drill the holes in the lower flange, the jig is reversed. Chips are removed by the chip deflector C consisting of a split ring with a V-shaped cross section. Only one-half of the chip deflector is removed to unload and load the jig.

**Relief and Protection**

The greatest problem, one that confronts the fixture designer at all phases of his work, is caused by chip fragments and dirt; i.e., essentially chip dust and particles of rust, scale, paint, and remnants of foundry sand from castings. Collecting in corners and cavities they cause misalignment by preventing proper contact between part and locators.

Vertical surfaces are, naturally, less exposed to dirt accumulations. Horizontal locating surfaces are relatively easily cleaned; this is one reason why locating surfaces should be kept small and why side and end stops preferably should be installed from vertical walls.

When a flat surface with sharp edges slides over another flat surface the leading edge acts as a scraper. In this way the edges of locating pads help to clean the surface of the part. Occasionally, additional grooves with sharp edges are cut into locating pads to improve the cleaning effect. In the same way, the edge of the part cleans the locator surface, provided the dirt has a place to go. This leads to the general rule for dirt-proof design: A contact surface shall be surrounded by a relief space. Incidentally, relief spaces for dirt will also serve as relief spaces for burrs from previous operations. They do not accommodate large chip accumulations, but they allow space for the dirt to escape when it is pushed, swept, or scraped ahead of an edge on an entering part.

Any horizontal locating surface satisfies the condition above if it is elevated sufficiently over the fixture base. (Examples were seen in Fig. 6-5.) Additional constructive action must be taken in the corners by providing relief space either in the fixture base or in the locator. Pins and buttons can be installed in bored holes with a chamfer or a straight recess, as in Fig. 8-5a, b, and c. (This solution requires an additional and rather expensive machin-
In some cases it is possible to let a chamfered recess on a side or end stop perform two functions as shown in Fig. 8-7. The part is flat and the side locator is made higher than the part. The locating surface is inclined to a point above the edge of the part. With the clamping pressure applied, as indicated by the arrow, the part is not only located and clamped sideways, but is also forced down on the base. The angle of inclination is from 7 to 10 degrees. This provides sufficient holding pressure without risk of jamming or wedging. The same arrangement can be used for other types of locators (nesting blocks, V-blocks, etc.). It requires fairly close tolerances on the height of the part. The horizontal line through A does the locating. The coordinates x and y are critical dimensions because they define the position of A, within the fixture. In cases where the two perpendicular base and side locating surfaces meet in a corner, a relief groove is cut in the corner. The details may vary from case to case. Four different configurations are shown in Fig. 8-8.

Fig. 8-7. A side locator with an inclined locating surface.

The design of dirt relief spaces for circular locators follows the same lines as those previously described. An example is shown in Fig. 8-9. In this case, as well as for pins and buttons, the groove actually
Fig. 8-9. Relief space in a circular locator.

performs three functions; it provides space for dirt, relief for burrs, and clearance for grinding operations.

Shields and Seals

A fixture with moving parts, such as bearings and sliding pins and wedges, is actually a piece of machinery. As such, it requires protection against dust getting to its bearing surfaces. Reasonably good protection is offered by the use of dust caps, as shown in Fig. 8-10a. More effective protection is obtained with felt washers (Fig. 8-10b) but they have the disadvantage of a limited life; they must be kept oiled and renewed from time to time. Modern technology offers a variety of seals with excellent, although not infinite, service life. The O-ring is a well-known example. Where an effective dust seal is required, the fixture designer should consult catalogs on available sealing components.

Indexing fixtures represent an important class of fixtures with moving parts. The required precision can only be maintained if all bearing surfaces are well protected. The typical design of an indexing fixture is shown in Fig. 8-11. The center pin (king pin) is shielded by means of a cap, A. Seals B are provided to protect the index pin and the main bearing surface.

Shielding and sealing, as described, is required when a movable component is in an exposed position. Many fixture designs present an automatic shielding device at no cost, namely, the part itself. A large flat part provides quite an effective shield against chips falling on base locators and intermediate supports. To fully utilize this effect, locators are placed a short distance inside the contour of the part.

Fig. 8-10. Chip protection by a shield (a), and a seal (b).

Fig. 8-11. Shield and seal applied to an indexing fixture.
Centralizers

Definitions

Centering is an advanced method of locating. While locating, as previously described, brings one surface at a time into the proper place relative to the fixture, centering is applied to two surfaces at a time and locates a plane within the part—almost always the middle plane between the two surfaces.

Centering has three degrees. Single centering is when one middle plane is located, double centering is when two middle planes (usually perpendicular) are located, full centering is when three middle planes (likewise usually perpendicular) are located. The components for centering are termed "centralizers." Arrows 1,1; 2,2; and 3,3 in Fig. 9-1 indicate three pairs of centralizers. Single centering with one pair of centralizers 1,1 locates the middle plane aa and nothing else. Double centering with the additional pair of centralizers 2,2 locates two middle planes aa and bb, as well as the axis 3,3 where aa and bb intersect. Full centering with the further addition of centralizers 3,3 locates three middle planes aa, bb, and cc, three axes 1,1; 2,2; and 3,3; and the common center for middle planes and axes.

Advantages

In the introductory discussion to locating, it has been stated that one purpose of a fixture is to approximately substitute for the scribed lines and punched centers provided in the initial layout of a rough part—prior to machining. It was also explained that the part locates within the fixture from one or several of its surfaces. With tolerances, sometimes quite wide on a rough part, there is no guarantee that the physical middle planes, axes, and centers of the rough part will coincide with the corresponding planes, axes, and centers of the fixture. The fixture can only guarantee the correct location of machined surfaces relative to the locators and to each other, regardless of rough part tolerances. The application of centering devices has advanced the function of the fixture to the point where it provides a true substitute for the scribed and punched layout markings. The physical middle planes, axes, and centers within the part are now exactly located in the fixture (within tolerances). Machining allowances are evenly distributed, depth of cut is constant on all sides, and excessive cutting forces are avoided. Center of gravity is also correctly located and any unbalance of rotating parts (in turning operations) is eliminated. Surfaces that remain unmachined are more accurately located relative to system lines and planes in the part and in the completed product. Entire machining operations may be eliminated by the use of cold rolled and cold drawn stock located and clamped in accurately centering devices.

Centralizers and Locators

Centralizers are single or multiple components. They act as locators, as clamps, or both. A fixed single component centralizer is a locator. Multiple component centralizers have at least one movable part. They can have one fixed (a locator), and one or more movable components (clamps). When all components of a centralizer are movable, they can be considered as either clamps or locators. Then there is no real distinction between locators and clamps.

Typical combinations of locators and centralizers are shown schematically in Fig. 9-2. Double centering is accomplished whenever an axis is located, regardless of what system of centralizers is used. By this definition the common three-jaw, self-centering chuck and the collet chuck are double-centering devices (see diagram g), as is the two-jaw chuck frequently used on smaller turret lathes (see diagram e). Drill chucks are also double-centering devices. The
Fig. 9-1. Single, double, and full centering.

A machine tool vise with two V-grooves (see diagram f), is a single-centering device. Centralizers are not nesting components. They provide positive contact and pressure without clearance. Locating with centralizers is, therefore, more closely defined and more accurate than nesting, particularly on rough parts.
Centralizers can be classified into three categories:

1. The angular block type
2. The linkage controlled multiple centralizer (automatic or scissor-type)
3. Self-centering chucks of commercial types.

The term “angular block” is the common designation for block-type locators with converging or diverging locating surfaces. They are used in three
forms, the V-block, cone locator, and spherical locator. The V-block has two flat surfaces. In the most widely used type of V-block the V is concave; the two surfaces form a slot or groove which receives the part. An inverted configuration is used occasionally where the V is convex and forms the two surfaces of a triangular prism which centralizes by entering the space between the prongs of a fork-shaped part. The cone locator has a convex (male) or concave (cup, female) conical surface. The included angle within the locating surfaces is significant for the function of the locator. The locating surface of the spherical locator is formed as a spherical cap or ring and can be convex or concave.

Linkage controlled multiple centralizers are those in which the movable components are so connected that they maintain equal distance from the middle plane, the axis, or the center. One example is a pair of scissors, and scissor-like linkages are frequently, but not exclusively, used. The term "linkage" is here used in a wider sense; it includes mechanisms that act with cams, wedges, telescoping rods, symmetrically arranged springs, etc. In mechanical language these are all called kinematic chains.

Centralizers and Equalizers

In a description of fixture design there are two terms that must not be confused: "centralizers" and "equalizers." Both use linkages, but they serve entirely different, almost opposite, purposes. In centralizers, the linkage system controls the motion and position of the locating and clamping points, and forces the part into a position defined by these points. Equalizers are also linkages and are used for clamps for the purpose of equalizing the clamping forces on uneven surfaces of the part. The equalizers carry the clamping points and enable the individual points to adjust themselves back and forth to accommodate local irregularities. They do not force the part into a position; they are forced into position by the part.

Commercial Centralizers, Chucks

Self-centering lathe chucks have jaws which are made to move in a concentric relationship to the spindle axis. They can therefore be classified as centralizers and are available with two or three jaws. The four-jaw chuck is also a device for centering, but the jaws are adjusted individually and manually. All these devices are commercially available, general-purpose work holders; as such, they do not qualify as fixtures. However, they become fixtures, or rather fixture bases or fixture bodies, when they are provided with special jaws or jaw inserts and locators to fit specific parts. Machine tool vises are also commercially available, general-purpose work holders, not fixtures. They are not, in themselves, centralizers, but they too can be fitted with special inserts and other components for special work.

A general-purpose work holder, "borrowed" from its parent machine and mounted on a base, can serve as a major fixture component with other components built upon it or around it. Drill chucks are used to good advantage for holding small parts. Instead of physically acquiring a piece of general-purpose equipment, the fixture designer may feel inspired by its design principle and apply it to the assignment in hand. The principle of the collet chuck with one or two sets of contracting or expanding elastic fingers can be applied to centralizers that can be mounted on an arbor or on a base. Fixtures of this type are excellent for locating and clamping on internal and external cylindrical or conical machined surfaces where high precision is required and the load from the machining operation is relatively light. The particular advantage is that they exert a uniform pressure on the part and do not force it out-of-round. They do have one serious limitation, however, as shown in Fig. 9-3; the fingers bend and their slope varies as they expand or contract. Theoretically, there is only one position where they grip and locate with their full surface. In actual practice this means

Fig. 9-3. The principle of the collet chuck.
that they operate satisfactorily only over a small diameter range; inside and outside of that range they grip only with the edge.

The inclined angle on the conical surface of the centralizer is 30 degrees; the activating (mating) cone is made with 31 degrees if it acts on the inside of the fingers as shown in Fig. 9-3, and 29 degrees if it acts on the outside. The thickness of the fingers must be small to ensure sufficient flexibility. For work diameters \( D \), ranging from 1/4 inch to 6 inches (6 to 150 mm), the recommended thickness \( t \) varies from 5/64 inch to 13/64 inch (2 to 5 mm).

The question may arise whether hollow parts should be located (centered) on the inside or on the outside; if so, the following points may be considered: Any radial locating error (eccentricity) is reproduced to true size, not more, not less, regardless of where the part is located. A misalignment error (a wobble) is reproduced to true size if the part is centered on the outside, but is reproduced with a magnification if the part is centered on the inside and on a small diameter—the magnification of the error increases with the diameter ratio.

For a given clamping pressure, the transmitted torque and the maximum permissible size of cut is greater when the part is clamped on the outside; also, with outside clamping the part is less likely to slip in case of an accidental overload in the cut.

Centering by Means of V-Blocks

The common method of centering cylindrical pieces or surfaces in a V-block is shown in Fig. 9-4. The V-block, as a rule, is stationary, held in place by screws and dowel pins, as indicated in the figure. However, the V-block may also be adjustable in order to take up the variations of the pieces placed in it, and in order to act as a clamp. A V-block of this type is shown in Fig. 9-5. Here, \( A \) is the adjustable V-block, having an oblong hole \( B \), to allow for adjustment. The block is held in place by a collar-head screw \( C \), which passes through the elongated hole. The underside of the block is provided with a tongue \( D \), which enters into a slot in the jig body itself, the V-block thereby being prevented from turning sideways. The screw \( E \), passes through the wall of the jig, or through some lug, and prevents the V-block from sliding back when the work is inserted into the jig. It is also used for adjusting the V-block and, in some cases, for clamping the work.

V-blocks are usually made of machine steel, but when larger sizes are needed they may be made of cast iron. Little is gained, however, in using cast iron, as most of the surfaces have to be machined, and the difference in the cost of material on such a comparatively small piece is very slight.

![Fig. 9-5. An adjustable V-block used as a locator and centralizer.](image)

For large size V-blocks it is economical to use finish machined, cast iron V-block stock, commercially available in widths up to 4 1/2 inches (120 mm) and in lengths from 2 to 3 feet (600 mm to 1 m). When a V-block is used for locating round parts, there is so much empty space left that there is no particular need for a relief groove for catching chips and dirt. When a relief groove is used, as it often is, the purpose is to provide clearance for the grinding wheel used for finishing the flats of the V-block. The groove must be made with rounded corners or as a semicircle, to reduce stress concentration.

Much mathematics has been applied in attempts to find and to justify an optimum value for the included angle, but no convincing calculation has yet appeared in the literature. Some extreme limits can be easily established. A V-block with a 30-degree angle will hold a part very firmly and is near the point where the part becomes wedged by friction. A small diameter variation causes a large variation in the height at which the part rests in the V. Thirty degrees is clearly a lower limit, and not even a practical one. A V-block with a 120-degree angle will receive the part freely and is not very sensitive to diameter variations. The position of the part is not very stable; it takes a relatively small horizontal force at the clamping point to roll the part out of
CENTRALIZERS

its resting place. 120 degrees is clearly an upper limit and not a desirable one.

Industry has solved the problem by accepting, almost universally, the value of 90 degrees for the included angle. Most V-block components in commercial fixtures are made with this angle, and V-block stock (rough and machined castings) is commercially available with tolerances down to 90 degrees ±10' for the included angle and ±0.002 inch per foot (0.2 mm per m) for straightness.

The 90-degree V-block is popular and rightly so. It provides a good stable support for circular cylindrical parts and is insensitive to even grossly inaccurate application of the clamping force (Fig. 9-6). With the clamping force acting on the top of the cylinder, it can deviate ±22 1/2 degrees from the vertical direction before the position of the part becomes unstable and it starts rolling. The point of action of the clamping force can move 45 degrees to either side before stability is lost. Other advantages of the V-block are that it is solid, strong, and rigid; it provides good bearing areas, is suitable for long as well as for large parts, lends additional stability and strength to the fixture, is versatile in its applications, and is inexpensive.

Fig. 9-6. The stability range of the 90-degree V-block.

Limitations of the V-Block

Since the V-block has so many easily recognizable good points, there is also the danger that it may be used for the wrong purposes. Its capability as a centralizer is quite limited; taken individually it provides only single centering, but it does that well. The plane in which it centers the part is the bisector of the angle. This centralizing effect is independent of the diameter of the part, up to the limit of capacity of the V-block, and is used in locating for the machining of any surface and configuration that is symmetrical with respect to the bisector plane, or which is dimensioned entirely and solely relative to this plane. Examples of such configurations are (see Fig. 9-7) holes and slots passing through or across the part in the plane of symmetry, and planes parallel to that plane. The words “through or across” are significant. Consider a blind hole or longitudinal keyseats (see diagram e). They are machined in perfect symmetry, but to a depth that depends on the physical diameter of the part. In a great many cases this objection is academic only, as diameter variations within tolerances are small, and blind holes, keyseats, and similar configurations are usually designed with a generous depth tolerance that can absorb the small error from the diameter tolerance.

More serious, perhaps, is the effect of diameter variations on the location of configurations that are dimensioned relative to the diameter, that is, perpendicular to the bisector (see diagram f). A symmetrically designed hole, keyseat, or slot will move clearly out of symmetry with a diameter variation. This effect is so obvious that many sources call it a wrong use of the V-block. The criticism is correct in principle, but exaggerated in reality; the real problem is again a problem of tolerances (see Fig. 9-8a and b). In a, the V-block is used as a centering device for a cylindrical part. When Δ is the variation of the part diameter, then the center of the part is located on the bisector with a locating error e, determined by

\[ e = \frac{1}{2} \Delta \sqrt{2} = 0.707 \Delta \]

In b, the V-block is used as a base and side locator; a use for which it is eminently suited. In this application the error in the horizontal (or vertical) direction is clearly

\[ e = \frac{1}{2} \Delta \]

The Sliding V-Block

With the few reservations stated, the V-block with a single clamp is very suitable for locating and single-centering circular and cylindrical parts. Long parts of ample stiffness require a V-block at each end rather than one long V-block. The area of application of the V-block principle is significantly expanded by combining one fixed and one movable V-block, the movable V-block acting also as the clamp. This system is widely used for elongated
parts with rounded (partly circular) ends. A typical example is shown in Fig. 9-9.

The drill jig shown is designed for drilling fork links. The form of the links is indicated by dot-and-dash lines in both views. The link has a round boss at one end and rounded forks at the other. It is accurately held between two V-blocks, one adjustable and the other stationary. The adjustable V-block A is clamped against the work by a star-wheel and screw, and it travels between finished ways, thus providing an accurate as well as a rapid method of clamping. These V-blocks have inserted steel plates B and C. The latter, which is in the stationary V-block, carries a drill bushing for drilling the lower fork, and an upper shoulder on this plate provides a support for the upper fork; thus there are two bushings in alignment for drilling the two ends. The inserted plate B in the adjustable block supports the opposite end of the fork link. With this arrangement, a double V-clamping jig is obtained having a three-point support. This drill jig was accurate, rapid, and easily operated.

The principle of the sliding V-block can be applied to parts of the most diversified shapes, as long as they present bosses or other contours with at least a little more than 90 degrees of a circle.

V-blocks are not often used for locating square and otherwise prismatic parts. One reason is that such use of the V-block is actually nesting, with its inherent lack of accuracy, mainly in the matching of the angles. Parts with flat surfaces are better located on base points with side and end stops, on strips, or in a vise-type of fixture. Another reason is that the V-block provides centering with respect to a diagonal, a feature rarely called for.

Conical Locators

The conical locator is well known in the machine shop, although not necessarily by that name. To
center drill a part prior to a turning operation and set the part up on the lathe centers is actually locating with conical locators. The tapers in and on machine tool spindles and the corresponding tapered shanks on drills, arbors, chucks, etc., are conical locators, but these tools are not fixtures. The lathe mandrel is a work holder with a tapered seat for the work. The taper is very small, 0.006 inch per foot (0.5 mm per m). A common characteristic of these devices is that they transmit torque by friction. The lathe mandrel does not define the axial position of the part; it is actually determined by the bore tolerance and the amount of pressure used when the part is mounted. The lathe mandrel is a general-purpose work holder, not a fixture.

From this brief resume it follows that conical locators cannot duplicate these devices. The conical locator can center and does that well. It can provide some degree of axial locating but not with precision. Integrating a conical centralizer and a flat axial locator (see Fig. 9-10) in one piece requires extremely close tolerances and is impractical except for special, high-precision work. A good workable solution is to mount a sliding conical locator within a flat axial locator and provide independent clamping devices. The example shown in Fig. 9-11 is typical of the application of this principle and contains some additional features necessitated by the need for gripping and clamping the work by a thin rim, without distorting it.

The work A is a special clutch flywheel which has been partially machined. In order to obtain concentricity of the various surfaces, it is necessary to locate the work from the taper in the hub. In order

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**Fig. 9-8.** The effect of diameter tolerance on the locating error.

**Fig. 9-9.** Locating and single centering by means of a fixed and a sliding V-block.
to compensate for slight variations between the taper and other finished surfaces, a tapered, shell-locating bushing $B$ is centrally located on the stud $C$, which is held in place in the faceplate fixture $E$ by the nut and washer at $D$. A light coil spring $M$ insures a perfect contact with the tapered surfaces, while a small pin $N$ restrains the movement. As the outside of the work is to be finished during this setting, it is necessary to grip the casting in such a way that the clamps will neither interfere with the cutting tools, nor cause distortion in the piece itself. With this end in view, the three lugs around the rim of the fixture are provided with shell bushings $K$, each of which is squared up at its inner end to form a jaw which is bored to a radius corresponding with the rim of the casting $L$. It is splined to receive a dog screw $J$, which prevents it from turning, and it also gets a good bearing directly under the point where the work is held so that there is no danger of it springing out of shape.

The bolts $F$ pass through the shell bushings and are furnished with nuts $G$ at their outer ends, the nuts having a knurled portion $O$, which permits of rapid finger adjustment before the final tightening with a wrench. It will be seen that this construction automatically obtains a metal-to-metal contact with the thin flange of the casting, without distorting it in the least, as the floating action of the bushings equalizes all variations and yet holds the work very firmly. After the clamps have been set up tightly, they are locked in position by the set-screw $H$, at the rear of the fixture. This application of the floating principle may be adapted to many kinds of work, and the results obtained leave nothing to be desired.

The machine for which this device was designed is a turret lathe of the horizontal type.

Conical locators do not necessarily require tapered bores to work with; they also work very well with circular edges. Any part with a cylindrical outer surface or a cylindrical bore and one or two flat end surfaces can be centered by means of conical locators provided that the edge is really circular. This requires that the end surface be perpendicular to the axis. If the part has been machined, the edges must be inspected and any machining burr removed. Castings with cored holes are likely to have fins around the core prints that must be cleaned away before the edge of the hole can be used for locating.

Whenever it is essential that a cylindrical part of the work be located centrally either with the outside of a cylindrical surface or with the center of a hole passing through the work, good conical locators can be designed as shown in Figs. 9-12 and 9-13. In Fig. 9-12 the stud, $A$, is countersunk conically (cup locator) to receive the work. The stud is made from machine or tool steel, and may, in many cases, serve as a bushing for guiding the tool. In Fig. 9-13 the stud is turned conically in order to enter a hole in the work. These two cone locators are stationary; they are only used for locating the work and would require additional means for clamping.

### Clamping with a Moving Cone

Clamping devices for use with cone locators can be separate and independent, but it is also possible, and very convenient, to make one of two locators movable and use it for clamping. The bearing area on the edge is small and the clamping load must be kept light to avoid deformation of the edge, or other damage. Clamping by means of a movable cone locator is widely used in connection with drill bushings. Drill bushings with an external screw thread are known as "screw" bushings and may be used for locating and clamping purposes by making them long enough to project through the walls of the jig and by turning a conical point on them, as shown in Fig. 9-14, or by countersinking them, as in Fig. 9-15. In all cases where long guide bushings are used, the hole in the bushing ought to be counterbored or recessed for a certain distance of its length.

In some instances the screw bushing must be movable sideways, i.e., when the piece of work to be made is located by some finished surfaces, and a cylindrical part is to be provided with a hole drilled exactly in the center of a lug or projection, the relation of this hole to the finished surfaces used for locating is immaterial. The piece of work, being a casting, would naturally be liable to variations between the finished surfaces and the center of the lug, particularly if there are other surfaces and lugs to which the already finished surfaces must corres-
In such a case, the fixed bushing for drilling a hole that ought to come in the center of the lug, might not always suit the casting and so-called "floating" bushings, as shown in Fig. 9-16, are used. The screw bushing $A$ is conically recessed and locates from the projection on the casting. It is fitted into another cylindrical piece $B$, provided with a flange on one side. The piece $B$, again, sets into hole $C$ which is large enough to permit the necessary
adjustment of the jig bushing. When the bushing has been located concentric with lug E on the work; the nut F, having a washer G under it, is tightened. The flange on piece B and washer G must be large enough to cover hole C, even if B is brought over against the side of the hole. It is seldom necessary, however, to use this floating bushing, for a drilled hole in a piece of work rarely can be put in without having any direct relation to other holes or surfaces.

Fig. 9-16. A floating drill bushing used as an outside conical locator.

**Linkage Controlled (Automatic) Centralizers**

For the control of the moving components the following mechanisms (kinematic chains) are used:

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<td>Screw and nut</td>
</tr>
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<td></td>
<td>Screw with right- and left-hand thread (turnbuckle principle)</td>
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<tr>
<td>Rotating arms and cams</td>
<td>Linkage systems</td>
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<td>Symmetrically moving pairs of levers</td>
<td>Scissor-type linkages</td>
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</table>

The list is representative, but not comprehensive; a complete list would require a kinematics encyclopedia. There is still room for the inventiveness and ingenuity of the fixture designer. As a guide, a few fundamental rules are given:

1. Make it simple
2. Prefer rotation to sliding
3. For rotating members—apply forces perpendicular to radii
4. For sliding members—provide support below the points of force application
5. Specifically for lever arms in linkages—
   a. Make arms of equal length; if not feasible, then operate on the long arm and let the short arm perform the clamping
   b. Should this not be feasible, operate through a force-magnifying device, such as a screw mechanism with a handwheel
6. For all mechanisms—
   a. Watch for rigidity of all individual members, and for
   b. backlash in all bearings and other points of contact.

To illustrate these rules, a few typical examples follow.

Kinematic chains do not squeeze and clamp as hard as single clamps, because the available force is divided between several locating and clamping points. Primarily used for drill jigs rather than for milling fixtures, they are recommended when drilling flat plates and covers which are not usually machined at the sides but have to be gripped or located in a jig by their rough cast edges. Similarly, such self-centralizing features will be found advantageous, timesaving, and economical in drilling parts having a similar shape, but whose overall dimensions differ.

One of the simplest forms of a self-centralizing device for a drill jig is shown in Fig. 9-17. This jig is an example of the type with sliding wedges actuated by opposing springs. It can also be characterized as a jig with a split V-block and contains a rectangular cast-iron body A, which is flanged at the bottom for hold-down purposes. A swinging arm B is pivoted on a pin C that is pressed into both side walls of a slotted boss on top of the body. Arm B carries the drill bushing D. Pressed into the underside of the arm on each side of the bushing, are two bearing pads E. When the arm is in the horizontal position, as shown, these pads will press equally on the workpiece X, holding it firmly to the top of the jig body. The right-hand end of arm B has an open-end slot for the cylindrical shank of clamping stud F. A
knurled nut $G$, threaded on the upper end of this stud, enables the arm to be clamped to the work. The stud can be swung about a pivot pin $H$ pressed into the body. The centralizing action on the workpiece is obtained from identically shaped spring-loaded slides $J$, which are mounted in a guide hole $K$, drilled completely through the body. Springs $L$ are held in pockets in these slides by stop-plates $M$, which are fastened to the sides of the jig body by screws. Each slide has a vertical projection at the inner end, with the hardened end faces of these projections inclined at an angle of approximately 10 degrees. The projections slide in slots which extend from the top edge of the jig body into the bearing holes $K$. The springs force the slides inward, toward each other, and the extent of this movement is limited by either the end faces of the slots or the workpiece, as shown.

A large-diameter vertical hole extends down through the body, directly below the hole to be drilled in the workpiece, to permit chips to fall out of the jig. A short cylindrical plug, cross-drilled as shown, is tightly pressed into hole $K$ to lie midway between slides $J$. This plug prevents chips from entering hole $K$. To mount a workpiece in the jig, arm $B$ is swung upward. The two opposing sides $J$ will be in their innermost positions, in contact with the end faces of the slots, and the workpiece is placed between the inclined faces of the slides. Transverse location of the workpiece is obtained by butting it against a plate $O$ fastened to the top face of the jig body. Arm $B$ is then lowered into the horizontal position and clamping stud $F$ is swung into its vertical position, as shown. As nut $G$ is tightened, pads $E$ bear down on the workpiece and press it into contact with the top of the jig body. This action will cause the slides to move apart an equal amount, and the workpiece $X$ will become located centrally with relation to bushing $D$. Parts differing considerably in width can be accommodated without difficulty in such a jig, since the slides can be arranged with an appreciable amount of opening and closing. Also, the jig may easily be adapted to increase these facilities merely by altering the angle of taper on the slides.

A typical design of a fully linkage-operated centralizer is shown in Fig. 9-18. It differs from the one shown in Fig. 9-17 by two very important features. The first is that the centralizing motion is positively controlled by the linkage, while in the previous case it was dependent on the symmetry in the spring arrangement. The second feature is that the effective opening has a large operating range so that this fixture can be successfully employed in drilling parts having considerable variation in width or length. As with the jig previously described, a swinging arm $B$ is pivoted on a pin $C$, which is pressed into jig body $A$. A drill bushing $D$ is carried in the arm, and located on each side of this bushing are the identical bearing pads $E$. The right-hand end of arm $B$ is slotted to admit clamping stud $F$, which is fitted with a knurled nut $G$ and pivoted on pin $H$.

The bellcrank locating and clamping levers $J$ are a sliding fit within a narrow slot in the jig body, and pivoted on pins $K$ pressed into the body. Permanently fitted into a transverse slot in the body is a platform $L$ for supporting the workpiece $X$. Vertical clearance holes are provided in this platform, and in the jig body, to permit the chips to fall through.

The upper inner edges of levers $J$, which contact the sides of the workpiece, are rounded and hardened, and can be serrated to provide a better grip. The lower ends of the levers are reduced to half their total thickness so that they overlap, and the left-hand lever is slotted to fit over pin $M$ pressed into the tail of the right-hand lever. When the lower halves of the levers are in a horizontal position, the center of pin $M$ is aligned with the vertical centerlines of the jig body and drill bushing. This arrangement insures that the levers will be swiveled equally.
Actuation of the levers is obtained by means of rod $N$, the slotted shackle end of which is pinned to the right-hand lever. The cylindrical shank of rod $N$ is a running fit within an externally threaded sleeve $O$, which is screwed into the right-hand wall of body $A$. When handwheel $P$ is rotated, levers $J$ will be swiveled (due to the force of sleeve $O$) against the shoulder on rod $N$ or the collar pinned to the rod.

The manner of loading and using this jig is similar to the one previously described. With arm $B$ raised, workpiece $X$ is placed on platform $L$. The contact surfaces of levers $J$ will have been moved apart by rotating handwheel $P$. Transverse location of the workpiece is obtained by butting it against an adjustably mounted stop-plate $Q$. The handwheel is then rotated in the opposite direction until the work is firmly gripped and centralized by the contact surfaces of the levers. Arm $B$ is then returned to the horizontal position shown, and clamped by tightening nut $G$. Additional clamping pressure is thus exerted on the work by pads $E$.

An example of the use of rotating arms actuated by inclined flats is shown in Fig. 9-19. This type of self-centralizing jig has been proved economical and accurate in drilling uniformly central holes through thin cover plates and similar parts having large variations in width. In this jig, a swinging arm $B$ is again pivoted on a pin $C$, which is pressed into the jig body $A$. The arm carries a drill bushing $D$, and its right hand is slotted for a ring-head clamping bolt $E$ that carries nut $F$ and is pivoted on pin $G$.

A spring-loaded cylindrical plug $H$ is a sliding fit in the vertical bore of the jig body. Rotation of the plug is prevented by key $N$. The lower threaded end of the plug is screwed through handwheel $K$ which is carried in a horizontal slot in the jig body. The upper head of the plug, on which the workpiece $X$ rests, has two diametrically opposite slots into which
are fitted the triangular-shaped locator pads $J$. These pads pivot on pins $L$, pressed into the side walls of the slots. The lower, outer corner of each locator pad rests on the tapered bottom surfaces of slots cut in the side walls of the jig body. In operation, arm $B$ is raised, and a workpiece is placed on top of plug $H$, located against a fixed pin or plate, not shown. Then, by turning handwheel $K$, plug $H$ is drawn downward—against the action of spring $M$—and pads $J$ are pivoted toward each other, thus centralizing and gripping the workpiece. Arm $B$ is then lowered into a horizontal position and clamped by tightening nut $F$.

Wear can be minimized in this jig by screwing hardened head set-screws into the tapered bottom surfaces of the slots cut in the side walls of the jig body. The rounded contact points of pads $J$ would then bear on the heads, and slight adjustments could be made by tightening or loosening the set-screws.

An effective method of obtaining centralization of the work by means of a real linkage (in this case, a linkage of the pantograph type) is illustrated in the drill jig seen in Fig. 9-20. Secured to the top, near the rear edge of jig body $A$, is a slotted bracket $B$. A swinging arm $C$, pivoted about a pin pressed into the uprights of bracket $B$, carries drill bushing $D$. The forward end of this arm is slotted to admit the shank of a ring-head clamping bolt $E$ that carries a knurled clamping nut $F$. The ring head of the clamping bolt is a close fit in a slot on bracket $G$ secured

Fig. 9-19. A cam-operated centralizer.

Fig. 9-20. A linkage-operated centralizer with a pantograph mechanism.
to the front of the body and pivots about a pin pressed into the uprights of this bracket.

Workpiece $X$ is placed on the top surface of jig body $A$, bearing against stop-plate $H$ secured to the body for lengthwise location. The part is centralized transversely and gripped by means of two bars $J$, which rest on the smooth top surface of the jig body. The inner contacting surface of these bars are relieved slightly to reduce the frictional pressure on the workpiece. The ends of both bars are pinned to levers $K$, forming a pantograph mechanism. Levers $K$ can be pivoted about studs $L$. The front end of the extended right-hand lever $K$ has an elongated slot to fit around a pin $M$, pressed into the slotted end of rod $N$. The threaded shank of rod $N$ passes through plain holes in both walls of a slotted bracket $O$ secured to the right-hand edge of the jig body, and is screwed in the internally threaded handwheel $P$. Thus, when the handwheel is rotated, the pantograph lever system is swiveled about studs $L$, either moving bars $J$ together to clamp the work, or apart to permit loading and unloading. Arm $C$ is handled in the same manner as those on the jigs previously described; swung upward while unloading and reloading, and down into the position shown, when the workpiece is in place.

Centralizers for Gear Wheels

In the manufacture of gear wheels there are two types of operations which require locating of the gear from points on the tooth flanks in the pitch circle. They are the machining of a localized detail such as the drilling of a hole pattern or the milling or broaching of a keyseat, and the finishing of the bore concentric with the actual pitch circle as defined collectively by the teeth.

A localized detail is usually dimensioned from a tooth or tooth space and must be located accordingly. This is not just a case of radial locating from one side only, but the locator must pick up both sides of the tooth or tooth space and locate with respect to the bisector between them. The locating is a centralizing operation and can be done with a sliding V-block on a tooth or a sliding V-prism in a tooth space (see Fig. 9-21). The design of these components is based on the gear tooth geometry, as defined by the angles $\phi, b_1$, and $b_2$.

To ensure contact on the pitch circle, the included angle $a$ in the vee is determined by

for a V-block

$$\phi = \frac{a}{2} + b_1$$

$$a = 2(\phi - b_1)$$

for a V-prism

$$\frac{a}{2} = \phi + b_2$$

$$a = 2(\phi + b_2)$$

---

Fig. 9-21. Centralizers for gear wheels.
\( \phi \) is the pressure angle; the angles \( b_1 \) and \( b_2 \) depend on the gear data and the tolerances (backlash). All angles are in degrees. Using symbols in *Machinery's Handbook*, 19th edition, the gear data are:

- Diametral pitch \( P = \frac{N}{D} \)
- Number of teeth \( N = PD \)
- Pitch diameter \( D = \frac{N}{P} \)
- Chordal thickness \( l_c \)
- Tooth space chordal width \( s \)
- Backlash (circular) \( B \)

\[
t = D \sin b_1 \\
\frac{s}{D} = D \sin b_2
\]

Then, disregarding backlash

\[
b_1 = b_2 = \frac{90^\circ}{N}
\]

and with backlash

\[
b_1 + b_2 = \frac{180^\circ}{N}
\]

\[
b_2 = b_1 + \left(\frac{B}{2D} \times 57.3^\circ\right)
\]

\[
b_1 = \frac{90^\circ}{N} - \left(\frac{B}{2D} \times 57.3^\circ\right); \; b_2 = \frac{90^\circ}{N} + \left(\frac{B}{2D} \times 57.3^\circ\right)
\]

Suggested values for backlash:

- Minimum \( B = \frac{0.030}{P} \), Average \( B = \frac{0.040}{P} \),
- Maximum \( B = \frac{0.050}{P} \)

In connection with the metric system, gear dimensions are based on the *module* which is denoted \( m \), and is always expressed in mm. Modules are standardized in simple and round numbers. The relations between pitch and module are:

\[
P = \frac{25.4}{m} \left(\frac{1}{\text{inch}}\right)
\]

\[
m = \frac{25.4}{P} \left(\text{mm}\right)
\]

and the suggested backlash values, now expressed in mm, are:

Minimum \( B = 0.030 \text{ m} \), Average \( B = 0.040 \text{ m} \),

Maximum \( B = 0.050 \text{ m} \)

For centralizing in a tooth space, the V-prism can be replaced by a cone of the same included angle and by a circular pin or spherical ball, of radius \( R \) where

\[
R = \frac{s}{2 \cos \frac{a}{2}} = \frac{s}{2 \cos (\phi + b_2)}
\]

For centering a gear wheel from its pitch circle, a plurality (usually three, but sometimes four) of centralizers are used evenly spaced and mounted on the jaws of a self-centering chuck. If the centralizers are circular pins, the capacity of the chuck must be at least \( D_c \), determined by

\[
D_c = D \cos b_2 + 2R \left(1 + \sin \frac{a}{2}\right)
\]

\[
= D \cos b_2 + s \left(1 + \sin (\phi + b_2)\right) / \cos (\phi + b_2)
\]

\[
= D \cos b_2 + s \tan 1/2 (90^\circ + \phi + b_2)
\]

Cones and balls are also used as centralizers for holding helical gears. Straight solid pins cannot be used as they are not compatible with the helix (except when very short), but pins formed as cylindrical spring rolls and held in position so as to form a cage, are used for locating and holding helical gears between chuck jaws. Figure 9-22 shows four such pins in position.

The really "natural" centralizer for a gear is another gear or a section of a gear. A fixture for bevel...
Each centralizer is an insert with one gear tooth operating in a tooth space. The inserts can slide radially and are forced inward by the outer cone when the base plate is pulled lengthwise toward the spindle. The fixture shown in Fig. 9-24 has a cluster of small pinions acting as centralizers for external or internal spur and helical gears. Each pinion is mounted on an eccentric pivot and the work is clamped and centered relative to its pitch circle when all pinions are simultaneously rotated.

Example—A spur gear with 20 degree pressure angle has 6 diametral pitch and 72 teeth. A V-block locator shall be designed with and without considerations of average backlash.

\[
D = \frac{72}{6} = 12 \text{ inches}
\]

Without backlash

\[
b_1 = \frac{90^\circ}{72} = 1^\circ 15'
\]

\[
a = 2(20^\circ - 1^\circ 15') = 37^\circ 30'
\]

With backlash

\[
B = \frac{0.040}{6} = 0.0067 \text{ inch}
\]

\[
b_1 = \frac{90^\circ}{72} - \frac{0.0067}{2 \times 12} \times 57.3^\circ = 1^\circ 15' - 58''
\]

\[
a = 2(20^\circ - 1^\circ 14'02'') = 37^\circ 31'56''
\]

Using symbols in Machinery’s Handbook, 19th edition, a helical gear is characterized by the helix (spiral) angle \(a\), also known as the “tooth” angle, which is the acute angle between a tooth and the axis of the gear. The number of teeth \(N\), and the pitch diameter \(D\), have no relation to the helix angle. The chordal thickness \(t_c\) and the tooth space chordal width \(s\), refer to the pitch circle as shown in Fig.
In a section perpendicular to the tooth, the following dimensions are defined: The normal pressure angle \( \phi \), the normal diametral pitch \( P_n \), the normal chordal thickness \( t_n \), the normal tooth space chordal width \( s_n \), and the backlash \( B \). \( \phi \), \( P_n \), and \( B \) are selected from the conventional values as for a spur gear. In a section perpendicular to the gear axis are defined the tangential pressure angle \( \phi_t \), the diametral pitch \( P \), and the tangential backlash \( B_t \). The following relations hold:

\[
\tan \phi_t = \frac{\tan \phi}{\cos \alpha}
\]

\[
P = P_n \cos \alpha = \frac{N}{D}
\]

\[
B_t = \frac{B}{\cos \alpha}
\]

\[
t_n = t_c \cos \alpha
\]

\[
s_n = s \cos \alpha
\]

The included angle \( a \) for a "conjugate" (imaginary) V-section in the plane perpendicular to the gear axis, is calculated from the spur gear formulas and is converted to the included angle \( a' \) for the actual V-section (in the plane perpendicular to the tooth) by

\[
\tan \frac{a'}{2} = \tan \frac{a}{2} \cos \alpha
\]

Dimensions \( R \) and \( D_o \) relating to a pin or ball centralizer are calculated as before, by substituting \( a' \) and \( s_n \) for \( a \) and \( s \).

Example—A helical gear with \( 33^\circ 33' \) helix angle and 20 degree normal pressure angle has 6 normal diametral pitch, 60 teeth and minimum backlash. A circular locator (spring roll or ball) must be designed.

\[
\tan \phi_t = \frac{\tan 20^\circ}{\cos 33^\circ 33'} = \frac{0.36397}{0.83340} = 0.43673
\]

\[
\phi_t = 23^\circ 35' 33"
\]

\[
P = 60 \cos 33^\circ 33' = 5
\]

\[
D = \frac{60}{5} = 12 \text{ inches}
\]

\[
B = \frac{0.030}{6} = 0.005 \text{ inch}
\]

\[
B_t = \frac{0.005}{\cos 33^\circ 33'} = 0.006 \text{ inch}
\]

\[
b_2 = \frac{90}{60} + \frac{0.006}{2 \times 12} \times 57.3 = 1^\circ 30'52"
\]

\[
\frac{a}{2} = (23^\circ 35'33" + 1^\circ 30'52") = 25^\circ 6'25"
\]

\[
\tan \frac{a'}{2} = \tan 25^\circ 6'25" \times \cos 33^\circ 33' = 0.39052
\]

\[
\frac{a'}{2} = 21^\circ 19'55"
\]

\[
s = 12 \sin 1^\circ 30'52" = 0.3172 \text{ inch}
\]

\[
s_n = 0.3172 \cos 33^\circ 33' = 0.2643 \text{ inch}
\]

\[
R = \frac{0.2643}{2 \cos 21^\circ 19'55''} = 0.1419 \text{ inch}
\]

\[
D_o = 12 \cos 1^\circ 30'52" + 2 \times 0.1419 (1 + \sin 21^\circ 19'55'')
\]

\[
= 12.3828 \text{ inches}
\]
Clamping Elements

Classifications

Here, as in other cases within shop terminology, the word "clamp" has more than one meaning; it is the common designation for all devices by which a part is secured in a fixture against the acting forces and it is also the name for a specific type of holding device described as a "strap." While the most important phase in fixture design is the locating phase, the clamping phase takes its place as second in importance. Its technical importance lies in the fact that clamps must generate and direct the acting forces in such a way that the part is securely locked in place without suffering injury in the form of elastic distortion (springing). The economic aspect of clamping is just as significant, because clamping and releasing the part absorbs a portion, perhaps the largest portion, of the total operating time. Therefore, clamps must be designed for safe and fast operation.

To accomplish this, clamping devices have been developed in an extremely wide assortment of diverse types and details. Almost all of them are now commercially available and are, to some degree, standardized. The fixture designer's task is no longer to invent and design new clamping components, but to select the right type and size from those on the market. A comprehensive display of clamping components is found in Chapter 17 and the present discussion is confined to an explanation of principles, the description of representative applications, and references to details.

The basic types of clamping devices are: screws, straps, wedges, cams, toggles, and rack and pinions. Racks and pinions are discussed in Chapter 21. The action of most clamps is based on friction, and they are actuated either manually, pneumatically, or hydraulically. Predominantly, clamps are of the strap type; structurally, they are modifications of the simple beam.

The Mechanics of Wedges

The action of wedges, screw threads, and cams are all based on the same friction relations. A screw thread is a wedge rolled around a cylinder. A flat cam is a wedge folded around a circle. To clamp a workpiece by means of a wedge requires, in the simplest case, that one side of the wedge bears against the work surface, as shown in Fig. 10-1a, while the other side is supported by a surface in the fixture. With the coefficient of friction \( \mu \), the forces on the two sliding surfaces are \( P \) and \( \mu P \), and it requires the force shown as \( F_1 \) to insert the wedge. To calculate \( F_1 \) the forces \( P \) are resolved into their components with respect to the axis of the wedge, as shown in Fig. 10-1b. Equilibrium requires that the sum of all force components parallel to the wedge axis equals zero, from which we get

\[
F_1 = 2P \sin\frac{\alpha}{2} + 2\mu P \cos\frac{\alpha}{2}
\]

\[
= 2P \left( \sin\frac{\alpha}{2} + \mu \cos\frac{\alpha}{2} \right)
\]

To withdraw a wedge that holds the work with a pressure \( P \) requires a force \( F_2 \) found by

\[
F_2 = -2P \sin\frac{\alpha}{2} + 2\mu P \cos\frac{\alpha}{2}
\]

\[
= 2P \left( -\sin\frac{\alpha}{2} + \mu \cos\frac{\alpha}{2} \right)
\]

Since the parenthesis includes a minus term, it may become zero, which gives

\[
-\sin\frac{\alpha}{2} + \mu \cos\frac{\alpha}{2} = 0
\]

\[
\tan\frac{\alpha}{2} = \mu
\]
The result, $F_2 = 0$, means that the wedge is no longer self-locking. The condition for self-locking depends strongly on the coefficient of friction, but so do the values of $F_1$ and $F_2$. The coefficient of friction depends not only on the contacting surfaces, but also on the pressure ($\mu$ increases with increasing unit pressure) and the operating conditions. Everything in a machine shop carries an oil film (unless it has been recently degreased). This condition is assumed for clamping operations. While clamped under pressure the surfaces have a tendency to break through the film and “bite,” and the coefficient of friction for release is significantly higher. On the other hand, if exposed to vibration, the surfaces may work loose resulting in a reduced coefficient of friction. Recommended values of $\mu$ for wedges, cams and pivots and bearings are given in Table 10-1.

![Fig. 10-1. The mechanics of the wedge.](image)

**Table 10-1. Coefficients of Friction $\mu$ for Wedges, Cams, Pivots and Bearings Made of Hardened Steel**

<table>
<thead>
<tr>
<th>Wedge or Cam Acting on</th>
<th>Coefficient of Friction $\mu_1$</th>
<th>Coefficient of Friction $\mu_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Clamping</td>
<td>Release</td>
</tr>
<tr>
<td>Hardened steel</td>
<td>0.19 to 0.20</td>
<td>0.19 to 0.20</td>
</tr>
<tr>
<td>Machine steel</td>
<td>0.17 to 0.19</td>
<td>0.20</td>
</tr>
<tr>
<td>Cast iron</td>
<td>0.15 to 0.17</td>
<td>0.17 to 0.19</td>
</tr>
<tr>
<td>Aluminum alloy</td>
<td>0.17 to 0.18</td>
<td>0.18 to 0.20</td>
</tr>
<tr>
<td>Laminated plastic</td>
<td>0.12 to 0.16</td>
<td>0.15 to 0.18</td>
</tr>
<tr>
<td><strong>Pivots and bearings</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Good Condition</td>
<td>0.03 to 0.06</td>
<td>0.10 to 0.15</td>
</tr>
<tr>
<td>Neglected</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Self-locking wedges and cams are made with tapers from 1:20 ($\alpha = 2^\circ 52')$ through 1:10 ($\alpha = 5^\circ 44'$) and up to a wedge angle of 7 degrees (taper 1:8.18). For wedges exposed to vibrations it is recommended that the taper should not exceed 1:15 ($\alpha = 3^\circ 47'$). Wedges designed within these limits have a short operating range and are only used on parts with fairly close tolerances. When the wedge is positively held in place, larger wedge angles can be used. Wedges that actuate plungers or other movable components are made with at least 15 degrees (taper 1:3.8) and up to 45 degrees wedge angle (plungers with ends cut under 45 degrees).

*Example*—Assume a 7-degree wedge acting on a cast iron part with $\mu = 0.15$ for clamping and $\mu = 0.18$ for release.

$$F_1 = 2P \left(\sin 3\frac{1}{2}\circ + 0.15 \cos 3\frac{1}{2}\circ\right)$$

$$= 0.422P$$

$$P = \frac{1}{0.422} F_1 \approx 2 \frac{1}{3} F_1$$

The pressure on the part is, in this case, $\approx 2 \frac{1}{3}$ times the applied force.

$$F_2 = 2P \left(-\sin 3\frac{1}{2}\circ + 0.18 \cos 3\frac{1}{2}\circ\right)$$

$$= 0.237P$$

The force to release the part is, in this case, approximately one-half of the clamping force. The wedge is self-locking for coefficients of friction down to

$$\mu = \tan 3\frac{1}{2}\circ \approx 0.06$$
The Mechanics of Screws

When a screw is used as a clamping device, the clamping force \( P \) on the work, or on a strap, is developed by application of torque \( T \) to the head of the screw or to the nut (see Fig. 10-2). In addition, the torque must overcome friction on the thread and under the head. All forces can be calculated by use of the wedge formulas with the helix angle and the thread angle taken into account. Detailed analysis of the system is available from textbooks on machine elements. However, considerable simplification is possible, because the standardized screw thread systems have constant thread angle and nearly constant ratios between the significant dimensions (nominal, pitch, and root diameter, etc.). With a coefficient of friction of 0.15 as representative of average workshop conditions and nominal screw diameter \( D \), the formula connecting torque \( T \) and clamping pressure \( P \) is

\[
T = 0.2 \, D \, P \quad (\mu = 0.15)
\]

With well cleaned and lubricated screw threads, the coefficient of friction can go down to around 0.10, which is about the best that can be expected under workshop conditions. Values for \( T \) for various values of \( \mu \) are

\[
\begin{align*}
T &= 0.164 \, D \, P \quad (\mu = 0.12) \\
T &= 0.139 \, D \, P \quad (\mu = 0.10) \\
T &= 0.115 \, D \, P \quad (\mu = 0.08)
\end{align*}
\]

The formulas as written are valid for coarse threads. For fine threads \( T \) is 3 to 5 percent less; a rather surprising result. One would expect a greater difference, but the gain in mechanical advantage is partly absorbed by the increased friction losses. All standard UNF, UNC, and all of the machine screw series of screw threads are designed to be self-locking and no further analysis is required with respect to this point.

Manual Forces

The torque on a clamping device is produced by manual force. Mechanical clamping devices do not use screws but apply their pressure directly, or through a simple linkage.

The force, that can be provided by an operator, is not a constant quantity and cannot be calculated as are mechanically generated forces. Nevertheless, numerous attempts have been made to measure them; their average and mean values and their upper and lower limits. Most of these have been directed towards the manipulation of machinery controls in general and have little bearing on the actuation of fixture clamps with the exception of a few results of a general nature. Test results show acceptable correlations and normal distributions around a mean value; the maximum value is 3 to 4 times the minimum value, and the minimum value is 0.45 times the mean.

Data for two operations are needed in manual fixture design: to pull (or push) a lever with one hand, and to turn a knob. The lever can be a wrench, a handle on a nut, or a cam lever. From evaluation of laboratory data it appears that a mean value for pulling a lever with one hand is 90 to 110 pounds (400 to 490 N). Data of this nature, however, must be taken with some reservation. The force a man exerts during a laboratory test is not necessarily the same as that which he would use while on his job, where he (unknowingly) is influenced by his physical condition, his willingness, the frequency of the effort, his degree of fatigue, and his age. It was found in one case that the maximum one-hand pull was 125 pounds (556 N) for a 25-year-old man and 103 pounds (458 N) for a 60-year-old man. It is not known to what extent the equipment used in the various tests is representative of common shop equipment. The length of the lever is of importance because it affects the mode of the grip and the geometry of the motion. A rule-of-thumb, applicable to levers of up to 8 inches (200 mm) in length, gives 2.8 pounds per inch (5 N per cm) lever length as the average applied force in one-hand operations.
Laboratory results agree well on the significance of fatigue. The force applied repeatedly by one hand should not exceed 30 to 40 pounds (135 to 175 N). European design practice uses 145 to 175 newtons (33 to 40 pounds) as the force applied to a lever. American practice is to take 30 pounds (135 N) for calculating the operating clamping force, but to calculate dimensions for an occasional maximum force of 90 pounds (400 N).

The turning of a knob is clearly a one-hand operation. The torque that can be exerted depends on the shape and dimensions of the knob; the maximum torque in repeated application can be taken as 47.5 inch-pounds per inch (211.3 N mm per mm) knob diameter for round and four-lobe hand knobs.

The above are design values which means limiting values, and not necessarily operating values, and must be applied with due consideration of all facts including the dimensions of the equipment. No workman would use from 30 to 90 pounds on a wrench to tighten a 3/8-inch screw. The operator adjusts his force to the job at hand; he pulls the wrench until he “feels” that the screw is “tight.”

The results of a test program performed under actual shop conditions are listed in Table 10-2. The operator was instructed to “pull normally until the screw is tight.” Screw thrusts were measured by a load cell. Threads were clean and normally lubricated. Data of this type are closer to reality and more reliable for use in design than data based on more or less uncertain assumptions. It is interesting to note the drop in thrust from the 5/8 inch- to the 3/4-inch screw. It reflects the combined effect of larger pitch, lower mechanical advantage, more friction loss, and greater physical effort required of the operator.

### Table 10-2. Clamping Force from Hand-Operated Screws—Experimental Results

<table>
<thead>
<tr>
<th>Screw Thread NC</th>
<th>Screw Turned by</th>
<th>Hand Knob</th>
<th>Wrench</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Clamping Force, Pounds</td>
</tr>
<tr>
<td>1/4 – 20</td>
<td></td>
<td>300</td>
<td>2200</td>
</tr>
<tr>
<td>5/16 – 18</td>
<td></td>
<td>400</td>
<td>2500</td>
</tr>
<tr>
<td>3/8 – 16</td>
<td></td>
<td>700</td>
<td>3000</td>
</tr>
<tr>
<td>1/2 – 13</td>
<td></td>
<td>900</td>
<td>6800</td>
</tr>
<tr>
<td>5/8 – 11</td>
<td></td>
<td>1300</td>
<td>7800</td>
</tr>
<tr>
<td>3/4 – 10</td>
<td></td>
<td>1200</td>
<td>7500</td>
</tr>
</tbody>
</table>

 according to the cam geometry and as a function of $\theta$. The surface and lubrication conditions at $D$ and on the pivot are different and are reflected by the two different coefficients of friction $\mu_1$ and $\mu_2$ ($\mu_1 > \mu_2$; see Table 10-1).

### The Mechanics of Cams

A cam may be thought of as a wedge folded around a cylinder. Figure 10-3 shows a cam actuated by a force $F$, on a handle of length $L$, measured from the pivot center $C$, and exerting a clamping force $P$ and a frictional force $\mu_1 P$ at the point of contact $D$. These forces generate a reaction $R$ with a frictional force $\mu_2 R$ on the pivot. Any point on the cam contour is defined by radius vector $r$ and position angle $\theta$ measured from the low point $A$. The location of $D$, relative to $C$, is defined by the initial eccentricity $e$ and the height $H$, which vary

---

1 Tomco, Inc., Racine, Wisconsin.
Assume, for a moment, that there is no friction; the acting forces are now \( F_o \) and \( P \); the reaction \( R \) does not enter into the equilibrium equation which is

\[
F_o L = Pe
\]

\[
\frac{P}{F_o} = \frac{L}{e}
\]

The pressure \( P \) exists only as long as the force \( F_o \) is maintained. When \( F_o \) is removed, \( P \) subsides. This situation is unusable and unrealistic. Frictions are always present. Their effect is to drastically increase \( F \) for a given or required \( P \) and to provide the possibility for self-locking.

With friction, the equilibrium equation for the clamping phase is

\[
P e + \mu_1 P H + \mu_2 R r_2 = FL
\]

\( R \) is composed of contributions from \( P \) and \( F \). Cam devices are designed so that \( P \) is much greater than \( F \) (10 to 20 times) and dominates the value of \( R \) which, as a good working approximation, can be taken as 1.03 \( P \). Then

\[
P (e + \mu_1 H + 1.03 \mu_2 r_2) = FL
\]

\[
\frac{P}{F} = \frac{e + \mu_1 H + 1.03 \mu_2 r_2}{L}
\]

With the assumption that the cam is clamped in a self-locking position with the force \( P \), it is now desired to release the cam by applying a force \( F' \) to the handle. This means that \( F \) changes direction and so do the friction forces \( \mu_1 P \) and \( \mu_2 R \). By changing sign for these terms in the equation it becomes

\[
P e - \mu_1 P H - \mu_2 R r_2 = -F'L
\]

\[
\frac{F'}{P} = \frac{-e + \mu_1 H + 1.03 \mu_2 r_2}{L}
\]

As long as the equation gives a positive \( F' \) the cam is self-locking. The theoretical limit for self-locking is when \( F' = 0 \), which makes \( R = P \) and gives

\[
-e + \mu_1 H + \mu_2 r_2 = 0
\]

\[
\frac{\mu_1 H + \mu_2 r_2}{e} = 1
\]

as the theoretical condition for self-locking. For any practical application a safety factor \( (FS) \) is required, defined by

\[
\frac{\mu_1 H + \mu_2 r_2}{e} = (FS)
\]

where \( (FS) \) is taken as 1.5 to 2. For \( \mu_1 \) it is recommended to use the lower range of values (column "clamping" in Table 10-1). The friction losses are evaluated by the cam efficiency

\[
\frac{F_o}{F} = \frac{e}{e + \mu_1 H + 1.03 \mu_2 r_2}
\]

The cam contour has five points of interest; the low point \( A \), the high point \( B \) (dead center), the lower and upper limits \( E \) and \( G \), and the midpoint \( M \), of the operating range. The upper limit \( G \) must never get close to \( B \) because of the danger of the cam snapping through in case of overload. This danger increases as the cam and its pivots and bearings wear. Safety against snap-through is obtained by proper selection of the location of \( M \) and the length of the operating range \( EG \), as measured on the contour, or expressed as \( 2\theta \) in angular measure. Recommended values for \( \theta \) are from 30 to 45 degrees.

