THE THEORY AND APPLICATION OF PRECISION MECHANICAL COMPONENTS

the theory and application of PRECISION MECHANICAL COMPONENTS

the theory and application of PRECISION MECHANICAL COMPONENTS

a guide for engineers, designers, draftsmen, and technicians

by Winfred M. Berg, Me.

Second Edition

RUSSELL E. SACKEN, Inc.
East Rockaway, N. Y.
1965

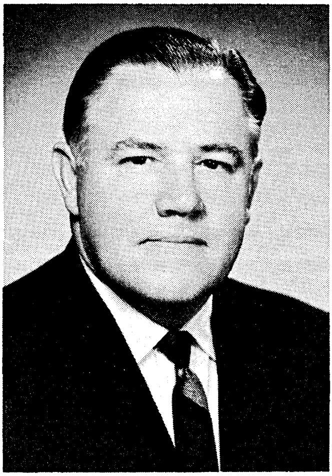
PUBLISHED & COPYRIGHTED 1965
RUSSELL E. SACKEN inc.
EAST ROCKAWAY, NEW YORK
PRINTED IN U.S.A.

PREFACE

This handbook for designers, engineers and draftsmen is concerned with the precise transmission of motion from one precision instrument component to another.

Its purposes are to: (1) familiarize designers, engineers and draftsmen with standardized precision mechanical instrument components; (2) provide basic design information which will help build better systems; and (3) explain the design and engineering advantages and disadvantages of a wide variety of such precise components and equipment available in today's market.

The electronic-mechanical precision instrument field is a relatively new one in which practitioners have independently developed their own ways of working – some good, some not so good. This handbook is the first attempt to centralize in one volume preferred methods of practice and to present helpful techniques, skills and basic tricks of the trade. It will be of particular aid to electronic and mechanical engineers, laboratory technicians, beginners, draftsmen and to designers who work only occasionally in the precision instrument field. It will also be of value to experienced designers and engineers whose present knowledge may be expanded and improved.



Winfred M. Berg

ABOUT THE AUTHOR

The material contained in this publication has been prepared by Winfred M. Berg, Chief Engineer of PIC Design Corp., East Rockaway, L.I., New York. Much of the material contained herein has been presented at seminars, conferences and meetings with engineers, designers and draftsmen associated with companies all over the United States and Canada. Certain portions of this material have also been published in leading technical publications. It has been consolidated for easy reference. It should be a distinct aid in building better products for the ever-growing precision field of electro-mechanical, servomechanism instrumentation and tooling.

CREDITS

"Thanks are in order to the following for allowing the use of referenced material, data and photographs":

PIC Design Corp. .... East Rockaway, N. Y.
Barber Coleman Corp. .... Rockford, Ill.
Gleason Machine Works .... Rochester, N. Y.
Fellows Machine Co. .... Springfield, Va.
National Std's Assoc. .... Washington, D. C.

Mr. Berg is the author of the following published technical articles:

"Musts and Mistakes of Precision Assembly"
Publication ★ Automatic Control .... Dec. 1957

"Backlash in Instrument Gears"
Publication ★ Military Systems Design .... July 1959

"Miter and Bevel Gear Differentials"
Publication ★ Design News .... March 1960

"Breadboards Speed Electro-Mechanical Designs"
Publication ★ Electronic Industries .... May 1961

"Gear Precision and Quality"
Publication ★ Magazine of Standards .... July 1962

"Servosystem Breadboards"
Publication ★ Machine Design .... Jan. 1963

"Designer's Tip on How to Use Mechanical Instrument Components"
Publication ★ Systems Design .... April 1964

"Plastic Transmission Belts with Geared Pulleys"
Publication ★ Engineering Materials and Design .. Dec. 1964

"Small Drives Designed for Reliability"
Publication ★ Power Transmission Design .... Feb. 1965

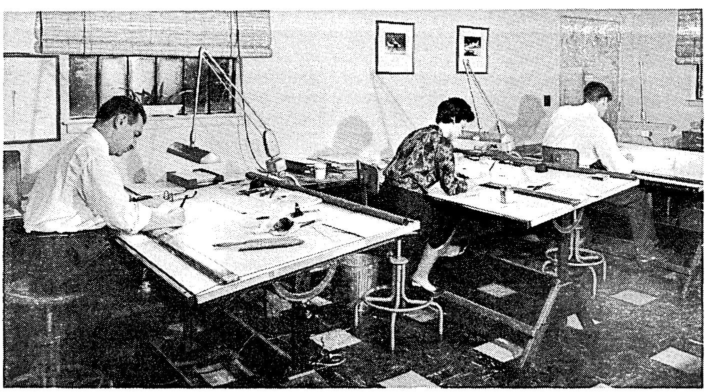
TABLE OF CONTENTS

Chapter	*Title*	*Pages*
1	Basic Problems	1
2	Spur Gears	19
3	Gear Boxes and Gearhead Speed Reducers	55
4	Worm and Wheel Gear and Packaged Speed Reducers	61
5	Helical, Miter and Bevel Gears	69
6	Mechanical Differentials	75
7	Chain and Belt Drives	83
8	Couplings	97
9	Magnetic Clutches	105
10	Dials	113
11	Servo Breadboards	119
12	Tool and Fixture Components	125
13	Miscellaneous Components	141
14	Engineering and Reference Data	151
Index		165

Chapter One

BASIC PROBLEMS

In this exciting era of space and automation development *time* is of utmost importance. *Cost* is also becoming increasingly important. Yet, precious time and countless dollars are being wasted in the design and production of instruments and instrumentation systems because of lack of utilization of *standard components* which are now readily available. Instead, millions of dollars worth of skilled man hours are expended in the design and fabrication of special components which would not be needed if standard components had been specified.



Typical Design Drafting Section of Small Development Company

There are two ways to attack a mechanical or electronic-mechanical design problem. The *hard way*, which is usually the most costly way, is to specify non-standard components, which must be designed, engineered, and custom-made, and for which replacements are not readily available.

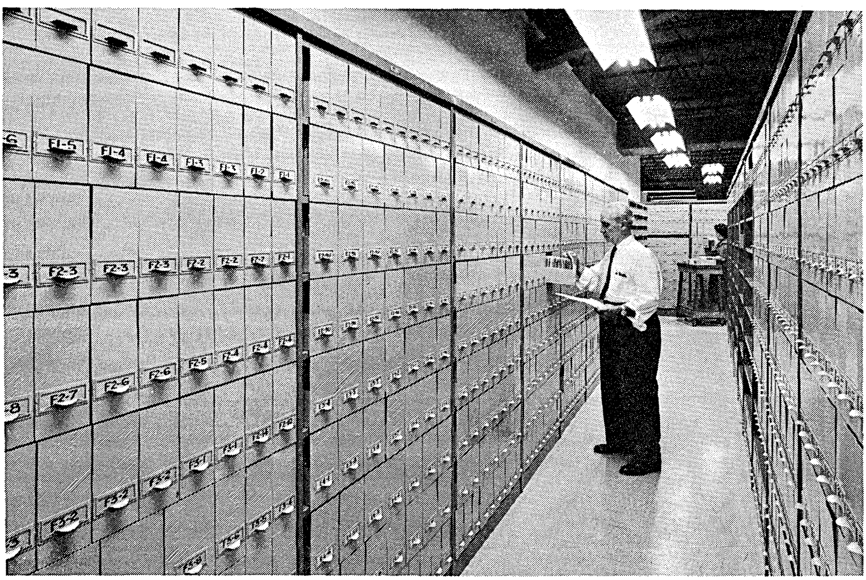
The *easy way*, which is usually the least costly, is to design around standard components which have already been designed and are readily available at modest cost, with improved accuracy, reliability and immediate replacements.

Of course, there is always the *N.I.H.* factor to contend with, which means *"Not Invented Here."* This factor adds greatly to cost and slows down delivery of the end-product. In addition, it aggravates and compounds the problems of the customer and his maintenance personnel.

The pressures of accelerated delivery time and cost reduction are causing engineers, designers and manufacturers to thoroughly explore the availability of standard components which can be employed in their products. They are finding that they can obtain greater precision at lower cost and can save considerable time by avoiding the design of special components for uses where standard components will do the job as well if not better.

SPACE AGE PRECISION

Most of the requirements of space program instrumentation can be met using standard components, including ever-tightening specifications for precision. Until recently, one degree of accuracy for 5000



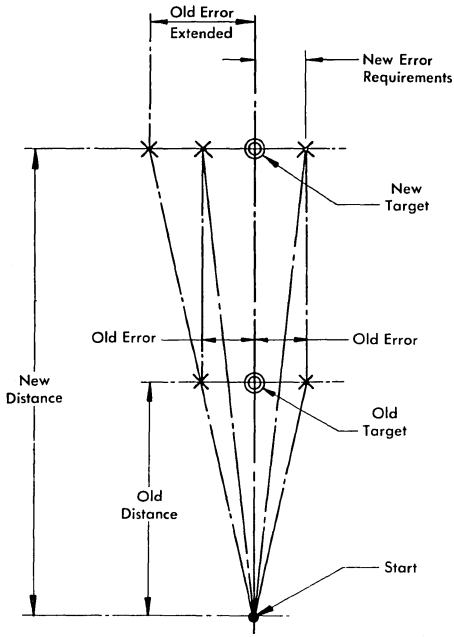
Stock Room of "Off the Shelf" Stock
Standardized Precision Component Supplier

miles of air or target projection was acceptable. Now, however, new armaments, missiles and rockets cannot be harnessed to 5000-mile tolerances since some must be projected thousands upon thousands of miles into space. Unless the one degree of relative accuracy is at least split in half to reduce the error by 50%, it is apparent that missiles traveling 10,000 miles may be as much as two degrees off target.

Error becomes even greater as range increases. Even at the relatively short distance of 10,000 miles, we can't hit the broad side of a barn door unless starting error is radically reduced. (See Figure 1-1 which illustrates the relationship of starting error to distance.)

ERROR EFFECT ON DISTANCE

This margin of error is being reduced by the joint efforts of manufacturers of precision electrical and mechanical components all working together. As we probe farther and farther into space, we must reach into the resources of our minds to keep pace with the exacting demands of these new, more-distant targets while at the same time reducing costs, size and weight.



Error Effect on Distance
FIGURE 1-1

Constant supervision and ingenuity are required to keep the costs of new products within practical limits. The factor that will keep costs within sight, with the assurance that the job will be done right, is *standardization.*

The *AN* and *MIL* standards applicable to mechanical and electronic-mechanical hardware are well known. The establishment of *JAN* standards started us off in the *right direction.*

There is an unspoken desire for more coordination among components manufacturers, equipment and systems manufacturers and end-users. This desire, however, needs to be transformed into action. A good example of what can be achieved by such action is the notable achievements gained through

AN & MIL Standards – Published by National Standards Assoc., Wash., D. C. Precision Instrument Component Std's – PIC Design Corp., East Rockaway, N. Y.

the formation of *The Standards Engineers Society* whose aims are to improve communications among manufacturers and customers and the establishment of tolerance and manufacturing standards and test procedures.

PRECISION INSTRUMENT COMPONENTS

Precision instrument components are basic building blocks, consisting of such small precision-made components as shafts, bearings, gears, differentials, speed reducers, couplings, clutches, cams, brakes, adapters and other elements used to transmit precise mechanical motion with as little motion lost as possible. Typical precision instrument components are shown in Figure 1-2.

Precision instrument components carry information from one Mechanical or Electro Mechanical mechanism to another. For example, the mechanism which aims a missile, gun, etc. must be supplied with a stream of information pertaining to many factors, such as changes in the speed of the target, wind velocity, elevation of

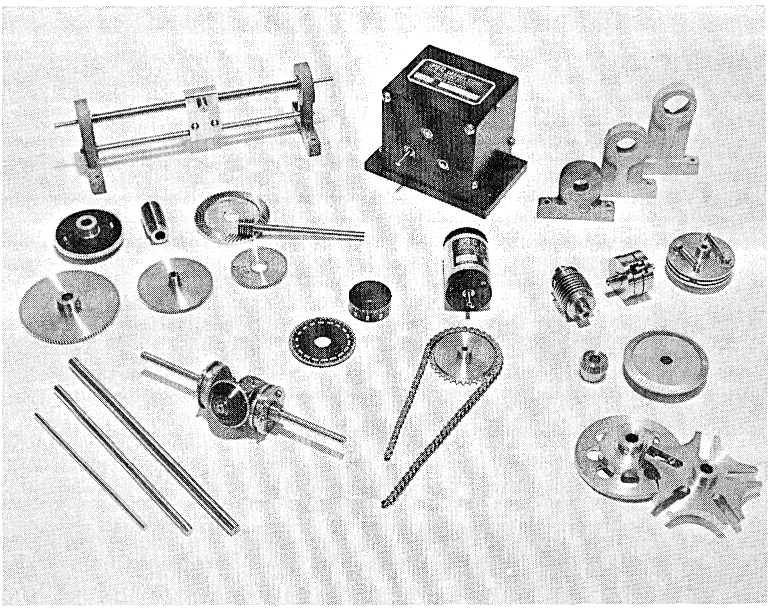


FIGURE 1-2 *Typical Precision Mechanical Components*

target, range, and so on. Changes in these conditions are continuously obtained by other mechanisms, and this information is instantly fed to the aiming mechanism by interconnecting electrical or mechanical precision instrument components. Obviously, the accuracy with which the end mechanism performs is largely dependent on the precision of the interconnecting and signal transfer components.

HANDLING PRECISION INSTRUMENT COMPONENTS

Precision instrument components must be handled carefully by those who receive and inspect them as well as by the engineer or technician who uses them. Otherwise their built-in precision may be destroyed and systems using them will not perform as expected.

Precision instrument components should not be scooped or tossed into trays or boxes or dumped onto tables. Each part should be picked up and set down individually. If placed on a table, care must be taken that parts will not be jostled onto the floor as illustrated in Figure 1-3. When storing, each part should be wrapped in tissue and placed in its own individual envelope or correctly placed in its jewel case if one is provided.

PRECISION SHAFTS AND GEARS

Skillful engineering requires that precision-made items be used in a manner that will not alter their inherent precision. For example, in selecting standard precision instrument shafts and bearings, loose-

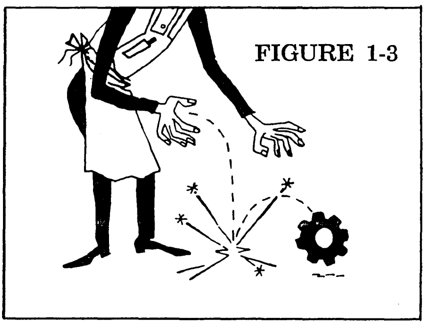


FIGURE 1-3

Handle Precision Parts With Care

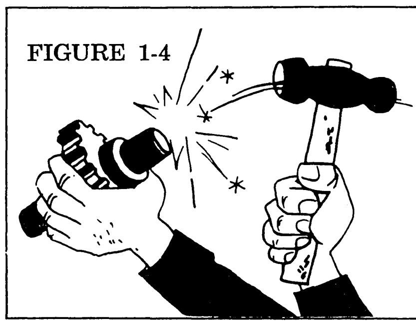


FIGURE 1-4

Wrong Method of Fitting a Shaft to a Gear Bore

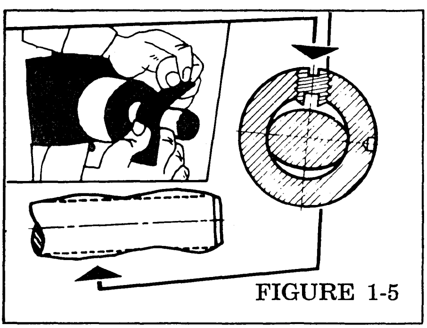


FIGURE 1-5

Do Not Polish Shafts to Fit Gear or Bearing Bores

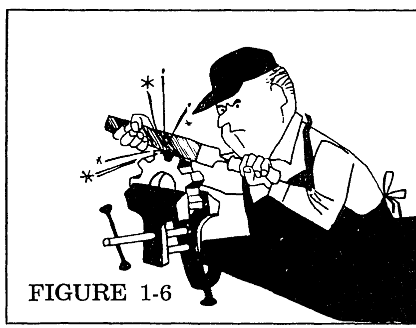


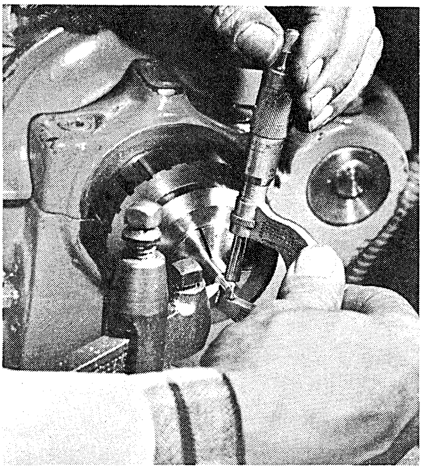
FIGURE 1-6

Do Not Rework or Modify a Precision Gear

fitting components should be specified rather than components which will require a drive or interference fit. The latter involves polishing an oversize shaft until it can be driven or forced into the bearing bore. Polishing a shaft, ready-made to precision-instrument tolerances, egg-shapes the once-perfect circumference, regardless of how much care is taken, and shaft contact with the inner bearing race falls, not on a completely uniform shaft, but on the few remaining high spots. This can cause the inner race to cock at unsupported sections between shaft high points. If the shaft is driven to fit, (See figure 1-4), the inner race of the bearing may be expanded, thus reducing the clearance between the balls and the races and causing the balls to pit the races and convert a normally smooth bearing operation into one which becomes rough, noisy and perhaps eccentric. (See Figure 1-5.)

Likewise, stock precision-instrument gears should not be modified. (See Figure 1-6.)

If a stock item is not available for a given requirement, order the special gear at once. It is impossible to drill or ream the bore of a ready-made precision instrument gear and expect to maintain the same concentricity between the established pitch diameter and bore. It is also impossible to machine down the face without pushing burrs



Shaft Journal Being Turned in Lathe

Precision Gear Teeth Being Dipped and Protected to Eliminate Possible Damage During Shipping

into the teeth, spoiling the initial precision. The engineer makes a tragic mistake when he specifies the modifications of a gear since the *error* he thus introduces multiplies and the entire system in which it is used is thrown out of control.

WORK IN A CLEAN AREA

When assembling and using precision instrument components, care and attention should be given to the cleanliness of the working area. Engineers should not smoke or drop ashes in an area where precision components are being handled or stored.

Cleanliness starts right at the beginning, *in* the *basis* shop, where rough machining chips and dirt are generated. (See Figure 1-7.)

Many R&D labs and manufacturers now have *clean rooms* (also called *white rooms)* in which precision parts and assemblies are handled under controlled conditions. Employees are required to wear lint-free smocks and hats when entering clean areas.

SHAFT AND BEARING SPECIFICATIONS

To avoid damaging shaft or bearing and destroying the initial precision of these parts, bearings and shaft should be specified so that a sensitive finger push will insert the shaft in the bearing. For example, a Class 7 bearing of ¼″ nominal bore will have a bore of .2498″ plus .0002″. To insure a loose fit, the mating shaft should be

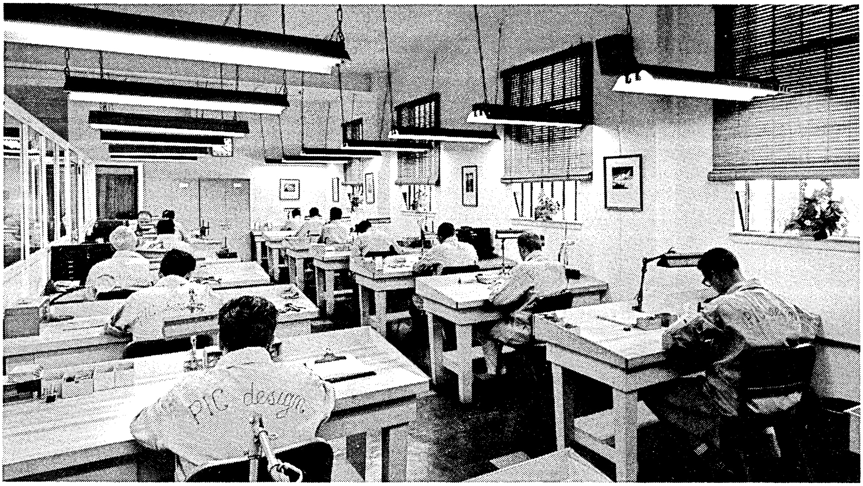
Typical Precision Lathe Metal Turning Dept.

specified .2497″ minus .0002″. This will provide a clearance of .0001″ to .0005″ and satisfy the push fit requirement. Shafts to this dimension are commercially available for push fit assembly with standard precision bearings, so no machine work is required to obtain proper shafts for standard bearings. (See Figure 1-8.)

IMPORTANCE OF SHAFTS

The rotation of a precision instrument shaft represents a change in some condition. It may be change in speed, distance, angle or of a hundred and one other things. Therefore, since the shaft is a basic information-carrying component, it is essential that this relatively simple element not be overlooked in considering the accuracy of the system in which it is used.

Standard precision instrument shafts have 10 micro finishes, or better (see Figure 1-9) achieved through double-pass, centerless



Clean Instrument Assembly Department

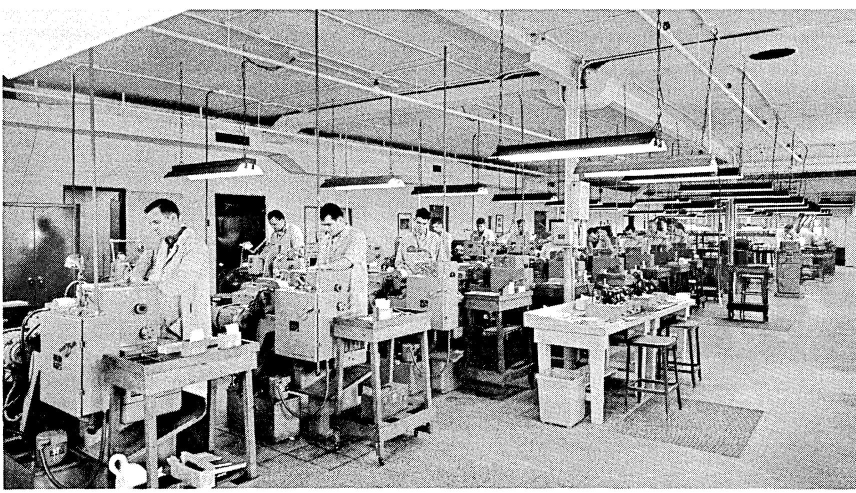


FIGURE 1-7 *Typical Clean Precision Machine Shop*

grinding and careful dimension roundness and straightness control. The ends of these shafts are *chamfered;* the chamfer acts as a funnel to help assemble parts properly. The chamfer also does away with sharp edges and burrs which might cause interference cocking, or cutting of the inner race of the ball bearing or gear when the parts are assembled.

Many stock shafts may be purchased and then slightly modified to give any particular requirement per Figure 1-10.

ULTRA PRECISION

Ultra-precision techniques can be used to hold clearances between shaft and bearing between .00005″ and .00025″. Shafts can be ground to minus .0001″ instead of minus .0002″, and finished with an 8-microinch finish (See Figure 1-8) instead of a 10-microinch finish. By

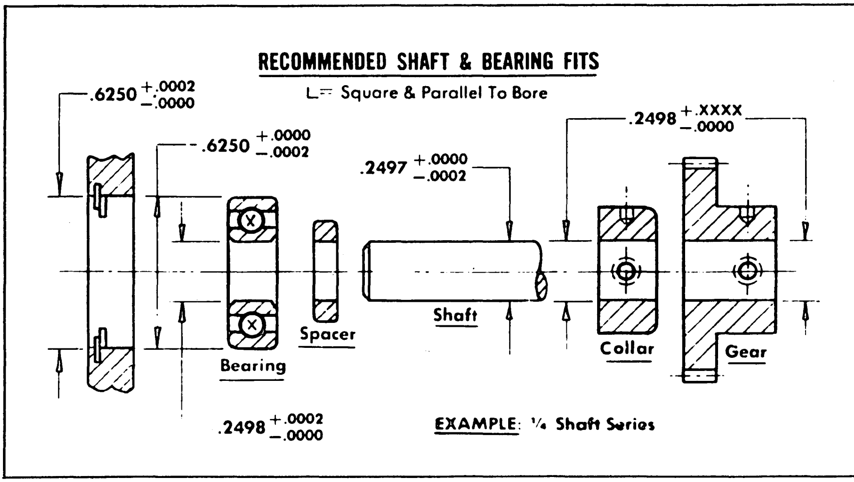


FIGURE 1-8

mating a standard ultra precision shaft of diameter .24975″ minus .0001″, with a standard select ultra precision bearing of bore size .2498″ plus .0001″, the above .00005″ to .00025″ clearance is obtained.

Ultra-precision shafting is roughly twice as expensive as precision-ground shafting and can only be produced in lengths up to 6″ with present day equipment. Select ultra-precision bearings cost approximately one dollar more than Class 7 bearings. These costs may be

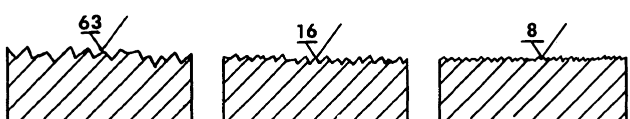


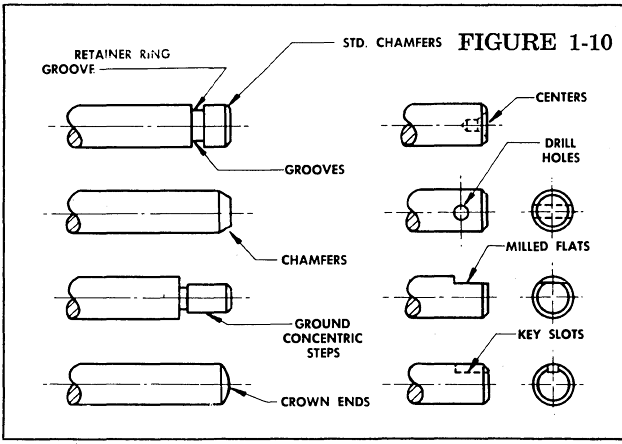
FIGURE 1-9 *Typical Micro-Inch Comparisons*

well worth paying when the utmost in precision is required. Ultra-precision finishes are illustrated in Figure 1-9.

GEAR RATIOS

In transmitting motion from a high-speed device to a low-speed device, or vice versa, there are generally a number of gear-train arrangements which can be used. Figure 1-11 illustrated a practical combination. The designer usually tries to use as few gears as possible but at the same time avoid high gear ratios which would involve gears of widely varying sizes and accuracy. Ratios greater than five to one are not recommended.

FIGURE 1-10



Typical Modifications to Standards to Give Special Shafts

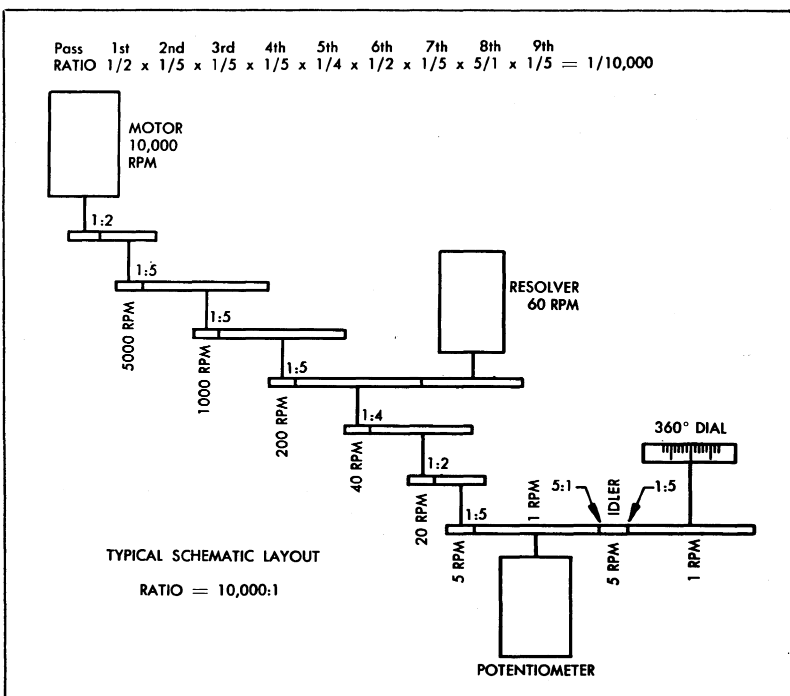
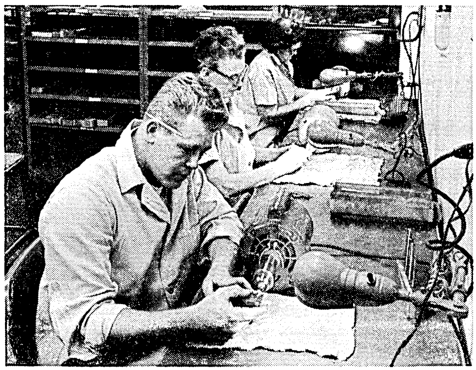


FIGURE 1-11 *Typical Gear Schematic*

BORE AND PITCH

Bore, of course, depends on shaft size, and in a speed-reducing gear train generally is increased as speed is reduced and as torque increases, thus the basic strength of the unit must be increased.

Pitch, or diametral pitch, as previously stated, is the ratio of the number of gear teeth to the pitch diameter. In a gear of given diameter, the greater the pitch, the greater the number of teeth. More teeth make for smoother action and less tooth-to-tooth error. However, more teeth means smaller and weaker teeth, and under high



De-Burring Precision Gears

torque conditions, coarse pitch rather than fine pitch gears would be used. As speed is reduced, pitch and shaft size should be increased.

Example:	*Pitch*	*Shaft*
Input pinion and gear	96	1/8″
Next mesh	96	1/8″
Next mesh	72	3/16″
Next mesh	72	3/16″
Next mesh	64	1/4″
Next mesh	64	1/4″

Maximum recommended rule of thumb loads for various pitches are:

Pitch	*Maximum load (oz.)*
24	500
1/10″	500
32	400
48	300
1/20″	200
64 & 72	100
80 & 96	50
120	25
200	10

Pitch may be selected by determining the load placed on the gear by the driving medium, and selecting the nearest pitch which will handle the given load. The suitability of selected gear may also be checked from the strength standpoint by determining the load placed upon it and checking the manufacturers load table to see if the selected gear will handle the applied load.

GEAR THICKNESS (Face Width)

In most pitches the designer can specify two or more gear thicknesses. In the precision field this primarily enables him to mis-mesh gears but, of course, a wider tooth is also a stronger tooth. By meshing wide-face pinions with narrow-face gears, the possibility of the introduction of error due to burrs pushed in between gear teeth by the gear cutting tool is minimized, since only one or neither potentially burred diameter will be in contact with the other gear. Burrs

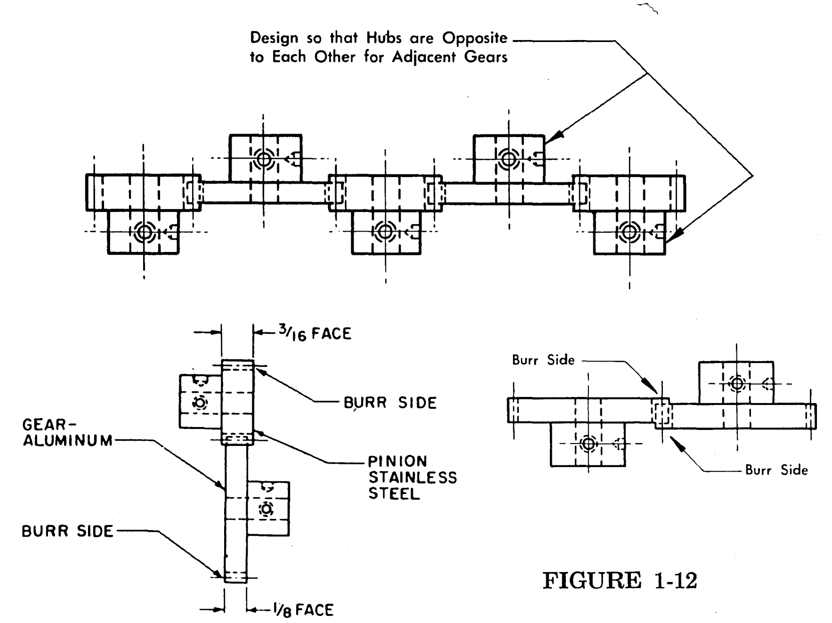


FIGURE 1-12

formed along the face of the gear can largely be eliminated by hand de-burring. Those pushed between the teeth as the cutting tool completes the cutting of a given tooth or during de-burring, are more difficult to remove without damaging the precision of the gear. Mismeshing minimizes inaccuracies which might be caused by such residual burrs. (See Figure 1-12.)

BACKLASH DUE TO ASSEMBLY OPERATIONS

Although it is possible to calculate backlash caused by variations of the gears themselves, it is not possible to calculate backlash factors caused by bearing variations, and by the many assembly tolerance factors. Therefore, a system of mounting bearings on alternately fixed and variable centers has been developed and encouraged. (See Figure 1-13.)

ACCURACY OF PRECISION GEARS

The accuracy of a gear system depends on many factors, one of the most important being the size or diameter of the gear. The toler-

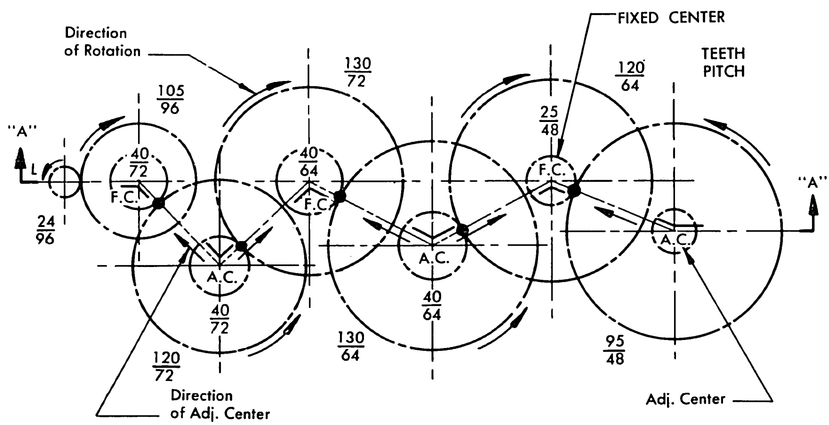


FIGURE 1-13

ances for various classes of standard and ultra-precision gears are shown in Figure 1-14.

The class of the gear specified is determined by the gear size. For example, a large diameter gear would require a Precision 1 class gear,

Class Gear	P.D. Tol.	O.D. Tol.	Bore Tol.	Side Wobble at 1″ Rad.	O.D. Concentricity to Bore
Prec. 1	–.001	–.002	+.0005	.001	±.0005
Prec. 2	–.0007	–.0015	+.0003	.0007	±.0003
Prec. 3	–.0005	–.001	+.0002	.0005	±.0002
Ultra Prec. 1	–.0004	–.0008	+.00015	.0004	±.00015

FIGURE 1-14

not Precision 2 or 3. The rule of thumb for selecting the appropriate class is given below:

Gear Class	*Maximum Gear Size*
Precision 1	Approx. 15″ diameter
Precision 2	Approx. 8″ diameter
Precision 3	Approx. 6″ diameter
Ultra-Precision 1	Approx. 2-1/2″ diameter

Other factors which may affect basic accuracy are the gear blank configuration, face width, material and quantities involved. Most

General Milling and Jig Boring Shop

manufacturers will accept stringent accuracy requirements on a best effort basis only.

FIXED AND ADJUSTABLE CENTERS

Fixed-center bearings are mounted in the jig bored or milled bearing plates and variable-center bearings are mounted in small diamond-shaped plates. Two fixed center shafts are first installed and a variable center shaft is then located so its gears mesh properly with the gears on the fixed shafts. (See Figure 1-15.) The diamond-shaped plate is then secured to the main bearing plate, with assurance of minimum backlash conditions.

BEARING INSTALLATION

Bearings must be installed accurately if the components carried on their shafts are to perform to precision. Bearing-plate holes must be drilled true, to the nominal outside of the bearing plus .0002″. The outside diameter of the precision instrument bearing is to the nom-

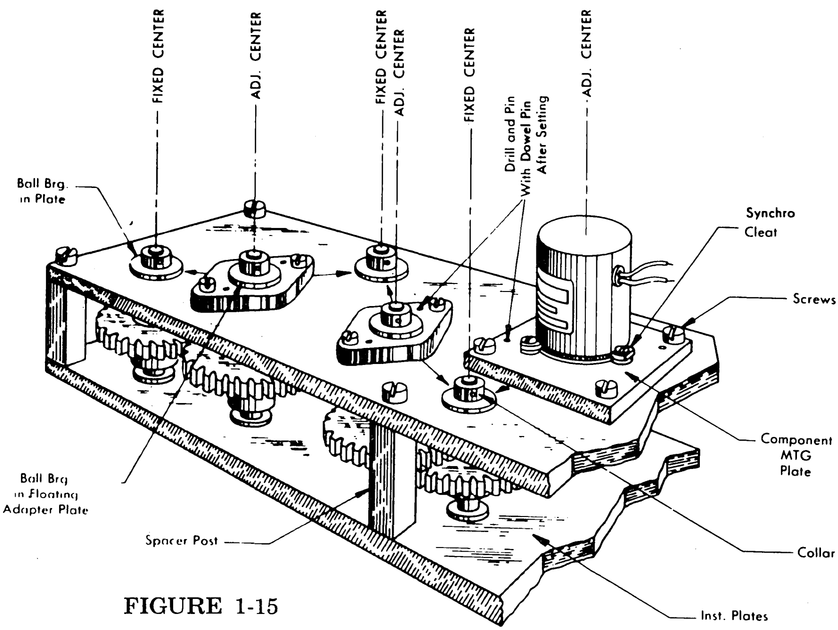


FIGURE 1-15

inal dimension minus .0002″, allowing a non-binding but close fit. Bearings may be secured by means of an inset in the bearing-plate hole, an integral flange or by retainer ring and shim spacer. (See Figure 1-16.)

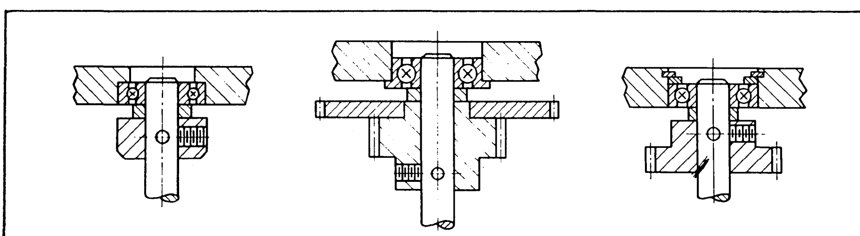


FIGURE 1-16 *Types of Bearing Preloading*

To assure alignment of holes in opposing bearing plates, plates should be clamped together and drilled as a unit to required gear class accuracy, dowel pinned and screwed together. (See Figure 1-17.)

If fixed centers are a must, then precise center distance control is of utmost importance. (See recommended tolerances.) (See Figure 1-18.)

Jig boring for production and uniformity can now be done on pre-established punched tapes and fed into a tape control jig borer for more exacting control.

DOWEL PINNING

Dowel pins are used to retain parts in an exact fixed position and to maintain positive alignment when parts are to be assembled and disassembled and exact positioning required.

Use no more than two (2) dowel pins at all times when holding two pieces together. For parts which have to be taken apart fre-

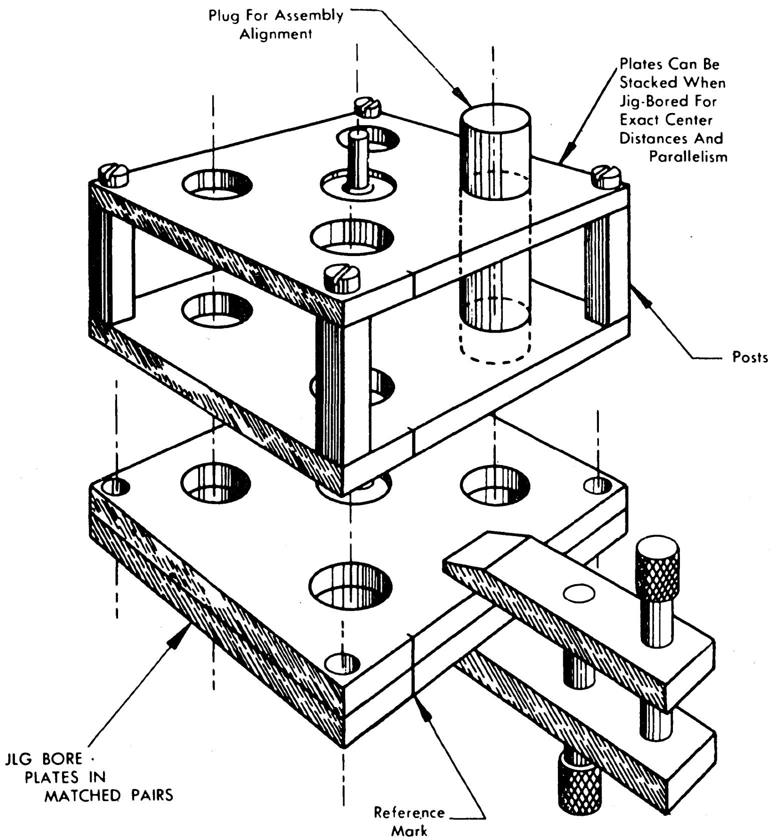
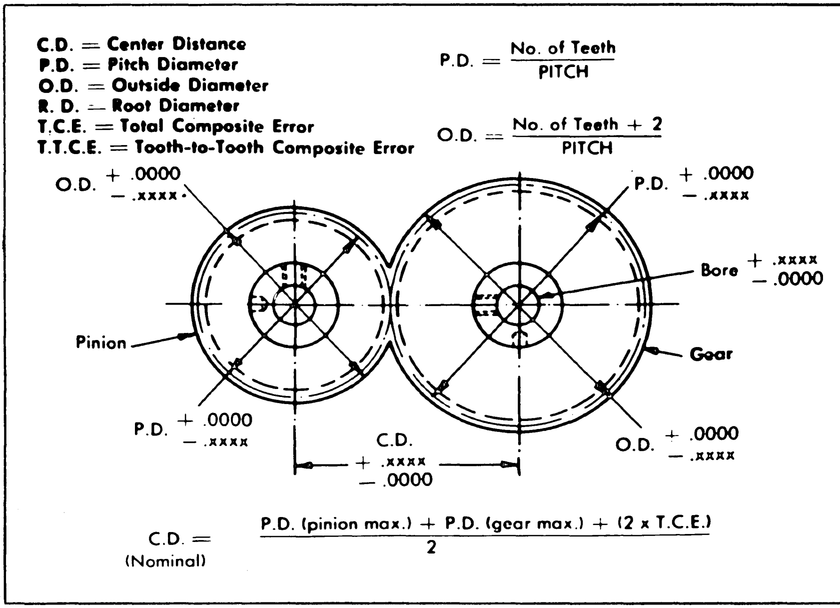


FIGURE 1-17 *Jig Boring Matched Plates*

quently, it is recommended that the one hole be slightly larger for ease of removal. The dowel pin should have a drive or interference press fit in the fixed or permanent member and loose fit in the removable piece. (See Figure 1-15.)

The size and length of the dowel pin is usually governed by the application. The general rule is that the dowel pin should be about the same diameter as the holding screws and length at least two times the diameter in each member.

Never depend on just dowel pins for holding units together; screws should be used for fastening the work permanently together.



RECOMMENDED TOLERANCES

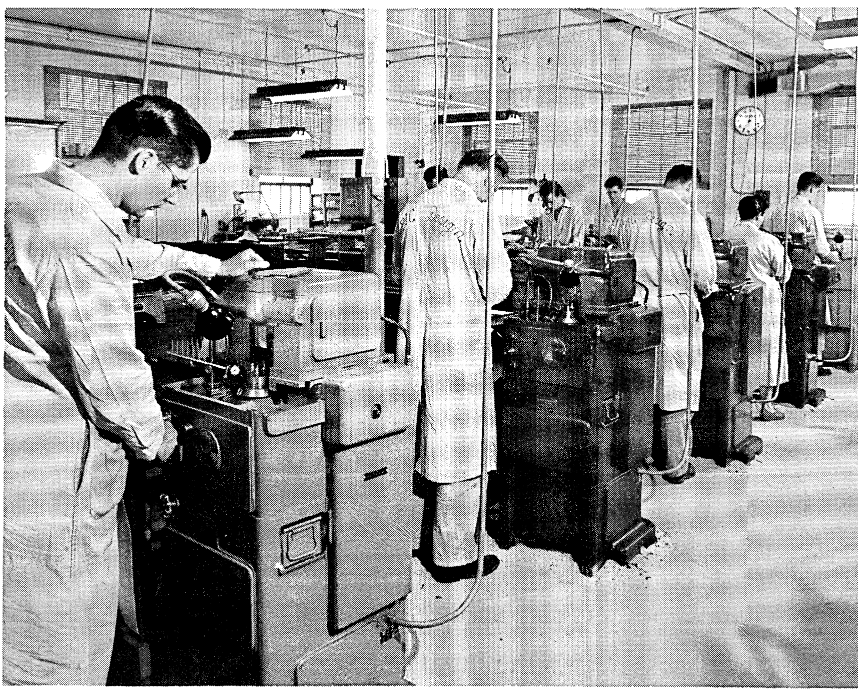
Class Gear	P.D. Tol.	O.D. Tol.	Bore Tol.	C.D. Fixed Centers	C.D. Adjustable Centers	Side Wobble at 1" Rad.	O.D. Concentricity to Bore
Prec. 1	−.001	−.002	+.0005	+.0005	+.002	.001	.0005
Prec. 2	−.0007	−.0015	+.0003	+.0003	+.0015	.0007	.0003
Prec. 3	−.0005	−.001	+.0002	+.0002	+.0012	.0005	.0002
Ultra Prec. 1	−.0004	−.0008	+.00015	+.00015	+.001	.0004	.00015

FIGURE 1-18

Chapter Two

SPUR GEARS

The most commonly used precision instrument components are spur gears. They increase or decrease the speed of one shaft with relation to another, and consequently are a means of advantageously increasing or decreasing the value of a given factor as represented by the rotation of a given shaft.



