SCR-83

Sandia Corporation

REPRINT

POSITIONAL TOLERANCING AT SANDIA CORPORATION

DESIGN DEFINITION DEPARTMENT

MAY 1959

Prepared for: The Industrial Education Institute Conference on Dimensioning and Tolerancing, Chicago, Illinois, May 1959.

Printed in USA. Price $1.00. Available from the Office of Technical Services, Department of Commerce, Washington 25, D.C.

SCR-83
DRAFTING
(TID-4500, 14th Ed)

SANDIA CORPORATION REPRINT

POSITIONAL TOLERANCING AT SANDIA CORPORATION

Design Definition Department

May 1959
First Reprinting May 1959

ABSTRACT

This brochure presents a brief introduction to the Positional Tolerance Method of Dimensioning as employed by Sandia Corporation. The emphasis is placed on the elimination of ambiguities and increase in tolerances provided by this method as compared to the older, bilateral method.

EDITED BY

F. F. Eichert
P. A. Nicovich

CONTRIBUTORS

R. O. Dell
V. D. Melillo
W. G. Parkhurst
E. S. Roth
C. S. Wolowicz

TABLE OF CONTENTS

INTRODUCTION

Within the past ten years, a modification to the standard method of dimensioning engineering drawings has been developed. Eliminating ambiguities and increasing tolerances as compared to the older method, the new method, Positional Tolerance Dimensioning, has been received with favor and is now widely accepted. To be fully effective, however, the system should be used, or at least understood, much more widely than it now is. Sandia Corporation, which has used the new method since 1955 with much success, presents this brochure in the hope that it will contribute to the more extensive use and understanding of positional tolerancing dimensioning.

In this brief brochure, the aim is not to give a definitive analysis of the new method. Rather, the aim is to introduce the reader to its advantages by presenting some of the background and some of the basic concepts as applied at Sandia Corporation. A larger view of the field of positional and geometrical tolerancing is given in more detailed reports such as those by W. H. Harrington in the American Machinist Reference Book Sheets (June 16, 1958; August 11, 1958; and October 6, 1958).

We will seek to show that positional tolerancing can coordinate more precisely than heretofore possible the data requirements of the engineer, designer, machinist, inspector, and manufacturer. In so doing, we shall try to point out where the new system is less ambiguous, more logical, more lenient in tolerances (subject to requirements of design), better able to specify such dimensions as clearance hole sizes of mating parts, better able to specify sizes of inspection gages, better able to provide parts for assembly with 100 percent assurance of fit, and finally, accomplish all these improvements without an excessive rejection rate.

BACKGROUND

Sandia Corporation interest in positional tolerancing derives from its relationship to its supplier and assembly plants. Primarily a laboratory assigned by the Atomic Energy Commission with the responsibility for engineering design of nuclear weapons, Sandia Corporation is separated—often by thousands of miles—from the plants that manufacture and the plants that assemble the parts it designs. The separation of functions obviously necessitates extreme design and manufacturing care. Just as obviously, accurate and unambiguous transmission of engineering information is essential.

The difficulties inherent in manufacture with large quantities of subcontracted units became painfully apparent during World War II. The British, embroiled in the same problems, found that engineering information was not being accurately transmitted and that the essential reason was the inherent ambiguity in the bilateral tolerancing system.*

Research into the problem disclosed that too great a latitude of interpretation of drawing information was allowed. The designer, the machinist, and the inspector could pick different points of origin for their measurements. Each could have a valid reason for his choice, but parts could be accepted or rejected almost arbitrarily with respect to actual function.

The entire dimensioning system was re-examined, and the reconstruction resulted in the Positional Tolerance Dimensioning system. This dimensioning method was accepted as the standard for British industry in 1953 with the publication of British Standard 308, Engineering Drawing Practices.

The Canadian Standard B78.1; was published a year later; except for detail changes, it was a duplicate of the British standard.

American participation in and investigation of this system has been carried out by S.A.E., A.S.A., and the Department of Defense.

The American-British-Canadian (ABC) Committee for International Drawing Standards has agreed unanimously to adopt positional tolerancing.

Sandia Corporation investigation of the system began with the publication of the British standard. The system was adopted by us officially with the publication in 1955 of SC-7016(Sp) Tolerances of Position and Form, including MMC.

The standards, with publication dates, are listed below in Table I.