**Example**—A cam clamp is designed with the following values:

**In English Units**

\[
L = 5 \text{ inches} \quad r_2 = 0.25 \text{ inch}
\]

\[
H = 0.84 \text{ inch} \quad e = 0.081 \text{ inch}
\]

\[
\mu_1 = 0.18 \quad \mu_2 = 0.05 \quad \text{required } P = 600 \text{ pounds}
\]

\[
\frac{600}{F} = \frac{5}{0.081 + (0.18 \times 0.84) + (1.03 \times 0.05 \times 0.25)}
\]

\[
F = 29.4 \text{ pounds}
\]

\[
(FS) = \frac{(0.18 \times 0.84) + (0.05 \times 0.25)}{0.081} = 2.02
\]

efficiency =

\[
\frac{0.081}{0.081 + (0.18 \times 0.84) + (1.03 \times 0.05 \times 0.25)} \times 100 = 33.0\%
\]
In SI (Metric) Units

\[ L = 125\text{mm} \quad r_2 = 6\text{mm} \]
\[ H = 21\text{mm} \quad e = 2\text{mm} \]
\[ \mu_1 = 0.18, \mu_2 = 0.05, \text{ required } P = 2670\text{N} \]
\[
\frac{2670N}{F} = \frac{125}{2 + (0.18 \times 21) + (1.03 \times 0.05 \times 6)}
\]
\[ 125F = 2670 \times (2 + 3.78 + 0.31) \]
\[ 125F = 16260 \]
\[ F = 130\text{N} \]
\[
FS = \frac{(0.18 \times 21) + (0.05 \times 6)}{2} = 2.04
\]

Efficiency =
\[
\frac{2}{2 + (0.18 \times 21) + (1.03 \times 0.05 \times 6)} \times 100
\]
\[ = 32.8\% \]

**Spiral Cams**

Fixture clamping cams are formed as Archimedes spirals or as circular eccentrics.

The fundamental equation for an Archimedes spiral (see Fig. 10-4) is
\[ r = \frac{l}{2\pi \theta} \]

where the angle \( \theta \) is in radians, and \( l \) is the lead, that is, the increase in radius vector \( r \) for one revolution.

![Fig. 10-4. The mechanics of the Archimedes spiral cam.](image)

(The angle equivalent of one revolution is \( \theta = 2\pi \) radians which, when substituted in the above equation, gives \( r = l \). The rise is the increase in \( r \) along a given length of contour. The rise over the entire operating range is the "throw.")

The slope of the tangent and the normal to the curve is determined by
\[ \tan \alpha = \frac{dr}{rd\theta} = \frac{l}{2\pi r} \]

The angle \( \alpha \) is equivalent to a wedge angle. While \( l \) is constant, \( r \) and \( \alpha \) vary with \( \theta \); \( \alpha \) increases with \( l \) and decreases with increasing \( r \). For fixture clamping cams the variation of the angle \( \alpha \) over the useful length of the cam is small, usually only around 1 degree. It is not difficult to make the cam self-locking over a wide operating range. There is no low or high point on a spiral.

For practical applications it is convenient to measure \( \theta \) from a fixed point \( A \) with radius vector \( r_0 \). This modifies the equation to
\[ r = r_0 + \frac{l}{2\pi \theta} \]

At an arbitrary point \( D \) we have
\[ e = r \sin \alpha \]
\[ H = r \cos \alpha \]

There are several empirical recommendations for the selection of the lead \( l \) for fixture clamping cams. One such recommendation says 0.001 inch (0.025 mm) per degree per inch radius which, on a 2-inch (50-mm) cam radius, gives \( l = 0.72 \) inch (18.3 mm) and \( \tan \alpha = 0.0573 \). This cam is self-locking for \( \mu = 0.06 \). Another recommendation says that the throw over 90 degrees shall be 1/6 times the radius. This gives \( l = 2/3r \) and, when applied to a 1 1/2-inch (38-mm) radius, \( \tan \alpha = 0.106 \). This cam is intended to be self-locking for \( \mu = 0.1 \).

**Eccentric Cams**

The circular eccentric cam (see Fig. 10-5) is characterized by the eccentric radius \( R \) and the fixed eccentricity \( E \). This cam has low and high points \( A \) and \( B \). The point of contact \( D \), is defined by \( r \) and \( \theta \) as before, height \( H \) and eccentricity \( e \) have, likewise, the same meaning. For calculating \( H \) and \( e \) it is convenient first to find angle \( \psi \), which is done by

\[
\sin \psi = \frac{E}{R} \sin (180 - \theta)
\]
\[ \sin \psi = \frac{E}{R} \sin \theta \]
An auxiliary angle $\phi$ is needed temporarily; it is found by

$$90 + \phi + (180 - \theta) + \psi = 180$$

$$\phi = \theta - \psi - 90$$

Then

$$H - R = E \sin \phi = -E \cos (\theta - \psi)$$

$$H = R - E \cos (\theta - \psi)$$

$$e = E \cos \phi = E \sin (\theta - \psi)$$

It should be noted that $e$ is independent of $R$ and depends only on the fixed eccentricity $E$ and the angle $\theta - \psi$ is the cam rotation angle. $\psi$ is always a small angle and $\psi = 0$ degrees at $\theta = 0$ degrees and $\theta = 180$ degrees. $e$ is zero at 0 degree and 180 degree cam rotation and has a maximum $e = E$ at 90-degree cam rotation. $\psi$ has here its maximum value determined by

$$\sin \psi = \frac{E}{R}$$

Since $\tan \psi$ is the slope of the tangent, it follows that the cam is least likely to be self-locking in this position, or, if it is self-locking in this position, it is always self-locking.

For $\mu = 0.1$ a cam is self-locking for $\frac{E}{R} = 0.09961$ or $\frac{2R}{E} = 20.8$. The preferred operating range is symmetrical with respect to the position for maximum $e$. The self-locking range for other values of $\frac{2R}{E}$ is listed in Table 10-3. The location of these ranges is shown in Fig. 10-6.

![Fig. 10-5. The mechanics of the eccentric cam.](image)

![Fig. 10-6. The location of self-locking ranges on an eccentric cam.](image)

<table>
<thead>
<tr>
<th>$D$</th>
<th>Self-Locking Operating Range Angle $\theta$, deg.</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>0 to 17, 163 to 180</td>
</tr>
<tr>
<td>8</td>
<td>0 to 23, 157 to 180</td>
</tr>
<tr>
<td>10</td>
<td>0 to 30, 150 to 180</td>
</tr>
<tr>
<td>12</td>
<td>0 to 37, 143 to 180</td>
</tr>
<tr>
<td>14</td>
<td>0 to 44, 136 to 180</td>
</tr>
<tr>
<td>16</td>
<td>0 to 53, 127 to 180</td>
</tr>
<tr>
<td>18</td>
<td>0 to 64, 116 to 180</td>
</tr>
<tr>
<td>20</td>
<td>0 to 85, 95 to 180</td>
</tr>
<tr>
<td>20.08</td>
<td>0 to 180</td>
</tr>
</tbody>
</table>

The Mechanics of Toggle Clamps

Toggle clamps are linkage operated clamps and are based on the same kinematic principle as the eccentric clamp but with widely different dimensions of the moving parts. They possess enough elastic flexibility to allow the actuated link to pass through dead center. A positive stop just beyond that point defines the locking position and the resistance at dead center secures the link in that position.

At dead center the initial eccentricity $e$ equals zero and, in the absence of friction, the mechanical advantage equals infinity. This holds for any eccentric cam, including toggle clamps. Mathematically
this means that the clamping pressure $P$ becomes infinity for any finite value of the actuating force $F$, or conversely, that a finite clamping pressure $P$ can be generated by an infinitely small actuating force $F$. In reality, we have neither infinitely large nor infinitely small forces, and the mathematical model

---

**Fig. 10-7.** The action of the toggle clamp.  
*a.* A toggle clamp of the push-pull type in the open and closed position;  
*b.* The force system in the toggle clamp at the moment of clamping.
means simply that a small (but finite) actuating force \( F \) can generate a large (and still finite) clamping pressure \( P \), as indicated in Fig. 10-7a.

The technical interpretation of these seemingly paradoxical statements is, that a perfectly rigid eccen-
tric cam actuated by a finite force can operate only to the point of positive contact with the part to be clamped. The clamping is effective if this point is near dead center. If it is at dead center, then the clamping can only be effective when all di-

ensions in the system are mathematically accurate; a completely unrealistic assumption. In mechanical language, the system is statically indeterminate; the height of the part to be clamped is the redundant dimension.

Actually, there are only two possibilities, either the cam clamps before dead center or it slips through. Togglo clamps are not designed with the same rigid-

ity as solid cams, and their inherent flexibility dra-

stically changes the force conditions during the clamping operation. After contact has been established (a short distance before dead center) pressure builds up and the entire toggle mechanism is elas-
tically deflected. The maximum deflection and corres-

ponding maximum pressure occur simultaneously as the actu-
ating link passes through dead center; the pres-

sure is regulated by means of an adjusting screw in the pressure pad or elsewhere in the linkage system. The actuating force required for moving the link from the point of initial contact to dead center is small, because the mechanical advantage is large; on the dead center, where the mechanical advantage is infinity, the actuating force would vanish if the mechanism were frictionless. The actuating force necessary on dead center is the force required to over-

come the frictional forces on the pivots and bear-

ings, and is calculated as follows: Maximum pressure \( P \) produces friction forces as shown in Fig. 10-7b. Since \( P \) is many times \( F \) (example: \( F = 15 \) pounds, \( P = 1000 \) to 2000 pounds), it is permissible to ignore any transverse reactions from \( F \). The actuating force is transmitted to the pressure link by a direct force \( F_1 \), so far unknown, and a friction force. First, consider the forces on the pressure link. Taking moments about the center of the right-hand pin gives

\[
F_1 B + \mu P (B - R) = \mu P R
\]

\[
F_1 B = \mu P (2R - B)
\]

\( F_1 \) may well come out negative. This is not disturbing; it means simply that the friction force is significant in the transmission of the actuating force. Now, consider the forces on the operating lever. Taking moments about the center of its bearing pin gives

\[
FL = F_1 A + \mu PR + \mu P (A + R)
\]

\[
= 2 \mu PR \frac{A + B}{B} = 2 \mu PR \left( \frac{A}{B} + 1 \right)
\]

Note that the ratio of \( F \) and \( P \) does not depend on the individual lengths \( A \) and \( B \), only on their ratio.

In English Units

**Example**—A toggle clamp has: \( L = 12 \) inches, \( A = 3/4 \) inch, \( B = 2 \) inches and \( R = 1/4 \) inch. Assume \( \mu = 0.15 \). Then

\[
F \times 12 = 2 \times 0.15 \times P \times 1/4 \times (3/8 + 1)
\]

\[
\frac{P}{F} = 116
\]

In SI (Metric) Units

**Example**—A toggle clamp has: \( L = 300 \) mm, \( A = 19 \) mm, \( B = 50 \) mm and \( D = 6 \) mm. Assume \( \mu = 0.15 \). Then

\[
F \times 300 = 2 \times 0.15 \times P \times 6 \times \frac{19 + 50}{50}
\]

\[
\frac{P}{F} = 120.8 \approx 121
\]

The Mechanisms of Beams

Straps are beams and are loaded in bending. The loads are the applied force \( F \), the clamping force \( P \) and a reaction \( R \) at the point of support. The application of straight straps as clamping elements in fixtures includes the five different force arrangements shown in Fig. 10-8, a through e. The angle strap is shown in f.

In the design and stress analysis of a strap clamp it can be assumed that \( P \) is known, and it is required to calculate the applied force \( F \), and the maximum bending moment \( M \), which always occurs at the load that is located in the middle part of the strap. The formulas for \( F \) and \( M \) are:

Case (a)

\[
\frac{P}{F} = \frac{L_1}{L_2}
\]

\[
M = RL_1 = P(L_2 - L_1) = F \frac{L_1(L_2 - L_1)}{L_2}
\]
Fig. 10-8. The mechanics of the beam type strap and angle clamp. Applied force: \( F, F_1, F_2 \); clamping pressure: \( P \); support reaction: \( R, R_1, R_2 \).

Case (b)
\[
\frac{P}{F} = 1 \quad M = F \frac{L_1 L_2}{L_1 + L_2}
\]

Case (c)
\[
\frac{P}{F_1 + F_2} = 1 \\
M = P \frac{L_1 L_2}{L_1 + L_2} = F_1 L_1 = F_2 L_2
\]

Case (d)
\[
\frac{P}{F} = \frac{L_1}{L_2} \quad M = FL_1 = PL_2
\]

Case (e)
\[
\frac{P}{F} = \frac{L_1}{L_2} \\
M = F(L_1 - L_2) = P \frac{(L_1 - L_2)L_2}{L_1}
\]
When comparative = PL = 0.06 = possibility; 0.16 = based = 1.50 = 0.11 = 0.05 = Time, 0.16 = 1, It Contact 0.02 = the = the = 0.09 = They = mechanical = Ott = They = minutes = only (disregarding = the = 0.07 = 20x671 Ch. 21x524 <e), 22x546 = Cases = common = actuated. All F = and = advantage = than = the = 22x458 = Economics = clamps = The = to = working = pur = chasing) = is = for = screw = of = 25 = variables = should = close = as = close to P as possible, and in cases (d) and (e), P should be as close to R as possible.

Economics

The clamping devices used in connection with jigs and fixtures may either clamp the work to the jig or the jig to the work, but very frequently the clamps simply hold a loose or movable part in place in the jig, the part can then be swung out of the way to facilitate removing and inserting work in the jig. The work, in turn, is clamped by a set-screw or other means passing through the loose part, commonly called the “leaf.”

Most clamping devices have some basic features in common: 1. They are made from high-strength material; 2. Contact surfaces are made wear-resistant; 3. They are designed for quick operation; and 4. When released they can be moved clear of the working area.

When making a selection between two or more different types of clamp the fixture designer must weigh savings in operating time against fabricating (or purchasing) cost. Time-study data are valuable for this purpose, if available; if not, comparative estimates can be based on average empirical values for the common clamping operations. A list of such values is given in Table 10-4. Actuating a valve or a switch for an air or hydraulically powered clamp takes about 25 percent of the time required for manual clamping of the same part. Cams are faster to operate than screw clamps, but are also more expensive. Cams are preferred for large series and short operations. With operations lasting more than 5 minutes, the advantages of the cam clamp become insignificant.

<table>
<thead>
<tr>
<th>Type of Clamp Operated</th>
<th>Time, Minutes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Socket head screw</td>
<td>0.22</td>
</tr>
<tr>
<td>Hexagonal nut or set-screw</td>
<td>0.16</td>
</tr>
<tr>
<td>Screw jack</td>
<td>0.16</td>
</tr>
<tr>
<td>Bar knob, bar screw</td>
<td>0.16</td>
</tr>
<tr>
<td>Spoke nut</td>
<td>0.12</td>
</tr>
<tr>
<td>Hand wheel</td>
<td>0.11</td>
</tr>
<tr>
<td>Backup screw (on a strap)</td>
<td>0.10</td>
</tr>
<tr>
<td>Star-shape or other form of hand knob</td>
<td>0.09</td>
</tr>
<tr>
<td>Fixed jack</td>
<td>0.09</td>
</tr>
<tr>
<td>Hand-operated cam</td>
<td>0.07</td>
</tr>
<tr>
<td>C-washer</td>
<td>0.06</td>
</tr>
<tr>
<td>Sliding strap and other quick-acting clamps of various design</td>
<td>0.02 to 0.04</td>
</tr>
<tr>
<td>L- and T-pins (for locating through a hole) 1/2-inch diameter, and up</td>
<td>0.09</td>
</tr>
<tr>
<td>less than 1/2-inch diameter</td>
<td>0.05</td>
</tr>
</tbody>
</table>

**Example**—An operation of 5 minutes duration requires either a screw clamp costing $12.00 or a cam clamp costing $50.00. Clamping time for the screw clamp is 0.16 minute, for the cam clamp, 0.07 minute. Release time is 50 percent of clamping time. Labor plus overhead is $10.00 per hour.

Higher cost of cam: $50.00 - $12.00 = $38.00

Savings in time per part with cam operation:

\[(0.16 - 0.07) \times 1.50 = 0.135 \text{ minute}\]

Savings per day in $ (disregarding losses, “breaks,” etc.):

\[
\frac{8 \times 60 \times 0.135}{60} \times 10.00 = 2.16
\]

Time required for break-even:

\[
\frac{38.00}{2.16} = 17.6 \approx 18 \text{ working days}
\]

Production required for break-even:

\[
18 \times 8 \times \frac{60}{5} = 1728 \text{ parts}
\]

**Clamping Screws**

Clamping fasteners (screws and nuts) fall into two groups; fasteners turned with a wrench, and hand-tightened fasteners. Fasteners for wrench operation
are set-screws with hexagonal head, collar-head (square-head) screws, and socket-head screws. Nuts used are hexagonal nuts, hexagonal flanged nuts, and acorn nuts. Collars, flanges, and washers are used to spread and reduce the pressure. Spherical washers are used to equalize the pressure when clamping on irregular contacting surfaces. Acorn nuts are used to protect the thread against damage and dirt. The height of hexagonal nuts should be 1 1/2 times standard height to reduce the load on the thread.

The acorn nut can be modified as shown in Fig. 10-9. The purpose of this design is to permit lifting the wrench off the "hex," and moving it back for a new grip. The round part of the nut serves to keep the wrench in place to be slipped back onto the hexagon nut, while the pin at the top of the nut makes the wrench an integral part of the fixture so that it cannot get lost.

![Fig. 10-9. Acorn nut for clamping screw, modified for retention of wrench.](image)

Machine steel of 60,000 psi (414 N/mm²) tensile strength is satisfactory in cases where the load is light; however, most fasteners for clamping purposes are made from steel with 120,000 to 150,000 psi (830 to 1035 N/mm²) tensile strength. Socket-head screws, are usually made from a 185,000 psi (1275 N/mm²) tensile strength steel. They are now widely preferred because of their strength and reliability, the fact that they require less space than older types of fasteners, and because they are safer in operation; the wrench cannot slip when it is properly inserted in its socket.

Hand-tightened fasteners are designed to provide a good grip for the hand and at the same time eliminate the need for a wrench, which could result in overloading the clamp or the part. Simple and inexpensive hand-grip fasteners that do not require purchase of commercial items are shown in Fig. 10-10.

A special type of bolt used in drill jigs is the "hook" bolt shown at left in the illustration. It is very cheap to make and easily applied. The bolt A, passes through a hole in the jig (having a good sliding fit in this hole) and is pushed up until the hook or head B, bears against the work; the nut is then tightened. When great pressure is not required, a thumb or wing nut provides a better grip and more satisfactory means than the knurled nut (shown at right), for tightening down upon the work, and permits the hook-bolt to be applied more readily. When work is removed from the jig, using the hook-bolt clamping device, the nut is loosened and the head, or hook, of the bolt is turned away from the work, thus allowing the workpiece to be taken out and another placed in position. Figure 10-11 shows an application of a bent hook-bolt. Generally speaking, the type shown in the previous illustration is better suited to its purpose, as the bearing point on the work is closer to the bolt body and it can be drawn more tightly.

![Fig. 10-11. Clamping with hook bolts.](image)

A screw with a pin through the head can be used for light clamping, but is inconvenient to work with. More convenient is the knurled circular nut shown previously, and also screws with knobs of the same type. Because of the better grip obtained, greater force can be applied by the use of wing-nuts, wing screws, and hand knobs of various shapes, used as screw heads and nuts. An application is shown in Fig. 10-12. These fasteners are all commercially available. Hand knobs come with 3, 4, and 5 lobes and in a range of sizes. The amount of force which the operator can apply to the knob increases with its size. With respect to shape, test results show that
Ch. 10 CLAMPING ELEMENTS

Fig. 10-12. Clamping screw with hand knob.

the 4-prong knob best fits the anatomy of the human hand.

If screws are to be firmly tightened without the use of a wrench, the method of using a pin through the screw-head can be used on large fixtures. The pin is 1/2 inch (13 mm) in diameter, or more, and requires the use of both hands (see Fig. 10-13). A more sophisticated version is the speed nut, a commercial component consisting of a nut with one or two arms terminating in ball knobs. This is for two-hand operation and permits fast spinning and hard tightening.

Fig. 10-13. Clamping screw with pin handle.

The use of manually operated screws, however, does not completely eliminate the possibility of overloading the clamp. Full safety against overload can be achieved by the use of torque head screws. The screw has a knurled head with a built-in spring-loaded clutch that slips automatically when the maximum load is applied. Many screw fasteners are, or can be, provided with swiveling pads for even pressure distribution and protection of the surface of the part.

Examples of the direct application of screws to substitute for the use of clamps are shown in Fig. 10-14. The method shown in diagram a is simple and self-explanatory. The method shown in diagram b requires screws with conical tips and is used in milling fixtures for light milling operations; the part is clamped horizontally and vertically at the same time. The method shown in diagram c is not recom-

mended for general use as the screws are severely loaded in bending, but as a last resort, it may be acceptable in cases where only light cuts are taken.

Fig. 10-14. Direct clamping with screws: a. The general method; b. An inexpensive method suitable for light-duty operations; c. A clamping method suitable for light-duty only—not recommended for general use.

Jack Screws

The ordinary jack screw is frequently employed as a supporting device in ordinary setups on a machine tool, but rarely in fixtures as it is a loose part and is quite apt to get lost. In Fig. 10-15 two simple devices are shown, that work on the same principle as the jack screw, but they have the advantage of

Fig. 10-15. Swing jack screws.
being connected to the jig by pin B. At A, a set-screw screws directly into the end of the eye-bolt, and at C, a long square nut is threaded on the eye-bolt. These nuts must be of a particular length, and made especially for this purpose. The eye-bolts are fastened directly to the wall of the jig, and the set-screw, or nut, is tightened against the work. The eye-bolt can be set at different angles to suit the work, thereby providing a means of double adjustment. This is a very convenient clamping device which works satisfactorily and can be easily swung out of the way. Several types of jack screws for permanent mounting in fixtures are commercially available.

Straps

The simplest form of clamping device is the strap. A number of different forms of commonly used straps are shown in Figs. 10-16 and 10-17. Perhaps the most common of all is that shown in diagram 10-16a. It is simple, cheap to make, and, for most purposes, it gives satisfactory service. The clamp shown in diagram 10-16b is practically the same principle, but with several added improvements. It is recessed at the bottom for a distance b, to a depth equal to a, so as to give a bearing only on the two extreme ends of the clamp. Even if the strap should bend somewhat because of the pressure of the screw, it would be certain to bear at the ends and exert the required pressure on the object being clamped. The strap is also provided with a ridge at D, located centrally with the hole for the screw. This insures an even bearing of the screw-head on the clamp, even if the two bearing points at each end of the clamp should vary in height, as illustrated in Fig. 10-17, left. The clamp at a in Fig. 10-16 would not bind very securely under such circumstances, and the collar of the screw might break off as the entire strain, when tightening the screw, would be put on one side.

A further improvement in the construction of this clamp may be had by rounding the underside of the clamping points A (Fig. 10-17, right). When a clamp with such rounded clamping points is placed in a tilted position it will bind the object to be held fully as firmly as if the two clamping surfaces were in the same plane.

The hole in these straps is very often elongated, as indicated in Figs. 10-16 a and b by the dotted lines, which allows the strap to be pulled back far enough to clear the work; making it easier to insert and remove the piece to be held in the jig. In some cases, it is necessary to extend the elongated hole, as shown in Fig. 10-16c, so that it becomes a slot (going all the way through to the end of the clamp) rather than simply an oblong hole. Aside from this difference, the clamp works on exactly the same principle as those previously shown.

To suit different conditions, instead of having the strap or clamp bear on only two points, it is sometimes necessary to have it bear on three points, in which case it may be designed similarly to the strap shown in Fig. 10-16d. In order to get equal pressure on all three points, a special screw, with a half-spherical head may be used to advantage. The head fits into a concave recess of the same shape in the strap. When the bearing for the screw-head is made in this manner, the hole through the clamp must have a generous clearance for the body of the bolt.

When designing clamps or straps of the types shown, one of the most important considerations is to provide enough metal around the holes, so that the strap will stand the pressure of the screw without breaking at the weakest place, which, naturally, is in a line through the center of the hole. As a rule, these straps are made of machine steel, although large clamps may occasionally be made of cast iron.
At e and f in Fig. 10-16, bent clamps and their application to the workpiece are shown. These clamps are commonly used for clamping work in the planer and milling machine, but are also frequently used in jig and fixture design as well.

Screws used for clamping these straps are either standard hexagonal screws, standard collar-head screws, or socket-head screws. When it is unnecessary to tighten the screws very firmly, thumb-screws or screws with hand knobs are frequently used, especially on small jigs as explained earlier.

The simple strap clamp is now commercially available in a large number of very sophisticated modifications, comprising single- and double-end straps, center and rear-end applications of the clamping screw, and fixed and adjustable height end support. Without exception the strap has an elongated hole to permit withdrawal from the work area, and a spring to keep it in the lifted position when released. Strap clamps are also made with cam actuation instead of screw actuation for quick clamping, and also with automatic withdrawal from the work area.

**Strap Clamp Applications**

A simple and common type of fixture requires clamping with one or two screws only at the center of the part, combined with easy access to the work area from above. The solution is a clamping screw (or screws) located at the center of a removable strap clamp, straddling the part. The strap, or clamp, is arranged as shown in Fig. 10-18, the screw passing through it at the center and bearing upon the work, either directly or through the medium of a collar or a swiveling pad, fitted to the end of the clamping screw. This type of clamping arrangement is commonly used for holding work in a drill jig. The strap used in this type of arrangement can be improved upon by making it in one of the forms shown in Fig. 10-19. Here the ends of the straps are slotted in various ways, to make it easy to remove the strap quickly, when the work is to be taken out of the jig.

![Fig. 10-19. Strap clamp with slots for easy removal.](image)

Another way of making the strap removable is to support it in grooves in the fixture wall so that it can slide when the clamping screw is released. Two ways of doing this are shown in Fig. 10-20. Shown in Fig. 10-20a is a strap that can slide lengthwise through slots in the fixture walls; the slots must have clearances below the strap to allow for passage of the screw. In Fig. 10-20b is a strap that slides in grooves in the fixture walls.

![Fig. 10-20. Strap clamps, sliding, for easy removal.](image)

The clamping pressure can be transmitted to the center of the part by means of the strap, with or without a pressure pad. Examples are shown in Fig. 10-21. Diagram a, shows a strap with slots for easy removal. The strap shown in diagram b, is clamped
by means of swinging bolts, a principle that has numerous applications. The strap shown in diagram c, is clamped by means of bolts with C-washers under the nuts and large bolt holes. When the nuts are released, the C-washers are easily removed, and the nuts can pass through the holes. The C-washers as shown are loose pieces. Loose pieces in a fixture are not desirable as they may be lost. In the present case this condition can be improved by the use of swinging C-washers. The C-washer is pivoted around a shoulder screw; when the nut is released, the C-washer is rotated out of engagement and the bolt can be withdrawn. Swinging C-washers with their pivot screws are standardized and commercially available.

Angular Clamps

Angular clamps are those that redirect the clamping force. In most cases, the force direction is rotated 90 degrees so that, for example, a vertical clamping screw generates a horizontal clamping force on the part. The clamping force is "turned around a corner." Some of these devices have the double effect of simultaneously clamping vertically and horizontally. Angular clamps are primarily necessitated by narrow space and restricted access conditions in the fixture.

The basic, and simplest, form for an angular clamp is the angle strap shown in Fig. 10-8f. Mechanically, it is a bell-crank lever. There are several other means by which the clamping force can be redirected 90 degrees or some other angle. The most important of these are the hinged strap (which is a modified version of the bell-crank lever), the linked strap, the combination of a strap with a wedge, and the combination of a strap with a plunger with a 45 degree inclined end surface. A widely used clamping device, the "gripping dog" is, in effect, also a modification of the bell-crank lever. The number of possible combinations is too large for a systematic clas-
sification, but the principles and their application will be demonstrated by representative examples.

Irregularly shaped castings which must be machined often present no apparently good means of holding by ordinary gripping appliances for drilling, shaping, or milling. In such cases, gripping dogs, as illustrated in Fig. 10-22, may be used. The basic type is the one shown in diagram a. The base block C is inserted in the T-slot of the machine table with a sliding fit and is prevented from slipping backwards by the backstop F, which is firmly bolted in position. The base block is slotted to receive the jaw D which is fulcrumed on a cross-pin. In the tail of the dog a set-screw E is threaded. By turning this set-screw the jaw is caused to "bite" inward and downward at the same time, firmly gripping the casting and forcing it down on the table. Since the position of the backstop is adjustable, the same gripping dog can be used for castings of different sizes. A gripping dog can also be solidly mounted on a fixture plate, eliminating the need for a backstop.

Gripping dogs have a tendency to draw the work down firmly and forcefully onto the rest-pins, or stops, and are useful in all classes of fixtures. A different type is shown in diagram b. Care should be taken to see that the stop is pivoted above point A. Another, and more rigid, device is shown at c. Plunger A, carried in plunger B, is forced down against the 45-degree side of stop C, compressing spring D. A fixture that provides two clamps which exert a "down-and-in" pressure is illustrated in Fig. 10-23a. Slides B are equalized by strap C and ball-and-socket washers D and E. This fixture is useful for milling and profiling, as the clamps and stops are below the surface of the work. A modification of this fixture is shown in diagram b. It has two down-and-in equalized clamps for holding a round piece of bored work for a milling operation. Lever A is tapped to receive screw B, and the clamping pressure equalizes with lever C by means of rod D. Levers A and C impart a down-and-in pressure to plungers E. Equalization is necessary as the work is already centered on a circular locator. This fixture can be applied to flat work.

In the double-movement clamp shown in Fig. 10-24, clamp A is carried by hinge B, pivoted at C. Screw E gives clamp A a down-and-in movement by means of a 45-degree taper on the contoured and hardened block D, which is also milled off at F to

Fig. 10-23. Clamping with "down-and-in" pressure.

Fig. 10-24. A clamp with double movement.
give the clamp sufficient movement to remove the work.

The combined action of a strap and wedge simultaneously produces horizontal and vertical clamping. It is simple, strong, efficient, and fast, and has many applications. Figure 10-25 shows a fixture where two such straps are clamped by one screw, also resulting in a centralizing action. The wedge end of the strap can bear against a mating surface on the fixture, as shown in Fig. 10-26a, or the configuration can be reversed so that the wedge end bears against and clamps the part, as shown in Fig. 10-26b.

An example of the use of plungers is shown in Fig. 10-27. Plungers A and B are built into the strap and are actuated by means of screw C with hand-knob D. In this way it is possible to reach, with the end of the strap, an otherwise almost inaccessible place in the workpiece. Another example, Fig. 10-28, shows a small clamping device used when drilling rivet holes through beading A and plate B.

Steel bracket C is fastened by screws to the side of the fixture; the front face of the clamp bracket is used as a stop for the plate and the beading; and

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2E. Thaulow, Maskinarbejde (Copenhagen: G.E.C. Gad's Forlag, 1930) vol. II.
clamp $D$, with a small hole drilled in one end, is fitted loosely in the milled slot in the bracket. The set-screw is located a little higher than the hole in the clamp and, by a few turns of the screw, the clamp is brought down against the work and forces the beading up against the stop, ready for drilling.

**Swinging Leaves**

A "leaf" or a "swinging leaf" is a hinged cover on a box-shaped fixture. It is normally used on drill jigs. The elementary principles involved in the swinging-leaf clamping construction are shown in their simplest form in Fig. 10-29. Loose leaves which swing out, in order to permit the work to be inserted and removed, are usually constructed in some manner similar to that shown in Fig. 10-30 in which $A$ represents the leaf, being pivoted at $B$ and held by a pin at $C$, which goes through the two lugs on the jig wall and passes through the leaf, thus binding the leaf and allowing the tightening of set-screw $D$, which bears against the work. The holes in the lugs of the castings are lined with steel bushings in order to prevent the cast-iron holes from being worn out too soon by the constant withdrawal and insertion of the pin. This kind of leaf, when fitted in well, is rather expensive, but is used not only for binding but also for guiding purposes, making a convenient seat for the bushings. If leaves are fitted well into place, the bushings in the leaves will guide the cutting tools in a satisfactory manner.

Another method of clamping down the leaf is shown in Fig. 10-31, in which $A$ is a thumb-screw, screwed directly into wall $B$ of the jig, and holding leaf $C$ down, as indicated. The thumb-screw is a quarter-turn screw. To swing the leaf out, the screw is turned back about a quarter of a turn, so that the head of the screw stands in line with a slot in the leaf, which is both wide and long enough to permit the leaf to clear the head of the screw. This is a very rapid method of clamping, and is frequently used.

![Fig. 10-29. The general principle of the swinging leaf clamp.](image1)

![Fig. 10-30. Swing leaf clamp with center screw.](image2)

![Fig. 10-31. A quarter-turn screw used for locking a swinging leaf clamp.](image3)

![Fig. 10-32. An eye-bolt used for locking a swinging leaf clamp.](image4)
CLAMPING ELEMENTS

The threaded end of the eye-bolt is provided with a standard hexagon nut, a knurled-head nut, or a wing-nut, according to how firmly the nut must be tightened. When the leaf is to be disengaged, the nut is loosened just enough to clear the point at the end of the leaf, and the bolt is swung out around pin B, which is driven directly into lugs projecting from the jig wall; a slot being provided between the two lugs, as shown, so that the eye-bolt can swing out freely. At the opposite end, the leaves or loose parts of the jig swing around a pin, the detailed construction of this end being, most commonly, one of the three types shown in Fig. 10-33. It must be understood that to provide jigs with leaves of this character involves a great deal of work and expense, and they are used almost exclusively when one or more guide bushings are held in the leaf.

A hinged jig cover may also be conveniently held in place by a semiautomatic spring latch of the type shown in Fig. 10-34. The body of the jig is shown at A; the hinged cover at B. This cover swings on pivot C and drops onto latch D. In cases where the cover is merely used to carry bushings, a latch of this kind is entirely satisfactory, although it is not recommended for use on jigs where screws for hold-down the work are carried by the cover. To swing the cover clear of the work in the jig, latch D is pushed back in the direction of the arrow. After the cover has been raised, the latch springs back into place ready to catch the top of the cover automatically when it drops back onto the jig, requiring no attention from the operator.

When the leaf is used to transmit clamping pressure to the part, a latch-type lock is insufficient, and a screw-type lock is required. The quarter-turn screw (see Fig. 10-35) is used where the height of the part is within close tolerances so that no significant height adjustment is required. Should some height adjustment be necessary, such as in a case where the leaf clamps upon a rough surface, a clamping screw with a longer travel can be used. A typical arrangement, using a swinging bolt, is shown in Fig. 10-36. Since, in this case, the part requires two clamping points; the clamp is pivoted so that the total clamping force is equally divided between the
two points. Most swinging-leaf components, including locking devices, leaves, and boxes are commercially available.

'Clamping Mandrel-mounted Work

The "natural" and convenient way of holding small parts that have been bored through and faced on the ends, is to mount them on a mandrel and make the mandrel an integral part of the fixture. When in position, the part is clamped. The simplest way of clamping it is with a washer and nut on the free end of the mandrel. This is a cheap and reliable, but slow, operation, because the nut must be run off and on each time a new part is placed in the fixture. The time required for running the hexagon nut on and off is saved as shown in the design in Fig. 10-37, using a quarter-turn knob. Stud B has a flat milled on both sides of its threaded end portion. The slot in knob A slides on over this flat and a quarter turn clamps the work. If the variation in the length of the work is not too great, this makes a rapid clamping arrangement.

Figure 10-38 shows another means of clamping the same piece in which the variation in length of the work and the time required for turning the knob to match the flat on the stud has been considered. The slotted washer A and knob B are dropped over stud C; A is held against B, which can then be screwed up as freely as a solid knob. This can be used for a variety of bushings of various lengths; stud C being made to suit the longest piece of work. Using a square or Acme thread is recommended, since these have less tendency to tilt the nut than would a 60-degree thread.

Clamping parts of varying length on a mandrel.

Cams, Eccentrics and Toggle Clamps

Clamping cams, eccentrics and toggle joint clamps differ from clamping screws in that they are more expensive to buy or make; they are faster to operate; and they have only a short effective clamping range. For a cam, the effective clamping range is 1/8 inch (3 mm), for a toggle joint clamp it is 1/16 inch (1.5 mm). As these mechanisms close, they tend to exert a slight lateral movement to the contacting surface. They are, therefore, used primarily to operate on another clamping member, such as a strap or leaf, rather than to clamp directly on the part. There are extensive analyses of the relative merits of the spiral cam and the circular eccentric cam, and industry has made its choice: the commercially available cam components are, as a rule, made with circular eccentric cams. There is little principal difference between this type of cam and an eccentric shaft; a toggle joint can be considered to be an eccentric device with a very large eccentricity. In addition, a toggle mechanism is fairly elastic, and this feature enables the toggle clamp to move beyond dead center when closing.

Eccentric shafts are often used for moving and closing a clamping strap. In Fig. 10-39 two applications of the principle of the eccentric shaft are shown. In diagram a, the eccentric shaft A has a bearing at both ends; the eye-bolt B is connected to it at the center and is forced down when the eccentric shaft is turned, causing the two end points of clamp C to bear on the work. This clamping arrangement has a very rapid action with good results. The throw of the eccentric shaft may vary from 1/16 inch (1.5 mm) to about 1/4 inch (6 mm), depending upon the diameter of the shaft and the accuracy of the work. In cases where it is required that the clamp bear in the center, an arrangement such as that in diagram b may be used. Here the eccentric shaft A has a bearing in the center and eye-bolts B are connected to it at each end. As the eccentricity
is the same at both ends, the eye-bolts or connecting-rod will be pulled down evenly when lever C is turned, and strap D will get an even bearing on the work in the center. If the force of the clamping stress is required to be distributed equally at different points on the work, a yoke may be used in combination with the eccentric clamping device.

When it is essential to use strap D for locating purposes, guides, which are necessary for holding it in the required position, must be provided for the strap. These guiding arrangements may consist of rigid rods, ground and fitted into drilled and reamed holes in the strap, or square bars held firmly in the jig and fitted into square slots at the ends of the strap. The bars may also be round, and the slots at the ends of the strap half round, the principle in all cases remaining the same; but the more rigid the guiding arrangement, the more accurate the locating.

The ordinary eccentric lever works on the same principle as the eccentric rods described above. There are a great variety of eccentric clamping devices frequently used and commercially available in several different models. For convenient and efficient operation the cam or eccentric lever should be located so that it is actuated by a straight horizontal pull and rotated to its position of maximum mechanical advantage with little change in body position. At any position of the handle it must, however, have a finger clearance of at least 5/8 inch (16 mm).

Figure 10-40a shows a cam specially intended for clamping finished work. It is not advisable to use this type of lever on rough castings, as the castings may vary to such a degree that the cam or eccentric would require too great a throw for rigid clamping.

The extreme throw of the eccentric lever should, in general, not exceed 1/6 of the length of the radius of the eccentric arc, if the rise takes place during one-quarter of a complete turn of the lever. This would give an extreme throw of, say, 1/4 inch (6 mm) for a lever having 1 1/2 inches (38 mm) radius of the cam or eccentric. It is obvious that as the eccentric cam swivels about center A, the lever being connected to the jig with a stud or pin; face B of the cam, which is struck with the radius R from the center C, recedes, or approaches the side of the work, thereby releasing it from, or clamping it against, the bottom, or wall, of the jig. The lever for the eccentric may be placed in any direction, as indicated by the broken and unbroken lines. Another eccentric lever, is shown in diagram b. It is frequently used on small work, for holding down straps or leaves, or for pulling together two sliding pieces, or one sliding and one stationary part, which, in turn, hold the work. These sliding pieces may be V-blocks or some kind of jaws. The cam lever is attached to the jig body, the leaf, or the jaw by a pin through hole A. Hook B engages the stud or pin C, which is fastened in the opposite jaw, or part, to be clamped to the part in which this pin is fastened. The variety of design of eccentric cam levers is so great that it is impossible to show more than the principles, but those examples which are shown embody the underlying action of all the different designs.
Intermediate adjustable supports require a quick-acting, safe, and hand-operated locking device. A cam-operated locking device for that purpose is shown in Fig. 10-41. The actual support member is the plunger $A$, loaded by spring $D$. Plunger $A$ has a tapered (conical) shank while the binder plunger $B$ has a matching tapered flat. When the fixture is loaded, spring $D$ keeps plunger $A$ up against the work; by actuating cam $C$, the binder is pulled outward, and the tapered flat engages and locks the tapered shank on $A$. The double taper on both plunger and binder makes it impossible to press the plunger down, away from the work.

The clamp is, by its design, a quick-acting device. This property can be combined with other design elements to produce devices that perform more than one function in one stroke.

The quick-acting jig clamp, Fig. 10-42, has a handle $A$, threaded to fit screw $B$, and a cam lobe $E$ that engages strap $C$. As handle $A$ is turned, cam $E$ applies pressure to strap $C$. A movement of handle $A$ of approximately 90 degrees, produces the clamping action on the work. This allows for a variation in the thickness of the piece to be clamped, equivalent to one-fourth the lead of the screw thread advancement. For example, with a 5/8-11 screw, a tolerance of plus or minus 0.011 inch would be allowed in the work. A groove is cut in the upper surface of strap $C$, and when the strap is loose, the cam rests in this groove (see sectional view). About 30 degrees movement of handle $A$ is required to cause cam $E$ to ride on top of the strap, as shown by the sectional view at the left.

The head of screw $B$ has six grooves (lower right-hand corner), which are engaged by set-screw $D$ to prevent it from turning. To adjust the lever or tighten the strap when parts wear, screw $B$ is turned to a new position and locked in place by set-screw $D$ which also serves to keep screw $B$ from dropping out of the jig. It is advisable to make a positive stop for handle $A$, so that the cam will not fall into the groove by a 180-degree turn and so loosen the strap.

Work can be clamped with one quick stroke, in the milling fixture shown in Fig. 10-43, by a cam-actuated clamping device. The workpiece is shown secured between the clamp and the form block, ready for the milling operation. It will be noted that the cam is provided with a handle having a ball at one end. At the completion of the cut, the handle is raised to a vertical position. This causes a tooth on the underside of the cam to enter a notch in the top of the clamp, thus moving it away from the form block and permitting the part to be unloaded from the fixture. The clamp is held in contact with the cam by a spring-loaded support finger which slides up and down on a dowel-pin. When another part has been placed on the form block, the ball is again lowered to the position shown. The tooth provides a positive engagement between cam
and clamp, moving the clamp to the left, over the part. The cam surfaces then force the clamp down on the part, holding it securely during the milling operation. The weight of the ball prevents the part from working loose due to chatter or vibration. Various modifications of this type of quick-acting clamp are commercially available.

A bayonet-lock is a type of cam and the bayonet-lock type of clamping device, Fig. 10-44, is fast in operation and positive. The bayonet slot is milled in ram C, and the point of screw D (which is locked in place by a check-nut), slides in it. In operation, the part is slipped over stud A with one hand, while with the other hand, handle E, attached to the ram, is pushed in and rotated with a single continuous motion. The shoulder stud A, extends into the work for about two-thirds of the length of the hole. This insures accurate location of the work and provides ample support against the thrust of the drill. The stud is flattened, as shown, to give ample drill clearance. The revolving cap B turns on a crown at the end of the clamping ram C and provides for a slight amount of float to compensate for possible variations in the work. As clamp B remains stationary during the actual turning or clamping motion of ram C, scoring of the face of the work is avoided. The drill bushing F, in the jig illustrated, is permanently fixed to the base.

Toggle clamps are commercially available in so many types and models that they satisfy all, or almost all, ordinary fixture clamping requirements. They occupy quite a large space, however, and the need for the design of a special toggle device arises when the clamping device has to fit within narrow space limits. Figure 10-45 shows a clamping device of this category that has been found useful on large work. It consists of four arms A with the ends bent to a right angle, and knurled, to hold the work firmly in place. These arms are pivoted on stud B, and

**Fig. 10-43.** A cam-operated clamp for quick withdrawal.

**Fig. 10-44.** A bayonet-lock type of cam clamp.

**Fig. 10-45.** A multiple toggle clamp actuated from the center of the fixture.
their action is guided by blocks C. The spring handle E, is pinned to the shank of the stud, and the upper edge of the handle is beveled to fit rack D, which is fastened to the side of the base. By turning the handle in the direction indicated by the arrow the work is securely clamped. If necessary, ordinary straps may be added for holding the work.

The location of drilled hold-down bolt holes through the steam cylinder heads for duplex piston pumps was often inaccurate when flat bushing plates of the same shape as the casting were used as jigs. These bushing plates were equipped with vertical pads around their peripheries to form nests for the castings. However, due to variations in the size of the castings, many of the workpieces would have a loose fit in the jigs, resulting in inaccurate location of the drilled holes. To overcome this difficulty, the drill jig seen in Fig. 10-46 was designed to accurately clamp the work at four points by means of a duplex toggle action.

The two clamping arms A are mounted to slide on bushing plate B by means of studs C; the central portions of these studs passing through large holes in the arms to permit their free movement. Pins D are a loose fit in the centrally located projections on the clamping arms, and their lower, enlarged diameter ends are provided with flats to fit slots milled in the bushing plate. This permits the arms to pivot about these pins and to slide along the slots when operating handle F is rotated. Cam F, which is rotated by handle E about stud G, is connected to clamping arms A by links H. These links can pivot about the loose-fitting studs J joining them to the clamping arms and cam. A spring-loaded latch K holds the cam, levers, and arms in the work-clamping position shown, or in the loading position, when the cam is rotated counterclockwise. As the cam is rotated counterclockwise, latch K will be rotated clockwise and links H will become aligned with each other. This forces clamping arms A outward, away from each other, so that the jig can be placed over workpiece X. The cam is then turned clockwise to the position shown, and arms A are pulled together firmly to clamp the work for drilling. Ten holes 3/4 inch in diameter, are drilled through the cylinder-head castings in this operation.

The Use of Adapters

Relatively inexpensive yet efficient fixtures are made by using a commercial work holder as the fixture base, then attaching special inserts to the jaws. Examples of lathe chucks converted to fixtures are shown elsewhere in the book (Chapter 9, Centra-lizers; Chapter 20, Miscellaneous Fixtures). The most common, versatile, and least expensive work holder suitable for conversion into a fixture is the machine tool vise.

![Fig. 10-46. A duplex toggle action clamp with four clamping points.](image-url)
The cheapest kind of milling fixture that can be built is a pair of detachable vise jaws, as shown in Fig. 10-47. Made of cold-rolled steel and case-hardened, they are inexpensive. They can be removed from the vise quickly and replaced by other jaws. Detachable jaws are widely used where great accuracy is not required, such as when cutting to length or milling clearance cuts. The jaws shown here are used for cutting off pieces from a bar of stock, which is pushed up against the stop and then cut off to the desired length. However, when the jaws are made with adequate precision, and the vise is in good condition, this type of fixture can be used for work with tolerances down to ±0.001 inch (±0.03 mm).

![Fig. 10-47. Detachable vise jaws for holding bar stock.](image)

Accuracy is improved if the detachable jaws are fastened to the vise jaws by screws and also secured in position by dowel pins. The fixed jaw is the fixture base. It carries the locators for the part, and the machining pressure must always be directed against the fixed jaw. With this simple device, the vise has become a fixture of wide applicability. The possibility of using a vise with inserts should always be investigated in the early stage of planning for a small part.

Almost all the rules of locating, and many of the rules of centralizing, can be applied to the design of vise jaw inserts. Round parts are held in V-blocks. Detachable jaws can be made larger than the vise jaws, thereby expanding the capacity of the vise and, at the same time, reducing its rigidity. Precision alignment of the two jaws is obtained by providing guide pins or matching tongues and slots. Insert faces can be machined to an angle for parts requiring angular cuts. Parts can be located by stops, pins, and nests. Inserts can be designed to hold more than one part, equalizing yokes can be attached to a detachable vise jaw, and even ejectors can be built in. While most applications of the vise with fixture inserts are for milling operations, it can also be used as the base for a drill jig by the addition of a drill jig plate with bushings. Various types of air-operated vises are commercially available; naturally, they offer the same conversion possibilities as the hand-operated vise and, in addition, faster operation. The same is the case with cam-actuated vises. Hydraulically operated vises offer greater clamping pressures than any other type of vise; they are available with a clamping force of up to 20 tons (178 kN).

Combinations of commercial work holders can be used to advantage. For example, a drill chuck, acting as a centralizer, can be fitted to be held in a vise, thus acting as the fixture base. A conventional drill chuck may be adapted to hold small workpieces that might otherwise be distorted if clamped directly in a vise.

**MISCELLANEOUS CLAMPING METHODS**

**Magnetic Chucks**

Magnetic chucks are available in two main types, as a surface plate (see Fig. 10-48)\(^3\) usually of rectangular shape and as a face plate (for mounting on a rotating spindle) usually of circular shape. The body of the chuck can be trunnion mounted for precision machining. For precision work the chuck has a reference pin mounted on one end and a matching flat reference surface on the base. The distance between the two reference surfaces is measured with gage blocks; in this way the chuck functions as a sine plate. The magnet poles terminate flush with the face of the chuck, and are separated from the

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chuck body by strips of nonmagnetic material (copper, brass, austenitic stainless steel, aluminum, or plastic) of a thickness of approximately 1/8 inch (3 mm). The polarity of adjacent poles alternates, or all individual poles have one polarity, and the surrounding face of the chuck has the opposite polarity. Either permanent magnets or electromagnets are used. Chucks with permanent magnets have less holding power than electromagnetic chucks, but have the advantage of not requiring an electric power supply. As long as the chuck is empty, the magnetic circuit is open at the chuck face. When a workpiece of a ferromagnetic material is placed on the chuck, the circuit is closed. In electromagnetic chucks the circuit is permanently closed within the chuck; when the current is switched on, the magnets are energized, and the magnetic flux passes through the workpiece clamping the work to the chuck. Chucks with permanent magnets have mechanical devices for opening and closing the magnetic circuit inside the chuck; the magnets may rotate or slide in and out of their closing position, or they are connected at their lower end by a sliding armature with nonmagnetic inserts which interrupt the circuit when the armature is moved to the “off” position. While most magnetic chucks are purchased for use as general-purpose work holders, it occasionally may be necessary to design a special magnetic chuck for incorporation into a fixture, and this requires estimating the holding power from some given data. The holding force has two components, the tensile holding force which prevents the part from being pulled off the chuck, and the shearing holding force, which prevents the part from sliding along the surface. The shearing holding force is essentially a friction force and is significantly weaker than the tensile holding power. The tensile holding force depends on the strength of the magnets, the position of the part relative to the poles, the size of the contacting surface, and the material, configuration, and surface quality of the workpiece. The overall strength of the chuck is expressed by its energy consumption in watts. Specifically, the strength of the individual magnet depends on the
number of ampere-turns in its coil. The significant material property is the magnetic permeability. Low carbon steel has the highest permeability of the common ferrous materials, and this property decreases with increasing content of carbon and alloying elements (Cr, Ni, Mo, etc.). The clamping force for cast iron is about 60 percent that of steel. The best position for the part is across the poles, covering at least three poles. Long and narrow parts may have to be clamped parallel to the poles, straddling over two poles only. The largest holding force is obtained with well fitting, finish machined surfaces. For rough machined surfaces the holding power is about 75 percent and for unmachined, but fairly regular, surfaces it is about 40 to 50 percent of the best value.

Values for AISI 1018 steel for various clamping conditions are listed in Table 10-5 and show that the height perpendicular to the chuck is highly significant, while the width in contact with the chuck has much less effect on the force per unit area.

Compared to other clamping methods, magnetic clamping is relatively weak. It is widely used for grinding, and can be used for light milling and turning. It is fast and convenient. Magnetic chucks are relatively inexpensive. Ferrous materials acquire remanent magnetism by this clamping method and must be demagnetized.

There are several means for improving the performance of the magnetic chuck. They can all be described as field shapers, as they affect the shape of the magnetic field. Their purpose is to draw more magnetic flux through the work either by increasing the area of access for the flux or by locally increasing the flux density. Figure 10-49 shows several such devices. A is a block of steel called a “binder block.” It is occasionally called a locator although it does not perform the functions of real locators as they are commonly used in fixtures. It collects flux from the chuck and delivers it to the work, thereby increasing the area of clamping as well as the clamping force. Binder blocks are used parallel to the work and also as end stops. B is a hold-down plate for the purpose of clamping thin workpieces C of nonmagnetic material; it is made of steel and collects some flux, sufficient to produce a clamping force on the work. The inserts D are flux dams; they are made of nonmagnetic material and divert the flux in such a way that the flux density is locally increased where greater flux penetration is required. A field shaper of a different design is shown in Fig. 10-50.4

Fig. 10-50. A field shaper of the V-block type for holding cylindrical parts.

It consists of narrow elements with thickness and spacing matching the poles in the chuck. The elements are separated either by air gaps or by nonmagnetic spacers, and function as pole extensions. Field shapers of this type come closer to the original definition of a locator. The design shown is a V-block for clamping cylindrical work.

Electrostatic Chucks

Electrostatic chucks (see Fig. 10-51) work on the principle of the attraction between surfaces with opposite electrical charges. The chuck body consists of a ceramic material with an additive to make it a semiconductor. It is charged to about 3000 volts from a power supply while the workpiece is electrically connected with the chuck frame which is

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permanently grounded. The amperage of the current delivered by the power supply is very low, just enough to make up for any leakage that may occur in the dielectric film. These features eliminate any electrical hazard. The two surfaces are insulated from each other by a hard resin coating on the chuck surface supplemented by a film of a liquid dielectric to prevent air from entering the interface. The same dielectric can also be applied to the grinding wheel for cooling and lubrication. A small heater maintains the chuck at a temperature a few degrees above ambient temperature to prevent condensation of moisture.

In accordance with Coulomb's Law the attractive force is proportional to the product of the charges and inversely proportional to the square of the distance between the two surfaces. Since this distance is small, the force is quite significant, approximately 30 psi for clean and smooth surfaces and independent of the thickness of the workpiece. Electrostatic clamping can be applied to any conductive material; for nonconductive materials it is sufficient to give them a conducting surface by spraying them with a quick-drying, conductive aerosol lacquer, which can be dissolved after the operation; or by mounting them on a metal foil, with an adhesive.

**Adhesive Clamping**

Clamping by means of primitive adhesives such as resin, varnish, and shellac have been used presumably since the beginning of the machine shop trade. Modern technology has developed a number of new adhesives for the joining of metallic surfaces. The use of adhesives for clamping in a fixture is deceivingly simple but has a number of drawbacks.

The method does not necessarily provide precision clamping because the adhesive film has a finite and significant thickness that is not always easily controlled. The curing of the assembly after adhesive has been applied always takes some time, it may be a few hours, or it may be overnight. The adhesive must have a matching solvent, so that it can be removed, and the removal is also a time-consuming process. On the other hand, the adhesive must not be soluble in cutting fluids. Considering all this, adhesive clamping is selected only when no other acceptable method can be found, and is limited to parts with a flat clamping surface and to light machining operations.

**Vacuum Clamping**

Clamping by vacuum is a versatile, fast, and clean clamping method. There are no requirements with respect to thickness or size of the part nor to the electrical or magnetic properties of the material. The method is applicable to flat as well as to curved surfaces and is not very particular with respect to surface quality, since the clamping area is sealed off all the way around. The method is distortion-free because the surface of the vacuum chuck is made to match the work surface. The theoretical maximum clamping force is 14.7 psi (0.1 N/mm²) with full vacuum and is reduced in proportion if the vacuum is less. The available clamping pressure is sufficient for many machining operations on aluminum, and vacuum clamping is widely used in the aircraft industry.

Vacuum fixtures are made from cast iron or aluminum. A flat clamping face is machined; a curved clamping face is cast to form, then ground and polished smooth. A net of crossing grooves is formed or machined into the clamping face to act as a manifold for distributing the vacuum to the entire surface (see Fig. 10-52). The seal along the edge consists of a rubber hose in a groove. Vacuum chucks for very thin sheets do not have grooves as they

*Fig. 10-51. An electrostatic chuck.*

*Fig. 10-52. Design details of a vacuum chuck.*
would leave permanent markings on the plates. Small parts can be clamped on individual suction cups shown at the right in Fig. 10-52; the center plug is machined so that it supports the part without leaving permanent marks.