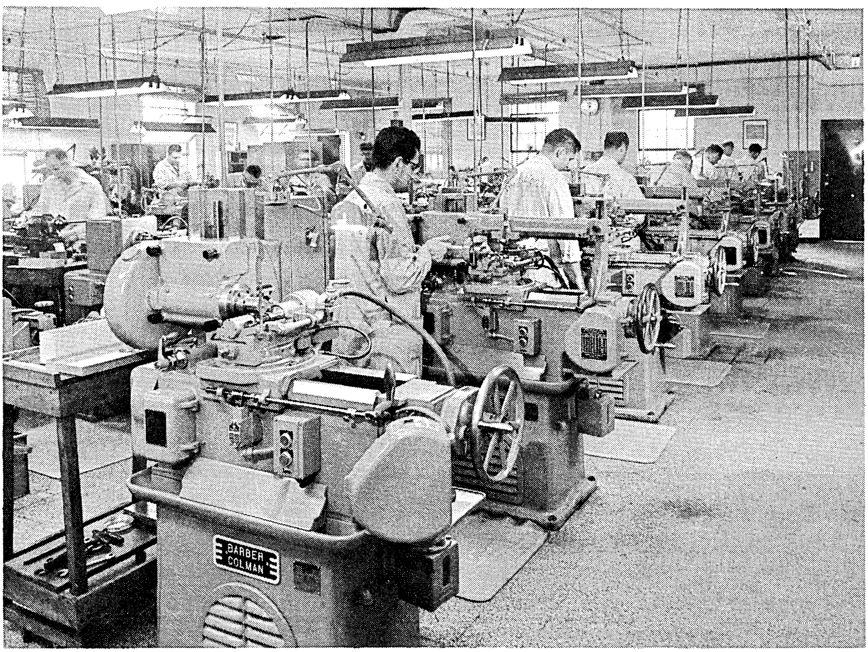
Bank of Fellow Spur Gear Shapers

Spur gears are usually produced by either the hobbing or shaping method.

Using a small gear on a hand-crank shaft to turn a large gear on a dial shaft will make the dial turn more slowly than the hand crank. Since slight turns of the hand crank will result in even slighter movements of the dial, more accurate settings can be made than if the dial turned in direct ratio to the hand crank. (See Figure 2-1.)

The value of one revolution of the input shaft of one mechanism may be different than the value of one revolution of the output shaft of the mechanism that is feeding it. By connecting the input and output shafts with the proper gears of different diameters or numbers of teeth the output of one mechanism can be adjusted to the input of the other. (See Figure 2-2.)

Many electrical devices operate efficiently only at comparatively high speeds, and each revolution of the shafts of these devices has a low value. In feeding information from these devices to other mechanisms, the shaft speeds are reduced and the shaft values increased



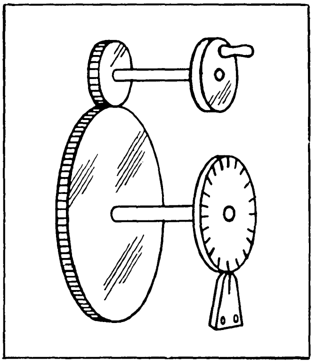
Bank of Barber Coleman Spur Gear Hobbers

by gear ratios. Conversely when feeding information to an electrical device from a mechanical device, the speed of the output shaft is usually increased and its value decreased by gear ratios. (See Figure 2-3.)

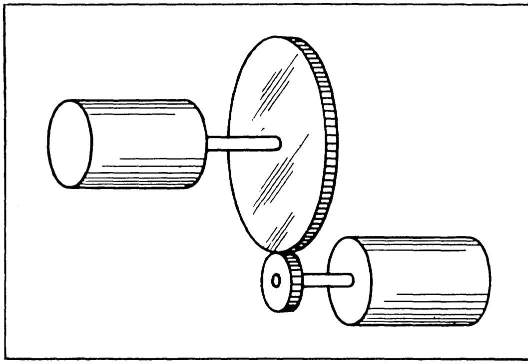
When a large increase in the number of revolutions is required, the increase can be made in several steps, using intermediate shafts, each of which carries two different size gears. For example, suppose a 12:1 ratio is needed between two shafts, as shown in Figure 2-4.

To do this in one step would require the driving gear to be 12 times as big as the smaller gear. This would be inconvenient, wasteful of space and would introduce additional gear system errors, since it is much easier to make small close-tolerance gears than to make large ones which have many errors such as eccentricity, side wobble, tooth to tooth errors, etc.

A more feasible approach is shown in Figure 2-5. Gears A and B have a 2:1 ratio. For each turn of driver A, gear B makes two revolutions. Since B and C are on the same shaft, gear C also turns twice



Spur Gear Reduction System
FIGURE 2-1



Differing Speed Requirements
FIGURE 2-2

for one turn of gear A. Gears C and D have a 3:1 ratio. Gear D turns three revolutions for each turn of C or (3x2=) six revolutions for each turn of A. Gear E is on the same shaft as Gear D; E also turns six times for one turn of A. Since E and F have a 2:1 ratio, F turns twice for each turn of E, or (2 x 6 =) 12 times for each revolution of the driving gear A. The ratio between A and F is, therefore, 12:1, and

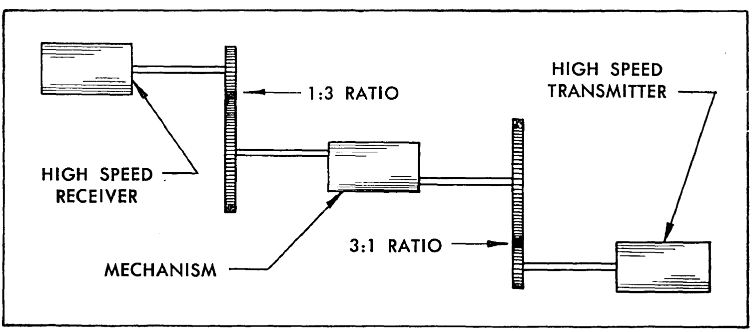


FIGURE 2-3 *Mechanical Coupling Between Two Electrical Devices*

this ratio is achieved without the use of large gears.

To find the ratio of a train of this kind, multiply together the ratios between each pair. The ratio in this example is: 24/12 x 30/10 x 20/10 = 12/1. Other gear-selection factors such as bore, face, pitch, material, load, etc., will be discussed later in this chapter. See recommendations Figure 2-6.

TYPES OF SPUR GEARS

Precision instrument spur gears which might be used in such a gear train are classified as hubless, split hub, pin hub, cluster and anti-backlash types, as shown in Figure 2-7.

12:1 Ratio — One Pass

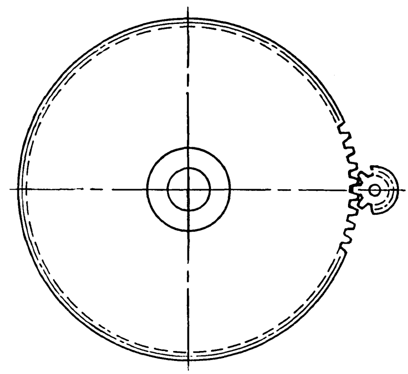


FIGURE 2-4

HUBLESS GEARS

The hubless gear is the least expensive type and also offers space-saving advantages when used with cluster gears and other applications (see Figure 2-8). Hubless gears have .3750″ bores, are press-fit and are generally staked to hubs or cluster gears to order by the manufacturer. Giving a wide variety and selection of bores, hub styles

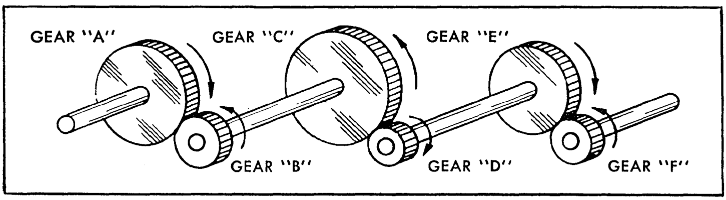


FIGURE 2-5 *Gear Train for 12:1 Ratio — 2:1 × 3:1 × 2:1 = 12:1*

RECOMMENDATIONS

(1) Mesh stainless pinions with aluminum gears.

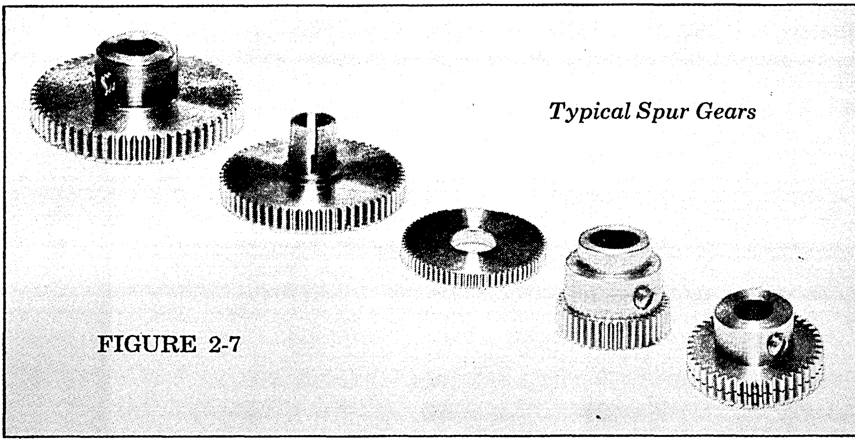
(2) We recommend in selecting pitches and shaft sizes to follow the below sequence if possible, for a reliable and efficient gear design and assembly.

Input Pinion—Fine Pitch—Example—96 Pitch—Shaft Size 1/8
Next Mesh—Example—96 Pitch—Shaft Size 1/8
Next Mesh—Example—72 Pitch—Shaft Size 3/16
Next Mesh—Example—72 Pitch—Shaft Size 3/16
Next Mesh—Example—64 Pitch—Shaft Size 1/4
Next Mesh—Example—64 Pitch—Shaft Size 1/4

Always reduce pitch and increase shaft size as ratio and torque increase

FIGURE 2-6

and number of teeth. They may also be secured to hubs by means of lock nuts by the user. Hubless gears are less expensive than hub gears since they are made from discs cut from bar stock. Hub-type gears involve the waste of stock in turning down from the gear's



Typical Spur Gears

FIGURE 2-7

outside diameter to hub diameter. The value of this material is greater than the cost of assembling a hubless gear to a hub or to a cluster gear. This is illustrated in Figure 2-9.

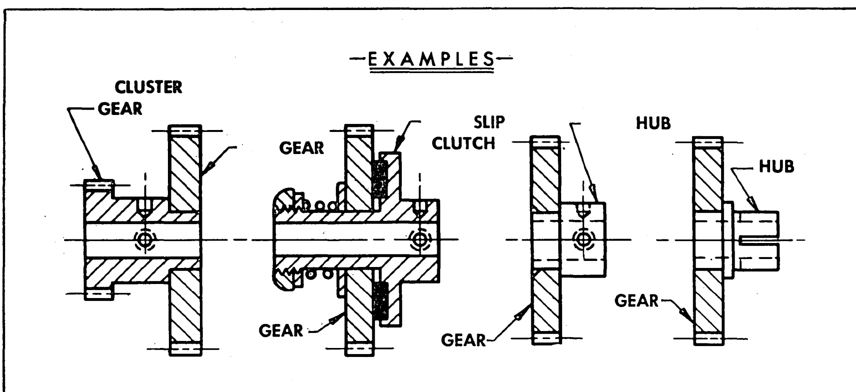


FIGURE 2-8

THREE-NOTCH STAKING

Three-notch staking of hubless gear to a hub or cluster gear is recommended since it keys the gear and prevents it from turning on the hub and also prevents the gear from working off the end of the hub. In using this method, as shown in Figure 2-10, three notches are cut in the face of the center hole of the gear, parallel with the gear teeth. The gear is then assembled with the hub and hub material forced into the notches. Unless the user has the special tools required,

HUBS

The two basic hubs used with hubless gears are the *pin hub* and *clamp hub.* Each basic type is available in two styles, one of which enables the gear to be secured with a lock nut, the other being designed for three-notch staking or dutch pinning. (See Figure 2-12.) These hubs are available in a wide variety of bores and materials.

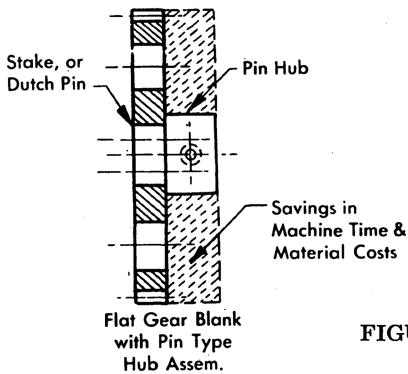


FIGURE 2-9

it is recommended that staking be done by the supplier when ordering gear and respective hub or cluster gear.

DUTCH PINNING

Dutch pinning, as shown in Figure 2-11, involves assembling of gear and hub, drilling a common dowel hole and pinning with a

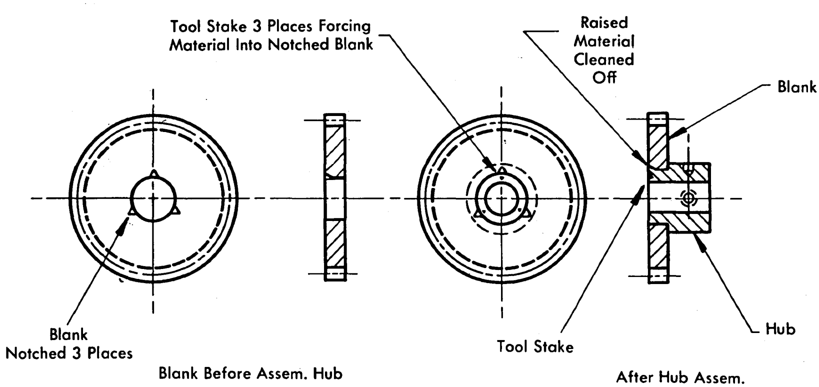


FIGURE 2-10 *A Hubless Gear Should Be Staked at Three Places to the Hub*

drive-fit dowel pin. This method is not always effective in preventing the gear from working off the hub, Thus if dutch pinning is required it should be in addition to three-notch staking to secure the gear and also to prevent it from working off the hub.

PIN HUB

The pin hub has a set screw, which temporarily secures the hub to the shaft, and a guide or sub drill hole for drilling hub and shaft for final taper pin or roll pin assembly. Pinning has a disadvantage in

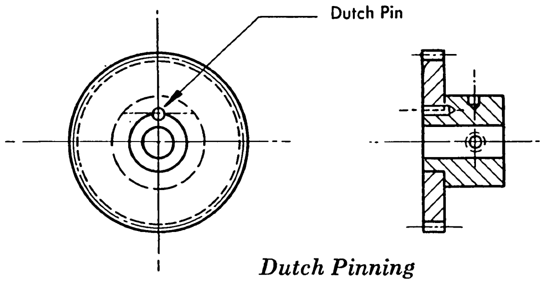


FIGURE 2-11

Dutch Pinning

that it weakens the shaft. Also, it is impossible to remove the hub from the shaft without damaging the shaft, the hub and its gear, or both. After pinning, the set screw is removed so it cannot later work itself loose and fall into the mechanism.

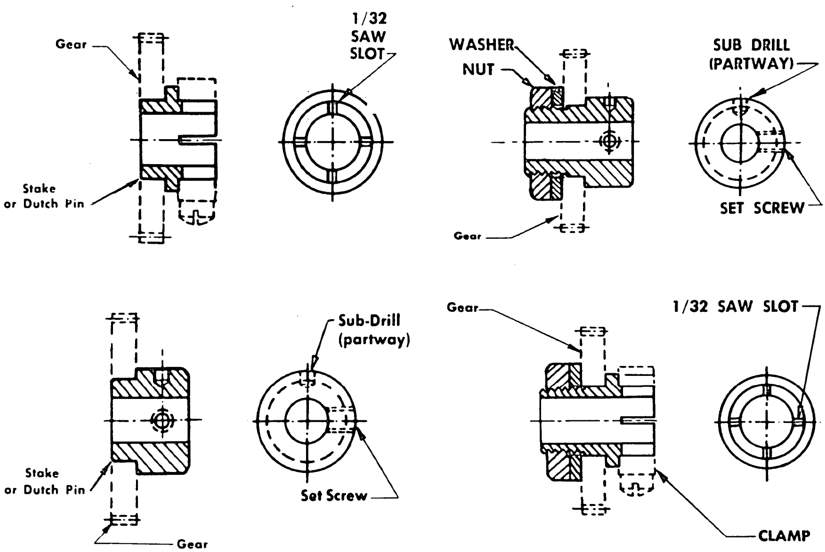


FIGURE 2-12

Pin Hubs and Clamp Hubs Are Two Basic Types Used With Hubless Gears

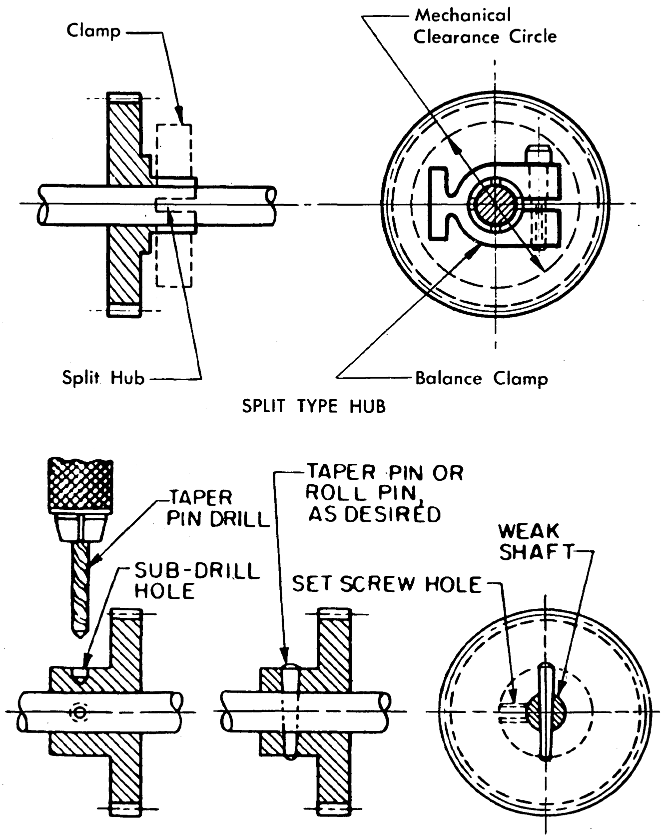
SPLIT OR CLAMP HUB

One end of the split hub is slit axially so that it may be secured to the shaft with a standard hub clamp. This type is easier to assemble and disassemble than the pin-hub type. It does not damage or weaken the shaft and makes it easier to adjust the gear on the shaft.

However, it takes up more space, introduces extra parts, extra weight and moments-of-inertia, which may not be desirable.

PIN AND CLAMP-TYPE GEARS

Pin-hub and clamp-type gears are also available with hubs integral with the gears, as shown in Figure 2-13. These types provide greater precision than the hubless type.



Integral Hubs of Pin Hub and Clamp-Type Gears

FIGURE 2-13

PINNING – TAPER

When taper pinning a gear, collar or coupling to a shaft is required, the hub usually has a spot or sub-drill hole in the hub to act as a guide or center punch for the pre-drill operation. Usually this drill size should be equal or slightly smaller than the small end of the taper pins.

Up to about a 1/8 drill, this is done in one drill operation. Then taper is reamed for the pin when shafts are larger than 1/4″. It is recommended that step pre-drilling be done (see Figure 2-14).

This drill operation is followed up with a taper pin reamer and the taper pin is then driven home. It is also recommended the top of the pin be staked to prohibit the taper pin from working loose under operation or by vibration.

PINNING – GROOVED PINS

A grooved pin is a cylindrical pin with longitudinal grooves raised in the cylindrical body to increase the diameter at these certain points. The pin is then driven in a straight drilled hole and the main advantage is that no reaming is required.

Grooved pins are resistant to loosening from shock and vibration and are easily removed and replaced without reducing the holding power.

Such pins are usually used for pinning gears or couplings to shafts. (See Figure 2-15.)

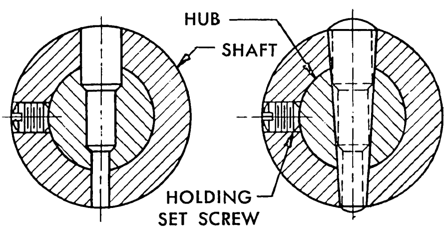
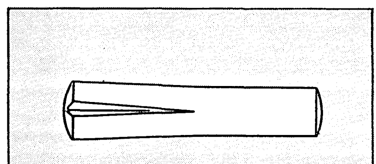


FIGURE 2-14



Typical Grooved Pin

FIGURE 2-15

CLUSTER GEARS

Cluster gears, as shown in Figure 2-16 are used in gear trains to allow large gear reductions with minimum space requirements. Essentially they are pin-hub spur gears with shoulders to which hubless gears are pinned.

The shoulder for the hubless gear may be next to the hub (Style A), next to the gear face (Style B), or between the gear and the hub (Style C). For continuous reductions, "A" style is meshed with "B" style, "B" with "C" style and the sequence is repeated. Always mesh stainless pinions with aluminum gears to eliminate possible galling effect of similar materials.

ANTI-BACKLASH GEARS

Backlash is generally explained as the play between mating teeth, as illustrated in Figure 2-17. It is the shortest distance between non-driving tooth surfaces of adjacent teeth in mating gears.

To help avoid backlash problems, anti-backlash gears, as shown in Figure 2-18, have been developed.

These gears have a *floating* gear member mounted alongside a fixed gear. The gears each have two through-slots perpendicular to a common diameter and on opposite sides of center. Coil springs are positioned in these matched slots, and one end is connected to the

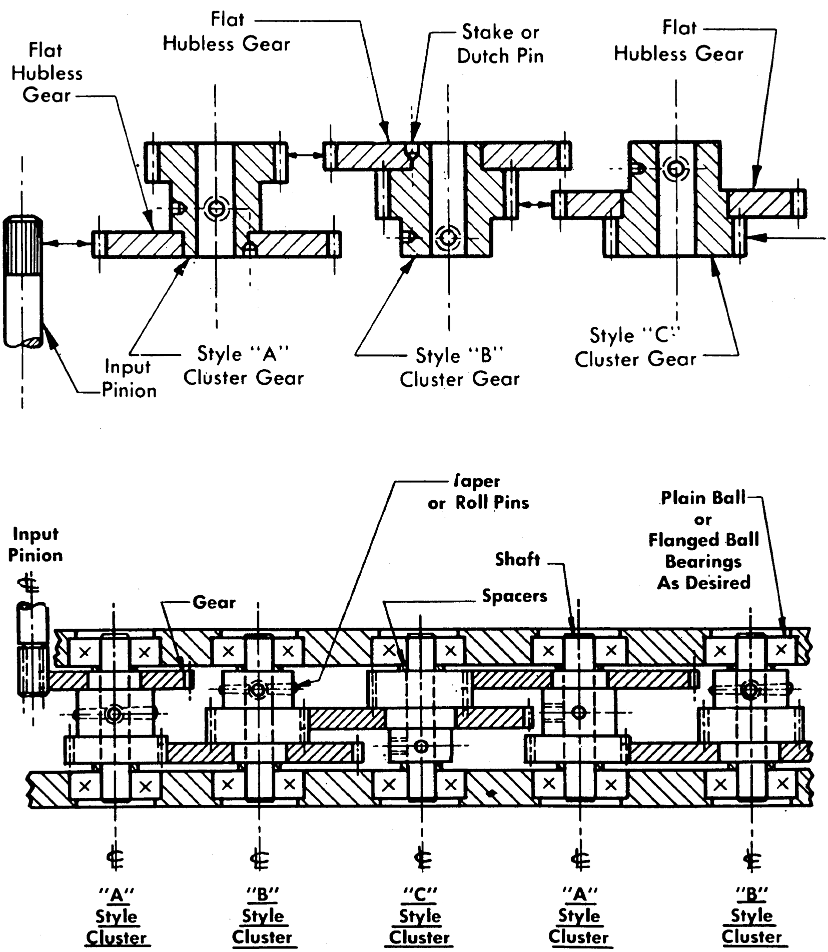


FIGURE 2-16 *Unique Cluster Gear Design System*

fixed gear and the other to the floating gear. As a composite anti-backlash-gear tooth engages a space of a mating gear, *spring action* takes up any backlash by pulling the non-driving face of the floating gear tooth tight against the non-driving face of the tooth with which it is in mesh.

MATERIALS

Hubless and pin-hub gears are made of aluminum, stainless steel, nylon and phenolic. Clamp-hub gears are made of aluminum and stainless steel. Cluster gears are made of stainless steel. Aluminum gears are anodized to protect them from corrosion and stainless steel

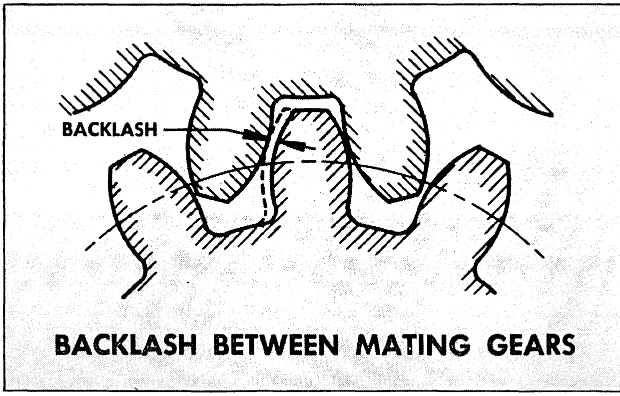


FIGURE 2-17

gears are passified after gear cutting to clean off all fine dust formed or impressed during machining operations.

In laying out gear trains, the smaller gear of a pair should be of stainless steel and the larger gear of aluminum to minimize noise and wear of materials.

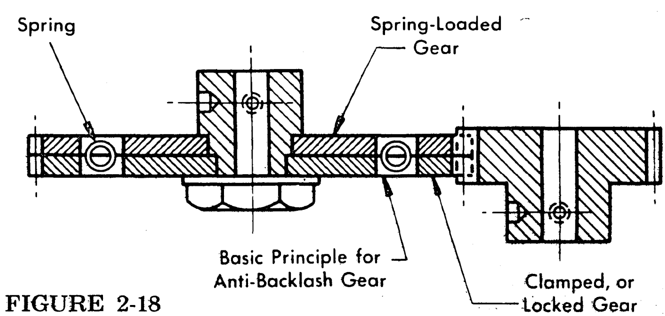


FIGURE 2-18

Anti-Backlash Gear and Standard Spur Gear in Mesh

The *nylon* gear has its most useful application as an *idler gear* (see Figure 2-19) or as a functional part of a slow-running system where silent operation is desired. Nylon expands and contracts with changes in the weather and, at high speeds, centrifugal force increases the outside diameter to the point where it will bottom on the root diameter of the mating gear.

Phenolic gears are used exclusively in systems requiring silent operation. They lack tooth-to-tooth strength and are limited to light load applications and slow speeds. Warm humid climates encourage a

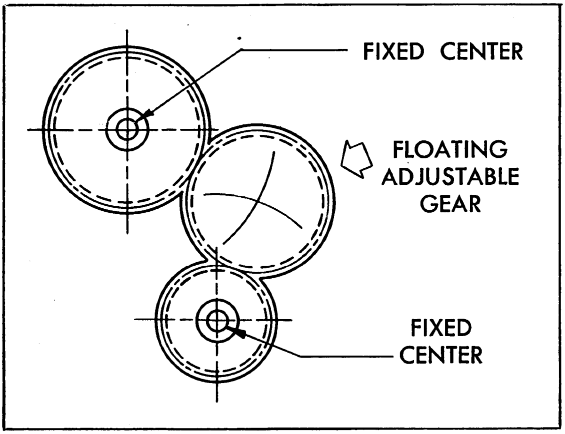
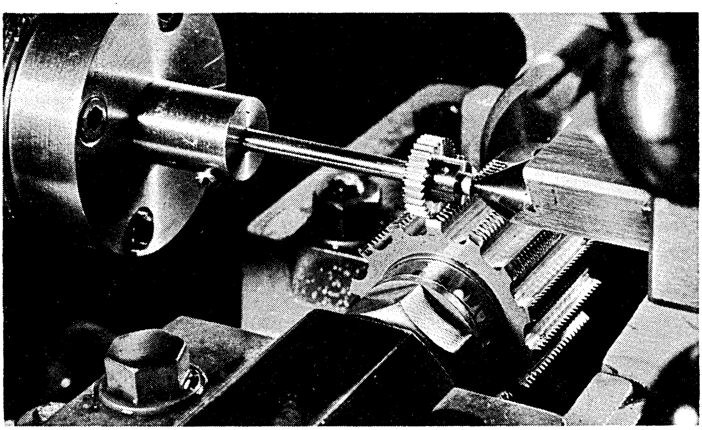


FIGURE 2-19

fungus growth on the material which hampers its use, unless operated under closely controlled or sealed conditions.

STANDARD PRECISION INSTRUMENT SPUR GEARS

Standard precision instrument spur gears are available in four degrees of precision, per Figure 1-14, and are available in *diametral*



Closeup of Spur Gear Being Hobbed

pitches of 24, 1/10, 32, 48, 64, 72, 80, 96, 120 and 200. *Diametral pitch* is the number of teeth divided by the diameter of the pitch circle, the circle formed by the points on the gear teeth which contact the teeth of the mating gear. The greater the pitch, the greater the number of teeth. (See Figure 2-20.)

The quality or microinch finish is most important in precise gearing. All gear teeth rub or slide against each other. When the teeth are in operation and rub, the rough surface will act as a file and

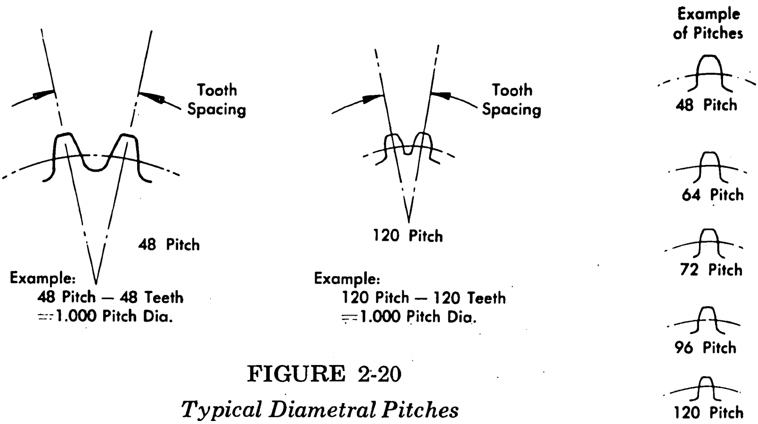
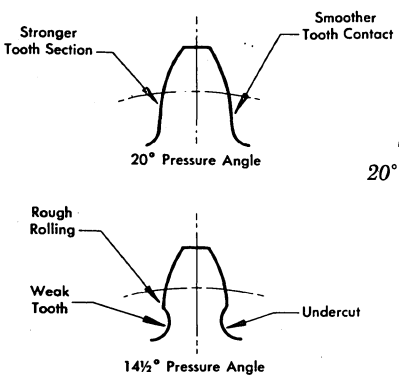


FIGURE 2-20

Typical Diametral Pitches

clean up the mating gear surface. Thus, the smoother the tooth finish, the longer the life and the smoother the tooth action.

Spur gears are also available in *circular pitches* of 1/10″ and 1/20″, this being the distance between corresponding points of adjacent teeth along the pitch circle. Circular pitches are sometimes desirable, since they provide a ready means of measuring motion on



Spur Gear Teeth With 20° and 14½° Pressure Angle

FIGURE 2-21

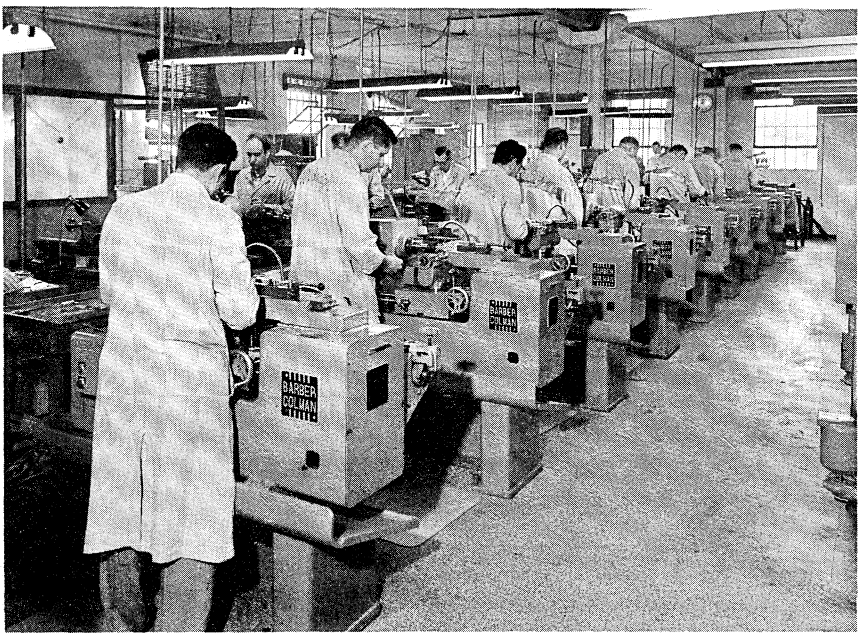
the basis of number of teeth advanced. Gear face thicknesses are 1/16″, 3/32″, 1/8″ and 3/16″, depending on other gear dimensions.

Standard precision instrument spur gears are made with 20° pressure angles, as shown in Figure 2-21 for greater tooth strength. *Pressure Angle* is the angle of the tangent-to-the-tooth at the pitch circle, with reference to the radius through the center of the gear.

The 14-1/2° pressure angle tooth is gradually becoming obsolete with a trend toward and demand for finer pitches. The 14-1/2° pressure angle tooth does *not give as smooth* a rolling or running gear mesh as the 20° pressure angle gear and has a tendency to stick in the undercut recess. The resultant hesitancy causes an unsteadiness in the overall gear train. The 14-1/2° pressure angle does have one advantage, however, in that it has slightly less backlash than the conventional 20° pressure angle gear.

HOW TO SELECT PROPER SPUR GEARS

The choice of which precision classification to specify depends on the accuracy with which motion must be transmitted from one device to another. The precision required to transmit motion accurately

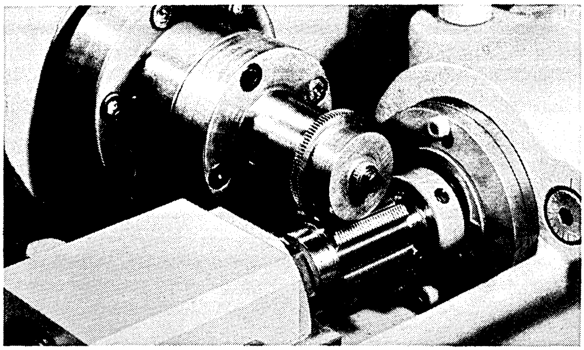


Bank of Precision 3 Spur Gear Hobbers

within 5 min. of arc is obviously much greater than that required to transmit motion within 30 min. of arc. Therefore, gears with a minimum of backlash, or those made to the highest degree of precision, will be selected where high accuracy is required in relaying motion from one device to another.

Backlash is necessary in order to smooth gear operation. If each tooth were exactly the same size as the space with which it meshed, there would be an interference fit and gears would bind, wear, and not run smoothly. The slight unavoidable manufacturing variations from nominal dimensions constitute one factor which contributes to backlash. These are:

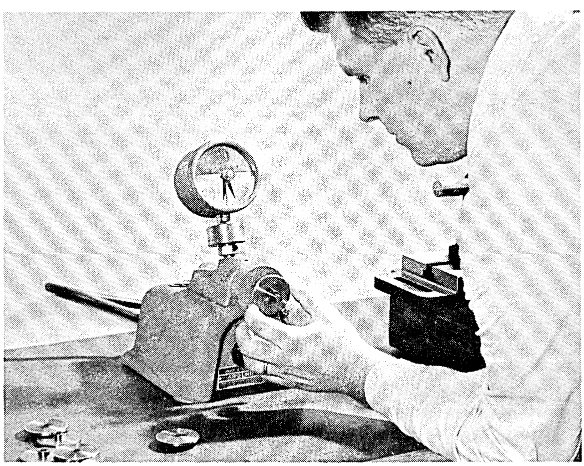
1. Variations in *pitch diameter* or size of gear.
2. Variations of *center hole size* and location.
3. Variations from one *tooth* to another.



Closeup – Prec. 3 Spur Gear Being Hobbed

Radial play and eccentricity of ball or bronze bearings are two other factors contributing to backlash. Assembly factors constitute another group of potential contributors to backlash. These include:

1. *Shaft fits* to bearing bores.
2. *Bearing fits* to housings.



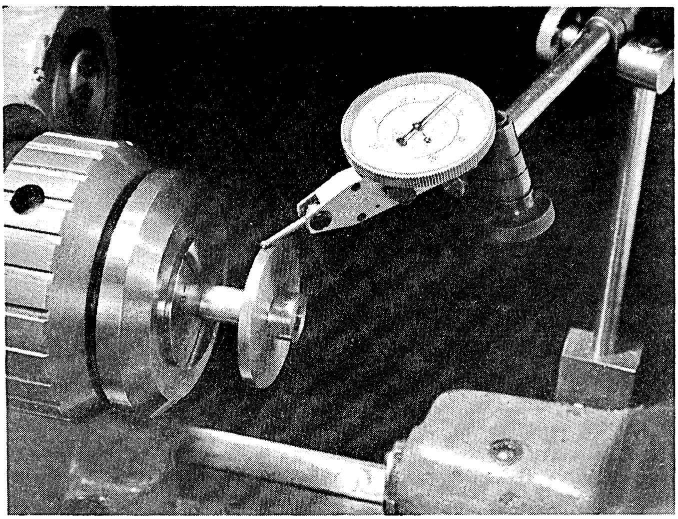
Air Gauging Gear Blank Bores within .0001"

3. Tolerance variations in *distances* between centers of holes for bearings carrying shafts of mating gears.

From the point of view of gear selection, backlash due to manufacturing tolerance variations can be calculated using data provided by the gear manufacturer.

DIAMETER VARIATIONS

The gear blank is turned to its nominal diameter, plus or minus the specified tolerances (see Figure 1-14). In the precision gear field, each gear blank must be checked by the manufacturer to make sure the diameter is within tolerance limits for the precision classification for which it is intended. This closely held bore and outside diameter of the gear blank is used as an indicator point and as a reference in the gear hobber to maintain the pitch diameter concentricity, and size in relation to the bore (see Figure 2-22).



Turned Blank Being Checked for Concentricity

CENTER HOLE AND TOOTH-TO-TOOTH ERROR

For purposes of calculating backlash, center hole and tooth-to-tooth variations are considered in one factor known as "total composite error," the magnitude of which depends on the precision classification of the gear.

Gear Class	*Total Composite Error*
Prec. 1	.00100
Prec. 2	.00050
Prec. 3	.00025
Ultra-Prec. 1	.00020

Total composite error may be determined by meshing and rotating the gear under test with a master gear of known accuracy. The master gear is mounted on a fixed shaft and the test gear is mounted on a shaft which moves toward or away from the master-gear shaft, depending on center hole and tooth-to-tooth variations in the gear under test.

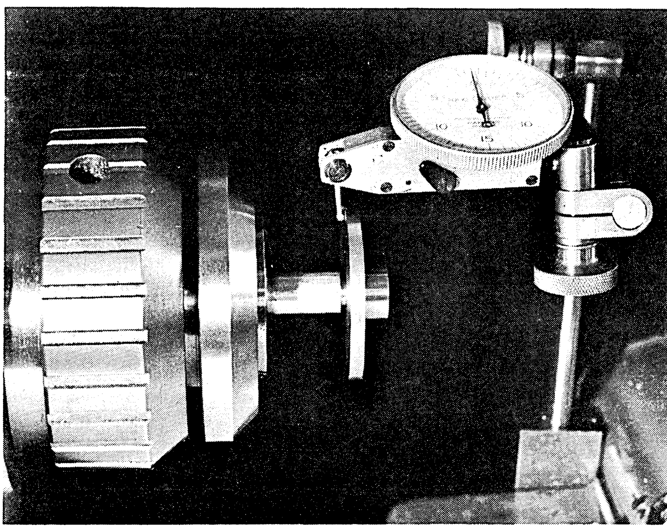
The motion of the movable shaft can be recorded and total composite error determined by examining the graph that is produced, an example of which is shown in Figure 2-23.

Total composite error – the total of the forward and backward distances traversed by the movable shaft due to tooth-to-tooth and center hole variations – must be within limits established for the particular precision classification for which the gear is intended. (See Figures 2-24 & 2-25.)

To calculate backlash in a given pair of gears, the gears are first assumed to be made to maximum tolerances and then to minimum tolerances. Center distances between gears are determined on the basis that each gear is to maximum tolerance diameter and has the largest allowable total composite error. Under these conditions, distance between centers would be:

$$\text{C.D.} = \frac{\text{PD max. (pinion)} + \text{PD max. (gear)} + 2\ \text{T.C.E.}}{2}$$

If gears mounted at this center distance are actually to minimum tolerances, then there will be a small space between the mating gears. This distance can be calculated by finding the center distance which would be used if both gears were to minimum tolerances, and sub-



Turned Blank Being Checked for Side Wobble

tracting this from the actual center distance, based on gears to maximum tolerances.

This is one element used in calculating backlash – the maximum space which might exist between the non-driving faces of meshed gear teeth along the pitch diameter. Another element is the pressure angle of the gear tooth. Without going through its derivation, the accepted formula for backlash is:

Backlash = Change in center distance x (2 x Tan of the pressure Angle).

For 20° pressure angle gears this would be:

Backlash = Change in center distance x (2 x Tan 20°)

= Change in center distance x 2 x .36397

= Change in center distance x .72794

Having found this value of backlash, a ratio can be made to deter-

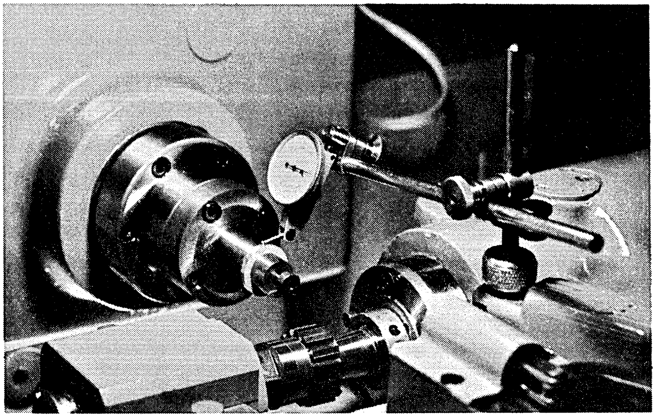
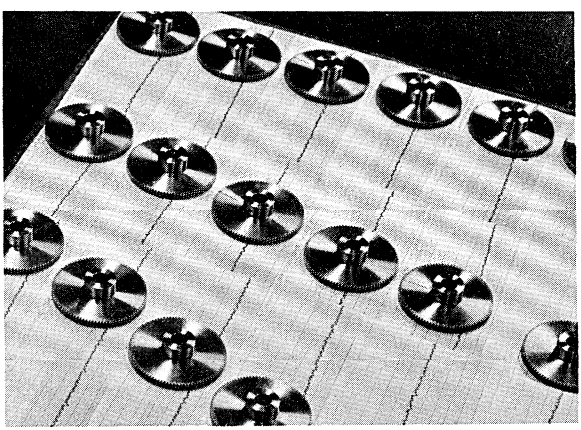


FIGURE 2-22 *Gear Blank Being Checked for Concentricity Before Gear Cutting*

mine the degree of arc error it represents:

$$X^\circ \text{ of arc} = \frac{\text{pitch circumference}}{360^\circ \times \text{backlash}}$$

In a given gear train it is necessary to add the backlash in each

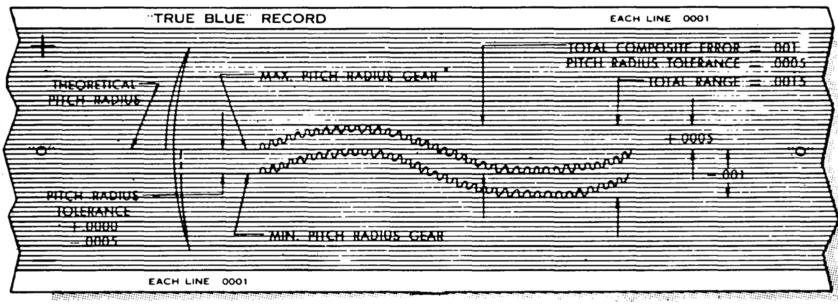


Precision Gears With Certified Tapes

pair of gears to get the total backlash of the train. If the calculated degree of arc is within tolerances, then the selected precision classification may be used. If not, a higher degree of precision may be required. It may also be possible to go to a lower, and less expensive classification. It can also turn out that the degree of precision desired is not attainable, in which case some other method must be investigated or the system accepted as the best available at the time.