TABLE I
Positional Tolerance Standards

June 1953	Military Standard 8A	December 1954	S.A.E. Standard
December 1953	British Standard 308	September 1955	Sandia Corp. Standard SC-7016(Sp)
July 1954	Canadian Standard B78.1	September 1957	A.S.A. Standard Y14.5

*"Bilateral tolerancing" refers to the method of tolerancing by means of ± callouts.

THE BILATERAL METHOD OF DIMENSIONING

We have said that the Bilateral Method of Dimensioning is inherently ambiguous. Let us look now at this charge and see whether we have been unjust. To do so, let us consider a fairly simple example of a part dimensioned by the bilateral method. This example, given below in Figure 1, is a plate with a rectangular pattern of four clearance holes. The intent is to describe a part that may be fastened to an identical plate by four bolts, 1/4 inch in diameter. Misalignment of the edges is permissible.

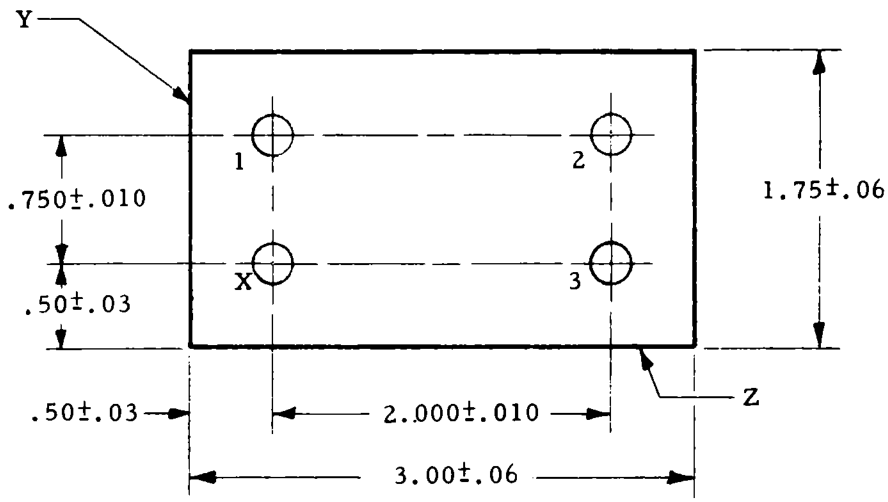


Figure 1

The drawing appears to be complete, except for clearance hole sizes, which have been intentionally omitted. Can we now go ahead and transmit the drawing with complete assurance that the manufacturing and inspecting functions will interpret the drawing as we intended? Probably not, as the following questions will show.

1. Does the .50 ± .03 dimension apply to both holes X and 1 from surface Y?

2. Does the .50 ± .03 dimension apply to both holes X and 3 from surface Z?

3. Can holes 2 and 3 each vary from the surface Y by the total amount of tolerances indicated?

4. Can holes 1 and 2 each vary from surface Z by the total amount of tolerances indicated?

5. Are holes 1, 2, and 3 located from hole X as a pattern within the ±.010 tolerance shown?

6. If question 5 is answered with a "yes," what is the horizontal tolerance allowed for hole 1, or the vertical tolerance for hole 3?

7. By the dimensions shown can the shape of the locational tolerance zones for holes 1, 2, and 3 be determined?

8. What would be the correct clearance hole size to meet the design requirements?

9. Is the part defined sufficiently for open-setup inspection?

10. How square is surface Y to surface Z?

11. Where is the starting point for taking measurements?

12. Assuming that the pattern of holes is intended to be within the ±.010 shown, are the dimensions shown parallel and perpendicular to holes X and 1, holes X and 3, surface Y, or surface Z?

13. Does the tolerance of location determine the allowable squareness for each hole?

14. Could simple fixed-pin gages be used to inspect the pattern of holes?

15. What size gage pins would be used?

16. May holes 1, 2, and 3 be outside of the limits of location specified and still be functional?

Answers can of course be given to each of the questions. But remembering the separation of design, manufacturing, and assembly locations, do we have any assurance that the answers will everywhere be the same? Experience at Sandia Corporation has been that they will not.

POSITIONAL TOLERANCE DIMENSIONING

It will be the contention of the remainder of this brochure that the ambiguities pointed out by the questions in the previous section are eliminated by the positional tolerance method of dimensioning.

To begin our description of the new method, let us refer to our earlier example of a plate with four holes, as redrawn in the positional tolerance method.