**Clamping with Low-melting Alloys**

Low-melting alloys can be used for clamping parts of awkward shapes, large overall dimensions, and thin wall sections. The use of cast materials for nesting purposes is described in Chapter 6 and several cast materials differing in melting points and pouring temperatures are listed. Most of these require access to the facilities of a nonferrous foundry. For clamping purposes the preferred metal is Cerro Bend, a bismuth alloy which melts at 158°F and can be handled in the machine shop without the assistance of a foundry. One technique consists in making a set of nesting clamps. The part serves as the pattern; it is coated with a parting agent, and located in a split mold or flask similarly coated with the parting agent. The metal is poured, and after solidification and removal from the mold, the metal block is machined to fit the jaws of a work holder, or fixture. The block is cut in two or three pieces, the part is removed, and the pieces of the casting are finally installed on the jaws, thereby completing the fixture. A similar technique is used for fixtures with one nest and an ordinary clamp. An example is shown in Fig. 10-53, where the part is a valve hous-

![Fig. 10-53. A fixture with a cast nest.](Image)


slots for the legs of the frame and serve as molds for the casting process. Additional equipment comprises the two removable centering arbors F. The arbors are placed in the tubular legs; the fixture is placed on a surface plate and the part is set up and leveled by means of the screw jacks and arbors resting in V-blocks E. The slots around the legs are sealed with clay, the metal is poured and when it has solidified the part is clamped. V-blocks and arbors are removed, and the loaded fixture is moved to the milling machine and clamped on the table with clamps acting directly on the cast-metal blocks. After machining, the fixture is set in a tray, hot water is poured over the metal blocks, the metal melts away easily and is collected in the tray, ready for reuse.

Another, and more sophisticated, application is the clamping of gas turbine compressor blades for the machining of the root. The shape of the blade is an air-foil which, in itself, provides virtually no usable clamping surface. The blade is inserted and located in the casting die in a regular die-casting machine and a low-melting alloy is cast onto and
around the blade, thereby providing a set of suitable clamping surfaces.

In the aircraft industry, honeycomb in the expanded condition is clamped for machining by means of ice. With the honeycomb block placed on a refrigerated platen, the cells are filled with water. As the ice is formed it supports the cell walls and clamps the block to the platen.

RESUME AND CONCLUSION

When designing clamping devices, the fewest number of operating screws or handles should be used as will accomplish the desired result. Making the screw with a double or triple thread is sometimes employed to advantage in decreasing the number of turns necessary to release the piece. Jig lids should be hung on taper pins in order to compensate for wear in the hinge and to prevent any resultant inaccuracy due to lost motion in the hinge. The included angle of taper on hinge pins should be only one or two degrees. The hinge pin should be a tight fit in the central portion of the hinge, which is usually the jig body, and a bearing fit in the ears of the lid.

All manually operated clamping screws and similar parts should be long enough and so located as to be convenient and easy to operate, and of sufficient size to prevent hurting the operator's hands because of the manipulating pressure necessary. The screws should be located so that they will resist the tilting action of the block, and dowel pins should be fairly close to the screws and of liberal dimensions in order to resist the shearing strains to which they will be subjected. When clamping or locating the work in the jig, it is essential to have the clamping pressure exerted in a direct line against some solid point of support to prevent the tendency to tilt, and the thrust should also come at such a point of the work that it will be resisted by solid metal.
Equalizers

Definitions

The purpose of an equalizer is, in the broadest sense, to distribute a single force between two or more separate points of action. Strictly speaking, the name implies an equal force distribution; but there are also equalizers that distribute the force in a constant ratio not necessarily equal to one. Predominantly applied to clamping mechanisms equalizers are also used on locators.

When an equalizer is applied to a clamping mechanism, the force to be distributed is the "actuating" force; when applied to locators it is more difficult to visualize the "force" because it is, in most cases, a reaction to one or more clamping forces. Equalizers can also be applied simultaneously to a clamp and a locator for the purpose of either equalizing a clamping pressure and the locator reaction, or maintaining a constant ratio, not necessarily equal to one, between these two forces.

Individual screws, even when threaded through the same fixture component, are, as a rule, not equalizers. When clamping with two screws as was shown in Figs. 10-14a and 10-20b, the screws are tightened one at a time, and whether or not the part is clamped with equal pressure depends on the operator. In contrast, the two pointed screws shown in Fig. 10-14b constitute an equalizer. When one screw is tightened to a certain pressure, it generates an equal reaction from the other screw. The same is true with the two eye-bolts in Fig. 10-21b.

Representative Examples

Although some equalizers are complicated, this is not necessarily always true. The plain double-end strap with a screw in the center, clamping simultaneously on two parts, as shown in Fig. 10-17, is an equalizer, as is the three-pronged strap shown in Fig. 10-16d. A typical equalizer is shown in Fig. 10-36; it is the balancing clamp in the center of the fixture.

In Fig. 10-22a, an equalizing effect is obtained by a gripping dog. When screw E is tightened, the pressure on the work from the dog D, is increased, as is the pressure on the locator below A. Strap C in Fig. 10-23a is an equalizer for the two slides B. In the fixture shown in Fig. 10-23b, the equalizer is rod D in conjunction with screw B. Actuating screw B has the same effect as an increase of the length of rod D which pushes with equal pressure on levers A and C.

Clamp B in Fig. 10-24 contains an equalizer, namely, the V-notch that grips over the corner of the work. The horizontal and vertical pressures on the work increase in the same ratio when screw E is tightened, assuming that the coefficient of friction between the tip of the screw and block D remains constant. The clamping device shown in Fig. 10-25 is an equalizing mechanism because of its symmetrical design.

Many centralizers, including those shown in Chapter 9, are also equalizers. A V-block holding a cylindrical part is an equalizer. The toggle-action drill jig shown in Fig. 10-46 contains two equalizing systems; one equalizes the pressure on the two ends of each clamping arm A (the arm A being the equalizer), the other equalizes the two arms A against each other (the equalizer is actually the workpiece X). A similar effect is found in the clamping device shown in Fig. 10-45, where it is the workpiece itself that equalizes the pressure from the four arms A.

The Floating Principle

An analysis of these examples shows that in all of them the forces are transmitted through a floating component; that is, a component which is movable with at least one degree of freedom. The floating component is statically determinate and adjusts its position freely until force equilibrium is reached. In some cases the floating member is represented by the workpiece.
The principle of action of an equalizer is often referred to as "the floating principle." In mechanical language the function of an equalizer is to eliminate a redundancy. The required movability in the system is obtained either by forming the equalizer as a beam that can swivel around an axis, or by using a stiff member—such as a rod—that floats between two springs. Although the details may vary, most equalizing systems are based on one of these two principles.

### Classification of Applications

Basically, equalizers are used for the following purposes:

1. To distribute an otherwise concentrated clamping force more evenly over the surface of a part
2. To align clamping forces with locators
3. To clamp on rough surfaces
4. To clamp on surfaces (rough or machined) of different heights
5. To clamp simultaneously on a horizontal and a vertical surface
6. To spread the clamping forces (and their matching locator reactions) evenly over a wide area to avoid distortion of a thin-walled and elastic workpiece
7. To center a part
8. To clamp simultaneously on more than one part (multiple clamping).

### The Problem Areas

The satisfactory function of an equalizer requires that the design assumptions are fulfilled. This requires attention to several factors which, under adverse conditions, can restrict the movability of the equalizer. As applied to clamping or holding methods, the greatest care must be exercised in order to make sure that the floating action is not constrained in any one direction, but will operate equally well, and with uniform pressure, on the required area. Frictional resistance may, in cases of this kind, be sufficient to cause imperfect work by reason of unequal pressure on the work itself.

Friction is invisible although it occurs wherever there is sliding motion. Most equalizers close with a slight rotation and friction forces are generated at the points of contact. The fixture designer must visualize the direction of the friction forces, estimate their size, and evaluate their possible effect. Significant friction forces occur on rough surfaces and also where forces are transmitted through surfaces that are not perpendicular to the force direction, such as on plungers with their ends at 45 degrees.

When clamping action is applied to a rough surface, great care must be used and the amount of float must be so proportioned that it will take care of a considerable variation in the castings or forgings. When a great number of pieces are to be handled, several patterns are often used and these will be found to vary somewhat, thus, there are differences in the resultant castings. Allowance must be made for extreme cases of dimensional variations.

When applied to methods of locating the work, or to supporting points on which it rests, the construction must be such that it will not, by any possibility, cause distortion. If springs are used under supporting plugs which are later to be locked in position, the springs must be so proportioned that they will not be strong enough to cause any trouble by forcing the piece out of its true position. Also, when supports are placed against finished surfaces they should be so arranged that they will not injure them. In locating a piece of work from two previously machined surfaces which are in different planes, the float action must be very carefully studied, to be certain that the contacts are positively assured, and no tilting of the work will result. In such cases, the equalizer may occupy a tilted position and it is necessary to check for the possibility of false contacts between the equalizer and other parts of the fixture and the workpiece.

There are occasional instances which require the location of a piece of work from a previously machined surface, in connection with a threaded portion by which it must be clamped. In a case of this kind, the "float" must be made so that it will take care of a possible lack of concentricity between the thread and the other finished surfaces and, at the same time, provide means of equalizing variations in the alignment of the thread. Locking devices for floating members must be so arranged that the members can be positively locked or clamped without causing any change in their position. A turning action, such as might be caused by the end of a screw against a locating point, is often sufficient to throw the work out of its correct position. The interposing of shoes between screws and floating members will prevent any trouble of this kind. The swiveling shoe or pad on the end of a screw is, in itself, an equalizer applied to a small area, as is the spherical washer.

### The Rocker Equalizer

The basic and most commonly used equalizer type is the "rocker." In principle, it is a beam, supported
at the center and loaded at the ends. It can be a straight beam, such as the double-end strap, or it can be curved, as a yoke, when it is supported from above and has to reach down to the workpiece. This form is typical and a good example is shown in Fig. 11-1. A is the work to be clamped, and B is the yoke which fits into a slot in the center of the strap, or clamp, C. The yoke is held by pin D, around which it can swivel to adjust itself to the work. The amount of pressure at the two points E and F will be equal, even though the screws at the ends of the strap may not be tightened to exactly the same height. Pin D takes the full clamping strain, and should therefore be designed to be strong enough. The strap, which is weakened by the slot and the hole in the center, should be reinforced, as indicated, at this place. It is preferable to have spiral springs at each end of the strap to prevent it from slipping down when the work is removed. The strap may be of either cast iron or machine steel, while the yoke should always be made of machine steel.

Fig. 11-1. The rocker-type equalizing clamp.

The plain rocker acts on two points only. By placing a small rocker on each end of the main rocker its action is expanded to four points; this development can be carried further and is used for simultaneous clamping of several parts (multiple clamping). Parts with large, flat areas requiring three clamping points can be clamped with a rocker-type equalizer formed as a plate, as shown in Fig. 11-2. The fixture is a drill jig provided with a floating clamp to work on a rough surface of a piston casting A, which has been previously machined at B. The body of jig G is of semi-box section and is provided with feet D on which it may rest, both during loading and when under the drill. A hardened and ground steel stud F is let into the casting at one end and serves as a locating point for the machined interior of piston B. A stud C is also provided to give the correct location to the wrist-pin bosses.

As the end of the piston is of spherical shape and in the rough state, it is necessary to provide a means of clamping which will so adjust itself to the inequalities of the casting that an equal pressure will be obtained so that there will be no tendency for the work to tilt. A heavy latch M is pivoted on a pin L, and is slotted at the other end to allow for the passage of a thumbscrew N which is used to clamp it in position. A special screw O is threaded into the latch, and is ball-ended at P so that it has a spherical bearing against the floating clamp Q. The screw S keeps it in position, but clearance is provided to allow for the floating movement around the body of the screw. Three pins R are set 120 degrees apart, in the face of the floating clamp, so that a firm three-point bearing is assured. In order to assist in supporting the work under the pressure of the drill, two spring-pins T are provided, set in the form of a vee near the front end of the piston. They are encased in a screw bushing U and are locked in position by means of set-screws (not shown) after they have been allowed to spring up against the piston casting. (In order to avoid confusion in the drawing, only one of these pins is shown and that at an angle of 45 degrees from its actual position.)

Steel liner bushings F are provided in the body of the casting so that the main bushings, which are of
the removable type, as shown at $H$, may not produce too much wear in the jig body itself. A slot is provided in the head of the bushing so that pin $K$ will prevent it from turning under the twisting action of the drill. It should be noted that in the construction of the spring-pins, which are used to help support the casting, the springs themselves should be very light so that they will not force the piston out of its true position, determined by the locating stud.

Equalizing on three points can also be accomplished by a system of 45-degree wedge-end plungers. A very rigid mechanism of this type is shown in Fig. 11-3. It is shown applied to a drill jig, but it is rigid enough to permit its use in milling or planing fixtures. In these cases, the clamping pins become rest pins and are subject to the thrust of the cut.

Screw $A$ thrusts against equalizing plunger $B$. Plunger $B$ is of smaller diameter than the drilled hole and rests on piece $C$. This piece is cut from a rod of the same diameter as the hole and is used to afford a flat base for plunger $B$ to rest on and insure full contact of the wedge end against plungers $D$ and $E$. Plunger $G$ is a duplicate of $B$ and equalizes plungers $F$ and $H$ by means of the same mechanism.

The Floating Screw Type Equalizer

The second basic principle used in the design of equalizers is that of the floating screw. A screw and its nut are mounted without lengthwise fixation between two clamps or other force-transmitting components. When actuated, the screw and its nut exert equal but opposite forces on the two adjacent components. A typical example, shown in Fig. 11-4, is the locating mechanism for a milling fixture in which two pieces are located by two plungers each, all operated by a single clamping operation. Lever $A$ draws out plunger $B$ and throws in sleeve $C$, operating plungers $D$ and $E$. Plungers $E$ are smaller in diameter than plungers $D$ and permit enough lateral movement to equalize plungers $G$ through auxiliary plungers $H$.

A fixture that shows a double application of the 45-degree wedge-cut plungers is shown in Fig. 11-5. Two pieces are each clamped simultaneously, at both ends, all by tightening one nut. The forces are fully equalized regardless of irregularities on the individual parts. Rod $A$, running through the fixture, carries ball-and-socket washers at each end and draws the end clamps $B$ and $C$ together. These clamps are given a down-and-in movement against the 45-degree wedge ends of rods $D$ and $E$. The clamping thrust against the rods imparts a downward movement to the inner clamps $G$ and $H$, pulling the work down on the inner rest-pins. The clamps are returned by means of plungers $K$ and spring $J$. This fixture is also an example of a modification of the strap with a wedge end, which was shown in Figs. 10-25 and 10-26.

Double Movement Clamps

Double movement clamps are equalizers that produce two force components with a constant ratio,
usually in different directions. They are used in a large variety of forms and applications and offer a wide field for the fixture designer. A few examples will be shown.

Figures 11-6 and 11-7 illustrate small, double-movement clamping mechanisms for hand milling or profiling use. In Fig. 11-6 the clamping pressure against clamp A also pulls out plunger B, raising plunger C and throwing the work against stop E, by means of plunger D. Spring plunger G is used to return plunger D. In Fig. 11-7 the pull-through clamp A on the plunger B throws the work against stop C, by means of plungers D and E. Figure 11-8 illustrates half a fixture for milling a cylindrical concave surface on an unusual piece. The work is clamped against pads A and B, on previously milled surfaces, by means of two differentially operated plungers C and D. To prevent springing under cut, the work is backed up with the floating plunger E on one side and F and G on the other. The plungers are operated by push-rods H and J. These push-rods are hand operated and are clamped by bushing K and star knob L.

Multiple Clamping

Multiple clamping consists of clamping several parts in one fixture for simultaneous machining. The most primitive form of multiple clamping is simply to mount a number of identical fixture units on a common base. Each fixture is equipped with its own clamping devices and is unloaded and loaded separately. When the fixture is fully loaded, all parts are machined in one operation. The obvious first step to improve this simple setup is to combine the common fixture base and the individual fixture housings into one part, a casting or a weldment, but maintain the individual clamping devices. Besides its simplicity, this type of multiple clamping has other distinct advantages. It is cheaper, more rigid, and faster to set up than are separate fixtures. It
also ensures proper clamping of nonuniform parts. The loading and clamping operation is faster since it utilizes shorter and more repetitive movements. The positioning of the cutter is utilized for a plurality of parts. The total length of travel of the cutter, and therefore the machining time, is also less, as there is a saving in idle time in the approach, end run-out, and traverse from one part to the next. This saving is obviously greater, the closer the parts are set. The number of idle return strokes, or travels, is reduced to one.

This simple application of the multiple clamping principle is used to some extent in milling fixtures (string milling) and is widely used in fixtures for reciprocating machining operations; notably planing and surface grinding.

The largest expense factor in this type of multiple clamping remains the extent of time required for clamping the parts individually. It would be desirable to be able to clamp them all simultaneously, thus the most drastic step to take is to hold all the parts in one fixture and, if possible, clamp them together in one operation.

An approach to this solution is the string milling fixture shown in Fig. 11-9. The entire package of parts is clamped lengthwise by one screw; each piece is clamped sideways by its own individual clamping screw. All side screws are mounted in a swinging bar which can be quickly removed for unloading, cleaning, and reloading of the fixture. The use of individual screws ensures that each part is fully located and aligned with the side wall in the fixture. It also prevents any "bulging" or "buckling" of the whole package by the pressure from the end screw.

Bulging of a Package

Bulging or buckling of a package of parallel pieces may occur as a result of accidental irregularities and thickness variations. However, buckling may occur even if the pieces have flat and parallel machined surfaces. In that case it is an instability effect like the buckling of a column. The tendency to buckling increases with the ratio \( L/H \) (see Fig. 11-10). To prevent buckling, setting the end screws under a small angle with the horizontal, pointing downward, is often recommended. Another very simple and effective means of preventing upward buckling is to place the end screws above the half height of the package. The clamping force is now an eccentric load and generates an uneven pressure distribution over the contact surface. Already having an eccentricity \( E = 1/6 \ H \), the pressure at the lower edge drops to zero, and with \( E > 1/6 \ H \) an area along the lower edge has zero pressure. The package is compressed along the
upper surface and opens the joints along the lower surface. It will try to buckle downward, resulting in a well-controlled pressure against the fixture base. If no means are applied to prevent buckling, the number of pieces within a clamped package should not exceed 5.

The principle of clamping, in a package, can be extended to parts that are not flat by the use of suitably formed spacers. An example of this is shown in Fig. 11-11. The spacers are mounted in the fixture (no loose pieces) and define the place where each part is to be clamped. In the present case, the spacers are formed as half V-blocks, which match the upper ends of the parts while the lower ends rest in full V-blocks.

A somewhat unorthodox, but very versatile, design for multiple clamping of small parts is shown in Fig. 11-12. It can be used for surface grinding and for straddle milling, slotting, and form milling. The base A may be fastened to a secondary base so that the fixture can be easily attached to the machine table. The work-holding members of the fixture consist of two pieces B, right- and left-hand, and a piece C, which is dovetailed to match a corresponding dovetail in parts B. Four swinging clamp members D are mounted on body A and are arranged so that one end of each bears against a tapered portion of pieces B. As a result, when the work-holding unit is forced in the direction indicated by arrow F, through the action of the eccentric clamp E, clamps D will pivot on their bearing pins and exert a force against the sides of pieces B, moving them slightly on the dovetail on part C and clamping them tightly on the workpieces. When the clamp is first being lined up for holding a given type of work, clamp E is tightened very slightly. An attempt is then made to pull the workpieces at the two extreme ends from the holder with the fingers. This is done to permit adjusting the clamping members D, so that the two pairs will have an equal clamping force.

If it is easier to pull the workpiece from one end than from the other, the clamping pressure is not equal. One of the two clamping members D at the tight end of the holder is then removed, and a small amount is ground from one of its pressure areas, after which it is replaced and another try is made. By this means, repeating the test process until it takes a very sharp tug with the fingers to remove either of the end pieces when E is lightly set, the device is considered correctly adjusted, assuming the workpieces to be uniform in size.

In most cases, operations performed on work held in a fixture of the type described will include only cuts so light that there will be little danger of the assembly being lifted from base A during machining. Should such trouble be experienced, however, it is only necessary to mill vees in the edges of pieces B where the clamping members D take their bearing, and grind the engaging ends of members D for a good line bearing in the vees.

In the design shown, the two pieces B were originally one, the holes being drilled first and the piece cut in two, afterward. Because of the way parts B
are mounted on piece C, it is possible to separate them for milling any desired shape of work seats in these parts. Thus, when the cross section of the work is not symmetrical, a work seat of a certain shape can be milled in one member B, and a differently shaped work seat in the opposite member. If required, it is also possible to have two or more sets of pieces B for different types of workpieces, which can be used with the same retainer piece C and the same base A.

A device such as that described can be made without any recesses for workpieces and used for holding a number of strips of material in a line pattern for surface grinding or milling their edges, machining first one edge and then the other. Other variations in the design are also possible. For example, both pieces B can be drilled with two pairs of registering holes in their inner edges to receive compression springs, so that the clamping action will take place against the resistance of these springs. The device will then automatically open, permitting the work to be unloaded easily and new parts inserted when the clamping pressure has been removed.

**Multiple Clamping with Rockers**

The rocker principle finds many applications for multiple clamping. One rocker clamps two parts; two smaller rockers mounted on the ends of a larger rocker clamp four parts. Carrying the principle one step further leads to an arrangement for clamping eight parts, all by actuating only one clamping component, a screw, a cam, or a hydraulic cylinder.

An example of multiple clamping is the fixture which was shown in Fig. 7-22. Each of the two jaws actuates two rockers, and each of these clamps, two parts. Another fixture working on a modified version of the same principle is shown in Fig. 11-13. The clamping pressure on eight small washers is equalized, and the washers clamped with a down-and-in movement in the fixture. Rod A clamps the equalizers B and C, which equalize the pressure against D and E on one side, and F and G on the other. Clamps D, E, F, and G are given a downward pull by four plungers H, which also impart a downward pull on the inner clamps J, K, L, and M. The clamps are bored to receive the washers, and are returned to normal position by the spring plungers N.

**Fig. 11-13. A vise type fixture with rocker type equalizers.**

Rocker systems can be designed for equalized clamping of any desired number of parts, not just 2, 4, 8, etc. In such cases, it is necessary to use rockers with arms of unequal lengths. Figure 11-14 shows the arrangements for the clamping of 3, 5, and 6 parts with equal pressure.
Multiple Clamping with Rollers

Multiple clamping systems can be made with cylinders (rollers) or spheres (balls) instead of rockers. Examples are shown in Fig. 11-15. They can also be used for the clamping of single parts with uneven surfaces as indicated in diagram d. However, they differ from rockers in their basic mechanics and also in some practical aspects. Rockers are beams and the force distribution is calculated with the equations for the equilibrium of parallel forces. Rollers are solid bodies; the forces acting on a roller are not parallel, but converge in its axis (the center of the circle in the drawings) and the equilibrium condition for each roller is established by its free body diagram.

Hardened steel rollers and spheres have a high load-carrying capacity, and a roller or ball equalizer can, with some skill (see Fig. 11-15d), be designed within less space than the equivalent rocker equalizer. The cost is moderate, as the rollers require no machining other than cylindrical grinding. There are no pins and bearings and no carrying structure other than an enclosing housing which can be made up by the fixture base. There are no parts that can break. To damage a hardened roller by direct pressure takes a high degree of overload and, with proper dimensioning, is a very remote possibility.

A roller-type equalizer looks, at first sight, as if it is statically indeterminate with a redundancy at each contact point on a horizontal center line. However, this condition changes as soon as the device is actuated. As each roller is forced downward, it attempts to separate the next two rollers. This eliminates the contact and from now on the system is statically determinate and can be analyzed by elementary methods. The direction of the enclosing walls is significant for the mechanics of the system. The device shown in diagram a produces two clamping forces, each of: \( P = \frac{1}{2} F \); but the device shown in diagram b has two clamping forces, each of \( P = \frac{2}{3} F \); indicating that this device is not only an equalizer, but is also a force multiplier. The explanation lies in the fact that each of the two lower rollers is forced into a V-block by a force parallel to one side of the V.

The two devices shown in diagrams a and b are true equalizers. There is symmetry and the two clamping forces \( P \) are equal. The device shown in diagram c is symmetrical, and the clamping forces are equal, two and two, but the two inner forces \( P_2 \) are greater than the two outer forces \( P_1 \):

\[
P_1 = \frac{1}{3} F \quad \text{and} \quad P_2 = \frac{1}{2} F
\]

The results quoted above assume absence of friction. Friction, however, is always present; on a roller the frictional component and the actuation forces act on the same radius, while on a rocker the frictional component acts on the radius of the pivot, but the actuating force acts on the length of the

![Fig. 11-15. The mechanics of roller type equalizers.](image)
rockers arm which is several times greater. Therefore, the frictional components are of greater significance in the roller type equalizer and should be taken into account in the detailed analysis with coefficient of friction \( \mu = 0.1 \) (hardened rollers and some lubrication).

### Hydraulie Equalizers

With adequate sealing of the housing the system of rollers can be replaced by a fluid; if pressure is applied to the fluid it is transmitted equally to all clamping points. Hydraulic pressure is excellent for the multiple clamping of identical parts and single parts of irregular contour. A general discussion of hydraulic fixtures is presented in Chapter 21, Automatic Fixtures. A few simple applications not requiring an outside hydraulic power source will be given below.

A vise can be equipped with a hydraulic jaw, as shown in Fig. 11-16, so that a uniform pressure is exerted on all of the castings, regardless of variations in dimensions or irregularities in their surfaces, such as are produced by raised part numbers or company names cast on the work. In the set-up shown, six lever castings \( A \) are clamped simultaneously for machining both sides and the top with straddle milling cutters \( B \). Hydraulic vise jaw \( C \) is drilled to provide oil reservoirs \( D \), with two rows of six plungers \( E \) fitting into the reservoirs. The oil reservoirs are sealed by means of pipe plugs \( F \), and the connecting oil passage \( G \) is sealed by welding plug \( H \) to the jaw after drilling. Snug-fitting rubber washers \( J \) are placed in the groove of each piston to prevent oil leakage. Plate \( K \), machined to fit the tapered sides of the castings, is screwed to the stationary vise jaw \( L \).

In preparing the hydraulic vise jaw for operation, the reservoirs are filled with oil to within 3/4 inch of the face of the jaw. The plungers are then carefully inserted in the jaw, and the assembly is clamped against some parallel surface to align the tops of all the plungers. While clamped, one of the pipe plugs is loosened to permit air and excess oil to "bleed" from the system. This plug is then secured tightly, the assembly is unclamped, and the hydraulic vise jaw is ready for service. In this case, the pressure is applied by closing the vise. The hydraulic system preferably is installed in the moving jaw so that the rigidity of the fixed jaw is not compromised. The system is completely self-contained and the vise is actually converted to a universal fixture.

A similar hydraulic system comprising the oil reservoir \( D \) and the desired number of pistons \( E \), can be installed in other types of fixtures, however, a modification is required to supply the oil pressure. One of the pipe plugs \( F \) is omitted; and a straight screw thread is provided in its place to accommodate an actuating screw with a piston that has a sliding fit in the end of the reservoir. The screw and piston constitute a primitive pump; as they are actuated, oil is displaced from the reservoir lifting the plungers. When the plungers have established contact with the part, pressure builds up and the part is clamped. When the screw is released, the pressure disappears, the plungers retract, and the part can be removed.

The system is versatile; clamping plungers can be arranged in any pattern and direction as long as they

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**Fig. 11-16. A vise type fixture with a hydraulic equalizer.**
communicate with the pump piston through the reservoir. High oil pressures can be produced by these simple means. The fixture base with the reservoir must be designed as a pressure vessel and dimensioned accordingly, but this usually does not present any problem. The installation of a pressure gage on large fixtures is recommended. A practical upper limit for the pressure in oil-hydraulic systems is 10,000 pounds per square inch (69N/mm²).

The only weak spot in the system is the possibility of oil leakage. There is no continual replenishment of the oil, and even a minor leak can soon result in loss of the oil pressure. Should this occur during machining, the part may be spoiled.

Plastic Fillings

The leakage problem is eliminated by substituting a plastic medium for the oil and modifying some design details accordingly. Paraffin, grease, and beeswax have been used in the past and are still usable, but are being replaced by a polyvinyl chloride resin (PVC). It is heated to 350F for filling the fixture. Theoretically it has no upper limit for the applicable pressure, but for reasons of design safety it is recommended that the fixture not be designed to operate at a pressure above 15,000 psi (103N/mm²). Since it is a plastic and not a liquid, it does not offer the same high degree of versatility in the design as does oil. The reservoirs must be fairly straight from the pump to the most remote clamping plunger. The cross-section area of the reservoir must not be less than 1 1/2 times the plunger area. Plungers and pistons shall be guided in their cylinders over a length of at least 1 1/2 times the diameter. The fit shall be an intermediate between sliding and running; a class RC2 fit is suitable; mating surfaces shall be machined to 32 AA roughness and then lapped to a true cylindrical shape; a bell mouth in the cylinder or a tapered end on a plunger is not acceptable because it would provide a tapered gap that could provide access for the plastic medium. For operating pressures above 7000 psi, the pressure end of plungers and pistons shall be bored out to a cup shape to provide a 10-degree feather edge sliding against the cylinder surface. It is recommended that return springs be installed on the plungers. The plastic medium does not provide rust protection. For fixtures that are out of operation for long periods of time, it is recommended that pistons, plungers, and cylinders be made of a corrosion resistant material.

1Plastiflex® proprietary to Hastings Plastics, Inc., 1704 Colorado Ave., Santa Monica, Cal. 90404.
Supporting Elements

Definition and Classification of Intermediate Supports

Intermediate supports are those elements in excess of what is basically required for geometrically complete and statically determinate definition of the position of the part within the fixture. They are used when the part does not have sufficient rigidity to withstand the operating forces without distortion.

In mechanical language the intermediate supports are redundant, and to avoid displacing or distorting the workpiece, they must be compatible with the locators, as was explained in the text to Fig. 6-37. An ideal solution would be to make them of a soft plastic material which would yield on contact with the part and freeze solid after the part has been located. This solution is generally hypothetical (except in such unusual cases as clamping with a cast-able metal, see Fig. 10-54), but it illustrates the principal mechanism of intermediate supports.

Intermediate supports are manually operated screws and plungers, and spring, wedge, air, and hydraulically operated plungers.

Method of Operation

Manually operated devices rely on the "feel" and judgment of the operator for the correct application pressure. Screw-type supports are thumbscrews, wing screws, knurled-head screws, hand-knob screws, and torque-head screws. Common to all of them is the fact that they are operated directly by hand and usually do not permit the use of a wrench except in very special cases, such as the one shown in Fig. 9-11. Thus, the operator is not deprived of the necessary "feel" for the proper contact pressure which enables him to avoid overloading the workpiece. The ideal screw-type device is the torque-head screw with a spring-loaded clutch built into the head which sets an upper limit for the transmitted torque and resulting pressure.

Springs for plungers are weak springs. Air and hydraulically operated plungers are designed to exert light pressures only. An intermediate support always has a locking device for rigid locking in the operating position. Screw-type supports have check nuts, while plungers are locked by means of a set-screw, a wedge, or a cam. Where possible, the locking device engages a tapered surface of the plunger so that the locking is positive and does not depend on friction only.

Screw-type supports are frequently provided with a swivel head. On irregular surfaces, this serves to equalize the support over a larger area; on previously machined surfaces it prevents marring the work with circular scratches. This can also be achieved by using a copper or nylon tip on the screw.

Commercial Components

Many individual components for intermediate support devices are commercially available. Several types of spring-loaded plungers are available as complete units (jacks) ready for mounting on the fixture base. Since they contain movable parts and are exposed to chips, they are protected by caps, shields, or seals in the manner shown in Fig. 8-10.

There is no very sharp distinction in the design between adjustable locators and intermediate supports. The two adjustable locators shown in Figs. 6-41 and 6-42 can also be used as intermediate supports.

Screw Type Supports

The screw type of intermediate support is the cheapest one to make, is rather slow to operate, and can only be used where there is convenient access for the operator's hand—in a side wall of the fixture. The plunger type, much more versatile, is therefore used extensively.
Plunger Type Supports

Figure 12-1a shows the simplest form of plunger support. It is provided with a helical spring beneath the plunger to press it against the work. One objection to this type of support is that the plunger A, will slip back under the pressure of the clamps or cutting tools bearing upon the work. There is also the danger of the milled flat on the plunger becoming clogged with dirt, so that it will not work properly. Considerable time is lost, therefore, in using this type of support. The method of clamping the plunger is also slow, as it is necessary to use a wrench in tightening or loosening the set-screw B. Shown in Fig. 12-1b is a support which is an improvement over that shown in diagram a. The flat on the side of plunger A is milled at a slight angle instead of parallel with the center line, as in diagram a. This prevents the plunger from slipping after it is clamped. A pressure shoe B—made of hardened drill rod, which is kept from turning by a small pin C, engaging a flat, milled in piece B—is used between the plunger A and the clamp. A wing nut D is fastened to the end of the screw, as shown, in order to eliminate the use of a wrench.

In Fig. 12-2 another design is shown, which presents a further improvement over those in Fig. 12-1. A bronze bushing B is driven into the base of the jig and allowed to project above the base. Plunger A is a sliding fit in the bushing. A cap C is driven onto the end of the plunger and extends down over the outside of the bushing, as indicated, making the support dirt-proof. This device, however, as well as those in Fig. 12-1, is not entirely satisfactory since it will shift as it is tightened, although when tightened, it will remain in position.

Fig. 12-1. Simple plunger type intermediate supports.

Two other improved designs are shown in Fig. 12-3, diagrams a and b. The end of the pressure shoe in diagram a is machined to fit almost half-way around the cylindrical surface of the plunger. This increases the pressure-transmitting surface and permits the use of a much larger locking pressure, resulting in a rigid locking of the plunger. The device in diagram b is an example of the use of the 45-degree wedge-end plunger which produces a much smaller side load on the plunger than the pressure shoe B, in Figs. 12-1b and 12-2.

Fig. 12-2. A plunger type intermediate support with bushing and cap.

Fig. 12-3. Improved plunger type intermediate supports.

An intermediate support of the wedge-operated plunger type is shown in Fig. 12-4. It represents a modification of, and an improvement upon, the adjustable locator shown in Fig. 6-40. The design of the device is low and is for use in cases where the plunger is located in the middle part of a fixture

and the actuating means must be accommodated in the fixture base. The plunger is adjusted up and down by the horizontal movement of the wedge. In Fig. 6-40 the wedge is located in a groove in the bottom of the fixture base and receives its support from the machine tool table. In Fig. 12-4 the wedge and its actuating rod are housed in a bore within the fixture base, resulting in a fully self-contained device, a stronger and more rigid fixture, and a more accurate operation of the wedge. The wedge is locked by clamping the actuating rod in the casting B, by means of a knurled nut which also serves as the handle for actuating the wedge. The wedge is supported on the lower edges of two holes in bushing A. For easy alignment, bushing A is made with a sliding fit and is secured in its proper position by screw C and dowel pin D. Full alignment is facilitated by the hinged connection between the wedge and the actuating rod.

Equalizers and Floating Supports

Intermediate supports can be combined with equalizers. An equalizer can carry two or more intermediate supports, or an intermediate support can carry an equalizer. Equalizers and "the floating principle" are used mainly where a plurality of intermediate supports are applied to a thin plate or rim. In the lathe fixture shown in Fig. 9-11 three intermediate supports are formed by shell bushings K with hook bolts F acting together as the jaws of a small vise. Each pair of jaws can float separately and adjust its position to the rim of the casting A after the casting has been located (centered) on the cone locator B. Finally, the three pairs of jaws are clamped separately in position without distorting the rim.

The fixture shown in Fig. 12-5 is for a combined lathe operation and includes three intermediate supports mounted on a common floating ring. The work A is to be bored, shouldered, and faced, complete in one setting. Because of its length, it was considered necessary to provide additional supporting points besides the jaw surfaces. A set of special jaws B was keyed to the sub-jaws in the table at D, with each special jaw shouldered at C to support the work. The brackets E are tongued at F to fit the special jaws and are secured by screws G. These brackets act as a support for the steel floating ring M in which the three spring-pins J are placed.
Elongated holes at points $N$ allow for the required floating action, as the ring is clamped by collar-head screws. Each bracket on which the ring rests is provided with a shelf $H$ which is offset slightly from center to allow the necessary width for the screws. In using the device, screws $L$ and $N$ are loosened, and the work is placed in the jaws, which are then tightened, while the ring floats sufficiently to allow for variations. It will be noted that the pins, being spring-controlled, adapt themselves to the casting and are locked there by screws $L$, after which the ring itself is clamped by the collar-head screws $N$.

Although the floating action of this device was satisfactory, the driving or gripping power was found insufficient to hold the work securely, thus it was necessary to replace the spring-pins with square-head set-screws, cup-pointed, and the ring was then tapped out to receive them. The ring was also allowed to float while these screws were lightly set up on the work, after which the clamping screws $N$ were tightened. After this change in construction, the action of the mechanism was much improved, and the driving power was found to be sufficient.

A different case of a floating intermediate support is shown in Fig. 12-6, also used for gripping a thin wall of a workpiece. The work has a narrow flange (at $x$) and clamp $A$ has a hook that grips over the flange. The rim of the part is clamped between point $x$ and the dog $C$ when wing nut $B$ is actuated. The left end of the clamp has an elongated hole that permits floating so that each clamp adjusts itself to irregularities in the shape of the part.

Fig. 12-6. A floating intermediate support applied to a thin wall.
Cutter Guides

Definitions and Principal Types

Cutter guides (setting gages, setting blocks, set-up gages) are used for correctly positioning the cutting tool relative to the work and thereby eliminating the necessity of taking trial cuts, measuring the part, resetting the cutter, etc. The cutter guides are usually small, flat or profiled blocks, or complete templates, and are permanently, or semipermanently, mounted on the fixture. When the cutter is correctly set, relative to the fixture, and the work is correctly located within the fixture, then the cutter is also correctly positioned relative to the work.

Cutter guides are used extensively on fixtures for: milling, planing, and shaping operations. Cutter guides for drilling and boring operations take the form of bushings and are described in Chapter 14. On lathe fixtures, flat cutter guides are used for positioning the cutter for facing cuts, while curved cutter guides with special curved feeler gages are needed for positioning the cutter for cylindrical turning operations. Turning cuts taken with cutters...
CUTTER GUIDES

Ch. 13

mounted on a turret do not require a cutter guide as the tools are already preset; other turning operations may require a higher accuracy than that obtained by using a cutter guide so that diameter measurements must then be taken. Fixtures for grinding operations do not employ cutter guides, but may be equipped with pre-positioned grinding-wheel dressers.

Tooling blocks with preset cutters do not, as a rule, require cutter guides. Cutter guides are always made of wear-resistant material, usually of hardened tool steel, but sometimes of tungsten carbide. They are mounted by means of screws and secured in position by dowel pins. They can also be manually held in position against a reference surface on the fixture. The reference surface of a cutter guide is usually set back a certain distance from the path of the cutter and the cutter is positioned against a feeler gage, or a gage block, placed on the reference surface. In this way the cutter does not have to come into contact with the reference surface which is a hardened precision surface, and both the reference surface and the cutting edge are protected against accidental damage as well as excessive wear.

It is a recommended practice to standardize the setback distance, which should be not less than 1/32 inch (0.8 mm). However, when the setback is not standardized, the required feeler gage dimension must be clearly marked at a place close to the reference surface. The principal types of cutter guides for single and multiple operations are shown in Fig. 13-1.

Cutter Guides for Milling Fixtures

An example of the use of a cutter guide in a milling fixture is shown in Fig. 13-2. The operation is the simultaneous milling of a contour consisting of seven parallel surfaces, by one gang of milling cutters. The side positioning is not critical and is done in the setup by direct measuring between the fixture and the milling cutters. The positioning for depth of cut must be repeated after each cutter grinding, and one cutter guide is provided, located in the center line of the fixture. In this case, the cutter guide is in the form of a button and is mounted by a press fit in the fixture base.

A cutter guide (set-up gage) for side positioning is shown in Fig. 13-3. The operation is the straddle milling of the flanged edges of a pinion bearing, shown in chain-dotted lines. The part is located on a cylindrical locator and is clamped between the locator and the modified movable jaw of a vise. This jaw also carries the cutter guide which is made 1/8 inch (3 mm) less in width than the distance between the two cutters. Consequently, the cutters can be positioned by means of one 0.0625-inch (1.50-mm)-thick feeler gage.

Another, more complicated, case involving the simultaneous positioning of three sets of milling cutters is shown in Figs. 13-4 and 13-5. Figure 13-4 shows the part and the fixture. The part is a pressure plate with three sets of lugs spaced 120 degrees apart. Each lug is milled on two sides, and the gang of cutters consists of one slitting cutter and two half side mills (Fig. 13-5). The operation is per-
formed on a special manufacturing type milling machine with three spindles operating simultaneously on the three lugs. The part is located on a circular locator (the supporting stud) designed for jam-free entering and is clamped on its periphery by three angular clamps simultaneously actuated by a floating cam for equalized clamping.

A cutter guide in the form of a stepped gaging lug is provided for each of the three places and the three cutter guides are mounted on a common dummy plate of the same dimensions as the part itself. The dummy plate is clamped in the fixture and the cutters are positioned, one set at a time. After removal of the dummy plate, the set-up is ready for production. With the dimensions shown, the mean thickness of a lug is 0.4965 inch (12.61 mm), leaving a total clearance of approximately 1/16 inch (approx. 1.5 mm) between cutters and gaging lug. The thickness of the feeler gage is selected to be 0.0625 inch (1.5 mm), and the reference surface on the lug must, therefore, be offset from 0.2505 inch (6.363 mm) to 0.2520 inch (6.501 mm), relative to the center line of the plate.

A cutter guide for the setting of a planer tool is shown in Fig. 13-6. The part is a lathe bed, and the cutter guide is formed as a template for the complete contour of the ways on top of the lathe bed.

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Drill Bushings

Definitions, Action, and Classifications

Bushings are used as cutter guides for drills, counterbores, reamers, and other cutting tools in the same category. They serve a triple purpose of positioning, guiding, and supporting the cutting tool.

The twist drill is a tool that is not well-suited for precision work. Its leading point, the chisel edge, has a rake angle of approximately minus 60 degrees. With a negative rake angle of this magnitude, metal removal is effected more by squeezing it away than by cutting. The chisel edge, therefore, is constantly exposed to a large axial cutting force. The angle between the two main cutting edges is 118 degrees (nominally, sometimes slightly less). An angle of this magnitude is not very effective in centering the tool. The shank of the drill is relieved with a slight “back taper,” so that there is no contact (and therefore no support) between the body of the drill and the walls in the hole in back of the two corners of the lips. The circular cross section of the drill is reduced by the two large flutes, leaving a cross section similar to that of an I-beam, and consequently, each cross section of the drill has one direction of low rigidity. In addition, the grinding of a twist drill is, for economical reasons, a very fast operation and, therefore, not a high-precision process. When starting the cut, the drill’s chisel edge has a tendency to “walk” on the surface before it starts to bite. Also, during cutting, the drill is sensitive to local variations in the hardness of the metal and may react by running out to the side of the softer metal. The result is that without taking proper precautions, holes drilled with a twist drill, are oversize, out of round, displaced, out of alignment, and not even straight. These deficiencies are greatly reduced and the quality of the work significantly improved by the application of a drill bushing so located that it provides positioning, guidance, and support to the drill at a point as close as possible to the surface of the work.

Bushings are constantly subject to wear when in use and must be made of wear-resistant material. No single fixture component offers such a large variety of types, shapes, and sizes, as bushings. USA Standard bushings are available in more than 50,000 different configurations, counting all differences in types, sizes, and individual dimensions. In comparison, the cross-reference conversion tables of other fixture components include approximately 14,000 items. Drill bushings are precision parts. They are commercially available at prices that are a fraction of what it would cost to make them individually. This fact has greatly contributed to the reduction in the cost of fabricating drill jigs.

Bushings can be classified as stationary press fit bushings and renewable (loose) bushings. The term “fixed bushings” for press fit bushings is not recommended, because it is used with a different meaning in the text of the USA Standard. By shape they can be classified as headless and head type bushings; by use, as liner and renewable wearing bushings. By basic design they are classified as conventional and special bushings. Conventional bushings are classified as standard and nonstandard bushings, depending on individual dimensions.

Standard bushings satisfy the majority, but not all, of the fixture designer’s needs. In some cases a standard bushing can be modified to suit special requirements; in other cases, it may be necessary for the fixture designer to design a nonstandard bushing. Empirical rules for such designs will be presented. Many of the statements that will be made in the following concerning standard bushings are actually of a general nature and apply also to nonstandard bushings.

Standard Bushings

Illustrations, data, and other information about USA Standard bushings have been extracted from
American Standard Jig Bushings (ANSI B94.33-1962, redesignation of B5.6-1962), with the permission of the publisher, The American Society of Mechanical Engineers, United Engineering Center, 345 E. 47th St., New York, N.Y. 10017. A condensed but comprehensive extract from this standard is found in Machinery's Handbook.¹

Standardized bushings are shown in Fig. 14-1 and the six basic types are shown in Fig. 14-2. They comprise press-fit and renewable bushings. Press-fit bushings are either liner bushings or press-fit wearing bushings, however, for the same diameter, liner bushings are shorter. Liner bushings are provided with and without heads and are permanently installed in a jig to receive the renewable wearing bushings. They are sometimes called "master bushings." Press-fit wearing bushings to guide the tool are for installation directly in the jig without the use of a liner and are employed principally where the bushings are used for short production runs and will not require replacement. They are also intended for use where the closeness of the center distance of holes will not permit the installation of liners and renewable bushings. Press-fit wearing bushings are made in two types, with heads and without.

Renewable wearing bushings to guide the tool are for use in liners which, in turn, are installed in the jig. They are used where the bushing will wear out or become obsolete before the jig, or where several bushings are to be interchangeable in one hole. Renewable wearing bushings are divided into two classes, "fixed" and "slip."

Fixed renewable bushings are installed in the liner with the intention of leaving them in place until they are worn out. Slip renewable bushings are interchangeable in a given size of liner and, to facilitate insertion or removal, they are usually made with a knurled head. They are most frequently used where two or more operations requiring different inside diameters are performed in a single jig, such as where drilling is followed by reaming, tapping, spot facing, countering, or some other secondary operation.

All standardized outside diameters are in multiples of 1/64 (0.0156) inch and all lengths are in multiples of 1/16 (0.0625) inch. The standard lengths of the press-fit portion of these jig bushings are based on standardized or uniform jig plate thicknesses of 5/16, 3/8, 1/2, 3/4, 1, 1 1/8, and 1 3/4 inches. Drill bushings in metric sizes are readily available from major bushing manufacturers, although not at this time considered shelf items.

Standard jig bushings are designated by the following system:

Inside Diameter:
The inside diameter of the hole is specified by decimal, letter, number, or fraction.

Type Bushing:
The type of bushing is specified by letters:
- **S** for Slip Renewable
- **F** for Fixed Renewable
- **L** for Headless Liner
- **HL** for Head Liner
- **P** for Headless Press Fit
- **H** for Head Press Fit

Body Diameter:
The body diameter is specified in multiples of 1/64 inch. For example, a 1/2-inch body diameter = 32/64 = 32.

Body Length:
The effective or body length is specified in multiples of 1/16 inch. For example, a 1/2-inch length = 8/16 = 8.

Unfinished Bushings:
All bushings with grinding stock on the body diameter are designated by the letter U following the number.

Example—

- **a. Inside Diameter Hole Size:**
  1. Decimal
  2. Letter
  3. Number
  4. Fractional

- **b. Type Bushing:**
  1. **S** — Slip Renewable
  2. **F** — Fixed Renewable
  3. **L** — Headless Liner
  4. **HL** — Head Liner
  5. **P** — Headless Press Fit
  6. **H** — Head Press Fit

- **c. Body Diameter:** 3/4 inch = 48/64 = 48

- **d. Body Length:** 1 inch = 16/16 = 16

Tolerance on fractional dimensions where not otherwise specified shall be plus or minus 0.010 inch.

The maximum and minimum values of hole size, A, shall be as follows:

<table>
<thead>
<tr>
<th>Nominal Size of Hole</th>
<th>Maximum, inches</th>
<th>Minimum, inches</th>
</tr>
</thead>
<tbody>
<tr>
<td>Above 0 to 1/4 in., incl.</td>
<td>Nominal + 0.0004</td>
<td>Nominal + 0.0001</td>
</tr>
<tr>
<td>Above 1/4 to 3/4 in., incl.</td>
<td>Nominal + 0.0005</td>
<td>Nominal + 0.0001</td>
</tr>
<tr>
<td>Above 3/4 to 1½ ins., incl.</td>
<td>Nominal + 0.0006</td>
<td>Nominal + 0.0002</td>
</tr>
<tr>
<td>Above 1½ inches</td>
<td>Nominal + 0.0007</td>
<td>Nominal + 0.0003</td>
</tr>
</tbody>
</table>

Diameter A must be concentric to diameter B within 0.0005 T.I.V. on finish ground bushings. The body diameter B, for unfinished bushings, is larger than the nominal diameter in order to provide grinding stock for fitting to jig plate holes. The grinding allowance is:

- 0.005 to 0.010 inch for sizes 5/32, 13/64, and 1/4 inch,
- 0.010 to 0.015 inch for sizes 5/16 and 13/32 inch, and
- 0.015 to 0.020 inch for sizes 1/2 inch, and up.

The length C is the overall length for the headless type and length underhead for the head type.

When renewable wearing bushings are used with liner bushings of the head type, the length under the head will still be equal to the thickness of the jig plate, since the head of the liner bushing will be countersunk into the jig plate. All bushings ranging from 0.0135 through 0.3125 inch will be counterbored to provide for lubrication and chip clearance. However, bushings without counterbore are optional and are furnished upon request. The size of the counterbore is the inside diameter of the bushing plus 1/32 inch. The included angle at the bottom of the counterbore is 118 degrees, plus or minus 2 degrees. The depth of the counterbore ranges from 1/4 inch for the smallest bushings to 5/8 inch for the largest bushings and is adjusted to provide adequate drill bearing length.

Bushings are straight both inside and out. Older literature shows jig bushings that are tapered on the outside but this can be considered obsolete. The upper corners, on the inside, are given a liberal radius (radius D in Fig. 14-2) to allow the drill to enter the hole easily, while the outer corners, at the lower end of the outside, are chamfered so that it is easier to drive the bushing into the hole when making the jig, and also to prevent the sharp corner on the bushing from cutting the metal in the hole into which the bushing is driven. In addition, it is recommended (but not standard practice) to relieve the outer surface at the lower end by 0.001 to 0.002 inch on the diameter, on a length equal to 1 1/2 to 2 times the length of the chamfer.

Bearing length for the drill within the bushing ideally should be a function of the drill diameter. A bearing length that is too short causes premature wear, while one that is too long causes excessive friction (the twist drill is never a precision tool!). If there are no overriding conditions, a bearing length of 2 times the drill diameter can be taken as a good, workable average. There are, however, other considerations, such as the need for sufficient length of press seat in the jig wall and for adequate thickness of the bushing head. An analysis of the standard tables shows a wide range of bearing length for each drill size. The ratio of bearing length (taken as the average of the table values for each drill size) to drill diameter varies from 7 for 1/4-inch-diameter drills to about 1 1/4 for 1-inch-diameter drills and down to 0.8 for large drills about 2 inches in diameter.

**Mounting of Bushings—Press Fit Bushings**

Stationary bushings of all types are mounted, as a general rule, by a press fit. In any press fit installation, metal is displaced and distortion occurs in the bushing and in the jig plate. It is recommended to always use the minimum interference necessary to safely retain the bushing in the jig plate. A diametral interference of 0.0005 to 0.0008 inch is adequate for the installation of headless press fit bushings and liners with 3/4-inch to 1-inch OD. For sizes from 1/2-inch to 3/4-inch OD, the use of 0.0003 to 0.0005-inch interferences is recommended; for sizes below 1/2-inch OD, an interference of 0.0002 to
Fig. 14-2. The six basic types of drill bushings.

Courtesy of ASME
0.0003 inch should be used. These values are for jig plates made of cast iron and low carbon steel.

Example—When a bushing of 1/2-inch ID, 3/4-inch OD, and 3/4-inch length is pressed into a 3/4 inch thick jig plate with an 0.0006-inch interference fit, the ID of the bushing will be reduced by approximately 0.0002 inch. The bore of the bushing is manufactured with a plus tolerance of 0.0001 to 0.0005 inch relative to the nominal drill size, and the compression of the bushing quoted above reduces the bushing diameter to almost exactly the nominal drill diameter. At the same time, the distortion of the jig plate is held to a negligible amount.

Head type press fit bushings and liners can be installed with 0.0003- to 0.0005-inch interference because the contact between the head and the surface of the drill plate provides additional support and rigidity in the assembly. Head type bushings are particularly recommended for installation in relatively thin jig plates where a headless bushing would require an excessive interference fit for adequate retention. The hole in the jig plate must be finished by means of a reamer, a jig borer, or a jig grinder, never with a twist drill. A twist drill cannot be relied upon to produce a hole of the required tolerance and roundness.

The OD tolerances on bushings were established for the purpose of matching holes finished with a chucking reamer. Reamers are commercially supplied with a plus tolerance and produce holes slightly larger than their own physical diameter.

Example—A 3/4-inch chucking reamer can be expected to measure 0.7505 inch when in good condition and to produce a hole very close to 0.7510 inch in diameter. With the standard tolerances of 0.7515 to 0.7518 inch on a 3/4-inch OD headless press-fit bushing, this leaves an interference of approximately 0.0005 to 0.0008 inch.

Particular precaution must be taken if two bushings are close together; the thin bridge of metal between them will yield excessively, and the bushings will “walk.” In such cases, it is recommended either to use bushings that are so large that the holes will blend together (and flatten the bushings on one side so that they can contact each other), or to make a special insert with two bushing holes.

A jig borer or jig grinder can hold tolerances to 0.0001 inch economically. A psychological peculiarity of some jig borer operators is that they sometimes develop the habit of working to the lower limit of the tolerances, resulting in interferences that are systematically on the high side.

The recommended method of installing press fit bushings is with an arbor press, but if the jig is too large, alternate methods must be considered such as pulling the bushing into place with a bolt. The bushing is first carefully started into its mounting hole; drilled pressure plates are placed both on the free end of the bushing and on the far side of the jig plate; a bolt is inserted, a nut is screwed on, and as the nut is gradually tightened, the bushing slides safely into place. When the bushing is too small to permit the use of this procedure, it can be carefully driven into place by means of a hammer. The hammer must never strike the bushing itself; a punch of soft metal such as aluminum, lead, or brass must be used as a cushion for each hammer stroke.

Before installing the bushing, the contacting surfaces must be carefully cleaned and lubricated. White lead is suitable for this and has the advantage of facilitating later removal of the bushing if replacement is needed. Recommended also is the handstoning of the leading ends of the bushing, the chamfer edge, and the edge of the relief.

Excessive interference may reduce the diameter of the bore to below a working clearance, which can cause tool seizure or prevent the insertion of a renewable bushing. An undersize bore must be relapped, and the operation must be performed with great care to prevent the bushing from becoming “bell mouthed.” Another method is to forcefully run a drill up and down in the bushing; this will either destroy the bushing or open it up; at any rate, it certainly damages the drill and is not a recommended procedure.

Distortion of a bushing as a result of excessive interference is essentially limited to the lower portion of the bushing. This is due to an “ironing” effect on the metal in the jig plate whereby small irregularities in the surface are squeezed down and the effective interference is reduced accordingly. Thus, a point in the surface at the top has been in contact with and “ironed” by the full length of the bushing, while points further down suffer less and less “ironing” effect due to the shorter length of bushing to which they are exposed.

Modern technology has provided means for eliminating problems associated with the use of an interference fit. An adhesive\(^2\) is now available that will bond a steel bushing in a clearance hole in a metal plate as securely as an interference fit. The surfaces are carefully cleaned, adhesive is applied,
the bushing is installed, and the assembly is left to cure for about four hours at room temperature. The curing time can be reduced to 15 or 20 minutes by the application of heat. Maximum temperature recommended is 250°F and the curing process can be performed in an oven or by locally applying heat from a heat lamp or other mild heat source.

For extremely precise location in the jig plate, the clearance should be 0.0001 to 0.0003 inch, resulting in a location accuracy of approximately 0.0001 inch. For average conditions, the clearance can be 0.0003 to 0.001 inch, and for noncritical conditions it is even possible to go to 0.003-inch clearance with good retention of the bushing.