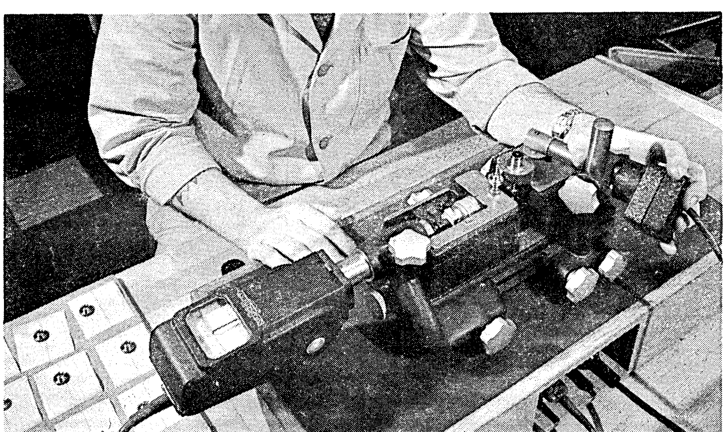
WORN OR CHEWED UP GEAR TEETH

The largest single problem in the precision gear field is *worn or chewed up gears.* Many designers and engineers use gears of known



Graphic Recording of Tooth Action and Total Composite Error
FIGURE 2-23

and established pitch diameter on nominal center distance, but do not put into the center-distance formula the required manufacturing tolerance variations such as eccentricity, center to center or jig boring tolerances, tooth-to-tooth errors. This is classified as TCE *(total*

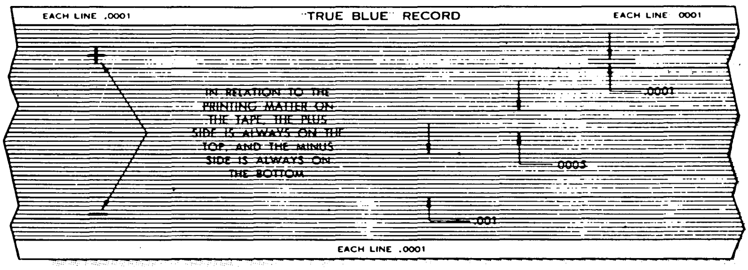


Gears Being Checked and Recorded to Verify Accuracy

composite error). If *not* included in the center distance formula, at some point in mesh the gears will bind causing damage or wearing of the teeth. The usual solution is to go immediately to harder gears or change the basic material. This will not eliminate the mechanical interference. Thus, the solution is to allow for all possible interferences in the center distance formulae. (See Figure 1-17.)

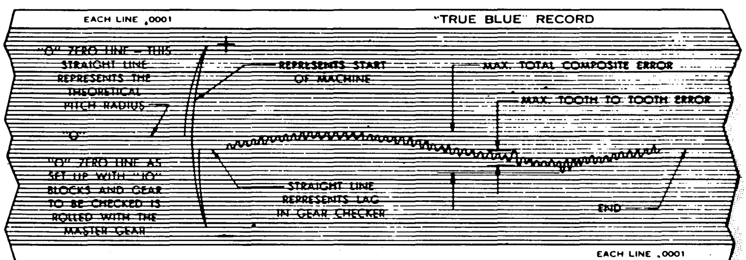
How To Read a "True-Blue" Tape

SAMPLE "TRUE-BLUE" TAPE SHOWING VARIOUS RANGES OF TOLERANCES



Standard "True-Blue" Tape

SAMPLE "TRUE-BLUE" TAPE SHOWING VARIOUS PHASES OF RECORDING



Sample Gear Recording

GENERAL INFORMATION:

All readings on tape are based on pitch radius.

- Gears can fall anywhere within the max. and min. limits indicated — but in no instance exceed total composite error.
- P.D. = Pitch Diameter.
- Theoretical Pitch Radius Formula:

$$\frac{\text{P.D. (Master Gear)} + \text{P.D. (Gear to Be Checked —Max.)}}{2}$$

Length of tape not critical — depends on speed of gear checker and diameter of gear.

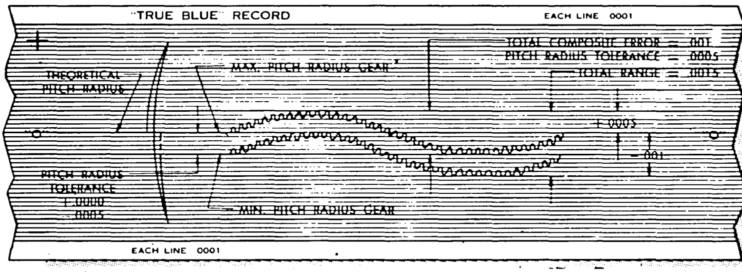
Many customers check P.D. "over wires." Unfortunately, this is not exact or precise enough and can only be considered approximate. True P.D. can only be verified on a gear checker against a certified master.

- All Precision Gears are cut to pitch diameter and total composite error tolerances

FIGURE 2-24

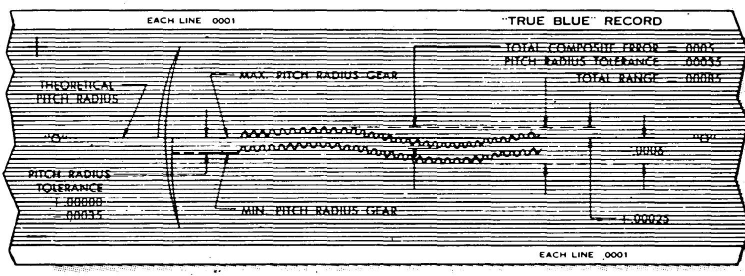
Precision 1, 2, & 3 Tape Comparisons

SAMPLE "TRUE-BLUE" TAPE SHOWING RECORDING OF PREC. 1 GEAR



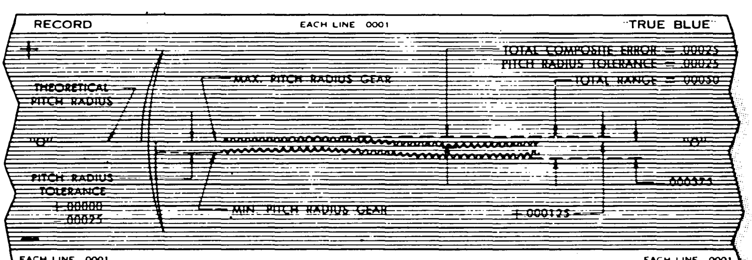
Precision 1 Tolerance Range

SAMPLE "TRUE-BLUE" TAPE SHOWING RECORDING OF PREC. 2 GEAR



Precision 2 Tolerance Range

SAMPLE "TRUE-BLUE" TAPE SHOWING RECORDING OF PREC. 3 GEAR



Precision 3 Tolerance Range

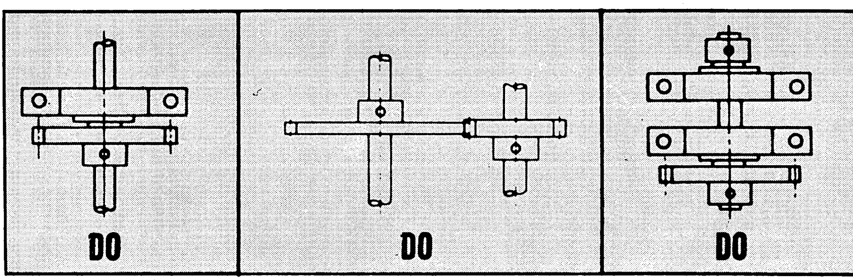
NOTE— "True-Blue" Gear Tapes are supplied with all Prec. 2, 3, or Ultra Prec. 1 gears at no additional charge; and are available with Prec. 1 gears on request, at additional charge based on quantity involved.

FIGURE 2-25

DOs and DON'Ts of MECHANICAL ASSEMBLY

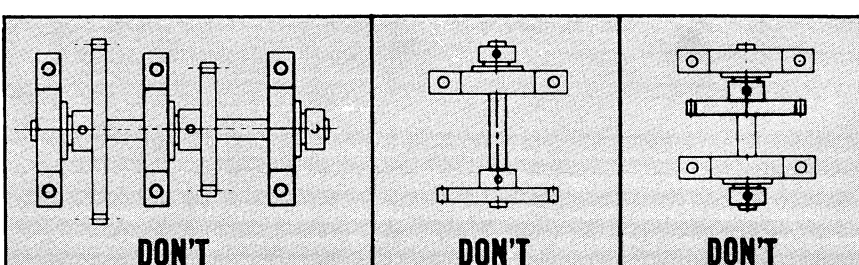
DO

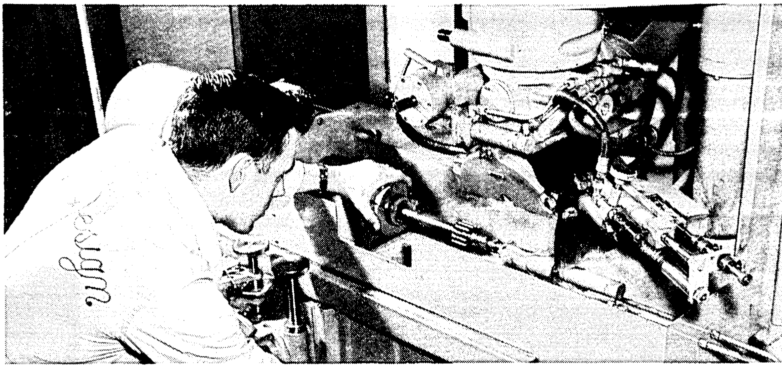
- assemble gears so that face is as close to bearings as possible—to eliminate bearing loading and overhang.
- mesh gears, wherever possible, with different face widths—1/8 to 3/16 or 3/16 to 1/4 or mismesh faces of same width to eliminate possibility of burrs hitting.
- alternate hubs in gear meshes so that possible burrs are opposite each other.
- always preload bearings in opposite directions.
- preload bearings, wherever possible, with spacer and component such as gear or coupling. Use collar only when necessary.



DON'T

- use one or three bearings to support a gear shaft. Always use two bearings.
- allow too great an overhang on extended shafts.
- mesh gears of similar material when high speeds are involved. Use dissimilar materials to eliminate galling. Example: Mesh stainless pinion with aluminum gear.
- depend on set-screws for permanent locking; only for temporary or setting positions.
- rework finished gears—only basic blanks. Gear cutting should be final operation.





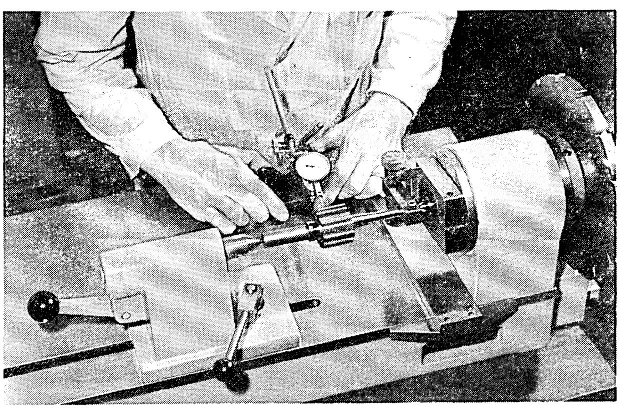
Hob Grinding

HOB & CUTTER CONTROL

One very important control of good quality gears is in the grinding and controlled sharpening of gear hobs and cutters. Many times improperly sharpened hobs and cutters introduce additional errors to the product.

SPECIAL GEARS

To eliminate the cost of detail drawings and calculation time for special, non-standard gears which must be made to order, a unique



Hob Inspection

standard numbering system has been developed for precision spur gears. It saves a tremendous amount of time and work. Many of the basic gear blanks are usually stocked by gear manufacturers. Thus the purchaser can enjoy considerable savings in unit cost and earlier delivery time.

The following specifications and standards are for "STD." custom-made Certified Military type fine pitch spur gears.

GEAR SPECIFICATION

Spur, Fine-Pitch

1. This specification covers requirements for 24, 32, 48, 64, 72, 80, 96, 120, and 200 diametral pitch spur gears having a 20 degree pressure angle.

*14-1/2° pressure angle gears are available by adding 14.5 to end of completed STD-number.

1.1 **Method of Designation**

1.1.1 **Basic Number** — The basic number of the STD Standard Part applicable to this specification consists of three letters STD and numerals per standard stock numbering specification. See 1.1.3

1.1.2 **Dash Number** — The dash number, which follows the STD — is separated from the basic number by a dash and consists of six numeral groupings designating type of gear, diametral pitch, numbers of teeth, gear blank characteristics, material and quality classification.

1.1.2.1 **Type of Gear** — The type of gear such as Pin Hub, Split Hub or Hubless gear is the first numeral grouping after the STD number—

"G" = Pin type hub
"H" = Split type hub
"J" = Hubless type hub

1.1.2.2 **Diametral Pitch Numeral** — The diametral pitch numeral is the second numeral group on the left and designates the diametral pitch according to the following schedule.

STOCK DIAMETRAL PITCHES	STOCK CODE DESIGNATOR
24	1
32	2
48	3
64	4
72	5
80	6
96	7
120	8
200	9

1.1.2.3 **Number of Teeth Numerals** — The third numeral group to the right of the diametral pitch numeral significantly identifies the number of teeth in a gear.

For example: 032 means 32 teeth
132 means 132 teeth

1.1.2.4 **Gear Blank Characteristic Numeral** — The basic gear blank characteristics are designated by the numeral to the right of the number of teeth numerals. It is the same numeral as given in the Gear Blank Numeral column of the Tables "G", "H" or "J" of STD-X-X-etc. in the detail specification. In effect, this numeral defines the bore size, hub diameter, face width, overall length, and other geometric characteristics shown in the detail specifications. The same gear blank characteristic numeral appearing in a part number of two gears having different diametral pitches or different numbers of teeth does not necessarily mean that the gear blanks are identical. The gear blank numeral identifies the geometry of the particular gear described by the preceding diametral pitch and number of teeth numerals shown in STD-X-X-etc.

1.1.2.5 **Material Numeral** — The fifth numeral from the left designates the gear material according to the following schedule:

STOCK MATERIAL	STOCK CODE DESIGNATOR
ALUMINUM	
2024-T42 Aluminum Alloy (QQ-A-268 Temper T42 or QQ-A-355B Temper T4) anodized before cutting of teeth per MIL-A-8625B	1
STAINLESS STEEL	
Corrosion Resistant Steel AISI Type 303 annealed MIL-S-7720 Comp 303 Cond A. Shall be passivated according to MIL-S-5002 after gear cutting	2
NYLON — MIL — 17091B — Type 1 May have stainless steel insert in bore to maintain stability	3
PHENOLIC — MIL-P-15035B — Type FBE Fungus-proofed before gear cutting, per Jan-T-152	4
BRONZE (Tobin) — MIL — B-994 a Compo A. Natural	5
***SPECIAL MATERIAL ALSO AVAILABLE**	
*Leaded Phosphor Bronze (ASTM B139 Alloy B1, Hard)	6
*2024-T42 Aluminum Alloy (QQ-A-268 Temper T42 or QQ-A-355B Temper T4)	7
*2024-T42 Aluminum Alloy (QQ-A-268 Temper T42 or QQ-A-355B Temper T4) anodized after final machining per paragraph 3.7 of this specification	8
*Corrosion Resistant Steel AISI Type 416 (QQ-S-00763A Class 6 Type A). May be heat-treated to 125,000 to 140,000 psi to improve wear, shall be passivated according to MIL-S-5002 after final machining	9

*Blanks not stocked, made to order.

1.1.2.6 **Quality Classification Numeral** — The quality classification numeral is the **sixth** numeral in the dash number and designation, the quality classification of the gear in accordance with Gear Standards as outlined in STD-XXX and SPEC-100

GEAR QUALITY CLASSIFICATION	DESIGNATOR
Class, Precision 1	P1
Class, Precision 2	P2
Class, Precision 3	P3
Class, Ultra Prec. 1	UP1

1.1.2.7 **Pressure Angle** — The pressure angle will be assumed to be 20° P. A. unless otherwise specified for 14½° P. A. add —14.5 to end of STD number.

1.1.3 **SAMPLE DESIGNATION**

STD — G — 4 — 056 — 02 — 2 — P3

- STD — Basic Number
- G — Type Gear "G" = Pin Hub
- 4 — Diametral Pitch—64
- 056 — Number of Teeth—56
- 02 — Blank Characteristics per Table "G"—"H" or "J" STD-X-X-etc.
- 2 — Corrosion Resistant Steel AISI Type 303 annealed (MIL-S-7720 Comp 303 Cond A). Shall be passivated according to MIL-S-5002 after gear cutting.
- P3 — Class, Precision 3

2. **APPLICABLE SPECIFICATIONS AND STANDARDS**

2.1 The following specifications and standards of the issue in effect on the date of invitation for bids form a part of this specification to the extent specified herein.

MILITARY

MIL-S-5002	Surface Treatments (except printing and painting) for metal and metal parts for Aircraft.
MIL-S-7720	Steel, Corrosion Resistant (18-8) Bars and Forging Stock (Aircraft Quality).
MIL-S-7742	Screw Threads, Standard, Aeronautical.
MIL-A-8625	Anodic Coatings for Aluminum and Aluminum Alloys.
MIL-STD-105	Sampling Procedures and Tables for Inspection by Attributes.

FEDERAL

QQ-Q-268	Aluminum Alloy 248; Bars, Rods, and Wire-Rolled or Drawn.
QQ-A-355	Aluminum Alloy (2024), Plate and Sheet.
QQ-S-00763	Steel Bars, Shapes and Forgings—Corrosion Resisting.

COMMERCIAL

A.S.T.M. B139 Leaded Phosphor Bronze

3.1 **Requirements**

Bore Size — The basic bore size shall be as specified in the tabulated gear blank dimensions. (Table "A" of STD.) Bore tolerances shall be in accordance with the gear classification as noted in the tabulated tolerances. (See Table "A" of STD.)

3.1.1 **Bore Size of Anodized Aluminum Blanks**
Bores may be plain or anodized. Basic tolerances must prevail under either condition.

3.1.2 **Bellmouth of Bore** — Bellmouth of bore is an extreme taper occurring at the ends of the bore. The bellmouthed portion must not exceed 20 percent of the overall length of the bore.

3.2 **External Corners** — All external corners except those adjacent to the contact surface of the gear teeth, shall be broken either by chamfer or radius. The amount of broken edge shall not exceed 20 percent of the addendum in either dimension normal to the surface.

3.3 **Deburring** — All gears shall be deburred. Burrs visible under a 10 power magnification shall be cause for rejection. Edges adjacent to the contact surface of the teeth may be broken a maximum of 0.015 inches.

3.4 **Surface Roughness** — The roughness of the surface finish of the working surface of the gear teeth and the bore shall be as smooth or smoother than that shown for the indicated class in accordance with the values shown in the tabulated tolerances of (Table "B" of STD).

3.5 **Hubs**

3.5.1 **Assembled Hubs** — Gears may have integral or assembled hubs, Stainless Steel #303. Where assembled hubs are used, they shall be rigidly fixed to the gear by notching, staking, or other suitable technique capable of meeting the requirements of 4.3.5.

3.6 **Anodizing** — When specified by material call-out designator 2 (paragraph 1.1.2), aluminum gears shall be provided with a chromic-acid anodic coating (anodized) after final machining. All specified tolerances apply after anodizing, except for bore tolerances. (See 3.1.)

3.7 **Masking of Bores** — The gear bore and set screw hole may or may not be masked during anodizing, at the discretion of the manufacturer. However, tolerances apply after anodizing.

3.7.1 **Workmanship** — The anodic coating shall be continuous, smooth, adherent, uniform in appearance, and shall be free from powdery areas, discontinuities such as breaks and scratches, or other damage. The size and number of contact marks shall be at a minimum consistent with good practice. Chromic-acid stains, although objectionable, shall not be cause for rejection unless they appear on working surface of the teeth.

3.7.2 **Rework** — Any gear anodized after final machining which is rejected may be reworked or re-anodized.

3.8 **Marking** — When practical, the sides of the gear shall be marked for identification with STD or manufacturers No. or shall include the part number according to 1.1.1 and 1.1.2.

4. **QUALITY ASSURANCE PROVISIONS**

Standard Test Conditions -- Unless otherwise specified, all tests and measurements required by this specification shall be made at a temperature of approximately 65°F to 85°F. Where tests are made at a temperature different from the above value, correction shall be made for the change in reading of instruments or gages due to the difference.

4.2 **Government Source Inspection** — Shall prevail at source or at destination as specified by purchaser.

4.2.1 **Sampling Procedure** — Sampling shall be in accordance with MIL-STD-105. The acceptable Quality Level shall be in accordance with the following schedule.

TEST	TABLE OF REQUIREMENT PARAGRAPH	DEFECT CLASS	AQL
Pitch Diameter	Table "A"	Critical	10%
Total Composite Error	Table "A"	Critical	10%
Bore Diameter	Table "A"	Critical	10%
Tooth-to-Tooth Error	Table "A"	Major	1%
Side Wobble	Table "B"	Major	1%
Surface Finish of Bore	Table "B"	Major	1%
Surface Finish of Tooth	Table "B"	Major	1%
Overall Length	Tables "G", "H", "J"	Major	1%
Torque Test, Assembled Hubs	3.5.1	Major	1%
Deburring	3.3	Major	1%
Taper of Bore	3.1.2	Major	2.5%
Outside Diameter	Table "A"	Major	2.5%
Face Width	Tables "G", "H" or "J"	Minor	1%
Anodizing, Workmanship	3.6	Minor	1%
Hub Diameter	Tables "G", "H" or "J"	Minor	2.5%
Marking	3.8	Minor	2.5%

4.3 **Test Methods.**

4.3.1 **Mechanical and Visual Inspection** — The gears shall be inspected to verify that the materials, design, construction, marking, dimensions, and workmanship are in accordance with this specification.

4.3.2 **Surface Roughness** — The surface roughness shall be inspected and evaluated optically under 10 power magnification by comparing the test surface with a standard surface finish of known roughness produced on the same material. (See 3.3.)

4.3.4 **Measuring and Checking Pressures** — When making measurement-over-wire measurements or when checking the gears on a variable-center-distance device, the amount of pressure applied to the measuring wires or applied to maintain intimate contact between the gear to be checked and the master gear shall be as follows:

DIAMETRAL PITCH	CHECKING PRESSURE, OZ
24	26 to 30
32	22 to 26
48	18 to 22
64	10 to 14
72	10 to 14
80	6 to 10
96	7 to 9
120	3 to 5
200	2 to 4

4.3.5 **Torque Test** — Gears having assembled hubs shall have 5 inch-pounds torque applied between the gear blank and the hub. The torque shall be applied gradually in not less than 30 seconds and shall be maintained for not more than 30 seconds.

5. **Preparation for Delivery.**

5.1 **General:**

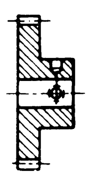
All gears after inspection shall be packaged in such a manner that the teeth shall not be damaged in transit or stock handling.

PRECISION SPUR GEARS
STANDARD STOCK SPECIFICATIONS
SAMPLE NUMBERING

STD – G – 4 – 056 – 02 – 3 – P3

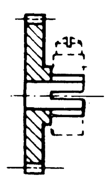
- G: Type Gear
Pin Hub =
- 4: Diametral Pitch
64 pitch =
- 056: No. of Teeth
56 Teeth
For limits
see Table "C"
- P3: Class of Gear Designator
= Precision Class 3. See Table "A" and "B" for tolerances.
- 3: Material Designator
= Nylon-Mil-17091B Type 1 may have stainless steel insert in bore to maintain stability.
- 02: = Basic blank designator characteristics per table "G", "H" or "J". Bore, face, Hub Style, etc.

PIN HUB



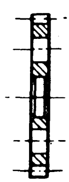
TYPE "G"

SPLIT HUB



TYPE "H"

HUBLESS



TYPE "J"

Type of Gear

–G = Pin Hub
–H = Split Hub
–J = Hubless

Basic Blank Characteristics

Obtain number from Gear Blank Table "G" "H" or "J"

Pitch Designators

Pitch Numerals

24 = –1
32 = –2
48 = –3
64 = –4
72 = –5
80 = –6
96 = –7
120 = –8
200 = –9

No. of Teeth Designator

Use 3 Digits,
Example:
.056 = 56 teeth
156 = 156 teeth

Prec. Class Designator

Class	Designator
Precision 1	= –P1
Precision 2	= –P2
Precision 3	= –P3
Ultra Prec. 1	= –UP1

STANDARD GEAR MATERIAL DESIGNATORS

MATERIAL	Code Designator
ALUMINUM	
2024-T42 Aluminum Alloy (QQ-A-268 Temper T42 or QQ-A-355B Temper T4) anodized before cutting of teeth per MIL-A-8625B	1
STAINLESS STEEL	
Corrosion Resistant Steel AISI Type 303 annealed (MIL-S-7720 Comp 303 Cond A). Shall be passivated according to MIL-S-5002 after gear cutting	2
NYLON — MIL-17091B — Type 1	
May have stainless steel insert in bore to maintain stability	3
PHENOLIC — MIL-P-15035B — Type FBE	
Fungus-proofed before gear cutting. Per Jan-T-152	4
BRONZE (Tobin) — MIL-B-994 a Compo A.	
Natural	5

OTHER SPECIAL GEAR MATERIALS

*Leaded Phosphor Bronze (ASTM B139 Alloy B1, Hard)	6*
*2024-T42 Aluminum Alloy (QQ-A-268 Temper T42 or QQ-A-355B Temper T4)	7*
*2024-T42 Aluminum Alloy (QQ-A-268 Temper T42 or QQ-A-355B Temper T4) anodized after final machining per paragraph 3.7 of this specification	8*
*Corrosion, Resistant Steel AISI Type 416 (QQ-S-00763A Class 6 Type A) may be heat-treated to 125,000 to 140,000 PSI to improve wear and life. Shall be passivated according to MIL- S-5002 after final machining	9*

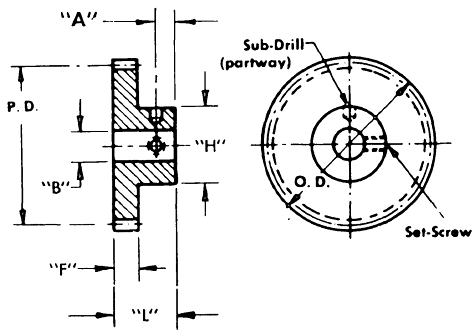
*Blanks not stocked, made to order.

TABLE "A" TOLERANCES					
Class of Prec.	Pitch Diameter	Outside Diameter	Bore	Total Composite Error	Tooth to Tooth Error
—P1	+.000 —.001	+.000 —.002	+.0005 —.0000	.001	.0004
—P2	+.000 —.0007	+.000 —.0015	+.0003 —.0000	.0005	.0003
—P3	+.0000 —.0005	+.000 —.001	+.0002 —.0000	.00025	.0002
—UP1	+.0000 —.0004	+.0000 —.0008	+.00015 —.00000	.0002	.00015

TABLE "B" TOLERANCES				
Class of Prec.	Taper of Bore per Inch	Side Wobble at 1″ Rad	Surface Finish of Tooth	Surface Finish of Bore
—P1	.0004	.001	48	48
—P2	.0003	.0007	32	32
—P3	.0002	.0005	24	24
—UP1	.00015	.0004	16	16

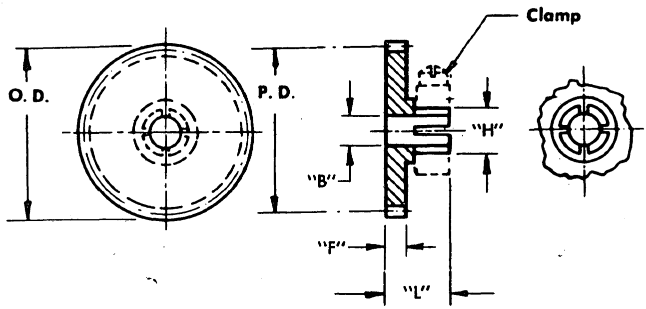
TABLE "C" DIAMETRAL PITCH AND TEETH NUMBER LIMITS			
Pitch	Designator	Min. Number Teeth	Max. Number Teeth
24	1	15	96
32	2	20	128
48	3	21	144
64	4	23	192
72	5	23	216
80	6	30	240
96	7	32	288
120	8	36	360
200	9	76	600

PIN HUB TYPE



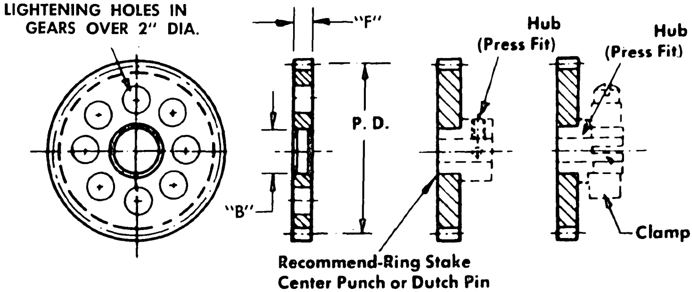
TYPE "G" DIMENSIONS							
Gear Blank Designator	"B" Bore Diameter	"F" Face Width	"H" Hub Diameter	"L" Overall Length	Set Screw	Sub Drill	"A"
—01 —02	.0900 .0935	.062	.250	.250	#0-80	#76	.09
—03	.1200	.093	.312	.281	#2-56	#69	.09
—04 —05 —06	.1248	.062 .093 .125	.312	.250 .281 .312	#2-56	#69	.09
—07	.1562	.125	.312	.312	#2-56	#69	.09
—08 —09 —10	.1873	.062 .125 .187	.375	.281 .343 .406	#4-40	#60	.11
—11	.2405	.187	.375	.406	#4-40	#60	.11
—12 —13 —14	.2498	.062 .125 .187	.500	.312 .375 .437	#6-32	#50	.12
—15 —16	.3123	.187 .250	.500	.375 .437	#6-32	#50	.12
—17 —18	.3748	.187 .250	.562	.375 .437	#8-32	#44	.12

SPLIT HUB TYPE



TYPE "H" DIMENSIONS					
Gear Blank Designator	"B" Bore Diameter	"F" Face Width	"H" Hub Diameter	"L" Overall Length	Shoulder Length
–01 –02	.0900 .0935	.062	.188	.343	.250
–03	.1200	.093	.188	.375	.250
–04 –05 –06	.1248	.062 .093 .125	.188	.343 .375 .406	.250
–07	.1562	.125	.250	.406	.250
–08 –09 –10	.1873	.062 .125 .187	.250	.343 .406 .468	.250
–11	.2405	.125	.312	.406	.250
–12 –13 –14	.2498	.062 .125 .187	.312	.343 .406 .468	.250
–15 –16	.3123	.187 .250	.375	.468 .531	.312
–17 –18	.3748	.187 .250	.437	.468 .531	.312

HUBLESS TYPE



TYPE "J" DIMENSIONS		
Gear Blank Designator	"B" Bore Diameter	"F" Face Width
—01 —02	.3748	.062 .125
—03 —04	.4998	.125 .187
—05	.6875	.125
—06	.9375	.125
—07	1.125	.187

To Find:

$$\text{P. D.} = \frac{\text{No. of teeth}}{\text{Diametral pitch}} \qquad \text{O. D.} = \frac{\text{No. of teeth} + 2}{\text{Diametral pitch}}$$

$$\text{Wire Dia. G} = \frac{1.728}{\text{pitch}}$$

Measurement over wires = Ref. Van Keuren Tables

Chapter Three

GEAR BOXES AND GEARHEAD SPEED REDUCERS

SERVO GEAR BOXES

The outputs of servo motors, which operate best at high speeds, generally have to be geared down so they can be used with slower-speed servo devices. For this reason standard servo gear boxes or reduction units have been developed, which save the user the trouble of erecting his own speed reducers. These gear boxes may also be used as speed increasers under *certain* limited conditions.

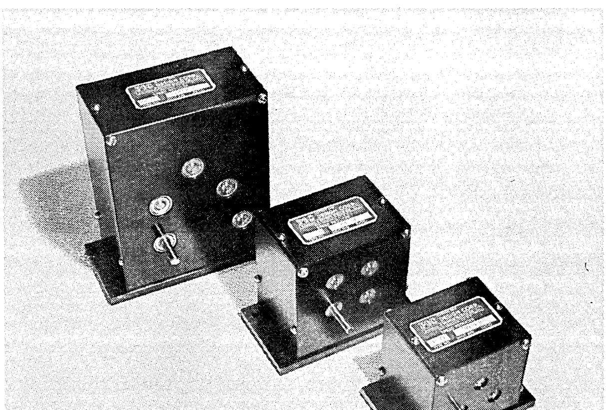


FIGURE 3-1

Standard gear boxes (see Figure 3-1) house various spur gear combinations and provide nearly 800 reduction ratios. The boxes are assemblies of machined plates, flat and parallel within ± .0005″. Shaft bearings, ball or oil-less bronze, are mounted in the side plates. The factory-assembled units observe all the rules for maximum precision set down in Chapter 2 – mis-meshed gears, aluminum pinion meshed with stainless steel gear, removal of pin hub set screw, etc. Bearing location, components assembly, and use of spacers, collars and washers all follow best mechanical engineering practice.

Input and output shafts are in the same horizontal plane and may be on the same or opposite sides of the box. The base plate may be perpendicular or parallel to the shafts. (See Figure 3-2.)

Many mounting arrangements are possible: a typical example is shown in Figures 3-3 & 3-4.

GEAR BOX CONSTRUCTION

Standard servo gear boxes are made with 1/8″, 3/16″ or 1/4″ diameter shafts for light, medium, and heavy-duty applications. All provide reduction ratios ranging from 2:1 to approximately 625:1. The light-duty boxes are for maximum rated torque output of 250 oz. in., the medium duty units 350 oz. in., and the heavy duty reducers 500 oz. in. Maximum backlash, as measured at output shaft with input shaft locked, is 30 minutes. Precision servo gear boxes can be supplied in custom built units with anti-backlash gears. (Recommended at output pass only.)

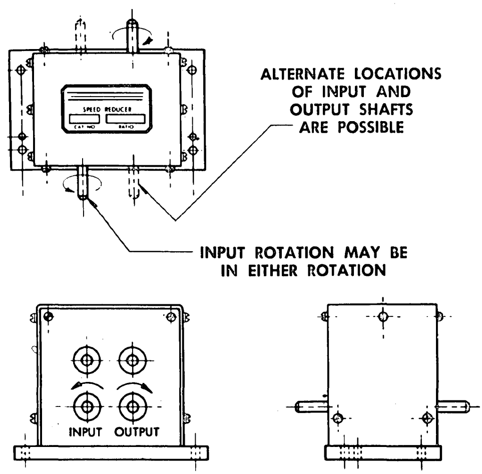


FIGURE 3-2

Servo gear boxes are most often used in design and development work. The more compact gearhead speed reducers, also covered in this chapter, are more commonly used for production systems.

A special multi-ratio speed reducer-increaser has been developed particularly for research development work. It embodies five reducers

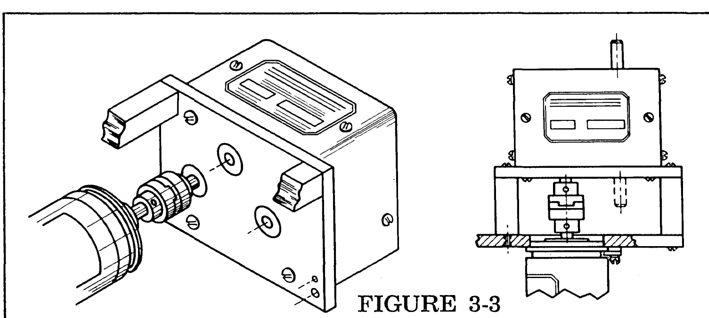


FIGURE 3-3

and/or five increasers in one unit, with ratios of 4:1, 3:1, 2:1, 4:3, and 3:2. Selection of a desired ratio is made by gearing other servo system components to the proper shafts of the multi-ratio speed reducer-increaser. (See Figure 3-5.)

A dial-a-ratio unit has been designed incorporating six different ratios which may be changed, even under fully loaded conditions. (See Figure 3-6.)

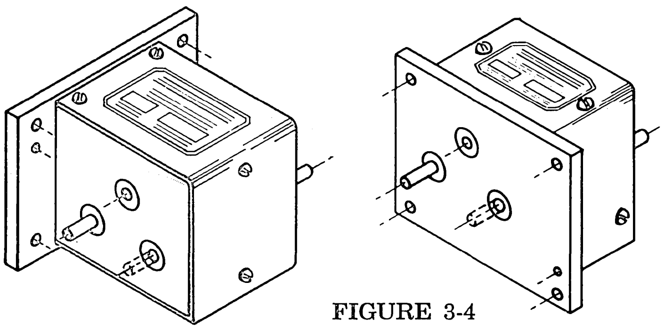


FIGURE 3-4

GEARHEAD SPEED REDUCERS

Gearhead speed reducers are cylindrically packaged trains of spur gears spatially arranged about the center axis of the cylinder. An illustration of gearhead speed reducers appears in Figure 3-7, which also shows a list of specifications.

Gearheads are designed to be attached or coupled to any particular type of servo motor (see Figure 3-8). The foremost end or head of the motor with the gears is called the gearhead end, while

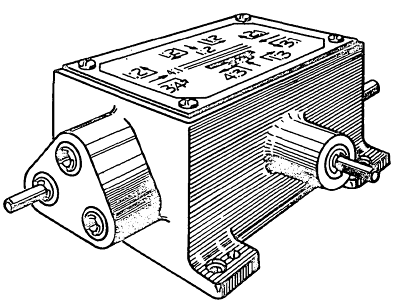


FIGURE 3-5

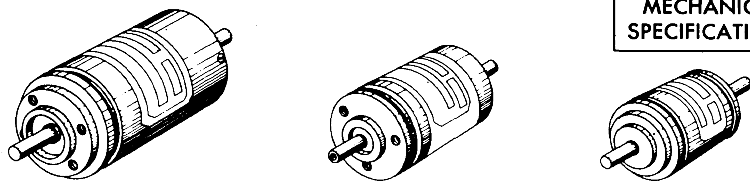
the package of gears itself becomes the gearhead or gearhead speed reducer. The shaft of the motor carries a gear and becomes the input shaft of the gearhead speed reducer.

REDUCTION RATIOS

Gearhead speed reducers cannot be used as speed increasers, since for practical purposes the torque required to turn the gears through the output shaft is greater than provided by most servo mechanisms. They are compact and lightweight, offering high reduc-

SPEED REDUCER DATA

MECHANICAL SPECIFICATIONS



1-3/4 Dia. 1-1/16 Dia. 15/16 Dia.

SPECIFICATION	Size 10 Series	Size 11 Series	Size 18 Series
Maximum Rated Output Torque	40 oz. in.	65 oz. in.	125 oz. in.
Frame Size and Dia. of Unit	Size 10 15/16 Dia.	Size 11 1-1/16 Dia.	Size 18 1-3/4 Dia.
Shaft Size	1/8″ Dia.	1/8″ Dia.	1/4″ Dia.
Operating Load Torque	25 oz. in.	30 oz. in.	65 oz. in.
Maximum Momentary Load Torque	60 oz. in.	90 oz. in.	150 oz. in.
Starting Torque (Max.)	.004 oz. in.	.005 oz. in.	.03 oz. in.
Backlash (Max.) Measured at Output Shaft with 5 oz. in. Rev. Load	30 Min. (Max.)	30 Min. (Max.)	30 Min. (Max.)
Lubricated Per Mil. Specs.	Oil Per Mil-L-6085a Grease Per Mil-G-3278		
Ball Bearings	A. B. E. C-5 or Better		
Gear Tolerances	Precision 2 Tolerances or Better		
Shaft Radial Play	With 4 oz. Gage Load .002 In. Per In. Length Max.		
Shaft End Play	With 1 Lb. Gage Load .004 In. Maximum		
Weight (Approx.)	2 oz.	3 oz.	8 oz.
Moment of Inertia at Input Shaft	.02 Gm-Cm²	.03 Gm-Cm²	.15 Gm Cm²

FIGURE 3-7

tion ratios – from 2:1 to 3000:1. However, because of the large number of gears, backlash is greater than in other types of speed reducers and alignment problems make it necessary to keep torque output low.

The units may be driven in either direction, but the output shaft does not always rotate in the same direction as the input shaft, since there may be either an odd or an even number of internal gear meshes. Shafts are either 1/8″ or 1/4″ in diameter. Maximum output

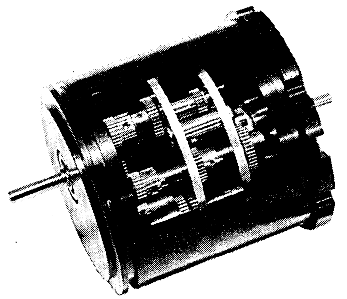


FIGURE 3-6

torques of two 1/8″ diameter shaft gearhead models are 35 oz. in. and 65 oz. in., respectively. Maximum output of the 1/4″ diameter shaft model is 125 oz. in.

Overall dimensions of the three models are roughly 7/8″ in diameter by 1-3/8″ long, 1″ in diameter by 1-3/4″ long, and 1-1/2″ in diameter by 2-13/16″ long. Gear boxes range in size from 2-1/4″ x 2-1/2″ x 3-1/2″ to 2-1/4″ x 4-3/4″ x 5″. Gearhead speed reducers are normally used for production models, in contrast to the servo gear boxes which are used primarily in development work.

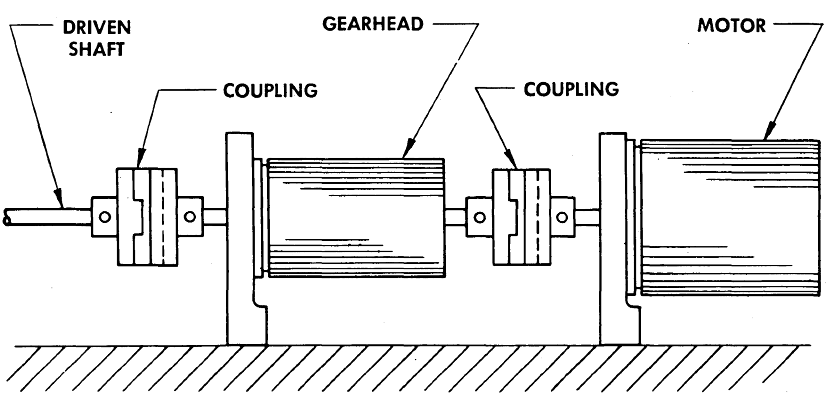


FIGURE 3-8

Chapter Four

WORM-AND-WHEEL GEARS AND PACKAGED SPEED REDUCERS

The *pinion* (or worm) of a worm-and-wheel is a section of tubing with a helical thread turned on its outer surface. The mating worm gear, or wheel, has spur gear-like teeth that are concaved across the top to better mesh with or wrap around the worm.

The *worm-and-wheel* transmits motion between perpendicular shafts that are not in the same plane (usually at 90 degrees). The full faces of the wheel teeth make contact with the worm, which minimizes both backlash and wear and makes the worm-and-wheel particularly good for power and torque transmission.

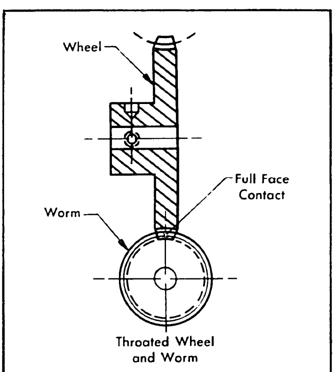


FIGURE 4-1

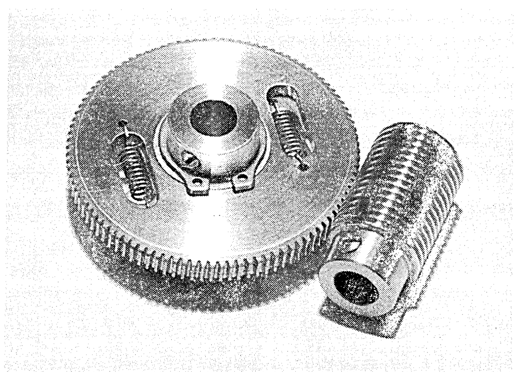


FIGURE 4-2

The principal advantage of the worm-and-wheel is that it provides high speed reduction ratios up to 120:1 or more with a single mesh. It may *not* be used as a speed increaser as the teeth lock when an attempt is made to transmit motion via the wheel.

Smaller speed reduction ratios can be achieved through the use of double-thread and four-thread worms and still allow the gear and worm to run on the same centers.

For example, a 120-tooth gear and a single-thread worm provides a 120:1 ratio. Using a double-thread worm and a 120-tooth gear, the ratio is 60:1.

DISADVANTAGES OF WORM GEARS

It is more difficult to maintain the precision of single-double-and four-thread worm gears because of the greater *helix angle error* than of conventional spur gears which have straight teeth. It is also difficult

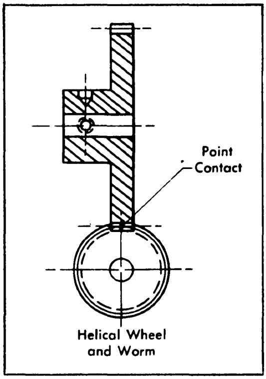


FIGURE 4-3

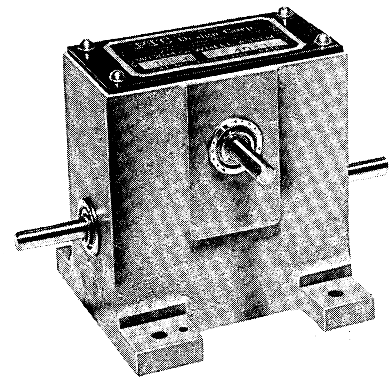


FIGURE 4-4

to align a worm or worm gear because of the curved face. The center of the curve must be directly over the center of the worm as illustrated in Figure 4-1. The shafts must be held at exactly right angles to each other to avoid a binding condition in the mesh.

Anti-backlash worm wheels are available to help minimize backlash on critical high accuracy positioning devices (see Figure 4-2).

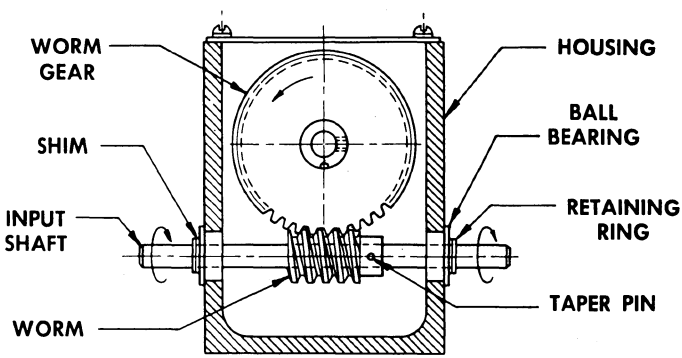


FIGURE 4-5

DESIGN ERRORS

The engineer or designer makes a mistake when he selects a stock worm gear to suit his application, and then changes the pitch diameter of the worm to suit his space requirements. Every time the worm diameter is changed, a special hob is required to produce the wheel. Thus, *much* time is lost waiting for special tooling, and at prime cost.

It is possible to design around the special hob requirement by using a straight helical gear mesh with a special worm pitch diameter. The alignment is much easier. However, there is only point contact at the tooth instead of full-face contact as illustrated in Figure 4-3. In time, this point wears and excessive backlash is introduced.

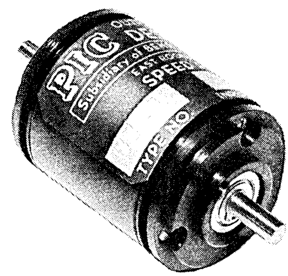


FIGURE 4-6

WORM AND WHEEL SPEED REDUCERS

Worm and wheel speed reducers provide a wide range of speed reductions ranging from 12.5:1 to 120:1. They provide low-cost high-reduction ratios with but one gear mesh.

PACKAGED SPEED REDUCERS

Packaged speed reducers are speed reduction gear trains assembled at the factory. Their use results in savings of time and money,

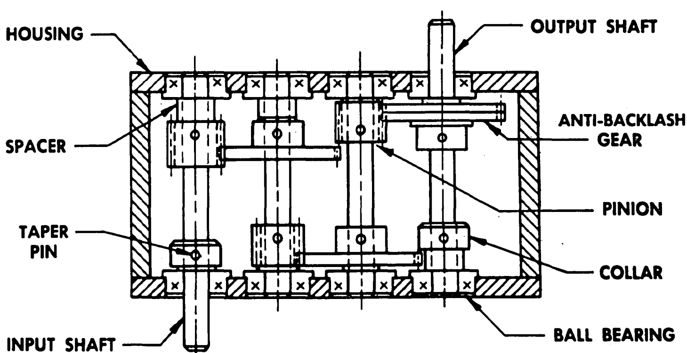


FIGURE 4-7

since it is unnecessary for the user to design and assemble his own speed reduction units. Packaged worm and gear speed reducers made up of most of the gear types previously discussed are available from stock, (Figures 4-4 and 4-5) and include spur gear, helical gear and bevel gear types. Spur gear speed reducers are provided in two forms – servo gear boxes and gearheads. Both types are widely used with servo motors. A typical package speed reducer is shown in Figure 4-6.

SERVO GEAR BOXES

The servo gear box (see Figure 4-7) houses a spur gear train, designed according to the standards set forth earlier. Shaft bearings are mounted in two opposing sides of the box, one of the other sides extended at both ends and drilled for mounting. The side plates are

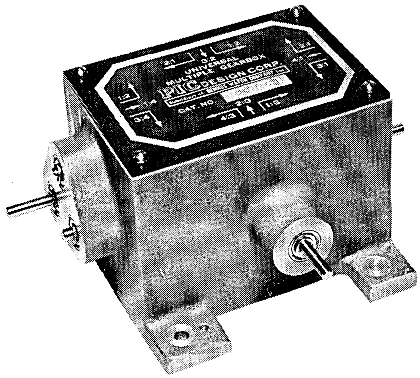


FIGURE 4-8

precision ground and are flat and parallel within .0005″; corners are 90° square within 5′. Ready-made servo gear boxes provide speed reduction ratios up to 625:1.

Many variations and mounting possibilities are available with these units. For example, mounting plates may be perpendicular or parallel to the gear shafts, input and output shafts may be on the

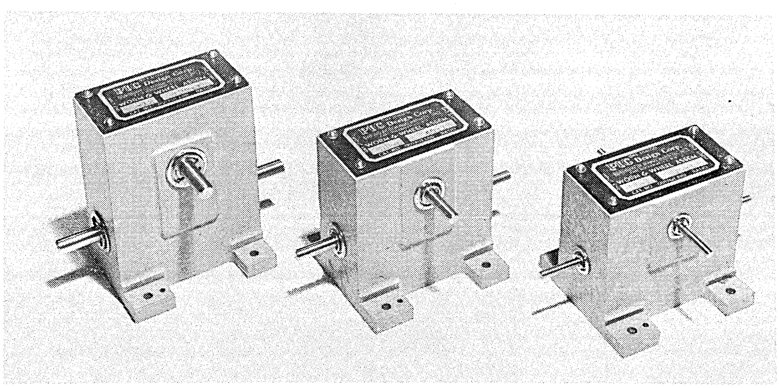


FIGURE 4-9

same or opposite sides of the box, special shaft lengths and diameters may be specified, anti-backlash gears may be used in the gear train, etc.

Leading suppliers will recommend specific arrangements for particular installations upon receipt of pertinent information and a rough sketch showing the requirements of *multi-ratio gear boxes*.

MULTI-RATIO GEAR BOXES

Servo gear boxes are often used in development work. To avoid the necessity of multiple gear boxes for commonly used low-reduction ratios, a multi-ratio gear box is available which provides five ratios

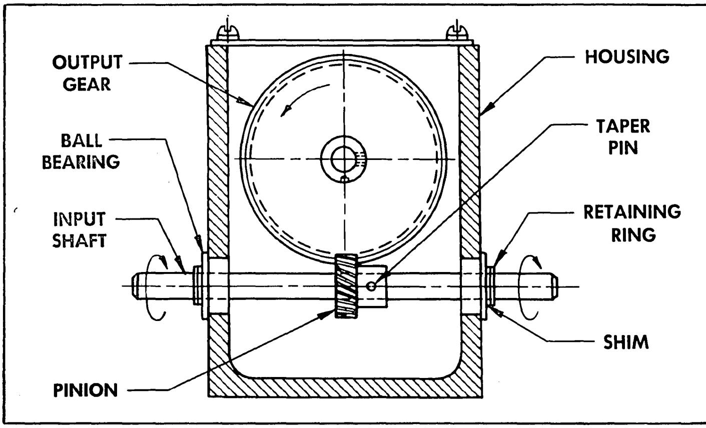


FIGURE 4-10

merely by using different shaft extensions. These are 4:1, 3:1, 2:1, 4:3 and 3:2. (See Figure 4-8.)

GEARHEAD SPEED REDUCERS

Gearhead speed reducers are so called because they are generally secured to the head of a servo motor. They are cylindrical enclosures

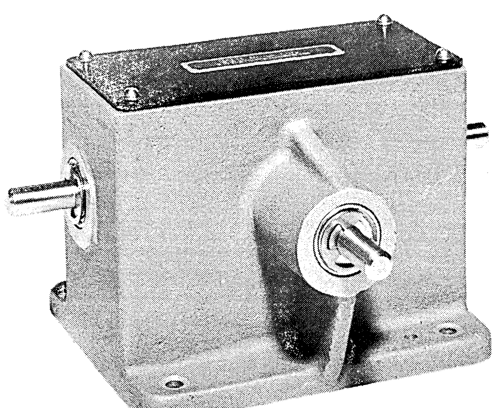


FIGURE 4-11

housing trains of spur gears spatially arranged about the center axis of the cylinder. The special arrangement saves space in comparison with the servo gear box and makes for lightweight, compact reducers. Reduction ratios up to 310:1 are available.

HELICAL SPEED REDUCERS

Helical gear speed reducers house a pair of helical gears at right angles to one another. (See Figures 4-9 and 4-10.) They provide small reductions in speed – 2:1, 3:1 or 4:1. One-to-one ratios are

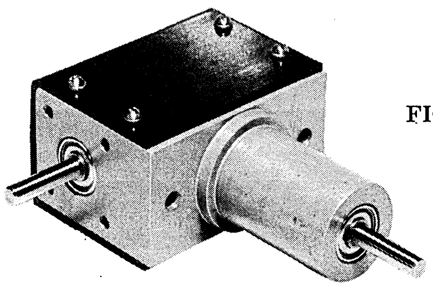


FIGURE 4-12

also available where 90° change of direction is required but the same speed is to be maintained.

BEVEL GEAR SPEED REDUCERS

Bevel gear speed reducers house a pair of bevel gears and reduce speeds in ratios of 2:1, or 3:1. One-to-one ratios are provided for use

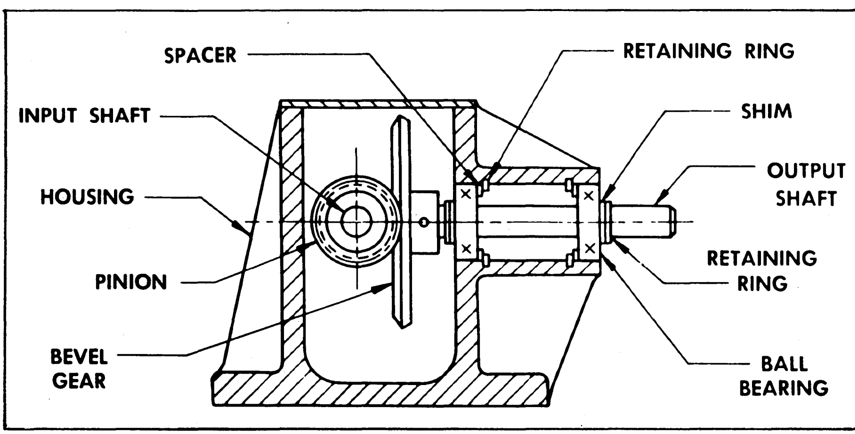


FIGURE 4-13

when motion is to be transmitted at right angles, but no speed reduction is required. (See Figures 4-11, 4-12, 4-13, and 4-14.)

TIME AND COST SAVINGS

Thus, the designer has a wide range of packaged speed reduction units to choose from to meet virtually any design requirement. These ready-made units eliminate the delays which would be encountered if the designer engineered his own speed reducer, had parts made or purchased them and had them assembled. In addition, standard

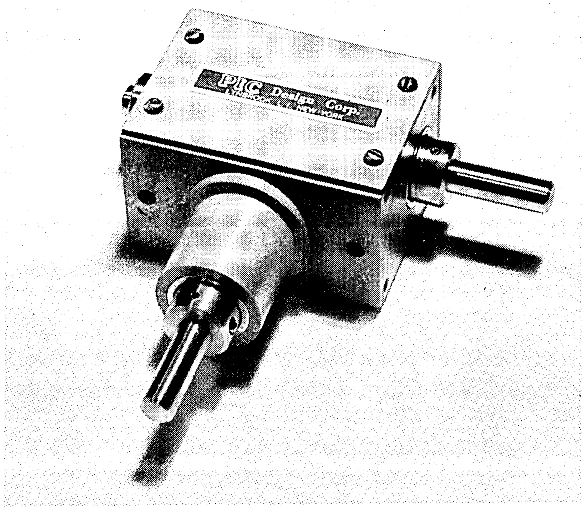


FIGURE 4-14

reducers have been designed by specialists, the designs have been tested in service, and specified performance is assured.

Chapter Five

HELICAL, MITER AND BEVEL GEARS

HELICAL GEARS

Helical or spiral gears have oblique teeth which are segments of helixes. A typical helical gear is shown in Figure 5-1.

The principal advantage of helical gears in the precision instrument field is that they transmit motion between non-parallel shafts that are not in the same plane, as shown in Figures 5-2 and 5-3. They may also be used to transmit motion between parallel shafts. (See Figure 5-4.)

However, helical gears are seldom used on parallel shafts in precision instrument work. As a tooth meshes, the leading edge first

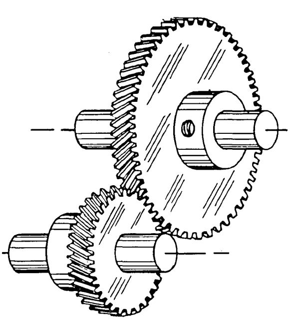


FIGURE 5-1

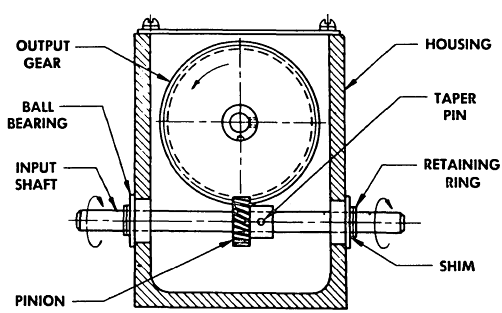


FIGURE 5-2

engages the mating gear and the balance of the tooth slides into mesh as the gears rotate. This makes the helical gear less efficient and more power consuming than the spur gear, which is generally used for parallel shaft applications.

Helical gears are used in parallel shaft operations in the power field because their teeth are stronger than spur gear teeth due to their longer base. Also, since several teeth are in mesh at one time, these gears offer smoother power transmission and quieter operation than ordinary spur gears cut to commercial tolerances.

TRANSMITTING ANGULAR MOTION

Helical gears are particularly useful for transmission of motion in the precision instrument field when motion is transmitted between shafts that are at an angle with one another and not in the same plane. The most common angle at which motion is transmitted between shafts is 90°, and standard gears with teeth cut at 45° helical

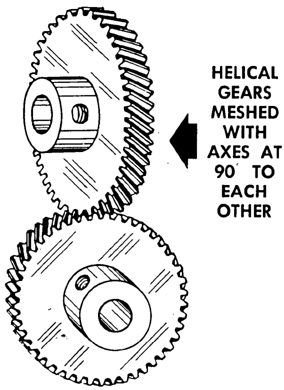


FIGURE 5-3

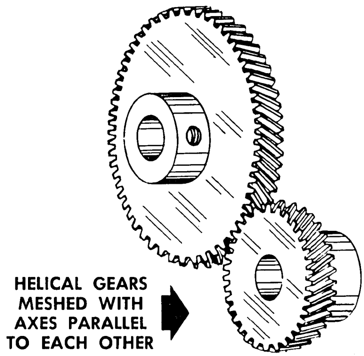


FIGURE 5-4

angles are available for this case. Where motion is to be transmitted between shafts at some other angle, helical gears must be made to order with teeth at helical angles which will accommodate the shaft relationship.

Due to the angularity of the teeth, helical gears produce an axi thrust, and in the power transmission applications this must be ab sorbed by thrust bearings. However, in the precision instrument fiel

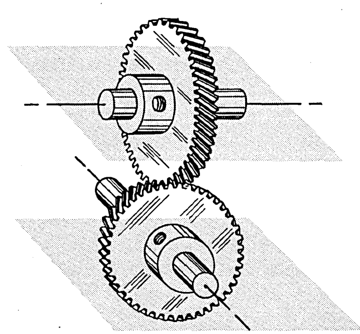


FIGURE 5-5

the forces are sufficiently small as to be absorbed by ordinary precision bearings. The precision built into the gears minimizes friction to the point where thrust forces are not a factor requiring compensation.

Standard helical gears are of the pin-hub type. They are termed right-hand helical gears when the teeth bear the same relation to the gear axis as the threads on a right-hand screw bear to its axis. A left-hand helical gear bears a similar relation to a left-hand screw.

MESHING OF HELICAL GEARS

Helical gears of the same hand mesh at right angles. Helical gears of the opposite hand mesh on parallel shafts. The meshing of helical gears is illustrated in Figure 5-5. Helical gears of the proper hand

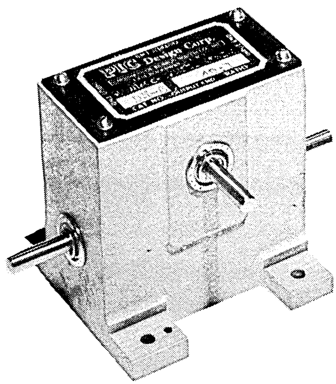


FIGURE 5-6

must be selected with care for given applications to be sure they provide motion in the direction desired.

Standard precision instrument helical gears are available in stainless steel and aluminum in Precision 1 classification as follows:

Bore	*Pitch*	*Teeth*
1/4	48	20, 25, 30
3/16	48	35, 40, 45
3/16	64	50, 50, 70
1/8	64	80, 90, 100

The same center distance formula is used for helical gears as is used for spur gears, and the same backlash considerations apply.

HELICAL GEAR BOXES

Accurate mounting of shafts at right angles in different planes presents assembly problems which can be avoided by purchasing helical gear boxes with factory-assembled gears. These boxes have through shafts at right angles to one another, one extending through the sides of the box and the other through the ends of the box. (See Figure 5-6.)

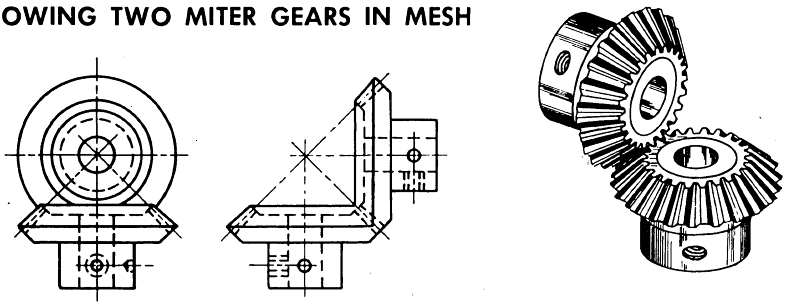
Gear Ratios of 1:1, 1:2, 1:3 and 1:4 are available, and the units can be used to reduce or increase speed. Because of the four-shaft protrusions, the box is a versatile modular unit which can be installed to suit an infinite number of layout possibilities. Helical gear assemblies have backlash of less than 30 minutes.

MITER AND BEVEL GEARS

Miter and bevel gears are designed to transmit motion at an angle, usually but not always, at an angle of 90°. (See Figure 5-7.) Such gears are called miter gears when they have a 1:1 ratio. Bevel gears have a ratio other than 1:1, usually 2:1 or 3:1.

Bevel gears have their teeth cut on the surface of the frustum of

PROJECTED AND ISOMETRIC VIEWS SHOWING TWO MITER GEARS IN MESH



PROJECTED VIEWS **ISOMETRIC VIEW**

FIGURE 5-7

a cone instead of on the edge of a disc or cylinder as in the case of spur and helical gears. They are used to transmit motion between shafts that are at an angle with one another and in the same plane. They are most frequently used with shafts that are at right angles.

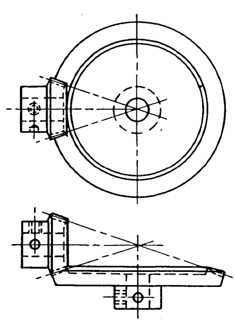


FIGURE 5-8

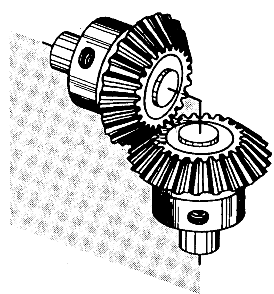


FIGURE 5-9

However, they can be designed to provide meshes at other angles. (See Figure 5-8.)

Bevel gears can increase or decrease speed as well as change the direction of motion. When used to transmit motion at the same speed between right angle shafts, they are known as miter gears. (See Figure 5-9.)

Standard bevel and miter gears are available for the precision instrument field as matched sets or individual units. They are made with pin or split hubs in Precision 1 and 2 classifications. Stock items provide ratios of 1:1, 1:2 and 1:3.

Package bevel and miter gear boxes provide factory-aligned gears which can be mounted as a unit. This eliminates independent mounting and aligning of gears on shafts, shafts in bearings and bearings in hangers or bearing plates. (See Figure 5-10.)

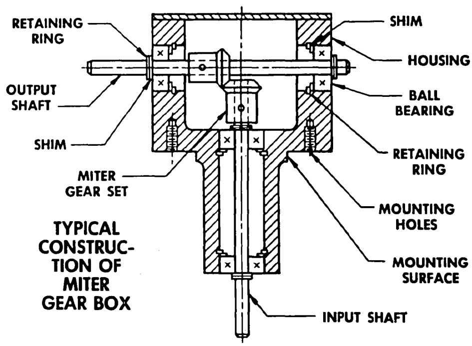
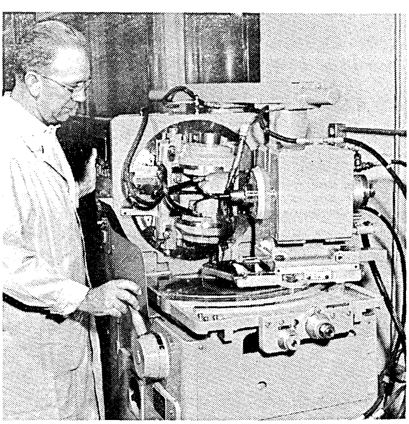


FIGURE 5-10

More precise bevel or miter gears are produced on such equipment as the "Gleason" Coniflex generating equipment. (See Figure 5-11.)

The shaft of one of the gears of a bevel gear box extends through the box and projects from each end. It may be turned in either direction from either end. The shaft for the other gear enters the box through a bearing adapter which is integral with the side of the case aluminum box. It may also be turned in either direction.



Gleason Coniflex Miter and Bevel Gear Generator

FIGURE 5-11

Right Angle Miter and Bevel Gear Checker and Recorder

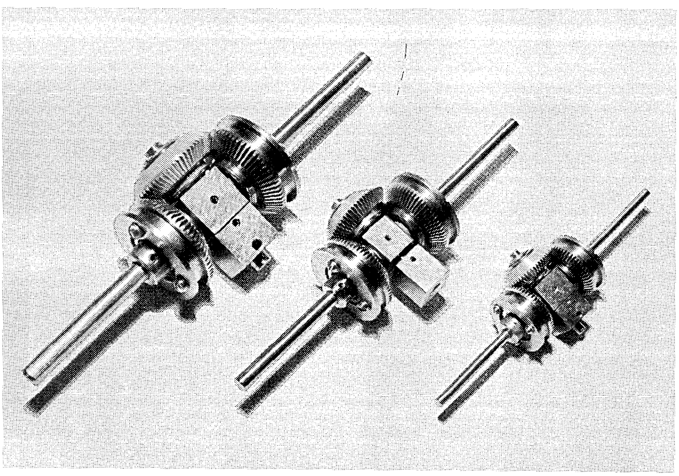
FIGURE 5-12

Chapter Six

MECHANICAL DIFFERENTIALS

The most universally used differentials in precision instrument work are bevel-gear differentials because of the fine accuracy which can be obtained. These are gear systems which add or subtract angular movements transmitted to two of their members and deliver the answer to a third. A typical mechanical differential system is shown in Figure 6-1.

Bevel-gear differentials are widely used for adding and subtracting shaft movements in servo systems and for addition and subtraction



Typical Differentials Less End Spur Gears

in computing machines. They can be geared with input and output shafts to multiply or divide inputs and outputs from and to these shafts. Differentials are also used to measure torque in a rotating shaft and to control the operation of other equipment.

Bevel-gear differentials consist of a spider and a spider shaft assembly. (See Figure 6-2.) The spider comprises a junction block with a cross arm extending from two opposite sides. A bevel gear is bearing-mounted on one arm and a balancing block is mounted on the other arm, or bearing-mounted bevel gears can be mounted on end of the cross arm. The balancing block is less expensive and offers greater accuracy since one pair of meshes is eliminated.

A spider shaft extends from two other opposing sides of the junction block. Bearing-mounted bevel gears on this shaft, one on each side of the junction block, mesh with the spider bevel gear. The spider shaft bevel gears have spur gears secured to their hubs, and the spider shaft has a spur gear fixed to one end. The spider shaft is supported on bearings.

A miter gear differential has all three bevel gears of equal size,

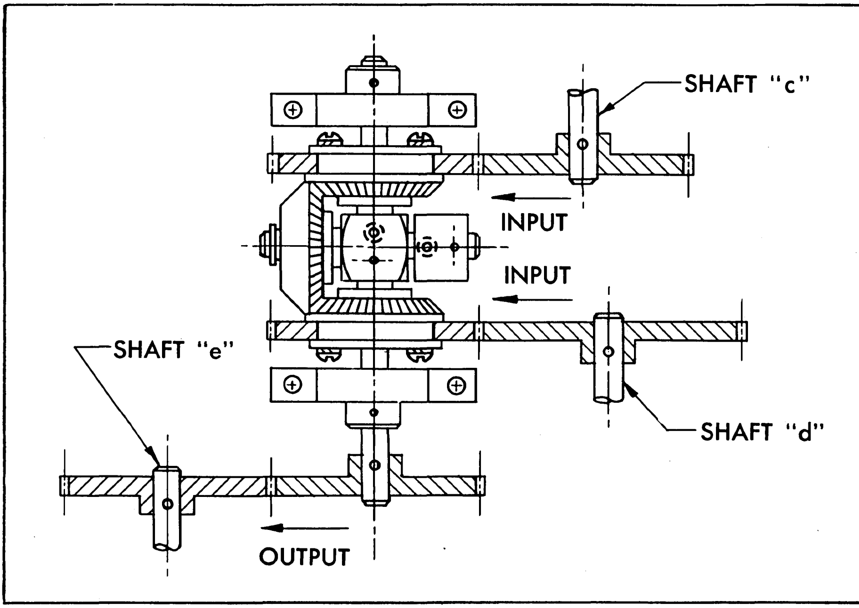


FIGURE 6-1

providing a 1:1 ratio between the spider miter gear and other two gears. The bevel gear differential has a spider gear which differs in size from the other two, resulting in a ratio other than 1:1.

Spur gear differentials are also available but are less precise than the miter and bevel gear types because of unavoidable backlash. For example, in the spur differential illustrated in Figure 6-3, teeth of the face gear are cut radially to the spider-shaft axis whereas those of the spider-arm spur gears are cut parallel to the spider-arm axis. Thus, there is not a perfect mating of the meshing teeth and some play is unavoidable. Figure 6-4 illustrates the ideal mesh.

FUNCTIONAL EXPLANATION

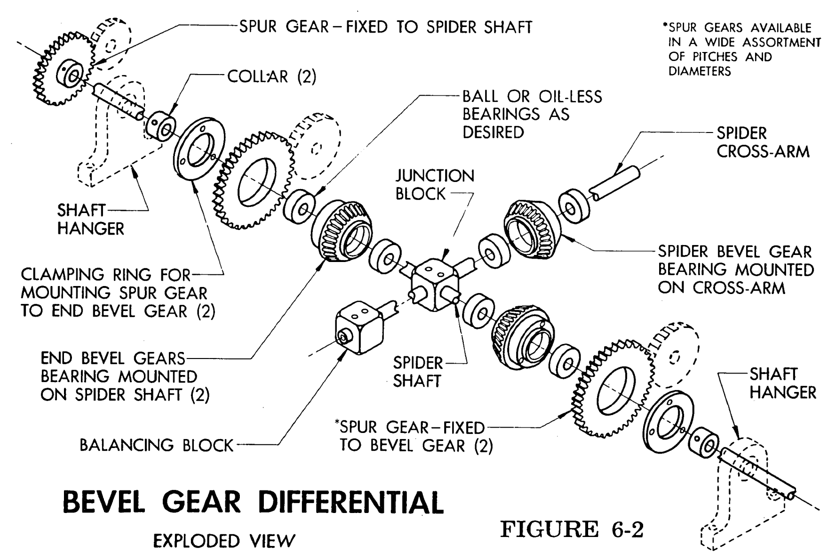
The operation of a mechanical differential system is illustrated in Figure 6-5. Assume that bevel gear *a* is held stationary, that the spider shaft is rotated clockwise and that bevel gear *b* is free to turn on its bearings. As the spider shaft rotates, the spider gear is rotated about axis YY. At the same time, it also rotates about the spider-arm axis XX through an angle equal to the one through which the spider shaft is turned. Thus, we have two equal motions simultaneously carried to bevel gear *b*. This gear therefore will run through an angle twice the one through which the spider shaft was turned. This will always be true regardless of the diameter of the spider gear.

If we reverse the above situation, and rotate *b* with the spider being free to turn, the spider shaft will rotate through half the angular displacement of gear *b*.

If bevel gear *a* is no longer held stationary but is rotated at the same time that *b* is rotated, this will affect the motion of the spider shaft and its angular displacement will equal one-half the vector sum of the angular displacement of the bevel gears.

$$D_s = \frac{D_a + D_b}{2}$$

If both bevel gears rotate in the same direction at different speeds, the spider shaft will rotate in this direction at a speed halfway be-



BEVEL GEAR DIFFERENTIAL

EXPLODED VIEW

FIGURE 6-2

tween the two and the differential adds.

If both bevel gears rotate in the same direction at the same speed, the spider gear will not rotate on the spider arm but the spider arm and shaft will rotate in the same direction as the gears and at the same speed.

If the bevel gears are turned in opposite directions at different speeds, the spider shaft will turn in the direction of the more rapidly moving gear at one-half the difference of the speed of the two bevel gears, and the differential subtracts.

If the bevel gears are turned at the same speed in opposite directions, the spider gear will turn but the spider arm and shaft will not.

The foregoing formula may, of course, also be used in determining outputs when the inputs are to the spider shaft and one bevel gear

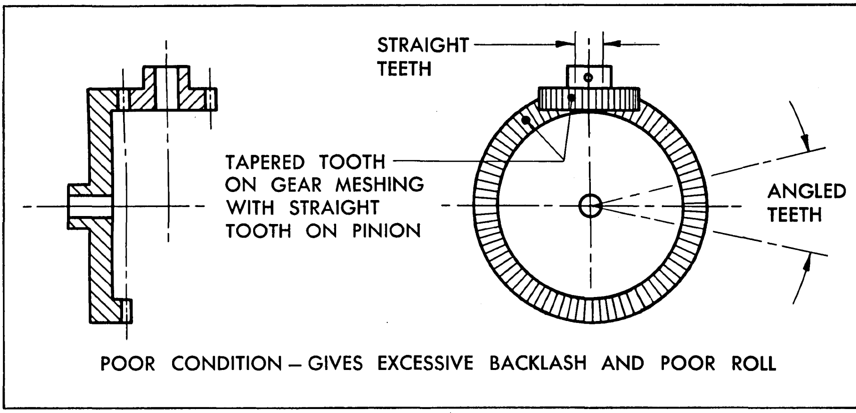


FIGURE 6-3

rather than to the two bevel gears.

In addition to the above two input and output possibilities, there is a third condition which can exist. This would result from restraining the spider shaft and driving one of the bevel gears, the other bevel gear being free to turn on its bearings.

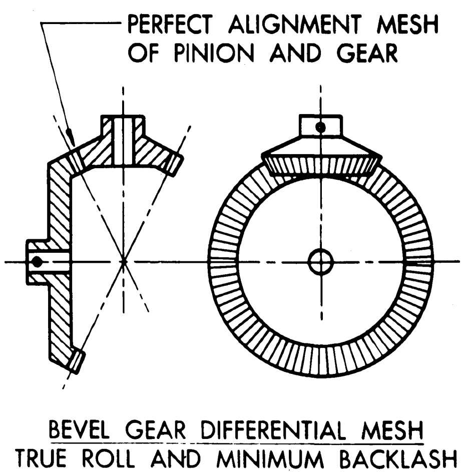


FIGURE 6-4

Under these conditions the differential acts as a simple gear train, which transmits motion from one bevel gear to the other but in the opposite direction. There will be a force tending to rotate the spider arm equal to one-half the force transmitted from one bevel gear to the other multiplied by the ratio of the spider bevel gear to the driving bevel gear. This force can be used to measure torque or to control equipment.

The differential can be used to multiply and divide by using differential spur gears of a different diameter than those with which

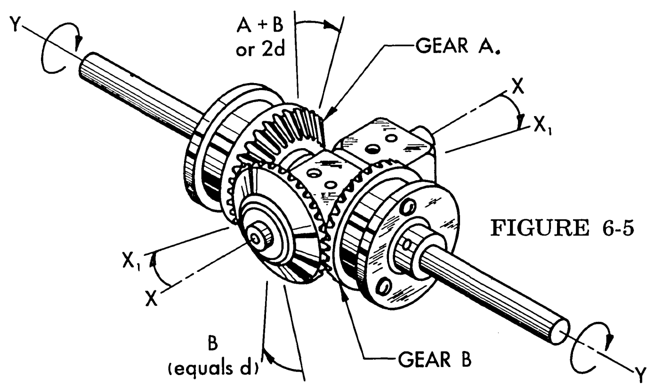
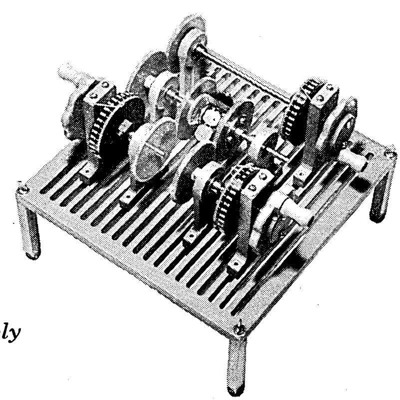


FIGURE 6-5

they mesh. This technique is illustrated in Figure 6-1, in which the two end miter gears provide the inputs, and the spider shaft the output. If it is desired to obtain the algebraic sum of X times the rotation of shaft *c* plus Y times the rotation of shaft *d*, the ratio of the respective spur gears would be specified accordingly. Further, by sim-



Differential Breadboard Assembly

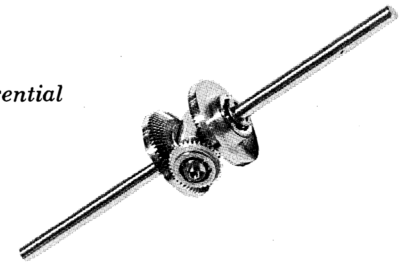
ilarly specifying the ratio of the spider shaft spur gear to its mate, the answer can be multiplied or divided. This can be expressed as follows:

$$D_s = \frac{XD_c + YD_d}{2} \text{ and } D_e = ZD_s$$

where D = the displacement of the respective shafts as indicated, and X, Y, and Z = the ratios of the differential spur gears to the spur gears on shafts *c*, *d*, and *e*, respectively.

These formulae may also be transposed for use with other input-output combinations.

Miniature Differential



APPLICATIONS

Of the many applications of miter and bevel gear differentials, perhaps the simplest is its use for changing the phase relations of one shaft with reference to others in the system. For example, in Figure 6-6, if shafts A and B are geared through the differential as

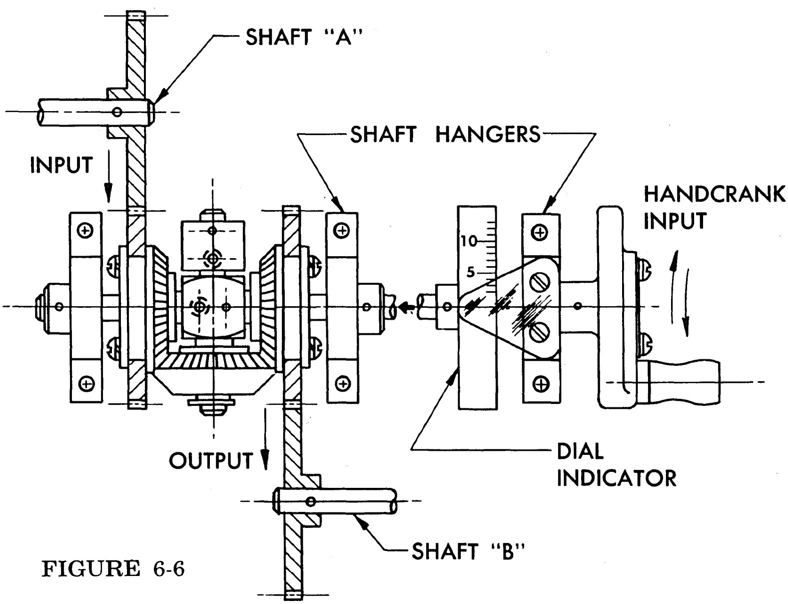


FIGURE 6-6

shown, the position of the output portion of the shaft, with reference to the input section, can be changed by means of a crank input. This will, of course, alter the phase relation of the output section (shaft B) with other shafts in the system.

Advancing one step further, miter and bevel gear differentials may also be used to alter the speed of a shaft. This is analogous to the phase shift application, except that the crank input would be replaced by a continuously driven input. In this instance, the speed of the output section of the shaft differs with the speed of the input section, and the effective speed of the shaft with relation to others in the system is changed.

By relating the rotation of differential components to elements of computer equations, the bevel gear differential may be used to introduce data to computers without disrupting their operation. In the example illustrated in Figure 6-1, computer information normally flows in terms of shaft rotation into the differential via one bevel gear and out the other without alteration. If, however, it is desired to add or delete information, the hand crank is turned a distance which correlates with the data change to be incorporated, and the differential output is altered accordingly.

Miter and bevel gear differentials may also be inserted in a shaft system to measure torque, as shown in Figure 6-7. The torque trans-

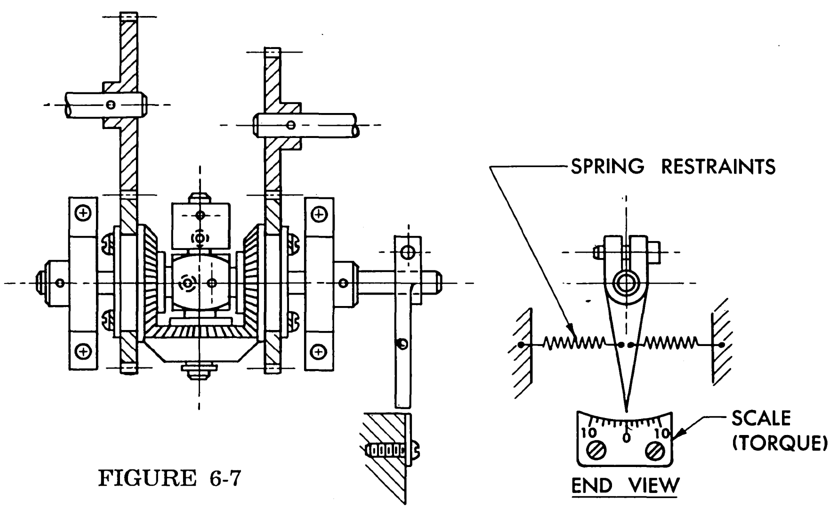


FIGURE 6-7

mitted reacts against a calibrated restraint, thus providing a means of torque measurement.

Another application of miter and bevel gear differentials is as a clutch or brake. In this case the differential is inserted between the drive and the load and one leg is restrained by means of a clutch as shown in Figure 6-8. As long as the load is less than a predetermined value, the clutch stays engaged and the load is driven. However, when the load exceeds the predetermined value, the transmitted torque opens the clutch and frees the restrained leg. This member, having no load, now rotates and the load remains stationary. Clutches can be arranged to engage and disengage as the load rises above and falls below a preset value.

Miter and bevel gear differentials are also used for error measurements and mechanical comparison by employing one leg to indicate differences in input to the other two. They are also used in conjunction with pilot motors as speed controllers, by arranging for

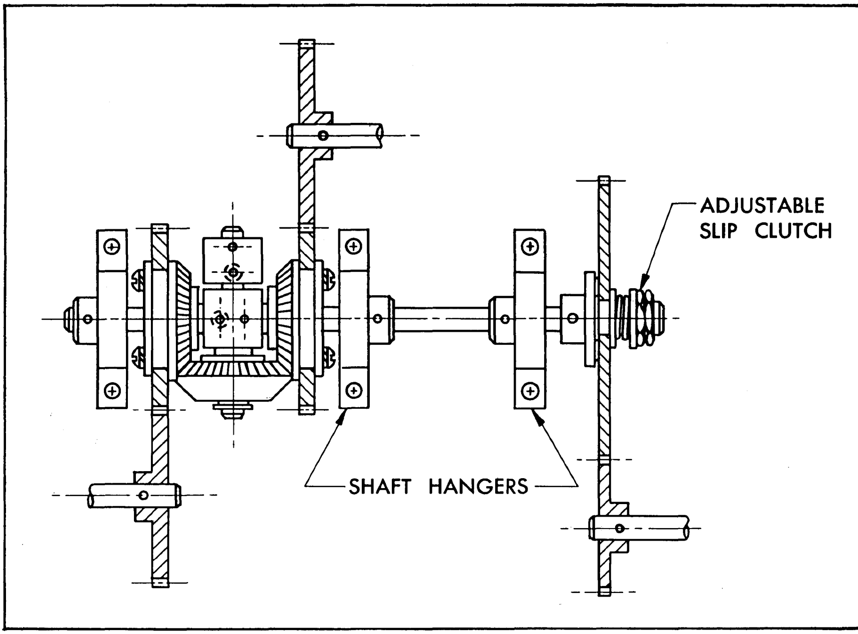


FIGURE 6-8

the output leg to operate a rheostat (or valve) when the input speed from the prime mover deviates from the input speed from the pilot motor.

Practical adaptations of the techniques described and other practical applications are numerous and will readily suggest themselves to the alert designer.

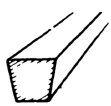
Chapter Seven

CHAIN AND BELT DRIVES

Chain and belt drives are used where there is distance between the drive and drive shafts.

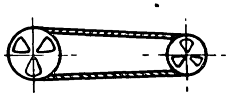
There are many such chain or belt drives in the precision small component field and each has advantages and disadvantages which I will try to outline as follows. Refer to Figure 7-23 for performance characteristics of various drive systems.

"V" BELTS



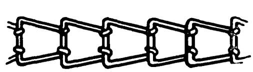
These are very good for power transmissions, however, they have not been developed for the fine precision field and depend on friction or excessive loading for driving.

"SPRING BELTS"



This type of belt drive is ideal where center to center distance is not controlled as the belt is stretched to suit. The bearings of this drive have to be loaded excessively and wear is quite noticeable, in time.

"LADDER CHAIN"



This chain is made of multiple links of wire and runs with sprockets. It is ideal for very slow speeds but has no accuracy because of the nature of manufacture. The "ladder chain", although rather inexpensive can become complicated to connect.

"BEAD CHAIN"

This type of drive is made up of multiple stamp balls such as pull chain and mesh with special pulleys which are indented to receive each ball. Here again, they are not quite accurate enough for instrumentation application as there must be a large amount of play between each ball for proper operation.

Although the "bead chain" is rather cheap, the pulleys are costly because of the dimpling process. This drive is ideal for various angle drive similar to the "No-Slip" plastic drive.

"PRECISION CHAINS"

A small precision stainless steel chain and sprocket line has been developed and standardized to offer positive acid or chemical free drive systems for long span distances. (See Figure 7-1.)

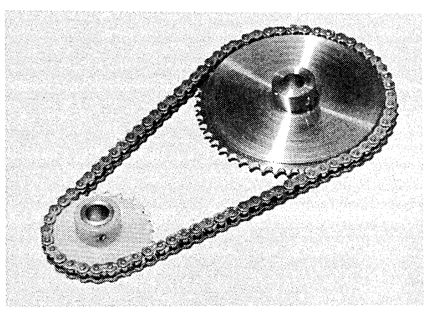


FIGURE 7-1

Unfortunately, these are multiple link connections with many moving parts which must be kept well lubricated in order to keep the noise level down. (See Figures 7-2 and 7-3.)

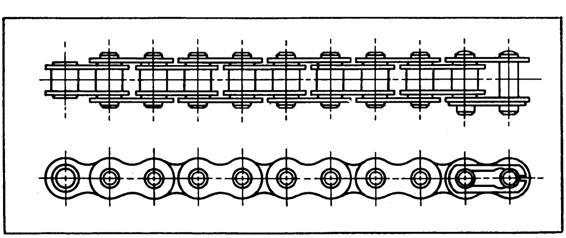


FIGURE 7-2

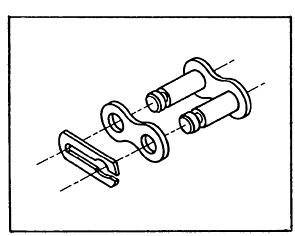


FIGURE 7-3

Sprockets to run with these stainless chains are available in differ-types of materials for various chemical and environmental conditions and in a wide variety of teeth, hub types and bores. (See Figure 7-4.)

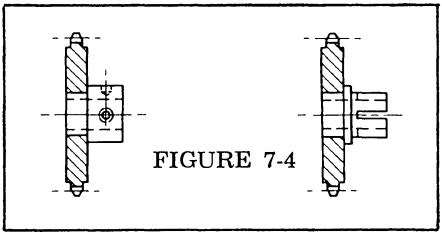
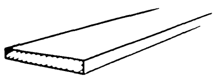
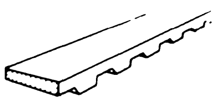


FIGURE 7-4

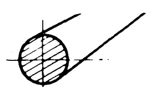
"FLAT BELTS" 

Flat belts are ideal for compact areas where they can be used as conveyors or drives where slip is permissible and metal clips or splicing staples are acceptable.

Pulleys are usually crowned to restrain the belts from side walking.

"COG BELTS" 

This cog or tooth belt is ideal for positive drive & timing purposes. However one of the wheels or pulleys must have side flanges to retain the belt from lateral walk.

ROUND BELT DRIVE 

Another type of belt drive has been the round rubber tension type drive which has the silent drive potential. (See Figure 7-5.)

However, it has certain other disadvantages such as loading the

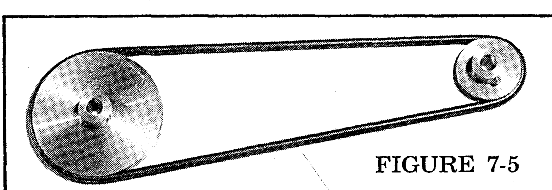


FIGURE 7-5

shafts due to required loading to have the belt drive and not slip. Lubrication cannot be used with this type of drive as it would cause possible slip.

PLASTIC TRANSMISSION BELTS & GEARED PULLEYS

New approaches to instrument design have recently been opened by the introduction of a new patented* "No-Slip" positive drive plastic belt and geared pulley transmission system. (See Figure 7-6.) The belt runs in grooved gear pulleys and has lateral projections which mesh with each tooth in the rims of the geared pulleys to provide positive no-slip belt drive (Figure 7-7).