**Installation of Bushings—Renewable Bushings**

When removable bushings are used, they should never be placed directly in the jig body, unless the jig is to be used only a few times. The hole, however, should always be provided with a lining that is made in the form shown in Fig. 14-3a. If the hole bored in the jig body receives a loose or removable bushing directly, its insertion and removal (if the jig is frequently used) would soon wear the walls of the hole, and in a short time, either the jig would have to be replaced, or at least the hole would have to be rebored and a new removable bushing made to fit the now larger-sized hole. In order to overcome this, the hole in the jig body is bored wide enough to receive a lining bushing, which is driven into place. This lining bushing, in turn, receives the loose bushing, the outside diameter of which closely fits the inside diameter of the lining bushing, as shown in Fig. 14-3b in which A is the jig body, B the lining bushing, and C the loose bushing. When no removable bushings are required, the lining bushing itself becomes the drill bushing or reamer bushing, and the inside diameter of the lining bushing will then fit the cutting tool used. The bushing may project, as shown in Fig. 14-3c, to provide the drill with the proper guidance and support close to the work. If the jig plate is thin, it can be locally increased in thickness by means of a boss, as shown.

Head type press-fit bushings are used to prevent the bushing from being pushed through the jig plate by the cutting tool. The most important application is to take the thrust of a stop-collar, which is clamped on the drill, to allow it to go down to a certain depth, as shown in Fig. 14-4a in which C is the stop-collar, D the wall of the jig, and E the press-fit bushing; F is the work. If the work to be drilled is located against a finished seat, or boss, on the wall of the jig, and the wall is not thick enough to take a bushing of standard length, then it is common practice to make a bushing having a long head, as shown in Fig. 14-4b. The length A of the head, can be extended as far as is necessary to get the proper bearing. As the bushing is driven into place and the shoulder of the head bears against the finished surface of a boss on the jig, it will give the cutting tool a bearing almost as rigid as if the jig metal surrounded the bushing all the way up.

Removable bushings (Fig. 14-4c) are frequently used for work which must be drilled, reamed, and tapped. Each of the cutting tools has its own bushing, and all these bushings have the same outside diameter so that they fit into the liner. There is a sliding fit so that they can be gently pressed into the liner by hand. These bushings are termed “slip bushings.” They are also used when different parts of the same hole are to be drilled out to different diameters, when the upper portion of the hole is counterbored, or when a lug has to be faced off. A slip bushing belongs to one tool only, and must not be interchanged. Slip bushings for drills and reamers are nearly the same size and cannot be recognized by sight alone. They must, therefore, be identified by conspicuous markings, usually the letter R, on a reamer bushing.

![Fig. 14-3. Mounting of liner bushings.](image)

![Fig. 14-4. Head type bushings.](image)
that is cut immediately under the head \( B \), and is also shown in Fig. 14-2. Although its dimensions are not standardized, the groove is important because it provides clearance for the grinding wheel. A width of 0.080 inch is suitable. Occasionally, a bushing having a large outside diameter is required as, for example, when a large counterbore must be used in a small hole, which makes it necessary to have a large opening in the jig body. If several operations with tools of different diameters are required, then all the slip bushings for these tools must be made with the same large outside diameter.

Slip bushings as well as other renewable bushings must be secured (locked) in place; otherwise they may rotate with the tool, wear rapidly on the outside, or be forced out by the chips. A number of design details for such locking devices have been developed. Their common requirements are that they must be safe, effective, simple, foolproof, chip-proof, and, preferably, inexpensive.

A very strong, safe, and effective device is that shown in Fig. 14-5a. A collar with a projecting tail, called a “dog,” is pressed fit around the head of the bushing and is bent at the end of the tail, with one end resting against some part of the jig. The tail is sometimes left straight, if there is a possibility for the tail to strike against a lug in the same plane. Making such dogs involves some extra expense, but they are very effective in avoiding troubles with bushings turning and working their way out of the holes. The tail also provides a convenient grip for placing and removing the bushing. Large bushings may be provided with two handles for this purpose.

One solution, which is also inexpensive, is to work a semicircular groove \( B \) in the edge of the head to fit over a pin driven into the jig plate, as shown in Fig. 14-5b. Although it is effective against rotation of the bushing it does not prevent lifting by chips, nor does it facilitate the removal of the bushing. The arrangement shown in Fig. 14-5b is commonly used for making bushings more easy to remove. A step \( A \) is turned down on the head, which, in this case, will have to be a trifle larger in diameter. This step permits a tool—a screwdriver, for instance—to be placed underneath, and with a quick jerk the bushing may be lifted enough to offer a good hold.

Three methods of holding bushings to prevent them from turning are shown in Fig. 14-6; all of them on the same principle described. \( A \) shows a bushing with a pin inserted which then slips into a slot cut in the lining bushing; \( B \) shows a bushing with a slot milled through the collar and a pin is located in the jig to engage this slot; and \( C \) illustrates a more elaborate device that is sometimes used, where the stop button which is fastened to the jig prevents the bushing from being drawn out of the liner while drills or reamers are withdrawn, as well as preventing it from turning.

Standardized bushing locking components have been developed for fixed renewable type bushings and for slip bushings. The fixed renewable bushing

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**Fig. 14-5. Special bushing details.**

**Fig. 14-6. Devices for preventing a bushing from rotating.**
is provided with a partly circular recess as shown in Fig. 14-7 and is held in position against rotation and push-out by a lock screw with its head engaging into the recess. Almost any type of screw could be used as long as it does not have a countersunk head; however, using a screw with a cylindrical head of substantial dimensions, such as a socket-head cap screw is recommended. Preferably a shoulder screw of the type to be seen in Fig. 14-10, is recommended, as the shoulder absorbs some of the bending moment from the head. Head bushings without the lock screw recess can be locked by means of a separate ring-shaped clamp (Fig. 14-8) that covers the flange of the bushing and is provided with a recess for a standard socket-head locking screw.

These locking devices require the removal of the lock screw before the bushing can be removed and are therefore not suitable for slip bushings that have to be changed quickly. The standardized locking device for slip bushings consists of a bayonet-type combination of a rounded notch and a curved recess, as shown in Fig. 14-9, and works with the lock screw shown in Fig. 14-10. The bushing is inserted and locked with a push and a twist. The notch clears along the head of the lock screw and the bottom of the recess slides under the head of the screw and locks the bushing. The bushing is kept in place by the friction from the tool as it rotates. In those rare cases where a left running tool is used, the recess must be located in the opposite direction.

Lock screws are only suitable for use with fitless mounted (headless or countersunk head type) liners, usually in light-duty applications. For heavy-duty applications clamps provide a better means of locking the bushing against the effects of vibration and torque from rotation. Clamps provide a larger bearing surface against the jig plate and are secured by standard socket-head cap screws. The bending moment on the screw is also greatly reduced. Clamps can also be used for locking removable bushings in projected mounted liners, that is, head type liners where the head is not countersunk. Typical
**FIXED RENEWABLE BUSHINGS**

**FLUSH MOUNTED (HEAD OR HEADLESS LINERS)**

<table>
<thead>
<tr>
<th>LOCK SCREW</th>
<th>ROUND CLAMP</th>
</tr>
</thead>
<tbody>
<tr>
<td>• RECOMMENDED FOR LIGHT DRILLING APPLICATIONS</td>
<td></td>
</tr>
<tr>
<td>• SMALL HEAD DIAMETER PERMITS CLOSE BUSHING PLACEMENT</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>ROUND END CLAMP</th>
</tr>
</thead>
<tbody>
<tr>
<td>• FOR LOCKING STANDARD FIXED RENEWABLE BUSHINGS</td>
</tr>
<tr>
<td>• PROVIDES LARGE BEARING SURFACE AGAINST JIG PLATE</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>FLAT CLAMP</th>
</tr>
</thead>
<tbody>
<tr>
<td>• PROVIDES MAXIMUM SECURITY AGAINST VIBRATION AND TORQUE</td>
</tr>
</tbody>
</table>

**PROJECTED MOUNTED (HEAD LINERS ONLY)**

<table>
<thead>
<tr>
<th>ROUND END CLAMP</th>
</tr>
</thead>
<tbody>
<tr>
<td>• FOR LOCKING STANDARD FIXED RENEWABLE BUSHINGS IN PROJECTED MOUNTED LINERS</td>
</tr>
</tbody>
</table>

**SLIP RENEWABLE BUSHINGS**

**FLUSH MOUNTED (HEAD OR HEADLESS LINERS)**

<table>
<thead>
<tr>
<th>LOCK SCREW</th>
<th>ROUND END CLAMP</th>
</tr>
</thead>
<tbody>
<tr>
<td>• RECOMMENDED FOR LIGHT DRILLING OPERATIONS</td>
<td></td>
</tr>
<tr>
<td>• SMALL HEAD DIAMETER PERMITS CLOSE BUSHING PLACEMENT</td>
<td></td>
</tr>
</tbody>
</table>

**PROJECTED MOUNTED (HEAD LINERS ONLY)**

<table>
<thead>
<tr>
<th>ROUND END CLAMP</th>
</tr>
</thead>
<tbody>
<tr>
<td>FOR ANY APPLICATION USING SLIP RENEWABLE BUSHINGS INSTALLED IN PROJECTED MOUNTED HEAD LINERS</td>
</tr>
</tbody>
</table>

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Examples of clamp locks are shown in Fig. 14-11; although they are commercially available, they are not standardized.

Lock screws must be accurately located at the correct distance (dimension R in Fig. 14-9) from the liner axis. Small drill jigs for this purpose, shown in Fig. 14-12, are commercially available. Lock screws are eliminated by the use of liners with integral locking tabs\(^3\), as shown in Fig. 14-13. The configuration of the tab is similar to a part of a standard lock screw, so that it engages in the locking recess of a standard slip bushing. In every other respect the liner is standard; it is pressed in place by means of an adapter arbor that centers in the bushing and has a milled slot for clearing the lock tab.

\(^3\)Proprietary to the American Drill Bushing Company.
Nonstandard Bushings

These are bushings of conventional configuration but some or all dimensions deviate from standardized dimensions. They are used where the workpiece presents dimensional problems for the design of the jig.

The theoretical minimum center distance between holes is equal to the outside diameter of the bushing. However, the practical distance must be greater to allow for a metal wall between adjacent bushings and practical considerations require a certain minimum wall thickness to avoid uneven distortion when bushings are pressed in place. The center distance can be reduced if the bushing wall is reduced, and thin-wall bushings of the basic types are available. The wall thickness is approximately half the wall thickness of standard bushings in the normal series. "Extra thin" wall bushings are also available.

When the guide bushings are very long, and, consequently, would cause unnecessary friction in their contact with the cutting tools, they may be recessed, as shown in Fig. 14-14a. The distance \( H \) of the hole in the bushing is recessed sufficiently wider than the diameter of the tool so as not to bear on it. The length \( L \) is about twice the diameter of the hole, to provide guiding surfaces for the cutting tool which are long enough to prevent its running out. If the outside diameter of the bushing is very large compared to the diameter of the cutting tool, as indicated in Fig. 14-14b, the expense of making the bushings may be reduced by making the outside bushing of cold-rolled steel or cast iron and inserting a hardened tool-steel bushing, mounted with a press fit. This bushing can be considered as a standard press fit bushing. The reason why a bushing may need to have so large an outside diameter and so small a hole is that it might be necessary to remove it for counterboring part of the already drilled small hole by a counterbore of large diameter, in which case the hole in the jig body has to be large enough to accommodate the large counterbore. If a slip bushing is longer than the lining bushing, as illustrated in Fig. 14-14c, it will be advantageous to make the projecting portion of the bushing about 1/32 inch (0.8mm) smaller in outside diameter than that part of the loose bushing which fits the lining bushing. This lessens the amount of surface which must be ground, and, at the same time, makes it easier to insert the bushing, forming a point, so to speak, which will first enter the lining bushing. This does not interfere in any way with the proper qualities of the bushing as a guide for the cutting tool.

In some cases, the holes in the piece to be drilled are so close to one another that it is impossible to find space in the jig for lining bushings. It is then necessary to make a leaf, a loose wall, or the entire jig, of machine steel or tool steel and harden the entire jig or a portion of it.

Removable bushings are sometimes threaded on the outside and made to fit a tapped hole in the jig, as shown in Fig. 14-14d. The lower part of the bushing is usually turned straight, and ground in order to center it perfectly in the hole in the jig. The head of the bushing is either knurled or milled hexagon for a wrench. When these bushings are used, they are not, as a rule, used for the purpose of
guiding the cutting tool alone, but frequently combine the functions of locating and clamping of the work as well. Examples are shown in Figs. 9-14, 9-15, and 9-16. These bushings are not commonly used as slip bushings, as it would take considerable time to unscrew, and to re-insert into the jig body, a bushing of this type.

**Drills with Attached Bushings**

When machining several small holes requiring two or more operations, changing slip bushings becomes relatively time-consuming. Considerable time is saved by attaching them to their respective tools, so that they participate in the rotation and feed. Slip bushings used for this purpose are without heads and are termed "guide" bushings. Figure 14-15 shows a guide bushing A attached to a drill. The free overhang of the drill must at least be equal to the depth of the hole to be drilled and should not exceed one inch in order to maintain the rigid support of the drill point, particularly when drilling a rough surface. Since the guide bushing is rotating against the liner, a clearance must be provided. To minimize the amount of frictional heat developed, this clearance should be made as large as permitted by the required accuracy in the hole location. In drilling steel, the use of a guide bushing offers the advantage of providing plenty of room for curled chips.

A technique that is widely used in connection with multiple-spindle drill heads is shown in Fig. 14-16. A Z-shaped bracket is bored to the size of the drill (or other tool) and is machined on the outside to form a pilot which enters a drill bushing as the spindle is fed towards the work.

Some drill presses have provision for the mounting of a bushing bracket carrying a drill bushing concentric with the machine spindle. The bracket with the bushing is adjusted to a height slightly above the surface of the work. The bushing guides and supports the drill, and the work is clamped or held in a positioning fixture on the drill press table.

A combination of these techniques is found in the aircraft industry and is applied to portable power tools used for drilling, spot facing, tapping, etc., of small holes in large parts. The drill jig is made of relatively thin metal, fiber plate, or plastic laminates and neither the drill jig, the bushings, nor the drills are capable of guiding and stabilizing the rather heavy portable power tool. The operation of this type of equipment is shown in Fig. 14-17a. The bushings in the drill jig are liners. They are secured with a nut on the far side of the jig and are provided, directly or indirectly, with two locking prongs on the forward side. The drill bushing is a long and heavy bushing (the tip) that is screwed into the nose of the power tool; it carries a flange with two projecting locking lugs matching the prongs on the jig bushings. In essence the tip is a slip bushing mounted on a power tool; when in use the tip is pushed all the way into the liner and the power tool is rotated counterclockwise so that the lugs engage the prongs. The power tool is now positioned and is so well supported that it takes little effort on the part of the operator to hold it up and operate it. The drill is rotated and fed through the tip onto the work.

The locking prongs can be integral with the jig bushing and for light-duty work they can be made of two ordinary lock screws of the type shown in Fig. 14-10. Individual lock buttons can be used instead of lock liner bushings where the holes are spaced too closely. When holes are closely spaced
in a line, locking is done by two undercut locking strips along the line of holes as shown in Fig. 14-17b.

**Empirical Formulas for Design of Bushings**

Very wide, very long, and very large bushings are the three most common types of nonstandard bushings that must be individually designed and specially made. Very wide bushings are for sequences of operations such as those shown in Fig. 14-18, where one hole is drilled to two diameters, or where a drilling operation is followed by counterboring, countersinking, or spot facing to a diameter significantly larger than the hole diameter; perhaps followed by reaming. The diameter of the liner is slightly larger than the diameter of the counterboring tool. The drill bushing fits the liner with the sliding fit for slip bushings. The hole diameter in the bushing is the drill diameter with normal clearance, and other dimensions on this bushing can be calculated by the formulas below. The counterbore requires no bushing since it is guided by the pilot in the drilled hole. The reamer bushing differs from the drill bushing only in hole diameter.

Very long bushings are of one of the types shown in Fig. 14-4b and Fig. 14-14. The letter symbols used in the following formulas are the standard letter symbols from Fig. 14-2 plus the following: Bearing length = L, Wall thickness = T = 1/2 (B-A) and Flange width on head = G = 1/2 (E-B).

The basic dimension is the hole diameter A. According to an old rule-of-thumb, the bearing length can be taken as

\[ L = 2A \]

This bearing length will generally work well. It is long enough to provide sufficient bearing surface but not long enough to cause excessive friction. However, it is larger than is needed for large drills, while small drills, from 1/4-inch (6-mm) diameter on down, can well use a longer support. A more sophisticated approach would be to take:

<table>
<thead>
<tr>
<th>Inch Dimensions</th>
<th>Millimeter Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>[ L = \sqrt{A} + 0.4 \text{ inch} ]</td>
<td>[ L = 5 \sqrt{A} + 10 \text{ mm} ]</td>
</tr>
<tr>
<td>[ L = 0.8 \sqrt{A} + 0.4 \text{ inch} ]</td>
<td>[ L = 4 \sqrt{A} + 10 \text{ mm} ]</td>
</tr>
</tbody>
</table>

for tools with sharp edges, such as drills; and

For minimum wall thickness, take:

\[ T = 0.2 \sqrt{A} + 0.04 \text{ inch} \]

\[ T = \sqrt{A} + 1 \text{ mm} \]

Body diameter will then be:

\[ B = A + 2T, \text{ or} \]

\[ B = A + 2T + 1/32 \]

\[ B = A + 2T + 0.8 \text{ mm} \]
depending on whether or not the bushing is counterbored (Fig. 14-14a.)

For the corner radius at the inlet end, take:

\[ D = 0.1125 \sqrt[3]{A} \text{ inches} \quad D = \sqrt[4]{A} \text{ mm} \]

For the normal height of the head, take:

\[ F = 0.6 \sqrt{A} \text{ inches} \quad F = 7^{4/3} \sqrt{A} \text{ mm} \]

This value does not apply to bushings of the type shown in Fig. 14-4b.

**Inch Dimensions**

For the diameter of the head, take:

\[ E = B + F - 1/8 \text{ inch} \quad \text{for } B \leq 1/2 \text{ inch, and} \]

\[ E = B + F - 1/16 \text{ inch} \quad \text{for } B > 1/2 \text{ inch} \]

where \( F \) is calculated by the previous formula for inch dimensions.

For the length of seat, take:

\[ B \leq C \leq 3B \quad \text{for } B \leq 3/8 \text{ inch} \]

\[ 0.67B \leq C \leq 3B \quad \text{for } 3/8 < B \leq 3/4 \text{ inch} \]

\[ 0.6B \leq C \leq 2B \quad \text{for } B > 3/4 \text{ inch} \]

The range of values for \( C \) corresponds approximately to the values in the ANSI Standard. Calculated diameters are converted to multiples of 1/64 inch and calculated lengths are converted to multiples of 1/16 inch. Other calculated dimensions are converted to the same selection of fractions as are used in the ANSI Standard.

The following fits based on ANSI Standard Tolerance limits (from ANSI B4.1-1967), are recommended:

- for liner in jig plate (press fit) H7-n6
- for slip renewable bushing in liner F7-m6
- for fixed renewable bushing in liner F7-h6

Tolerance limits for H7, n6, and h6 are found in reference books, such as *Machinery's Handbook*; tolerance limits for F7 and m6 are found in ANSI B4.1-1967, Appendix I, p. 16 and p. 19.

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Example—Calculate dimensions of 3/4-inch diameter drill bushings of the types shown in Figs. 14-4b and 14-14a.

\[ A = 0.750 \text{ inch}; \sqrt{A} = 0.8660; \]

\[ \sqrt[3]{A} = 0.9086; \sqrt[4]{A} = 0.9306 \]

**Bearing Length:**

For Fig. 14-14a

\[ L = \sqrt{0.750 + 0.4} = 1.2660 \approx 1 5/16 \text{ inches} \]

**Corner Radius:**

\[ D = 0.1125 \sqrt[3]{0.750} = 0.1022 \approx 3/32 \text{ inch} \]

**Total Length:**

For Fig. 14-4b

\[ L + D = 1.2660 + 0.1022 = 1.3682 \approx 1 3/8 \text{ inches} \]

**Wall Thickness:**

\[ T = 0.2 \sqrt{0.750} + 0.04 = 0.2132 \text{ inch} \]

**Body Diameter:**

For Fig. 14-4b

\[ B = 0.750 + 2 \times 0.2132 = 1.1764 \approx 1 1/2 \text{ inches} \]

1 1/2/64 = 1 3/16 inches

For Fig. 14-14a

\[ B = 0.750 + 2 \times 0.2132 + 1/32 = 1.2077 \]

1.2077 \approx 1 7/32 \text{ inches} \]

**Height of Head:**

For Fig. 14-14a

\[ F = 0.6 \sqrt{0.750} = 0.5584 \approx 9/16 \text{ inch} \]

**Diameter of Head:**

\[ E = 1.2077 + 0.5584 - 1/16 = 1.7036 \]

1.7036 \approx 1 45/64 \text{ inches} \]

**Length of Seat:**

\[ 0.6 \times 1.2077 \leq C \leq 2 \times 1.2077 \]

\[ 0.7246 \leq C \leq 2.4154 \]

\[ 3/4 \text{ inch} \leq C \leq 2 3/8 \text{ inches} \]
**Millimeter Dimensions**

For the diameter of the head, take:

\[ E = B + F - 3 \text{ mm for } B \leq 13 \text{ mm and} \]
\[ E = B + F - 1.5 \text{ mm for } B > 13 \text{ mm} \]

where \( F \) is calculated by the previous formula for millimeter dimensions.

For the length of seat, take:

\[ B = C \leq 3B \text{ for } B \leq 10 \text{ mm} \]
\[ 0.67B \leq C \leq 3B \text{ for } 10 \text{ mm} < B \leq 19 \text{ mm} \]
\[ 0.6B \leq C \leq 2B \text{ for } B > 19 \text{ mm} \]

For millimeter dimensions, the following fits based on ISO Recommendation R286, are recommended:

- for liner in jig plate (press fit) \( \text{H7-n6} \)
- for slip renewable bushing in liner \( \text{F7-m6} \)
- for fixed renewable bushing in liner \( \text{F7-h6} \)

The tolerance limits are found in reference books, such as *Machinery's Handbook*.\(^5\)

**Example**—Calculate the dimensions of 19-mm-diameter drill bushings of the types shown in Figs. 14-4b and 14-14a.

\[ A = 19 \text{ mm}; \sqrt[3]{A} = 4.36; \sqrt[4]{A} = 2.67; \sqrt[4]{A} = 2.09 \]

**Bearing Length:**

\[ L = 5\sqrt{19} + 10 = 31.8 \approx 32 \text{ mm} \]

**Corner Radius:**

\[ D = 3\sqrt{19} = 2.67 \approx 3 \text{ mm} \]

**Total Length:**

\[ L + D = 32 + 3 = 35 \text{ mm} \]

**Wall Thickness:**

\[ T = \sqrt{19} + 1 = 4.36 + 1 = 5.36 \approx 5.5 \text{ mm} \]

**Body Diameter:**

For Fig. 14-4b

\[ B = 19 + 11 = 30 \text{ mm} \]

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\(^5\) Ibid., pp. 1532 to 1537.
made of cast iron, but only when the cutting tool is so designed that no cutting edges come within the bushing itself. For example, bushings used simply to support the smooth surface of a boring-bar or the shank of a reamer might, in some instances, be made of cast iron. But hardened steel bushings should always be used for guiding drills, reamers, taps, etc., when the cutting edges come in direct contact with the guiding surfaces. If the outside diameter of the bushing is very large, as compared with the diameter of the cutting tool, the cost of the bushing can sometimes be reduced by using an outer cast-iron body and inserting a hardened tool-steel bushing, as seen in Fig. 14-14b.

**Special Bushings**

Many jigs are now made of materials other than steel and cast iron. The commonly used materials in this category are cast or laminated plastics, plastic or fiber sheets, aluminum and magnesium sheets, and tooling plate. These materials do not permit conventional press fit mounting of bushings.

In jigs made of cast or laminated plastics, bushings are either cast in place as the jig is made, or potted in cavities that are formed by cores as the jig is laminated. In either case, the bushings are made with an outer surface texture, or pattern, that permits the plastic to grip and lock the bushing. In each case the surface configuration is such that the bushing is positively locked against rotation as well as against axial displacement. These patterns are a combination of deep circular grooves with shallow and sharp longitudinal serrations or polygonal flanges, or are systems of two sets of V-grooves crossing each other to form a pattern of diamond-shaped projections similar to a knurled surface with a coarse pitch. Samples of these patterns are shown in Fig. 14-19.

These bushings are also available with a 0.030-inch (0.8-mm)-thick ceramic coating to act as a heat barrier for the protection of the potting or bonding material against the frictional heat developed within the bushing. Other jig materials, such as aluminum, magnesium, Masonite®, and even wood, which is occasionally used for lightweight jigs, can be equipped with bushings that are specially designed for press installation. The lower half of the bushing is cylindrical and precision ground for locating the bushing in the mounting hole. The upper half is larger in diameter and has longitudinal serrations (see Fig. 14-19). The step in diameter prevents axial displacement, and the serrations cut into the jig material and lock against rotation. Template bushings are those used for template tooling, that is, jigs made from metal plates in thicknesses from about 1/16 to 3/8 inch (1.5 to 10 mm). This jig material is too thin for conventional press fit and for serrated bushings. Template bushings also require a fastener, which can be a nut or a deformable fastener. An example is shown in Fig. 14-20. The hole in the template (the jig plate) is drilled, reamed to 0.001 to
0.003 inch (0.03 to 0.08 mm) oversize, and countersunk 90 degrees. The bushing is placed in the hole, and an aluminum locking ring is crimped around it and into the groove. The crimping is done with a special adapter arbor in an arbor press, or with a rivet gun. Bushings of this type can be removed by cutting the locking ring, and then can be reused.

Circuit board bushings are small bushings for drills of 1/4-inch (6-mm)-diameter, down to #80 (0.35mm) for the drilling of circuit boards. With standardized hole diameters, they are available in a variety of outside configurations to match the circuit board drilling machines currently on the market (see Fig. 14-21).
Design of Fixture Bodies

Drawings

At this stage the choice of locating and clamping devices and intermediate supports, if these are required, has been finalized. A drawing is made showing these devices in their correct position relative to the part. The part outline is also shown. Using a color code for the lines, to differentiate the various items, is helpful.

Tooling holes are those drilled in the part for the purpose of locating it in a fixture or in a series of fixtures, one after another. All other dimensions are directly or indirectly referenced to the tooling holes. Where tooling holes are used, they must be shown and identified.

The drawing, as it now stands, is a phantom drawing with the details floating unsupported; the next step is to outline the fixture body so that it connects all loose parts and satisfies several other requirements as well.

The Use of Existing Components

Regardless of the type of fixture, the first design step is to examine the possibility of using existing equipment or components. The most promising of these is to use a manually or air-operated machine vise, with jaw inserts. The vise can be used as a base for milling and planing fixtures and for drill jigs, in which case it is also provided with a jig plate. In the case of a drill jig, the next possibility is the universal drill jig, or "pump" jig. It supplies the jig structure and clamping mechanism and needs only to be provided with locating devices and a jig plate with bushings. The use of this type of jig is described in detail in Chapter 21. The third possibility, applicable to box-type drill jigs, is the use of a commercially available jig box, an example of which, equipped here as a leaf jig, is shown in Fig. 15-1. These boxes are made of cast iron or of aluminum with cast-iron corner posts, and are available in sizes up to 4 by 8 inches (100 by 200 mm), and 6 by 6 inches (150 by 150 mm). Plain fixture bases are made in the two styles shown in Fig. 15-2. The difference lies in the location of the lugs. They are applicable to fixtures and jigs of many types, and are available in sizes up to 12 by 18 inches (300 by 450 mm). In any case, the final make-or-buy decision is based on a cost estimate. If commercial components do not fit, however, the design procedure continues.

Drill Jigs

In the case of a drill jig, the designer has three choices: A plate jig, an open jig, or a closed jig. If all holes are parallel and drilled from one side, and the part is large and stable, the plate jig is the probable solution. If all holes are drilled from one flat surface, the plate jig takes the simplest possible form, a flat plate. If the holes are located in surfaces at different levels, the designer has the choice of making a flat plate jig with projecting bosses of varying lengths, or to form the plate with bends and offsets so that it follows the contour of the part.
surface. The first possibility is usually recommended, except in extreme cases.

If the part is small and not easily supported, and all holes are parallel and drilled from one side, an open jig is the solution. If there are holes in more than one direction, a closed box jig is needed. The design procedure for these two cases is described in detail in Chapter 18, Drill Jigs. All drill jigs, with the exception of plate jigs, must have feet which can be either attached to the jig body, or integral with the jig body. The preferred forms of integral feet are the L and T; their overall dimensions must be large enough to bridge the width of the slots in the machine tool table.

### Fixture Clamping

All other types of fixtures require a fixture base that is aligned with and clamped to the machine tool table. The fixture base must always have slots, not holes, for the clamping bolts so that the nuts do not have to be completely unscrewed to remove the fixture. The traditional form for the clamping bolts is the T-bolt or screws with T-nuts, which must slide all the way to the end of the table to be removed from the T-slots. There are commercially available clamping nuts and bolts, however, that can be lowered into the T-slot and rotated into the gripping position.

### Alignment

Alignment is obtained in principle by means of a key in the fixture and an alignment slot in the machine table. While T-bolts and nuts must have a loose sliding fit in the T-slot, a key must have a close fit in its slot, and the fixture designer must have the data for the dimensions of alignment slots in the machine tables for which he is designing the fixtures. Most machine tables do not have separate alignment slots, but the T-slots serve both purposes. Since the clearance in a T-slot may vary, it is a firm rule that a fixture must align against only one side of the T-slot. Several types of combination clamping and aligning devices are commercially available. A representative example of an aligning clamp that is not proprietary is shown in Fig. 15-3.

The construction of the clamping device is as follows: Fitting into the conventional T-slot in the
machine table is a hardened and tempered, cast-steel locating T-block of rectangular cross section with one edge of its tongue machined to provide a flat positioning ledge inclined at an angle of 60 degrees from the horizontal plane. The block should be a snug sliding fit within the T-slot, with clearance kept as small as possible. A standard, square-headed clamping bolt is inserted through a hole in the center of the T-block, with the head of the bolt fitting closely into a slot machined across the underside of the T-block.

A cylindrical cam-sleeve with a sliding fit in a hole bored through the lug of the fixture is drilled to provide a clearance of at least 1/32 inch (0.8 mm) for the clamping bolt. Two flats are machined on the lower end of this sleeve, with the flat on the left machined so that it will be in vertical alignment with the left-hand edge of the T-block. The lower portion of the flat on the opposite side of the sleeve is inclined at an angle of 60 degrees, as shown, to mate with the positioning ledge on the T-block. Both the left-hand flat and the right-hand inclined surfaces should be hardened and polished, since they are the parts most subject to friction and wear. A clearance of 1/16 inch (3 mm) should be provided between the lower face of the sleeve and the top surface of the T-block, as indicated.

The upper end of the cam-sleeve, which projects beyond the top face of the fixture lug, is provided with a fine-pitch external thread to accommodate the circular ring-nut. The periphery of the ring-nut is knurled to facilitate manual rotation. A standard hexagonal lock-nut is screwed on the upper, threaded end of the clamping bolt.

In the illustration, the fixture and parts of the clamp are shown in the correct relative positions they will take when the fixture has been properly located and clamped to the machine table. The cam sleeve has been moved downward by simply tightening the lock-nut. This movement causes the inclined flat on the right-hand side of the sleeve to contact the positioning ledge on the T-block, thus pressing the fixture toward the left until it is stopped by the flat on the left-hand edge of the sleeve coming into contact with the side of the T-slot in the machine table. Before tightening the lock-nut, the ring-nut should be backed off slightly to clear the top surface of the fixture lug and allow the sleeve to pass through the hole in the lug.

When the fixture is aligned, the ring-nut is tightened—by hand pressure only—and the lock-nut is then given a final, partial turn to insure rigid clamping and positive alignment of the fixture with relation to one side of the T-slot. The cam-sleeve is prevented by the ring-nut from pressing too forcibly against the positioning ledge on the T-block.

The Fixture Body

The body of the fixture is now built up from the base. It consists, essentially, of walls, forming a channel or a complete box (a pot, in the case of a lathe fixture), or of individual uprights or brackets. Again, the phantom drawing shows where material is needed.

The principal consideration, apart from rigidity and strength, in the application of material is clearance. At this stage, clearance is easily arranged; later, it may be unavailable. Clearance is required at the following places: Around the part to allow for dimensional tolerances, around the path of the part as it is being loaded and unloaded, and around the fixture to prevent collision with any part of the machine. Wherever the operator's hand is applied, a finger clearance of at least 5/8 inch (16 mm) must be provided; more, if a full hand-grip on the part is anticipated. No part of the fixture should obscure the view of the cutter's action area. If possible, the locating areas should be visible. Windows in side walls may be needed that also serve to reduce the weight and to provide access for chip cleaning and for the free flow of cutting fluid. Inaccessible pockets that can accumulate chips must be avoided. Projecting points, corners, and edges are a hazard to the operator; these must be blunted and rounded.

Fixtures that require little handling are made of steel or cast iron; those that require a great deal of handling are made of a selection of lightweight materials now available, such as: aluminum, magnesium, cast or laminated plastics, and plastic or fiber sheet.

Three Construction Principles

A fixture body may be of the built-up type, a casting, or a weldment. Historically, the first two types have been used from the inception of the use of fixtures, with cast fixtures dominating; the arrival of the welding process, particularly arc welding, has changed the picture, and today the welded fixture is the dominant type. It has, however, not completely eliminated the two older types for each has its advantages and therefore its limited area of application, therefore a discussion of the principles and merits of all three types is justified.

Typical Examples

As an introduction, it will be interesting to see how the three construction principles can be ap-
plied to the same assignment. Already the simple case of a channel fixture, as shown in Fig. 15-4, demonstrates some principal features. It is obvious that the rigidity of the three designs increases in the sequence: built-up, welded, cast; because the cast channel is fully integral, the welded channel is partially integral, while the built-up channel depends for its rigidity on the size and number of fasteners. A screw joint is never completely solid because of the required hole clearances, and any screw joint in a built-up fixture must therefore be additionally and permanently secured by means of tightly fitting dowel pins, as shown. The three types differ also with respect to “clean contours.” The built-up fixture, if made from fully machined or cold-finished components, presents well defined inner and outer contours with clean inner corners. The welded fixture has projecting weld beads in the inside corners and on the outer sides. The cast fixture has rounded fillets in the inner corners and clean, but not parallel, outer sides because of the draft. These features require consideration in the planning and layout of areas to be machined.

![Fig. 15-4. Three designs of a channel fixture.](image)

A somewhat more complicated case is the box jig with hinged leaf shown in Fig. 15-5, a through c. It is assumed that the three jigs must have a machined base surface of dimensions $A \times B$, as shown. The three jigs are drawn to the same scale and the thicknesses shown are representative. The built-up jig, Fig. 15-5a, is made of mild steel plate. It is designed strictly to dimensions $A$ and $B$, because the components are machined before assembly. All fixed joints are assembled and secured with screws (shown as larger circles) and dowel pins (shown as smaller circles). The screws provide the forces that hold the pieces together, but since screw holes normally are drilled with a clearance around the screws, they do not guarantee the exact position of the parts relative to each other. The dowel pins secure the parts in their exact position. Therefore, dowel pins are made to fit exactly in their holes. Normally, it takes two pins to secure a part, and for best control of the position, the pins are located as
Design of Fixture Bodies

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far apart as possible; therefore, dowel pins are usually placed diagonally opposite each other. In the present case, the base is closely fitted into grooves in the end walls; therefore, one pin would be sufficient, theoretically, at each end of the fixture; however, most designers would choose two pins in accordance with traditional practice. Where a removable part is to be secured with dowel pins, the pins are mounted with a press fit in the fixed part, and holes in the removable part are made with a sliding fit over the pins. Thicknesses of the individual parts are selected approximately equal to those in the cast fixture to provide sufficient bearing surfaces in the joints. The two straps improve the rigidity against longitudinal forces while they retain accessibility to the base for chip cleaning.

Dowel Pin Applications

Dowel pins are used extensively in all categories of tooling, and the correct design and application of these small components is of fundamental importance. Dowel pins are made of soft steel or drill rod. A hardened dowel pin can be made from commercial hardened drill rod. Standard dowel pins can be purchased either soft or hardened, and are made with a 5 degree taper at the leading end for easy and safe start. For hardened dowel pins, the surface hardness is 60 to 64 Rockwell C, the core hardness 50 to 54 Rockwell C. The shear strength ranges from 150,000 to 210,000 pounds per square inch (1035 to 1450 N per square millimeter). Diameter tolerance is plus and minus 0.0001 inch (0.003mm) with a surface roughness of 4 to 6 microinches (0.10 to 0.15 μm). Normally, they are manufactured with 0.0002-inch (0.0005mm) oversize to provide a secure press fit, but are also available with 0.001-inch (0.003mm) oversize for repair work in cases where a hole has been worn or accidentally machined oversize.

As a general rule for jigs and fixtures, the dowel diameter is selected one size smaller than the assembly screws. For presswork dies, dowels are made the same size as the screws because of the conditions of shock and vibration under which the dies operate. The length of engagement, or the bearing length, i.e., the length which the pin protrudes into the second member of the assembly, is 1½ to 2 times the dowel diameter. Soft dowel pins can be used for plain locating purposes where no heavy load is applied to the pin. However, hardened pins are sometimes preferred because they are usually ground to closer tolerances. A locating dowel pin that is also subjected to a severe shear load should always be hardened.

Locating the Dowel Pin

Dowel-pin locations are not usually specified by dimensions, but are shown by center lines on the drawing. The toolmaker usually knows that he is to locate the holes somewhat at random in the area of the center lines; a note to this effect is sometimes placed on the drawing. An exception occurs when the dowel holes are to be jig ground, an operation which is not commonly practiced on jigs and fixtures, but is occasionally, on dies. Dowel pins should be so located that the assembly of the components is foolproof. Symmetrical parts can be located in more than one relative position, and dowel pins are used to make certain that they go together in the one and only correct location.

In an assembly where one component must be removed frequently, the use of dowel pins together with straight drill bushings is sometimes recommended. Using a hardened pin in a hardened bushing, results in a precise and wear-resistant fit.

The general rule is that dowel pins should be so located that the holes pass entirely through the two components. This is done for easy removal of the dowel pin when needed. If, however, a dowel pin necessarily must be pressed into a blind hole, then the hole should be drilled deeper than is required to hold the pin. A good practice (which is not always followed) is first to drill the hole deeper and then to ream it to the depth to which the pin must penetrate. It is not possible to exactly specify just how much deeper the dowel-pin hole should be drilled because this depends upon the restrictions inherent in the design of the component into which it is to be drilled. The purpose of drilling the blind hole to a greater depth is to reduce the build-up of air pressure behind the pin, in the bottom of the hole. The pressure build-up follows Boyle’s Law, but does not have to be calculated.

Assembly Screws

Assembly screws are usually hexagonal socket head cap screws made of a high-strength steel. The minimum length of engagement of the screw thread should be as follows (where D is the screw diameter):

- in steel \( \frac{3}{2} \times D \)
- in cast iron \( 2 \times D \)
- in magnesium \( \frac{3}{4} \times D \)
- in aluminum \( \frac{3}{2} \times D \)
- in fiber and plastic \( 3 \times D \) and up
The welded jig in Fig. 15-5b, is also made of mild steel plate. The inherent rigidity in the welded joints permits the use of thinner plates. On the other hand, the inner space must exceed the A and B dimensions with sufficient clearance to avoid removing the weld beads in machining.

The cast jig in Fig. 15-5c is designed with larger material thicknesses than the welded jig, since cast iron has less tensile strength and a lower modulus of elasticity. The part is fully monolithic and, therefore, has no weak areas. Again, the machining of the base requires full clearance all the way around. The cast design, however, requires less machining than the other two designs, and needs no cutting and fitting. In accordance with most common drawing practice, the jig is drawn without showing the draft, but when proper draft is provided on all vertical surfaces, the casting can be made from a single pattern. In this case, the side walls would be solid, which is excellent from a structural viewpoint, but sacrifices access to the base for chip cleaning. Should this point be essential, either machined or cast windows could be provided, as indicated by the chain-dotted lines. Forming windows in the casting is perfectly possible, but requires the use of cores, thus complicating the foundry work. No evaluation or choice between the three principles will be made here on the basis of these two simple examples, because such a choice would depend on many factors, such as size, available equipment, time, etc.

Built-Up Fixtures

For each of the three design types, there exists some general rules and recommended practices which may provide useful guidance in the design. Built-up fixtures offer the greatest freedom in the design, essentially because there are no thermal-metallurgical limitations involved. The material is usually low-to-medium carbon steel, from AISI 1025 to AISI 1040; steels with very low carbon content do not machine well to produce smooth surfaces. Hardened or otherwise heat-treated steel can, without difficulty, be assembled with softer steels when desired. Small fixture bodies may be made in one piece by machining (carving) them from a block of steel. The whole body may be heat treated, thereby eliminating the need for separate hardened components such as drill bushings, locating points, etc.

About the only two rules regarding material thicknesses refer to the stability and strength of joints. Experience has shown that when thicknesses are selected as if they were intended for castings (see later), they will usually provide sufficient bearing areas for stiffness, and they preferably should be two times the OD of the assembly screws used, with 1.6 times as the absolute lower limit. There are no upper limits.

When additional rigidity is needed, it is necessary to use straps, as shown. Gusset plates are not practical in built-up fixtures. A number of different joint patterns are shown in the composite structure in Fig. 15-6. A different example is shown in Fig. 15-7, consisting of a channel bracket mounted on a plate flange. The channel is machined from a solid block. Standard commercial rolled sections offer but little opportunity for use in built-up fixture construction because of their rounded fillets and thin wall thicknesses. Therefore, where channels and angles are needed in a built-up jig, they will have to be machined from the solid block. As a rule-of-thumb, this method can be assumed to be economical for dimensions up to 2 to 4 inches by 8 to 12 inches (50 to 100 mm by 200 to 300 mm). Beyond these dimensions, it is cheaper to weld them.

Contoured flat components can be cut advantageously from plate stock on a contour bandsaw.
Fig. 15-7. A channel bracket mounted on a plate flange.

Fig. 15-8. A bracket type drill jig made of flame-cut plate.

Fig. 15-9. A jig made of one flame-cut plate and two straight plates.

Fig. 15-10. A modification of the jig shown in Fig. 15-9, made entirely of straight plates.

Fig. 15-11. A bracket fixture in cast design. The design to the left requires a split pattern; the design to the right is made from a one-piece pattern.

or with a cutting torch, and machined afterwards where necessary. An example is seen in Fig. 15-8, showing a drill jig of the bracket type. The built-up principle offers the advantage of having the top side of the base machined before assembly, an operation that would be more difficult if the jig was welded together. Modified approaches to related problems are shown in Figs. 15-9 and 15-10.

The foundry trade has a large bag of tricks and devices by which it can solve almost any design problem, and the use of castings, therefore, presents a great flexibility of form to the designer. However, these devices have their price; there are a few rules to which a casting must conform in the interest of economical production. These concern the easy withdrawal of the pattern from the mold, the free flow of metal in the form, uniform shrinkage, and the avoidance of "hot spots."

The first condition can be stated as "no undercuts with respect to the direction of withdrawal." Every fixture has, in a sense, a work space in the form of a more or less enclosed cavity for receiving the part, also, a form that permits easy and unobstructed loading and unloading will, usually, also permit easy withdrawal from the mold. Exceptions occur when the fixture has localized projections or depressions perpendicular to the direction of motion. One example was the window indicated in the cast
fixture in Fig. 15-5c. Another example is the bracket fixture seen in Fig. 15-11. The design to the left, with a circular boss and a machining clearance groove in the base, requires a split pattern and the mold to be parted along a-a. By the small changes shown in the design to the right, the parting line can be b-b, and the pattern can be made in one piece. Numerous examples of this and similar concepts are found in foundry literature. Only a few examples with direct reference to fixture design shall be given here.

A box-type fixture with a dividing wall may be designed as in Fig. 15-12a. This requires two cores, which are eliminated by either one of the designs in Fig. 15-12b and c. If the purpose of the upper flange is additional strength, this is compensated for in design b, by increasing the thickness in the dividing wall. If the purpose is to provide a flat surface for assembly with other components, then this is accomplished by design c.

A related case is seen in Fig. 15-13. The box to the left requires a core, while the box to the right can be formed without a core and is, for all purposes, at least equivalent to the design at the left; perhaps even better, as it eliminates two metal accumulations in the two T's.

A prototype for a widely used class of drill jigs is the casting shown in Fig. 15-14. It contains a number of details, providing for angular feet on the top and bottom surface, and long straight strips that can serve as feet on all four sides. Nevertheless, with the necessary draft, the pattern can be withdrawn and the casting made without cores. If it is now desired to reduce the length of the strip feet by cutting back as indicated at A, and to make these feet angular by adding horizontal ribs B, then the feature of pattern withdrawal in this casting is lost, and four cores of two different shapes will be required.

Fig. 15-12. Three different designs of a box type fixture. View a requires two cores; views b and c can be made without cores.

Fig. 15-13. Two different designs of a flanged channel jig. The design to the left requires a core; the design to the right is made without a core.

Fig. 15-14. A typical drill-jig casting.

Rules for Dimensioning Cast Fixtures

The condition "free metal flow" is essentially equivalent to the setting of a lower limit to the metal thickness in walls. If below such a limit, the metal will suffer excessive heat loss and solidify before the wall cavity is properly filled, forming what is known as a "cold run." These lower limits depend on the length of flow for the metal and, therefore, on the size of the part. Broadly, the following values are quoted:
For average size castings 3/8 to 1/2 inch (10 to 13 mm)
For smaller castings 1/4 to 3/8 inch (6 to 10 mm)
For very small castings down to 1/8 inch (3 mm)

The wall thickness can be more specifically related to the overall dimensions. In many cases, a wall serves as the web in a beam, either an I-beam, a T-beam, or an angle, as indicated in Fig. 15-15. With beam height \( H \), the thickness \( t \) can be taken as

\[
t = 0.2 \sqrt{H} \text{ inches} \quad \text{or} \quad t = \sqrt{H} \text{ mm}
\]

and for double web beams, as indicated in Fig. 15-16, the thickness in each web can be taken as

\[
t = 0.16 \sqrt{H} \text{ inches} \quad \text{or} \quad t = 0.8 \sqrt{H} \text{ mm}
\]

The reason for this thickness reduction is twofold; with double walls, the temperature in the mold is higher and metal can flow satisfactorily in a thinner wall cavity without cold run; statically, the double web beam has sufficient strength with less web thickness.

![Fig. 15-15. Dimensions of open beams.](image1)

![Fig. 15-16. Dimensions of double web beams.](image2)

In many cases, a flat plate within a casting forms a series of panels within a frame. Examples are indicated in Fig. 15-17. The length \( L \) between cross members (ribs, dividing walls, etc.) may be taken into consideration by taking

\[
t = \frac{1}{4} + \frac{1}{15} \sqrt{L} \text{ inches} \quad \text{or} \quad t = 6 + \frac{1}{3} \sqrt{L} \text{ mm}
\]

The calculated thickness should be interpreted in each case as a lower limit; in case of a discrepancy between the two formulas (which usually will be insignificant with well-proportioned castings) it is safer to use the higher value. The formulas are entirely empirical and cannot be proved mathematically; experience has shown that their results are satisfactory to the foundry and make a casting of well-balanced dimensions, which usually also satisfies the static conditions except, perhaps, in extreme cases.

The condition of uniform shrinkage means, theoretically, uniform thicknesses; in practice, it means an upper limit to the thickness ratio between adjoining sections. A good limiting value for this ratio is 2 to 1, normally somewhat less. It is actually not desirable always to strive for completely uniform thickness; those parts of the casting that are exposed to more effective heat loss to the mold and which, therefore, would tend to cool faster, should be heavier than those parts where the cooling is slower; the end result is a casting with uniform shrinkage and low residual stresses.

Corners should be rounded; this is imperative for internal corners, to avoid cracking; it is of lesser importance for external corners. The corner radius \( r \), can be related to wall thickness \( t \), as follows:

For internal corners \( r = 0.5 t \) to 1.0 \( t \)
For external corners \( r = 0.18 t \) to 0.2 \( t \)

These are minimum values. However, radii in internal corners should not be uncritically increased, particularly not at places where ribs and walls cross or join, to avoid unnecessary accumulation of metal which would cause "hot spots," i.e., slow-cooling areas which invariably collect slag and develop porosities.

While the principal dimensions of a cast fixture can be determined or confirmed by calculation, as explained in Appendix III, many details are not
amenable to such analysis, and will have to be designed by a combination of experience and “feel” on the part of the designer.

Lugs for hold-down bolts should be provided with prongs of generous width because they may be accidentally stressed far beyond the limits anticipated in any calculation. The same applies to ribs; in particular, ribs on brackets and other projecting parts with a large overhang. Examples are seen in Figs. 15-18 and 15-19.

Bends in thin walls should not show sharp corners, neither externally nor internally; in fact, they should be given generous radii of curvature, exceeding those quoted previously, as shown in Fig. 15-20.

Effect of Machining on Castings

As a general rule, machined surfaces should be as small as possible, partly because of the cost of machining, but also because the skin of the casting is of a particular structural value. The cast iron immediately below the skin is strongest, and this strength diminishes gradually towards the center of the section.

When unstabilized castings are machined they are apt to distort as a result of the removal of metal in which the stresses were previously balanced against the stresses in the remaining metal. The casting distorts (warsps) until a new balance of residual stresses is obtained. These stresses consist of the remaining stresses originally in the casting, plus new stresses set up by the action of the cutting tool. Any casting for a precision fixture must therefore be given a stabilizing treatment, either a normalizing, an anneal, or at least a stress relief. It is good practice, but not widely used, to sandblast and paint castings after this treatment to remove scale, oxide, and any remaining sand from mold and cores.

Welded Fixtures

Welded fixtures are, with few exceptions, made from low-carbon, hot-rolled steel assembled by electric-arc welding. This construction principle puts few restraints on the designer. There are virtually no thickness limitations, neither down nor up; metal of any thickness large enough to be used in a fixture can also be welded, and the proficient welder can deal with any special problem by, e.g., preheating prior to welding, and selecting a proper welding sequence to prevent heat accumulation, to cope with heavy sections or sections of widely differing thicknesses.

Every known type of weld joint may be used; most frequently employed are those shown in Fig. 15-21. Chamfered corners, and V- and U- joints are used less in fixture welding than in other structural welding, since fixtures are usually designed with generous dimensions so that fatigue is not considered a serious hazard. Full penetration is, therefore, not necessarily a requirement in joints between heavy sections.

Weld Details

A collection of some typical weld details for fixtures is shown in Fig. 15-22, which also gives
Fig. 15-22. Typical weld details and relative weld dimensions. Fillet sizes are determined by the thinner of the two adjoining sections. Where extra strength is required, use heavier fillets. The allowable stress under shocks is 5000 pounds per square inch (34.5 N per mm²).

Relative weld dimensions. A few typical fixture details are shown in Figs. 15-23 and 15-24; a composite fixture structure is shown in Fig. 15-25. Small projections for bosses and pads can be made by building up welding material and subsequent machining of the surface; examples are shown in Fig. 15-26.

Components for welding should be precut as far as possible by sawing, milling, shearing (small thicknesses only), and torch cutting. Large openings

<table>
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<th>Thickness, A or B</th>
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<th>Fillet Size</th>
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<td>mm</td>
<td>in.</td>
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<tr>
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<td>6 - 9.5</td>
<td>1/4 - 3/8</td>
</tr>
<tr>
<td>7/16 - 1/2</td>
<td>11 - 13</td>
<td>7/16 - 1/2</td>
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Diagram a.

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<td>14 - 25</td>
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<tr>
<td>1 1/16 up</td>
<td>26 up</td>
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</table>

Diagram b.

<table>
<thead>
<tr>
<th>Depth (in. mm)</th>
<th>Rod Size (in. mm)</th>
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<tbody>
<tr>
<td>1/4 - 3/8</td>
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<td>11 - 13</td>
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<td>1 1/16 up</td>
<td>26 up</td>
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Diagrams c, d, and e.

Fig. 15-23. A welded U-shape.

Fig. 15-24. A welded bracket.

Fig. 15-25. A composite welded fixture structure.

Fig. 15-26. Welded pads.
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DESIGN OF FIXTURE BODIES

DESIGN RULES

Welded design differs from cast design in one important limitation: Curved shapes should be avoided. Straight plates, strips, and bars are cheap; and except with thin sections of no interest for fixture design, bending involves a serious cost increase. Welded fixtures, therefore, often will lack the more artistic look associated with cast shapes. Appearance, however, is a matter of taste and fashion, not of economy and efficiency.

A seemingly small detail, actually a great asset to welding by increasing rigidity, is the use of straps and gussets; two reinforcing components which are inexpensive in their application. Assume, for example, that a U-shaped open box, such as that shown in Fig. 15-23, lacks rigidity; this deficiency is easily eliminated by adding two straps as shown in Fig. 15-30, or four gusset plates, as shown in Fig. 15-31. On brackets and shelves, the use of a simple diagonal strap, as shown in Fig. 15-32, may be cheaper and even more efficient than a gusset.

In addition to flat plates, the welded design makes extensive use of standard rolled structural shapes. The use of flats, with and without full corners, saves a considerable amount of cutting; the same is the case with square sections. Round sections are used in short lengths for bosses. Angles are widely used in plates are precut. Many contoured components are cut or rough machined before they are welded. Examples are the three-point clamp in Fig. 15-27, three different types of hinges in Fig. 15-28, and the built-up T-slot in Fig. 15-29.

**Fig. 15-27.** A welded strap clamp.

**Fig. 15-28.** Welded hinges.

**Fig. 15-29.** A welded T-slot.

**Fig. 15-30.** The use of straps for increased rigidity.

**Fig. 15-31.** The use of gusset plates for increased rigidity.
and are available in a great variety of dimensions, including large thicknesses. The other standard structural shapes, the channel and the I- and Z-beam, have relatively small wall-thicknesses and are seldom used, except in special cases, such as for large T-slot bases, as shown in Fig. 15-33. Typical examples of combinations of plates and standard sections are shown in the following illustrations. A drill jig with angle legs and an extensive use of flat sections, a design typical of many box-type jigs, is shown in Fig. 15-34. The same is the case with the drill jig shown in Fig. 15-35, which is built from plates and flats, with short lengths of T-sections for feet.

There is no upper limit for the size of welded fixtures. For very large welded fixtures used in the aerospace industries, tube sections of medium wall thickness are virtually indispensable. The material may be of steel or aluminum. Circular as well as square tubes are used, and although square tubes are easier to cut and fit, they are slightly less economical with respect to material, in relation to strength and rigidity, and are not available in such large sizes as are circular tubes.

To eliminate the danger of later distortion, welded fixtures should be annealed or normalized, then sandblasted and painted before machining.

Within certain limits, welding permits the joining of steels of different hardness within the same structure. Jig feet can be made of low-grade tool steel, heat treated, and then welded to the main structure. During annealing or normalizing, the material in the feet will be drawn and the resulting hardness will be approximately 35 Rockwell C, sufficient to provide good wear-resistance and still be machinable.

Welded fixtures should be designed to have minimum surface areas for machining. In this respect, the designer has a little more freedom than with cast fixtures where certain compromises may have to be accepted for the sake of simplicity in the pattern design. He must visualize all machining operations and be certain that they do not also remove his weld beads. Unbelievable as it may seem, this sometimes happens!
Comparison and Conclusions

Some general rules can be laid down for the areas of application and relative merits of the three types of fixtures:

The built-up fixture, including those carved out of one piece, is preferred for small parts in general; and for medium-size parts where the shape is simple; when welding and foundry facilities are not available; or when delivery time is critical. It may take advantage of available standard sections, in the same manner as the welded fixture. It can be disassembled and changed, or its components may be reused in other fixtures. The cast fixture can, in principle, be designed for any desired size of part; however, its natural area is the medium size. It allows great freedom to the designer, tends to be heavy, requires access to a foundry, and involves the time and cost of pattern making, which may be substantial. On the other hand, the cast fixture may be economically superior if more than one casting from the same pattern is needed. When a larger number of nearly equal fixtures are needed, it may be economical to design and stock a supply of standard cast shapes, notably thick-walled channels and angles. However, it is not practical to attempt to change a cast fixture.

The welded type of fixture has now become the most widely used. It is extensively covered in the literature and its superiority, relative to cast fixtures, has been widely expressed; some of the statements made are quite correct, while others overly generalize; although they may signify the trend, in specific cases they do not always hold true.

The welded fixture can be made lighter in weight than the cast fixture without sacrificing strength and rigidity. Procurement time can be less for a weldment than for a casting because of the time required for pattern making. It has been said that it takes less time and money to make a complete weldment than to make the pattern for it. It is probably nearer the truth to say that these two items are, generally, of the same order of magnitude. The foundry trade is full of pitfalls, and perhaps it takes less experience and skill to design a welded structure than a successful cast structure.