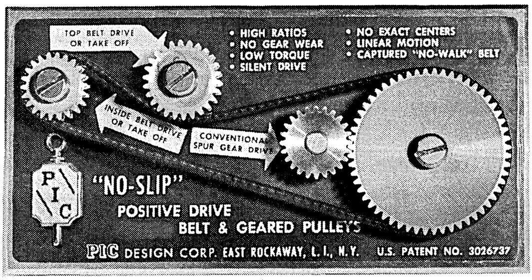


FIGURE 7-6

This innovation offers many design advantages which, in given instances, makes it superior to conventional gears and pulleys, cogged timing belts, roller chains and sprockets, or bead and ladder chains. In addition, unique features of the belt enable it to transmit power in ways that hitherto have not been possible.

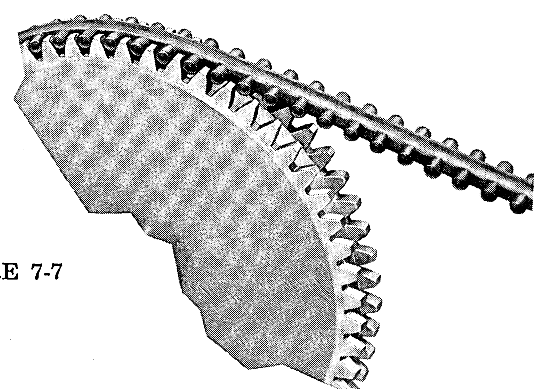


FIGURE 7-7

DEVELOPMENT OF SYSTEM

The concept of the new system was recently originated, primarily to provide a silent drive for sound and recording devices. However, practical considerations of finding suitable materials, and molding procedures delayed its introduction until all methods were tried and tested. At the present time, belts and pulleys are available for instrument-type applications only, although plans are under way for

*Patented by Author Winfred M. Berg.

extending their use through licensing arrangements for large-size power applications such as on lawn mowers, home appliances, toys, bicycles, etc.

The polyurethane belt employed in this system has a core of multiple strands of pre-stretched dacron cord. Many attempts were made before an acceptable, non-stretch, long-wearing belt (Figure 7-8) was developed, having the combination of flexibility and strength required to adapt it for precise instrument and power transmission

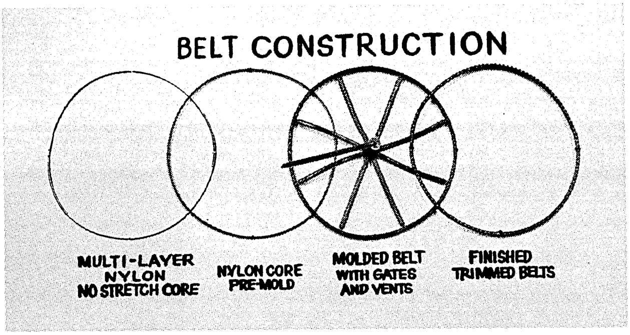


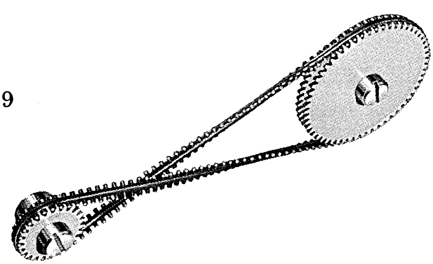
FIGURE 7-8

applications and enable it to offer the anticipated design benefits. The belts may be used at temperatures ranging from –65 degrees to 180 degrees F., within the range of most military applications.

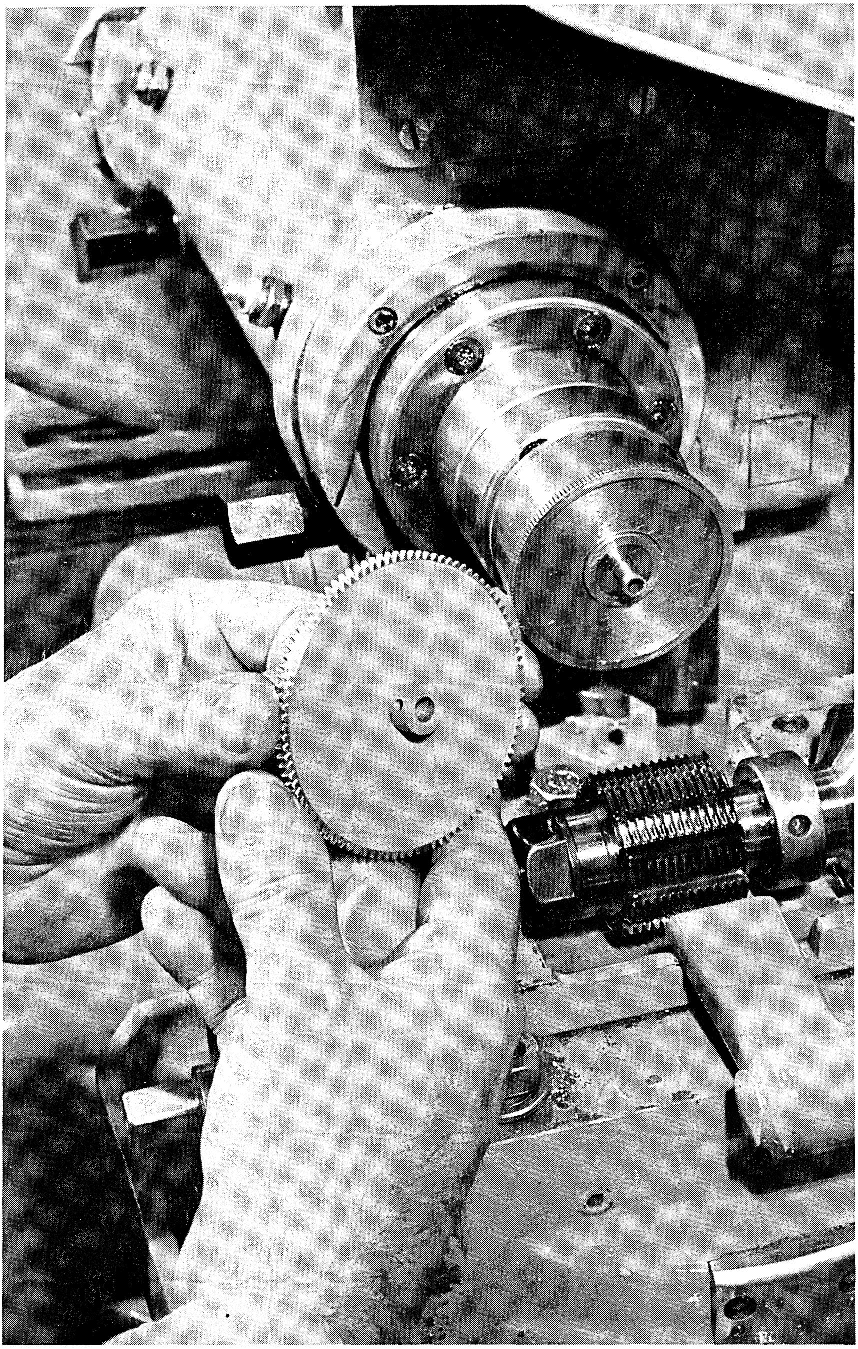
SPEED CHANGING APPLICATIONS

The "No-Slip" system provides advantages in many applications in which gears are now used. For example, in speed increasers and speed reducers, sizeable changes in speeds are presently made in

FIGURE 7-9



stages involving multiple gear meshes. Ratios greater than 1:5 are seldom used with one pair of gears. However, with the new system, ratios which may be achieved are limited only by the size of geared pulleys available. Thus, the belt and pulleys can eliminate intermediate gear meshes to produce more compact, less expensive designs. Not only are pairs of gears eliminated but the accompanying shafts and bearings are also eliminated, as well as the work involved



Geared Pulley Being Hobbed

in assembling and mounting these components.

The projections of the belt engage from approximately one-third to more than one-half of the teeth of the pulleys at any given time, depending on the difference in pulley sizes and the distance between centers. Therefore, the transmission of power is smooth and there is no cogging effect as in the case of gears, where only two teeth are in contact at any given moment. This offers smooth linear action. Also, the drive is silent, since there is no metal-to-metal contact friction or wear. This provides an ideal advantage for driving recording mechanisms, office machines, home entertainment, professional TV, motion picture and similar equipment.

REVERSING MOTION

Another common gear application which can be handled to advantage by the new transmission system is in reversing motion. This is often accomplished by an extra gear or pair of gears. It can now

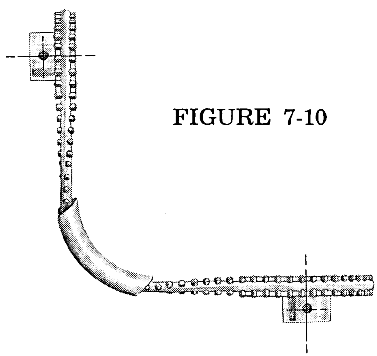


FIGURE 7-10

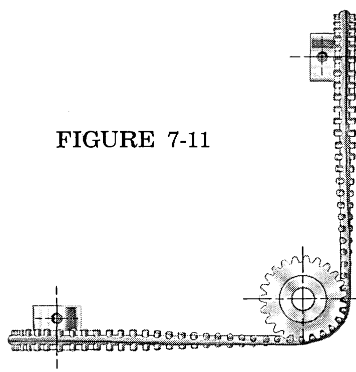


FIGURE 7-11

be accomplished simply by crossing the belt between pulleys (Figure 7-9), eliminating the need for additional gears and supporting members. In this case, pulley ratios should not exceed 1:1-1/2, otherwise the lateral belt projections will rub and cause wear.

PERPENDICULAR MOTION

The belt and pulley system can also be used in place of helical or bevel gears to transmit motion at right angles. When the right-angle shafts are in the same plane, a guide made of teflon or similar material directs the belt "around the corner" in going from one pulley to the other (Figure 7-10). The twist in the belt between pulley and guide is not harmful, nor is the rubbing against the guide. Belts under life test have shown no wear after months of continuous operation under various conditions of twist and rubbing contact with each other and guide members, as long as the lateral projections do not rub or an additional idler gear may be added at the bend (Figure 7-11).

Belts between right-angle pulleys not in the same plane require no guide (Figures 7-12 and 7-13). They merely twist appropriately in transferring motion from one plane to the other. Direction reversal is made simply by reversing the twist in the belt as it passes from one pulley to the next.

Pulley shafts do not have to be at exact right angles in using the new belt system. This provides a distinct advantage over conventional gear systems in which shafts at odd angles present expensive design and machining problems, since no special gears have to be designed and manufactured to transmit power from one shaft to the other.

ALIGNMENT OF PULLEYS

Another advantage of the plastic "No-Slip" system over gears is that pulleys on shafts not in the same plane do not have to be directly in line or above and below one another. Pulleys can be offset

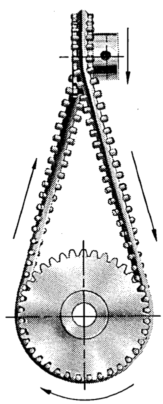


FIGURE 7-12

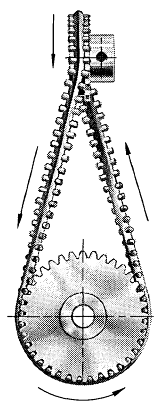


FIGURE 7-13

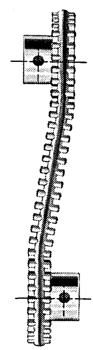


FIGURE 7-14

and the belt will travel obliquely from one pulley to the other as shown in Figure 7-14. The belt can be handled like a piece of string and run between parallel or non-parallel pulleys in the same or different planes (Figure 7-15).

In present design practice, shafts are generally parallel and in the same plane, gears are in line, and changes in direction are made in 90 degree bevel or miter gear turns. These requirements may now be eliminated in favor of freer and more direct transmission of power and motion from one device to another.

POSITIVE DRIVE

The round belt and grooved pulleys presently used to a limited extent in instrument design work do not provide positive drive but are subject to slippage under load conditions. They also slip when the belt becomes loose for any reason. To avoid slippage, round belts are sometimes installed under tension. Spring belts are also used to avoid slippage. Pre-tensioning, however, places a load on supporting pulley shaft and bearings and requires additional driving energy or torque.

The new transmission system overcomes these limitations by means of the positive engagement of pulley and belt teeth. The positive drive belt and pulley system handles loads equal to those handled by meshed gears, and the positive pin and gear engagement eliminates slippage.

The "No-Slip" system also overcomes disadvantages of the conventional cogged timing belt. Pulleys for this belt are expensive, since pulleys of different diameters require individual hobs, and side flanges are necessary to keep the belt from walking off the pulley. Side

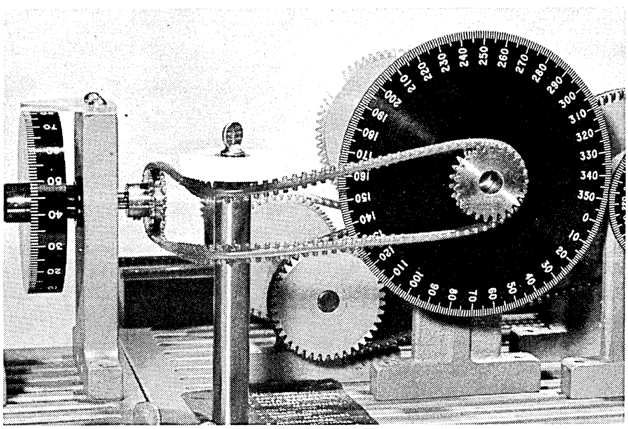


FIGURE 7-15

flanges must be machine tapered to guide the belt onto the pulley, introducing additional rubbing or scraping noise. In addition, the timing belt is not flexible laterally. Therefore, the pulleys must be in line and cannot provide the freedoms of design offered by the flexible plastic belt.

INTERMEDIATE DRIVE

Besides the design possibilities already mentioned, others offered by the new transmission system include taking power from the belt by intermediate geared pulleys positioned to mesh with either the outer or inner circumference of the belt (Figure 7-16). Most important, standard spur gears can be meshed with any geared pulley. Geared pulleys normally have 32 pitch teeth of 20 degree pressure angle, and the spacing of the lateral projections on the belts is designed to mesh with geared pulleys of this pitch. The belts will also mesh with 14-1/2 degree pressure angle gears (Figure 7-16).

In comparison with chain and sprockets, the new belt is one integral unit, rather than an assembly of multiple components. Unlike chain, the plastic belt is not subject to changes in longitudinal dimension, requires no lubrication, and is laterally flexible.

BELT SIZES

Polyurethane belts are presently available in 14 sizes, ranging from approximately 9″ to 48″ in circumference. Geared pulleys are

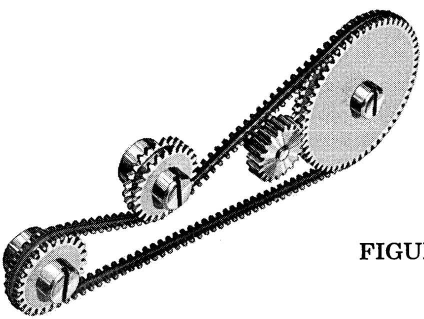


FIGURE 7-16

available in 18 sizes, having from 20 to 128 teeth and ranging in pitch diameters from .6250″ to 4″. Geared pulleys come in three standard bore sizes, 1/8″, 3/16″ and 1/4″, and are also available with 1/4″ diameter pinion-type shafts 3-3/16″ long. The pinions have 14 to 16 teeth and may be located at one end or in the center of the shaft.

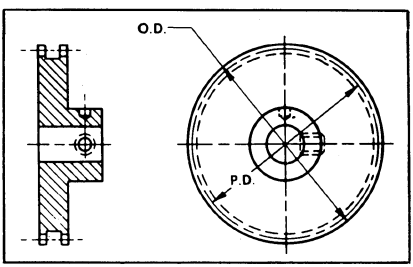


FIGURE 7-17

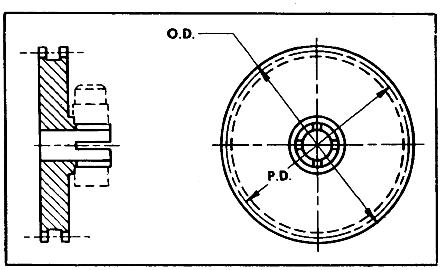


FIGURE 7-18

PULLEY TYPES

Geared pulleys are of three types – hubless, pin hub and split hub. The hub types have the hub integral with the pulley (Figures 7-17 and 7-18). With the pin-hub type, the hub is temporarily held to the shaft by a set screw while the hub and shaft are drilled through for permanent pinning. The set screw is discarded after pinning.

The hub of a split pulley has four axial slits 90 degrees apart. The hub is held to the shaft by a clamp and is easily removed or adjusted by loosening the clamp.

In using the new power transmission system, pulley centers can be located and the appropriate belt selected, taking up any slack with an idler pulley, or the pulleys can then be more accurately centered to avoid slack and take up pulley.

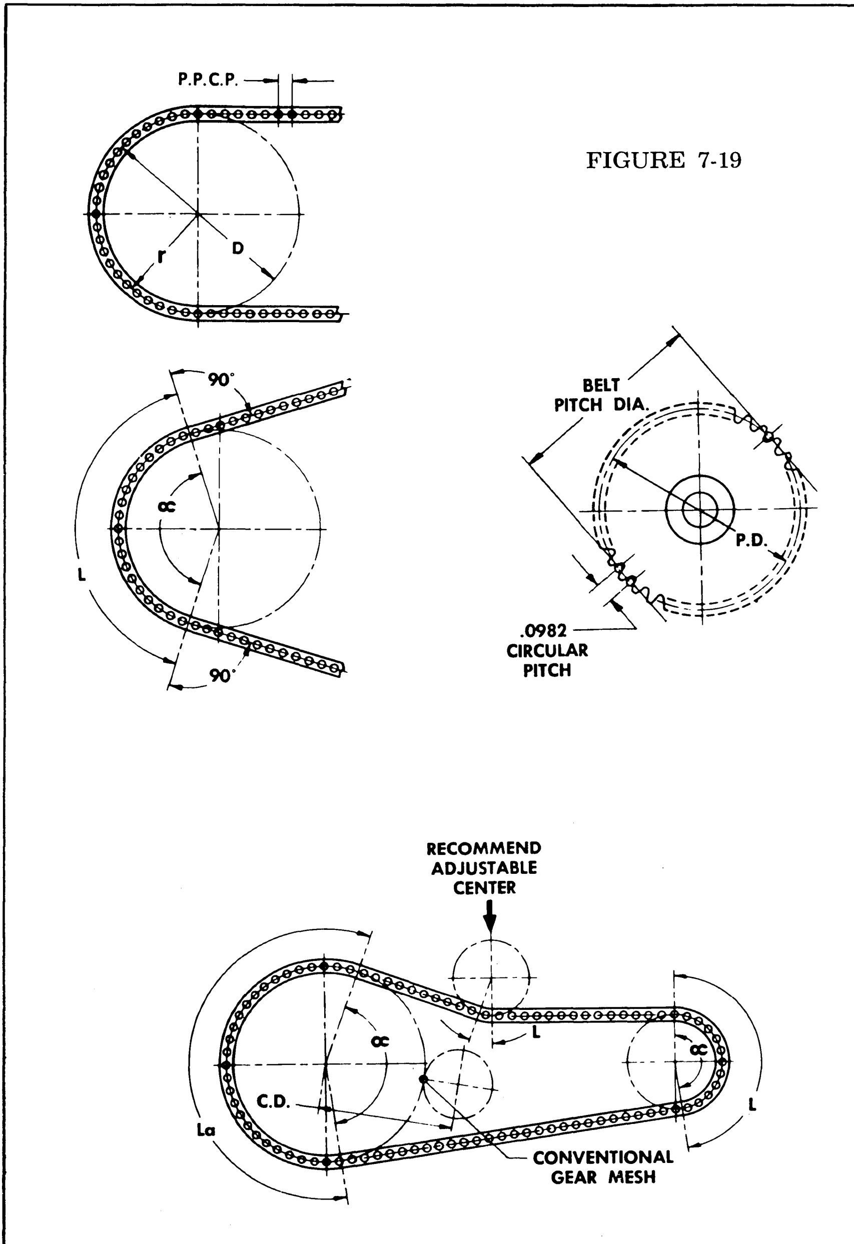


FIGURE 7-19

DETERMINING BELT LENGTH

To determine the length of belt for given center distance, the following formula is used:

$$L = \frac{NP\ (180 + 2a)}{360} + \frac{nP\ (180 - 2a)}{360} + 2D \cos a$$

L = length of belt, in inches
N = number of teeth in large pulley
n = number of teeth in small pulley
P = circular pitch of belt, in inches
D = center distance between pulleys, in inches
a = angle between tangent made by belt in contacting pulley and pulley centerline perpendicular to centerline common to both pulleys (See Figure 7-19).

$$a = \sin \frac{-1\ R - 5}{D}$$

R = pitch radii of large pulley
r = pitch radii of small pulley
Select the belt of the next larger stock size and take up the slack

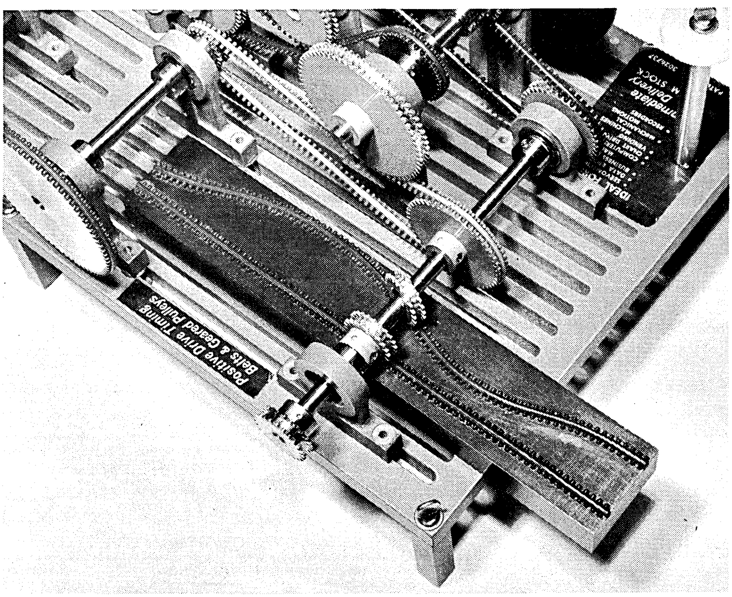


FIGURE 7-20

with an idler or by moving the pulleys further apart. (See Figure 7-19.)

To calculate the exact center distance for given pulleys for a given belt length, use the above equation to solve for D. The recommended practice is to use this center distance, making the center of one pulley adjustable.

OTHER DESIGN APPLICATIONS

In addition to transmitting power and motion between pulleys, the new belt has other design applications. Sections of belt can be secured permanently to flat surfaces with adhesives to serve as racks (Figure 7-20). Sector gears and complicated gears can be made by securing lengths of belt to the outer edges of disc segments (Figures 7-21 and 7-22) and mating with a geared pulley.

Cam blanks can similarly be fitted with lengths of belt to enable the cam follower (geared pulley) not only to advance and retract with cam rotation, but also to rotate as it follows the cam eliminating the usual cam climb and friction problem.

The new belt can be cut as a string or cable and wound around two drums offering positive cable drive for paper or chart drives. Successful experimental belt drives have been set up in which belts

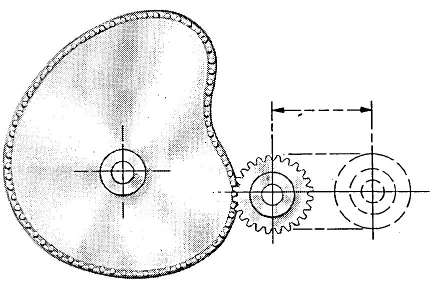


FIGURE 7-21

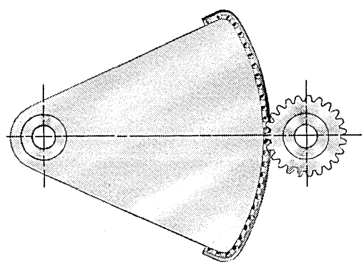


FIGURE 7-22

travel through metal tubing bent in various snake forms, or through special snap plastic channel tubing. With these guide devices, the new system is readily translated into an ideal positive three-dimensional drive.

Another extension of the "No-Slip" system would be the application of belts with a third set of projections, at right angles to existing projections, so they might be meshed with tape perforations and used as chart drives. Another possibility, where large amounts of power are to be transmitted, is the use of ganged gear pulleys and multiple belts.

Thus, the plastic belt and geared pulleys immediately open new approaches to instrument-design technique and add new possibilities to the use of these devices. The system readily lends itself to breadboard testing. As belt and pulleys become available in larger sizes, it is expected that these advantages will be extended throughout the design field.

PERFORMANCE CHARACTERISTICS OF VARIOUS DRIVE SYSTEMS

CODE: E-EXCELLENT F-FAIR G-GOOD N-NO

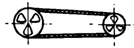
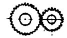
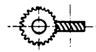
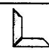
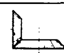
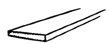
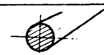
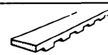
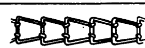
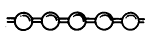
Drive System		Positive Drive	Silent Drive	Without Lubrication	Long Life Minimum Wear	No Pre-Tension	No Exact Centers	Permits Non-Parallel Shafts	Can Be Used As Pulley And Conventional Gear	Minimum Backlash	Drives Around Corners	Simple Direction Reversal	Permits High Ratios	Permits Drives Thru Bent Tubing	Inside or Outside Belt Drive	Linear Motion	Low Torque	Permits Third Plane Drive	Permits Dirt and Dust
"V" BELTS		G	E	E	G	F	F	F	F	F	F	F	E	N	N	F	F	N	E
SPRING BELTS		G	G	F	G	F	E	E	F	F	F	F	E	N	E	F	F	N	E
GEARING – SPUR		E	F	F	G	E	F	F	E	E	N	F	F	N	N	E	F	N	F
ROLLER CHAIN		E	G	G	G	E	E	E	F	F	N	F	E	N	E	F	F	N	E
GEARING – HELICAL		E	F	F	G	E	F	F	E	E	E	E	F	N	N	E	F	N	E
GEARING – BEVEL		E	F	F	G	E	F	F	E	E	E	E	F	N	N	E	F	N	F
GEARING – MITER		E	F	F	G	E	F	F	E	E	E	E	F	N	N	E	F	N	F
FLAT BELTS		F	E	E	E	F	E	E	F	F	F	F	E	N	E	F	F	N	E
ROUND RUBBER BELTS		F	E	E	E	F	E	E	F	F	E	F	E	N	E	F	F	N	E
TIMING OR COG BELTS		E	E	E	E	E	E	F	F	G	F	F	E	N	N	E	G	N	E
LADDER CHAIN		E	F	E	E	E	E	E	F	F	F	F	E	N	E	F	E	E	E
BEAD CHAIN		E	E	E	E	E	E	E	F	F	E	E	E	E	E	F	E	E	F
"NO-SLIP" POSITIVE DRIVE BELT		E	E	E	E	E	E	E	E	E	E	E	E	E	E	E	E	E	E

FIGURE 7-23

Chapter Eight

COUPLINGS

Coupling of one shaft or rotating component to another must be done properly if the accuracy of precision instruments is to be maintained. There are a number of standard couplings available, each of which will perform the coupling function. However, each coupling has features which make it better for certain applications than for others, and for optimum operation it is to the designer's best advantage to know the outstanding features of each type coupling.

The 12 most popular multi-purpose precision instrument couplings are:

1. Sleeve Coupling
2. Bellows Coupling
3. Bellows Coupling – With Zero Adjustment*
4. Oldham Coupling – Basic Type
5. Oldham Coupling – Miniature
6. Oldham Coupling – Anti-Backlash Type
7. Oldham Coupling – For Blind Assembly
8. Oldham Coupling – With Slip Clutch
9. Flexible Couplings
10. Multijaw Coupling
11. Universal Joints
12. Shaft Extensions

Bellows, Oldham and Flexible Couplings are available with pin or split hubs.

SLEEVE COUPLINGS

The *sleeve coupling* (see Figure 8-1) is essentially a short length of round stainless steel bar stock which has been bored through axially to receive the shafts to be coupled. A shaft fits into each end of the sleeve and set screws temporarily secure the coupling to the shafts while drilling for final assembly by pinning. Pilot holes guide the drilling operations.

The disadvantage of this type coupling is that it does not allow

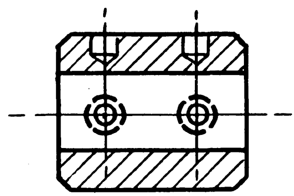
*Patented

for shaft misalignment but rigidly joins the two shafts, so that any shaft misalignment is transferred to other components in the system. Where precise accuracy is required this, of course, is not allowable. The use of sleeve couplings is *not recommended* when more than two bearings are involved.

BELLOWS COUPLINGS

The *bellows coupling* (see Figure 8-2) bends axially and allows for angular shaft misalignment up to 3° and shaft-to-shaft offset up

FIGURE 8-1



to .010″. It is made of stainless steel throughout and comprises two hubs, one for each shaft, with an interconnecting convolution-type mechanical bellows. The bellows is secured to each hub with three drive screws, 120° apart. It is available with split or pin-type hubs and, also, with a split hub at one end and a pin hub at the other. The pin hub is the most common type, but the split-hub type offers the advantage that it is easier to turn the shaft within the hub when zero-adjusting during assembly.

The bellows coupling is desirable for high-speed (to 12,000 rpm) servo-slewing applications because there is no backlash inherent in

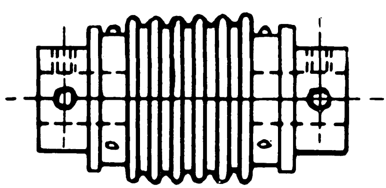


FIGURE 8-2

this type of coupling. It is not recommended for continuous loads because it has a limited torque of 10 oz. in. max. The coupling will tend to fatigue if excessive torques are applied and shaft misalignments are beyond a maximum.

BELLOWS COUPLINGS – WITH ZERO ADJUSTMENT

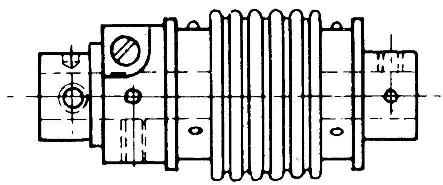
A special bellows coupling (see Figure 8-3) has been developed for making final shaft adjustments after securing the shafts in the coupling. This is especially desirable for use in servo mechanisms. One hub of the basic bellows coupling is replaced by a two-member zero-adjustment comprising a hollow stainless steel cylinder having a collar at the center and a transverse, off-center machine screw near one end, and a short length of round stainless bar stock turned down at one end and partially drilled out at the other. The turned down end has gear-like teeth near the shoulder and fits into the cylinder so that the transverse machine screw engages the teeth. The other end of the cylinder fits into the flange of the bellows and is secured with special drive screws.

One shaft to be coupled fits into the hole drilled in the short-length-of-bar stock member and is pinned in place, a set screw and guidehole for drilling being provided. The other shaft fits into the hub-end of the coupling and is pinned in place. (See Figure 8-4.) Then, by turning the transverse machine screw, one shaft may be turned with reference to the other to obtain an exact relation or synchronization with it. When this is obtained, the two parts of the special assembly are locked with a set screw, drilled as indicated by a pilot hole, and pinned for in-service use. Thus, perfect synchronization is obtained quickly, accurately and permanently.

OLDHAM COUPLINGS

The basic *Oldham coupling* (see Figure 8-5) comprises three interlocking disc-like members. The two end-members have integral hubs and are secured to the shafts to be coupled. The center disc engages the end-discs in such a manner that it "floats" and enables

FIGURE 8-3



the coupling to accommodate shaft-to-shaft offset up to .010″. The center disc has a wide groove or channel machined across each face, one at right angles to the other, and the outer discs have projections which fit into the slide in these grooves. This enables the center member to shift position during rotation as may be required by shaft misalignment, at the same time maintaining the coupled relationship.

Since this coupling does not bend axially except for a slight amount of play between parts, not more than 1° of angular misalignment compensation can be obtained. Also, due to the play between parts, there is some backlash (not more than 8 minutes). Maximum speeds at which this coupling is recommended is 2500 rpm. Torque rating is 100 oz. in. It is available in split and pin-hub types.

THE OLDHAM COUPLING – MINIATURE

For applications where space is at a premium, a miniature Oldham coupling is available (see Figure 8-6). It has an oil-less bronze center member (or nylon if preferred) to avoid sliding of like metal on like metal. The design is slightly different in that the center member is basically a square with the corners machined off.

One of the coupling's end pieces has a projection that fits into a groove in one face of the center-piece, just as in the basic Oldham coupling. However, the coupling's other end-piece has a wide groove instead of a projection, and the center-piece fits into and slides in this groove. This allows the same degree of motion as in the basic design, but takes up less space. This unit accommodates axial mis-

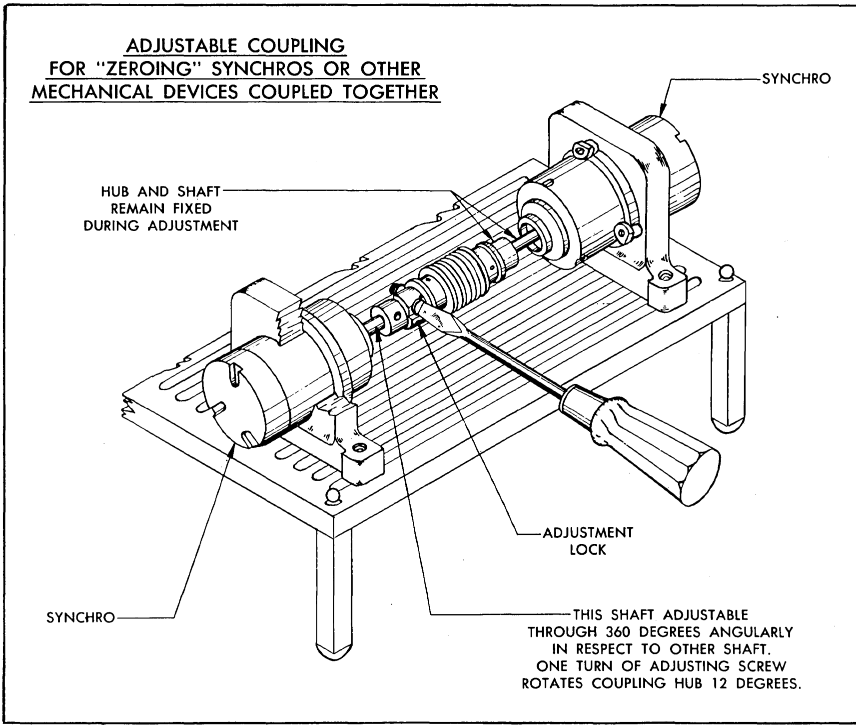


FIGURE 8-4

alignment up to 1°, has maximum backlash of 10 minutes and is recommended for speeds to 1000 rpm.

THE OLDHAM COUPLING – ANTI-BACKLASH TYPE

To avoid backlash in an Oldham coupling, an anti-backlash type is available, as shown in Figure 8-7. It is larger in diameter than the other types and is relatively short. It comprises three thin interrelated discs, the outer ones with hubs. One of the outer discs has a stud riveted near each end of a diameter. These studs extend inward through slots in the middle disc and through holes in the third disc. Stud-mounted springs between the discs space the discs apart, and retainer rings at the ends of the studs hold the assembly together.

Studs are also secured to the third disc. They are near the ends of a diameter, which is 90° from the one on which the through holes for the first pair of studs are located. The second pair of studs extend inward through slots in the center disc. They also project outward, and a spring connects each outward projection with the end of one of the retainer-ring studs, just beyond the retainer ring. This construction provides a floating third disc, and the spring loading pre-

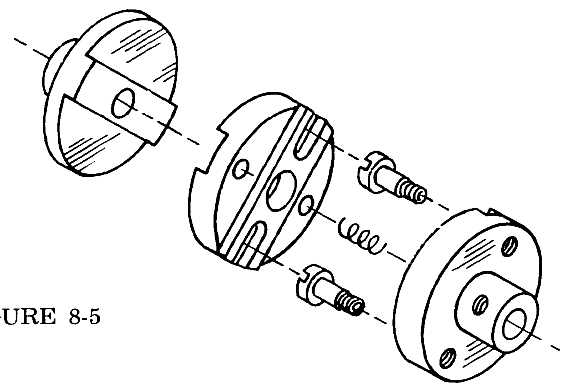


FIGURE 8-5

vents backlash. This coupling also deflects axially, holes being drilled in the first disc to allow the studs secured to the third disc to pass partially through and avoid restriction of the axial movement of the middle and third disc with respect to the first.

OLDHAM COUPLING—FOR BLIND ASSEMBLY

A modified Oldham coupling (see Figure 8-5) is available for applications where the coupling must be installed blind in hard-to-

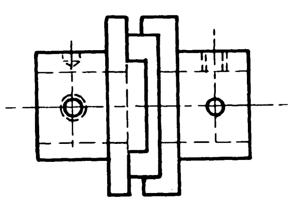


FIGURE 8-6

reach places. It is similar to the basic Oldham coupling, but one face of the center member has a projection instead of a channel, and this projection keys into a channel in one of the mating end-discs.

The center disc has through "U" slots at the ends of the projection, and special machine screws fit into these slots and screw into the tapped holes in the mating outer disc. Non-threaded shanks of the machine pass through stepped down portions of the "U" slots, the wider portions accommodating the heads. Thus, the center disc can slide back with respect to the outer disc, but cannot fall off. Relation of the center disc to the other end disc is as in the basic Oldham coupling. The three basic parts now are two, which can readily be assembled with the shafts to be coupled without dropping the center piece down into the chassis. To accommodate the special screws, the basic coupling diameter has been increased to 1". The length, however, remains approximately the same. Maximum recom-

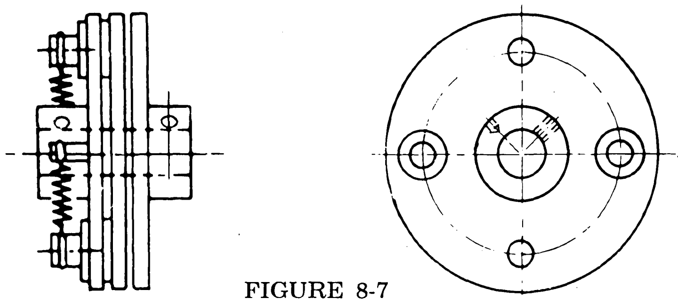


FIGURE 8-7

mended rpm is 2000. Maximum backlash is 8 minutes. The coupling can accommodate angular misalignment to 1°. Torque rating is 150 oz. in.

OLDHAM COUPLING – WITH CLUTCH

The Oldham coupling with clutch has a clutch mechanism replacing one of the disc-and-hub components of the basic Oldham coupling, as shown in Figure 8-8. When the clutch is engaged, the two shafts are coupled; when it is disengaged, they are uncoupled.

The clutch mechanism consists of two major assembled parts – a hollow cylinder, one end of which engages the coupling's center

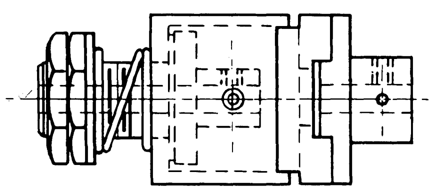
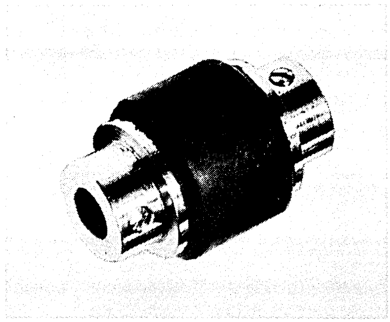


FIGURE 8-8

member, and a sleeve, into which one shaft fits. The sleeve has a collar and is assembled with the cylinder so the collar is inside the cylinder, the sleeve projecting outward through a hole in the cylinder and wall. A cork washer is installed on the sleeve between the collar and the end wall. The end of the cylinder side wall which engages the coupling's center member is machined so there are two opposing projections which fit into a channel in the coupling's center disc. Thus, when the sleeve is positioned so the collar pulls the cork disc against the cylinder wall, the cylinder turns with the shaft, and couples it with the other shaft pinned to the hub of the standard disc-and-hub member of the coupling. When the cork disc is not in contact with the cylinder, the sleeve turns in the cylinder end-wall opening, and the shafts are not coupled. The shaft that fits into

FIGURE 8-9



the sleeve is pinned in place, an access hole being provided in the cylinder side wall for set screw turning, for drilling the pin hole and installing the pin.

The clutch is spring loaded so the coupling is normally engaged. Spring loading is provided by a coil spring which fits on the portion

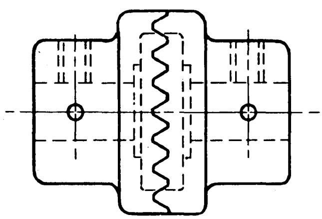
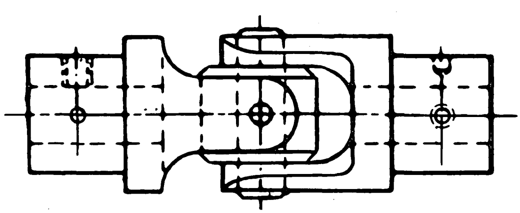


FIGURE 8-10

of the sleeve projecting outside of the cylinder. One end of the spring pushes against the end of the cylinder, and the other end is confined by a nut that fits on the threaded external end of the sleeve. The nut is locked in place by a second nut that is tightened against it.

FIGURE 8-11



FLEXIBLE COUPLINGS

The flexible coupling comprises two hubs, either pin or split, and an interconnecting length of neoprene, as shown in Figure 8-9. This coupling isolates torsional vibration and insulates between units. It allows shaft to shaft and angular misalignment.

MULTI-JAW COUPLING

Multi-Jaw couplings are available for engaging and disengaging shafts. (See Figure 8-10.) This coupling essentially comprises a pair of pin-type hubs with interlocking teeth in the engaging faces. They are rated for 200 oz. in. torque. They measure 7/8″ long by 3/4″ in diameter. A smaller miniature multi-jaw coupling, 0.79″ long by 17/32″ in diameter, is available for use when minimum space is available.

UNIVERSAL JOINTS

Universal joints with pin or split hubs are available for use when one shaft meets another at an angle of up to 30°. (See Figures 8-11 and 8-13.)

SHAFT EXTENSIONS

Shaft extensions are provided for use when it is desired to lengthen a shaft. (See Figure 8-12.) The extension has an enlarged shoulder at one end and this shoulder is drilled out to receive the shaft that is to be lengthened. An end section of the shoulder is turned down and slotted for clamping to the shaft that is to be extended.

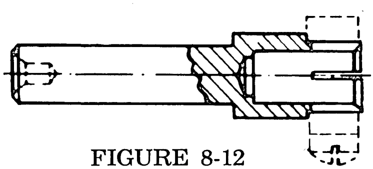


FIGURE 8-12

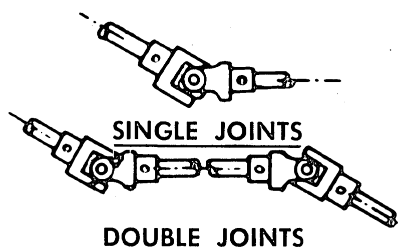


FIGURE 8-13

CYCLE ERROR

All couplings are designed to *permit* driving conditions whenever there is a shaft-to-shaft or angular misalignment error. However, there will always be a *cycle error*.

For example, shaft A which is misaligned .100″, travels through a 30° angle. Then shaft B will not follow through the 30° angle but at some larger angle. When shaft A has traveled 45°, shaft B will have traveled at an angle greater than 45°. Thus, it is obvious that there will always be a *cycle error*. The bigger the angular or shaft-to-shaft misalignment, the greater the cycle error. Couplings *permit* but do not cause angular and shaft-to-shaft misalignment. They cannot eliminate the *cycle error*.

Chapter Nine

MAGNETIC CLUTCHES

There are many types of clutches for engaging and disengaging shafts, but the one generally used in automatic control systems is the *magnetically* operated friction-disc type. This clutch best achieves the speed of response, long life, and reliability necessary for modern automatic control systems.

The *magnetic clutch* (see Figure 9-1) is a rotating electro-magnetic device having input and output shafts. It looks like a small motor with a shaft coming out of each end, or two concentric shafts coming out of one end. Coupling between the shafts is controlled by

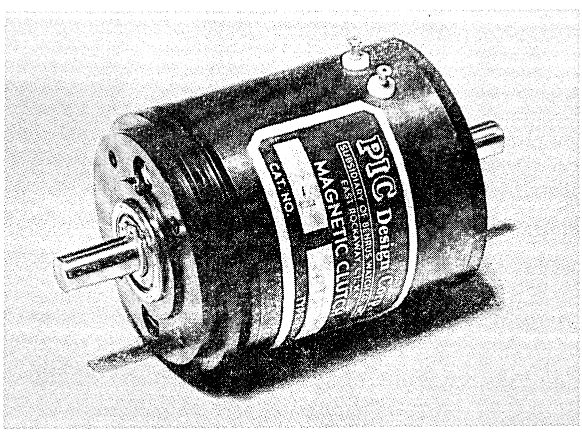


FIGURE 9-1

a magnetic field set up by passing DC through a coil built into the clutch housing. (See Figure 9-2.)

Input and output shafts have fixed magnetic cylinders attached to adjacent ends. The output shaft is free to slide axially along its shaft. Under the action of the applied magnetic field (DC), the output cylinder makes strong physical contact with the input cylinder, thus coupling the two shafts. Rotation of the input shaft results in rotation of the output shaft. Upon removal of the DC signal, a spring forces the moveable cylinder away from the input cylinder and opens the coupling.

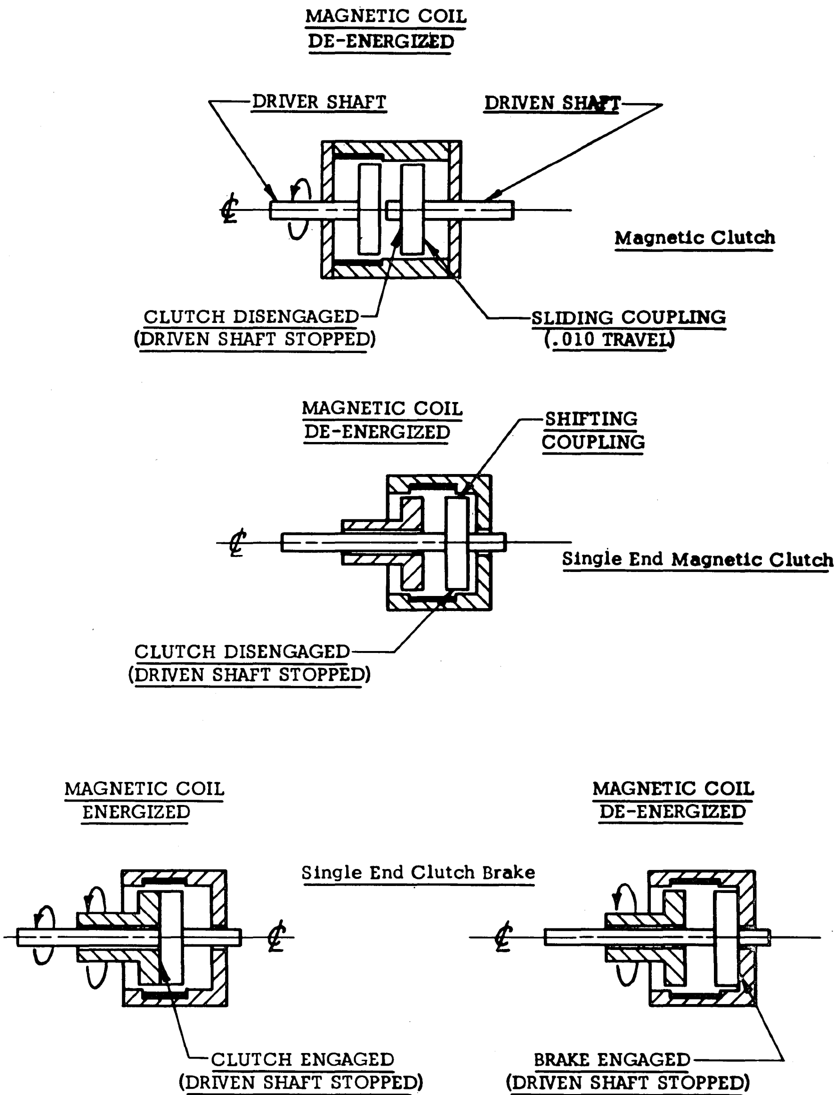

MAGNETIC COIL
DE-ENERGIZED
DRIVER SHAFT
DRIVEN SHAFT
Magnetic Clutch
CLUTCH DISENGAGED
(DRIVEN SHAFT STOPPED)
SLIDING COUPLING
(.010 TRAVEL)
MAGNETIC COIL
DE-ENERGIZED
SHIFTING
COUPLING
Single End Magnetic Clutch
CLUTCH DISENGAGED
(DRIVEN SHAFT STOPPED)
MAGNETIC COIL
ENERGIZED
Single End Clutch Brake
MAGNETIC COIL
DE-ENERGIZED
CLUTCH ENGAGED
(DRIVEN SHAFT STOPPED)
BRAKE ENGAGED
(DRIVEN SHAFT STOPPED)

FIGURE 9-2

MAGNETIC CLUTCH BRAKE

The magnetic clutch brake (see Figure 9-3) is similar to the magnetic clutch. It has the added feature that the output shaft is positively stopped and arrested when disengaged. The output shaft is mechanically locked to an opposing braking face on the frame by a spring when the DC signal is removed.

The magnetic field of the magnetic clutch must be sufficiently strong to overcome the spring force, and at the same time yield good firm contact between the clutching surfaces. The spring, on the other hand, must be strong enough to overcome the holding action of residual magnetism when the DC signal has been removed. In the clutch brake the spring must not only open the clutch, but must also hold the output disc against the braking surface in opposition to any torque imposed by the driven device.

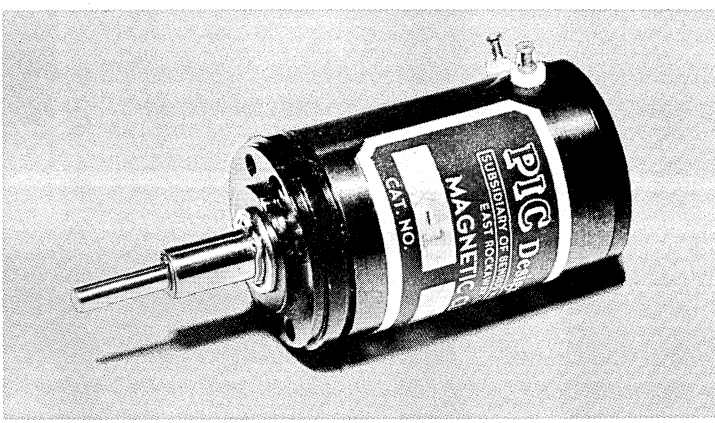


FIGURE 9-3

Standard magnetic clutches and clutch brakes for control systems are designed for operation from a 24- to 28-volt DC source and consume three watts of power. Special units are available for voltages ranging from 1.5 to 300 volts DC.

EFFECTS OF VOLTAGE VARIATIONS

Variation of applied voltage will affect torque output, as shown in the typical curves in Figures 9-4 and 9-5 and makes the magnetic clutch valuable as a speed changer. Reduction of the voltage will result in clutch slippage and lower output speed, but output torque will remain relatively high. Specific curves and specification tables for clutches of various capacities, as well as data explaining how output torque, breakaway torque, inertia and other specification data are derived, are available from magnetic clutch manufacturers.

In addition to developing high output torques at intermediate applied voltages, magnetic clutches provide angular accelerations

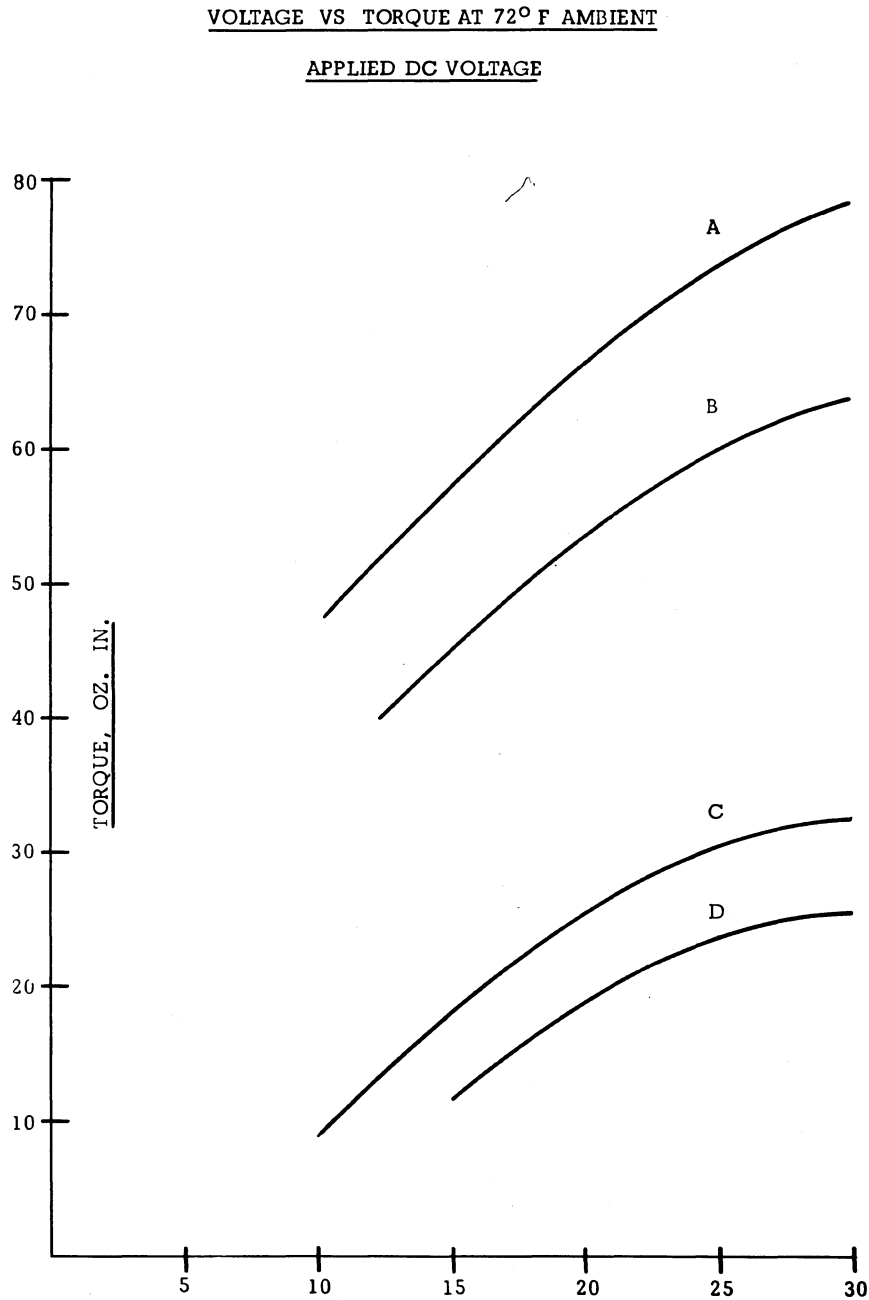

VOLTAGE VS TORQUE AT 72° F AMBIENT
APPLIED DC VOLTAGE
80
70
60
50
40
30
20
10
TORQUE, OZ. IN.
A
B
C
D
5
10
15
20
25
30

FIGURE 9-4

far exceeding those of conventional motors. They also can reduce shock loading such as incurred during sudden reversal of high-inertia loads, or when engaging limit stops. They may be incorporated into the design of speed changers, rapid cycling systems, accurate shaft positioners and other specialized devices.

HIGH HOLDING TORQUE

When extra high holding torques are required, toothed faces can be supplied on clutching or braking surfaces. Such couplings provide torques up to four times that of smooth-face types. They are not recommended where slipping is required or where angular displacement between shafts cannot be tolerated. It is necessary for teeth to line up in order to engage and hence an angular error is introduced. Also, the teeth create a ratcheting effect in engaging while

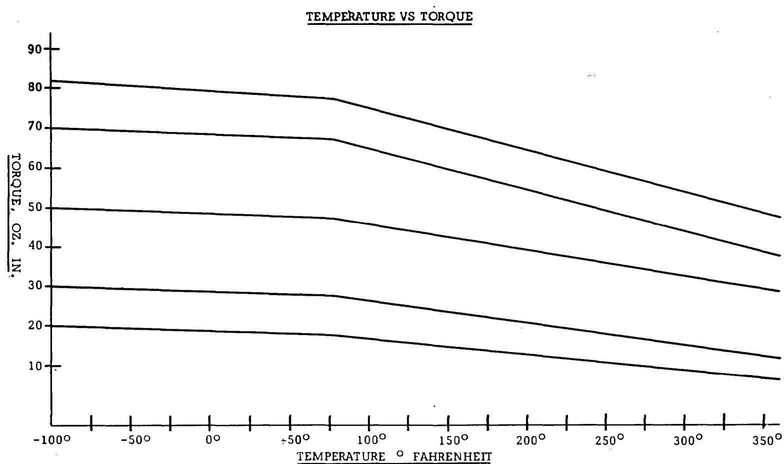


FIGURE 9-5

rotating, which results in damaging the teeth, reducing their life and reducing torque characteristics. Crown tooth magnetic clutches are not recommended for speeds over 300 rpm.

TIME DELAY

The time required for the clutch to engage after the coil has been energized is caused, for practical purposes, by *spring resistance* to the magnetic pull set up by the energized coil. Typical delays are indicated by curves B and C in Figures 9-6 and 9-7. Delays as low as three milliseconds can be achieved, as indicated in Figures 9-4 and 9-5, by special adjustment during assembly of the gap between discs and the tension in the return spring.

There is some loss of output torque at relatively high operating temperatures, as indicated by the curves in **Figure 9-8**.

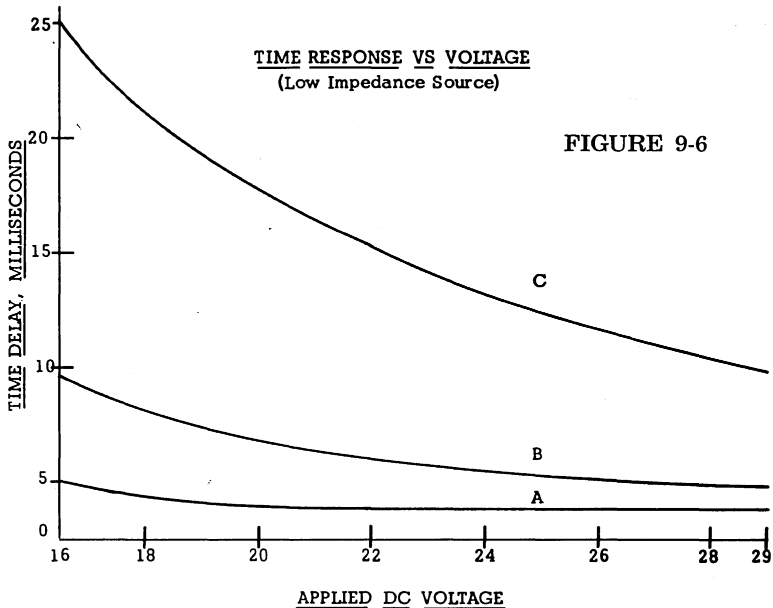


FIGURE 9-6

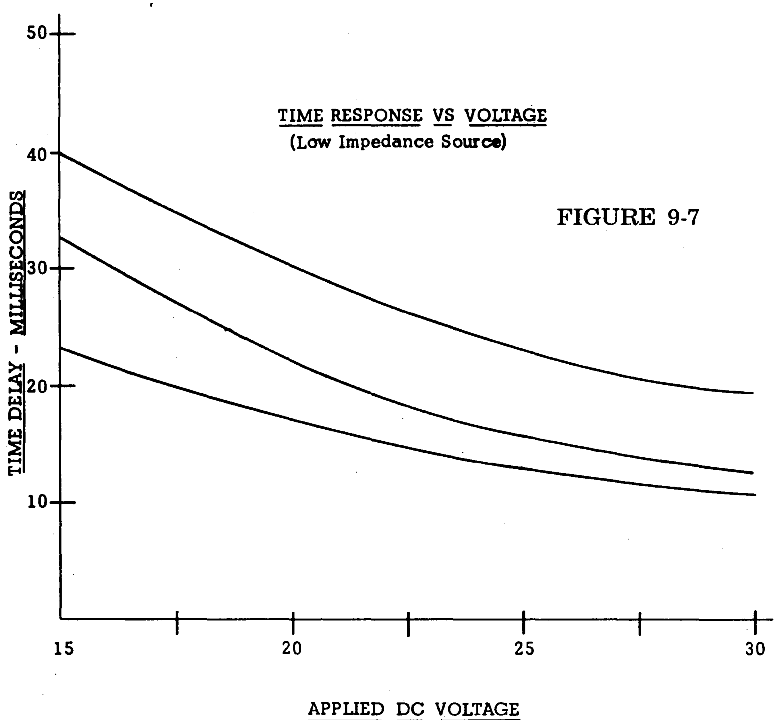


FIGURE 9-7

SELECTING MAGNETIC CLUTCHES

In purchasing magnetic clutches, detrimental factors to avoid include:

1. *Soft metal clutch faces* which quickly break down creating small particles that destroy torque output and eventually get into the bearings.
2. *Composition faces* which have the following inherently poor characteristics: relatively short life; slippage when wet; are fungus nutrients; harden, glaze, burn or slip when hot.
3. Metal faces of *corrosive material* which corrode, destroying initial torque. An attempt to correct this is sometimes made by plating with

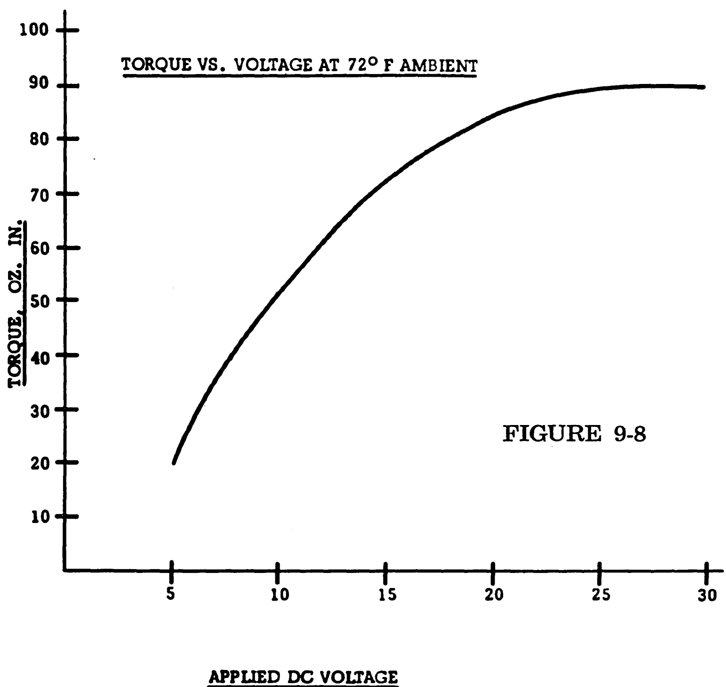


FIGURE 9-8

chrome. If the chrome is thick enough to be wear resistant the torque is reduced. When thin, the chrome quickly chips off, exposing the base metal to corrosion and all of the difficulties of soft faces mentioned above.

4. *Eccentricity* of moving parts which creates camming (rotation of shafts when turning clutch on and off), uneven breakaway torque and eccentricity of shafts when clutch is turned on.

In contrast, hardened metal clutch faces will provide high torque and long life, even when slipped continuously, corrosion resistance,

resistance to effects of temperatures between −100°F and +400°F, no loss of torque when wet, and no nutrients for fungus.

Additional magnetic clutch features required for precision instrument application include:

1. Concentricity of .001 and squareness of less than 5′ of shafts to mounting diameters and faces.
2. Low breakaway torque.
3. No rotation of shafts when clutch is turned on and off.

Chapter Ten

DIALS

Dials indicate shaft positions. They are usually read against fixed indexes. Two types of dials commonly used are the disc dial and drum dial. (See Figures 10-1 and 10-2.)

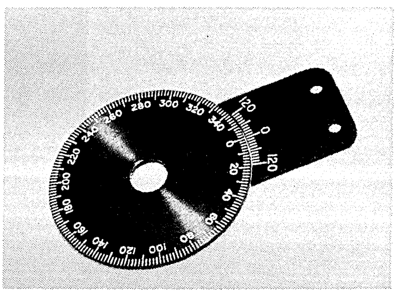


FIGURE 10-1

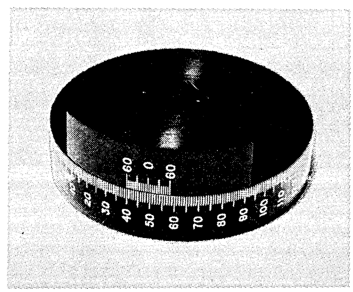


FIGURE 10-2

DISC DIALS

The *disc dial* has equi-spaced radial graduations precisely engraved around its circumference. Standard discs range in diameter

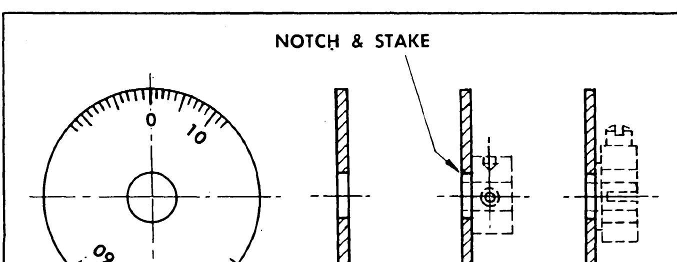


FIGURE 10-3

from 1-1/2″ to 4″ and are available with from 10 to 360 graduations, clockwise or counter-clockwise. The disc dial is generally staked to a pin hub or split hub by the manufacturer. The split hub enables the dial to be shifted or zeroed out with relation to the shaft whereas the pin type hub is fixed once pinned. (See Figures 10-3 and 10-4.)

The *disc dial index* is a metal or clear plastic plate with an engraved index line. It is mounted in a fixed position on the indicator panel. Metal plates are mounted alongside the dial so the index lines up with graduations as they rotate into position opposite it. Clear plastic indexes extend over the dial and the index line coincides with graduation lines as they come into position under it.

Vernier indexes are available for particularly accurate readings. Discs for vernier readings are graduated in one or two degree increments and the index, or vernier, is marked off in six- or twelve-minute

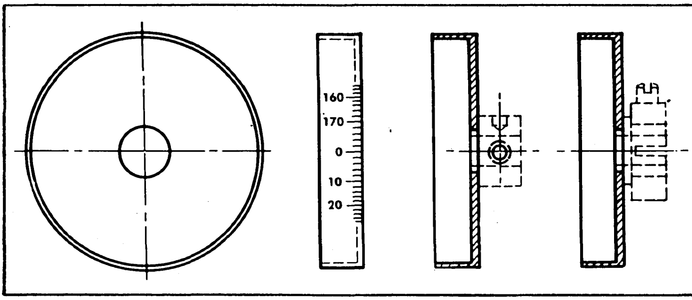


FIGURE 10-4

graduations. When a given graduation on the disc lines up with a given graduation on the vernier, the total reading is the degrees indicated by the disc graduation plus the minutes indicated by the vernier graduation, as with micrometers. (See Figure 10-5.)

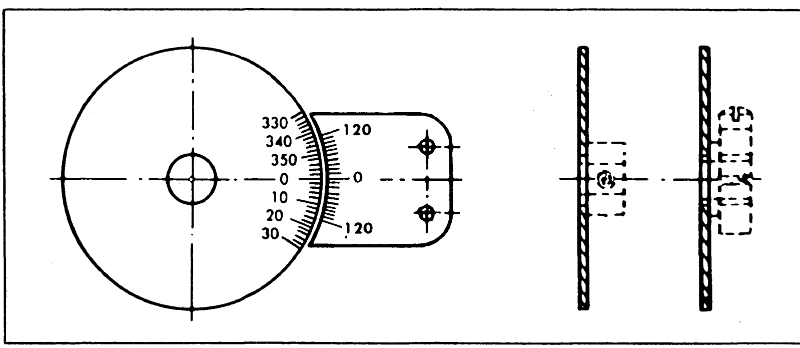


FIGURE 10-5

DRUM DIALS

The *drum dial* has equi-spaced graduations engraved parallel to its axis around its cylindrical surface. It also comes ready-staked to a pin or split hub of various bore sizes. It is the preferred type for breadboard applications.

Standard hangers are available for mounting drum dials in breadboards. Transparent plastic indexes may be mounted to the tops of hangers to extend over the dial, and drum dial indexes and verniers are made to fit between the hanger (or panel) and the dial, as shown in Figure 10-6.

TOLERANCES

Angular tolerances of dials graduated in one degree steps are within ten minutes. Dials graduated in two degree steps or greater are within fifteen minutes of nominal.

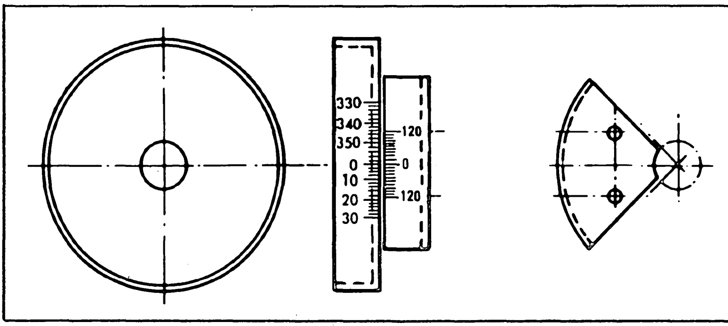


FIGURE 10-6

STANDARDS

In order to standardize on engraving of characters, height, width, thickness, depth and length of lines, basic standards have been established and are most commonly used by engravers and users. (See Figures 10-7 and 10-8.)

DISC DIALS & INDEXES

Dia. of Dial	CHARACTERS				LINES		
	Height	Width	Thickness	Depth	Long	Med.	Short
1½″	3/64 R	.035	.010	.005	7/64	–	1/16
2″	1/16 R	.050	.012	.006	1/8	–	1/16
3″	3/32 R	.080	.012	.006	1/8	–	1/16
4″	1/8 R	.092	.014	.007	3/16	–	1/8

Average Depth of Engraving .006 Line width .010

SERIES DISC DIALS & VERNIERS

Dia. of Dial	CHARACTERS				LINES		
	Height	Width	Thickness	Depth	Long	Med.	Short
1½″	3/64 C	.025	.010	.005	7/64	–	1/16
2″	1/16 C	.030	.012	.006	1/8	–	1/16
3″	5/64 C	.035	.012	.006	1/8	3/32	1/16
4″	3/32 C	.050	.012	.007	3/16	1/8	1/8
LENGTH VERNIERS LINES					1/8	3/32	1/16

Average depth of engraving .006 Line Width .008

R=Regular Gothic Type

C=Condensed Gothic Type

MATERIAL — Aluminum

FINISH — Black Anodized

Engraving Filled With White Enamel

Angular Tolerance

1° Steps=Within 10′

2° Steps or More=Within 15′

Series=Within 10′

FIGURE 10-7

DRUM DIALS & INDEXES

Dia. of Dial	CHARACTERS				LINES		
	Height	Width	Thickness	Depth	Long	Mod.	Short
1″	5/64 C	.055	.012	.005	1/8	—	1/16
1½″	5/64 C	.055	.012	.005	1/8	—	1/16
2″	5/64 C	.055	.012	.005	9/64	—	5/64
2½″	3/32 R	.080	.014	.006	5/32	—	3/32
3″	3/32 R	.080	.014	.006	5/32	—	3/32

Average depth of engraving .006 Line width .010

SERIES DRUM DIALS & VERNIERS

Dia. of Dial	CHARACTERS				LINES		
	Height	Width	Thickness	Depth	Long	Mod.	Short
1½″	5/64 C	.055	.012	.005	1/8	—	1/16
2″	5/64 C	.055	.012	.005	9/64	—	5/64
2½″	3/32 R	.080	.014	.006	5/32	—	3/32
3″	3/32 R	.080	.014	.006	5/32	—	3/32

Average Depth of Engraving .006 Line Width .008

R=Regular Gothic Type
C=Condensed Gothic Type

MATERIAL — Aluminum
FINISH — Black Anodized
Engraving Filled With White Enamel

Angular Tolerance
1° Steps=Within 10′
2° Steps or More=Within 15′
Series=Within 10′

FIGURE 10-8

Chapter Eleven

SERVO BREADBOARDS

The availability of some 20,000 assorted, standardized precision instrument components make possible a relatively new design technique which eliminates detailed mechanical drawings, models and prototypes. Servo system components are mounted on metal plates

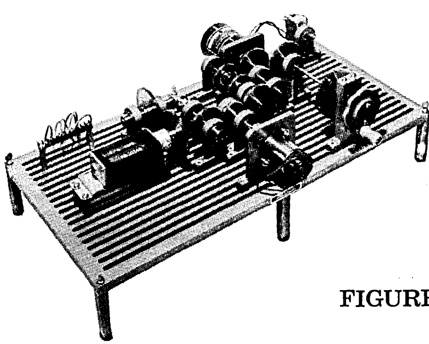


FIGURE 11-1

according to a template-made drawing and are adjusted or changed until the system is perfected and ready for production. (See Figure 11-1.)

Components are mounted in vertical plates, or hangers, with cutouts for bearings and electrical components. The hangers have

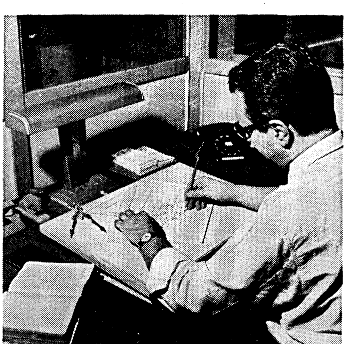


FIGURE 11-2

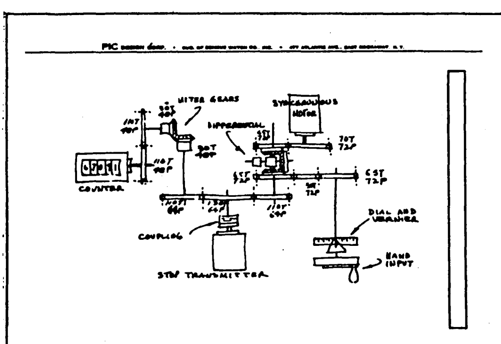


FIGURE 11-3

feet with tapped holes. The feet are secured to the breadboard plate through slots from below. Hangers are readily available for standard-size components. Blank hangers can be machined to size for mounting special components.

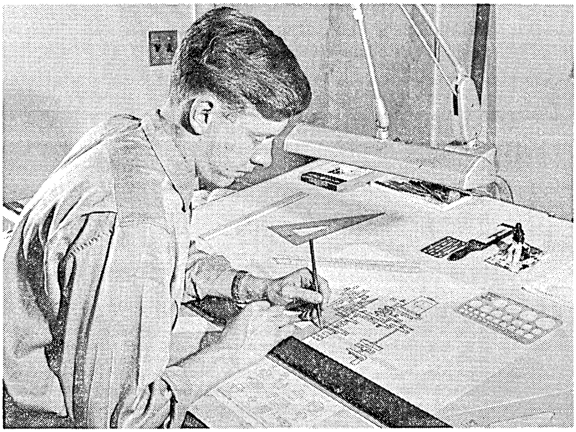


FIGURE 11-4

The first step in using the breadboard technique is to prepare a rough sketch of the proposed servo system as shown above. Normally the next step would be to prepare a scale drawing from the rough sketch. By using templates, however, a draftsman can prepare a working drawing from which to assemble components in a fraction

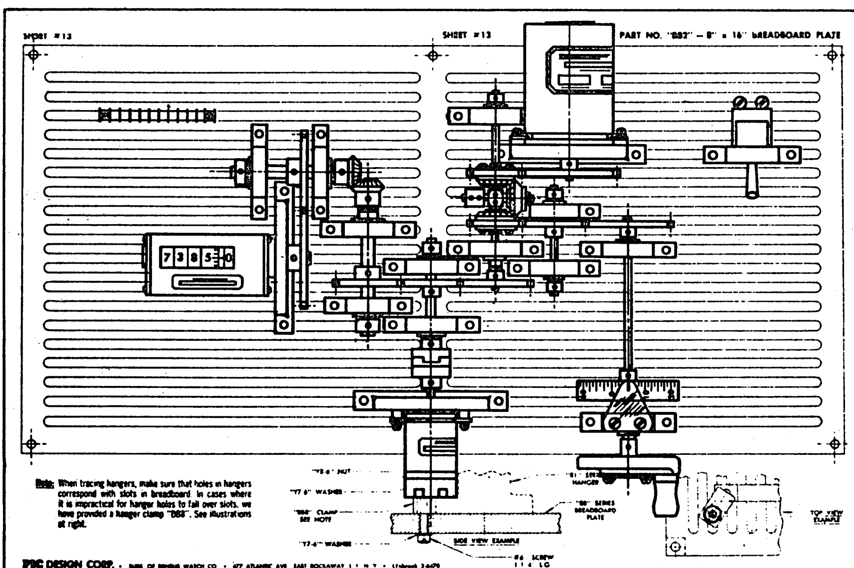


FIGURE 11-5

of a day. (See Figures 11-2 and 11-3.)

The templates comprise full size outlines of precision instrument components on light-weight paper. The draftsman slips these under his drawing paper in the position the various components will occupy

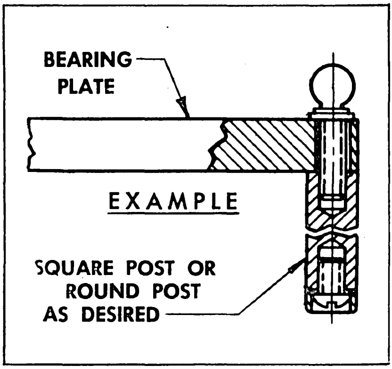


FIGURE 11-6

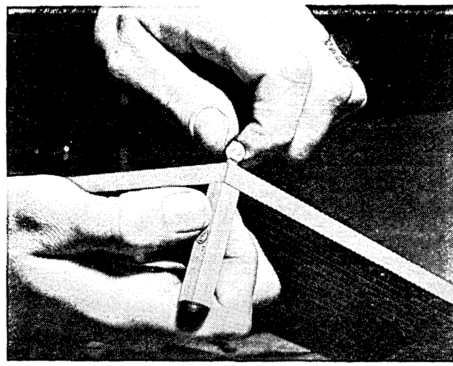


FIGURE 11-7

according to the rough sketch, and traces them off. Templates are available free of charge from component suppliers. (See Figures 11-4 and 11-5.)

After the drawing has been finished, the necessary parts are ordered. They are often shipped the same day the order is received

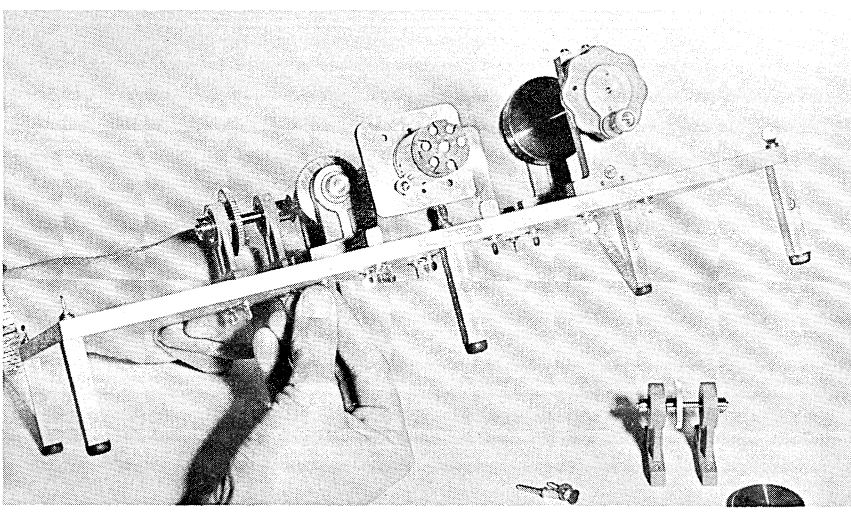


FIGURE 11-8

and should be in the hands of the assembly group within a matter of days.

The first step in assembling a breadboard is to assemble the leg posts to the slotted mounting plate as shown in Figures 11-6

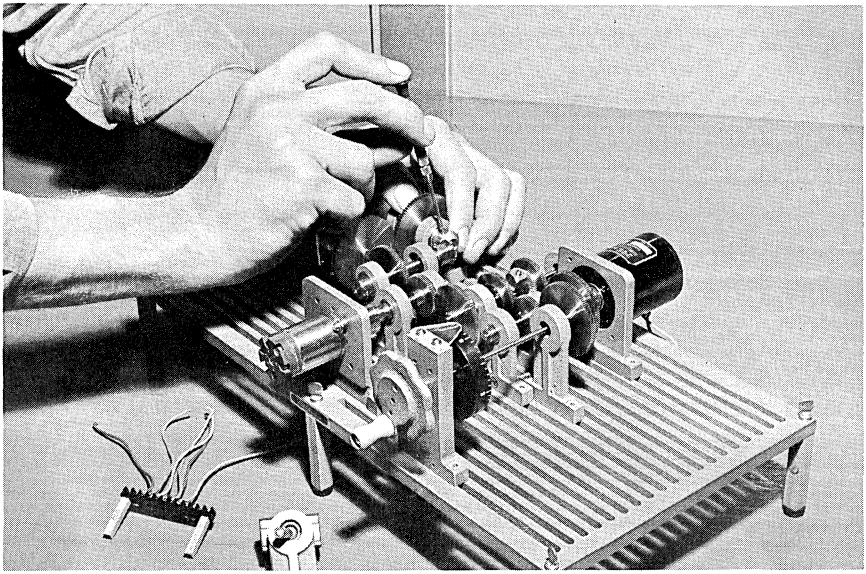


FIGURE 11-9

and 11-7. This is accomplished by inserting wing nut screws from above into plate cornerholes and tightening the screws into holes tapped in the upper ends of the leg posts. The leg posts have rubber feet to prevent marring finished surfaces or slipping on them.

Next, spur gears, couplings, differentials, clutches, etc., are fitted

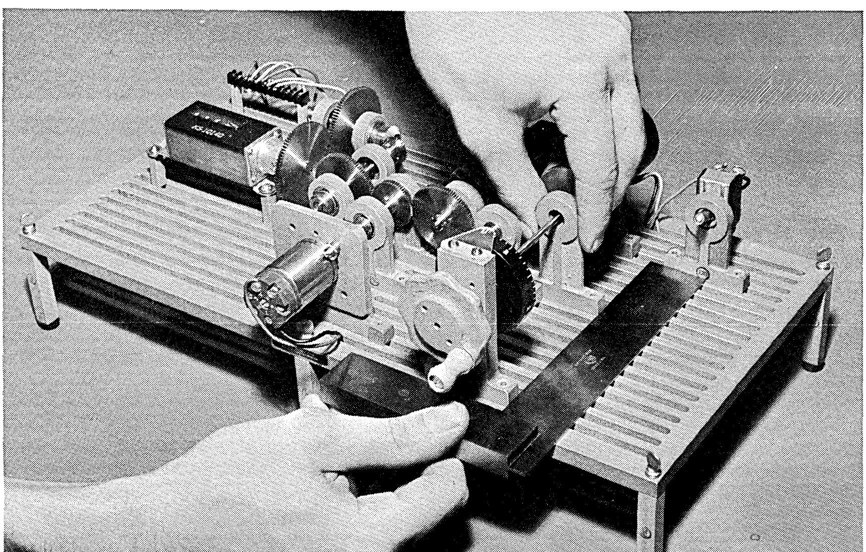


FIGURE 11-10

to the shafts of electrical components. The electrical components are then secured in component hangers, and the hangers mounted on the board in the approximate position indicated on the drawing. (See Figure 11-8.)

Thumb screws extend through the slots from below into tapped holes in the hangers. Thumb screws are not securely tightened until all equipment has been mounted and can be accurately aligned. General practice is to start installing at one side of the board and work across. (See Figure 11-9.)

Again working across the board, shafts are fitted with gears, shaft hangers, and other components such as differentials, dials, cams, etc., and mounted in place according to the drawing. (See Figure 11-10.)

Once all of the equipment has been assembled and placed on the board, it is aligned with a square using the machined edges of

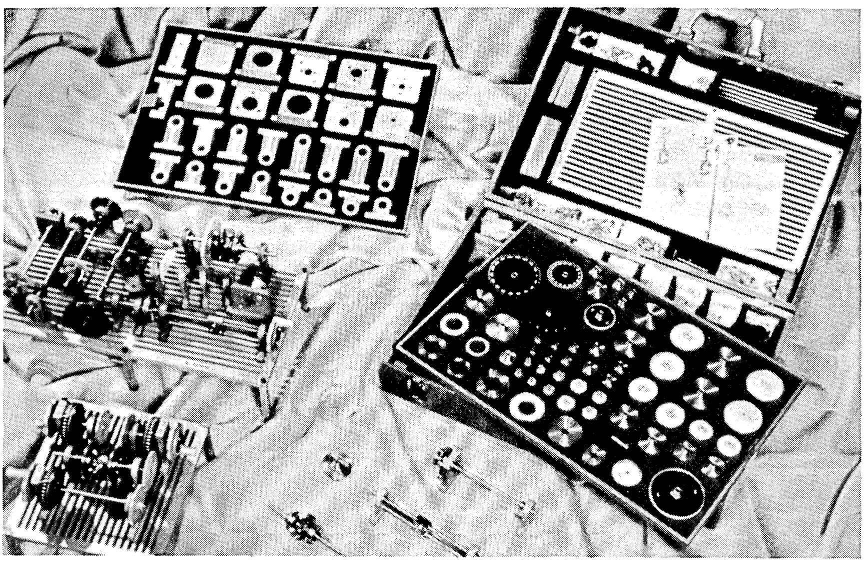


FIGURE 11-11

the slotted plate as a guide. The thumb screws are then securely tightened. Terminal strips are available to facilitate wiring of electrical circuits.

Once the breadboard has been completed, it can be operated under all electrical or mechanical conditions which might be encountered in practice. Weak points can be corrected, improved spacing arranged, and a complete range of operating characteristics obtained for evaluation and study.

Breadboard kits are available for use when frequent breadboard construction is encountered. These assortments reflect the requirements of many component users over a period of time. (See Figure 11-11.)

Chapter Twelve

TOOL AND FIXTURE COMPONENTS

This chapter has been prepared for the purpose of affording the general designer and engineer a clearer concept of standardized tool components readily available in today's market.

This knowledge of tooling applications should not be limited to the tool designer alone, but made available to every mechanical and electronic designer, as a valuable adjunct toward a better understanding in the development of his specific engineering projects and designs.

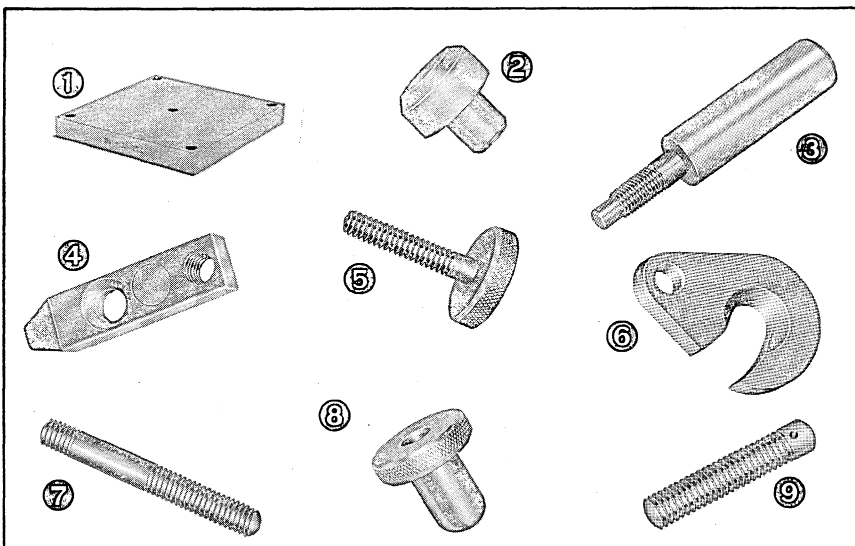


FIGURE 12-1

It is hoped that the information in this chapter will prove an important complement to the engineer's and designer's technical knowledge.

WHAT TOOL COMPONENTS ARE!

Tool components are standard parts for jigs, fixtures, tools and dies. In the past, the tool designer has always designed jigs and fixtures over and over again to suit the particular problem. All the parts of the jig and fixture were special and custom made. Now, he can save valuable detail and design time through the use of the more common part or components. (See Figure 12-1.)

They include very accurate (1) jig plates manufactured under

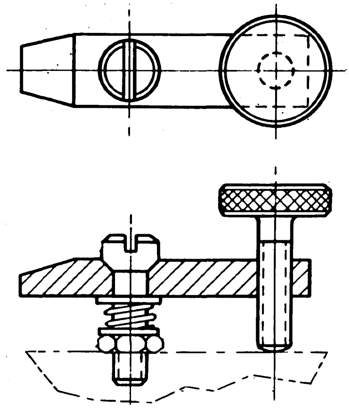


FIGURE 12-2

controlled conditions, (2) jig buttons, (3) jig legs, (4) several types of clamps, (5) numerous interchangeable thumb screws and (6) swivel "C" washer. Also (7) double threaded studs, (8) swivel and thumb nuts, (9) heel pins and many others.

Some are available in tool steels, heat treated and plated, ready

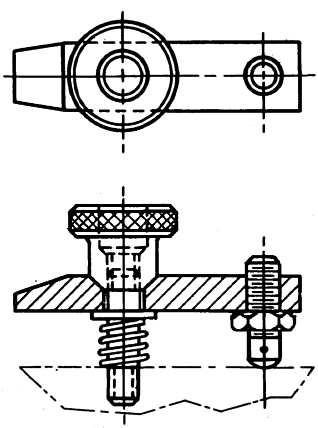


FIGURE 12-3

for immediate use and some are made of stainless steel to eliminate rust and lubrication problems, and to assure long life and reusage.

Many tool component parts come in a wide range of sizes. Complete clamp assemblies are also available as units or made up from individual components as desired.

ADVANTAGES OF USING STANDARD TOOL COMPONENTS

Design time is saved since many jig and fixture components can be specified from the catalog and need not be individually designed. (Figure 12-2.)

Machine shop time is saved in that these components do not have to be custom made but can be purchased from stock at production cost. (Figure 12-3.)

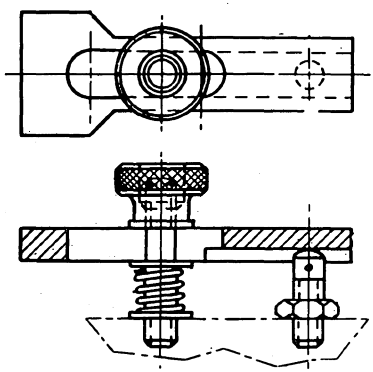
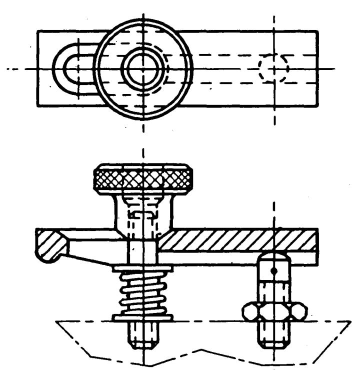


FIGURE 12-4

Delays are avoided since components are available from stock in almost a matter of hours instead of in weeks or months as when custom made. (Figure 12-4.)