Areas requiring machining occasionally may be kept smaller in the welded fixture; it is also claimed that a weldment requires less additional thickness for machining allowance; this is, after all, a consequence of the amount of distortion to be expected. One literature source suggests machining allowances of: 1/8 inch (3 mm) on medium-size and 1/4 inch (6 mm) on large-size welded fixtures. To this could be added 1/16 inch (1.5 mm) for fixtures consisting mainly of solid blocks with little welding. These figures are actually well in line with general practice for machining allowances for castings. Machining of steel takes about 5 percent more time than machining of cast iron for the same quantity of metal removed. On the assumption that weldments do require less machining than castings, it is also claimed that total machining cost will be 10 percent less, and, in conclusion, that the complete welded and machined fixture will result in a saving of 25 percent over the total cost of the cast fixture.

A welded fixture can, in principle, be changed by removing and adding components. This should also be taken with some reservation, because the altered fixture may well require an additional anneal and renewed machining. Rules such as these should be taken as guidelines only; sometimes they apply, sometimes they do not.

The design of a fixture is not necessarily confined to any single type of construction alone. A combined construction may well present the most advantageous solution. The frame or body can be welded or cast, and the precise locating surfaces can be screwed and doweled onto the frame. Thus, much, but not all, of the precision machine work can be done on the loose parts before they are attached to the body; often under more favorable conditions. Also, repair or alteration work on the fixture is more easily accomplished.

Solid Fixtures

Solid fixtures means those that are machined out of one piece of material. Simple jig plates would often fall into this category. The material is machine steel or light metal tooling plate. If hardened surfaces are required, the entire fixture is made of tool steel. A practical upper limit for fixtures made of machine steel is approximately 2 by 3 by 6 inches (50 X 75 X 150 mm). The limit is not an absolute one, but increases with increased capacity and efficiency of the machining facilities available.

A simple example of a solid fixture is shown in Fig. 15-36, a drill jig for drilling, countersinking, and tapping the hole in a split shaft collar. The jig body is milled square and bored out, and a drill bushing, a handle, and a locator for the split in the collar are installed. The bushing is a slip bushing to allow for countersinking and is beveled to approximately fit the curved surface of the collar.

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1 E. Thaulow, Maskinarbejde (Copenhagen: G.E.C. Gad's Forlag, 1930) vol. II.
A different type of drill jig is shown in Fig. 15-37. The part is a bearing bushing; the jig body is a pot with a handle. The bushing plate has a long stem terminating with a screw thread and the large nut that locks the bushing plate also provides the base on which the jig is supported during drilling. Almost all machining operations on the jig are performed in a lathe. An additional example of a solid fixture is shown later in Fig. 18-39.

Plastic Fixtures

Plastic tooling is essentially made by casting or by laminating. The strength of these materials is, at best, comparable to the strength of cast iron, but often it is less. For this reason, plastic tooling materials are used only for tooling that is not exposed to heavy loads. Their principal areas of application are drill jigs, routing fixtures, inspection fixtures, and various types of templates. Within their natural areas of application they do have several advantages. They are light and are easy to handle. Since they are fabricated (cast or laminated) directly from the part, they lend themselves to forms with complex contours. They closely reproduce the contour of the master without complicated and costly machining. Tool details such as bushings and liners, are either embedded in the fixture body as it is fabricated, or potted in place after the plastic has cured. They require less lead time than metal fixtures and are also less expensive.

If plastic fixtures are damaged they can be repaired, and if broken beyond repair, they can be replaced at a moderate cost. Design changes can be quickly and inexpensively incorporated in the fixture. Contours can be altered and detail parts added, deleted, or relocated without costly time delays or expensive machining. They are, therefore, applicable to prototype development. The low cost makes it economically feasible to use plastic fixtures for even short production runs. For their application to production in large quantities, they have the advantage that they can be duplicated at a low unit cost and with precise accuracy. Technical details about plastic fixtures are presented in Chapters 3 and 14.

2 Ibid.
CHAPTER 16

Drawings, Dimensions, and Tolerances

Economical Design

Design and drawing costs represent a relatively large share of the total fixture cost since usually only one fixture is made from a set of drawings. A few shortcuts can be applied to reduce the cost of the drawings, but their integrity with respect to accuracy, completeness, readability, and clarity cannot be sacrificed. The fixture drawings are successively used by the checker, the planner, the production shops, and the inspection department and must quickly convey to each, the construction as well as the intended operation of the fixture. Overloaded drawings will require excessive time for study and deciphering, ambiguities generate requests for clarification, and inaccurate or incorrect drawings result in rejections, rework, and much loss of material and time.

Economical design requires systematic work. The first step is to acquire the detailed and complete part drawing; the next step is to accumulate additional pertinent information relating to the machine tool to be used and to the available accessories and general-purpose work holders. All fixtures required for the complete machining program for the part are first laid out in sketches and reviewed together, a procedure that quite frequently results in useful modifications of the initial program, sometimes even in a revision of the design of the part. With these steps finalized, the shop drawings of all the fixtures can be prepared, lessening the risk of unpleasant surprises.

Fixture Drawing Practices

The layout drawing shows the complete fixture with the part located for machining, including an outline of the raw material. In this way, the machining allowance on each machined surface is clearly recognized as is the clearance around the part. Adjoining details of the machine tool (table with T-slots, lathe spindle nose, etc.) are also shown. The assembly drawing is prepared from the layout and shows the fixture as seen from the operating side. It is also useful to indicate the direction of motion of the cutting tools, the direction of rotation of milling cutters, etc. Wherever possible, the details, as well as the assembly, are drawn to full size. Standard parts are not drawn in detail, but are shown and listed in the assembly.

Drawings are made in pencil, with strong black lines to ensure blueprints of maximum contrast. A blueprint is exposed to rough treatment in the shop and its readability deteriorates rapidly from wear, repeated folding, and dirt. The part outline is drawn in phantom red lines. They are highly conspicuous, not only on the original, but also on blueprints where they show as ghost lines.

General drafting practice, as used for product design drawings, is followed with respect to projections, sections, symbols, lettering, title block, material identification, heat treatment, surface roughness, standard tolerances, etc. The parts numbering system used for tooling detail parts is different than the system used for product detail parts. Sections are used generously as they are more informative than projected views. Sections are cross-hatched for easy recognition in the drawing. The viewing direction on sections follows established practice or is conspicuously identified by arrows.

A fixture drawing provides some specific information about the use of the fixture. It lists the number and type of cutters required for the operation. Where one fixture serves more than one operation and the part is differently located within the fixture for each operation, each location of the part is shown or clearly indicated. The same is done when a fixture is used for more than one part.
Tricks of the Trade

Templates for all commercial fixture components are available from the manufacturers and are used at the design stage as well as in the completion of the final drawings.

To improve the contrast of blueprints, dimensions; dimension lines; arrowheads; and extension lines (witness lines) are drawn in India-ink. The additional time required is negligible and the result is well worth the effort.

In some aircraft companies, the very large fixtures for welding and assembly are built only from sketches of the main structure together with the detail drawings of the parts to be welded or assembled.

In layout and assembly drawings it may sometimes happen that one component, or the workpiece will obscure another component because their lines coincide. In this case, if one component is deliberately drawn with a slight distortion, i.e., a trifle longer or shorter, or to a slightly different scale, the confusion of lines is eliminated and the obscured part will then be exposed to view. The drawing gains in clarity, and the possibility of misinterpretation and error is prevented by adding one or more dimensions to the distorted part.

Placing the drawing paper diagonally on the board has several advantages not generally recognized. The lower left-hand corner of the sheet is held near the front edge of the board, the right-hand edge is raised 20 degrees, and the corners are then fastened. The drafting machine is adjusted to work at this angle. Work is done faster as shadows are eliminated along straight edges, and it is easier to see the pencil point. Lettering is more easily done because of the slant of the drawing. The paper stays cleaner and the front edge does not become worn and torn, since the draftsman does not have to lean across the drawing.

Dimensions and Tolerances

Assembly drawings show the dimensions required for the inspection of the fixture; they are the dimensions to the locating and clamping surfaces, to the tool guides (including drill bushing centers), and those relative to the attachment of the fixture to the machine tool. Every part contains one or more critical dimensions, that are recognized and identified at the beginning of the dimensioning and tolerancing procedure. Critical dimensions are those that are significant for the function of the part or for its compatibility with other parts in the fixture. Typical critical dimensions are hole center distances, the distance from a hole center to a machined surface, or the distance between two machined surfaces. A distance from a machined surface (or a hole center) to an unmachined surface is seldom critical, nor is the distance between two unmachined surfaces.

Detail drawings of the fixture parts show critical dimensions, overall dimensions, and all others that define machining operations. Stock dimensions are listed in the bill of materials and are not needed on the detail drawings.

In the conventional dimensioning system, all co-linear dimensions are written as a long chain that reaches from one end of the part to the other. The advantage here is that it provides a good check on the nominal numerical values as all the single dimensions in the chain must add up to the overall length of the part. However, it also has a disadvantage in that the individual tolerances within the chain add or subtract in an unpredictable manner. If, for example, all tolerances come out with their plus maximum values, it would require an excessively large tolerance on the overall length of the part.

The conventional dimensioning system is now supplemented, and is gradually being replaced, by a new system, the "coordinate dimensioning system," or the "coordinate system." Modern manufacturing practices, as exemplified by the use of jig borers, jig grinders, and N/C (Numerically Controlled) machine tools, require each dimension to be defined as the distance between the particular point or surface and a common datum or reference line, point, or plane. Fundamentally, two perpendicular reference lines are required for dimensions in one plane. Three reference planes are required for complete definition of all dimensions on a three-dimensional part. Where more than two reference lines or points are needed in one plane, their relative location must be clearly defined. A set of two reference lines is equivalent to the x and y axes used in analytical geometry.

In a jig borer or jig grinder, the table with the work can be moved relative to the spindle in two perpendicular directions (the coordinate axis directions), as shown in Fig. 16-1, and the table settings (the x and y coordinates) can be read out with the accuracy of 0.0001 inch (0.003 mm). In most machines, the readings are direct, when the table moves toward the left and toward the column of the machine (away from the operator), which again corresponds to an apparent or relative movement of the spindle to the right (the x axis direction), and toward the operator (the y axis direction). For convenience in the operations and as a safety measure to avoid errors, the dimensions shown on the
part drawing should also go from left to right and from top to bottom, and the two reference lines in the plan view should originate from a point at, or near, the upper left corner of the part.

Numerically controlled machine tools are designed somewhat differently. The reference axes on the drawings should appear in the lower left-hand corner; i.e., the dimensions go left to right and from bottom to top. However, tool drawings should be dimensioned as shown in Fig. 16-2, since tooling components are normally machined on jig borers, not on N/C machines.

Fig. 16-1. Movements and positions of the jig borer table. A. Position of table relative to spindle before table movement; B. Position of table after moving in conventional direction by direct reading of coordinate measuring system. Note that the "effective" movement of the jig borer spindle is opposite in direction from the actual table movement.
Dimensioning of hole centers from two reference lines (coordinate axes) is shown in Fig. 16-2. In the upper view, dimensions are written on ordinary dimension lines with arrowheads and leader lines to the corresponding points in the drawing. The arrangement of the dimension lines shows clearly where the reference lines are assumed to be. Direct center-to-center dimensions, such as a in the illustration, are not part of the system, but are calculated and entered on the drawing, labeled "REF," and are to be used for inspection. When the number of holes is large, some drafting and lettering work is saved by simply numbering the holes on the drawing and tabulating them by number, diameter, and x and y coordinates. In this case it is necessary to define the coordinate axes on the drawing.

A different method of dimensioning from reference lines is shown in the lower view of Fig. 16-2. The reference lines are defined, and the dimension to each significant point is written on the leader line. This method is frequently used, not only because it simplifies the lettering work, but because it also makes it much easier to read the numbers.

With the coordinate dimensioning system, each single dimension and tolerance is now independent of all others. Sometimes, it is necessary to combine two or more dimensions in a short chain. This is permissible, provided the individual tolerances are
selected so that the total tolerance on the chain does not conflict with other tolerances within the part. The transfer of dimensions with or without tolerances from the conventional system to the coordinate system is governed by the following two principles:

1. For untoleranced dimensions, the difference of any pair of dimensions (coordinates) on the coordinate system must equal the dimension that they replace on the conventional system.

2. For tolerated dimensions, the sum of the tolerances of any pair of dimensions on the coordinate system must not exceed the tolerance of the dimension that they replace on the conventional system. The procedure is explained in full detail in Appendix II at the back of the book.

**Fixture Tolerances**

The accuracy of a machined part is less than the inherent accuracy of the machine tool and the fixture by means of which the part was made. This is known as the "degeneration of accuracy." To ensure interchangeability of the machined parts, it is therefore necessary to prescribe closer tolerances of the dimensions of the fixture than of the dimensions of the part. The crucial question is what tolerances shall be applied to the fixture dimensions to ensure correct tolerances on the part dimensions. The literature is generous with suggestions. They range from one half to one tenth of the part tolerances. Each recommendation is as good as any other recommendation, and none of them will guarantee the correct result.

Fixture tolerances are determined by a combination of common sense, practical judgment, logical conclusions, analysis, and calculations, usually of an elementary, sometimes trivial nature. The procedure can start with the part tolerances and work through to the fixture tolerances, or it can start with a set of assumed or selected fixture tolerances, calculating the resulting part tolerances, and comparing these with the required tolerances from the part drawing.

*Close tolerances are expensive, thus tolerances are always selected as wide as possible,* consistent with the proper functions of the part. A part that functions with wide tolerances will also function with close tolerances, but not necessarily vice versa. An initially selected tolerance that is found to be too wide can be reduced without risk. In the search for fixture tolerances it is therefore recommended to start with those dimensions that offer the best possibility of using wide tolerances.

Tolerances are needed on dimensions relating to the following fixture elements:

1. Part locators and cutter positioners
2. Devices for attaching the fixture to the machine tool (positioning keys, slots for clamping bolts, etc.), for attaching gages to the fixture, and surfaces for the installation of standard components and interchangeable or replaceable fixture parts (bushings, inserts, etc.).

**Example**—Two 0.191-inch-diameter holes are to be drilled in the round disc by means of the drill jig shown in Fig. 16-3. The part is located by its periphery in the inverted nest in the upper part of the jig. There is no clamping device, but when the first hole is drilled, a pin is inserted to lock the part in position relative to the jig. The two hole location dimensions are apparently critical dimensions. The most liberal tolerance is the .004 inch on the location of the hole to the left. If there were no other considerations this would permit the center of the part to move a total of .004 inch relative to the jig, and a part with the maximum diameter of 1.4375 inches could, so far, accept a nest diameter of 1.4375 + .004 = 1.4415 inches. This dimension, however, is not acceptable because it does not include any tolerance, it does not allow for wear, and it does not recognize the existence of parts with less than maximum diameter.

A part of this small size requires a minimum nominal clearance of .0005 inch to enter the nest which gives a minimum nest diameter of 1.4375 + .0005 = 1.4380 inches. To this is applied a manufacturing tolerance (machining tolerance, "toolmaker’s tolerance") of .001 inch resulting in the nest diameter 1.438 inches. If the nest is at its maximum of 1.439

1.439 it provides a clearance of .0025 inch against a part of minimum diameter 1.4365 inches. These relationships are shown in Fig. 16-4. The selection of the fixture dimension and tolerance is in accordance with sound economic principles. The cost is determined by the tolerance .001 inch, which is reasonable for this class of work, and not by the resulting minimum clearance of .0005 inch which is in a sense, incidental only, since it depends on the fixture and the selected part, not on the fixture as such. The resulting maximum clearance is within the maximum permissible value of .004 and also allows an ample margin.

As a part of minimum diameter is shifted through the maximum clearance of .0025 inch, the center of the part is shifted between positions, .00125 inch on
Fig. 16-3. A part with dimensions and tolerances, and its drill jig.

each side of the center of the nest. Let \( a \) be the distance from the center of the nest to the center of the drill bushing, first assumed to be without tolerance. Then, to satisfy the part tolerances, we have

\[
\begin{align*}
\quad a - .00125 & > .498 \quad \text{and} \quad a + .00125 < .502 \\
.49925 & < a < .50075
\end{align*}
\]

Like any other dimension, \( a \) requires a toolmaker's tolerance which again, is selected as .001 inch. There must also be an allowance of .00025 inch for wear in the bushing (actually also including the initial clearance between the drill and the bushing). These requirements are satisfied by selecting the center distance as \( .5005 \) inch, which provides the required wear allowance of .00025 inch and leaves an unclaimed margin of safety of .00025 inch.

The maximum tolerance on the hole center distance on the part is .002 inch; with a toolmaker's tolerance of .001 inch on the bushing center distance in the jig, this requirement is comfortably satisfied.

Fig. 16-4. Resulting clearances for the part shown in Fig. 16-3.
It is usual practice in such cases to split the part
tolerance. This is done by selecting the fixture dim-
ension as \(0.0005\) inches, which provides an allowance
of \(0.0005\) inch on each side. This allows the \(0.00025\)
inch for bushing wear and still leaves an unclaimed
margin of safety.

**Toolroom Tolerances**

The toolmaker’s tolerance of \(0.001\) inch (\(0.03\) mm)
can be maintained economically in ordinary tool-
room operations on conventional lathes, milling
machines, and grinders. Precision grinders with
hydraulic feed and positioning devices work to toler-
ances of less than \(0.0005\) inch (\(0.013\) mm). With the
addition of a high precision gage, tolerances of \(0.0001\)
to \(0.0002\) inch (\(0.003\) to \(0.005\) mm) can be main-
tained consistently. The jig borer maintains \(0.0005\)
inch (\(0.013\) mm) under average conditions, and this
can be reduced to \(0.0002\) to \(0.0001\) inch (\(0.005\) to
\(0.003\) mm) by careful work under the best condi-
tions.

*Example:* A crank arm as shown in Fig. 16-5, is to
be straddle milled on the two opposite sides of the
crank arm boss to a width of \(0.750\) inch. One side of
the crank arm boss is to be aligned with the side of
the center boss (which is already machined) within
\(\pm 0.002\) inch. The milling cutters are positioned with
a feeler gage from a setting block which is located
from the same locating surface in the fixture as the
center boss.

![Crank Arm Boss](image)

**Fig. 16-5. Tolerances for setting block and milling cutters.**

The cutters are mounted on an arbor. The space
between the cutters is checked after each regrind-
ing and is maintained at \(0.750\) inch. With a mean cutter
space of \(0.749\) inch and a mean thickness of the
feeler gage of \(0.120\) inch, the mean width of the
setting block is

\[0.749 - 2 \times 0.120 = 0.509\] inch

The mean width of the shoulder on the setting block
is \(0.120\) inch, the same as the thickness of the feeler
gage. The toolmaker’s tolerances are \(0.0004\) inch
on the feeler gage, \(0.0006\) inch on the setting block,
and \(0.001\) inch on the shoulder. The actual dimen-
sions, are, on the feeler gage \(1.198\) inch, on the set-
ting block, \(0.5093\) inch, and on the shoulder \(1.205\)
inch.

When the right cutter is positioned from the right
side of the setting block, the extreme alignments
with the locating surface (which must be within
\(\pm 0.002\) inch) are

<table>
<thead>
<tr>
<th></th>
<th>Inch</th>
<th>Inch</th>
</tr>
</thead>
<tbody>
<tr>
<td>shoulder, minimum</td>
<td>-.1195</td>
<td>maximum</td>
</tr>
<tr>
<td>feeler gage, maximum</td>
<td>+.1202</td>
<td>minimum</td>
</tr>
<tr>
<td>alignment</td>
<td>+.0007</td>
<td>-.0007</td>
</tr>
</tbody>
</table>

Plus and minus alignment means that the cutter is
positioned to the right and left of the locating
surface.

When the left cutter is positioned from the left
side of the setting block, the extreme alignments for
the right cutter are

<table>
<thead>
<tr>
<th></th>
<th>Inch</th>
<th>Inch</th>
</tr>
</thead>
<tbody>
<tr>
<td>cutter space, minimum</td>
<td>+.748</td>
<td>maximum</td>
</tr>
<tr>
<td>shoulder, maximum</td>
<td>-.1205</td>
<td>minimum</td>
</tr>
<tr>
<td>setting block, maximum</td>
<td>-.5093</td>
<td>minimum</td>
</tr>
<tr>
<td>feeler gage, maximum</td>
<td>-.1202</td>
<td>minimum</td>
</tr>
<tr>
<td>alignment</td>
<td>-.002</td>
<td>+.002</td>
</tr>
</tbody>
</table>

which is still permissible.

**Callouts**

Additional instruction for the machining of the
part is provided by callouts. Machining operations
that are obvious are not called out, but callouts are
used where a sequence of several operations is re-
quired at one location, such as for a hole, and where
an operation must be performed in accordance with
specific requirements that are not conveniently de-
scribed by conventional or coordinate dimensioning.
Callouts are short, often one word only, and even
that may be abbreviated, or they are formulated in a
sort of telegram style when more than one word is
needed. Some companies forbid the use of callouts
because they prefer to have decisions regarding
DRAWINGS, DIMENSIONS, AND TOLERANCES

COMMON AND TYPICAL CALLOUTS FOR USE ON PART DRAWINGS

<table>
<thead>
<tr>
<th>DRILL</th>
<th>CASE HARDEN .030-040 DEEP</th>
</tr>
</thead>
<tbody>
<tr>
<td>DRILL AT ASSEMBLY</td>
<td>CARB HDN &amp; GR (carburize, harden and grind)</td>
</tr>
<tr>
<td>DRILL .50 DIA 1.25 DEEP</td>
<td>GRIND AFTER HARDENING</td>
</tr>
<tr>
<td>.75 DRILL, .50 DEEP, 4 HOLES</td>
<td>DRILL ROD, HARDBR</td>
</tr>
<tr>
<td>DRILL 3/4 THRU</td>
<td>COLD-ROLLED STEEL, CASE HARDEN</td>
</tr>
<tr>
<td># 30 (.128) DRILL 2 PLACES</td>
<td>HARDEN SCREW AND NUT</td>
</tr>
<tr>
<td>SPOT DRILL</td>
<td>HT, H, T, HEAT TREAT</td>
</tr>
<tr>
<td>CENTER DRILL</td>
<td>HT TO 30-32 RC</td>
</tr>
<tr>
<td>SPOT FACE</td>
<td>STRESS RELIEVE</td>
</tr>
<tr>
<td>BACK FACE</td>
<td>ANNEAL</td>
</tr>
<tr>
<td>COUNTERBORE</td>
<td>SAW</td>
</tr>
<tr>
<td>CO BORE</td>
<td>TORCH CUT (or FLAME CUT)</td>
</tr>
<tr>
<td>C’BORE</td>
<td>BRAZE</td>
</tr>
<tr>
<td>COUNTERSINK</td>
<td>DIA TO FIT INSIDE BUSHING</td>
</tr>
<tr>
<td>CO SINK</td>
<td>FIT AT ASSEMBLY</td>
</tr>
<tr>
<td>C’S’K (countersink)</td>
<td>SLIDING FIT IN PART NO. ABC 235</td>
</tr>
<tr>
<td>BORE</td>
<td>.001 TIR (Total Indicator Reading)</td>
</tr>
<tr>
<td>NO. 3 MORSE TAPER (.60235 PER FOOT)</td>
<td>FIR (Full Indicator Reading)</td>
</tr>
<tr>
<td>.250 BASIC TAPER ON DIA PER INCH OF LENGTH (used for a conical taper)</td>
<td>ROUND WITHIN .010 ON R</td>
</tr>
<tr>
<td>TAPER .1500 ± .0015 PER INCH (used for a flat taper)</td>
<td>CYLINDRICAL WITHIN .010 ON R</td>
</tr>
<tr>
<td>REAM</td>
<td>PERP TO SURF (perpendicular to surface)</td>
</tr>
<tr>
<td>REAM 1.250 DIA</td>
<td>4 EQUAL SPACES</td>
</tr>
<tr>
<td>TAP</td>
<td>BOTH SIDES (used to save repeating a dimension)</td>
</tr>
<tr>
<td>TAP 1/2-20, 1 DEEP</td>
<td>DATUM LINE</td>
</tr>
<tr>
<td>TAP 1/2-13 UNC-2B, 1 DEEP</td>
<td>REFERENCE LINE</td>
</tr>
<tr>
<td>1/2-13 NC THREAD</td>
<td>DIA (diameter)</td>
</tr>
<tr>
<td>1/2-13 NC THD</td>
<td>R (radius)</td>
</tr>
<tr>
<td>DRILL &amp; TAP 1/2-13 UNC, 2 HOLES 180° APART</td>
<td>TRUE R</td>
</tr>
<tr>
<td>1.000 MIN FULL THD</td>
<td>R SPHER (spherical radius)</td>
</tr>
<tr>
<td>LEFT HAND THREAD</td>
<td>BEND R 1/4</td>
</tr>
<tr>
<td>THREAD RELIEF</td>
<td>TYP (typical)</td>
</tr>
<tr>
<td>1/16 WIDE X 3/64 DEEP-4 GROOVES</td>
<td>REF (reference)</td>
</tr>
<tr>
<td>GRIND</td>
<td>BSC (basic)</td>
</tr>
<tr>
<td>HONE</td>
<td>SYM (symmetrical)</td>
</tr>
<tr>
<td>LAP</td>
<td>¢ (center line)</td>
</tr>
<tr>
<td>LAP OUTSIDE (or INSIDE) AFTER GRINDING SCRAPER</td>
<td>MMC (Maximum Material Condition)</td>
</tr>
<tr>
<td>KNURL</td>
<td>DEG (degrees)</td>
</tr>
<tr>
<td>96 DP RAISED DIAMOND KNURL</td>
<td>GA (gage, indicating plate thickness, for example #12 GA = .105 inch)</td>
</tr>
<tr>
<td>CHAMFER</td>
<td>φ (diameter)</td>
</tr>
<tr>
<td>1/8 X 45° CHAMFER</td>
<td>mm (millimeter)</td>
</tr>
<tr>
<td>BREAK EDGES</td>
<td>cm (centimeter)</td>
</tr>
<tr>
<td>ROUND EDGES</td>
<td>m (meter)</td>
</tr>
<tr>
<td>DEBURR</td>
<td>¹ A callout “WELD” followed by the weld dimension may be used. However, using the standardized symbols for welded joints is recommended.</td>
</tr>
<tr>
<td>CASE HARDEN</td>
<td></td>
</tr>
<tr>
<td>CASE HDN</td>
<td></td>
</tr>
</tbody>
</table>
machining operations made by the workshop personnel. However, industrial callouts are widely used and the tool designer should be familiar with their application. Common and typical callouts are presented in the listing on the previous page, and some typical operations are shown in Fig. 16-6.

Fig. 16-6. Typical operations, frequently identified by callouts: A. Spot drill, center drill; B. Spot face; C. Back face; D. Counterbore; E. Countersink; F. Bore.
Standard and Commercial Fixture Components

Adventages of Standardization

Fixture components are, in general, small and are used in large quantities. Their design is closely determined by the function of the particular component, and no consideration of taste or style is involved. For these reasons, fixture components offer a wide field for standardization.

Standardized components offer significant advantages to the user. “Design” is reduced to selection of a component of suitable size from a table, and actual design time is eliminated. The quantity of identical parts required is increased, and production cost is reduced. When the standardization process transcends the boundaries of individual firms, it opens the way for mass production of components by specialized manufacturers with further possibilities for cost reduction and quality improvement. High-precision and high-quality parts (small drill bushings, hardened and ground to close tolerances) are available in today’s market at prices less than a dollar; this is less than the cost of pulling a vellum from the file and having a blueprint made!

N.I.J.F.C.M.

The drill bushing and fixture component industry in the USA is today a multimillion dollar industry and is steadily expanding. In 1958, the manufacturers within this industry organized the N.I.J.F.C.M. (National Institute of Jig and Fixture Component Manufacturers, Oakland, California). A major activity of this organization is the standardization of fixture components. As standards are adopted, specifications are made available to its members for incorporation into their manufactured products and are presented to consumers through the members’ marketing literature.

USA Standards

Standardization of fixture components in the USA (including drill bushings which, traditionally are mentioned separately) is now found on three levels. National standardization started in 1935 with the issue of American Standard for Jig Bushings. This standard has been frequently revised; the current edition is ANSI B94.33–1962 (R1971), Jig Bushings. It covers the types of bushings illustrated in Figs. 14-1 and 14-2 and described in Chapter 14.

Proposed Standards

The next level of standardization is a set of proposed standards prepared by the N.I.J.F.C.M. for submittal to the American National Standards Institute (ANSI); a package of individual standard proposals, covering most of the components needed for clamping and locating. With the unified recommendation of the industry concerned, it may be expected that these proposals will be approved and designated as USA Standards in the near future.

It should be noted that this proposal will standardize not only sizes, dimensions, tolerances, and, in some cases, materials, but also part numbers, which when adopted, will greatly simplify specifying and ordering these items.

Manufacturers’ Standards

The third level of standardization is at the manufacturers’ level. Much standardization and unification has been going on through the years, and many items have been virtually standardized on all significant dimensions. These are listed in individual manufacturers’ catalogs which should be consulted for the items required. One area with the least
standardization is the numbering system; but this condition will be simplified by the proposed standards. As it now stands, each manufacturer has his own serial number system. Cross reference tables have been prepared by which interchangeable or, at least, equivalent parts from different sources can be identified.

Drafting templates are also available for most commercial components. When used systematically, they can save much time in all phases of the design, from the initial layout to the final vellum.

Proprietary and Patented Fixtures and Components

Most of the devices and components described in this book are in the public domain and may be utilized freely. However, several of the commercially available components and fixtures are covered by some degree of legal protection, usually in the form of one or several patents. Where information about proprietary rights, patents or otherwise, has been available, it is so indicated, either in the text or in captions to the illustrations. Any device and design so designated is protected and the fixture designer is advised not to copy it nor to utilize it in any other unauthorized manner.

Commercial Fixture Components

A review of the presently available components with evaluations, brief descriptions, and data for their range of sizes and capacities follows. Dimension symbols on line drawings will indicate to the fixture designer the dimensional information that is available (from manufacturers’ catalogs) about each component. Drill bushings are not included since they are discussed at considerable length in Chapter 14.

Bolts, Screws, and Associated Parts (See Fig. 17-1a through ii.)

Studs (a) are made from high-strength steel and have a number of uses too diversified to be specified. They can be joined by coupling nuts (f) to create any required length. Threads are made to a nut fit on both ends. A secure fit at installation is obtained by the use of a bonding agent on the thread. Range: Thread from 1/4-20 to 1-8, length from 1 1/2 to 12 inches.

Eye bolts (b), jig latch bolts (c), and swing bolts (d), are used where the bolt must be swung out of place to allow a strap to be removed or a jig leaf to be opened. Another use is with cam clamps where it is desired that the cam rock back and forth for direct force application. Range: Thread from 1/4-20 to 3/4-10, length from 2 to 6 inches.

T-bolts (e) are used with related parts such as clamp straps, flange nuts (g), and spherical washers (1) to create various work-clamping arrangements. They are also used for clamping the fixture down to the machine tool table. These bolts are highly stressed in service and are made from heat treated alloy steel with 150,000 psi minimum tensile strength. Range: Thread from 1/2-13 to 1-8, length from 1 1/2 to 12 inches.

Coupling nuts (f) are long nuts and are used to couple two studs to create a stud of desired length. They are also used where a nut of exceptional length is needed for other purposes. Range: Thread from 3/8-16 to 1-8, length from 1 to 2 1/2 inches.

Flange nuts (g) have a large bearing area to ensure increased surface contact with a clamping component. For many purposes, combining a flange nut with a set of spherical washers (f) to eliminate unequal load on the screw thread, caused by slight misalignment, is recommended. Range: Thread from 1/4-20 to 1 1/4-7 and from 1/4-28 to 1 1/4-12.

Spherical flange nuts (h) combined with bottom spherical washers (1) are used to compensate for minor irregularities between clamp strap and part being clamped, and to eliminate unequal thread loads caused by slight misalignment. Range: Thread from 1/4-20 to 1 1/4-7.

Acorn nuts (see Fig. 17-3 e) are nuts that are closed at one end to protect screw thread against dirt and damage. Range: Thread from 1/4-20 to 7/8-9.

Knurled lock nuts (i) are used for quick thread locking. They can be tightened by hand or with a 1/4-inch rod. Range: Thread from 3/8-16 to 5/8-11.

T-slot nuts (j) are used with studs for clamping a fixture down to the machine tool table. They are adapted to standard machine tool table T-slots. Two series are available: “standard” and “N. I. J. F. C. M. standard.” The screw thread in the nut is so designed and cut that the stud cannot be turned through the nut and down into the table. Range: Thread from 5/16-18 to 3/4-10.

Interchangeable sine fixture keys (k) are keys for use in the machine tool table slots and permit the use of the fixture in slots of varying widths. They are inserted in the T-slot from above, rotated until they fit in the wide part of the slot, and locked into position in the fixture by a slight turn of the set-screw. Range: Width across flats (for entering the T-slot) from 1/2 to 1 1/8 inches.
Spherical washers (1) act as a ball-and-socket to compensate for any slight misalignment between clamp strap and part being clamped, and to eliminate undue stresses on threads. Hole size is 1/16 to 1/8 inch larger than the corresponding stud to allow for equalizing action. Range: Corresponding stud diameter from 1/4 to 1 1/2 inches.

C-washers (m) are for easy removal to speed-up clamping and release of the part. A wire hole is provided for attachment to the fixture. Range: Width of slot A from 9/32 to 1 1/32 inches.

Connecting cables (n) are used for C-washers and other loose parts, for attachment to fixture. They are made of nylon covered stranded-steel cable in...
Fig. 17-1 (Cont.). *Bolts, screws, and associated parts. i. †Knurled lock nut; j. ††T-slot nut; k. †Interchangeable sine fixture key (U.S. Pat. 2,707,419); l. §Spherical washer; m. §C-washer; n. †Connecting cable; o. ††Swing C-washer.

*Illustrations courtesy of the following companies: †American Drill Bushing Co.; ††Northwestern Tools, Inc.; §Morton Machine Works.

1.5-inch length and provided with ferrules for crimping.

Swing C-washers (o) are for easy removal to speed up clamping and release of the part. Washer is held in place by shoulder screws (p or q). Range: Radius $B$ from 1 to 1 3/4 inches, thread from 1/4-20 to 3/8-16.

Slotted shoulder screws (p) provide a precision ground shoulder diameter for use as a pivot for a swinging or rotating component. Range: Thread from 10-32 to 3/8-16.

Socket shoulder screws (q) provide a precision ground shoulder diameter for use as pivot pins for C-washers, swing clamp straps, and other rotating components. Screw head contains socket for socket hex wrench. Available as N. I. J. F. C. M. standard. Range: Thread from 10-24 to 5/8-11.
Fig. 17-1 (Cont.). *Bolts, screws, and associated parts. p. $Slotted shoulder screw; q. †Socket shoulder screw; r. †Hand knob and screw; s. †Hand knob screw assembly; t. †Knob swivel screw; u. **Knurled-head screw.

*Illustrations courtesy of the following companies: $Morton Machine Works; †American Drill Bushing Co.; **Monroe Engineering Products Inc.
Hand knobs and screws (r) are used for hand-tightened holding and clamping functions. Hand knob is cadmium plated cast iron; screw is heat treated steel with black oxide finish. Range: Thread from 1/4-20 to 5/8-11, length from 1 3/4 to 3 inches.

Hand knob screw assemblies (s) are used for hand-tightened holding and clamping functions. Relieved tip protects end threads from damage caused by slight peening. Range: Thread from 1/4-20 to 5/8-11, length from 1 to 3 1/2 inches.

Knob swivel screws (t) are used for hand-tightened holding and clamping functions. Swivel shoes prevent marring of finished surfaces of soft materials such as aluminum and copper. Shoe stops rotation immediately upon contact with workpiece, swivels...
Fig. 17-1 (Cont.). *Bolts, screws, and associated parts. bb. †4- and 5-prong hand knobs; cc. ‡Star hand knob; dd. †Knurled knob; ee. †Speed ball handle; ff. ‡Plastic ball knob; gg. §Finger handle.

*Illustrations courtesy of the following companies: †American Drill Bushing Co.; ‡Northwestern Tools, Inc.; §Morton Machine Works.

3 degrees in all directions to compensate for minor surface irregularities, pulls off easily and snaps on for installation in fixture mounting hole. Range: Thread from 1/4-20 to 3/4-10, length from 1 1/4 to 4 13/16 inches.

Knurled head screws (u) are used for light-duty holding and clamping applications. Knurled head provides for easy finger tightening. Relieved tip protects end threads from damage caused by slight peening. Threads are rolled, which provides increased
strength and wear resistance and a smoother surface.
Range: Thread from 10-24 to 1/2-13, length from 1 to 3 1/2 inches.

Torque head screws (v) have a spring-loaded clutch which is built into the head and releases at a preset torque. In this way they prevent overclamping and distortion of the part. In some models the releasing torque can be adjusted from the outside. Torque screws are provided with check nuts and can be supplied with swivel pads and nylon tips. Range: Thread from 10-32 to 5/8-11, end force from 10 pounds to 28 pounds for fixed torque types and from 0 to 50 pounds for adjustable torque types.

Quarter-turn screws (w) are quick-locking fasteners for leaf jigs and jig plates. Corners are chamfered to provide self alignment of screw head with latch slot when closing the leaf. Range: Thread from 10-32 to 1/2-13, length from 1 to 2 1/4 inches.

Half-turn screws (x) are quick-locking fasteners for leaf jigs and jig plates. Screw-shoulder locks and unlocks the plate by rotating the screw one-half turn. Range: Thread from 10-32 to 1/2-13, length from 1 to 1 1/4 inches.

Threaded adjustable locating buttons (y) are used together or in combination with fixed locating buttons and pins to accurately locate workpiece in fixture. Range: Thread from 10-32 to 5/8-18, length from 1 to 3 inches.

Jack screws (z) are used in fixtures to support irregularly shaped workpieces, such as castings, and to prevent elastic distortion (springing) of thin workpieces during machining. Range: Thread from 3/8-16 to 5/8-11, length B from 1 1/4 to 2 1/2 inches.

Bar knobs (aa) are used in heavy-duty clamping applications. They are made from high-strength ductile iron and can be tightened by inserting a bar between the vertical prongs for maximum leverage. Knobs can be obtained as unmachined blanks or as finished knobs, with choice of tapped or reamed mounting hole. Range: Hole, blank, tapped from 3/8-16 to 1-8, reamed from 3/8 inch.

Four- and five-pronged hand knobs (bb) are cast from gray iron, tumbled smooth, and cadmium plated. Faces of reamed or tapped knobs are machined square with hole. Four-pronged knobs—Range: Hole, blank, tapped from 10-32 to 5/8-11, reamed from 3/16 to 5/8 inch, diameter A from 7/8 to 2 1/2 inches. Five-pronged knobs—Range: Hole, blank, tapped from 5/8-11 to 3/4-10, reamed from 5/8 to 3/4 inch, diameter A, 3 inches.

Star hand knobs (cc) are available in choice of aluminum or cadmium plated cast iron. Knobs can be obtained as unmachined blanks or as finished knobs with tapped hole. Range: Hole, blank, tapped from 1/4-20 to 5/8-11, diameter A from 1 1/8 to 3 inches.

Knurled knobs (dd) are suitable for adjustment, clamping and locating devices. Knurled head provides nonslip finger grip. Knobs are made with tapped holes. Range: Thread from 10-24 to 3/4-10, diameter B from 3/4 to 2 1/2 inches.

Speed-ball handles (ee) are balanced to permit rapid spinning for quick clamping and release of the work. Wide handle permits greater leverage. Handles can be obtained as unmachined blanks or as finished handles with choice of tapped or reamed mounting hole. Range: Hole, blank, tapped from 1/2-13 to 3/4-10, reamed from 1/2 to 3/4 inch, width A from 4 3/4 to 8 inches.

Plastic and steel ball knobs. Lightweight plastic ball knobs (ff) provide a comfortable, rustproof handgrip for actuating levers. Plastic knobs, except
the largest size, have threaded brass inserts. Steel knobs are recommended for use where added weight is desirable for easier lever actuation. They are available as unmachined blanks or as finished knobs with hole drilled and tapped, ready for mounting. Plastic knobs—Range: Thread from 10-32 to 5/8-18, diameter $B$ from 1 to 1 7/8 inches. Steel knobs—Range: Hole, blank, tapped from 3/8-16 to 5/8-18, diameter $B$ from 1 1/2 to 2 inches.

Finger handles (gg) are miniature handles to be used as finger grips for actuation, control, or adjustment of small movable parts. Range: Thread from 10-24 to 5/16-18, length $C$ from 1 1/16 to 1 1/4 inches.

Handwheels (hh) are made from cast iron and can be obtained as blanks or finished machined with polished rims and unpolished surfaces painted. Smallest size handwheel is solid, larger sizes have four spokes. Range: Diameter $A$ from 3 to 12 inches.

Machine handles (ii) are used with handwheels and for various actuating and gripping purposes. They are machined and polished to a smooth finish. Shape is designed for convenient grip, and speed and ease of operation. Solid handles can be obtained with a press fit, or with a threaded shank, revolving handles are made with a press fit. Solid handles—Range: Shank diameter $A$ for press fit from 1/4 to 1/2 inch, shank thread from 1/4-20 to 1/2-13, length $B$ from 1 23/32 to 5 inches. Revolving handles—Range: Shank diameter $A$ for press fit from 5/16 to 1/2 inch, length $B$ from 2 5/8 to 5 1/8 inches.

Quick-acting Screw Components (See Fig. 17-2a through d.)

Fig. 17-2. Quick-acting screw components. a. Single locking lever; b. Double locking lever; c. Quick lock lever; d. Quick lock knob.

Range: Thread from 1/2-13 to 3/4-10, length of arm from 4 3/8 to 5 3/8 inches.

Quick-locking knobs (d) have a part of the screw thread removed to provide a smooth hole for fast removal when the knob is tilted. The knob locks or releases by rotating it one quarter turn. Range: Thread from 1/4-20 to 5/8-11, diameter from 1 1/8 to 3 inches.

Screw Clamp Assemblies (See Fig. 17-3 a through m.)

Strap clamp assemblies (a through h) are suitable for a wide variety of clamping applications. They
also offer considerable flexibility since individual parts are standardized and interchangeable; e.g., the hex nut can be replaced by an acorn nut or a hand knob, or the single end strap can be replaced by a double end strap. The studs are made of high-tensile steel, and straps; nuts; spherical washers; and clamp rests are made of heat-treated steel with black oxide finish. Hand knobs are cadmium plated. Single end straps (b, c, d) have machined finger grips for easy lateral adjustment. The straps are spring loaded (lifted) for quick release and for holding them in position when released. Spherical washers compensate for irregularities between strap and part and ensure rigid holding. Double end straps...
Fig. 17-3 (Cont.). Screw clamp assemblies. h. Removable clamp assembly; i. Swing clamp assembly for reamed hole mounting; j. Swing clamp assembly with flange base; k. Hook clamp assembly with socket head cap screw.

(e, f) are used for clamping flat parts in the position shown in the illustrations or can be inverted for holding round pieces in V-blocks or in nesting arrangements. With open hex nuts (a, b) the assembly provides maximum clamping range and has visual indication of thread engagement. The acorn nut (c, e) protects the stud thread from dirt and damage, but limits the clamping range. The hand...
knob (d, f) provides quick clamping and release without need of a wrench. With a double end strap, the assembly can clamp two parts in one operation, and can also be used as a single end strap against a clamp rest permanently mounted in the fixture. Single end strap—Range: Thread from 1/4-20 to 3/4-10, travel G from 1/2 to 1 3/4 inches, capacity J from 7/8 to 2 inches. Double end strap—Range: Thread from 3/8-16 to 3/4 10, travel from 1 to, 1 3/4 inches capacity from 1 5/8 to 2 inches.

End hand-knob clamp assemblies (g) combine the maximum clamping range of the hex nut with the quick operation offered by the hand knob. Rest plate made of heat-treated steel protects fixture body and assists in locating the strap prior to clamping. Range: Thread from 1/4-20 to 5/8-11, travel from 1/2 to 1 1/2 inches, capacity from 5/8 to 1 3/4 inches.

Removable clamp assemblies (h) permit complete and fast removal from the fixture of the entire assembly. The hardened bottom insert is pressed into a recessed hole in the fixture base, and the bayonet-type T-bolt is inserted into the slot and turned one quarter turn. The hand knob, locked into place with a jam nut, is used for inserting and removing the assembly. The speed handle is for clamping and releasing, and can be replaced by a flange nut for use where space is limited. Range: Thread from 5/8-11 to 3/4-10, capacity from 3 1/8 to 5 1/8 inches.

Swing clamp assemblies (i, j) are made both for reamed hole mounting and for mounting with a flanged base. While all strap clamps can be swung free of the part, they require considerable space but the swing clamp assembly swings free with much less space requirements. The clamping screw has a swivel head to protect the surface of the work and adjust to irregularities. The arm can be adapted to a left or right swing by installing a stop pin in a dowel hole to one or the other side of the tail lug. Range: Thread on clamping screw from 5/16-18 to 5/8-11, travel from 5/16 to 11/16 inch, capacity is not limited by assembly but is determined in the design of the fixture.

Hook clamp assemblies (k, l), available with socket-head cap screw and stud and hex nut, are ideal for use where space is extremely limited. The hook is spring loaded for quick release and easily swings away from the workpiece. The body is an alloy steel investment casting with precision ground diameter designed for reamed hole mounting. Range: Thread on cap screw from 5/16-18 to 1/2-13, on stud 5/16-18 to 5/8-11, capacity from 7/8 to 1 11/16 inches.

Hinge clamp assemblies (m) allow fast, and completely free, access to work area. They can clamp directly on the part or be used for locking a swinging leaf. A hinged mounted pad bolts to the fixture. A hand knob swivel screw provides quick clamping and release, protects work surface and compensates for minor surface irregularities. Range: Thread from 5/16-18 to 3/8-16, capacity $H_{max}$ from 1 1/4 to 2 5/8 inches, throat A from 2 to 3 1/2 inches.

**Cam Clamp Assemblies (See Fig. 17-4 a through e.)**

Long travel cam clamps (a) have a hand knob and a bayonet-type guide groove in the stem. The hand knob is pushed straight against the work to close,
and rotated to clamp and lock. This type of clamp combines long travel and quick clamping better than any other manual clamping device. The rotating pad on the end of the stem protects surface of workpiece. Range: Diameter of stem from 3/8 to 1 inch, rapid travel from 1/2 to 2 1/2 inches, locking travel, equivalent to a cam rise, from 1/16 to 3/16 inch.

Cam-actuated strap clamp assemblies (b, c, d) are available with single end and double end straps and center cam, and as single end straps with plain and automatic end cam. The clamp strap is actuated by a quick-acting clamping and release cam. The strap is lifted and carried by a spring. Spherical washers permit adjustment of strap to surface irregularities.
Nuts on stud in end cam assemblies allow for adjustment in height and compensation for wear. Center cam, single end—Range: Thread from 1/4-20 to 5/8-11, travel G from 1/2 to 1 1/2 inches, capacity J from 5/8 to 1 5/8 inches. Center cam, double end—Range: Thread from 1/2-13 to 5/8-11, travel from 1 to 1 1/4 inches, capacity from 1 3/8 to 1 5/8 inches. End cam—Range: Thread from 1/4-20 to 5/8-11, travel from 1/2 to 1 3/8 inches, capacity from 7/16 to 1 1/16 inches.

Automatic end cam clamp assemblies (e) have single end straps. Cam action automatically retracts strap after release and brings strap forward before clamping in one movement of handle. Handle may be mounted on either side of base block.

Range: Thread from 1/4-20 to 5/8-11, travel from 1/2 to 1 inch, capacity relative to base from 1 3/8 to 2 5/8 inches; capacity can be changed by changing fixture dimensions.

Individual Clamping Components (See Fig. 17-5 a through d, and Fig. 17-6.)

All parts incorporated in the assemblies previously described are individually available for use on jigs and fixtures. Various other parts for similar purposes include plain and serrated end steel clamp straps (a), matching steel and aluminum step blocks (b), and chuck jaws (c).
Serrations on blocks and clamps are precision machined for perfect fit between half blocks put together and between straps and half blocks. They are used for setups in machining operations, as shown in Fig. 17-6, for temporary fixtures, and as components on universal jigs. Plain and serrated end steel straps—Range: Bolt size from 5/16 to 1 inch, length from 2 1/2 to 10 inches, width from 1 to 2 inches, thickness from 1/2 to 1 1/2 inches. Step blocks—Range: Width from 1 to 2 inches, capacity (height of clamped part) from 3/4 to 9 inches.

Chuck jaw inserts (c), to be mounted on the master jaws of standard 3-jaw lathe chucks, are available in low carbon steel and 2024-T4 aluminum; in each case they are made of bar stock, not cast material. Since they are soft, they can be machined to fit a specific part, thereby converting the chuck into a turning fixture. The materials used permit easy machining. Steel inserts are preferred for most normal applications; they stand up well for medium-size production lots; for large production lots, they can be carburized and case hardened. Aluminum inserts offer some special advantages: They protect highly finished machined surfaces and parts made from soft materials, and their light weight reduces the moment of inertia of the chuck assembly and the load and wear on the spindle bearings. Range: Length from 2 5/8 to 6 inches, width from 1 to 2 1/2 inches.

To assist in the machining of soft inserts, boring positioners (d) are available. The boring positioner is a flat disc in the form of a three-lobe cam. It is placed inside the master jaws of the chuck, and is rotated with a screwdriver until the desired diameter is obtained. With this position of the boring positioner, the jaws are drawn tightly against the edge of the positioner to eliminate backlash, and the inserts are then machined to the desired diameter and profile. Range: Clamping diameter from 1 1/2 to 6 inches.

**Fixed Locating Components** (See Fig. 17-7 a through k.)

Rest buttons (a through f) are installed in a fixture to provide level support of the workpiece and also to prevent it from resting on accumulated chips and dirt. Rest buttons are made from heat-treated steel, ground to size on principal dimensions.

Round rest buttons (a, b, c) can be obtained with flat or spherical tops and for various methods of installation. The most accurate and least expensive way of mounting is by a press fit (a, b). Buttons with flat head are made with head thickness (dimension b) precision ground to final height or manufactured to 0.010 to 0.014 inch oversize to allow for finish grinding after installation in the fixture. A less accurate mounting method is by means of a screw thread on the shank of the button (c) and a tapped hole in the fixture base. These buttons are hexagonal and are made with oversize height for finish grinding.

Hollow rest buttons (d) can be mounted by means of a separate flat-head or socket-head screw. The button is counterbored for the screw head and precision ground to final height. This type of button is available in heights up to 2 1/4 inches and is, together with the hexagonal screw-mounted button, also used as feet for drill jigs. It is best not to use this button with the counterbore in the top position as small chips and dirt can lodge in the counterbore possibly resulting in inaccuracy when the workpiece rests against the button.

Rest buttons are also made with slip-fit shanks (e, f) for quick change. They are secured either by a separate lock screw (e) or by a screw thread (f) on the outer end of the shank. Slip-fit rest buttons require retainer bushings (g, h, i), press-fit mounted in the fixture base. Retainers are straight bushings (g) or shoulder bushings (h) for use with buttons with separate lock screws. The threaded slip-fit buttons require retainers (i) with a matching screw thread below the bore. Buttons of various types—Range: Diameter from 5/16 to 1 5/8 inches, height from 1/8 to 1 inch.

Rest pads (j) are used in lieu of buttons for level support of large workpieces in heavy-duty applications. They are precision ground, ready for installation by means of countersunk, socket-head cap screws. Range: Length from 2 3/8 to 3 3/8 inches, width from 1 to 2 inches.
Fig. 17-7. Fixed locating components. a. Rest button (locator) for press-fit installation; b. Spherical button for press-fit installation; c. Threaded hexagonal rest button; d. Hollow rest button, also used as buttons for jig feet; e. Rest button for slip-fit installation with lock screw recess; f. Threaded hexagonal rest button with straight shank for slip-fit installation.

Jig legs (k) are available in double end design. They are press-fit mounted in the jig base and secured by screw thread and nut. The small end is extended beyond the thread and formed as a small jig foot. Range: Thread from 1/2-13 to 3/4-10, length of large foot from 2 to 6 inches, length of small foot from 1/4 to 5/16 inch.

Intermediate (Adjustable) Supports (See Fig. 17-8a through d.)

Jack locks (a, b) are used for actuating (lifting and lowering) plunger-type intermediate supports and locking them in place when lifted to contact with workpiece. Typical applications are to support castings and other rough workpieces at points between the fixed locating points, to eliminate deflection during machining. When released, the jack lock can be moved freely in either direction. A quick twist of the hand knob locks the jack by the expanding action of two hardened steel shoes. A lock stop, mounted on the side of the jig by one screw, prevents the unit from being rotated or pulled out of the fixture. Range: Diameter from 0.624 to 1.249 inches, travel from 3/4 to 1 1/2 inches.

Spring jack locks (c) are completely self-contained units comprising plunger with cap, actuating spring, screw with hand knob for release and locking, housing and (optional) base for mounting on fixture base.
They compensate for irregularities in workpiece dimensions by automatically adjusting the plunger with cap to the work surface by the action of the spring. After contact is established, the plunger is locked tight by a twist of the hand knob. They are designed for fixed spring pressure and for screw-adjusted spring pressure. The working mechanism is protected against chips and dirt by a long skirt on the cap. Range: Travel 5/16 inch, height extended from 2 3/8 to 2 9/16 inches.

Eccentric leveling lugs (d) are circular discs with a pivot hole located off-center. They provide positive support with precise adjustment. Position after final adjustment is permanently secured with a dowel pin and two dowel holes are provided for this purpose. Discs are supplied in soft steel or carburized and heat treated. Range: Diameter 1 inch, thickness (height) from 1/4 to 3/8 inch.

Locating Pins (See Fig. 17-9 a through o.)

Locating pins are for the precise locating of parts that are already provided with mating holes. Locating pins are either pilot pins (a) or full-length pins; a pilot pin and a full-length pin are used together; the full-length pin "catches" the part first, then provides guidance and support as the part is lowered to "catch" on the pilot pin. Most pins are chamfered on the end (b) or bullet nosed (c) to facilitate catching and entering. Locating pins are either fully round or relieved by four flats forming a "diamond pattern" (b, c). The diamond pattern provides the combination of a close tolerance in one direction (the major axis in the diamond) with a wide tolerance in the other direction (the minor axis). Pin diameters (full round and diamond) are
Fig. 17-8. Intermediate (adjustable) supports. a. Jack lock; b. Adjustable support with a jack lock installed in a fixture; c. Adjustable spring jack locks (press fit and flange base); d. Eccentric leveling lug.

Locating pins are press fit (a, b, c, g) or slip-fit (e, f) mounted in the fixture. A slip fit is used where the pin must be interchangeable and requires a matching retainer bushing press-fit mounted in the fixture. Slip-fit pins are secured by means of a lock screw (e) or by means of a threaded shank and a nut (f). Round and diamond pins are available with a knurled portion (larger than the pin) for embedment in plastic or castable tooling (h, i). Locating pins are also used for locating of two fixture parts relative to each other, which requires mating liners (d) in the second fixture part. Standard drill bushings are suitable for mating bushings in most cases. Range: Most types of locating pins are

offered in an "A" and a "B" range; the "B" range is 0.001 inch smaller than the "A" range.
available in diameters from 1/8 or 1/4 inch as the lower limit, to 7/8 or 1 inch as the upper limit, and in lengths from 1/8 to 1 inch.

Floating pin locators (j), mounted in a press-fit bushing, serve the same purpose as the diamond pin, providing a close tolerance in one direction and a
Fig. 17-9 (Cont.). Locating pins. i. Knurled relieved (diamond type) locating pin; j. Floating pin locator, comprising pin with bushing; k. Slotted hole locator bushing, for use with "L" or "T" pins; l. Knurled slotted hole locator bushing for plastic tooling; m. L-pin; n. T-pin; o. Double-action captive L-pin.

Wide tolerance (1/8 inch total floating travel) in the perpendicular direction. Float direction is defined and secured by means of a roll pin. Range: Diameter of floating pin from 1/4 to 5/8 inch.

Slotted hole-locator bushings (k), press-fit mounted in the fixture base with position secured by means of a roll pin, serve the same purpose as the diamond pin. With a mating round pin they provide a close tolerance in one direction and a wide tolerance (1/8-inch travel allowed lengthwise in the slot) in the perpendicular direction. Slotted hole bushings are also available with knurled outer surface (1) for embedment in plastic or castable tooling. Range: Slot width from 1/4 to 1/2 inch for press-fit mounting, from 3/16 to 1/2 inch for mounting in plastic.

L-pins (m) and T-pins (n) are easily removable locators for temporary precision alignment of predrilled workpieces in jigs and fixtures, or for alignment of a drill jig plate on a part after the first hole or the first two holes have been drilled. Conventional L- and T-pins are loose parts and are removed from the fixture when not in use. They can...
be obtained with a cable for permanent connection of pin to fixture to prevent loss. Detent pins have one or (usually) two spring-loaded balls embedded in the pin body to provide a nonpermanent lock and protect against accidental withdrawal of the pin. Captive locating pins (o) are provided with a bushing in which they slide with a precision sliding fit. The bushing is permanently installed in the jig with a press fit or a slip fit (secured with locking screw). Bushings are available with knurled exterior for embedment in plastic tooling. The pins are captive in their bushings. A single-action pin can be pushed

**Fig. 17-10.** Indexing components. a. Rotary cam operated tapered (Left), and straight (Right), indexing plungers for standard mounting; b. Suggested methods of adapting head of plunger pin to actuating devices; c. Spring-loaded straight indexing plunger.
down the full length of the pin, retracted until the pilot end of the pin is inside the bushing, and held in this position by a groove in the pin. Double-action pins have an additional but reversed upper groove that limits the downward travel. Range: Pin diameter from 3/16 to 1/2 inch, length of travel up to 6 inches. L- and T-pins can be obtained with a screw thread instead of the pilot end and used as clamping screws.