Better components are frequently obtained since the skills of men devoted exclusively to this phase of metal-working are embodied in the design of these components. Further, the experiences of many

FIGURE 12-5



companies, as reflected back, are incorporated in the design of these elements. (Figure 12-5.)

HOW COMPONENTS ARE USED

The designer uses precision ground jig or instrument plates as foundation bases, on which to assemble other elements of the jig or fixture. Rectangular, square and round plates are readily available in a range of sizes in aluminum, stainless steel, tool steel and ground stock to meet the tool engineer's requirements. Since the base plate is the foundation of the jig or fixture, it is essential, for accurate machining, that the plate be precision made. Therefore, jig plates have a 32 micro inch or better ground finish on all six sides, and

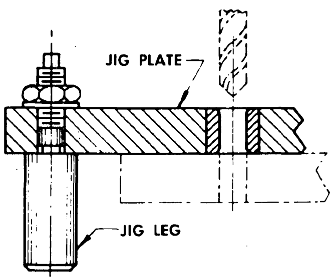


FIGURE 12-6

surfaces are flat and parallel within ±0.0005″. Corners are square to ± 5 minutes. Thus, we have ideal starting or banking conditions for reference use and jig boring. For drilling operations, jig buttons are inserted at the corners of the plate to provide four resting points and assure level positioning of the plate on the drill table. Jig buttons are press fit into reamed holes in the jig plate. Jig buttons also have other uses as will be brought out in typical applications to follow.

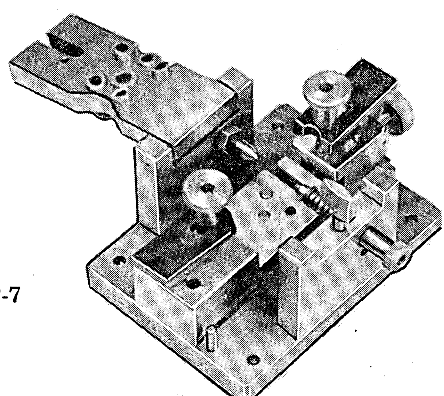


FIGURE 12-7

Jig legs are used to support a second plate above the jig plate (Figure 12-6), and also to permit tumbling of the jig so that it can be used in drilling more than one surface of the work, and to ease loading work. This application will be illustrated in one of the examples to follow.

Clamping assemblies are used to secure work to the jig or fixture. The choice of clamping assembly is determined by the size to be held, whether the surface of the part is smooth (machined), or rough (casting), and to some extent upon the preference of the designer or tool engineer.

The following discussion of typical jigs and fixtures will illustrate the various ways in which standardized components are used to save design and machine shop time.

Figure 12-7 shows a typical jig used in vertical drilling operations. Some of the components are custom made, some are standard parts and some are altered standardized parts. This particular jig has a

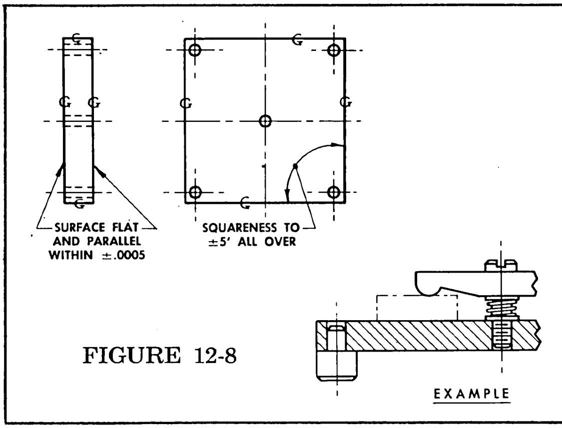


FIGURE 12-8

hinged leaf with guide bushing insets, which is folded over and locked in place once the work has been positioned and clamped.

The base plate of this jig is a jig plate, Figure 12-8 with jig buttons, Figures 12-9 and 12-10, at the corners. In placing work in the jig, the spring-loaded pusher in the block opposite the leaf-supporting block is pulled back, the work is placed in the center space, and the pusher released. The pusher secures the work against a horizontal pin in the leaf-supporting block. This item is fashioned from a stand-

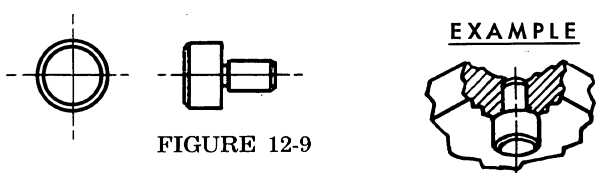


FIGURE 12-9

ard socket-head cap-screw, the threaded end against which the work rests being filed to a point. The pin is locked in place by a nut which is tightened against the block once the pin has been properly adjusted. The pin normally stays in one position in handling a given lot of castings. However, deviations among lots may require intermittent pin adjustment.

The pusher is made from standard components – a stud, knurled thumb nut, spring, and retaining pin. The stud passes through a hole in the supporting block and the thumb nut is screwed on the outer end. The spring is fitted over the inside end, and is secured to the stud with a retaining pin. The stud is drilled to receive the retaining pin.

The horizontal thumb screw in the block at the right end of the jig plate is a standard item. It passes through a tapped hole in the block and pushes against the work to secure it against the opposite block. It has been fitted at the threaded end with a swivel shoe that

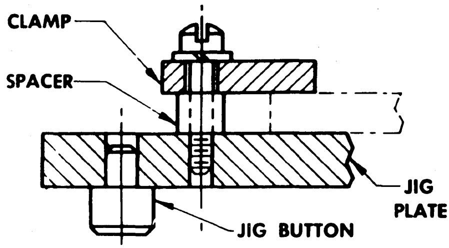


FIGURE 12-10

engages the work without turning while the thumb screw advances. Precision ground jig and fixture plates are available in #440C stainless steel which can be hardened and heat treated, if desired, for wear reasons.

The clamps at the ends of the jig are flat sliding clamps, Figure 12-11. The groove in the underside fits over a heel pin screwed into

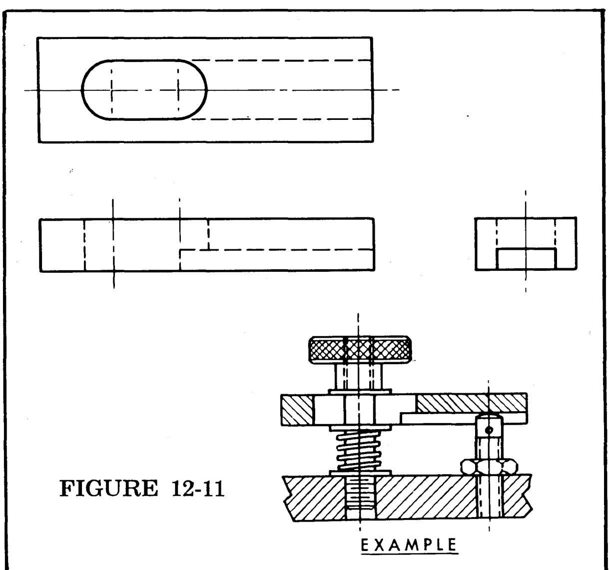


FIGURE 12-11

the jig plate. The center slot fits over a stud which screws (stud fit) into the supporting block. A spring and washer are interposed between block and clamp to provide spring loading. A thumb nut (loose fit) screws onto the end of the stud which extends above the clamp, and tightens and loosens the clamp. The clamp slides back and forth to engage and disengage the work, being guided by the heel pin and stud. The heel pin may be adjusted for height and is locked in place by a nut which is tightened against the block when the heel pin has been adjusted to the desired height.

The nose of one clamp has been cut back to a slight taper and

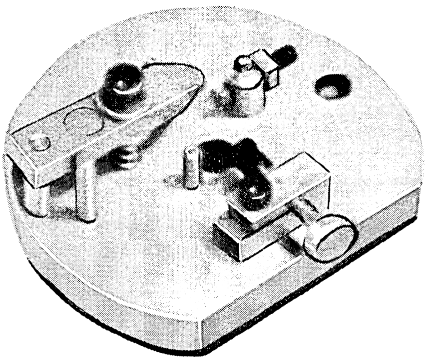
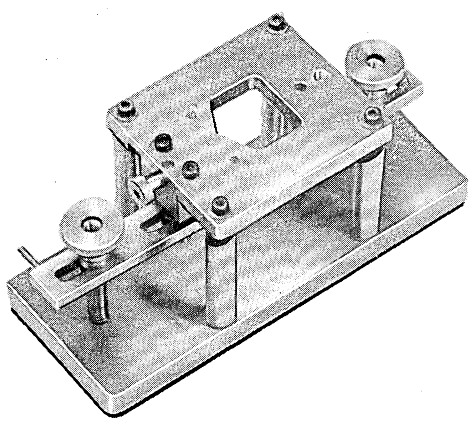


FIGURE 12-12

the nose of the other has been slightly concaved. This has been done to prevent the clamps from obstructing the path of the drill. One of the components also has been undercut to permit it to overlap a lip on the cast part.

Figure 12-12 shows a fixture used with a lathe. A segment has been cut from the plate to facilitate turning the adjacent thumb

FIGURE 12-13



screw. The round member at the back of the plate fits into a chuck adapter used with the particular lathe for which this fixture has been designed. The hole near the uppermost point of the plate is for a pin which comes through the back to aid in securing the fixture to the chuck.

Work to be turned is positioned against a pin on the far side of the clamp, and against the adjustable stop just below the hole for the chuck-pin. The adjustable stop comprises a jig button which has been drilled and tapped to receive a socket-head set screw. A nut on the set screw located between the jig button and the set screw head, locks the set screw in position. This screw generally requires adjustment only when changing from one lot of castings to another.

Work is tightened against the pin and adjustable stop by the

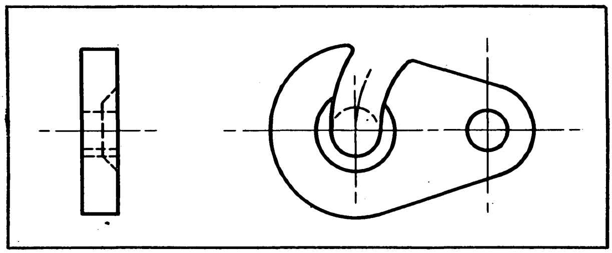


FIGURE 12-14

thumb screw at the flat side of the plate. The end of the screw which engages the work is fitted with a swivel shoe that does not turn as the screw is tightened. A slot has been cut from the edge of the block to the thumb screw hole and a vertical socket-head set screw provided to tighten the slot so the thumb screw may be locked in position during the machining operation.

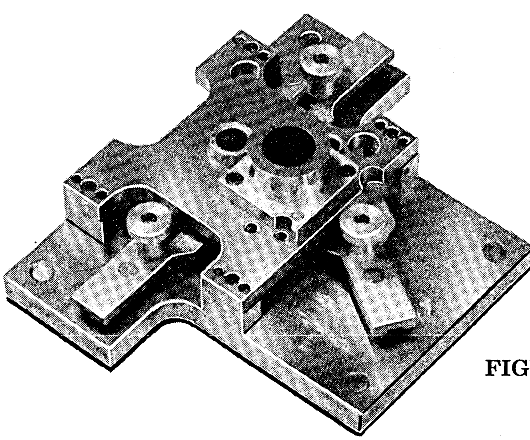


FIGURE 12-15

The clamp in this case is a swinging taper nose clamp. It swings about a set screw which passes through a hole toward the forward end. A spring and washer are interposed between the clamp and the plate and a washer is inserted between the clamp and the head of the screw. A rear end-supporting jig button fits into a hole at the rear of the clamp. Thus, the clamp can pivot about the set screw and the nose can be moved up and down by regulating the set screw. The pin in the plate near the rear of the clamp is a stop for the clamp. The shorter pin between the clamp and the thumb screw is a "fool-proof" pin to assure proper positioning of the part in the fixture.

Figure 12-13 shows a fixture for a vertical milling machine. In this case the work is clamped to the underside of a top plate, two

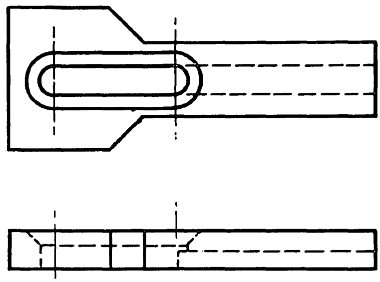


FIGURE 12-16

special rear tightening clamps being used for the purpose. The upper, or banking plate, is supported on four jig legs. The cutting tool extends into the opening in the banking plate and is guided by a pantograph type control.

The clamps are secured to the banking plate by socket head screws, one of which fastens to the plate itself, the other to a block

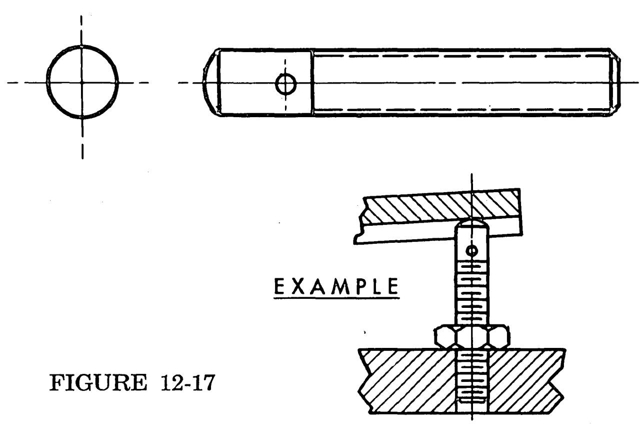


FIGURE 12-17

secured to the plate. Nuts tighten against the underside of the banking plate or block to hold the socket head screws in fixed adjustment. A spring and washer are inserted between the clamp and lock nut in each case. A swivel washer, Figure 12-14, is inserted between the head of the socket head screw and the clamp. This arrangement enables the clamp to swivel up and down to engage and disengage work, the center slot in the clamp being beveled or countersunk to receive it. The clamps are adjusted by means of thumb nuts at the rear of the clamps. These thumb nuts fit onto studs screwed onto the base plate. Between the thumb nuts and clamp are swivel

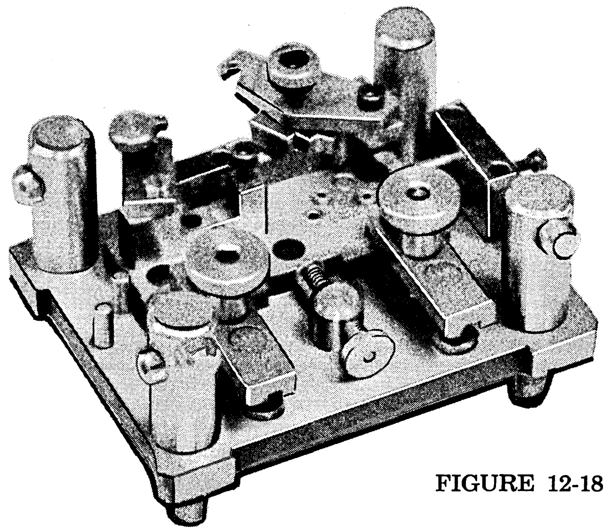


FIGURE 12-18

washers. Flat washers and springs are inserted between the clamps and base plate.

A spring loaded pusher, as described in the jig discussion, is inserted in the block which is attached to the banking plate, and positions the work against several pins in the plate's undersurface. The clamps used here are special in that they have slots at each end so they can slide back and forward for engaging and disengaging

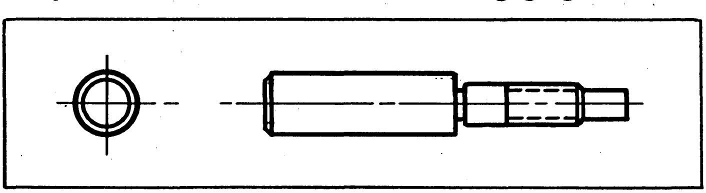


FIGURE 12-19

the work under the conditions illustrated. They have pins inserted in the ends as finger aids in sliding the clamps back and forth.

Figure 12-15 shows another drill jig. In this case, wide face clamps (see Figure 12-16) are used since the casting to be drilled has a broad surface area. Each clamp has two points of contact on its clamping surface so that it firmly holds the work regardless of irregularities in the surface. These clamps are of the sliding type and each has a groove in the underside. This groove fits on a heel pin, Figure 12-17, which screws into the base plate.

The forward end of the clamp has a through-slot which fits into a stud which screws into the base plate. A spring and washer are inserted between the clamp and plate to provide spring loading. A thumb nut screws on the upper end of the stud and controls tightening of the clamp.

This particular jig is equipped with slip bushings. One set of bushings fits into holes in the top plate and a second set fits into the first set. The slip bushings enable drilling of a small size hole which can then be enlarged to a certain depth by removing them and using the fixed bushings as guides.

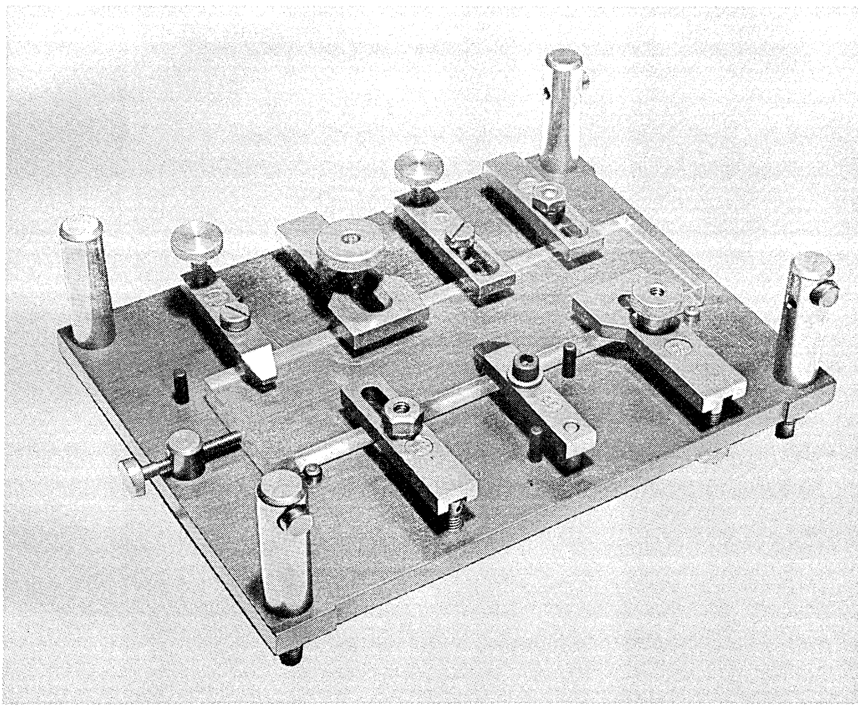


FIGURE 12-20

Tool components also adapt jigs to hold work for drilling holes in more than one surface. In Figure 12-18, a jig plate has been equipped with jig legs at each corner, and these legs in turn have been fitted with jig buttons near the upper extremities. With this arrangement, the jig may be accurately positioned in a horizontal position, tipped on end or tipped on its side.

The portions of the jig legs, Figure 12-19, which extend beneath the plate provide precise leveling when the jig is horizontal, and the jig buttons, in conjunction with machined plate projections, provide accurate four-point support of the work in the other two positions. A number of other components are also illustrated in Figure 12-20.

These include sliding swivel clamps for clamping unfinished surfaces, knurled swivel nuts which tighten and loosen the swivel clamps, studs on which the knurled swivel nuts turn, springs and washers which support the swivel clamps, and heel pins which guide and also support the clamps. Also used here is a front tightened taper nose clamp with knurled thumb screw and a swinging "C" washer and shoulder screw.

Figure 12-20 illustrates seven clamp assemblies used under various conditions. In the lower row, right, is a sliding wide-face clamp. This clamp slides back and forth to engage and disengage the work. It is tightened and loosened by means of a thumb nut which screws onto a stud that screws into the base plate. A spring and washer are inserted between the clamp and plate for spring loading, and a groove

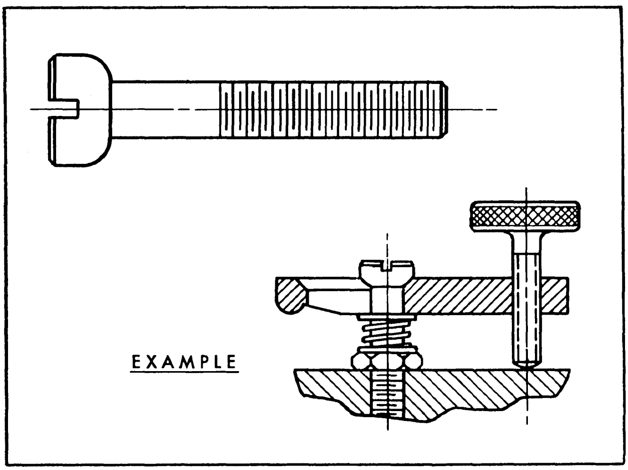


FIGURE 12-21

in the underside of the clamp fits over a heel pin screwed into the base plate. This clamp is used where wide surfaces are to be clamped.

Lower row, center, shows a swinging taper-nose clamp. It pivots about a socket head cap screw which passes through a hole toward the clamping end. The clamp is spring loaded and a washer is inserted between the head of the screw and the clamp. A jig button is inserted in the underside of the clamp at the rear and stop pins are inserted in the jig plate as guides in swinging the clamp. This clamp is used where the clamping area is small.

Lower row, left, shows a sliding clamp which is tightened and loosened by means of a hex nut. A washer is inserted between the nut and the clamp. The clamp is spring loaded and has a groove in the underside that fits into the heel pin.

Upper row, right, shows a sliding swivel clamp. It is similar to the sliding clamp except that the through-slot is beveled to engage a rounded nut, FIGURE 12-22, thus enabling the clamp to swivel up and down in addition to sliding back and forth in engaging and disengaging work.

Upper row, second from right, is a sliding swivel clamp assembly for rear adjustment. A spring-loaded swivel stud, FIGURE 12-21, passes through the center slot and screws into the base plate.

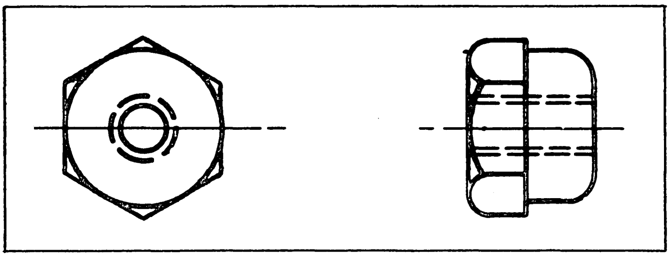


FIGURE 12-22

A knurled thumb screw which passes through a tapped hole at the rear of the clamp tightens and loosens it. The thumb screw terminates in a slot in the plate to provide control of the sliding motion when releasing or securing work.

Upper row, second from left, is a wide-face swivel clamp with knurled swivel nut. (Figure 12-23.)

Upper row, left, is a swinging taper nose clamp which is tightened

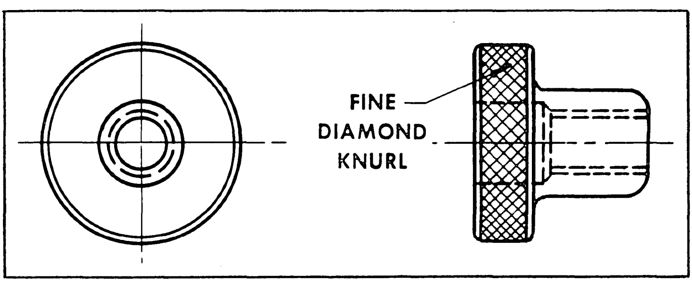


FIGURE 12-23

and loosened from the rear by means of a thumb screw. The forward fastener is a swivel stud, the forward hole being countersunk to receive it.

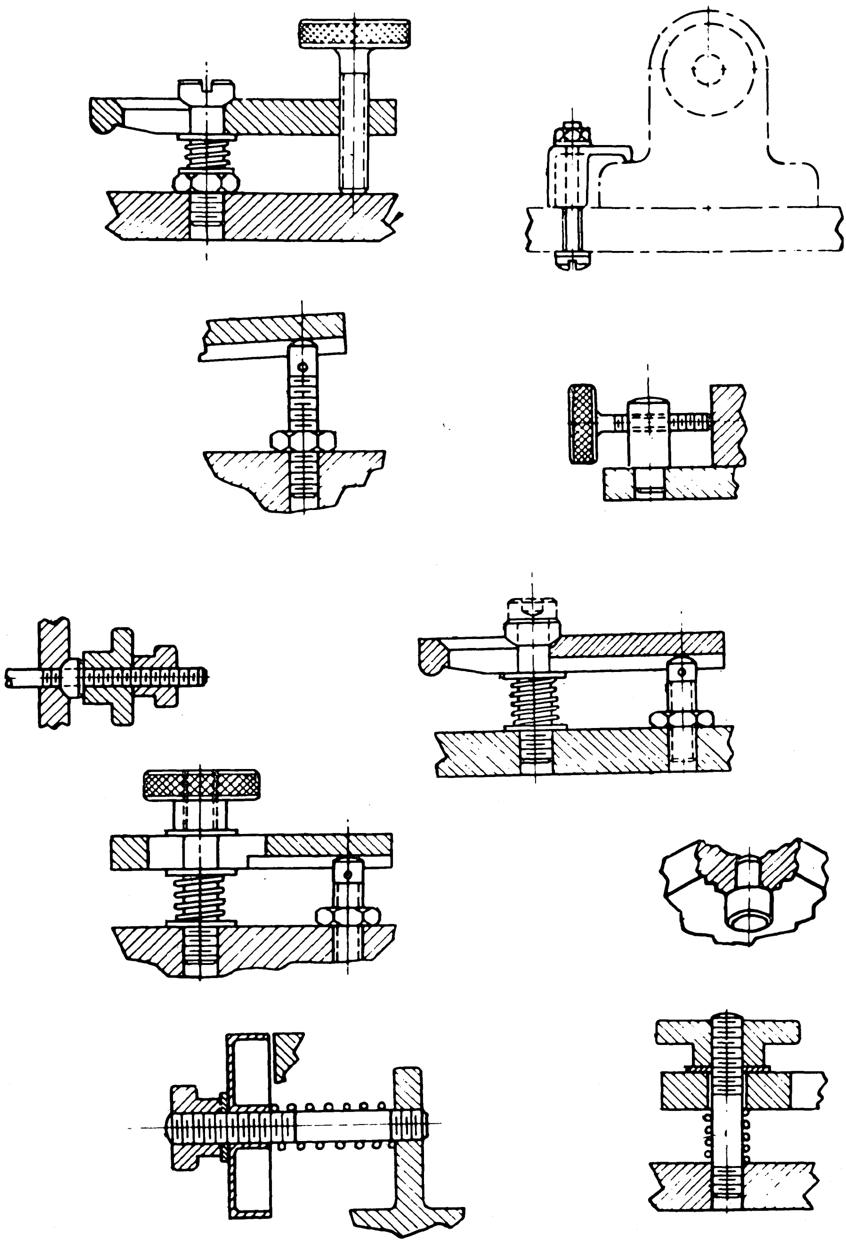
Also shown in this photo is a jig plate, jig legs, and jig buttons for tumbling, and a jig button with horizontal thumb screw for positioning work against a banking surface.

USING TOOL COMPONENTS IN YOUR PLANT

As indicated by the foregoing examples, standard tool components are easy to use and save substantial amounts of both design time and machine shop time. Once the components have been selected, no further design work is required, and, of course, no machine shop time is required. The availability of the components is as rapid as the carriers which link the designer with the plant.

Components are often ordered for a particular project and retained upon its completion for future use. Some of the larger companies maintain an appropriate inventory of the various types and sizes of components so that they are available from the company's own tool room. Some of these companies also include information on component application in their manuals on design procedure. See the following few suggested uses for some standard tool parts.

A FEW SUGGESTED USES FOR SOME OF STANDARD TOOL PARTS



Chapter Thirteen

MISCELLANEOUS COMPONENTS

PRECISION SPUR RACKS

Spur racks have been established to offer the conversion of rotary motion to linear motion or linear motion to rotary motion and/or for pick-up signals from rotating components such as potentiometers, resolvers, etc. (See Figure 13-1.)

Because of limited requirements for such gears, there is limited equipment and little or none is available to produce the precise, high accuracy racks that could run with the conventional Precision 1, 2 or 3 spur gears. The one basic problem is holding the straightness

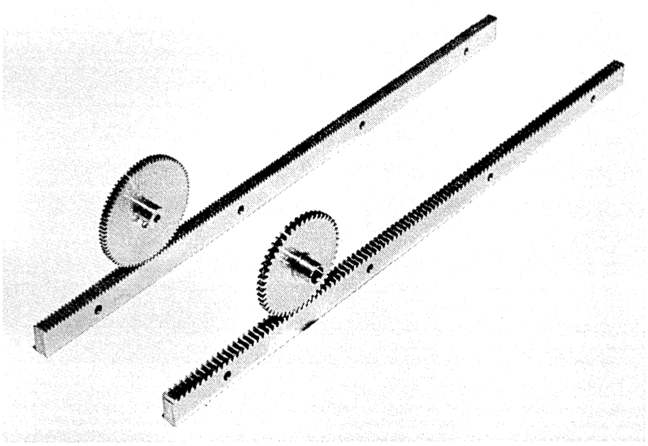


FIGURE 13-1

of the rack, after the teeth are notched all along the surface. The material by normal action must bow, distort and twist, and all accuracy is lost. Thus tremendous caution and care must be maintained in starting with dense, stress relieved material. Then the blanks must be rough cut and again stress relieved by heat treatment, aging or deep freezing, to remove additional stresses before the final finest teeth are cut. This is why certain very precise, Precision 3 racks are made of only short lengths, approximately one foot, to maintain the critical tolerances. These are classified as Certified racks. Such racks are designed for butting to enable the engineer or designer to have lengths of more than one foot to infinite lengths. (See Figure 13-2.)

Classifications and tolerances for such "Certified" racks are referenced in Figure 13-3.

The advantage the butt rack gives is that one can compensate for tooth to tooth accumulated errors in the butting operation.

Many of these precise racks are now used in numerically controlled machine tool table motion in conjunction with resolver and potentiometer pick-ups. Because table motion of thousandths of an inch must be measured, a new gear pitch, namely 1/10 circular

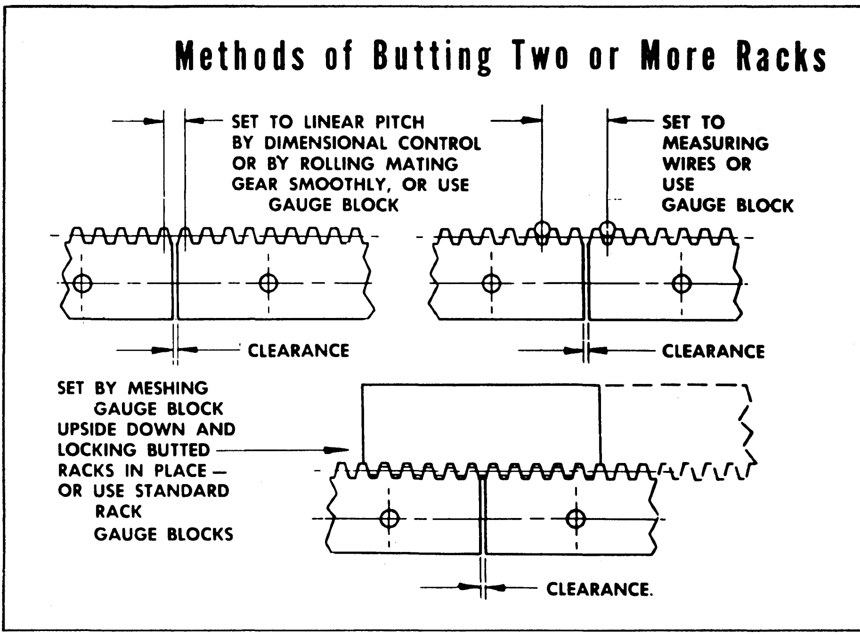


FIGURE 13-2

pitch, which gives .100 rack motion for one tooth of the gear, has been established. This can be amplified or reduced through gearing to receive or send the proper signal in tenths of thousands.

ROTATING COMPONENT MOUNTING CLEATS

Many methods of mounting rotating components such as motors, resolvers, potentiometers and syncros to instrument plates have been developed.

The most practical and commonly used method is the swing washer or clip method, as indicated in the following illustrations. (Figure 13-4.)

Available Pitches and Tooth Data

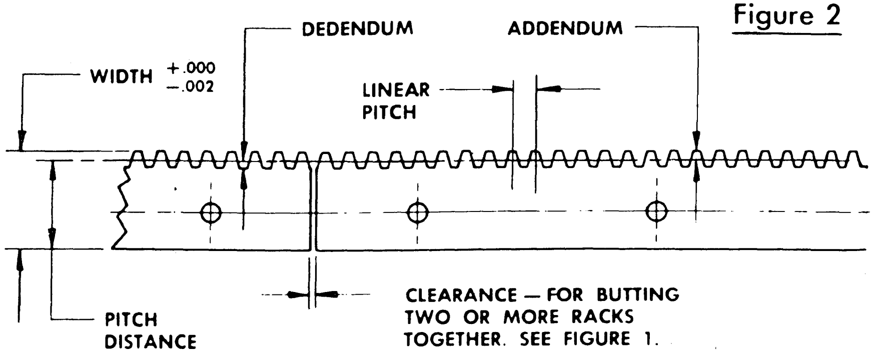
Pitch	Linear Pitch	Addendum	Dedendum	Whole Depth
24	.1309	.0417	.0520	.0937
1 10	.1000	.0318	.0402	.0720
32	.0982	.0313	.0395	.0708
48	.0654	.0208	.0270	.0478
64	.0491	.0156	.0208	.0364
72	.0436	.0139	.0187	.0326
80	.0393	.0125	.0170	.0295
96	.0327	.0104	.0145	.0249

Classifications and Tolerances

Class Rack	Adjacent Tooth-to-Tooth Error	Non-Adjacent Tooth-to-Tooth Error	Max. Overall Accumulated Error in 11″	Parallelism To Base
PREC. 1	.0005	.001	.002	.001
PREC. 2	.0004	.0007	.001	.0007
PREC. 3	.0002	.0005	.0006	.0005

Terminology and Tolerances

Figure 2



Material: #416 Stainless Steel
Stress Relieved (Not Heat Treated)

FIGURE 13-3

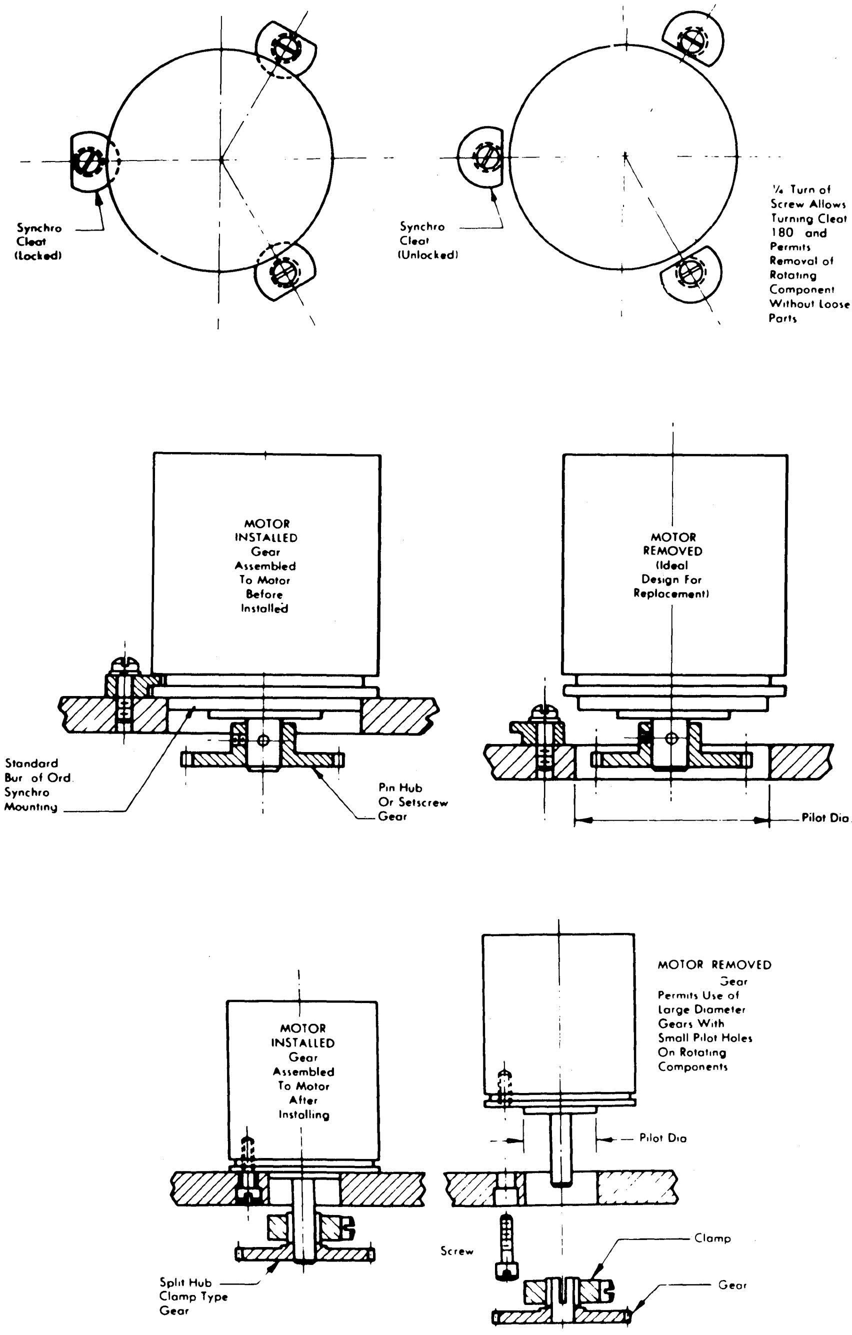

Synchro
Cleat
(Locked)
Synchro
Cleat
(Unlocked)
¼ Turn of
Screw Allows
Turning Cleat
180 and
Permits
Removal of
Rotating
Component
Without Loose
Parts
MOTOR
INSTALLED
Gear
Assembled
To Motor
Before
Installed
MOTOR
REMOVED
(Ideal
Design For
Replacement)
Standard
Bur. of Ord.
Synchro
Mounting
Pin Hub
Or Setscrew
Gear
Pilot Dia.
MOTOR
INSTALLED
Gear
Assembled
To Motor
After
Installing
MOTOR REMOVED
Gear
Permits Use of
Large Diameter
Gears With
Small Pilot Holes
On Rotating
Components
Pilot Dia
Split Hub
Clamp Type
Gear
Screw
Clamp
Gear

FIGURE 13-4

CAPTIVE SCREWS

Captive screws have been developed to eliminate possible loss of the screws for the cover, often removed from the mechanism. (See Figure 13-5.)

The panel or cover is tapped to accept the screw until it reaches the undercut or reduced area. Thus the screws always stay with the panel or cover and are used or assembled as indicated in Figure 13-6.

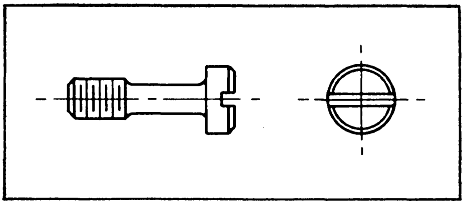


FIGURE 13-5

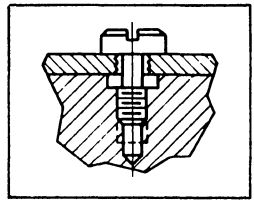


FIGURE 13-6

PRECISION GROUND INSTRUMENT PLATES

A wide variety of standardized and stocked precision ground instrument plates of high accuracy has been established. They are available in many different thicknesses, widths and lengths in both stainless steel and aluminum to meet almost any possible requirement or application. (See Figure 13-7.)

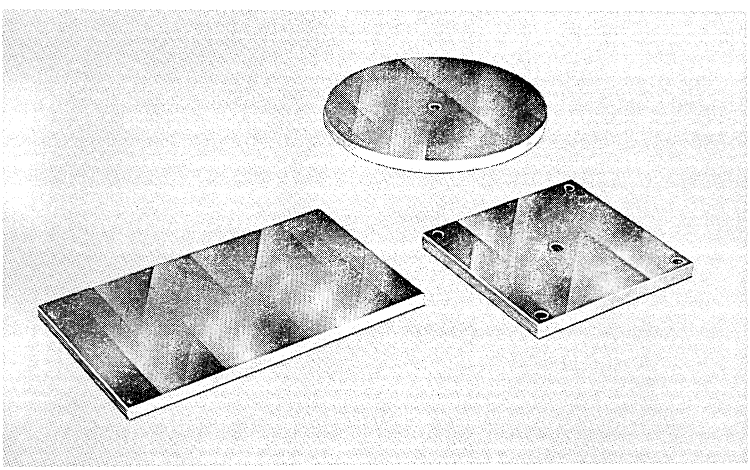


FIGURE 13-7

Although these are basic plates, they are square, flat and parallel within $\pm.0005$ and are ideal for the start of the critical jig boring operations, cage assembly, gear centers, tools and fixtures. (See Figures 13-8 and 13-9.)

Typical cross section of a basic assembly is indicated in Figures 13-10 and 13-11.

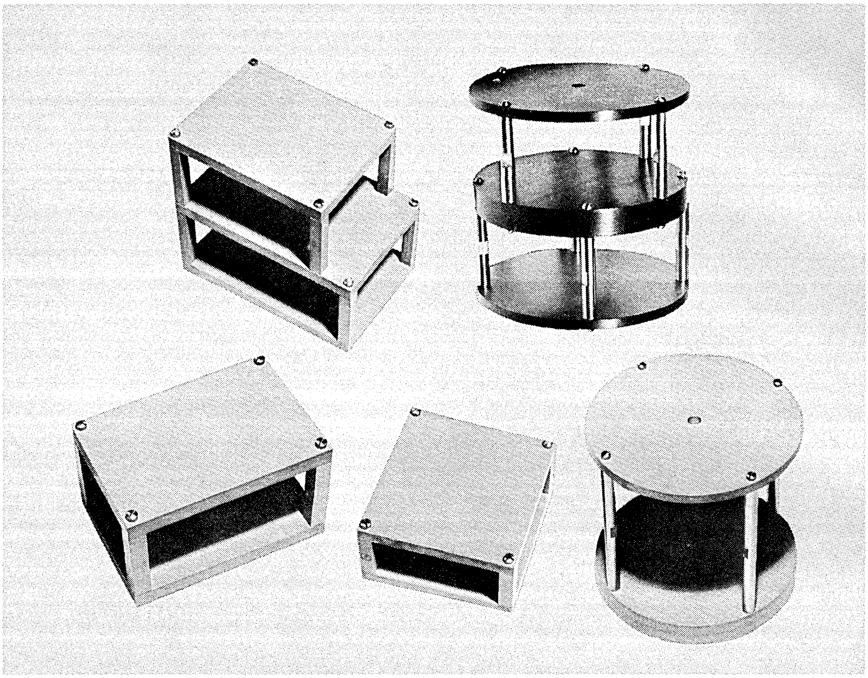


FIGURE 13-8

STAINLESS SPRINGS

A range of stock stainless steel springs, both open and closed wound, have been designed and made available in various diameters and spring wire sizes which can be slightly modified to suit almost

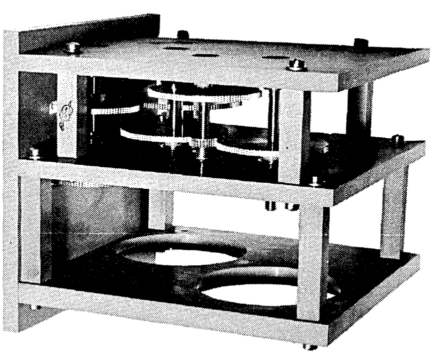


FIGURE 13-9

any individual engineer's or designer's basic problem. (See Figure 13-12.)

The ends of the spring can be squared and ground for proper seating in the holes or loops can be formed to fit on the spring posts. (See Figure 13-13.)

INTERNAL SPUR GEARS

A wide variety of internal spur gears ranging in outside diameter from two inches to four inches in diameter and in stainless or alu-

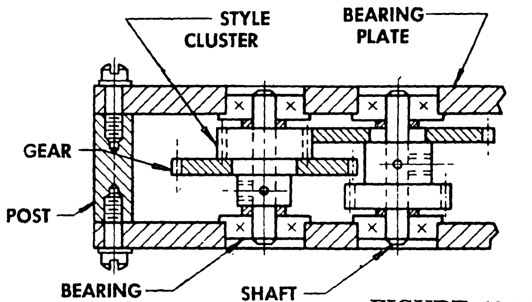
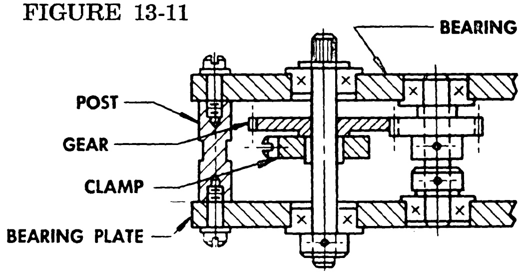


FIGURE 13-10

FIGURE 13-11



minum, are available. Figure 13-14 illustrates internal gears being shaped.