Indexing Components (See Fig. 17-10 a through i.)

Precision made indexing plungers and matching bushings are the most critical detail required in the

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Fig. 17-10 (Cont.). *Indexing components. d. §Spring plunger; e. †Spring plunger mounted in a blind hole; f. †Spring plunger mounted in a through hole; g. §Stainless steel ball plunger; h. †Ball plunger detent; i. †Spring stop.

*Illustrations courtesy of the following companies: †American Drill Bushing Co.; §Morton Machine Works.
construction of an indexing fixture. Plungers are straight or tapered with a 15-degree included angle (a). Plain indexing plunger units are made without actuating devices. The plunger head is soft so that it can be machined in accordance with the design of the plunger actuator supplied by the customer (b). The plunger housing is machined for press-fit mounting in the fixture. Cam-actuated plunger units are
manually operated. Handle rotation of 180 degrees completely retracts plunger. Plunger housings are standard mounted in a reamed hole in the fixture, and secured in the “whistle notch” by means of a nylon tipped set-screw. Housings are also available with mounting flange. Range: Housing diameter from 3/4 to 2 inches, length up to 3 inches.

Spring plungers (c through g) lock, support, and locate workpiece or fixture parts by spring-actuated plungers and require less space than any other device. Also used as ejectors, they are available with standard end force up to 68 pounds, and with light end force up to 31 pounds. A nylon plug is inserted in the screw thread and provides a positive, vibration-proof lock. The normal type is a self-contained unit where the spring and plunger are located within a threaded stud; loose parts are also available for mounting in a blind or through hole in the fixture wall. A modification is the ball plunger (g) where the plunger is a steel ball (low alloy steel or stainless steel); for accurate positioning the mating part should be provided with a hardened steel detent (h). Range: Thread from 6-32 to 1-8 (ball plungers down to 4-48), length from 7/16 to 2 13/32 inches.

Spring stops (i) with circular crowned or rectangular tapered pressure heads are used for holding parts against locators and for light-duty locking functions. Range: Head size from 3/8 to 3/4 inch, end force from 10 pounds to 32 pounds.

Miscellaneous Components (See Fig. 17-11 a through h.)

Toggle clamps (a through d) are manually operated linkage clamps based on the same kinematic principle as the eccentric clamp. When clamping, the toggle link is pushed slightly past dead center and stays locked in this position. Locking and release is done by a quick swing of the handle. When released, the clamping arm is swung at least 90 degrees away from the clamping position and provides excellent access to the clamping area. Normal working position of the clamping arm is horizontal (a, b), however, models are available with working position of the arm 45 degrees up or down (c) or 90 degrees down. A push-pull variation of the toggle clamp (d) performs the clamping by means of a sliding plunger. Toggle clamps require relatively large construction and operation space; they are manufactured in various series, differing in strength and power. Range: Clamping force from 50 to 3000 pounds, capacity below horizontal clamp from 11/32 to 3 inches.

Tooling balls (e, f, g) provide a precision reference point relative to a part or a fixture, for the adjustment of the machine tool spindle prior to critical machining operations. The ball and shank are concentric within 0.0002 inch T.I.R. (total indicator reading). The standard tooling ball (e) provides immediately a “visible” and measurable extension of the axis of the bore where the shank is located. By an additional measurement taken from the ball to the part, the location of the ball center is defined and available for axial adjustments. The shoulder tooling ball (f) provides a built-in reference dimension from the shoulder to the ball center. The toolmaker’s construction ball (g) provides the same dimension and can be locked in position by a screw in the internal thread in the shank. Range: Ball diameter from 1/4 to 1 inch.

The tooling ball pad (h) is a supplement to the standard tooling ball and is used where the part does not provide a suitable reference hole for the tooling ball shank. The tooling ball can be adjusted to the exact position desired and locked securely in place by a cap screw lock without marring the tooling ball shank. The pad is drilled for mounting on the part by a cap screw, and the pad position can be secured by dowel pins.

Cast Iron Fixture Stock Sections (See Fig. 17-12)

Fixture stock consists of cast iron plates, blocks, and profiled shapes and is used in the design and building of fixture bodies, thereby saving time and expense for patterns. The following sections are available: V-, U-, H-, L-, and T-sections, flats, and hollow squares and rectangles. T-sections are available in equal and unequal shapes. The material is high-tensile-strength cast iron; the stock is machined square and parallel to within 0.005 inch per foot on all external surfaces. The interior of hollow shapes is left unmachined. Range: V-sections have a 90 degree included angle, an opening width from 1 1/2 to 4 1/2 inches, and height from 1 1/8 to 3 inches. Other sections have overall dimensions from 3 to 8 inches, rectangular hollow blocks have up to a 10-inch-width, wall thickness of open shapes from 5/8 to 1 1/4 inches, wall thickness of hollow blocks from 1/2 to 1 inch.1

Other commercial products in larger sizes and for more general applications are also available. They are described in the chapters on fixture bodies, drill jigs, universal fixturing, and automatic fixturing.

1Tables with complete dimensions for a number of the basic fixture components are found in Eric Oberg and F.D. Jones, Machinery’s Handbook (New York: Industrial Press, Inc., 1971) 19th ed., pp. 1883-1911.
Fig. 17-12. Cast iron fixture stock sections. a. U-Sections; b. L-Sections; c. T-Sections (equal); d. T-Sections (unequal); e. V-Sections; f. H-Sections; g. Flats; h. Square hollow blocks; i. Rectangular hollow blocks.
Design Studies I—Drill Jigs

The Five Basic Design Steps

From the discussion in Chapter 3, The Fixture Design Procedure, it becomes evident that there is a great deal of similarity in the design and design procedures for jigs and fixtures with respect to the principles of locating, clamping, and supporting; the principal difference being in the guidance of the cutting tool and, to some extent, in the support against the cutting forces. As outlined in Chapter 3, the systematic design of a fixture is comprised of the following five steps:

1. Designing a method of locating in the jig or fixture which will correctly orient the surfaces on the workpiece, for machining or other manufacturing operations.
2. Designing a clamping method that will hold the workpiece firmly against the locators and against the cutting forces.
3. If required, designing additional intermediate supports that may be needed to prevent the workpiece from springing or bending when it is subjected to the clamping forces and the cutting forces.
4. Designing or selecting the cutter guides; for drill jigs this means the drill bushings.
5. Designing the jig or fixture body to consolidate all of the components previously designed, into one unified structure.

These five steps are fundamental and of universal validity. In their application, the designer must consider the dimensions, material, and weight of the part to be handled in the fixture; the already existing surface finish and accuracy of its surfaces; the accuracy required in the operations to be performed; the quantity to be manufactured; the probability for multiple machining; and safety for the operator, the equipment, and the part. Furthermore, when working on the design, the designer must keep in mind the possible procedures that a toolmaker will employ to construct the jig or fixture.

For the purpose of evaluating the degree of sophistication and perfection to which the fixture should be designed, production quantities can be classified as follows:

<table>
<thead>
<tr>
<th>Type of Production</th>
<th>Number of Pieces</th>
</tr>
</thead>
<tbody>
<tr>
<td>Small lot</td>
<td>up to 40</td>
</tr>
<tr>
<td>medium lot</td>
<td>from 40 to 100</td>
</tr>
<tr>
<td>large lot</td>
<td>from 100 to 1000</td>
</tr>
<tr>
<td>mass</td>
<td>over 1000</td>
</tr>
</tbody>
</table>

This systematic fixture-design procedure will be demonstrated by the application of the five basic steps to a number of cases. This chapter covers drill jigs; the following chapters will deal with typical fixtures for milling and other machining operations.

Case Series

Case 1. A Jig Plate for a Small Part

Design a drill jig for drilling two 3/8-inch (10 mm) diameter holes in the part shown in Fig. 18-1. The part measures 5 by 2 1/2 by 7/8 inches (127 by 64 by 22 mm), is made of cold-drawn AISI 1030 steel, cut to length, finish-machined on the ends, and weighs 3 pounds (1 1/2 kg). Holes are to be drilled with a twist drill; reaming is not required. The quantity to be made is classified as small-lot production and called for the cheapest possible type of tooling. Obviously, the size and weight of the part presents no handling problem. Surfaces and edges are sufficiently flat and straight for locating, supporting, and clamping purposes. The simplest and cheapest type of tooling is a plate jig made to the same length and width as the part, as shown in Fig. 18-2. This solution correlates with the five steps as follows:
1. The jig is located by laying it on the part and lining it up along its edges. The part is further located by setting it, directly or indirectly, on the drilling machine table.

2. Clamping is done by auxiliary components, not integral with the jig. As shown in Fig. 18-3, the jig and the part (in this case a stack of two pieces) can be clamped on the drilling machine table by a conventional strap clamp with a bolt and heel block, or a simple clamping plate with a stud, strap clamp, and heel block could be made. Two low parallels under the work will provide clearance for the drill. A bolt fastened into the T-slot of the drill press table will prevent the work from spinning.

3. With the substantial thickness of the part and its regular shape and flat surfaces, there is no need for additional intermediate supports.

4. For minimum cost, the jig plate is made with holes to match the drill to be used, and it is made without bushings. This is acceptable for small-lot production. For larger quantities, drill bushings should be used as shown in Fig. 18-2. The thickness of the jig plate can be taken as 3/4 inch (19 mm), equaling two times drill diameter, for good bearing length (see Chapter 14, the section on Standard Bushings).

5. This step is automatically accomplished by making the jig plate in one piece.

In a conventionally dimensioned part drawing, the dimensions $a$, $b$, $c$, and $d$, shown in Fig. 18-4, will be given. For the drawing of the jig plate, the dimensions are transferred to coordinates, as explained in detail in Chapter 16. The upper long side and the left-hand short side are selected as the axes, so that the coordinates represent the distances from two edges of the part to the hole centers. Starting from this drill jig, which represents the simplest possible design, a number of steps can be taken to meet the more severe requirements of larger production and/
or larger dimensions of the workpiece. Since the parts have flat, parallel surfaces they can be stacked so that more than one piece is drilled at a time, as indicated in Fig. 18-3. For other, more advanced requirements, bushings are used, as indicated before, and the plate is made of a carbon steel in the AISI 1025 to AISI 1035 range. These steels are inexpensive and readily machinable and have sufficient natural hardness to resist wear and surface damage, caused by accidental nicking and scratching, reasonably well. They are also available cold rolled which, for the present purpose, would save the machining of the two large surfaces.

The operation of the jig is improved by the addition of locating points acting against two adjacent sides of the work, two on the long side and one on the short side, as shown in Fig. 18-5. The locating points are cylindrical pins, installed with a press fit. Such points are either dowel pins or commercial locating buttons (see Chapter 17). Since they act against flat, machined surfaces, they are provided with flat contact surfaces, as shown. The addition of pins makes the locating operation faster, and the positive contact with three pins ensures that all parts are drilled identically alike. Figure 18-6 shows a modified shape of the jig plate, suitable for casting, and resulting in a saving in weight for large jigs. When it is required to positively clamp the part to the jigs, lugs A, shown in Fig. 18-7, are added to the jig plate to carry finger screws for locking the part in position against the locating pins. This eliminates the danger of accidentally losing the correct position. For long and narrow workpieces two lugs on the long side are required, to avoid springing (elastically deforming) the part. A short and rigid part, such as the one considered here, could be sufficiently clamped with one lug only, as shown by the dotted lines at B. However, sloppy manipulation could result in inaccurate clamping. Finally, the jig can be provided with feet or legs, as shown in Fig. 18-8, which are screwed or pressed in place to the depth defined by shoulder A. If screwed, they are locked by means of small pins or headless screws. The legs are made of a carburizing steel or a tool steel, the ends are hardened and, after installation, ground and lapped at the ends to exactly the same length. Commercially available jig feet and legs are described in Chapter 17.

![Fig. 18-7. A jig plate with clamping means.](image)

**Case 2. An Open Drill Jig**

Design a drill jig for drilling two 5/8-inch (16 mm) diameter holes in the part shown in Fig. 18-1. The part measures 7 by 3 1/2 by 1 5/8 inches (175 by 90 by 40 mm), is fully machined from a gray iron casting, and weighs 10 pounds (5 kg). It is manufactured in repeated lots of 100 pieces each with holes to be drilled and reamed.

The quantities required can be classified as upper limit of medium-lot production, and justifies a complete jig with all components integral with or attached to the jig, but without devices for automatic action. With proper design, such tooling will ensure interchangeability of the parts independent of the operator's skill and care. Application of the basic steps leads to the type of jig known as the "open drill jig" which is, essentially, a plate jig with the part suspended underneath, and standing on legs, as shown in Fig. 18-9. Since it is open to one side, it is suitable for casting, because it requires little or no core work. The design shown is cast, and proceeds as follows:

1. Holes can be drilled from one side. The part has a machined flat surface that is perfect for locating against a machined surface on the jig plate, and regular edges for side location with two plus one
locating pins. The jig plate with pins can now be drawn. Minimum thickness is 1 inch (25 mm), and the machined face is raised 1/4 inch (6 mm) to provide a clearance all the way around for machining.

2. The part must be clamped from below against the jig plate. It would appear natural to provide a strap clamp at each end; however, the part is so rigid that it can stand up to the pressure from the drill with one strap clamp only, located at the center. This solution requires substantial dimensions. The strap is 2 3/4 inches (70 mm) wide by 1 inch (25 mm) thick and is slotted for the 7/8-inch (22-mm) clamping screw so that it can be pulled back and clear the locating area for inserting and removing the part. The heel block for the strap is provided as a downward extension of the jig plate. Three finger screws are provided for locking the part against the locating pins.

3. No additional supports are required because the part is already well supported over its entire area.

4. Drill bushings are inserted in the jig plates. The bushings are press-fit wearing bushings for use with slip bushings for drilling and subsequent reaming. The length of each bushing is 1 1/4 inches (two times hole diameter).

5. 1/4-inch (6-mm) bosses are added on the upper side of the jig plate to avoid protruding bushings. Vertical side walls are formed along the perimeter of the jig plate, to carry the finger screws. On the four corners, the side walls are formed into legs with angular (L-shaped) cross section, and made long enough to lift all components clear of the machine table. Legs are tapered 15 degrees for maximum rigidity with minimum weight, as shown in the sketch in the upper right-hand corner. Width b at the lower end is 1 1/2 times thickness a, which again depends on the jig size, taken as overall face area (length times width of the jig plate), as follows:

<table>
<thead>
<tr>
<th>Face Area of Jig</th>
<th>Thickness a</th>
</tr>
</thead>
<tbody>
<tr>
<td>Square inches</td>
<td>mm²</td>
</tr>
<tr>
<td>up to 6</td>
<td>4000</td>
</tr>
<tr>
<td>6 to 60</td>
<td>4000 to 40,000</td>
</tr>
<tr>
<td>over 60</td>
<td>over 40,000</td>
</tr>
</tbody>
</table>

This jig measures 10 1/2 by 7 1/2 inches (270 x 190 mm), and the legs are 3/4 inch (19 mm) thick. Pads A are provided on one long and one short side, to support the casting when laid out and machined. A 4-inch (100-mm)-long handle is cast on one end to give the operator a secure grip during drilling and reaming.

Case 3. An Open Jig with an Intermediate Support

Design a jig for a bracket with a large boss. The part is shown in chain-dotted lines in Fig. 18-10. It is a gray iron casting, approximately 13 inches
(330 mm) long and 8 inches (200 mm) wide, weighing 35 pounds (16 kg), with a large boss for a 2 1/4-inch (57-mm)-diameter bearing hole A, to be faced on both ends. It has a flange, already machined flat on its free side, to be provided with three screw holes B and two dowel-pin holes C, bosses for screw holes B to be spot faced. The initial lot will be 300 pieces and may be repeated at a later date. The awkward shape, weight, and quantity, call for a good jig without excessive frills. All operations, except a few facings, can be done from one side which also offers an excellent, flat, machined locating surface. The solution is an open jig that can be turned upside down for the few facing operations from the other side.

1. The jig plate can now be drawn. It is offset (Z-shaped) to conform with the height difference between the flange and the end of the boss and is provided with a raised, machined locating surface to receive the machined surface on the flange. Endwise locating is against two locating pins at the small end of the flange, with a screw at the other end to ensure contact. The pins are round (no flats) because they bear on the unmachined edge of the flange. Crosswise, the boss is located between two screws which allows for centering the boss with respect to the bushing for hole A, and thereby adjusting for possible variations in the castings.

2. The part is clamped against the jig plate with three strap clamps D, measuring 1 1/2 by 5/8 inches (38 X 16 mm) with slots for 5/8-inch (16-mm) clamping screws to allow them to be pulled back and clear the part.

3. The boss is overhanging and not sufficiently supported against the heavy cutting-tool pressure to which it is exposed. No possibility exists for supporting it on the free side, but a substantial intermediate support is provided by the 3/4-inch (19-mm) pressure screw E, carried by a 1 1/2 by 3/4 inch (38 by 19-mm) strap F which is screwed to the jig by two 5/8-inch (16-mm) screws G. Screw E supports the rib at a point as near as possible to the boss. Strap F has a hole at one end and a slot at the
other end, so that it can be swung clear of the part by loosening, but not removing, the screws G.

4. Holes B and C are plain drilled holes and require press-fit wearing bushings. Hole A is first drilled 1/8 inch (3 mm) undersize, resulting in 1/16-inch (1.5-mm) stock allowance for the final operation, which is done with a chucking or machine reamer. These operations require one press-fit liner bushing in the jig and two slip bushings for, respectively, the drill and the chucking or machine reamer. The tools for the facing of A, and the spot facing of B, are guided by pilots in the already finished holes, and do not require bushings.

5. A long and a short leg is provided at each corner. The overall dimensions of the jig are 17 1/2 by 10 1/2 inches (450 by 270 mm) which calls for 3/4-inch (19-mm)-thick metal in the legs. The jig plate is 7/8-inch (22-mm)-thick. Compared to the 1-inch (25-mm) thickness used in Case 2, this may appear to be on the low side, but the plate is substantially strengthened, first by its Z-shape which virtually makes it a structural beam, and also by the boss for the big bushing. Lugs, bosses, and pads are provided for bushings, screws, and clamps. Windows are cored out in the large flat panels for weight reduction. The result is a cast fixture that is strong, rigid, and as light as it can be under the circumstances.

Case 4. A Modified Procedure for Case 3

A drastic saving in tool cost, bought by an increase in the total machining time, is accomplished by a combination of turning and drilling operations. After a preliminary layout for the center, the large boss is drilled, perhaps bored, and then reamed and faced, either on a faceplate in a large lathe, or preferably, on the horizontal table of a vertical boring mill (VBM) or a vertical turret lathe (VTL). Holes B and C are drilled by using the jig plate shown in Fig. 18-11. The jig is located by a plug in hole A, and manually aligned with the periphery of the flange; the assembly is then clamped on the drill table by a strong strap with a jack screw for supporting the overhanging end.

Case 5. Design of a Closed Jig (Box Jig), of a Relatively Simple Type, with Positive Locating Means

Design a drill jig for drilling the four 5/8-inch (16-mm) holes in the part shown in Fig. 18-12.

The part measures 6 by 3 1/2 by 1 3/4 inches (150 by 90 by 45 mm). It is fully machined from a gray iron casting and weighs 10 pounds (4.5 kg). The quantity is small-lot production and calls for an inexpensive jig. All holes are drilled; their relative positions must be accurately maintained, but their location relative to the outline of the part is not critical. Holes A are drilled through, but holes B and C are blind holes and must be drilled from opposite sides.

When holes must be drilled in several directions, and bushings are required, the drill jig must be of the closed, or box, type, because it has to be turned around for the drilling operations. A closed, or box,
Fig. 18-12. A sample part used for the development of the closed drill-jig design shown in Figs. 18-13 through 18-19.

jig can be described as an open jig with a floor. Since the jig, more or less, completely embraces the part, a door, gate, or port must be provided to get the part in and out. With this difference in mind, most design features in closed jigs are the same as in open jigs.

The simplest solution is to make two jig plates as in Case 1, one for each side of the part, and build them together to form a closed jig, as shown in Fig. 18-13. The part is placed between the two jig plates and is located by two plus one locating pins, which have flats to engage with the machined edges of the part. The jig plates are held in their proper position by large-diameter dowel pins which have a press fit in the lower plate and a sliding fit in the upper plate. To prevent separation, a screw with a large flat head or a fully countersunk, hexagonal socket-head screw is installed in one of the dowel pins. Holes are drilled in the lower plate opposite the bushings for holes A to provide clearance for the drill and for the escape of chips.

Case 6. A Closed Jig with all Components Integral with or Attached to the Jig

Design a drill jig for the part specified in Case 5 to be manufactured in repeated lots of 100 pieces each. The design proceeds according to five basic steps and results in the jig shown in Figs. 18-14 and 18-15.

1. The part is located with its flat side on the machined pad R of the lower plate L, and sideways and endwise against two plus one locating pins with flats to match the machined side and end surfaces of the part, and is locked against the pins by two 1/2-inch (13-mm) screws U and Q. Obviously, one of these screws may block the part from entering, once the jig is fully designed and its closed character has become apparent. However, this problem will be solved in a later step.

2. The part is clamped against the pad R, by two 5/8-inch (16-mm) screws J, from above. The illustration shows an additional screw J (in dotted lines) to indicate that for parts of small dimensions one clamping screw in a central position may be sufficient.

3. The part is so effectively supported that no intermediate supports are needed.

4. 5/8-inch (16-mm) press-fit wearing bushings are provided, three from above and one from below. Bushings are 1 1/4 inches (32 mm) long for a bearing length of 2 times the drill diameter.

5. Upper plate K can now be drawn; thickness is 1 1/4 inches (32 mm), equal to bushing length. No end clearance is required for burrs and chips, because there is a space between the part and the plate.

Fig. 18-13. A simple closed jig with locating pins.

Fig. 18-14. A well-developed closed jig.

Fig. 18-15. A closed jig supported on parallels.
Side plates, also marked L (left side of the illustration), are made integral with lower plate L; the complete lower part can be machined from a solid steel block or from a U-shaped gray iron casting. The locating pad R has a clearance groove on each side. Wall thickness is a uniform 1 1/4 inches (32 mm). The upper plate is fastened with 1/2-inch (12 mm) screws M, and secured in position with dowel pins N. To carry the locking screw Q, a swinging arm P is provided, which can be swung out of the way for loading and unloading the jig. For the drilling of hole C in the underside of the part, the jig is turned upside down and placed on parallels D (Fig. 18-15). The addition of the machining strips f will save the finish machining of some quite large areas of the jig.

Case 7. A Closed Jig with Legs

Design a drill jig for the part specified in Case 5, with integral legs for the two operating positions. The jig is shown in Fig. 18-16. Steps 1, 2, 3, and 4 are the same as in Case 6. In Step 5, a boss for the bushing and four foot pads are added to the lower part; four legs and a machining pad around the bushings and the clamping screws are added to the upper plate. Machined surfaces are marked f.

![Fig. 18-16. A closed jig provided with feet and legs.]

The screw Q is now carried by a 1 1/4 - by 5/8-inch (32 - by 16-mm) swinging strap E, which is supported at both ends and, therefore, provides a more rigid and secure position for the screw. A handle S is added on the end of the jig to give the operator a safe grip.

Case 8. A Closed Jig Designed as an Improved Type of Leaf Jig

Design a drill jig for the part specified in Case 5, with a swinging leaf, but without clamping means in the leaf. While it may be a convenience to use a commercial leaf jig with the clamping screw, or screws, placed in the leaf, it also has its valid objections.

The clamping pressure is carried by the leaf and causes an elastic deflection; if the tight fit in the hinge and in the seats against the jig body is loosened by use and wear, the position of the leaf may shift when the clamping screw is tightened, causing inaccuracy in the direction and location of the drill bushings and thus faulty work. For best results it is therefore generally recommended to separate bushings and clamping means and to use the leaf for only one of these two types of components, as shown in Fig. 18-17 where the clamping is done by two 1 1/2-by 7/8-inch (38- by 22-mm) strap clamps G with 1/2-inch (12-mm) screws, one at each end. As usual, the straps are slotted for easy withdrawal. This arrangement is always recommended when the clamps have to take a heavy drilling load as, for example, in operations using a multiple-spindle drilling machine.

![Fig. 18-17. A closed jig with a leaf and clamps.]

The leaf is here a 1 1/2- by 1-inch (38- by 25-mm) strap, carrying the bushing for hole B and locked by means of a thumb screw H, sometimes formed as a quarter-turn screw.

Case 9. A Closed Jig with a One-piece Body and No Swinging Parts

Occasionally, it is possible to design a jig that is sufficiently closed to carry bushings for all required drilling directions and which still provides an opening, a port, large enough to bring the part in and out. Essentially this depends on the hole location in the part. Chances are best if the holes are placed near the edges.

An example is shown in Fig. 18-18. The part is the one specified in Case 5. An examination of
Fig. 18-18. A closed jig with a port for loading and unloading the part.

Fig. 18-12 shows that hole B is located fairly close to the two edges. The jig is now designed with only hole B drilled with the jig upside down, and the other holes with the jig in the upright position. The bushing for hole B is carried by a boss held on a bracket D strengthened by a rib E. With the clamp to the right withdrawn, the end of the part can be lifted and the part pulled out in a tilted position. There must be enough clearance, not only for the part, but also for the operator’s fingers. The designer is warned against overoptimism in this respect. In reality, parts are always larger and clearances smaller, than they appear to be in a drawing.

Case 10. A Closed Jig for Drilling from Four Sides

Design a drill jig for the part specified in Case 5 with an additional hole in the long side and one in the end.

The jig with the part is shown in Fig. 18-19. Steps 1 and 3 are the same as in Case 8. In Step 2, clamp H is moved over 7/8 inch (22 mm) to clear the new bushing for the end hole, and clamp G is moved opposite to maintain the balance of the clamping forces. In Step 4, bushings E and F are added for the side and end holes. In Step 5, additional pads are added to act as feet for the drilling operations through bushings E and F.

General Definitions and Classifications

Drill jigs are used exclusively for drilling, reaming, tapping, and facing operations. Whenever a combination of these operations is required on a part, it is usually possible to design one single drill jig for all of them. Drill jigs may be classified as open jigs and closed jigs. Some sub classifications within these two main classifications can also be made. The open jig has all bushings mounted in the same plane and with parallel axes. It has no removable walls or leaves, thus it is easy to insert and remove the part. The simplest type of open jig is the template jig, widely used in the aircraft industry, consisting of a large sheet of aluminum, magnesium, fiber, or laminated plastic with the necessary bushings, and the means for locating it on the part and holding it in position.

The most typical form of open jig in the average machine shop is the plate jig; it is applied to, and supported by, the work. It may require clamping, but in many cases this type of jig is used on a finished portion of the part which provides means for holding the jig in position. The next step in development is the plate jig with feet, with the part clamped below the jig plate. If the four feet are replaced by two parallel walls, the jig becomes a channel jig. If two more side walls are added, the jig may still be a plate jig, but if these side plates now are used for installation of drill bushings, then the jig has developed into a box jig.

The open channel jig, as well as any one of the more closed type jigs, can be provided with a cover, often in the form of a leaf, for fast operation. The box jig is frequently formed as a tumbling jig, that is, it has feet in several directions and is simply turned over 90 degrees or 180 degrees when the next side is going to be drilled. If there are hole axes at angles other than 90 degrees, the jig can be supported in a cradle, and if there are a large number of axis directions, the jig will be made as an indexing jig.

Operating with Drill Jigs

Before a jig is designed, it must be decided whether the drilling operation will go to a fixed-spindle drill press or to a radial drill. Small parts that are easy to
handle should always be routed to a fixed-spindle drill press where the jig with the part is moved so as to bring one bushing at a time in line with the drill spindle.

The moving of the part is usually done manually, but when necessary, is assisted by a hoist or a crane. The use of an air cushion table greatly facilitates the shifting of heavy tooling and is gradually gaining acceptance in workshops. If there are holes in more than one direction, the jig is designed either as a tumble jig or as a bracket-type indexing jig, depending on the clamping possibilities found in the part.

Large and heavy parts go to the radial drill because this machine has a spindle that is easily moved from one hole position to the next, which permits the jig to be clamped to one of the machine tables. If the part has holes in more than one direction, there is again the choice between a tumbling jig and an indexing jig. A less than clearcut situation arises with medium-size parts, parts that can be manually handled on the table of an ordinary drill press, but not without some physical effort by the operator. The tendency is to prefer the radial drill if it is available. Modern radial drills have ample rigidity and a larger range of speeds and feeds and usually more power than drill presses of similar spindle dimensions, also all handles and levers for the control of the machine are located within easy reach of the operator. They are designed for convenient and fast manipulation with a minimum of physical effort.

A most important safety rule applies to the manual handling of drill jigs. The torque exerted by even a relatively small drill exceeds what can be safely held by hand. Any drill jig, if it is not physically clamped to the machine table, must be positively restrained against rotation. A stop block may be sufficient for this, but a straight bar or rail, contacting one full side of the jig, is better. A set of two such parallel rails, with the jig sliding between them, is frequently an excellent solution. A rule-of-thumb says that the work can be held by hand when drilling holes of 1/4-inch (6-mm)-diameter or less. Even in this case the jig must be of such a shape and size that the operator has a good grip on it, or it must be provided with a handle.

The planner must recognize that the drill bushing steals some of the available length of drill spindle travel. The following values are average and common:

![Fig. 18-20. Using a medium-size drill jig.](image-url)
Typical examples of medium- and large-size drill-jig work are shown in Figs. 18-20 and 18-21. The first illustration shows a medium-size part that is to be drilled in two directions. In the position shown, the jig is supported on a block of such thickness that the height to the top of the jig equals the length of the jig. When the jig is turned 90 degrees for the drilling of the end hole, the height is the same as before, and no adjustment of the clamps is needed. The second illustration shows a box jig with a separate jig plate. The jig is mounted on the indexing table of a large work positioner that is not part of the jig.

Placement of Jig Bushings

The fixture designer has no option with respect to the placement of the bushings, because they are determined entirely by the drawing of the part. A recurring problem is that holes are so close together there is no room for the drill bushings. The various possible solutions are shown in Fig. 18-22. The use of thin-wall bushings may solve the problem, or two standard headless or headed bushings are each ground with a flat side and are installed with the two flat sides in contact. In extreme cases it is necessary to include two or more holes in one single insert.

The distance from the end of the bushing to the surface of the work is important. Various possible arrangements are shown in Fig. 18-23. For maximum accuracy, the bushing should contact, or almost contact, the work to provide maximum support and guidance to the drill. When full contact with a tight close-up fit on the surface of the part can be established, this type of fit is preferable because it positively prevents chip jamming below the bushing. It can be achieved with the use of the threaded type of bushing shown in Fig. 14-15d, and later, in Fig. 18-28, view A, and also in many cases with plate jigs. In all other cases, it is preferred, even necessary, to maintain a short clearance between the bushing and the surface. The distance is taken as 0.5 times drill diameter for materials that produce short chips,

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<th>Maximum Drill Diameter $A$</th>
<th>Drill Spindle Travel</th>
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<td>inch</td>
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<td>19</td>
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Fig. 18-21. Drilling a large part with a drill jig plate.

Courtesy of LeBlond Inc.
and 1 to 1.5 times drill diameter for materials that produce long chips. Materials with long chips can also be drilled with the bushing in contact with the work. In this case, the chip is continuously guided up to the surface of the jig. Excessive chip clearance is not used because it reduces the guiding effect of the bushing. Burr clearance for highly ductile materials, such as copper, is 0.5 times drill diameter.

Drilling on non-perpendicular surfaces is facilitated (see Fig. 18-24) by carrying the bushing down to the work surface and cutting the end to the contour of the workpiece. In applications of this nature, the drill point has a strong tendency to skid or wander. For this reason, the distance between the bushing and the workpiece must be held to a minimum so that the full guiding effect of the bushing can be
obtained. The side load exerted by the drill in applications of this type is usually concentrated at a point near the drill-exit end of the bushing and causes accelerated bushing wear. Except in short production runs, the use of fixed renewable bushings should be considered, to simplify replacement of worn bushings and to facilitate proper orientation of the bushing with respect to the contoured work surface. When press-fit bushings are used, bushing end contours should be applied after the bushings are installed in the jig plate to assure proper contour placement with respect to the workpiece.

The Open Plate Jig, Reversible Jigs

This type of jig is simple, inexpensive, efficient, and can, if made reversible, be used for the drilling of matching parts. The ring-shaped jig shown at A in Fig. 18-25 is used for drilling the stud bolt holes in a cylinder flange and also for drilling the cylinder head, which is bolted to the cylinder. The position of the jig when the cylinder flange is being drilled is shown at B. An annular projection on the jig fits closely in the cylinder counterbore to locate the jig concentric with the bore. As the holes in the cylinder are to be tapped or threaded for studs, a tap drill, which is smaller in diameter than the bolt body, is used and the drill is guided by a removable bushing of the proper size. Jigs of this type are often held in position by inserting an accurately fitting plug through the jig and into the first hole drilled, which prevents the jig from turning with relation to the cylinder, when drilling the other holes. When the jig is used for drilling the head the opposite side is placed next to the work, as shown at C. This side has a circular recess or counterbore, which fits the projection on the head to properly locate the jig. As the holes in the head must be slightly larger in diameter than the studs, another sized drill and a guide bushing of corresponding size are used. The cylinder is, of course, bored and the head turned, before the drilling is done.

Fig. 18-24. Bushings for drilling through non-perpendicular surfaces.

Fig. 18-25. A reversible, open-plate jig.
The jig shown in Fig. 10-46 is an open, but not reversible, plate jig. The jig shown in Fig. 18-26 at A and B is a centralizing open-plate jig and is used for drilling the ring shown at C. For centralizing the jig on the work, it has three plungers D, which are held against the conical point of wing screw E by springs F. In operation, the wing screw E is turned back until the plungers D are well within the body G, at points H. The ring C is then drifted on and the wing screw is turned down until the plungers D are forced out and into contact with the inside surface of the ring. The ring is then drilled on a sensitive drilling machine, e.g., an upright drilling machine with hand feed only.

The Box Jig

The leaf jig is a commercially available box jig and one example, designed for drilling a hole having two diameters through the center of a steel ball, is shown in Fig. 18-27. The work which is shown enlarged at A, is inserted while the cover is thrown back, as indicated by the dotted lines. The cover is then closed and tightened by the cam-latch D, and the large part of the hole is drilled with the jig in the position shown. The jig is then turned over and a smaller drill of the correct size is fed through guide bushing B on the opposite side. The depth of the large hole could be gaged for each ball drilled, by feeding the drill spindle down to a certain position, as shown by graduation or other marks, but if the spindle has an adjustable stop, it is preferably used. The work is located in line with the two guide bushings by spherical seats formed in the jig body and in the upper bushing, as shown. The work can be inserted and removed quickly, and a large number of balls can be drilled in a comparatively short time.

A typical, custom designed box jig is illustrated in A in Fig. 18-28, where A, B, and C show the three positions in which this jig is being used. A is the
loading and unloading position and B and C are the two different drilling positions. The work, in this case, is a small casting with its form indicated by the heavy dot-and-dash lines. This casting is drilled at a, b, and c, with the two larger holes a and b finished by reaming. For inserting the work, the hinged cover of this jig is opened by unscrewing the T-shaped clamping screw s one-quarter of a turn, which brings the screw head in line with a slot in the cover. The casting is clamped by tightening this screw, which forces an adjustable screw bushing g down against the work. As this bushing is adjustable, it can be set to give the right pressure, and, if the height of the castings should vary, the position of the clamping bushing is easily changed.

The work is properly located by the inner ends of the three guide bushings a₁, b₁, and c₁ and also by locating screws l against which the casting is held by knurled thumbscrews m and n. When the holes a and b are drilled, the jig is placed with the cover side down, as shown in B, and the drill is guided by removable bushings, one of which is shown at r. When the drilling is completed, the drill bushings are replaced by reamer bushings and each hole is finished by reaming. The small hole c, is drilled in the end of the casting by simply placing the jig on end as shown in C. Box jigs which have to be placed in more than one position for drilling different holes are usually provided with feet or extensions, as shown, which are accurately finished to align the guide bushings properly with the drill.

These feet extend beyond any clamping screws, bolts, or bushings which may protrude from the sides of the jigs, and provide a solid support. When inserting work in a jig, care should be taken to remove all chips which might have fallen upon those surfaces against which the work is clamped and which determine its location.

Another jig of the box type, which is quite similar to the one shown in Fig. 18-28, but is arranged differently, owing to the shape of the work and location of the holes, is shown in Fig. 18-29. The work has three holes h, in the base, and a hole at i which is at an angle of 5 degrees with the base. The three holes are drilled with the jig standing on the opposite end y, and the angular hole is drilled while the jig rests on the four feet k, the ends of which are at such an angle with the jig body that the guide bushing for hole i is properly aligned with the drill. The casting is located in this jig by the inner ends of the two guide bushings w and the bushing o, and also by two locating screws p and a side locating screw q. Adjustable screws r and t₁ in the cover hold the casting down, and it is held laterally by the two knurled thumbscrews u and v.

**Jigs for Angular Drilling**

When the work is to have angular holes, that is, holes that are to be drilled at an angle with its basic surfaces or planes, the jig must be supported in an inclined position. For drilling only one angular hole,
or a group of such holes with parallel axes, the jig is designed as shown in Fig. 18-30. The work, shown here as a rectangular block, has one angular hole

which is to be drilled through the bushing A, and the feet on the opposite side are machined to a plane perpendicular to the axis of the bushing, as indicated by the angle $\alpha$. The feet $B$ are machined to a plane parallel to the faces of the work and are used for supporting the jig for the drilling of perpendicular holes. When it is required to drill one hole, or a group of parallel holes, at an angle and other holes perpendicular to the face of the work and from the same side, an arrangement such as that shown in Fig. 18-31 is needed. The bushing for the angular hole is A. On the opposite side, the jig has feet of equal length for support when the perpendicular holes are drilled. A separate base (also known as a stand, an angle block, or a cradle) $B$ is provided to support the jig in the required inclined position for drilling through bushing $A$. Separate bases are used not only for angular drilling, but sometimes to accommodate the jig in cases where it would be inconvenient to provide the jig with either feet, finished bosses, or lugs, for resting directly on the drilling machine table.

The use of a separate base is applicable to jigs of almost any size. It does require handling and is therefore constructed as light as possible, with a
large cored hole in the panel and with ribs and webs for rigidity.

Manual handling is greatly reduced by the use of a jig with a swinging leg as shown in Fig. 18-32. This jig is designed for the drilling of two holes, one of which is at an angle. When drilling the straight hole, the jig is in the position shown at A; for drilling the angular hole, the operator simply lifts the front of the jig, and the swinging leg C falls, bringing the jig into the position shown at B, and places the hole to be drilled in line with the drill. By using this jig, extra parts, such as a separate base or angle block, are eliminated, and the jig is very quickly moved between operating positions.

Jigs for Large Work

When a jig of large dimensions and weight is to be turned over, either for the insertion or removal of the work, or for drilling holes from opposite sides, it is advantageous to have a special device attached to the jig for turning it over. Figure 18-33 shows one such arrangement for use where a crane or hoist is available. A represents the jig which is to be turned over. The two studs B are pressed or screwed into the jig in convenient places, as nearly as possible in line with a gravity axis. These studs then rest in yoke C, which is lifted by the crane hook placed at D. The jig, when lifted off the table, can then easily be swung around. The yoke is made of round machine steel.

For work of medium size and weight (in the range from 8 to 25 pounds) where crane assistance is not feasible, much hard work can be saved and production increased by outfitting the jig with rockers where that can be done without interfering with the drilling operation. An example is shown in Fig. 18-34. The work requires drilling from two opposite sides, as indicated by the bushings and legs shown, and the third side is available for the rockers. They are made from steel plate, machined to a radius, and attached with screws. The machining of the curved contour does not require high precision, since the jig does not rest on the rockers during drilling operations.

The Vise as a Drill Jig

The machine vises such as are used for milling or planing operations may be used for drilling when they are provided with attachments for holding drill bushings or locating stops. By using suitable plates in these jigs, many odd-shaped pieces can be drilled, and Fig. 18-35 is a typical example. The method of using this plate is shown by the illustration. Bushings A are placed in plate B at the proper location to guide the drills into the work. The plate is screwed on top of the vise, stop C is adjusted to the proper location, and the work D is placed in the vise against the stop, after which the holes are drilled.

Another example of drilling in a vise is shown in Fig. 18-36, where a number of holes are drilled around a circle. The work is gripped between the jaws in the vise proper, and a bushing plate is located by pins A and B in the vise. By sliding the vise to various positions the holes are drilled in the usual manner. This bushing plate is removable for taking out the work.

A jig construction adapted to drilling holes on an angle is illustrated in Fig. 18-37. In this case, a
swivel vise is fitted with a plate A, which can be set at the proper angle in relation to the base B by swinging the vise around axis C.

Jigs for Multiple Spindle Drilling

Multiple-spindle drilling machines and multiple-spindle drill heads mounted on single-spindle drill presses have their drills already set in the required pattern. Some of these machines and drill heads carry their own drill jig, when they are provided with drill bushings mounted in the manner shown in Fig. 14-16; they require only a work-positioning fixture.

When no such devices are present, the multiple spindle machines and drill heads need a drill jig for locating and clamping the part and for guiding and supporting the drills. This is particularly important as the drills, in many cases, are small and long.

These machines are used in mass production, and fast manipulation of the drill jig is of the greatest importance. Depending on the number of drills in operation, the load on the drill jig is usually quite large.

Indexing Drill Jigs

Indexing devices are described in Chapter 6, Design of Locating Components, and two indexing drill jigs
are shown in Figs. 6-35 and 6-36. A special case of an indexing drill jig where the part is its own indexing plate, is shown in Fig. 18-38.

This jig was used for drilling dial plates of the form employed on automatic feed mechanisms for power presses. These dial plates had the center hole bored and the notches milled to suit the locating plungers on the power presses, but the holes had to be drilled later because they were located with reference to the particular presses on which the dials were used. Before using the drill jig it was necessary to make center punches to fit the punch-blocks on the different power presses and also to fit bushing A in the jig. Each dial plate B was then put on its bed and the press was set in the usual way, care being taken to have the locking device fit properly in one of the notches. The center punch was then mounted in the punch-block and one prick-punch mark was made on the dial in proper relation to one of the notches. The dial plate was next placed on the table of a drill press and the center punch was set in the chuck in the drill spindle so that the prick-punch mark on the dial could be lined up with the spindle. The plate was then strapped to the table and stud C

![Fig. 18-37. A tilting vise with a jig attachment.](image)

![Fig. 18-38. An indexing drill jig.](image)
driven into the center hole. The top of stud C was machined to fit the pivot hole in the arm D of the jig.

The next step was lining up the bushing A of the fixture with the center punch in the drill spindle. The bushing was made adjustable relative to the center C, about which the arm swung, so that it could be set in the required position before clamping the binding bolt. The bushing was located in proper relation to the notches in the dial plate by means of the locking pawl E, and the eccentric screw F adjusted the position of the pawl relative to the arm D of the jig. The pawl was held in the proper notch in the dial by spring G which was mounted on pins H and I; and stud J was used to hold the arm of the fixture true with the face of the dial plate. It will be evident that after this setting had been made, bushing A would be located directly over the center punch mark which was made on the dial plate while the prick-punch was mounted in the punch-block of the power press. The hole could then be drilled in the dial plate, after which successive holes were drilled by simply swinging the dial around pivot C, and locking it for drilling each hole by dropping pawl E into successive notches in the dial plate.

Miscellaneous Drill Jigs

Interesting and characteristic types of drill jigs have been used as examples in previous chapters. The jig shown in Fig. 11-2 is a clear-cut example of a box jig with some good design details. Drill jigs for small parts that require the hole exactly in the center of the part are shown in Figs. 9-17, 9-18, 9-19, and 9-20. A simple, inexpensive, and very versatile jig is the bracket type of drill jig shown in Figs. 10-37 and 10-38.

Where a large number of parallel holes (six or more) all require the use of one or several slip bushings, the time required for inserting and removing the bushings becomes substantial. Considerable time can be saved by using a bushing holder, a plate carrying all slip bushings of the same type, where all bushings are removed and inserted in one manual operation. The cost of a bushing plate is negligible.

The smaller the part, the more important it is that the operation is fast, and much can be gained by using sophisticated, yet simple, devices for closing the jig and clamping the part. The drill jig shown in Fig. 18-39 is a good example of the simplicity of design that can be attained by using the bayonet-lock type of clamp. A clamp of this type also keeps the loading time down to the minimum required for economical production. The jig is composed of only six pieces—the body, clamp, pin, and three bushings. The side of the body opposite the drill bushing for the angular hole in the workpiece is machined at an angle of 90 degrees to the axis of the bushing hole to serve as a base while drilling the hole. This arrangement eliminates the necessity of providing a separate angle-block.

Fig. 18-39. A jig with quick-acting bayonet clamp.
The jig is designed for drilling the angular hole at A and the two holes B and C at opposite sides of the work. The workpiece comprises a subassembly of a high-pressure valve and stud for a sensitive air-control valve which is part of an air brake.

The body of the jig is bored to a slip fit for the workpiece. The opposite end of the jig is flattened on both sides of the center to meet the bottom of this bore. This provides openings at D and E for the escape of chips. A clearance hole F is also drilled through this end to clear the stud in the workpiece. The three drill bushings are pressed into holes that are accurately positioned in the jig body. The clamp is a slip fit for the hole in the body, which is made larger than the locating bore for the work, so that the jig will be easier to load. The top of the hole in the body and the end of the clamp are chamfered to facilitate insertion of the clamp. Bayonet slots G on the sides of the body are made a slip fit for the pin pressed into the clamp. These slots have a radius bend and a lateral section which permit the pin to be given a clockwise turn. These lateral ends of the slots are machined at an angle of about 95 degrees to form cam surfaces which give the bayonet lock its clamping action against the workpiece. The lateral slot on one side extends in the opposite direction from that on the other side. The work should be clamped when the pin is at about the middle of the lateral part of the slots.

The length and diameter of the hub on the end of the clamp are such that the hub clears the plane of the flat, on the side of the body on which the jig rests when drilling the angular hole. This flat provides sufficient surface beyond the center line of the bushings to permit drilling one side hole, and the angular hole, without causing the jig to tip. Because it is necessary to have a small hub at the end of the clamp, a hexagon socket is machined in it to fit an Allen wrench, so that the clamp can be easily tightened or loosened.

The jig shown in Fig. 18-40 has an equalizing member for the combined clamping and closing operation, and is used for drilling and tapping stud A, which is made from 1/4 X 1/4-inch (6 X 6-mm) cold-drawn steel. The end of the stud enters hole B in the locating block, and this hole is milled to provide clearance for the head of the stud. The work rests on the drill bushing which is slightly counterbored to provide clearance for the tap. The interesting feature of the jig is that the cover and clamping mechanism are both secured by the same knob; clamp C is swung around its pivot to hold the stud securely in place when the knob is screwed down, and the same operation tightens the cover. This principle permits fast opening and closing of the jig, and can be employed on jigs and fixtures used for holding a great variety of parts.

Occasionally a hole must be drilled in the interface between two parts that are fitted together, and a pin is driven into the hole to act as a lock or key. In job shop work this is done at assembly; under manufacturing conditions it is preferable to perform these operations on the two parts separately, prior to assembly. To drill such a "half" hole, it is usually necessary to plug up the hole in the work in some way that will back up the side of the drill that is not cutting. This is accomplished, as shown in Fig. 18-41, by means of a hardened stud A with a semicylindrical groove that matches half of the surface of the drill. The stud has a push fit in the work and backs up the drill during the drilling operation. An angle iron or plate, B, is attached to stud A and held in position by bolt C; plate B is also doweled in place. A hole is drilled in this angle iron to receive bushing D, which guides the drill in the usual manner. The remainder of the jig consists of the key E which locks the jig in place on the work.

In using this tool, key E is pulled back, clear of the work, and stud A, which carries the angle iron, is pushed into the hole until the stud moves up against the shoulder of the work. By pushing up tapered key E until it binds on the flat of the work then tapping it lightly, the jig is held securely in place.
When drilling the hole, the work is set up on end on the drill press table and the drill is then fed through the bushing in the usual manner, the bushing holding the drill in position until it starts to cut. As the drill is fed down, there is a tendency to force it away from the work, but this tendency is resisted by the hardened stud A so that the half hole is drilled parallel with the axis of the work. Even with a drill jig, this is a difficult operation. When drilling with an ordinary twist drill there is a tendency for the drill to "hog in," which is apt to result in the tool breaking. For this reason, a drill with zero rake angle is recommended; either a straight-fluted or farmer's drill or an ordinary twist drill ground in such a way that it has no rake. A jig that clamps quickly, and with spring pressure, is shown in Fig. 18-42. It is a representative example of a homemade pump jig. The jig is shown empty. A drill bushing A is mounted in a movable traverse which, by spring pressure, is forced down and clamps on the work. One locator is installed under the bushing and another locator B is carried on a bracket C. To unclamp and open the jig, the operating handle D, is depressed. It swings around pivots E and lifts rods F which in turn lift the traverse with the bushing against springs G.

**Tooling for N/C Drilling Machines**

In N/C drilling machines, the locating of the spindle is automatically controlled from a tape- or card-operated electronic control unit. Consequently, these machines do not require the conventional type of drill jig with bushings for locating the drill relative to the work. Most, but not all, N/C drilling
machines are used for job shop or short run operations. This is a highly competitive type of business, tooling expenses must be kept low and the tooling is reduced to the simplest possible type of fixture for the sole purpose of supporting and clamping the work. Quite often, the job is done entirely without special tooling.

The clamps used are, as a general rule, hand-operated strap clamps. Another general requirement is low construction height to avoid collision when the table sweeps back and forth under the spindle. Two characteristic examples are shown in Figs. 18-43 and 18-44. In each case, the fixture consists essentially of an aluminum tooling plate with a few accessories. In Fig. 18-43, the part is centered in a circular recess in the plate which also carries four studs for the strap clamps. In Fig. 18-44, the part is located against locating pins (visible on the right-hand side) and clamped with strap clamps having knurled-head hand knobs.

Where N/C drilling is applied to large-volume production, it becomes economically feasible to use quick-acting and more sophisticated clamping devices. A typical example is the drilling of circuit boards. These boards are manufactured in fairly large quantities, in widely different sizes and are stack drilled. For these reasons, the clamping devices must be horizontally and vertically adjustable and quick acting, as shown in Fig. 18-45. The carriage provides the horizontal adjustment for varying board sizes, and the air clamp cylinder has sufficient length of travel to accommodate stacks of
boards of varying height. Any automatic clamping device should comprise a safety feature against accident or damage in case of failure of the operating pressure. In the present case, this is accomplished by a pressure switch. Should the air pressure drop, the machine will stop. In other cases it may be done by means of a toggle clamp or a self-locking eccentric, cam, or wedge.

N/C drill fixtures, as well as other N/C fixtures, usually incorporate a locating device to establish one spindle position in relation to a reference point or surface on the part. This position serves as the correct starting point for the programmed sequence of all following spindle locations.

With the absence of bushings, the drill point has no support and guidance as it meets the surface of the part, and it may "walk" a little before it enters the metal. The position accuracy is therefore less than with the use of bushings. If the position tolerance is ±0.010 inch (±0.25 mm), a total position variation of 0.020 inch (0.5 mm) or less, it is necessary first to spot drill with a short and rigid center drill, or to use a twist drill with a spiral ground point.

A drastic and illustrative example of the saving in fixture cost that may be realized by replacing conventional drilling machine equipment with an N/C drilling machine is shown in Fig. 18-46. The part is a casting for a fuel pump housing, and the fixture required for N/C drilling of this part consists of a base plate, an angle plate, a bolster plate, and a clamping stud with nut. The total cost of this fixture was less than $600.00, while the cost of the various drill jigs required for conventional drilling was over $5000.00.

An additional case is the instrument frame shown in Fig. 18-47a, which requires considerable machining with end mills, and the drilling of a large number of holes.
of holes some of which also require tapping. These operations are performed on an N/C machine tool in the setup shown in Fig. 18-47b. The fixture is built up on a tooling plate as the base and consists essentially of rectangular blocks, bolted to the base. They carry the part, one cam-operated clamp, two toggle clamps, and two circular locators matching the contour within the three small lugs inside the two circular openings. By these simple means, the part is supported, located, and clamped. The clamps which are here shown in their retracted positions, present a low profile relative to the part.
Design Studies II—Milling Fixtures

Milling operations are characterized by large, periodically varying cutting forces, producing a large volume of chips, usually of small size. The tool may be a single milling cutter or a set of cutters. The operation is normally completed in one pass of the cutter. In most operations the path of the milling cutter relative to the work is a straight line. However, the fixture may be clamped on a revolving table for cutting an arc of a circle, or some other curve may be cut as in tracer controlled and contour milling.

To meet these conditions, milling fixtures must be sturdy, with relatively large locating and supporting areas and very strong clamps. Wherever possible, cutting tool pressure is taken up by positive stops, rather than by friction, which may fail under vibration. To reduce loading and unloading time, fixtures for volume production are equipped with pneumatically or hydraulically operated clamps. Hydraulic operation is preferred, since oil has less inherent elasticity than air, and because hydraulic actuators can be made with smaller dimensions for the same clamping force. Pneumatic and hydraulic clamping devices must have a safety locking feature, as explained in Chapter 18, page 242, to prevent accidents in case of a power failure.

In principle, a milling fixture is a box, preferably of open design, i.e., open at the top or at one side for giving easy access to the cutter for reaching the surface to be machined, and also to the locating areas for cleaning away chips. These two cases are illustrated in Figs. 1-2 and 1-3, which show most of the locating, supporting, and clamping components that are typical for milling fixtures. Attention is drawn to the tool setting block (1 in Fig. 1-2) with which the milling cutter is positioned for the correct location of the cut. The gage pin in Fig. 1-3 has the same function.

These features are clearly seen in the fixture shown in Fig. 19-1. The fixture base is mounted vertically by means of an angle plate, as the face-milling operation is done on a horizontal milling machine. For the same operation on a vertical milling machine, the fixture would be mounted directly on the machine table.

The milling machine vise with detachable jaws or inserts, contoured to fit the part, provides many opportunities for the design of inexpensive milling fixtures. Design details are given in Chapter 10, Clamping Elements. In production milling it is often economically advantageous to use more than one fixture. The combined length of the run-in and run-out distance for the milling cutter is usually quite significant relative to the net length of the machined surface, as illustrated in Fig. 19-2. A considerable saving in operating time is accomplished by string milling, where a number of identical milling fixtures are mounted as closely together as
possible, in a line, on a common base. Duplex milling, i.e., milling of two parts in one operation, is a common as well as a profitable operation. Multiple-spindle milling machines naturally require multiple milling fixtures and an example is shown in Fig. 19-3.

Case 11. Design a fixture for the milling of the surfaces on the back side of the part shown in Fig. 19-4, the housing for the lead screw drive on a medium-size engine lathe.

The part is a gray iron casting, weighing 45 pounds (20 kg). In comparison, the complete fixture weighs empty, 95 pounds (43 kg). The system of surfaces to be machined on the back side consists of an upper and lower longitudinal recess extending over the entire length of the part, and recesses on two bearing parts at and near the right-hand-end of the part. They can be gang milled in one pass with the set of milling cutters shown in Fig. 19-4. This operation is selected as the first step because it constitutes the major single operation on the part, and it provides excellent locating surfaces for all subsequent procedures. The design develops as follows:

1. For the first operation, the part must be located and clamped entirely on raw cast surfaces. This presents no problem because the part has a regular geometry, with large flat surfaces at right angles to each other, and, in addition, there are three bosses in a triangular pattern on the front. Furthermore, all surfaces to be considered for locating, supporting, and clamping are free of casting contaminations, such as mismatch and flash. In fact, the part offers the possibility of a classical application of the 3-2-1 principle, using hardened spherical buttons as the locating points. The locators are shown in Fig. 19-5 and are identified by (1). Three buttons carry the part on the three bosses; two buttons on one side align the part, and one button on the end locates it endwise.

2. The part is clamped against the locators by three 5/8-11 UNC (16 × 2mm) bolts, (2), arranged opposite the side and end locators. The clamping bolts are inclined 5 degrees, so that they aim below the side and end locators and force the part down on the three base locators.

3. A critical examination at this stage shows that the part is not fully stable. If large forces are applied outside the locator triangle and near the two corners, the part may tilt over one side of the base triangle by slipping slightly under the clamping bolts. To prevent this, one or several intermediate supports are needed. Applying two more base supports near the corners will provide the classical, rectangular, and very efficient, support pattern; however, such supports must be individually adjustable, as they must act on raw surfaces. Screw jacks may be ruled out since they would not be easily accessible. Spring loaded jacks are quite long and would therefore raise the part a considerable distance above the machine table, thus substantially
sacrificing rigidity in the setup. Endwise, there is no such dimensional limitation, and a spring loaded jack is mounted symmetrically with, and parallel to, the end locator. The jack applies itself to the surface of the part by spring pressure, and is then secured by the hand knob locking screw. These parts are identified by 3.

4. The relative location of the individual milled surfaces is defined by the milling cutter assembly, and the cutter guide has to locate only one corner of one cutter relative to the part. The cutter guide, 4, is a hardened steel plug, provided with a 90-degree step with horizontal and vertical guiding surfaces. To avoid wear on the precision surfaces, a 0.120-inch (3.05-mm)-thick feeler gage is laid against the cutter guide when the cutter is adjusted.

5. The design of the complete fixture, as shown in Fig. 19-6, follows almost automatically from the pattern of the previously described components. It is a rectangular box with ribs beneath the bottom, and heavy section uprights on the sides for added rigidity. It lends itself well to casting and does not require much core work. The base locators are...
lifted from the bottom on bosses to facilitate machining and to ensure that they stay clear of chips. The side walls have windows for chip removal. The fixture is bolted to the machine table with four T-bolts, and is aligned with two keys in one T-slot. The closest tolerances on the part are those controlled by the mounting of the milling cutters on the arbor; the remaining tolerances are quite liberal. Therefore, there is no place where the conventional toolmaker’s tolerance of 0.001 inch (0.025 mm) is really needed, and all tolerances on the fixture except those for press fits) are 0.002 inch (0.050 mm) or multiples thereof.

Fig. 19-6. The complete milling fixture.

Fig. 19-7. A slide base for a special machine tool.
Case 12. Design a fixture for the milling of the upper and side surfaces of the part shown in Fig. 19-7.

Detailed drawings of the fixture body and the clamp strap are shown in Figs. 19-8 and 19-9, while the complete fixture is shown in Fig. 19-10.

The part is a gray iron casting, weighing 13 pounds (6 kg). In comparison, the complete fixture, empty, weighs 52 pounds (24 kg). The part is to be used as the base for a small slide in a special machine tool and the surfaces to be milled form the guideways for the slide.

The first operation to be performed on the part is, naturally, the machining of the bottom surface. This operation requires no fixture because the size and shape of the part permit it to be securely clamped in the milling machine vise. And once the bottom is machined, it offers an excellent locating and clamping surface for the subsequent milling operations of the upper surfaces. The outline of the milling cutter assembly for this operation is shown in Fig. 19-7. The design proceeds as follows:

1. With some modifications, the 3-2-1 principle can be applied. The fixture body has a large machined flat surface to receive the machined bottom surface of the part. This is the equivalent of the first three locators. Dimensions and tolerances for the part indicate that a considerable degree of symmetry is required, which includes the two unmachined edges of the base flanges. It is therefore necessary to provide a system of centralizers, acting on the side edges of the flanges. In the present case, this is accomplished by an unconventional design of the clamp straps. Each strap is fork shaped and the fork prongs have downward projecting strips arranged in a V-shape, as seen in Fig. 19-9.

These two Vs act on each two corners of the flanges. To confine the clamp straps to symmetrical positions, and still permit longitudinal movement, each strap has a longitudinal slot with a sliding fit around the clamp stud, and a tail which is guided, also with a sliding fit, in the rest block which is integral with the fixture body. When the two Vs are brought into contact with the four corners of the flanges on the part, the part is confined to the required symmetrical position, but so far, longitudinal motion is still possible. To finally eliminate this motion, one end stop, in the form of a round rest button, is mounted with a press fit in the fixture body. The locating components, thus described, are identified by Fig. 19-10.

2. The part is clamped against the bottom locating surface by the two fork-shaped clamp straps and two 3/4-10 UNC (20 X 2.5 mm) clamp studs with
nuts and spherical washers. Each strap clamps on the part at two points, and is reacted by the rest block under its tail end. The design is such that the stud is located approximately in the center of gravity for the three pressure points, so that the total clamping force is distributed quite evenly on these three points. When in operation, the strap opposite the end stop is moved forward so that the part is brought into contact with the end stop. The force from the milling cutter acts in the same direction. In view of these facts, and the substantial vertical clamping forces exerted by the two straps, no additional longitudinal clamping means are needed. The clamping components are identified by (2) in Fig. 19-10. Note the details of the design of the tail end which permits the strap to tilt and adjust itself to any unevenness in thickness of the flanges without binding of the tail end. Had there been lifting springs under the clamp straps, the operator would have found them convenient. However, the available space is too narrow to allow the installation of such springs.

3. Because of the rigidity of the part, and the uniform support which it receives from the base, there is no need for any intermediate supports.

4. As in Case II, the relative location of the individual milled surfaces is defined by the milling cutter assembly, and the cutter guide has to locate the cutter assembly in the vertical and horizontal directions. The cutter guide, (4), in Fig. 19-10, is a hardened plug of tool steel, mounted with a press fit in an extension of the rest block, at that end of the fixture which is opposite the end stop. The cutter guide has horizontal and vertical locating surfaces, with dimensions that allow the use of a .120-inch (3.05-mm) feeler gage when setting the cutter. In this case, no detailed drawing is provided of the cutter guide, but it is recommended that the reader, as an exercise, make a complete drawing of the cutter guide (including grinding clearances, if needed), and calculate the required tolerances.

5. The design of the fixture body, shown in detail in Fig. 19-8 and identified by (5) in Fig. 19-10, follows almost automatically from the previous discussion. Essentially, it consists of a heavy base with the locating surface for the part, two rest blocks at the ends, and slots for the keys and T-bolts which align and secure it to the machine table. All tolerances, except those for the cutter guide (to be calculated in the recommended exercise), are quite liberal. There are no closed spaces and no chip cleaning problems. The design lends itself well to casting and requires no core work. However, it is equally well suited for welded construction.

Fig. 19-10. The complete milling fixture for the part shown in Fig. 19-7.

Fig. 19-11. A bracket with two bearing bosses.
**Case 13.** Design a fixture for the milling of the base surface of the bracket with two bearing bosses as shown in Fig. 19-11.

Detailed drawings of the fixture body, a V-block locator, and the clamp straps, are shown in Figs. 19-12 and 19-13, while the complete fixture is shown in Fig. 19-14. The part, a gray iron casting, is a bracket with two bearing bosses, and weighs 28 pounds (13 kg). In comparison, the complete fixture, empty, weighs 136 pounds (62 kg).

The part comes unmachined, and it is natural to select the machining of the base surface as the first operation since this provides excellent conditions for fixing the following operations. This choice is not without its problems, since the part has no other flat surfaces on which it could be located and clamped for the first operation. However, the bearing holes are bored out to 1 3/8-inch (35 mm)-diameter, so that the part can be well clamped in the bored holes while it is located and carried on the cylindrical outer surfaces of the bosses. This method of locating assumes that there is no parting plane with its inherent danger of mismatch across the bosses. Depending on the type of milling machine to be used (horizontal or vertical), the milling cutter is either a plain milling cutter with helical teeth, slightly longer than the width of the part, or a face milling cutter with a diameter substantially greater than the width of the part. It is left to the reader, as an exercise, to make a recommendation for the diameter of the face milling cutter. When it comes to the detailed planning of the operation, the direction of the cutter tooth helix, or the teeth in the face milling cutter, together with the rotation of the milling machine spindle, must be such that the side component of
the cutting force is acting against the side stop in the fixture. The design proceeds as follows:

1. The part can be located and carried with the two bosses supported in a double V-block, shown in detail in Fig. 19-13. The 3-2-1 principle is not directly applicable, but the support in the V-block eliminates four degrees of freedom; namely, two in the vertical direction and two in the side direction. At the same time, the bearing axis is centered. The part can still rotate in the V's around this axis, and it can slide longitudinally, thus having two degrees of freedom. These two freedoms are now eliminated by the addition of a side stop and an end stop. The side stop is formed by a 5/8-11 UNC (16 x 2 mm) hexagonal head screw, acting against the side of the base. The screw is locked in position by a jam nut. In this way, the position of the side stop can be adjusted when necessary, as, for example, if a batch of castings should fall outside of dimensional tolerances. Only one side locator is required, as the direction of the bearing axis is already defined by the V's. The end stop is a spherical button. The locating components are identified by Fig. 19-14.

2. The part is clamped down into the V-block by two finger-type clamp straps, shown in detail in Fig. 19-13, and two 3/4-10 UNC (20 x 2.5 mm) clamp studs with nuts and spherical washers. The straps have relief grooves cut across at a distance of 1/2 inch from either end. These details allow the straps to seek the lowest point in the cored holes, as the nuts are tightened, and to adjust themselves to slight dimensional variations in the parts. Lifting springs around the studs are provided to hold the clamps up when in the retracted position, for the convenience of the operator. A 5/8-11 UNC (16 x 2 mm) hexagonal head-clamping screw is provided to clamp the part against the side stop. A 5/8-11 UNC (16 x 2 mm) hand-knob screw is provided to clamp the part against the end stop. Here a hand-operated screw is preferred because it provides more “feel” in clamping, than a hexagonal screw operated with a wrench. In clamping, the part will be laid down into the V's, and manually held against the side and end stops, while the hand-knob screw is applied. Subsequently, the hexagonal head screws are tightened. The clamping components are identified by in Fig. 19-14.
3. The area to be machined is quite wide and a large cutting force is generated, including a substantial side-force component. It is therefore necessary to-supplement the side stop and its mating clamping screw with additional supporting means, equivalent to what is known as “intermediate supports.” They must be hand operated, to be applied with proper “feel” so as not to strain the part out of the position in which it is held by the previously applied clamps. These additional supports comprise two more 5/8-11 UNC (16 x 2 mm) hand-knob screws, in Fig. 19-14, arranged opposite each other.

4. The cutter can be sidewise located by sight, since it is visibly wider than the part. The vertical locating of the cutter is done with a cutter guide consisting of a round rest button, installed with a press fit in one of the four uprights that form part of the fixture body. This particular upright is 1/8 inch (3 mm) higher than the three other uprights and is machined at the top. An allowance of .120 inch (3.05 mm) is provided for the feeler gage used in the setting of the cutter. A recommended exercise, is to calculate the tolerance on the height dimension of the cutter guide, in Fig. 19-14.

5. The design of the fixture body, shown in detail in Fig. 19-12 and identified by in Fig. 19-14, follows almost automatically from the positions of the locating and clamping components. It consists essentially of a heavy base, four uprights, and two upper cross bars, carried by the uprights. The base has the locating surface for the V-block, two rest blocks at the ends, and slots for the keys and T-bolts which align and secure it to the machine table. The four uprights carry the side stop and its clamping screw, and the two additional hand-knob screws. One upright carries the cutter guide. The two uprights in either side are connected by a 5/8-inch (16-mm)-thick wall for added rigidity.

The two uprights at either end are connected by two cross bars which carry, respectively, the end stop and its hand-knob clamp screw.

The fixture body is designed as a casting. It requires some core work, but the core is of a regular and simple geometry and presents no technical or economic problems. However, this fixture body

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Fig. 19-15. a. A milling fixture with manual clamping. b. A line drawing of the same fixture showing workpiece (a cylinder head), milling cutters, locating points, and the gage block.

Courtesy of Cincinnati Milacron Inc.
Typical Milling Fixtures

A typical milling fixture is shown in Fig. 19-15, a and b. The work is a cylinder head, outlined in Fig. 19-15b. It has a previously machined surface and is located by this surface on five blocks, each with two narrow bearing surfaces. The clamps are hand-operated clamps of normal design. Further locating is done by two locating pins fitting into two holes in the block (tooling holes). The cutter setting gage is located on a bracket on one side of the fixture. This fixture is aligned with the milling machine spindle by means of two keys, called “tongue strips,” one at each end, which fit into a T-slot in the milling machine table. The fixture also has two ordinary slots at each end for the clamping bolts. While this is a widely used practice, it may be less practical when a fixture is expected to be used on several milling machines as the spacing of the T-slots may be different on different machines. It is, in this sense, more practical to use end clamps, as shown in Fig. 19-16.

The milling machine vise with modified jaws provides many opportunities for the design of inexpensive milling fixtures. An example is shown in Fig. 19-17. String milling is used extensively, and some examples are shown in Fig. 19-18.

Duplex milling, that is, the milling of two parts in one operation, is also a common and profitable operation.

Contour and Profile Milling Fixtures

Contour or profile milling fixtures are used on profiling or contour milling machines. These fixtures are basically similar to other fixtures; however, a distinctive characteristic of this type of
equipment is that it must provide for a bracket for holding the cam or template which controls the operation of the machine. The fixture must also have setting gages for aligning the fixture and the template bracket with the machine and the tracer spindles, as explained in Chapter 13, Cutter Guides.

Two or three spindles are frequently found in tracer controlled and contour milling machines. They require the corresponding number of identical fixtures for simultaneous milling of several parts. An example is shown in Fig. 19-19. Here, two fixtures are used to hold two cylinder heads. The template bracket is a trunnion-mounted box holding templates for several different operations. This box indexes between operations so that only one template at a time is brought into the active position. In this case, various operations require different angular positions of the part, therefore the fixtures are built as trunnion bases with cradles.

**Adjustable and Movable Milling Fixtures**

Milling fixtures are made adjustable or movable for several reasons. An adjustable fixture designed for the milling of the nonparallel sides of the block shown in Fig. 19-20, is illustrated in Fig. 19-21. Three operations are involved; the parallel sides $A$ are milled by means of the straddle cutters, and the two sides $B$ and $C$ are then milled in two subsequent operations. The three operations are all performed without requiring more than one setting of the work. The block is cut off from bar stock, and drilled and counterbored to receive two fillister-head screws which hold it in place on the machine of which it forms a part. These holes are also utilized for holding the block in position on the fixture.

The milling fixture consists of an upper plate $A$, which is pivoted on stud $B$. This stud is mounted in the cross slide $C$, which operates on base $D$. Plate $A$ is provided with two tapped steel bushings which are a forced fit in holes drilled and counterbored for the purpose. These bushings receive the two screws which secure the work in position on the fixture, with the purpose of preventing the rapid wear of the threads which would take place had they been tapped directly into the cast iron. The fixture is shown set in position for milling the parallel sides $A$, of the work. There are two tapered pins $E$ and $F$, which
are used for locating the work in the required position. For milling the parallel sides of the work, pin $F$ is inserted in hole $N$ to locate the cross slide $C$ in the required position. Similarly, pin $E$ is located in the central hole to locate swivel plate $A$. These pins are merely used to locate the fixture; bolts $G$ and $H$ are provided to secure it in the required position. When the fixture is set for milling the angular side $C$ of the work, pin $E$ is inserted in hole $J$, and pin $F$ in hole $O$. This sets swivel plate $A$ at the required angle and also locates the cross slide $C$ at the required off-center distance to enable the work to be milled by the outer edge of the cutter. After this operation has been completed, swivel plate $A$ is then swung over to enable pin $E$ to enter hole $K$. Similarly, cross slide $C$ is moved so that pin $F$ will enter hole $M$. This brings the work into position to enable the angular side $B$ to be milled by the outer edge of the other cutter on the arbor.

The fixture described above is, in a sense, a primitive indexing fixture. Fully indexing fixtures are used extensively, and dividing heads of various types are available for the control of the indexing function.

Movable milling fixtures are those that move while the cutter passes over the work. Radial fixtures perform a slow rotation around a fixed center or axis, with the result that the cutter generates an arc of a circle. Other movable fixtures, similarly rotating around an axis, are controlled in their motion by a template and are, therefore, capable of generating curved surfaces of any desired shape.

### Locating Pins in Milling Fixtures

Locating pins in milling fixtures can be fixed or retractable. An example of a fixture using fixed locating pins is shown in Fig. 19-22. The part is an aluminum cylinder head, and the fixture is rotating. The cylinder head is supported and located on the large diameter circular locator which centers it on the inside bore. The final accurate location for milling the fins is obtained by means of a diamond-shaped pin which can be seen in the background. This pin engages the locating hole in the joining surface of the cylinder head. In this operation, the cooling fins are milled to the required depth by guiding the cutter with a tracer which follows the contour of the template as the piece rotates with the fixture.

Retractable pins are used where a workpiece must slide into position before the pins can engage. An arrangement of retractable pins is shown in Fig. 19-23, where the locating pins are mounted on the ends of a cross bar. This cross bar, balanced by two springs placed at equal distances from a centrally located eccentric, is moved up and down as the eccentric is operated by a hand lever. The lever is
Fig. 19-22. A rotating fixture for milling cooling fins on a cylinder head with one fixed diamond pin for locating the part.

Located at the front of the machine (Fig. 19-24), which is used for milling various surfaces on a cylinder block held in a fixture (not shown in the picture).

Fig. 19-23. A milling fixture with retractable locating pins.
Fig. 19-24. The same milling fixture shown in position on the milling machine.

The locating pins are normally in the retracted position. When the work is moved into position, the operator rotates the lever to raise the pins, and also moves the cylinder block slightly to ease the engagement of the pins with the locating holes. After the part has been located and clamped, the pins are again retracted. To further facilitate the insertion of the pins and thus reduce the time required for locating the part, both locating pins are made with the diamond-shaped head and both are located with their major axis perpendicular to the surfaces to be milled. This will permit a slight variation in the location of the part in the direction parallel to the surface to be milled. The movement provides ease in positioning of the parts and has no appreciable effect on the accuracy of location of the cylinder block for the milling operation.

**Gear Hobbing Fixtures**

Successful gear hobbing depends equally on the accuracy of the gear blanks and the accuracy and rigidity of the hobbing fixture. A typical hobbing fixture (Fig. 19-25) for a vertical spindle hobbing machine consists of a base bolted to the machine table, a bottom support plate, a mandrel, an upper clamping plate, and a clamping nut. The mandrel extends above the nut with a pilot, which is supported by the supporting arm of the machine. The following points are highly significant: The gear blanks are accurately centered on the mandrel; they are supported and clamped on the largest possible diameter, and before the base is finally clamped to the table, the entire fixture is centered, with respect to the axis of rotation, by a dial indicator. To allow for this centering adjustment, the base must not be solidly centered in the machine table; there must be about 1/8-inch (3-mm)-clearance in the hole in the machine table for the pilot of the base. The height of the base must be sufficient to allow for a clearance of about 1 inch from the work to the cutter at the lower end of the travel. Adequate approach and overtravel must be provided at both ends of the cutter travel, and the fixture designer is cautioned that overtravel, in the case of helical gears, is considerable yet not easily detected from a drawing.

**Fixtures for N/C Milling**

The rapidly expanding use of numerically controlled (N/C) milling machines has focused attention on the need for reduction of all phases of non-cutting time. The loading and unloading time is not a programmed operation, and even with skillfully designed fixtures, is still a burden on the economy of the operation. It can be drastically reduced, however, by dual fixturing; that is, by the use of two identical and interchangeable fixtures, a method which is used quite extensively. While one part, clamped in its fixture, is machined, the other fixture
Direction of hob rotation for conventional and climb hobbing methods of cutting.

is unloaded then reloaded. When space permits, both fixtures are mounted on the machine table. With very large fixtures, it is necessary to unload and reload one fixture on the floor, while the part in the other fixture is machined, and then the fixtures are exchanged. An example of this type of operation is seen in Fig. 19-26.

![Fig. 19-25. Gear hobbing fixtures. a. Plain. b. With reversible bottom plate.](image_url)

![Fig. 19-26. The use of interchangeable fixtures with an N/C milling machine.](image_url)
Design Studies III—Miscellaneous Fixtures

Lathe Fixtures in General

Lathe fixtures are, for the most part, used on vertical and horizontal turret lathes and high-speed production lathes. In the past they have not been used on engine lathes to the extent they deserve; a skillfully designed yet inexpensive fixture can well convert an engine lathe into a production machine. However, fixtures are now being used extensively on engine lathes equipped with N/C controls. Workpieces are centered, located, and clamped in lathe fixtures in essentially the same way as in conventional workholding devices used with lathes: accordingly, the fixtures can be classified as chucks with special jaws and inserts, collet-type fixtures, face-plate fixtures, pot-type fixtures, mandrels and arbors, and special fixtures.

Since cutting forces are unbalanced, lathe fixtures are always designed with large metal thicknesses and strong clamps. For the main fixture body, gray cast iron is preferred to mild steel for damping vibrations; however, for high-speed operation, weight and strength considerations may require the use of steel (in welded or built-up construction) rather than gray cast iron. With the exception of mandrels, which are supported on the tailstock, lathe fixtures are cantilevered. They must be designed with as little overhang as possible and, for operator safety, projecting screws and pins must either be avoided or shielded. At medium and high spindle speeds, centrifugal forces become significant and must be taken into account. They affect clamping forces and will cause vibration if the fixture with the part does not run true. Many fixtures for second operation cuts, taken at high spindle speeds, are therefore designed so that they can be adjusted with respect to the spindle axis. Examples of adjustable fixtures are shown in Figs. 6-43 and 6-44. Irregularly shaped workpieces must be counterweighted. Medium- and large-size lathe fixtures for roughing work in a first operation are usually designed on the three-point principle. Larger fixtures for second operations and for thin-walled parts are designed with equalizing clamping devices (the floating principle). Examples of applications are shown in Figs. 9-11 and 12-5.

Chuck Fixtures

The cheapest type of lathe fixture is the standard lathe chuck (3- or 4-jaws), with special jaws or inserts, machined to fit the part. Aluminum, gray cast iron, and steel inserts are commercially available in dimensions to match standard chucks. As a general rule, steel jaws with serrated or hardened surfaces are used for gripping on rough parts, while soft jaws with smooth surfaces are used on machined parts to prevent scratching or marring. These rules also apply to power operated chucks, found on many turret lathes and production lathes. Designs of more elaborate chuck fixtures are also shown in Figs. 9-11 and 12-5.

Since centrifugal force tends to draw the jaws away from the part (except when the part is clamped from the inside), and the standard chuck design is inadequate at spindle speeds used in high-speed production lathes, solutions must be sought; i.e., using a power operated chuck where a positive clamping force is constantly maintained on the jaws. Another solution is to make use of the centrifugal force for clamping. An example is shown in Fig. 20-1. The machine is a two-spindle production lathe and the part to be turned, in this case, is a cast-iron brake drum. The rear chuck is shown loaded; the front chuck, empty. The boss of the part is clamped by means of the hook clamp seen near the center of the chuck. In addition, the part is clamped on its periphery by 10 jaws. Each jaw can rotate around a fulcrum and at the back it is provided with an inertia block of greater mass than the forward
clamping end of the jaw. The result is that when the spindle rotates, the inertia block is forced outward by centrifugal force, and, in turn, forces the clamping end of the jaw against the work. Consequently, the clamping pressure increases with the spindle speed. Uniform pressure applied around the rim of the brake drum does not distort the part and eliminates chatter when the drum is machined.

The principle of the collet chuck was shown in Fig. 9-3, with design rules and data presented in the accompanying text. The total travel of the jaws or fingers is determined by elastic deformation within the collet and is, therefore, very short.

The maximum diameter range of the spring collet is 0.002 to 0.003 inch (0.05 to 0.08 mm) and it is intended for use on bar stock and bar-shaped parts. However, the term "collet chuck," as applied to fixtures in general, is now used in a wider sense. Any chuck that actuates its jaws by an axial motion relative to a conical surface is a collet-type chuck.

The two chucks for the centering of gear wheels shown in Figs. 9-22 and 9-23 are collet chucks. They are clamping chucks, but that is a matter of application rather than of design. The collet principle can be applied to chucks of any dimension and proportion, and for internal as well as for external clamping. A collet-type lathe chuck for internal clamping is shown in Fig. 20-2. It is operated by means of the central draw bar. The actuating cone is a solid conical plug in the center of the chuck. The sliding pads are kept in contact with the cone by springs and each carries two pins which constitute the actual jaws. As the cone is drawn in, the jaws are forced out. Dotted lines indicate buttons that can serve as axial end stops.

The collet chuck can be combined with other locating and clamping components. One example is shown in Fig. 20-3. The part is a pinion, integral with a long shaft, too long to be held in any type of chuck. For the same reason, the collet chuck provides excellent centering of the part and is well adapted to precision work. An application example is shown in Fig. 9-24.

### Face Plate Fixtures

The face plate fixture is the natural type of fixture for machining large diameter parts on the vertical turret lathe. It is highly versatile and any type and combination of locators and clamps can be built up on it. A typical example is shown in Fig. 20-4.

The work A is a cast-iron bracket which has previously been machined along the face D and has had the tongued portion cut approximately central with the cored hole at Y. Four holes have also been drilled at J. Two sizes of these brackets are made in lots of ten or twelve. An angle plate B, is tongued on the underside F, to fit one of the table T-slots and is held down by screws (not shown). The distance E, for the two sizes of brackets, is determined by placing a stud G in the center hole of the table and locating angle plate B from it. The bracket is placed in position on the angle plate so that tongue H fits into the groove, and bolts J are passed through the holes in the bracket and tightened by nuts K. Clearance is provided in the bolt holes.

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Fig. 20-3. A collet-type lathe chuck for external clamping of a pinion with a long shaft (inverted tailstock).

Fig. 20-4. A face plate fixture for noncircular bracket-type parts.
to allow the finished edge of the bracket to rest on pins C. Two special jaws Q are fixed in position on the table but may be adjusted radially, when necessary, to bring them into the correct position for the other size of bracket. The jaws are provided with set-screws O, which are adjusted to support the overhanging end of the bracket, after which they are locked by the check nuts at P. The jaws are keyed at S to the sub-jaws of the table; and clamps N are used on the unfinished portion of the bracket. The clamps are tightened by nuts at R, so that the surface to be machined is clear of interferences. The boring bar L is used to bore the hole, and the side head tool M faces the pad. This fixture illustrates that lathe fixtures can be made to accommodate workpieces of completely noncircular shapes and of more than one size.

On engine lathes and horizontal turret lathes it is often more convenient to have a special face plate integral with the fixture and to mount the complete assembly on the spindle nose, as was shown in Figs. 6-43 and 6-44. A fixture for the facing and internal machining of an aircraft fuel-pump housing on a numerically controlled lathe is shown in Fig. 20-5a and b. The fixture base is mounted on the face plate of the lathe. The part is located and clamped in V's, which are carried on a projecting plate and braced against the fixture base.

The pot type fixture is used for parts of large diameter and considerable axial length, which do not require external machining. A typical example is shown in Fig. 20-6. The work A is a large casting which, because of its dimensions (diameter, length, and wall thickness), could not be adequately supported and driven by a conventional chuck. The variations in dimensions which may be expected in large castings require a liberal clearance between the work and the fixture. In the present case, a clearance of one inch was provided. The pot fixture H is centered on the lathe table by the locator J and bolted down in the T-slots with bolts O. The part is located by its cylindrical surface against three screws B, C, and D, and is first positioned against these three points by the central clamping screw D. After the positioning is accomplished, the clamping is completed by application of the upper and lower screws D, and all three screws are then moderately and evenly tightened. In the vertical direction, the part is supported on three points F, G, and H (G is fixed, while F and H are adjustable). Windows P in the fixture wall allow for access to the adjustable supports. The part is clamped down by means of U-clamps L and nuts and washers M on studs K. This fixture presents several interesting details: the screwthreads E on clamping screws D and the adjustable points F are shielded against chips; screws D are provided with "snubbers," heavy rubber pads on their tips which prevent distortion of the casting wall because of excessive pressure and also assist in the damping of vibration (chatter) during machining.
Mandrel and Arbor Type Fixtures

Mandrel and arbor type fixtures will center, locate, and grip the work from the inside and are normally used for parts that already have a machined internal surface. The mandrel is supported at both ends as a simple beam; the arbor is carried at one end only, as a cantilever beam. In the simplest case, the work is centered with a sliding fit, located endwise against a shoulder, and clamped with a nut and washer. Commercially available mandrels and arbors hold the work between a fixed and a movable cone. Another type holds the work on an expanding sleeve. This design is shown in Fig. 20-7. The sleeve is cylindrical on the outside and fits inside on a taper on the mandrel body. When forced axially up onto the taper, the sleeve expands and clamps the part. The

sleeve is slit, with slots running alternately from either end. The large nut on the left-hand end of the mandrel serves the double purpose of locating the part endwise and of releasing the pressure by forcing the sleeve down on the taper. These mandrels are made with tapers varying, in general, from 1:50 up to 1:15; and occasionally up to 1:6. A 1:50 taper represents the limit for what conveniently can be operated with a release nut. This type of mandrel fixture is perhaps the most satisfactory of all. It locates and centers well and the sleeve remains practically cylindrical as it expands, exerting a uniform pressure on the part. Sleeves of different sizes can be used on the same mandrel body. For satisfactory results, the number of slots must be related to the sleeve diameter as follows:

<table>
<thead>
<tr>
<th>Diameter</th>
<th>Number of Slots</th>
</tr>
</thead>
<tbody>
<tr>
<td>inches</td>
<td>mm</td>
</tr>
<tr>
<td>up to 2</td>
<td>up to 50</td>
</tr>
<tr>
<td>up to 3</td>
<td>up to 75</td>
</tr>
<tr>
<td>up to 5</td>
<td>up to 125</td>
</tr>
<tr>
<td>up to 7</td>
<td>up to 175</td>
</tr>
</tbody>
</table>

A recent development, now commercially available, is a mandrel, or arbor, with one or several hydraulically expanded sleeves. An example is shown in Fig. 20-8, where two sleeves are used to clamp a part with a stepped hole. Hydraulic pressure is generated by means of a piston forced against the hydraulic fluid by an actuating screw, which is manually rotated with a socket-head screw wrench. For high-production work, the piston can be actuated by a push rod mounted centrally in the lathe spindle. Mandrels and arbors can be dimensioned by the same rules as boring bars. Details concerning the design and dimensioning of the expanding sleeve and the data for the hydraulic system are presented in Chapter 21, Universal and Automatic Fixtures.

Another recent development is the mandrel with an expanding sleeve made of an elastomer. It was developed by AEC-NASA and its description is published in a Tech Brief. The purpose is to support

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3An AEC-NASA Tech Brief from Cutting Tool Engineering, April 1969, p. 17.
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Fig. 20-8. An arbor with two hydraulically expanded clamping sleeves.

Fig. 20-9. A mandrel with an expanding elastomer sleeve.

rough, hollow castings during grinding and turning. The device is shown in Fig. 20-9. The elastomer sleeve is supported on a mandrel threaded at one end. The part is slipped over the sleeve, a heavy washer and nut are put in place, and when the nut is tightened, the sleeve expands on its diameter while maintaining constant volume, and centers, supports, and clamps the part. The elastomer must be nearly completely enclosed so that it cannot escape from the pressure. Under these conditions it behaves much like a rigid body after it has filled the cavity inside the part. It locates and centers the part on its average inside diameter or contour. The elastomer can be cast in aluminum molds. The diameter of the sleeve is made 0.010 to 0.015 inch (0.25 to 0.38 mm) smaller than the cast hole and the length approximately 1/8 inch (3 mm) larger. Elastomers suitable for this application are silicone RTV, PVC (polyvinyl chloride) plastisols, vulcanized rubber, and polyurethane. Generally, elastomers can be reused.

Workpieces with unmachined interior surfaces, such as cored holes, can be mounted on mandrels or arbors with movable internal jaws.

Parts with an internal thread can be mounted on a mandrel. The simplest case is that shown in Fig. 20-10. The screw thread serves to center, clamp, and drive the part, and alignment is provided by the flat shoulder. A plain arrangement such as this

Fig. 20-10. A simple type of arbor for holding threaded work.

Fig. 20-11. An improved type of arbor for threaded work, with a separate clamping and release nut.

causes the part to bind in the thread after machining. An improvement, shown in Fig. 20-11, uses a large clamping nut with a coarse left-hand screw thread. The nut provides the means for end-locating the shoulder and for aligning the part. After machining, the nut is backed off a fraction of a turn and the binding pressure on the part's screw thread is eliminated.

Miscellaneous Fixtures

Lathe operations on parts that are unusual because of their shape or dimensions offer many opportunities for the successful application of fixtures. At times the fixtures are complicated and expensive, at other times they are simple, if not primitive, yet they are always efficient with respect to the saving of time and the improvement of quality. A systematic classification will not be attempted here but a few examples will indicate the possibilities.

A crankshaft has one geometrical axis defined by the main bearing journals, and one or more additional axes, each defined by a wrist pin or a set of wrist pins. Each of these axes requires a turning operation, but only the axis through the main journals terminates in solid steel with surfaces that can be center drilled. The other axes primarily run through air.

Two different sets of fixtures for the turning of crankshafts are shown in Fig. 20-12, a and b. In example a, each end of the crankshaft is provided with a block that carries the two sets of centers required for the two wrist pins. The center block at the tailstock end can also carry a counterweight, as indicated by the dotted lines. Other fixture components needed are struts for taking the axial thrust between the tailstock and the spindle center, and spacers between two parallel arms. The fixtures shown in example b are comprised of two brackets; the one to the left is clamped to the spindle, and the one to the right is provided with a center drilled plug that is supported by the center in the tailstock.

The fixture shown in Fig. 20-13 carries, supports, and guides the free end of a tank for a space exploration rocket engine. The visible portion is the so-called “Y-joint,” which is seen being machined in preparation for welding to the end closure and the adjacent tank. The fixture is provided with adjustable bearing blocks all the way around to allow the tank to rotate accurately, just as an axle rotates in an ordinary steady rest.

Boring Fixtures

Boring is an operation whereby an existing hole is machined to a larger size. Boring fixtures differ from drill jigs in that they are to be used with boring bars. Drill jigs, however, can also be designed for combined drilling and boring operations. While the twist drill is supported and guided by the hole that is being drilled, the cutter in a boring bar receives its support entirely from the boring bar itself, producing holes of greater accuracy with respect to diameter, roundness, position, and alignment.

Boring bars differ in design and can be made in a rather wide range of sizes. Some boring bars, called “line” boring bars, are supported at both ends. Others, called “stub” boring bars, are supported only at one end by the spindle of the machine. Line boring bars are used to bore long, deep holes and holes of very large diameter. Stub boring bars are used more frequently than line boring bars and must be used for boring blind holes.

Boring operations are performed on many types of conventional or numerically controlled machine tools. Conventional or N/C lathes, milling machines, drilling machines, vertical boring mills or vertical turret lathes, and horizontal boring mills (HBM) are used extensively to perform boring operations. The horizontal boring mill is a remarkable machine, practically a one-man machine shop. With different accessories, this machine can perform almost any

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4 Ibid.
5 Ibid., vol. II.
conventional machining operation. Its spindle assembly is designed with the precision and rigidity required for work with boring bars, and it has the necessary power for the boring of large holes. By its design, and with experienced and careful operation, it produces holes that are exactly parallel to the surface of its table. Boring, in the toolroom, is performed on jig borers to obtain very precise hole locations on tools, dies, jigs, and fixtures. Jig borers sometimes are used to obtain precise hole location tolerances on machine parts where the number of parts is relatively small. Boring fixtures occasionally are used on jig borers to speed-up the location of the part on the machine or where the part cannot be conveniently held by any other means.

High-production boring is often done on special machine tools. Some are special purpose machines such as those found in the automotive industry. Often the part is automatically moved to and from the boring station by machines called "transfer machines." A different class of boring machine is comprised of the highly automated, high-speed production boring machines, known as "Bore-Matics,"® or equivalent trade names, which operate at high cutting speeds and fine feeds (borizing). These machine tools may be adapted to do a variety of jobs; however, they are usually set up to do a particular job for a prolonged period of time. Boring fixtures are always used with these production machines.

**Design of Boring Fixtures**

**Case 14.** Design a boring fixture for a box-shaped part with a bearing hole at each end.

The part is schematically represented by the rectangular outline A in Fig. 20-14. The distance between the holes requires a line boring bar; consequently, this type of fixture is called a "line boring fixture." The usual five design steps apply here again, with some modifications and simplifications, that are characteristic for most boring fixtures.

1. Usually, a part to be bored is already machined on one or several major flat surfaces. Here, the base surface of the part is machined, and is now located and supported on a matching flat surface on the jig base. In the side direction it is located by means of pins, stops, or with keys or dowel pins, if such components are provided for in the part, for use in the final assembly. Endwise, it is located to provide sufficient clearance at each end (dimensions B).
2. The part may be clamped by means of clamp straps, but often the part is already provided with bolt holes which can then be used for clamping it to the fixture.

3. Intermediate supports are usually not needed, since the part is well supported on its base.

4. The cutter guides serve as bushings for the boring bar. The fixture is provided with fixed bushings mounted in brackets K, and the boring bar has slip bushings of sufficient outer diameter to allow the bar with its cutting tools to be inserted and withdrawn endwise.

5. The fixture body consists essentially of a base with brackets K, for bushings. If needed, the brackets may be provided with stiffening ribs C. Side lugs D are machined exactly parallel to the boring axis and serve for the fixture alignment on the machine table. Bosses E are for measuring and checking the clearance B, to the end surfaces G, of the work. B must be large enough to allow for variations in the size of the raw part, for the escape of chips, and, if necessary, for the insertion of facing cutters or the mounting of shell reamers on the bar. The fixture is bolted to the machine table by means of three lugs H. The use of three lugs provides for statically determinate support and minimizes the possibility of elastic deformation (springing) in the fixture. To reduce weight, the fixture base is cored out from below, and the bushing bosses are tapered. The fixture is a one-piece casting. Gray cast iron is preferred over weldments because of its excellent damping properties.

Many modifications are possible from this general outline. The fixture can be built with a base plate and detachable brackets carrying the bushings. This is used when the dimensions are so large that the use of a single casting is impractical. It also provides for lengthwise adjustment of the brackets to accommodate workpieces of different lengths, for the insertion of an intermediate bracket in the case of very long workpieces, or for the use of different brackets on the same fixture base. For multiple boring operations the fixture can be provided with a multiple spindle gearbox which rotates and feeds the boring bars. It should be remembered that mating gear wheels rotate in opposite directions, and some cutters may have to be designed for left-hand cutting. A boring fixture can have bushings for holes in more than one direction and is then placed on a revolving or indexing table.

A somewhat different technique can be used when the boring operations affect only a small area of a much larger machine part or assembly. In that case, the boring fixtures are small and are carried by the larger unit. This method is frequently used in the manufacture of machine tools. One example is given in Fig. 20-15, which shows a machine tool bed with some accessories. The hole B which signifies the hole for the main bearing in the headstock E, is shown bored with a boring bar supported in two fixtures C and D which are located on the inverted V's of the machine bed. This ensures alignment between bed and spindle. Next it is assumed that hole B should be aligned with holes F and G in two already existing carriages or brackets, and in this case these same holes serve as guides for the boring bar, if necessary by the use of liner bushings. Finally, the front elevation shows how hole J is bored in the carriage and apron I by using the three bearings K, L, and M as guides for the boring bar. A tapered hole is bored by means of a boring bar mounted at the required angle in a bushing in such a way that it is fed through the bushing, rotating the bushing by means of a key and keyseat. The
The workpiece is a casting for a lathe headstock, is located on inverted V's 1 and 2, and clamps against end stop 4 by means of screw 3. When located, it is finally clamped down by screws 5. The fixture has boring bar bushings 6 for the spindle, 7 for the back gear shaft, and 8 for the rocker shaft. The fixture body is designed like a box-type drill jig, and all operations are performed in a radial drill.

**Design of Boring Bars**

The strength and rigidity of a boring bar is determined by its diameter $D$ and free length $L$. The diameter should always be taken as large as possible, allowing a chip clearance between the bar and the raw hole of not less than the machining allowance in the hole. The free length $L$ can be taken as follows:

- for line boring bars
  \[ L \leq 10\, D \]

- for stub boring bars
  \[ L \leq 6\, D \]

The length of a bushing should never be less than $D$, and for smaller bars, the bushing length may be taken up to 2 times $D$. This rule is for boring bars to be made of steel. Solid cemented carbide boring bars may be dimensioned by comparison with previously used boring bars. The critical property is the transverse rupture strength which can vary greatly (up to 200 to 300 percent) in any given carbide grade.

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Fig. 20-16. A boring bar with bushing for boring a tapered hole.

Fig. 20-17. The boring fixture for the operation shown in Fig. 20-16.

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Fixtures for Production Boring Machines

High-speed production boring machines use stub boring bars and other cutting tools (for facing, counterboring, etc., and sometimes for external turning) that do not require bushings in the fixture. The fixtures are designed for quick precision clamping of the part. As an example, Fig. 20-18 shows the fixture for machining a die-cast-aluminum engine front cover. With four spindles, the machine bores and faces pockets and shaft bores to a finish of 120 to 125 RMS at a rate of 57 pieces per hour. The part is positioned from its rear side and is clamped by eight clamps, all air-operated for fast opening and closing. Shown in Fig. 20-19 is an indexing fixture that holds two compressor crankcases for simultaneous machining (finish boring and chamfering of both ends of the cylinder bores).

Grinding Fixtures

The grinding operation is characterized by small cutting forces, high accuracy, and, in general, a large flow of coolant. Grinding fixtures must allow for the unrestricted access of coolant to the work, as well as free drainage of the used coolant, with no sumps or pockets where sludge can accumulate. Magnetic face plates and chucks, supplemented by electrostatic and vacuum operated devices for nonmagnetic workpieces, are widely used as standard work-holding devices for surface grinding, but are not usually part of the fixture. The magnetic face plate can be used, however, as a fixture base that offers a fast and convenient method of removing and replacing the fixture. Typical examples of grinding fixtures of the chuck type were shown in Chapter 9, Figs. 9-22 through 9-24. Clamping is mainly done with hand or finger operated mechanical elements, designed for light duty only, and is not apt to cause distortion of the part, nor restrain its natural thermal expansion.

The structural design of grinding fixtures is very similar to that of other fixtures, except that they are lighter. A fixture that structurally lends itself...
change the setup to the position for the second set of bores, the angle plate is placed on its back, thereby tilting it 90 degrees.

**Angle Plate Fixtures**

The internal grinder with planetary spindle motion is used for internal grinding operations on cylinder blocks and other similar type work. A fixture for this kind of operation is shown in Fig. 20-21. The grinder spindle is horizontal and the fixture is formed as an angle plate with stiffening end walls. The cylinder block is clamped by its base surface on the angle plate which has openings for the entrance of the grinder spindle. This fixture also has two diamonds for the truing of the grinding wheel, a feature that is characteristic of many grinding fixtures. There are two reasons why a grinding fixture cannot use cutter guides of the usual type for setting the grinding wheel; one is that the setting would soon be lost because of wheel wear; the other is that an ordinary cutter guide would soon be ground down and destroyed by accidental contact with the grinding wheel. Therefore, instead of cutter guides, grinding fixtures are equipped with diamonds for the truing of the grinding wheel. The diamonds are preset for the final work dimension and the grinding wheel is run back past the diamond and trued before it starts the finishing cut. The angle plate is also a characteristic feature of many grinding fixtures. An angle plate has sufficient rigidity to withstand the weak grinding pressures, and is frequently used to raise the work to a convenient position above the grinder table.

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Automotive Grinding Fixtures

Of the grinding fixtures used in the automotive industry, two are particularly significant because of their special design features which may well be utilized in other applications. They are the fixtures for the cylindrical grinding of crankpins and for the contour grinding of camshafts.

The principle of a crankpin grinding fixture is shown schematically, and somewhat simplified, in Fig. 20-22. The work spindle of the grinder has a large face plate $A$, which carries a locator $B$, $C$ for the main bearing journal at the end of the crankshaft, and an index plate $D$ with an indexing mechanism, here shown simplified as an index pin $E$. The crankshaft is nested in bearings $B$, and one crankpin is brought into contact with a retractable locator $F$, which brings one crankpin center line to coincide with the work spindle center line. These operations define the starting position for the crankshaft and when that position is reached the pressure feet $C$ close; the end of the crankshaft is clamped to the index plate $D$; and the crankpin locator $F$ retracts. The grinding wheel $G$ advances and grinds the crankpin. After grinding is completed, the index pin $E$ is withdrawn, the index plate with the crankshaft is indexed to the next position, the grinding wheel is aligned with the next crankpin, and grinding can continue.

On most crankpin grinders available, the fixture is integral with the machine. The operations: loading and positioning the crankshaft, clamping, indexing, start and stop, feeding, retracting, and repositioning the grinding wheel, are all mechanically controlled. An example of such a mechanized and automated crankpin grinder is shown in Fig. 20-23.

Cam grinding is a copying operation. The cam contours are copied from a set of master cams; contact between a master cam and the master roller is maintained mechanically by means of a spring or a hydraulic cylinder with piston. The principle of cam grinding is shown schematically in Fig. 20-24.

![Fig. 20-22. Schematic view of a fixture for the grinding of four crankpins.](image)

![Fig. 20-23. A crankpin grinding fixture with work positioner.](image)
master cam set \( H \) and a chuck \( I \), with a live center for locating, supporting, and clamping one end of the camshaft \( K \). The opposite end of the camshaft is carried by a tailstock, also known as the "footstock." The free part of the camshaft is supported against the grinding pressure by a "work rest."

As the work spindle with the master cam and the camshaft rotates, the cradle is oscillated around its shaft \( D \), and with the grinding wheel \( L \) in operating position, the master cam contour is transferred to the cam that is being ground. The master roller is moved lengthwise from one disc to the other, on the master cam, when the grinding wheel is moved into position for the grinding of the next cam on the camshaft. A camshaft grinding operation is seen in Fig. 20-25. The illustration shows the cradle with its inverted-V guide, the footstock, the work rest on the middle of the camshaft, and the driving chuck. The master cam mechanism is covered within the housing at the left in the photograph.

**Fig. 20-24.** Schematic view of a fixture for cam grinding.

**Fig. 20-25.** A camshaft grinding fixture.

### Fixtures for Planing and Related Operations

Planing, shaping, slotting, and broaching have many features in common which are reflected in the design of their fixtures. They are straight-line operations which, with the exception of broaching, are performed by a single-point tool and a reciprocating motion. In planing, the work moves, while in the other operations, the cutter moves. In planing and shaping, the motion is horizontal; in slotting, the motion is vertical. Broaching is done horizontally (pull broaching) and vertically (surface broaching).

Because of inertia forces at stroke reversal, the cutting speeds are rather moderate. Vibration is not a problem, but fixtures must be designed to withstand the impact from the tool each time it enters the work. Structurally, these fixtures are related to milling fixtures. They require substantial metal dimensions, positive and strong end stops to withstand the impact, solid bolting down upon the machine table, and strong clamps. The reciprocating operations require ample end clearance relative to the cut surface at each end of the stroke. The run-out distance at the exit end is relatively small (about 1/2 inch [13 mm] for shaping, and 1 to 2 inches [25 to 50 mm] for planing) to allow the tool to clear the work and be lifted before the start of the return stroke, and at least 1 inch (25 mm) for slotting (vertical) to allow for the accumulation of chips. The run-in distance at the entry end must be long enough to allow for deceleration, stroke reversal, re-acceleration, and some time for the side-wise feed motion to be completed. For planers this will require several inches, depending on the size, type, and condition of the machine, but for the other machines, 1 to 2 inches (25 to 50 mm) will suffice. Tool setting blocks are often used. For the machining (planing and shaping) of composite shapes, such as a dovetail or the inverted V's on a lathe bed, the setting block is formed as a template of the contour. (See Fig. 13-6.)

### Planing Fixtures

While planing is the natural operation for long parts, it can also be economically applied to smaller parts when they are clamped in a gang fixture. An example is shown in Fig. 20-26. Inexpensive shaping fixtures can be made by the installation of special inserts on the jaws of the standard machine vise. In the vertical machining operations (slotting, surface broaching) the cutter moves in a vertical path (see Fig. 20-27) and the main cutting force \( F_C \).
is directed downwards. Since the slotting machine table is horizontal, this cutting force assists in stabilizing the work. However, the thrust force $F_T$ is horizontal and quite significant. It acts with its full value right from the beginning of the cut and generates an overturning moment on the work. It is therefore essential that slotting fixtures for tall workpieces are designed with a wide base.

Similar considerations apply to fixtures for surface broaching. The thrust force $F_j$ on the average, is equal to $1/2$ of $F_C$ and may increase to the same value as $F_C$ as the cutting edges become dulled by wear.

**Broaching Fixtures**

Broaching is characterized by a short machining cycle and, for economical production, also requires a short loading and unloading time. Broaching fixtures for production work are, therefore, almost exclusively provided with automatic clamping devices. They can take many forms, even for almost identical workpieces. As an example of the variety in the design of broaching fixtures, Fig. 20-28a and b show two different designs of a fixture for the broaching of the large end of an automotive connecting rod. In each design, the closing and clamping motion of the clamping arm is controlled by a contoured slot in the arm, while, in a, the clamp is double and is actuated by a crank arm on a rotating shaft. In b, there are two individual clamps, each one actuated by a power cylinder.

The indexing principle is also frequently used for the purpose of reducing loading and unloading time. An application is shown in Fig. 20-29. The machine is not a duplex, but carries two identical broaches on the ram and machines two parts with each stroke. The parts are manually loaded into nests on the periphery of the indexing plate and are indexed into the machining station where they are hydraulically clamped. After machining, they are indexed to the
Fig. 20-28 a and b. Two different designs of broaching fixtures for an automotive connecting rod.

Fig. 20-29. An indexing broaching fixture with two parts in each station.

Fig. 20-30. A broaching fixture for pull broaching.

Fig. 20-31. A broaching fixture for pull broaching.

horizontal) machine table, and an internal part (a plug) to guide the broach relative to the work and to provide support against the thrust force. A thin-walled or otherwise flexible workpiece requires a substantial work support (Figs. 20-30 and 20-31). A broach with an asymmetrical cut, such as the keyseat broach, is supported by a plug, and on a horizontal machine the plug has an extension (a horn, see Fig. 20-32) to prevent sagging of the free end of the broach. The thrust force and its reaction are now internal forces within the part. By means of parallel or tapered inserts, the keyseat depth can be varied, or a plug can accommodate broaches of different heights. For tapered cuts, the face of the work support and the bottom of the slot in the plug are machined to the appropriate angle so that the work is tilted with respect to the broach.

Fixtures for Internal Broaching

A fixture for internal broaching consists essentially of a work support, known as an adapter, that transmits the main cutting force to the (vertical or horizontal) machine table, and an internal part (a plug) to guide the broach relative to the work and to provide support against the thrust force. A thin-walled or otherwise flexible workpiece requires a substantial work support (Figs. 20-30 and 20-31). A broach with an asymmetrical cut, such as the keyseat broach, is supported by a plug, and on a horizontal machine the plug has an extension (a horn, see Fig. 20-32) to prevent sagging of the free end of the broach. The thrust force and its reaction are now internal forces within the part. By means of parallel or tapered inserts, the keyseat depth can be varied, or a plug can accommodate broaches of different heights. For tapered cuts, the face of the work support and the bottom of the slot in the plug are machined to the appropriate angle so that the work is tilted with respect to the broach.
dimensions and in the integration of different fixtures into one master fixture system. The following description of a typical aircraft fixture system refers to the tooling for the aft wing section of a delta wing of a medium-size bomber. The section is 612 inches wide, 261 inches long, and 21 inches deep in the landing gear wheel well area. Each assembly consists of skin panels and spars, bulkheads, and box sections which form the understructure. The assembly working area in the plant is occupied by a large steel structure, shown in Fig. 20-33, which supports the individual fixtures, provides work platforms in three levels, and contains services for electricity and compressed air, with overhead tracks for electric hoists. For each of the wing sections, a vertical assembly fixture carrying pin locators, stops, and clamps is mounted on the main structure. Stops and locators, including locators for the fuel lines, are installed by means of tooling gages, which simulate the items to be installed. With the tooling gages in position, skin panel tooling samples are located and fabricated to minimum gap clearances at plane intersection joints. Hole locations in the tooling samples are checked against the tooling gages and corrected as necessary. The setup for a typical operation is shown in Fig. 20-34. By means of the locators in the fixture, the understructure is assembled, checked with gages, and riveted.

The predrilled skin panels are located to the bulkheads and spars. Holes are transferred to the understructure and step drilled to size. Attachment holes in the skin are counterbored and countersunk, as required, attachment holes in the understructure are tapped, skin panels are removed, and the understructure is deburred and cleaned. Master tooling gages simulating the main landing gear are used to establish the hole patterns for attaching the landing gear to the integral fittings of the corresponding bulkheads. Master tooling gages simulating the fittings for the elevon hinge, are used to control the installation of the locators for these fittings.

Design of Arc Welding Fixtures

The type of assembly tooling most widely used throughout the metalworking industry is tooling for arc welding. In welding shop terminology they are called jigs when they are stationary, and fixtures when they are movable. Their purpose is to locate and hold the parts in correct relative position for joining, to reduce distortion, and to orient the part so that each weld can be laid in the most convenient position: i.e., downhand and horizontal. For this purpose, the fixture usually is carried by a positioner.
with which the part can be raised, lowered, tilted, and rotated. Positioners can be commercially available machines, usually operated with a worm wheel drive to permit a wide range of positions, however, they operate rather slowly. An example is shown in Fig. 20-35. The fixture can be designed to incorporate its own positioner, preferably designed as an indexing positioner, with which each operating position is secured by a locking device that enters a hole or a notch in the index plate. The system's center of gravity should be placed in the axis of rotation, if necessary by the addition of counterweights. The balancing of the fixture is facilitated by the use of light alloys for the moving parts. Even large fixtures can be so well balanced that they can be operated manually and with one hand. Positioning is fast, and operating positions are accurately defined and safely maintained.

Distortion is caused by thermal expansion of parts during welding, and by subsequent shrinkage of deposited weld material. Plain thermal distortion is transient; it disappears as the material cools down, and is harmless. To allow for thermal distortion, the parts may be firmly clamped at one place (anchored), and allowed to slide against the friction under the clamps in directions away from the anchor point. Shrinkage distortion is of a different nature. Since the weld material is deposited from one side at a time, the initial shrinkage is essentially asymmetrical and tends to misalign the parts. It is calculated that the distortion is 1 degree per pass. Minimum lateral shrinkage is obtained by welding with large electrodes in as few passes as possible. The distortion is counteracted by the use of clamps, fixed locators, and stops in such places that they prevent or substantially minimize the anticipated distortion, and also by dimensioning the fixture body with adequate rigidity and strength. The active stresses to be encountered equal the yield stress of the welded material at the elevated temperature that prevails at the beginning of the cooling period after completed solidification.
A fixture is designed around or inside the completed workpiece, and must allow the finished piece to "get out" again. For this purpose, internal fixtures may be collapsible and external fixtures may be split or have a large hinged or otherwise detachable door. Rams, bumpers, or other types of ejectors may be added for the removal of binding workpieces.

Fig. 20-34. Locating and fitting of skin panel tooling samples for an aft wing section.

Fig. 20-35. An arc-welding fixture mounted on a boom type positioner.
The parts to be welded are fitted with a space or gap of from 1/32 to 1/16 inch (0.8 to 1.6 mm) between edges. Backing bars (of copper, aluminum, or stainless steel) are placed behind the gaps to act as a heat sink to protect against overheating, burning, or buckling, and to prevent blowout of molten weld material. Commercial copper suffices in most cases. Where severe wear is anticipated, Class II copper is preferred. This is a copper alloy with a chromium content, widely used for resistance welding electrodes. Stainless steel is used where preheating or postheating is required as it resists oxidation. To prevent contamination, some exotic materials, notably titanium, are not permitted to contact other materials. In such cases the backing bar is provided with a groove, behind and somewhat wider than the gap between the plates. The groove is purged with an inert gas under sufficient pressure to carry the penetration, which then forms a bead without contacting the backing bar. For large parts, two fixtures may be used. The first fixture is the "tacking" fixture and is for tack welding only; the second fixture, known as the "holding" fixture, is for the completion of the welding.

A different technique is the use of subassembly fixtures, followed by the main assembly fixture. During subassembly welding, the parts are allowed to distort as needed, but the pieces are cut with overlength, and free ends are trimmed back and fitted together when installed in the main fixture. Figure 20-36 shows a fixture of this category, used for the final welding of subassemblies for the engine frame of a Titan missile.

The flow of electric current must be controlled. The fixture or the parts must be grounded to provide a return path for the current. Besides, the current carries a magnetic field, here called the "magnetic flux." It can affect the direction of the arc and disturb the welding (arc blow). The flux in the part and in the fixture is controlled by the path of the return current, and it is essential that the magnetic flux has no opportunity to cross or concentrate near the path of the arc. A rule-of-thumb recommends that in the vicinity of the joint, steel members should be one inch (25 mm) below the part or two inches (50 mm) above the path of the arc. An effective means is to ground the part at the starting end of a long weld. If clamps and fixture are made

![Fig. 20-36. Main assembly welding fixture for the engine frame of a Titan missile.](image)
of nonmagnetic material, which will not provide a path for the magnetic flux, the magnetic field will be weak.

Welding fixtures are of simple and inexpensive design with liberal tolerances and little or no machining. As a general rule, no castings are used, but all the standard structural shapes, plates, angles, channels, and I-beams, can be employed. Preference is given to closed sections, such as circular, rectangular, and square tubes, because they combine high torsional strength and rigidity with low weight. Toggle clamps are used extensively; they are inexpensive and allow fast operation.