Unfortunately, again it is practically impossible to produce better than Precision 1 in internal gears because of the nature of the equip-

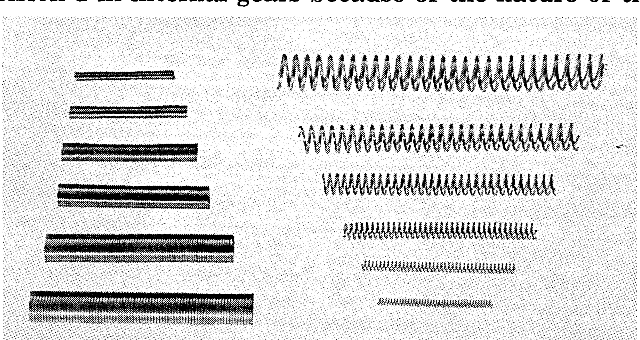


FIGURE 13-12

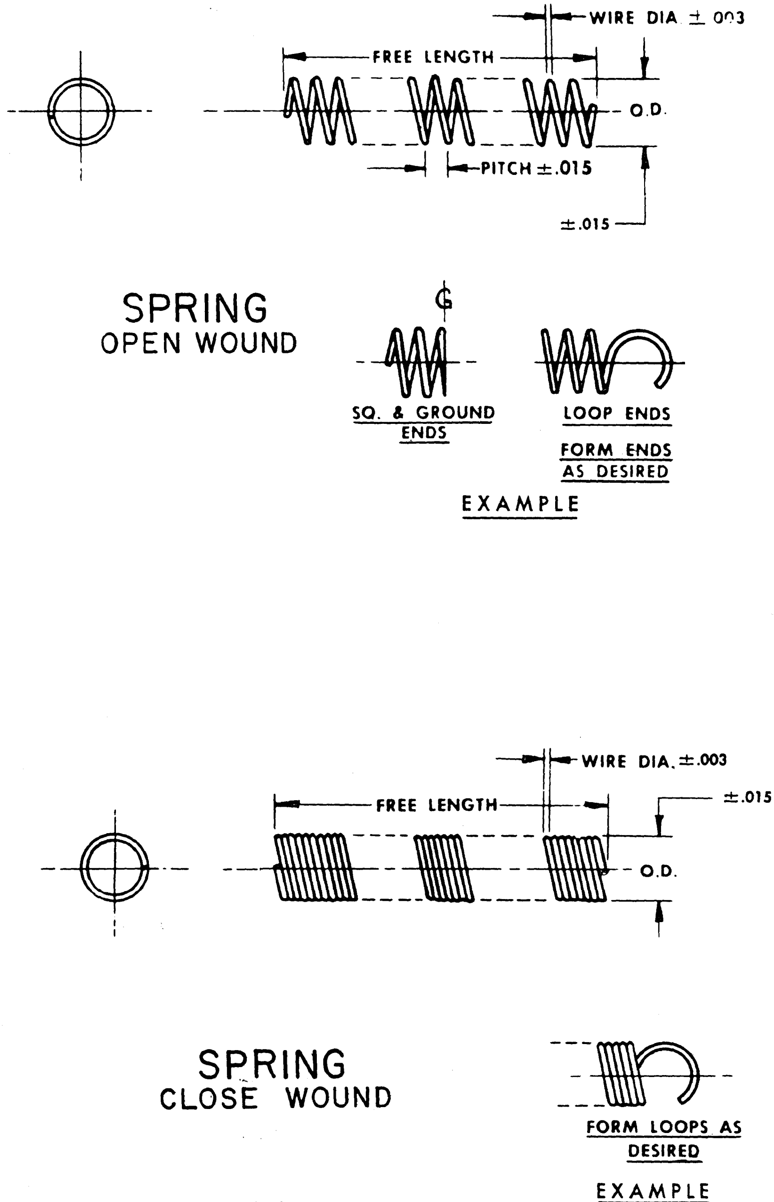

WIRE DIA ± .003
FREE LENGTH
O.D.
PITCH ±.015
±.015
SPRING
OPEN WOUND
SQ. & GROUND ENDS
LOOP ENDS
FORM ENDS AS DESIRED
EXAMPLE
WIRE DIA. ±.003
FREE LENGTH
±.015
O.D.
SPRING
CLOSE WOUND
FORM LOOPS AS DESIRED
EXAMPLE

FIGURE 13-13

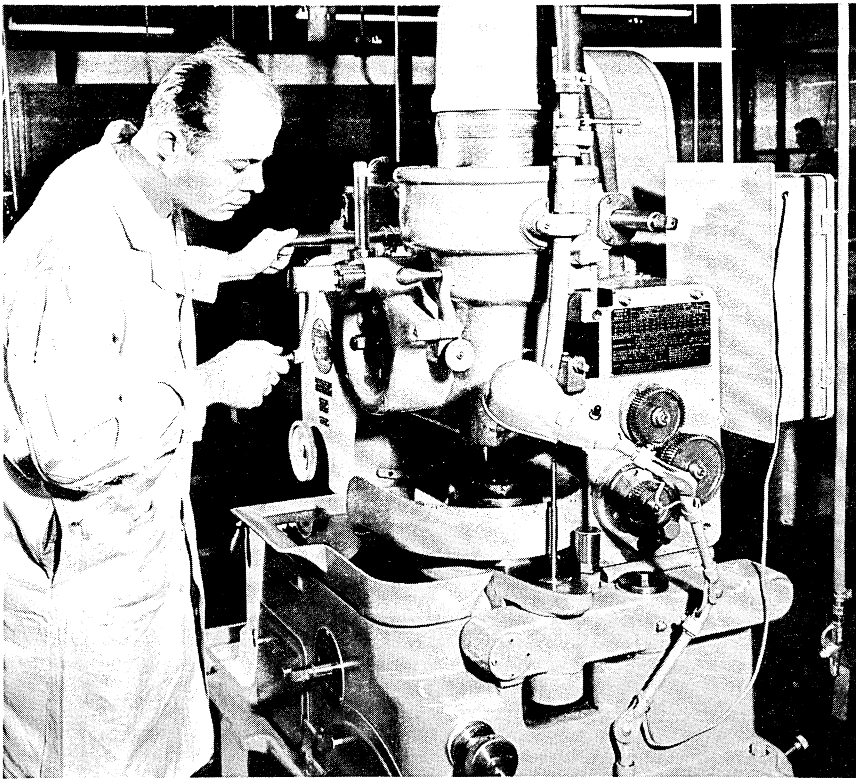


FIGURE 13-14 *Internal Spur Gears Being Shaped on Fellows Equipment*

ment available in today's market and blank distortion due to the basic shape. (Figure 13-15.)

One serious error usually made by engineers and designers is in specifying the size of the spur gear which can roll properly with a

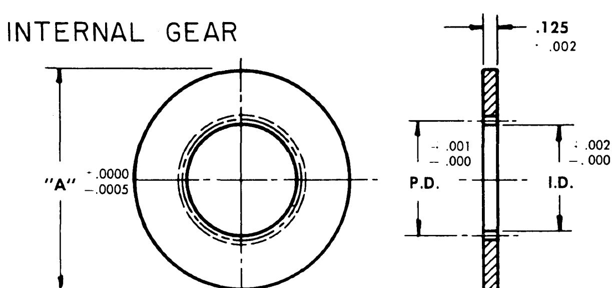


FIGURE 13-15

particular size of gear. A quick rule of thumb selection has been the number of teeth in the gear should never be more than 60% of the number of teeth in the internal gear.

Chapter Fourteen

ENGINEERING REFERENCE DATA

The following pages contain valuable engineering data compiled to assist you in expediting your specific project.

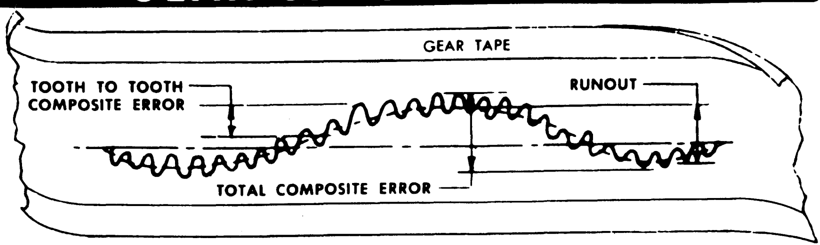
They contain such information as gear specifications, data, and formulae; military specifications; screw head, drill and tap data, standard symbols, torque measurement, and other data.

A.G.M.A. COMPARISON TABLE

Below is a comparison chart between the old and new A.G.M.A. Quality Numbers

Class of Gear	OLD A.G.M.A.	New A.G.M.A.
Commercial 1	Commercial 1	Quality #5 or 6
Commercial 2	Commercial 2	Quality #6 or 7
Commercial 3	Commercial 3	Quality #8
Commercial 4	Commercial 4	Quality #9
Precision 1	Precision 1	Quality #10 or 11
Precision 2	Precision 2	Quality #12
Precision 3	Precision 3	Quality # 13 or 14
Ultra 1	NONE	Quality #15 or 16

GEAR SPECIFICATIONS



Flicker of indicator of a variable center distance fixture shows the effect of circular pitch error. tooth thickness variation and profile error.

Gear Tapes are included with all Prec. 2, 3 or Ultra Prec. 1 gears

Gear Tapes are available with Prec. 1 gears upon request,

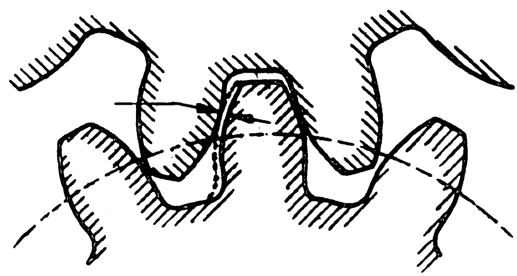
Class of Gear	Total Composite Error	Tooth to Tooth Composite Error
Comm. 1	.006	.0020
Comm. 2	.004	.0015
Comm. 3	.002	.0010
Comm. 4	.0015	.0007
Prec. 1	.00100	.0004
Prec. 2	.00050	.0003
Prec. 3	.00025	.0002
Ultra Prec. 1	.00020	.00015

Class Gear Recommended for Various Pitch Line Velocities	
Speeds R.P.M.	**Class Gear**
Up to 80	Comm. 1 & 2
Up to 400	Comm. 3 & 4
Up to 2000	Prec. 1
Up to 5000	Prec. 2
Up to 10,000	Prec. 3
Over 10,000	Ultra Prec. 1

SPUR GEAR DATA

Standard Stock Pitches	Tooth Size	Addendum	Dedendum	Whole Depth	Circular Pitch	Checking Pressure
24		.0417	.0520	.0937	.1309	28 oz.
32		.0313	.0395	.0708	.0982	24 oz.
48		.0208	.0270	.0478	.0654	20 oz.
64		.0156	.0207	.0363	.0491	12 oz.
72		.0139	.0187	.0326	.0436	12 oz.
80		.0125	.0170	.0295	.0393	8 oz.
96		.0104	.0145	.0249	.0327	8 oz.
120		.0083	.0120	.0203	.0262	4 oz.
200		.0050	.0080	.0130	.0157	2 oz.

BACKLASH IN GEARS



BACKLASH BETWEEN MATING GEARS

BACKLASH is the shortest distance between non-driving tooth surface of adjacent teeth in mating gears or the amount of clearance between the teeth or mating gears and is usually measured at the common pitch circle. It varies from .0000 to .0015 depending on the class of gear, the pitch, number of teeth and tolerance variations. So-called zero backlash applications are expensive and should be avoided if possible.

In general, backlash in gears is play between mating teeth. It occurs only when gears are in mesh. In order to measure and calculate backlash, it is defined as the amount by which a tooth space exceeds the thickness of an engaging tooth. Unless otherwise specified, numerical values of backlash are understood to be given on the pitch circles.

Backlash is provided for a variety of reasons and cannot be designated without consideration of machining conditions. The general purpose of backlash is to prevent gears from jamming together and making contact on both sides of their teeth simultaneously.

Any error in machining which tends to increase the possibility of jamming makes it necessary to increase the amount of backlash by at least as much as the errors. Consequently, the smaller the amount of backlash, the more accurate the machining of the gears must be.

Runout of both gears, errors in tooth profile, pitch, tooth thickness, pitch diameter tolerance and center distance, all are factors to be considered in the specification of backlash.

Backlash can be measured on a variable center distance fixture similar to that used in checking the composite error. In this case, the backlash is approximately equal to the change in center distance multiplied by twice the tangent of the transverse pressure angle.

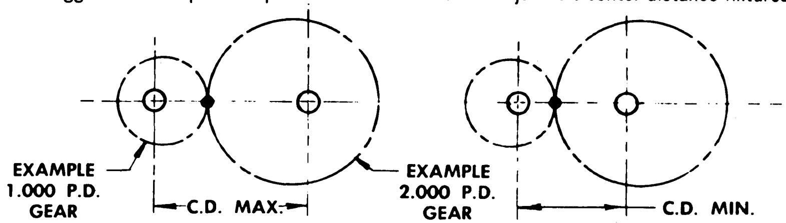
EXAMPLE: .001 Center Distance Change
20 P.A. Gears
Tangent of 20 P.A. = .36397

BACKLASH = Center Distance Change x (2 x Tan. 20 P.A.)
= .001 x (2 x .36397)
= .0007 Backlash

CONTINUED

BACKLASH IN GEARS

Exaggerated examples of Spur Gears as measured on adjustable center distance fixtures.



$$\text{C.D.} = \frac{\text{P. D. (Pinion)} + \text{P.D. (Gear)} + (2 \times \text{T.C.E.})}{2}$$

P. D. = Pitch Diameter

O.D. = Outside Diameter

T.C.E. ≐ Total Composite Error

$$\text{P.D.} = \frac{\text{No. of Teeth}}{\text{Diametral Pitch}}$$

$$\text{O.D.} = \frac{\text{No. of Teeth} + 2}{\text{Diametral Pitch}}$$

$$\text{No. of Teeth} = \text{P.D.} \times \text{Dia. Pitch}$$

$$\text{Dia. Pitch} = \frac{\text{No. of Teeth}}{\text{P.D.}}$$

$$\text{Cir. Pitch} = \frac{3.1416}{\text{Dia. Pitch}}$$

PIC Class	Total Composite Error	P.D. Tolerance
Prec. 1	.001	+.000 −.001
Prec. 2	.0005	+.0000 −.0007
Prec. 3	.00025	+.0000 −.0005
Ultra Prec. 1	.00020	+.0000 −.0004

Class	C.D. (Max.)	C.D. (Min.)	C.D. Differences	Max. Backlash
Prec. 1	1.501	1.499	.002	.00146
Prec. 2	1.5005	1.4993	.0012	.00087
Prec. 3	1.50025	1.4995	.00075	.00055
Ultra Prec. 1	1.5002	1.4996	.0006	.00043

PRESSURE ANGLE of a gear tooth is the angle between the tooth profile and the radial line at its pitchpoint or the angle between the line of action and the line tangent to the pitch circle. Standard pressure angles have been established in connection with standard gear tooth proportions to transmit smooth action at constant velocity with minimum intereference. The basic 20 pressure angle (P.A.) is most commonly used in the fine pitch field, and fine pitch gears are STOCKED in 20 P.A. only for stronger teeth and smoother rolling.

DIAMETRAL PITCH is the number of teeth to each inch of the pitch diameter.

CIRCULAR PITCH is the distance from the center of one tooth to the center of the next tooth, measured along the pitch circle circumference.

RULES & FORMULAE for SPUR GEARS

TO FIND	RULE	FORMULA
Diametral Pitch	Divide 3.1416 by Circular Pitch	$\text{Dia. Pitch} = \frac{3.1416}{\text{Cir. Pitch}}$
Circular Pitch	Divide 3.1416 by Diametral Pitch	$\text{Cir. Pitch} = \frac{3.1416}{\text{Dia. Pitch}}$
3.1416	Multiply Dia. Pitch by Circular Pitch	$\text{Dia. Pitch} \times \text{Cir. Pitch} = 3.1416$
Pitch Dia.	Divide No. of Teeth by Diametral Pitch	$\text{P.D.} = \frac{\text{No. Teeth}}{\text{Dia. Pitch}}$
Pitch Dia.	Multiply No. of Teeth by Cir. Pitch and Divide by 3.1416	$\text{P.D.} = \frac{\text{No. Teeth} \times \text{Cir. Pitch}}{3.1416}$
Outside Dia.	Divide No. of Teeth Plus 2 by Dia. Pitch	$\text{O.D.} = \frac{\text{No. Teeth} + 2}{\text{Dia. Pitch}}$
Outside Dia.	Multiply the No. of Teeth Plus 2 by Cir. Pitch & Divide by 3.1416	$\text{O.D.} = \frac{(\text{No. Teeth} + 2) \times \text{Cir. Pitch}}{3.1416}$
No. of Teeth	Multiply O.D. by Dia. Pitch & Subtract 2 from this Product	$\text{No. Teeth} = (\text{O.D.} \times \text{Dia. Pitch}) - 2$
No. of Teeth	Multiply O.D. by 3.1416 & Divide by Cir. Pitch & Subtract 2 from this Product	$\text{No. Teeth} = \left(\frac{\text{O.D.} \times 3.1416}{\text{Cir. Pitch}}\right) - 2$
Addendum	Divide 1 by Diametral Pitch	$\text{Add.} = \frac{1}{\text{Dia. Pitch}}$
Addendum	Multiply .3183 by Circular Pitch	$\text{Add.} = .3183 \times \text{Cir. Pitch}$
Working Depth	Multiply Add. by 2	$\text{Working Depth} = 2 \times \text{Add.}$
Full Depth of Tooth	Divide 2.157 by Dia. Pitch	$\text{Full Depth} = \frac{2.157}{\text{Dia. Pitch}}$
Full Depth of Tooth	Multiply .6866 by Circular Pitch	$\text{Full Depth} = .6866 \times \text{Cir. Pitch}$
Dedendum	Subtract Add. from Full Depth	$\text{Ded.} = \text{Full Depth} - \text{Add.}$
Tooth Thickness	One Half the Circular Pitch	$\text{Tooth Thickness} = \frac{\text{Cir. Pitch}}{2}$
Ratio	Divide Teeth in Driven by Teeth in Driver	$\text{Ratio} = \frac{\text{No. Teeth in Driven}}{\text{No. Teeth in Driver}}$

MILITARY SPECIFICATIONS

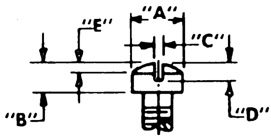
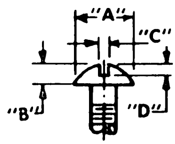
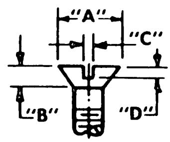
Military Specifications

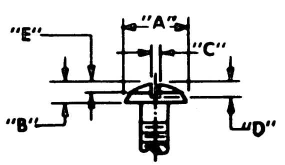
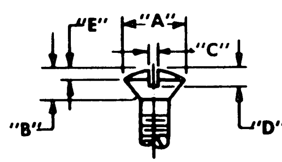
Materials
Finishes
Treatments
Lubrication

Hardness of #303 Stainless Steel
B75-83 Rockwell

MATERIALS		FINISHES		USED ON
Designation	**Mil. or Fed. Specification**	**Designation**	**Mil. or Fed. Specification**	**(Recommended)**
#303 Stainless St. (Bar)	MIL-S-7720 Comp. 303S or SE	**Clear Passivate**	MIL-F-14072 Finish E300	**Gears, Shafts, Couplings**
#302 Stainless St. (Spring Temper)	MIL-W-6713 Comp. B	**Clear Passivate**	MIL-F-14072 Finish E300	**Springs, Washers**
#416 Stainless Steel	QQ-S-763a Class 6 Type A	**Clear Passivate**	MIL-F-14072 Finish E300	
#440 Stainless Steel	QQ-S-763a Class 10 Type A	**Clear Passivate**	MIL-F-14072 Finish E300	**Ball Bearings**
#420 Stainless Steel	AMS-5506 or SAE-51420	**Clear Passivate**	MIL-F-14072 Finish E300	**Retainer Rings**
#1020 Carbon Steel	QQ-S-633 Comp. FS1020	**Black Finish** **Heat Treat**	57-0-2C Type III Cl. A AN-H-201 Par. D-2	**Adjustable Clamps**
24ST Aluminum	QQ-A-268 Cond. T4	**Chromic Acid Anodized**	MIL-A-8625a Type I	**Gears, Spacer Posts**
24ST Aluminum	QQ-A-268 Cond. T4	**Black Anodized**	MIL-A-8625a Type I	**Dials**
#108 Cast Aluminum	QQ-A-601A Comp. 8 Cond. T55	**Chromic Acid Anodized**	MIL-A-8625a Type I	**Hangers, Breadboards**
Bronze (Tobin)	MIL-B-994a Comp. A			**Worm Wheels**
Oil-Less Bronze	MIL-B-5687a Type I Comp. A	**Lubricated With SAE 40 Oil**		**Bearings Thrust Washers**
Brass (Laminated)	MIL-S-22499B Comp. 2 Class 1			**Shims**
Beryllium Copper	MIL-C-6942			**Retainer Rings**
Nylon	MIL-P-17091B Type I			**Gears**
Phenolic	MIL-P-15035B Type FBE	**Fungus-Proofed**	JAN-T-152	**Gears**
Phenolic (Molded)	MIL-P-14P			**Terminal Boards**
Neoprene (Molded)	MIL-R-6855 Class II			**Flex. Couplings Pulley Belts**
Plexiglass (Clear)	MIL-P-5425B Finish A			**Indexes**
Grease	MIL-G-3278			**Gears, Bearings −65 F to +250 F**
Oil	MIL-L-6085a			**Gears, Bearings −65 F to +250 F**

SCREW HEAD DETAIL DATA

	FILLISTER HEAD SCREW					ROUND HEAD SCREW				FLAT HEAD SCREW			
													
Size	"A"	"B"	"C"	"D"	"E"	"A"	"B"	"C"	"D"	"A"	"B"	"C"	"D"
0	.096	.059	.023	.025	.021	.113	.053	.023	.039	.119	.035	.023	.015
1	.118	.070	.026	.031	.024	.138	.062	.023	.043	.146	.043	.026	.019
2	.140	.083	.031	.037	.028	.162	.070	.031	.048	.172	.051	.031	.023
3	.161	.095	.035	.043	.032	.187	.078	.035	.053	.199	.059	.035	.027
4	.183	.107	.039	.048	.035	.211	.086	.039	.058	.225	.067	.039	.030
5	.205	.120	.043	.054	.039	.256	.095	.043	.062	.252	.075	.043	.034
6	.226	.132	.048	.060	.043	.260	.103	.048	.067	.279	.083	.048	.038
8	.270	.156	.054	.071	.050	.309	.119	.054	.076	.332	.110	.054	.045
10	.313	.180	.060	.083	.057	.359	.136	.060	.086	.385	.116	.060	.053

	BINDING HEAD SCREW					OVAL HEAD SCREW				
										
Size	"A"	"B"	"C"	"D"	"E"	"A"	"B"	"C"	"D"	"E"
0	.125	.028	.022	.019	.007	.119	.035	.023	.030	.021
1	—	—	—	—	—	.146	.043	.026	.038	.025
2	.181	.046	.031	.030	.018	.172	.051	.031	.045	.029
3	.208	.054	.035	.036	.022	.199	.059	.035	.052	.033
4	.235	.063	.039	.042	.025	.225	.067	.039	.059	.037
5	.263	.071	.043	.048	.029	.252	.075	.043	.067	.041
6	.290	.080	.048	.053	.032	.279	.083	.048	.074	.045
8	.344	.097	.054	.065	.039	.332	.100	.054	.088	.053
10	.399	.114	.060	.077	.045	.385	.116	.060	.103	.061

DRILL & TAP REFERENCE DATA

LETTER SIZE DRILLS

Let.	Dia.	Let.	Dia.	Let.	Dia.	Let.	Dia.	Let.	Dia.
"A"	.234	"F"	.257	"K"	.281	"P"	.323	"U"	.368
"B"	.238	"G"	.261	"L"	.290	"Q"	.332	"V"	.377
"C"	.242	"H"	.266	"M"	.295	"R"	.339	"W"	.386
"D"	.246	"I"	.272	"N"	.302	"S"	.348	"X"	.397
"E"	.250	"J"	.277	"O"	.316	"T"	.358	"Y"	.404

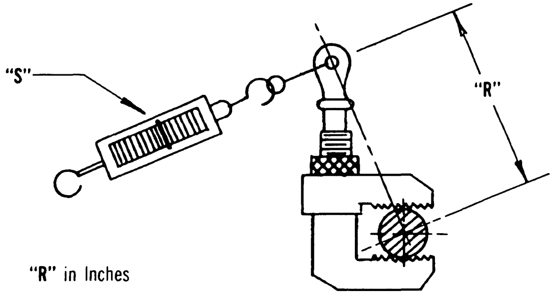
TWIST DRILL & STEEL WIRE GAGE

No.	Dia.	No.	Dia.	No.	Dia.	No.	Dia.	No.	Dia.
#1	.2280	#17	.1730	#33	.1130	#49	.0730	#65	.0350
#2	.2210	#18	.1695	#34	.1110	#50	.0700	#66	.0330
#3	.2130	#19	.1660	#35	.1100	#51	.0670	#67	.0320
#4	.2090	#20	.1610	#36	.1065	#52	.0635	#68	.0310
#5	.2055	#21	.1590	#37	.1040	#53	.0595	#69	.0295
#6	.2040	#22	.1570	#38	.1015	#54	.0550	#70	.0280
#7	.2010	#23	.1540	#39	.0995	#55	.0520	#71	.0260
#8	.1990	#24	.1520	#40	.0980	#56	.0465	#72	.0250
#9	.1960	#25	.1495	#41	.0960	#57	.0430	#73	.0240
#10	.1935	#26	.1470	#42	.0935	#58	.0420	#74	.0225
#11	.1910	#27	.1440	#43	.0890	#59	.0410	#75	.0210
#12	.1890	#28	.1405	#44	.0860	#60	.0400	#76	.0200
#13	.1850	#29	.1360	#45	.0820	#61	.0390	#77	.0180
#14	.1820	#30	.1285	#46	.0810	#62	.0380	#78	.0160
#15	.1800	#31	.1200	#47	.0785	#63	.0370	#79	.0145
#16	.1770	#32	.1160	#48	.0760	#64	.0360	#80	.0135

RECOMMENDED DRILL & TAP DATA
AMERICAN STANDARD — NATIONAL COARSE

Screw Size	Outside Dia.	Root Dia.	Tap Drill Data				Bottom Tap Drill Dp.	Clear Drill	
			Drill No.	Size	Dp.	Tap Dp.		Dr.No.	Size
# 1-64	.073	.053	#53	.059	.23	.16	.19	#46	.081
# 2-56	.086	.063	#51	.067	.25	.17	.20	#41	.096
# 3-48	.099	.072	#47	.0785	.30	.20	.23	#36	.106
# 4-40	.112	.080	#43	.089	.34	.23	.26	#31	.120
# 5-40	.125	.093	#38	.1015	.37	.25	.28	#29	.136
# 6-32	.138	.097	#36	.106	.42	.28	.31	#26	.147
# 8-32	.164	.123	#29	.136	.50	.33	.36	#17	.173
#10-24	.190	.136	#25	.1495	.56	.38	.41	#6	.204
AMERICAN STANDARD — NATIONAL FINE									
# 0-80	.060	.044	#56	.046	.18	.12	.15	#51	.067
# 1-72	.073	.055	#53	.059	.23	.16	.19	#46	.081
#2-64	.086	.066	#50	.070	.25	.17	.20	#41	.096
# 3-56	.099	.076	#45	.082	.30	.20	.23	#36	.106
#4-48	.112	.085	#42	.0935	.34	.23	.26	#31	.120
# 5-44	.125	.096	#37	.104	.37	.25	.28	#29	.136
# 6-40	.138	.106	#33	.113	.42	.28	.31	#26	.147
# 8-36	.164	.128	#29	.136	.50	.33	.36	#17	.173
#10-32	.190	.149	#21	.159	.56	.38	.41	#6	.204

How to MEASURE TORQUE



"R" in Inches

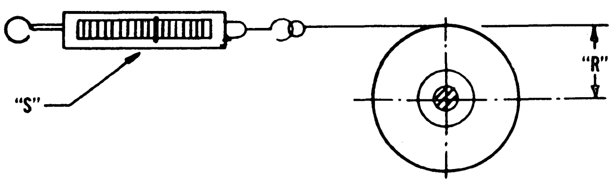
"S" in Pounds

"R" × "S" = Torque (in inch pounds)

STARTING TORQUE

Break-away friction is always greater than any succeeding friction. Therefore, starting torque is generally greater and must be considered so. Following above sketch, one can determine simply but accurately your starting torque.

RUNNING TORQUE

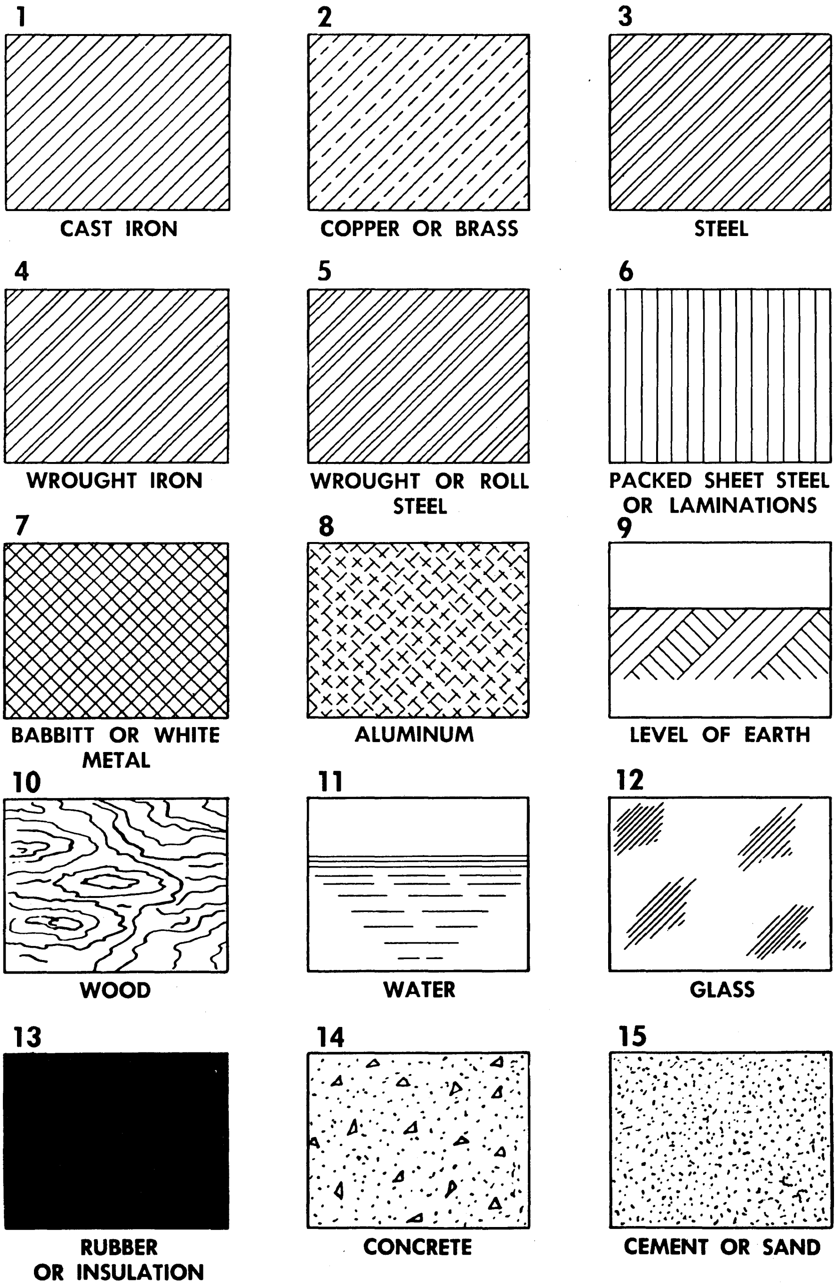


Running torque is the normal torque required for driving and therefore must be considered. Some uses have constant torque but most uses require varying torque. Therefore, take the highest reading when measuring the torque. To measure running torque, take several wraps of cord around a drum, attaching one end to the drum. The free end is pulled by the scale to rotate the drum at least 360° or more.

Multiply the highest scale reading by the drum radius, as in the sketch above, to obtain your torque.

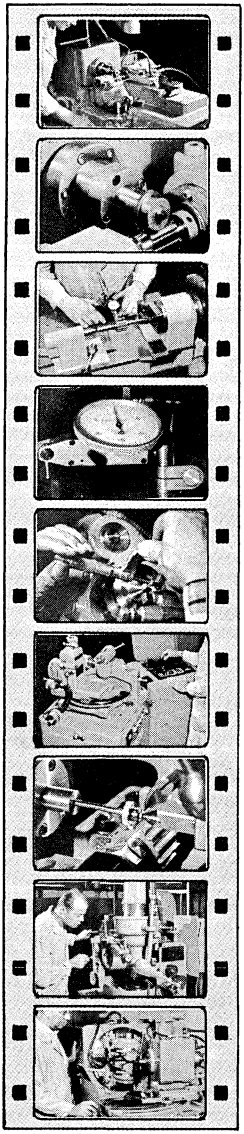
"R" × "S" = Torque (in inch pounds)

STANDARD CROSS HATCH SYMBOLS



DECIMAL EQUIVALENT

FRACTIONAL EQUIVALENT		DECIMAL EQUIVALENT	FRACTIONAL EQUIVALENT		DECIMAL EQUIVALENT
—	1/64″	.0156	—	33/64″	.5156
—	1/32″	.0312	—	17/32″	.5312
—	3/64″	.0468	—	35/64″	.5468
1/16″	—	.0625	9/16″	—	.5625
—	5/64″	.0781	—	37/64″	.5781
—	3/32″	.0937	—	19/32″	.5937
—	7/64″	.1093	—	39/64″	.6093
1/8″	—	.1250	5/8″	—	.6250
—	9/64″	.1406	—	41/64″	.6405
—	5/32″	.1562	—	21/32″	.6562
—	11/64″	.1718	—	43/64″	.6718
3/16″	—	.1875	11/16″	—	.6875
—	13/64″	.2031	—	45/64″	.7031
—	7/32″	.2187	—	23/32″	.7187
—	15/64″	.2343	—	47/64″	.7343
1/4″	—	.2500	3/4″	—	.7500
—	17/64″	.2656	—	49/64″	.7656
—	9/32″	.2812	—	25/32″	.7812
—	19/64″	.2968	—	51/64″	.7968
5/16″	—	.3125	13/16″	—	.8125
—	21/64″	.3281	—	53/64″	.8281
—	11/32″	.3437	—	27/32″	.8437
—	23/64″	.3593	—	55/64″	.8593
3/8″	—	.3750	7/8″	—	.8750
—	25/64″	.3906	—	57/64″	.8906
—	13/32″	.4062	—	29/32″	.9062
—	27/64″	.4218	—	59/64″	.9218
7/16″	—	.4375	15/16″	—	.9375
—	29/64″	.4531	—	61/64″	.9531
—	15/32″	.4687	—	31/32″	.9687
—	31/64″	.4843	—	63/64″	.9843
1/2″	—	.5000	1″	—	1.0000



TECHNICAL TRAINING FILMS

There is a growing trend among manufacturers and suppliers of precision mechanical components to provide technical and training films for use by their customers. These serve to introduce new products, and explain the proper use and handling of existing products.

Professionally produced, these films are usually available without charge, and vary in running time from ten to thirty minutes, depending on their subject matter.

Typical of these, are two films recently completed by PIC Design Corp., of East Rockaway, N. Y. – "Standardization of Precision Instrument Components" and "No-Slip Positive Drive Belts and Geared Pulleys", with a running time of thirty and ten minutes, respectively. Arrangements for the loan of these films, or for a showing by company personnel, may be made by contacting the above concern.

Most suppliers and manufacturers of precision parts and components will, upon inquiry, supply a list of such films available from their libraries.

INDEX

A

A.G.M.A.—Comparison Table....151
AN Standards 3
About the Author iii
Accuracy of Precision Gears 13
Adjustable Gear Centers 14
Adjust — Zero Couplings 98
Air Gauging — Parts 33
Alignment of Pulleys 90
American Gear Mfg's. Tables 151
Anodizing 156
Anti-Backlash Couplings 100
Anti-Backlash Spur Gears 27
Anti-Backlash Worm Wheels 61
Applications of Differentials 80
Assembly — Clean Rooms 6
Assembly of Hubs to Gears 26
Assembly Procedures 40

B

Backlash 29, 100, 153
Backlash Between Mating Gears 153
Backlash Couplings 100
Backlash in Spur Gears 29, 100
Barber Coleman Gear Generating 20, 41
Basic Problems 1
Bead Chain Drive 83
Bearing Installations 9
Bearing Preloading 9
Bearing and Shaft Fits 9
Bearings — Ball 9
Bearings — Fits 9
Bearings — Installation 9
Bearings — Plain 9
Bellows Couplings 97, 98
Bellows — Zero Couplings 98
Belt Drivers — Bead 84
Belt Drives — Chain 83
Belt Drives — Cog 85
Belt Drives — Flat 84
Belt Drives — Ladder 83
Belt Drives — Plastic Transmission 86
Belt Drives — Round 84
Belt Drives — Spring 83
Belt Drives — "V" 83
Bevel Gear Checking 73
Bevel Gear Differentials 77
Bevel Gear Speed Reducers....66, 73
Bevel Gears 72
Blind Assembly Couplings 101
Bore Size Specifications 11
Bores 11, 34
Brake — Magnetic 105
Breadboard Components 122
Breadboard Differentials 79
Breadboard Kits 123
Breadboards — Servo 119
Butting Racks 142

C

Captive Screws 145
Centers — Adjustable 14
Centers — Fixed 14
Centers — Gear 14
Chain Drives 83
Chain Drives — Bead 83
Checking — Bores 34
Checking — Outside Diameter....152
Checking Pressure for Gears.....152
Checking Spur Gears 37, 154
Circular Pitch 31
Clamp Hub 26
Clamped or Split Hub Gears 23, 25, 26
Clamps — Tool 125
Clean — Areas 6
Cleats — Motor and Syncro 144
Closed Wound Springs 148
Cluster Gears 27, 28
Clutch Brake 105
Clutch Coupling 97, 102, 103
Clutch Magnetic 105
Cog Belts 85
Coniflex — Bevel and Miter Generating 73
Contents — Table v
Couplings — Anti-Blacklash 97

Couplings — Bellows 97
Couplings — Clutch 97
Couplings — Errors 104
Couplings — Flexible 97
Couplings — In Line 97
Couplings — Multi-Jaw 97
Couplings — Oldham 97
Couplings — Shaft 97
Couplings — Sleeve 97
Couplings — Universal 97
Couplings — Zero Adjust 97
Cross Hatch Symbols 160
Cycle Error 97

D

Data for Screw Heads 157
Data for Tap and Drill Sizes 158
Deburring of Gears 11
Deburring Precision Gears 11
Decimal Equivalent Table 161
Design Errors 62
Dial Indexes 113
Dials — Disc 113
Dials — Drum 113
Dials — Vernier 113
Dials — Standards 116, 117
Dials — Tolerances 116, 117
Diamentral Pitch of Gears 31
Differentials 75
Differentials — Applications 75
Differentials — Construction 77
Disc Dials 113
Do's and Don't of Mech.
Assembly 40
Dowel Pinning 16
Drill and Tap Data 158
Drilling and Pinning Gears 27
Drives — Chain and Belt 83
Drum Dials 114
Dutch Pinning 24

E

Engineering Reference Data 151
Errors — Design 62
Errors in Couplings 104

F

Face Gears 75
Face Width Alignment of Gears 12
Fellows Gear Shaping 19
Fits — Shaft and Bearing 9
Fixed Gear Centers 14
Fixture Component 125
Flat Belt Drives 85
Flexible Couplings 97, 103
Formulae for Spur Gears 155

G

Gear Backlash 153
Gear Box — Construction 64
Gear Boxes 55,64
Gear Checking Tapes 36, 37
Gear Hobbing 20
Gear Materials 29, 156
Gear Schematics 10, 21
Gear — Specification .. 11, 13, 42, 151
Gear Tolerances 13
Geared Pulleys 93
Gearhead Reducers 57
Gearhead Speed Reducers 57, 65
Gears — Accuracy 33
Gears — Anti-Backlash 29
Gears — Bevel 69, 72, 73
Gears — Centers 28
Gears — Cluster 23
Gears — Helical 69, 70
Gears — Hubless 22, 23
Gears — Internal 147
Gears — Miter 67, 72, 73
Gears — Pin Hub 22, 25
Gears — Racks 141
Gears — Ratios 9
Gears — Split Hub 23, 25, 26
Gears — Spur 19, 23, 26
Gears — Thickness
(Face Widths) 12
Gears — Worm and Wheel 61
Gleason Coniflex Gear
Generator 73
Graphic Recording of Gear Data 38
Groove Pins 27
Ground Jig Plates 16, 125
Ground Shafting 9

H

Handling — Precision
Instrument Component 4
Helical — Gears 69
Helical — Speed Reducers 71, 66

Helical — Worm Gears 61, 69
Hob and Cutter Control 41
Hobbing Spur Gears 41
How to Measure Torque 157
How to Read a "True Blue" Tape 38
How to Select Spur Gears 19
Hubless Gears 22, 23
Hubs — Clamp 24, 25
Hubs — Pin 24, 25

I

In-Line Couplings 97
Instrument Components —
Basic 141
Instrument Plates (Ground) 16
Intermediate Drives 91

J

Jig Boring Plates 16
Jig Components 125
Jig and Fixture Parts 125
Joints — Universal 97

K

Kits — Breadboard 123

L

Ladder Chain 83

M

Machine Shop — Typical 8
Magnetic — Brake 105
Magnetic — Clutch 105
Materials 29
Materials for Gears 29, 156
Measuring and Checking
Pressures for Gears 152
Mechanical Differentials 75
Meshing — Helical Gears 71
Micro Inch Comparisons —
Typical 9
Military Material Specification .156
Mill Standards 3
Milling and Jib Boring 16
Miniature Chains 84
Miniature Couplings 99
Miniature Differential
Assembly 75
Miscellaneous Components 141
Miter and Bevel Gear Cutting
Equipment 73
Miter and Bevel Gear Drives 64, 69
Miter and Bevel Gears 72
Modifications to Precision Shafts 97
Motor Mounting 144
Multi-Jaw Couplings 97, 103, 104
Multi-Ratio Gear Boxes 57, 65

N

Non-Metallic Gears 30
"No-Slip" Belt and Geared
Pulley 86

O

Oldham Couplings 97, 99
Open Wound Springs 148

P

Packaged Speed Reducers 63
Passivating 156
Pin Hub 26
Pinning — Dowel 16
Pinning — Dutch 24
Pinning — Gear 26
Pinning — Grooved Pins 27
Pinning — Taper 26
Pitch Diameter 34
Pitch-Gear 11, 31
Plastic Transmission Belts 86
Positive Belt Drive 90
Precision Chain Drives 84
Precision 1, 2 and 3 Spur
Gear-Tables 32
Precision 3 Gear Hobbing 32
Precision Ground Instrument
Plates 16
Precision Instrument
Components 4
Precision Racks 141
Precision Shafts 5
Preface ii
Preloading Ball Bearings 40
Pressure Angle 31
Pulleys — No-Slip 93
Pulleys — Round Belt 84
Pulleys — Types 92

R

Racks — Spur Gears 141
Recommendations for Meshing
Gears 22
Reduction Ratios 57

Reversing Motion via Belt Drives ... 89
Right Angle Drives ... 69, 73
Round Belt Drives ... 85
Rules and Formulae for Spur Gears ... 152

S

Sampling Procedures ... 46
Schematic – Gear ... 21
Screw Head Data ... 157
Screws – Captive ... 145
Screws – Tool Component ... 125
Selecting Magnetic Clutches ... 111
Servo Breadboards ... 119
Servo-Gear Boxes ... 55,64
Shaft and Bearing Fits ... 7, 9
Shaft Couplings ... 97
Shaft Extensions ... 97
Shaft Modifications ... 97
Shafts – Precision ... 7
Shafts – Ultra Precision ... 7
Shaped Gears – Fellows ... 19
Side Wobble ... 17
Sleeve Couplings ... 97
Specifications – Military Material ... 156
Specifications and Standards for Gears ... 42
Specifications and Tolerances for Gears ... 156
Speed Reducer Data ... 58, 61
Spring Belts ... 83
Springs ... 146
Sprockets ... 83
Spur Gear Cutting Equipment ... 20
Spur Gear Data ... 152
Spur Gear Differentials ... 78
Spur Gear Formulae ... 154
Spur Gears ... 19
Staking – Three Notch ... 19
Standard Cross Hatch Symbols ... 160
Standard Gear Test Provisions ... 152
Standardized Tool Components ... 125
Stock Room ... 2
Surface Roughness of Gear Teeth ... 45
Syncro Cleats ... 144

T

Table of Contents ... v
Table of Decimal Equivalents ... 161
Tables – Formulae ... 155
Tap and Drill Data ... 158
Tape – Gear Checking ... 36, 37
Taper Pinning ... 26
Technical Reference Data ... 151
Throated Worm Wheels ... 61
Timing Belt Drives – Cog ... 83
Timing Belt Drives – No-Slip ... 83
Tolerances – Bores ... 17
Tolerances – Outside Diameter ... 17
Tolerances – Pitch Diameter ... 17
Tool Components ... 125
Tooth to Tooth Error ... 17, 34
Torque – How to Measure ... 157
Torque – Magnetic Clutch and Brakes ... 109
Torque Measurements of Differentials ... 81
Torque Tests for Gear ... 157
Total Composite Error ... 34
Transmitting Angular Motion ... 70
True Blue Tapes ... 38
Types of Spur Gears ... 22
Typical Machine Shop ... 8

U

Ultra Precision Gears ... 4, 9
Ultra Precision Shafts ... 9
Universal Couplings ... 97, 104
Universal Joints – Single ... 97, 104
Universal Joints – Double ... 97, 104

V

"V" Belts ... 83
Vernier Dials ... 113

W

Wheels – Worm ... 61
Work-Clean Areas ... 6
Worn Gear – Reasons ... 37
Worm Gear – Design Errors ... 61
Worm Gear – Disadvantages ... 61
Worm Gears ... 61
Worm – Speed Reducers ... 63

Z

Zero Adjust Coupling ... 98