The screw threads on clamping bolts must be shielded against weld splatter.

Welding fixtures within some size limitations can be made from castable epoxy and phenolic tooling resins. Since they are cast to shape, they frequently permit the work to be located in the fixture by simple nesting without the need of additional locators and clamps. The plastic materials are light, have satisfactory dimensional stability, minimum deterioration, and do not actively support combustion.

Case 15. Design a fixture for the arc welding of the structure shown in Fig. 20-37.

The structure is a rectangular frame with two cross bars. The frame consists of 6 by 2 1/2-inch (150 by 65-mm) angles, bent from 1/2-inch (13-mm) -thick plate. The cross bars are 6 by 7/16-inch (150 by 11-mm) flats. The length of the structure is 6 feet (1.8 m), equal to the height of a person. The weight is 325 pounds (147 kg); in comparison, the fixture, as shown in Fig. 20-38, weighs, empty, 428 pounds (194 kg). With the exception of the four short welds across the narrow angle flanges, all welds are 90 degree corner welds and they are parallel. Therefore, the fixture is designed to be rotated primarily around one axis which will successively bring each of these welds into a convenient position. As for the short welds on the narrow angle flanges, if the fixture is mounted on a commercial positioner, there may be at least one additional axis of rotation whereby the fixture can be laid down flat, or the part can simply be removed from the fixture and placed on a bench for these welds. Thus, it is a matter of available equipment. In the design shown in full lines, the fixture has a circular base with 6 holes for clamping it on the positioner table. If corner welds only are required to be done in the fixture, it can be provided with a shaft to be mounted horizontally in its own bearing bracket. The circular plate is retained and serves as an index plate for indexing the fixture to four positions 90 degrees apart. This alternate solution is indicated by chain-dotted lines.

The fixture is a rectangular frame with diagonals, provided with standing lugs that function as locators for the parts, and with push-pull toggle clamps for securing the four frame sides in position. The locators for the cross bars are welded on the diagonals in pairs, with sufficient space between for receiving and locating the parts. In this way, their position is fully defined, and no clamping is needed. Each clamp acts in a point, halfway up on the locator.

Fig. 20-37. An arc welded rectangular structure with cross bars.
so that the correct orientation of the frame members is secured when the parts are installed. The end members and the cross bars are positively positioned between the side members; the side members can slide longitudinally and be visually lined up, relative to the end members, at the corners. Thermal distortion is not significant, as the welds are small in relation to the mass of metal, and are located far apart. The effect of the heat is expansion, and the result is essentially a bending of the frame members away from the locators over the free spans between, and outside of, the clamped points. This distortion is slight, transient, and not harmful. The shrinkage distortion which tends to pull the parts together is effectively resisted by the wide and massive locators. All locators are rectangular blocks of substantial thickness. They could have been designed as T-section brackets, saving some steel, but would cost significantly more in cutting, fitting, and welding. The fixture body consists of square tubing, 4 by 4 inches (100 X 100 mm), 11 gage (.120 inch or 3.05 mm). With the weight of the part and the fixture, the required strength and rigidity might well be obtained with smaller dimensions, or with round tubing. However, the use of this size of square tubing provides large, flat areas for supporting the parts and permits assembly without the use of gusset plates. The width of the tubes provides areas for the mounting and welding of the locators. The two longitudinal tubes are made one inch longer than the part, to provide backing for the welding across the narrow angle flanges. The total weight of the tubes is 70 pounds (32 kg), in other words, only a small fraction of the total weight of the fixture. The pads for the clamps are flat plates. The width may seem somewhat excessive, relative to the base of the clamp, but again, the extra width provides the rigidity which otherwise would have required a bracket with a rib; a more expensive design. In the design of welding fixtures, it is often possible to economize by trading off labor cost for cutting and welding, against some additional material.

All dimensions shown in Fig. 20-38 are REF (untoleranced) dimensions. As an exercise, calculate the dimensions with their proper tolerances so that the fixture will produce the part with the tolerances shown in Fig. 20-37.

Some positioners, particularly those for automatic welding, have developed into full-fledged machine tools, usually of the lathe type. A welding lathe carrying an internal fixture is shown in Fig. 20-39. One segmented and collapsed backing ring is seen in the photograph. The part, a thin-walled cylinder, has runner rings on the outside, supported in the four large frames, which function as steady rests.

There is no physical upper limit to the size of welding equipment. Probably the largest existing welding positioner is shown in Fig. 20-40. It weighs 200,000 pounds (91,000 kg) and is designed for use at the U.S. Naval Shipyard on Mare Island. It rotates the work at speeds ranging from 0.052 RPM (19.2 minutes for one revolution) to 0.0052 RPM and in four minutes can tilt the table 60 degrees from the horizontal. The table diameter is 33 ft 0 in (10 m); the height, measured to the table in the horizontal position, is 20 ft 10 in (6.35 m), and its work capacity is 150,000 pounds (68,000 kg).
Universal and Automatic Fixtures

Definition of “Universal”

The term “universal fixtures” covers two different types of equipment. The first type consists of a drill jig body with a quick-acting clamping and locking mechanism, and which can be provided with interchangeable drill bushing top plates and sub-bases (adapters) to support the work. The second type comprises sets of building elements which can be temporarily assembled to a fixture and dismantled after use. Both types are available from commercial sources, but they can also be designed and built “in-house” to advantage. Examples of simplified designs for this purpose are included in the following sections.

UNIVERSAL DRILL JIGS

Custom-made Jigs

The merits of the machine tool vise (and other vises) for use as fixture bases have been described at length in other chapters. Examples of how various types of visses, not only machine tool visses, can be converted to universal drill jigs were shown in Figs. 18-35 through 18-37. The vise drill jig shown in Fig. 8-36 includes, in addition, two V-blocks and demonstrates a principle used in universal drill jigs of a simple type for cylindrical parts. It differs from the more commonly used type by having the V-block installed with horizontal V’s. The usual type consists of a V-block with a vertical V, one or more brackets, each with a drill bushing centered in the axis of symmetry of the V, a clamp, and an end stop. It is used for drilling holes along a diameter of cylindrical parts within the full range of diameters that can be accommodated in the V-block.

Pump Jigs

The most common type of universal drill jig is the pump jig, so named because it is operated by a pump movement of the operating handle. A pump jig, designed and built “in-house” is shown in Fig. 18-42. In the clamping position the top plate A with the drill bushing is held against the work by spring pressure. When the handle D is lifted, it lifts the top plate and releases the work.

A commercial drill jig with three posts is shown in Fig. 21-1a. The construction of universal drill jigs is quite simple. The outer style and some details may vary, but the principle remains the same. The top plate is secured to either one, two, or three vertical posts. The posts are raised and lowered through a lever arm, with the top plate maintaining a horizontal position at all times. The length of travel (the clamping range) of the top plate is quite limited, normally about 25 percent, or even less, of the maximum opening height. For most workpieces it is therefore necessary to provide a sub-base, known as an adapter, to lift them up so that the top of the work comes within the clamping range, as shown in Fig. 21-2. Locators, attached to the top plates or to the adapters, are also used. A few rules can be formulated for the design of the adapters and locators. Assuming that the part has one machined and one unmachined side, horizontal alignment is established from the machined surface. When the holes must be drilled from the machined surface, this side must be up, and to ensure full alignment with the bottom side of the top plate, the adapter must be made much smaller than the surface of the part. This condition is shown in the illustration. When the holes must be, or can be, drilled from the unmachined side, the machined side is down and the adapter is made large enough to align and support it on its entire width.

Internal location is preferred to external location. A concentric locator is attached to that member from which horizontal alignment is established. Preferably, a concentric internal locator is attached to the top plate, but a concentric external locator is attached to or made integral with the adapter.

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Locators are designed with a short locating surface to prevent jamming, and with conical or otherwise tapered lead surfaces (pilots). Locators on the top plate have long lead surfaces so that the operator can see that they catch the part. It may be necessary to machine a clearance space in the adapter if the lead surface is longer than the height of the part. Locators on the adapter have short leads because they are visible to the operator when the jig is open. A long lead would also require that the part be lifted higher when inserted and removed. In either case, there must be space enough over or under the locator in the open position to bring the part in and out.

Parts having fully concentric configurations (contours and holes) do not require radial location, i.e., components that prevent rotation away from the correct position. Most noncircular parts require radial locators, usually hardened steel blocks, fastened to the top plate or the adapter.

There is generally a locking mechanism connected with the operating mechanism. This device maintains the clamping pressure and prevents the work from shaking loose when it is drilled. One widely used type of operating and locking mechanism is shown in Fig. 21-1a, b, and c. As seen in the sectioned view a, the jig contains a helical gear pinion and a mating rack which is integral with the post. The pinion shaft (see b) carries the operating handle and at one end an integral brake cone. A counter cone is fitted on the opposite end, and mating conical seats are machined into the base of the jig. Operating the handle closes the jig, and as clamping pressure builds up, an axial thrust is developed which locks the brake cone into its seat. The cone at the opposite end is for locking the top plate in the fully open position. View c shows a unit consisting of a pinion and rack set mounted in a bearing bushing.

Fig. 21-2. General arrangement of the work, top plate, and adapter in a pump jig.

Fig. 21-1. a. Sectional view of a pump jig with three posts, showing the rack and pinion movement. b. A pinion shaft with integral braking cone. c. A rack and pinion locking unit with bearing bushing and operating handle.

Courtesy of Jergens Inc.

ing which contains the conical seats. Such units are commercially available for installation in custom designed jigs.

Other types of pump jigs employ braking devices based on the principle of the overrunning clutch, as in a bicycle wheel, or a pair of cam operated brake shoes. Locking units of these types are also commercially available.

The jig shown in Fig. 21-3 has several refinements. The top bushing plate is interchangeable and adjustable, the adapter for the work is a V-block, and an adjustable end stop is provided for locating the work. A drill jig with an air-operating clamping mechanism is shown in Fig. 21-4. Some drill jigs have a fixed top plate, and the adapter is mounted on
a post that can be raised for clamping the work. Some drill jigs have the clamping area located be-
tween the posts while still others provide the feature of improved access to the clamping area by allowing the top plate to swing 180 degrees out of the way in

the horizontal plane or tilt 45 degrees in the vertical plane. Some jigs have the rear side of the body precision-machined square with the bottom surface. In this way the jig becomes a tumbling jig; it can be laid down on its back to permit the drilling of holes in two directions, 90 degrees apart.

Cast iron top plates, fitted to the posts, but without bushing holes, are supplied by jig manufacturers. Blanks for top plates can also be economically produced by torch cutting them out of steel plate and drilling and reaming the post holes with a simple drill jig. A commercial punch holder, i.e., the upper half of a postless die set, makes a satisfactory and inexpensive adapter blank.

Advantages

Speed of operation is the greatest advantage of the universal drill jig. It is so significant that the use of this type of jig as a permanent component of a single-purpose tool can be economically justified in highly repetitive work.

The jig operation is fast since it is operated by a single sweep of the lever handle, which eliminates the need for loose tools and clamping parts. This manual operation is always the same, regardless of the part configuration, and a line of dissimilar parts is drilled as if they were all alike. The rate setting can be done without individual time studies, by deter-
mining and recording the handling times, once and for all, and calculating drilling time from speed and feed. The jig is loaded and unloaded in the upright and open position, and does not have to be turned over as do most other drill jigs. When one hole only is to be drilled, the jig can be secured to the drill press table; therefore, the drill will enter the bushing practically without touching, which results in pro-
longing drill and bushing life. Top plates and ad-
apters can often be so designed that chip cleaning is greatly simplified if not completely eliminated.

The cost of a top plate with adapter is generally less than the cost of a complete single-purpose drill jig, and top plates and adapters are interchangeable so that the main body and operating mechanism can be used for a variety of jobs. A top plate is usually more expensive than an adapter, although the ma-
terial costs are a minor consideration. The greatest expense item is the precision boring of the post and bushing holes. A top plate may be equipped with bushings for more than one hole configuration, and an existing top plate may be modified by the addi-
tion of more bushings. Different parts with the same hole configuration can be accommodated by changing adapters and locators. If a top plate is made with integral locators, then it can be turned over and the other side used. In such cases it may be necessary to use headless drill bushings.
Chips and Coolant Considerations

When coolant is needed it is directed onto the top plate. Commercially available cast top plates are formed as trays and provide a reservoir from which the coolant flows down along the drills. For use with a flat top plate, a ring large enough to encompass all the drills used is cut from 1/2-inch (13-mm) steel plate, and is laid on the top plate to hold the coolant. Chips are swept off by simply sliding the ring over the plate.

UNIVERSAL FIXTURES

Commercial Universal Fixtures

A simple, yet quite versatile and efficient fixture is shown in Fig. 21-5. It is in essence a glorified V-block. With the clamping screws shown, it can hold parts of any configuration within its own dimensional limitations. It can be rotated (like a tumble jig) 45 degrees and 90 degrees in its own vertical plane, and rotated 90 degrees to either side. A swivel base is available by which it can be rotated at an arbitrary angle. It can be used in any machine tool, including the lathe, where it is clamped on the face plate.

A different and more representative type of fixture is shown in Fig. 21-6. The principal component is the base plate. Edge strips are bolted on, so that a lying V-block is formed, and a part is clamped in this V by means of clamping screws of the same type as those shown in Fig. 21-5. It is used here for precision drilling. The drill bushing, mounted in a large boss, is located from the sides of the V by means of gage blocks.

The backbone of every universal fixture is the sub-base; the various systems differ in the types and number of components. The elements in general are of steel, hardened and ground to tolerances.
of the order of 0.0003 inch (0.008 mm) on the significant dimensions. The basic elements are rectangular blocks with T-slots, called stop elements (manufacturer's terminology), thrust elements, and adapter blocks with bolt holes for buttressing the stop elements, fixed and adjustable height elements, angular elements with fixed angles of 30, 45, and 60 degrees, adjustable angular elements (including sine bars), special elements for the attachment of locating pins (also adjustable by means of an eccentric), V-blocks, jack screws, clamps, holders for drill bushings, and bearings for boring bars. Straight and angle straps are provided for the joining of two sub-bases, and for bracing stop elements, for example, for forming rigid corners. Sub-bases, with T-slots of standard dimensions and spacing, are made of nickel cast steel and are available in square, rectangular, and round shapes. Typical elements are shown in Figs. 21-7, 21-8, and 21-9, and a completed milling fixture is shown in Fig. 21-10.

Such a fixture is not designed in advance but is built up with dummy blocks made of a castable plastic material around an actual production part,
or a replica of a part. When the dummy fixture is completed, it is photographed in detail. The photographs are used in the toolroom for assembling the actual fixture, and provide a permanent record for filing.

Another system uses holes instead of T-slots for the assembly. The holes are alternately straight precision holes and tapped holes and are closely spaced in a modular pattern. A drill jig built with components from this set is shown in Fig. 21-11. The jig was built in two hours and is used for drilling and reaming holes with 0.003-inch (0.08 mm) tolerance on the center distances.

**Custom-made Universal Fixtures**

An experienced fixture designer in cooperation with a good toolmaker can make any desired type of universal jig or fixture. An example of a universal jig construction is shown in Fig. 21-12. It is known as a toolmaker's universal drill jig and consists of a heavy plate A, containing adjustable locating rods B with locking screws, and a boring for interchange-

![Fig. 21-12. The toolmaker's universal drill jig.](image)

able drill bushings C, for different hole sizes. The application of this jig for drilling and reaming the holes in a large die block is shown in Fig. 21-13. Parallels are clamped to the edges of the die block, and the jig is positioned against the parallels with the locating rods; measurements are taken with micrometers and gage blocks. When the bushing C, is correctly positioned, the jig is clamped to the die block and the hole is spotted, drilled, and reamed. The procedure is repeated for each hole.

A drill press can be converted to a makeshift borer by installing a compound table with slides at right angles, on the drill press table. An upright with a bracket carrying a liner bushing for the insertion of different size slip bushings is installed with the bushing axis in line with the drill spindle. The slides are positioned from fixed stops by means of micrometer gages, gage blocks, or gage bars.

In developing and building a universal fixture set, the first task is to design the sub-base. Any base plate with parallel T-slots or mounting holes will serve the purpose, but a design with partly diagonal
ribs and T-slots, such as that shown in Fig. 21-14, has advantages over the conventional type. The T-slot pattern is more versatile, and diagonal ribs provide additional rigidity against torsion.

Some angle plates with single (Fig. 21-15)\(^2\) and multiple T-slots and some smaller and larger tooling blocks (Fig. 21-16) are added to the base. There is no rule that forbids the use of T-slots and mounting holes within the same set. Each system has its advantages and selection is made according to what is needed. Finally, an assortment of bushings, clamps, bolts, and sundry items is selected from fixture component catalogs, and the universal fixture set is ready for its first assignment.

\[\text{Fig. 21-15. Angle plate with a single T-slot for use in a universal fixture.}\]

\[\text{Fig. 21-16. A large tooling block for use in a universal fixture.}\]

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**AUTOMATIC FIXTURES**

**Definitions and Principles**

Automatic fixtures are those in which the part is clamped and unclamped by the use of a power medium, usually compressed air (pneumatic fixtures) or oil under high pressure (hydraulic fixtures). These devices are used for five purposes:

1. To apply a greater and more consistent clamping force than is possible by manual operation
2. To reduce operating time and operator fatigue
3. To operate the fixture by remote control (including foot operation)
4. To clamp simultaneously and uniformly in multiple fixtures
5. To be able to incorporate the fixture into an automated program (transfer machines, conveyorized production lines, or numerical control [N/C] machine tools).

The actuating member is always a cylinder with a piston or a plunger (referred to in the following as the power cylinder). The actuating force is applied directly or indirectly; direct actuation means that the force from the power cylinder acts directly on the part or on a clamp that is in contact with the part; indirect actuation means that the force from the power cylinder acts on the clamping element through a kinematic chain, which can be a linkage system or a combination of cams and links.

**Common Features and Advantages of Pneumatic and Hydraulic Fixtures**

There is no general preference for selecting one system over the other. It is not even possible to predict relative costs without making comparative estimates. For the selection of a system, the following guidelines can be applied:

Hydraulic fixtures utilize significantly higher working pressures than pneumatic fixtures. Hydraulic fixtures are preferred, therefore, where large clamping forces and short strokes are required and where available design space is limited. Permissible flow velocities in conduits are 15 feet per second (4.6 m per sec) in pressure lines and 4 feet per second (1.2 m per sec) in other lines. The maximum piston speed is 2 feet per second (0.6 m per sec). Hydraulic fixtures operate satisfactorily under average cycling conditions and at normal temperature levels. The oil temperature must not exceed 140°F (60°C). At higher temperatures oils lose viscosity, oxidize, and, in the long run, break down. Mating parts within the equipment that expand differently, may bind or
leak. Hydraulic cylinders are self-lubricating; air cylinders are not.

Pneumatic fixtures are used where medium clamping forces with almost any length of stroke are required, and where ample design space is available. Permissible flow velocities in conduits are significantly higher for air than for oil; pistons in air cylinders operate at speeds ranging from 1/4 inch per second (6 mm per second) to 10 feet per second (3 m per sec) with 2 to 3 feet per second (0.6 to 0.9 m per sec) as a commonly used average; this means that the clamping operation is practically instantaneous. Pneumatic fixtures operate satisfactorily under conditions of high cycling and elevated temperatures.

The use of a power medium (air or oil) under constant pressure permits close control of the clamping pressure which, first, ensures that the part is sufficiently gripped and, second, reduces the risk of distortion or even breakage of the part. The initial cost of pneumatic and hydraulic clamps is higher than the cost of manually operated clamps; they also incur some operating expenses (compressed air, power for the hydraulic pump), but these are insignificant. Air is cheap. A representative figure for the cost of compressed air in a factory is $0.12 to $0.15 per 1000 cubic feet ($0.42 to $0.55 per 100 m³), and it takes many piston strokes for a small cylinder to consume one cubic foot.

The dominating, if not decisive factor, in favor of using automatic fixtures is the saving in labor costs. The saving is about 80 percent on manual clamping operations of up to 1/2 minute duration, and 85 percent on longer operations. In addition, operator fatigue is virtually eliminated.

Each of the two systems can be manually or automatically actuated by electrical controls (solenoïd-operated).

Normally, the power medium is applied to the clamping operation. The release of the clamp and the return to the open position can be accomplished by application of the power medium in the opposite direction, by a return spring, or by a combination of the two. Where several clamps are used in the same fixture, they can be timed to function simultaneously or in a predetermined sequence. The force \( F \) (pounds) exerted by a power cylinder of diameter \( D_p \) (inches) with a piston rod diameter \( D_R \) (inches), and an operating pressure \( P \) pounds per square inch is calculated by

\[
F = 0.7854 \left( D_p^2 - D_R^2 \right) \times P \times (0.85 \ldots 0.90)
\]

For that side of the piston where there is no piston rod, \( D_R = 0 \). The factor (0.85 . . . 0.90) is the mechanical efficiency.

In metric units \( F \) is in newtons, \( D_p \) and \( D_R \) are in millimeters, and \( P \) is in newtons per square millimeter.

**Pneumatic Fixtures**

Pneumatic fixtures are operated with air from the compressed air supply system in the plant. The operating pressure is nominally 100 pounds per square inch (0.69N per mm²). It is usually assumed to range between 80 and 100 pounds per square inch (0.55 to 0.69N per mm²), but may well drop to 40 to 50 pounds per square inch (0.28 to 0.34N per mm²) at points a great distance from the source, or in cases where the compressor or the distribution lines are overloaded. Air cylinders are available with diameters up to 14 inches (350 mm) and lengths up to 3 to 4 feet (0.9 to 1.2 m). The larger sizes are not for clamping purposes, but are used for moving the part into or out of the fixture, for rotating the part, or for moving the fixture from one station to another, etc.

To ensure constant output from the power cylinder and constant clamping pressure in the fixture, the input pressure must be maintained constant and independent of pressure fluctuation in the air supply. This is done by means of a pressure reducing valve in the supply line to the fixture.

With constant power output, the power cylinder is essentially an elastic system. It maintains its own force as the clamping force on the clamped part and it holds the clamped part in a position of stable equilibrium as long as the clamping force is superior to any opposing force. However, if the opposing force temporarily or permanently equals the clamping force, the equilibrium is no longer stable, and if the opposing force, even only temporarily, exceeds the clamping force, the clamping element is pushed back and the part is likely to be thrown or pulled out.

For the stability of the operation, there must be provided some means by which release of the clamp is prevented if the air supply is cut off or the supply pressure falls below the pressure at which the reducing valve can maintain constant operating pressure. One method is to insert a pressure switch in the supply line which will stop the machine if the pressure drops below a safe limit or the supply is cut off. A different and frequently used method is to transmit the force from the power cylinder to the clamps.
through a mechanical device, a kinematic chain, which is self-locking when it is in the clamping position. Wedges, cams, and toggle joints are used. Once fully activated, they hold the clamps engaged, even if the air pressure vanishes. The introduction of a kinematic chain for the transmission of the force eliminates the elasticity in the system and has the additional advantage that it is now possible to increase the applied clamping force; in mechanical language this means to introduce a mechanical advantage greater than one. For a given required clamping force this permits the use of a smaller power cylinder or the use of a lower operating pressure. A self-locking device normally has a mechanical advantage that is significantly greater than one.

Air Cylinders

Rotating power cylinders for the actuation of various types of chucks are commercial items used on many semiautomatic and automatic lathes. Non-rotating air cylinders are used on commercially available vises. These devices are workholders, not fixtures.

A universal drill jig equipped with an air cylinder is shown in Fig. 21-4. Many commercial clamps are built with an attached or integral power cylinder, mostly for air operation, some of them capable of air operation and oil operation in the same cylinder (dual pressure clamps).

Air cylinders are available from many sources and with a variety of rod end and rear end accessories as shown in Fig. 21-17, for attachment to the fixture and the clamp. A power cylinder can perform the following functions; push, pull, raise, and lower. Cylinders are single-acting and double-acting. They can be supplied with single-end and double-end piston rods, protruding from one end only, or from both ends of the cylinder. For cylinders with single-end piston rods, where possible, clamping with pressure on the piston-rod side of the piston, and releasing with pressure on the full piston area are recommended. The additional release force may be needed to overcome any jamming or sticking in the clamps. When dimensioning air cylinders a length that is two times the net calculated length of travel plus the length of the piston is recommended to compensate in advance for future changes in work dimensions and for other unforeseen changes.

An air-operated clamping device with integral power cylinder is shown in Fig. 21-18. It illustrates most of the previously described principles. The
actuating element is the large horizontal plunger which carries the piston at its right end and a sloping double-acting cam at its middle portion. This cam engages a small vertical plunger which actuates the clamp through spherical equalizing washers. The operation is controlled by an air valve. For clamping, air is admitted to the small area on the left side of the piston, and moves the large plunger to the right. The sloping cam pushes the small plunger down and clamps the part. The inclination angle of the sloping cam surfaces equals the friction angle for dry surfaces so that the clamp is locked in position even without the help of cylinder pressure. In order to release the clamp, air is admitted to the full piston area on the right side of the piston, which moves the large piston to the left. The initial movement simultaneously releases the clamping pressure and the frictional forces, and further movement to the left causes the upper sloping cam surfaces to engage the vertical piston and lift it from the work.

Hydraulic Fixtures

The acceptable working pressure for a hydraulic fixture is determined by the fixture, the conduits, and the hydraulic power source. The design pressure for hydraulic fixtures is, in a sense, arbitrary, because a fixture can theoretically be designed for any desired pressure level. However, higher pressures require not only larger material dimensions, but also tighter fits, closer tolerances, and smoother (precision lapped) surfaces to ensure against leakage. As explained in Chapter 11, the practical upper limit for the working pressure is 15,000 pounds per square inch (103N per mm²) when a plastic (PVC) is used as the pressure medium, and 10,000 pounds per square inch (69N per mm²) when oil is used. Many hydraulic fixtures and fixture components are designed for significantly lower pressures. The dual pressure clamps referred to in the previous section are designed for operation with air pressures of 100 to 250 pounds per square inch (0.7 to 1.7N per mm²) or oil pressures up to 500 pounds per square inch (3.4N per mm²).

The maximum working pressure for standard hydraulic tubing and fittings is about 6000 pounds per square inch (41N per mm²) for stationary pressure lines and about 3500 pounds per square inch (24N per mm²) for flexible tubing. Special types and qualities are available for higher pressures.

Hydraulic fixtures can be powered from the machine tool, if it has a hydraulic drive, from a separate hydraulic pump, and from a pneumatic-hydraulic booster (pressure intensifier). A machine tool with a hydraulic drive is a convenient and inexpensive power source for the fixture. The operating pressures used in machine tools are low (1000 pounds per square inch [7N per mm²] or less) compared to the previously quoted values, and the fixture designer must always carefully check the available pressure. Separate and individual hydraulic pumps are available with almost any desired pressure, but the cost is frequently too high for consideration. The preferred power source, which is now widely used for fixtures, is the booster. It is simple, small, inexpensive, and versatile and is commercially available.

The booster consists of two coaxial cylinders with different diameters and a common piston. A low-pressure medium is applied to the large piston area and a high pressure is developed on the medium in the small diameter cylinder. The low-pressure medium is primarily air from the compressed air line (at around 100 pounds per square inch [0.7N per mm²]) or medium pressure oil, for example, supplied from the hydraulic system in the machine tool. A booster is characterized by the area ratio, the maximum output pressure, and the volume of high-pressure medium supplied per stroke. Representative ratios are 7:1, 15:1, 28:1, and 30:1. Output pressures from commercial units are 1500, 3000, and 7000 pounds per square inch (10, 20, and 50N per mm²). They supply from 1 to 14 cubic inches (16 to 230 cm³) of high-pressure oil per stroke. Some models are provided with an adjustable relief valve and provide infinitely variable pressures of from 1500 to 3000 pounds per square inch (10 to 20N per mm²). Boosters are small and since they

![Fig. 21-19. A fixture with a hydraulic clamp and with the air/hydraulic booster mounted on the machine tool.](Proprietary to Tomco, Inc.)
require only an air connection, they can be placed close to the fixture. It is often found convenient to mount the booster on the machine tool as shown in Fig. 21-19.

Power cylinders are designed for and rated at maximum working pressures of 3000, 5000, and 10,000 pounds per square inch (20, 35, and 70N per mm²) and are commercially available with up to approximately 8000 pounds (36 kN) maximum capacity. Matching holding brackets and complete clamping sets are also available. The most recent design trend is characterized by small dimensions and low profiles. The smallest hydraulic power cylinders are threaded on the outside with thread dimensions down to 1/2-13 UNC (see Fig. 21-20) and provide 150 pounds (0.7 kN) of force at 3000 pounds per square inch (21N per mm²) rating with 1/8 inch (3 mm) plunger travel. A typical low profile clamping unit is shown in Fig. 21-21. When power cylinders are incorporated in individual fixtures, utilizing only 75 percent of the rated maximum stroke to provide a reserve for workpiece variations, etc., is recommended.

An entirely different principle for hydraulic clamping consists of using a thin-walled sleeve expanded by hydraulic pressure into the part that is being clamped. An application to a lathe fixture is shown in Fig. 21-22. The fixture is completely self-contained and works only with the pressure medium (oil or plastic) that is confined in the cavities inside the sleeve. Pressure is applied from the rear end of the lathe spindle by means of the actuating rod located in the axis of the fixture. The principle is the same as that used in the fixture shown in Fig. 11-16. The sleeve expands very uniformly, except near the ends, and the pressure on the part is also very evenly distributed. The unit pressure is relatively low, and the part is not distorted. Accuracy on the clamping surface can be maintained within a tolerance of 0.0002 to 0.0004 inch (0.005 to 0.010 mm) TIR,

depending on the expansion. With a design stress of 60,000 pounds per square inch (400N per mm²) in the sleeve, the maximum expansion is 0.002 times the sleeve diameter. This type of fixture is built in sizes from 1/2-inch (13-mm) diameter minimum and can be used on the inside and outside of cylindrical and slightly tapered surfaces (up to a 6 degree taper angle), on stepped diameter cylinders, and for the clamping of several shorter rings on or in one fixture, as indicated in the illustration. It may be noted that when larger expansion is needed, the sleeve can be made of nylon with steel inserts.

Fig. 21-21. A drill jig with two low-profile swinging hydraulic clamps.

Fig. 21-22. An arbor-type lathe fixture with hydraulically expanded sleeve.

FIXTURES FOR TRANSFER MACHINES AND N/C MACHINE TOOLS

Fixtures for N/C Machining

Because N/C machining is characterized by a high hourly burden (overhead rate), several times higher
than that used for conventional machine tools, a reduction in non-cutting time is of first importance. One way to reduce the non-cutting time is by improved fixturing.

The principle of N/C machining has drastically affected the fixture design. A common type of N/C machine, the N/C machining center, performs milling, drilling, tapping, boring, and reaming in one setting of the part and reduces the number of fixtures from two or three, to one. The need for individual cutter guides, except for the starting position, is eliminated because all subsequent positioning of the cutters relative to the work is covered in the programming for the operation. Fixtures for N/C machining are thereby greatly simplified, in fact, they are reduced to locating and clamping devices for the workpiece.

Most N/C machines have what is called a “floating zero” which allows the programmed starting point to be adjusted to the actual starting point on the part. This point can be defined by a single tool setting block on the fixture or by a point selected on the part itself. This initial operation is not always necessary. Several types of N/C machines work with preset tools, and when the fixture is accurately located relative to the machine, it is also positioned relative to the cutters. Edge locators with a close fit in the slots in the machine table are provided for this purpose. Horizontal and vertical universal fixture bases are also available for most machines. An error in the mounting of a fixture endangers the operation, because the cutter may collide with the fixture body. Machines with more than one slide motion have interference zones for the protection of vital parts, as for example, an index table. Only one slide at a time is permitted in an interference zone. To fully utilize both slide motions on a part held in a fixture, it is sometimes necessary to lift the fixture above the interference zone by mounting it on a raiser block of sufficient height.

N/C fixtures must be strong and rigid to ensure correct part tolerances and have provision for quick loading and unloading of the workpiece. Weight considerations are unimportant when the fixtures are indexed from position to position and are moved on and off the machine by mechanical means.

One result of the simplified design is that there is easier access for the part to the interior of the fixture, and the handling of the part is correspondingly simplified and facilitated.

A simple and easily fabricated universal fixture base, suitable for short-run production, consists of a plate with key seats, T-slots, precision holes, and tapped holes in a regular pattern. Key seats and T-slots are spaced at 6-inch (150-mm) intervals, and holes are located with 3-inch (75-mm) center distances. This pattern permits side and end locators (strips and buttons) to be installed for any conceivable configuration of the outline of the part. It permits easy handling of the part and it provides the possibility for locating all necessary clamping devices. For fast operation, the clamps are hydraulically operated and all power cylinders are fed from the same source of high-pressure oil. In this way, all clamps are actuated simultaneously and with the same pressure, yet still independently so that they automatically equalize for any variations in part dimensions. The source of high-pressure oil can be a booster, but since the total oil volume is relatively small, it is more practical to use a hand pump. When the oil volume required is small enough, a screw pump will suffice. The uniformity in the pressure distribution eliminates or minimizes distortion of the part. The time saving obtained by the simultaneous application of all clamps is repeated after machining, when the clamps are released and automatically retracted, likewise, simultaneously. With automatic control of the hydraulic system, the operation of the clamps can be included in the program and closely coordinated with the machining cycle.

![Proprietary to Vler Engineering Corp.](image)

Fig. 21-23. A typical N/C fixture setup with dual fixtures and low profile swinging hydraulic clamps.

The most drastic reduction in loading time is realized by the use of dual fixtures. With smaller parts, two fixtures are mounted on the machine table; one fixture is unloaded and loaded while the part in the other fixture is being machined. With large parts and long machining cycles, two sets of fixtures and base plates are used. One set is in the machine, while the other set is being unloaded and loaded in the toolroom or on the floor. With the use of edge locators on the table, the exchange of
the base and fixture sets takes little time, and maximum machine "on-time" is achieved.

A typical N/C setup with dual fixtures and hydraulic clamps is shown in Fig. 21-23. The clamp straps lift and swing out automatically as the oil pressure is released.

This clamp satisfies the modern requirement of a low profile. In N/C machining, the machine follows, without human supervision or interference, the path through which the cutter is programmed and returns to the starting point after a completed cut. To prevent collision and damage it is therefore important that the air space above the part is free of obstructions.

**Fixtures for Transfer Machines**

The highest level of development of automatic fixtures is found in the fixtures that are used on or incorporated into conveyorized production lines and transfer machines as used in the automotive and other mass production industries. The design principle for the fixture depends on its mode of operation. There are two modes of operation, the traveling fixture and the stationary fixture.

The traveling fixture is either moved on a conveyor, is an integral part of the conveyor, or is pushed on a track by means of reciprocating transfer bars with fingers that push the fixture during the forward stroke of the bar, and are retracted from the path of the fixture during the return stroke of the bar. The part is loaded into the fixture and clamped at one end station and is not released until the fixture has reached the other end station. At each work station the entire fixture is located and locked in position for the machining operation.

With stationary fixtures, one at each work station, the part enters a fixture, is located, clamped, machined, and released, and is then moved on to the next fixture. The part is moved by means of a conveyor or a transfer bar. The direct use of these transfer mechanisms requires that the part has one or several flat surfaces. When that is not the case, the parts are nested and clamped in pallets with flat surfaces that can slide on the track.

Regardless of the mode of operation, these fixtures have a number of common features. The part is usually provided with two tooling holes for dual cylindrical location, and the fixture has movable locators, known as "shot pins." They are in a retracted position as the part enters the fixture; the part is stopped in an approximately correct position and is finally located as the shot pins enter the tooling holes. The shot pin has a tapered (conical or polygonal) pilot end, or a bullet nose end, or a combination of tapered flat surfaces and strips of the original cylindrical surface, as shown in Fig. 21-24. The tapered portion pushes the part over and takes the wear, and the cylinder defines the final position. It maintains its accuracy, because it is not exposed to any significant sliding motion.

![Fig. 21-24. The action of a tapered shot pin.](image-url)
(double-action hydraulic cylinders), not by springs. A broken or otherwise inoperative spring does not retract the component, and the result is serious damage as the motion of the part is actuated.

A typical cam operated work station is shown in Fig. 21-25. The cam shaft provides the successive movements required to locate and clamp, and later unclamp, the pallet, and to actuate the various work slides in the station.

The various machining operations are performed with preset tooling and do not require tool setting blocks. Drill jigs and boring fixtures are equipped with bushings for the support of the tools.

Chip disposal is mechanized. Rotating brushes are provided for the cleaning of clamping surfaces as the part enters a fixture. A simple way of removing chips from operating areas and cavities within the parts is to provide intermediate stations where the part is tilted so that the chips fall out. Other intermediate stations are used for turning the part over so that new surfaces are brought into position for subsequent machining operations. Intermediate stations are also provided for automatic gaging of previously machined surfaces. If a surface does not gage correctly, a warning signal is actuated or the machine is shut down. Similar warning devices are used to indicate if a part is incorrectly located at a work station.

A typical detail from a production line is shown in Fig. 21-26. The picture shows two loaded work stations and a section of the transfer line which moves the part from station to station by means of transfer bars. The electromechanical drive seen in the foreground serves to raise, transfer, and lower the transfer bars with the parts.

**Fig. 21-25.** Diagram of a cam-operated work station in a transfer line.

**Fig. 21-26.** A transfer line with two work stations and the drive for the transfer bars.
Economics

Classification of Fixtures by Grade

The purpose of making economic estimates and calculations for a fixture is to justify its cost. The cost is affected by the accuracy required of the fixture and particularly by the level of simplicity or complexity embodied in its design. This applies particularly to the clamping devices. In this respect, fixtures can be classified into four grades corresponding to four different levels of production.

Small lot production, up to 40 pieces, requires fixtures of the simplest possible type with manually operated screw or cam actuated clamps.

Medium lot production, from 40 to 100 pieces, justifies the use of quick-acting clamping devices for single clamping, and multiple clamping devices, where applicable. Multiple clamping is the simultaneous actuation of several clamps acting on a single part or the simultaneous clamping of several parts in one fixture.

Large lot production, from 100 to 1000 pieces, represents the area where well designed time-saving clamping devices are a necessity. Multiple clamping is used where possible and clamps are air or hydraulically operated.

Mass production, over 1000 pieces, uses, in principle, the same power operated clamping devices but with the addition of such refinements as electrical control, remote control, or semiautomatic control of the clamp-actuating components.

Estimate of Profit

An economic estimate for a fixture shows whether or not it will be profitable. This involves a comparison between the savings obtained by the use of the fixture over a fixed period and the cost of using it. The result depends on a number of factors. They can be expressed as mathematical variables and written into an equation, which, in turn, can be solved for any one of them. However, it is clearer to calculate separately each of the two items, the savings and the cost, and then compare them.

The period considered is one year. The following symbols\(^1\) for the variables and constants are used:

\[
\begin{align*}
N & \text{ number of parts produced in a year} \\
s & \text{savings in labor cost per part produced in the fixture (dollars)} \\
L & \text{overhead rate (burden) on labor cost} \\
C & \text{cost of fixture (dollars)} \\
i & \text{yearly interest rate} \\
u & \text{yearly maintenance cost rate for the fixture} \\
t & \text{yearly cost of taxes, insurance, etc.} \\
a & \text{number of years required or estimated for amortization of the fixture cost.} \\
S & \text{(assuming an old, but still usable fixture is replaced by the new fixture) the unamortized value of the old fixture less its scrap value (dollars)}
\end{align*}
\]

The factors \(L, i, u,\) and \(t\) are expressed as decimal fractions, not as percents.

The annual saving by using the fixture is

\[
X = Ns (1 + L)
\]

(a)

The annual cost \(Y\) of using the fixture is

\[
Y = C (i + u + t + \frac{1}{a})
\]

(b)

or, if the new fixture replaces an old one, the annual cost \(Y_f\)

\[
Y_f = C (i + u + t + \frac{1}{a}) + Si
\]

(c)

---

\(^1\)These symbols are the same as those used in *A Treatise on Milling and Milling Machines*. (Cincinnati, Ohio: The Cincinnati Milling Machine Co., 1951.) 3rd ed., p. 747.
If the cost of setting up and removing the fixture is substantial, it is added before, known, \( C = \) the length \( L \) years, substantial, machining fixture the the +0.90) (a)

\[
\text{the fixture is profitable, breaks even, or loses money.}
\]

**Example—With the following values**

\[
s = 0.10, \quad L = 0.90, \quad C = 600.00, \quad i = 0.06, \quad u = 0.04, \quad t = 0.12, \quad a = 2 \text{ years}, \quad S = 200.00,
\]

how many parts must be machined per year to break even?

\[
X = N \times \frac{0.10 \times (1 + 0.90)}{0.19} = 0.19N \quad \text{(a)}
\]

\[
Y_f = 600 \times (0.06 + 0.04 + 0.12 + 1/2) + 200 \times 0.06 = 444 \quad \text{(c)}
\]

hence \( 0.19N = 444 \)

\[
N = \frac{444}{0.19} = 2337 \text{ parts per year.}
\]

**Example—With \( N = 3000 \) parts per year and other values as in the previous example, the fixture is profitable. What is the profit?**

\[
X = 3000 \times 0.10 \times (1 + 0.90) = 570 \quad \text{(a)}
\]

\[
Y_f = 444 \quad \text{(c)}
\]

**Annual profit:** \( 570 - 444 = 126 \)

**Example—With \( N = 2000 \) parts per year and other values, except \( C \), as before, what is the maximum allowable cost \( C \) of the fixture?**

\[
X = 2000 \times 0.10 \times (1 + 0.90) = 380 \quad \text{(a)}
\]

\[
Y_f = C \times (0.06 + 0.04 + 0.12 + 1/2) + 200 \times 0.06 = 0.72C + 12 \quad \text{(c)}
\]

hence \( 0.72C + 12 = 380 \)

Maximum allowable cost: \( C = \frac{380 - 12}{0.72} = 511.11 \)

**Example—With the following values for a fixture for an N/C machine tool operation**

\[
N = 3000 \text{ parts per year}; \quad s = 0.20; \quad L = 3.5 \quad (350\%); \quad C = 1600.00, \quad \text{and other values as before, is the fixture profitable, and if so, what is the profit?}
\]

\[
X = 3000 \times 0.20 \times (1 + 3.5) = 2700 \quad \text{(a)}
\]

\[
Y_f = 1600 \times (0.06 + 0.04 + 0.12 + 1/2) + 200 \times 0.06 = 1164 \quad \text{(c)}
\]

**Annual profit:** \( 2700 - 1164 = 1536 \)

**Estimate of Fixture Cost**

The profit or loss estimate requires that the fixture cost is known or estimated. The safest way of getting this figure is to make an ordinary cost estimate from the drawings. However, fixture drawings may not be available with sufficient details for a cost estimate, there may not be enough time, or there may be other reasons why the fixture designer must make his own estimate of the fixture cost.

The total cost is composed of material, labor, and overhead, including the design cost. The overhead rate is known, the material cost can be estimated closely enough from sketches. The labor cost can be estimated with sufficient accuracy for the present purpose from the formula

\[
H = A \frac{W^3}{V^2}, \quad \text{where} \quad A = 105 \text{ with average present-day technology.}
\]

\( H \) is machining and assembly time in hours, \( W \) is the weight of the fixture in pounds, and \( V \) is the overall volume of the fixture in cubic inches. \( V \) is calculated as length \( \times \) width \( \times \) height or \( \frac{\pi}{4} \times \)

![Fig. 22-1. Overall dimensions for cost estimate.](image_url)
diameter^{2} \times \text{height}. The dimensions are measured over the main body of the fixture; projecting flanges are included with one third of their actual width; local projecting parts such as bosses and feet are disregarded. For the fixture shown in outline in Fig. 22-1, the volume \( V \) is defined by

\[ V = A \times B \times C \]

where the dimensions are taken as indicated.
Measuring Angles in Radians

For most applications, angles are measured in degrees, minutes, and seconds. However, another measuring system, "Circular Measure," is preferred for certain applications, e.g., when the angle is conveniently defined by the length of an arc of a circle. Another common application of this system is in formulas relating to revolving bodies. The unit in the system is called a radian (abbreviated rad or rad.) and is the angle for which the arc has the same length as the corresponding radius.

For a circle of radius $r$ the length of a 180-degree arc is $\pi \times r$, and it measures $\frac{\pi \times r}{r} = \pi$ radians.

Consequently,

$180$ degrees $= \pi$ radians

$1$ degree $= \frac{\pi}{180} = 0.0175$ radian, and

$1$ radian $= \frac{180}{\pi} = 57.2958$, or approximately $57.3$ degrees

For small angles, the chord can be substituted for the arc because it is almost the same length. This leads to a simplification in calculations, because when the length of the chord is known, the angle is readily measured in radians, and there is no need for the use of trigonometric functions.
Appendix II

Transfer of Tolerances from the Conventional Dimensioning System to the Coordinate System

The coordinate system of dimensioning, with the point of origin for the reference lines (the coordinate axes) located at or near the upper-left-hand corner of the workpiece, is designed to be compatible with the scale readings available on most jig borers. Many of the recently developed N/C machine tools are also compatible with this system. For tool and die work done on the jig borer, dimensions are usually given to .0001 inch without specified tolerances, as the jig borer operator will work to the limit of accuracy of his machine. In this case, dimensions are transferred from the conventional dimensioning system to the coordinate system by simply adding or subtracting. An example is shown in Fig. II-1. The horizontal coordinate 4.7500 shown in view B is obtained, for instance, by the addition of the individual dimensions of 7/8 inch, 2 1/8 inches, 1/2 inch, and 1 1/4 inches shown in view A. For other work, where tolerances are included in the conventional dimensions, the tolerances must be transferred together with the dimensions when the change is made to the coordinate system. The tolerances in the coordinate system will not be the same as in the conventional system. In most cases the tolerances will be reduced in the transfer process. The new tolerances must be carefully calculated, otherwise serious errors will result.

The principle of the “Transfer of Tolerances” can be stated as follows:

When transferring tolerances from the conventional dimensioning system to the coordinate dimensioning system, the sum of the tolerances of any pair of dimensions on the coordinate system must not exceed the tolerance of the dimension that they replace on the conventional system.

As the first step, all dimensions with unilateral or with bilateral, but unequal, tolerances are changed to make all tolerances equal. In the second step, the conventional dimensions are transferred to coordinate dimensions. In the third step, the tolerances are transferred to comply with the principle stated above.

The application of the principle and the three steps is demonstrated by the following example. In Fig. II-2, views A and B show a part with conventional dimensioning. In view A some of the dimensions have mixed (unequal and equal) tolerances; in view B the basic dimensions are changed, as needed, to make all tolerances equal. View C shows the resulting coordinate dimensions with their tolerances.

Step 1. The +.003 tolerance on the 2.125 dimension is changed to ±.002, and simultaneously, the 2.125 is changed to 2.126. This does not change the physical dimension, because

\[
2.125^{+.003}_{-.001} = 2.128 = 2.126^{+.002}
\]

Step 2. The 3.0010 dimension is obtained by the addition

\[.8750 + 2.126 = 3.0010\]

Step 3. The available tolerance on 2.126 is ±.002 and must now be divided between two coordinate dimensions. If evenly divided, this leaves ±.0010 for each dimension, resulting in .8750±.0010 and 3.0010±.0010.

Sometimes the tolerance of a dimension on the coordinate-system drawing is affected by more than one dimensional requirement. In this case, the final tolerance used must be that which fulfills all of the requirements. For example, consider the .502±.002 dimension in view B. It is replaced by the 3.0010 and the 3.5030 dimensions in view C. If the 3.0010 and 3.5030 dimensions are each given a ±.001 tolerance, the sum of their tolerance would not exceed the original ±.002 tolerance and, presumably, the requirements for the transfer of tolerances would be met. However, the 3.5030 dimension together with the 4.6280 dimension replaces the 1.125 dimension in view B, which has a tolerance of only ±.001 inch. Thus, the sum of the tolerances on the 3.5030 and the 4.6280 dimensions cannot exceed ±.001 inch which, when divided equally, amounts to ±.0005 inch. The 3.5030-inch dimension must therefore be given the lesser tolerance of ±.0005 inch.

For this reason, when transferring the tolerance it is usually best to start with the smallest tolerance. Also note that the sum of the tolerances replacing the .502±.002 dimension is less than the ±.002 inch. This is satisfactory since the sum of the new tolerances does not exceed the original tolerances.

When a tolerance is bound by a small tolerance, as in the case of the .502 dimension, it is sometimes possible to increase an adjacent tolerance on the coordinate dimension drawing. For example, the 2.000-inch dimension in view B, which has a tolerance of ±.005 inch is replaced by a 1.0000- and a 3.00000-inch dimension on the coordinate drawing. Each of these dimensions could be given a tolerance of ±.0025 inch; however, the tolerance of the 1.0000-inch dimension on the coordinate drawing is bound by the requirement of the ±.001 tolerance of the 1.000 dimension shown at the right side, in view B. Thus, the 1.0000 dimension in view C, together with the 2.0000 dimension, must have a total dimension tolerance of ±.0010 or ±.0005 inch on the 1.0000-inch dimension. Since the sum of the tolerances of the 1.000 and the 3.0000 dimensions (view C) can be ±.005 inch, the tolerance of the 3.0000 dimension can be increased to ±.0045 inch.

The basis for the determination of the final tolerances for the coordinate dimensions (view C) is summarized below:

| .8750±.0010 | The ±.001 tolerance together with the ±.0010 tolerance of the 3.0010 dimension is required to maintain the ±.002 tolerance for the 2.126 dimension. |
| 3.0010±.0010 | See requirements for the .8750±.0010 dimension given above. |
| 3.5030±.0005 | The ±.0005 tolerance together with the ±.0005 tolerance of the 4.6280 dimension is required to maintain the ±.001 tolerance for the 1.125±.001 dimension. |
| 4.6280±.0005 | See requirements for the 3.5030±.0005 dimensions given above. |
| 1.0000±.0005 | The ±.0005 tolerance together with the ±.0005 tolerance of the 2.000 dimension is required to maintain the ±.001 for the 1.000±.001 dimension given at the right, in view B. |
| 2.0000±.0005 | See requirements for the 1.0000±.0005 dimension given above. |
| 3.0000±.0045 | The .0045 tolerance, together with the ±.0005 tolerance of the 1.0000 dimension in view C, is required to maintain the ±.005 tolerance for the 2.000±.0005 dimension in view B. |
The Dimensioning of Fixtures by Stress Analysis

Although the structural design of fixtures has not been given much consideration in most textbooks on stress analysis, they can be designed systematically by the proper application of known formulas and calculation procedures. An underdimensioned fixture may be damaged or destroyed in use. An overdimensioned and, therefore, overweight fixture is a constant source of unnecessary expense for excessive work in handling, transportation, and storage, etc., of the fixture. The forces for which the fixture is analyzed are the external loads, the clamping loads, and the reactions. The external loads comprise the cutting forces, the weight of the part and the fixture, and inertia forces. Inertia forces are the centrifugal forces in the fixtures and rotating grinding fixtures, and the deceleration and acceleration forces at stroke reversal in fixtures for planers and surface grinders.

A rule-of-thumb says that a fixture stress analysis shall be performed when the weight of the part is 25 pounds (110N) or more. This weight is exemplified by a 3- by 3- by 10-inch (75- by 75- by 250-mm) solid block of steel, or by a hollow aluminum casting, open on one side, with 1/2 inch (13 mm) wall thickness and 8- by 8- by 16-inch (200- by 200- by 400-mm) overall dimensions. The cutting forces run into hundreds, if not thousands of pounds, and are always somewhat approximate. There is, therefore, no need to include the weight of the part and the fixture in a static stress analysis as long as these weights do not exceed 10 percent of the main cutting force.

Formulas for calculating centrifugal forces are found in the Mechanics sections of reference books, such as Machinery's Handbook. The acceleration of a planer table at stroke reversal is of the order of magnitude of from 0.01g to 0.1g and is insignificant except in special and extreme cases.

To calculate the load from the cutting tool, it is resolved into its three components as shown in Fig. III-1. They are:

- $F_C$, The main cutting force or, simply, the cutting force. It is the force component acting in the direction of the tool travel (the direction of cut) relative to the workpiece. In a cylindrical turning operation it is the tangential force component.
- $F_F$, The feed force. This is the force component acting in the direction of the feed, i.e., parallel to the surface which is being generated in the machining operation. In a cylindrical turning operation it is the longitudinal force component.
- $F_T$, The thrust force. This is the force component which acts in the direction perpendicular to the surface being generated. In a cylindrical turning operation it is the radial force component.

Force components are in pounds or newtons. A single-point tool has only one set of force components. For multiple-point tools (drills, milling cutters, broaches) there is a set of force components for each cutting edge which is actively cutting. $F_C$

---

is the major force component and is the component that determines the amount of work and horsepower absorbed in the cutting operation. \( F_F \) and \( F_T \) are significantly smaller than \( F_C \). Average values are
\[
F_F \approx \frac{1}{3} F_C \text{ to } \frac{2}{3} F_C, \text{ and}
\]
\[
F_T \approx \frac{1}{4} F_C \text{ to } \frac{1}{2} F_C
\]

\( F_F \) is maximum and \( F_T \) is minimum when the side-cutting-edge angle (SCEA) is zero; \( F_F \) decreases and \( F_T \) increases with increasing SCEA. The size of \( F_C \) and the other force components depends on the material, the dimensions of the cut, and the cutting speed. Detailed data are found in reference and text books. However, for the purpose of dimensioning fixtures it is sufficient to use the approximation that \( F_C \) equals the unit (specific) cutting pressure \( p_C \) multiplied by the area of cut \( A_o \):
\[
F_C = p_C A_o = p_C f d
\]

where,
\[
A_o = \text{area of cut (square inches or mm}^2\text{)}
\]
\[
f = \text{feed per revolution or per tooth (inches or mm)}
\]
\[
d = \text{depth of cut (inches or mm)}
\]

\( p_C \) is essentially a material constant and can be taken as 2.5 to 3.2 times the tensile strength for steel and other ductile materials, and 4.5 to 5.6 times the tensile strength for cast iron and other brittle materials,

where the effects of dimensions of cut and cutting speed are reflected in the ranges quoted for the co-efficients. The higher values are to be used for fine feeds or shallow depths of cut (small \( f \) and \( d \)) and lower cutting speeds (as used with high-speed steel tools), the lower values are for heavy cuts and/or higher cutting speeds (as used with carbide and ceramic tool materials). In the final calculation of \( F_F \) and \( F_T \) a contingency factor is introduced to allow for tool wear, cutter runout, and local variations in material dimension and hardness. For single-point tools and drills, this factor is 1.25. For milling cutters it is 2. For twist drills, \( F_F \) is further increased by a factor of 1.33 to allow for the additional resistance caused by the chisel edge. Data for drilling forces are found in text and reference books.\(^2\),\(^3\)

The clamping forces must secure the part against being pulled out of the fixture by the cutting forces. Detailed calculations for the various types of clamps are given in Chapter 10. The safety factor against pullout should be not less than 1.5, however, in most cases it will be found that a safety factor of 2 or better can easily be established.

With the forces calculated, the elements of the fixture can now be dimensioned. Regardless of how complicated the fixture may appear, with a little practice on the part of the designer it can always be subdivided into simple structural elements. These elements are cantilever beams, simple beams, shafts and bolts (loaded in torsion and/or bending), flat or curved plates of square, rectangular, or circular circumference, cylinders, angles, and, occasionally, columns. Formulas for dimensioning these are found in Machinery's Handbook.\(^4\)

\(^{2}\)Ibid., pp 1743, 1744.
\(^{4}\)Oberg, op. cit., pp 402-441.
## Metric Conversion Tables

### Fractional Inch—Millimeter and Foot—Millimeter Conversion Tables

(Based on 1 inch = 25.4 millimeters, exactly)

#### FRACTIONAL INCH TO MILLIMETERS

<table>
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<th>Fraction</th>
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#### INCHES TO MILLIMETERS

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#### FEET TO MILLIMETERS

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<tr>
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</table>

**Example 1:** Find millimeter equivalent of 293 feet, 3½ inches.

\[
293\text{ ft} = 89,452.053 \text{ mm} \\
3\frac{1}{2} \text{ in.} = 89.452 \text{ mm} \\
293.5 \text{ ft, } 3\frac{1}{2} \text{ in.} = 89,452.053 \text{ mm} 
\]

**Example 2:** Find millimeter equivalent of 71.86 feet.

\[
70\text{ ft} = 21,335 \text{ mm} \\
1\text{ ft} = 304.8 \text{ mm} \\
0.86 \text{ ft} = 262.88 \text{ mm} \\
71.86 \text{ ft} = 21,902.928 \text{ mm} 
\]
## METRIC CONVERSION TABLES

### Decimals of an Inch to Millimeters

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### Decimals of a Millimeter to Inches

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### Use previous table to obtain whole inch equivalents to add to decimal equivalents figures given in this table are exact. Figures to the right of the last place figures are all zero.
### METRIC CONVERSION TABLES

**App. IV**

#### Millimeters to Inches (Based on 1 inch = 25.4 millimeters, exactly)

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**Conversion Table**

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**Additional Data**

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