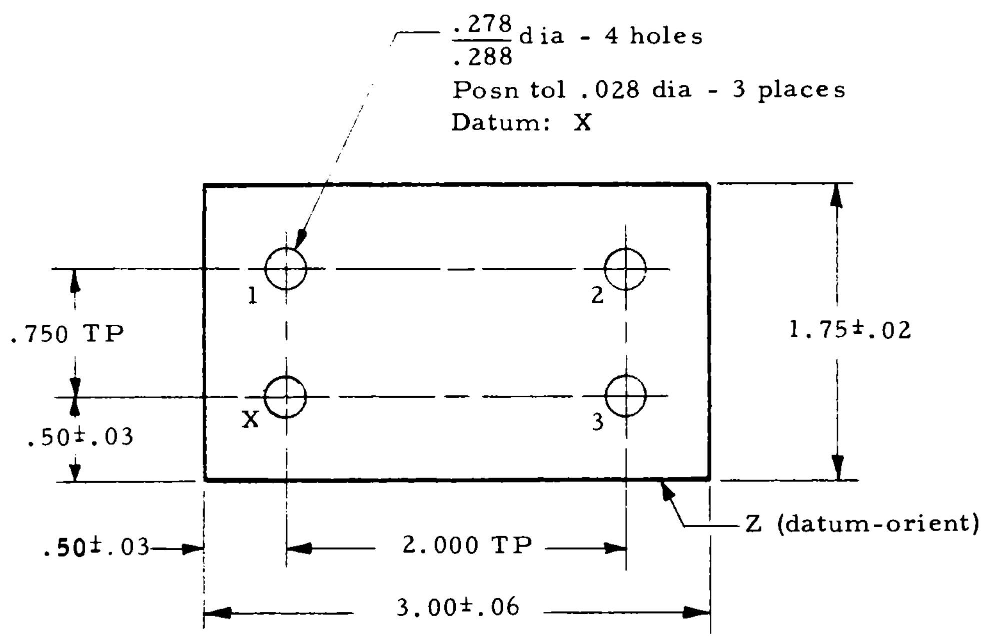


Figure 2

We can see at once that, although several callouts are different, positional tolerancing is not a total revolution from bilateral tolerancing. Rather, it is used in conjunction with accepted bilateral tolerance methods. The plate is still defined by coordinate methods, the overall dimension being the same as in Figure 1 and called out in the same manner. The differing callouts are Datum: X, Datum-Orient, and TP. These new terms are the keys to elimination of ambiguity.

Datum -- Datum is defined as a specific and clearly indicated point of origin for dimensions and measurements. As so defined, it must be used by everyone working with the drawing, and, therefore, no ambiguity can arise. In the event that a datum cannot be readily distinguished on the actual part, the drawing should specify that the part be marked with a symbol to identify the single feature chosen as datum.

In Figure 2, datum is indicated by the callout Datum:X on the lower left hand hole. The center of this hole is the datum, then, for dimensioning the rest of the drawing. The location of Datum:X is determined by the normal bilateral method.

Datum-Orient -- Datum-Orient stands for datum for orientation and is defined as a specific and clearly indicated surface or line that orients the direction of measurement. All measurements are therefore made perpendicular and/or parallel to the datum-orient. In Figure 2, the bottom edge of the plate is the datum-orient.

TP -- TP stands for True Position, defined as the theoretical location of a point, line, or plane. True position, that is, is the desired location. Coordinate dimensions have, of course, always indicated a desired position, with bilateral tolerances used to limit deviations from this position. In this respect, true position represents no change. True position, however, includes the additional concept of measurement only from Datum:X parallel and perpendicular to Datum-Orient. It specifically excludes measurement from features that appear to be more convenient.

True positioning dimensioning is not automatically done. Rather, it must be called out after any dimension to be measured by the positional tolerance method.

True position of hole 1, for example, is .75 inch from Datum:X on a line passing through Datum:X and perpendicular to Datum-Orient. True position for hole 2 is the intersection of a line 2 inches to the right of Datum:X and perpendicular to Datum-Orient and a line .75 inch above Datum:X and parallel to Datum-Orient. True position for hole 3 is 2 inches to the right of Datum:X on a line passing through Datum:X and parallel to Datum-Orient.

The vertical surface plays no part in locating the hole, and the squareness of the corner is unimportant. Figures 3a and 3b illustrate the difficulties under the bilateral system of locating the datum if the corner is not square. Figure 3c shows the method used in positional tolerance application of the bilateral system.

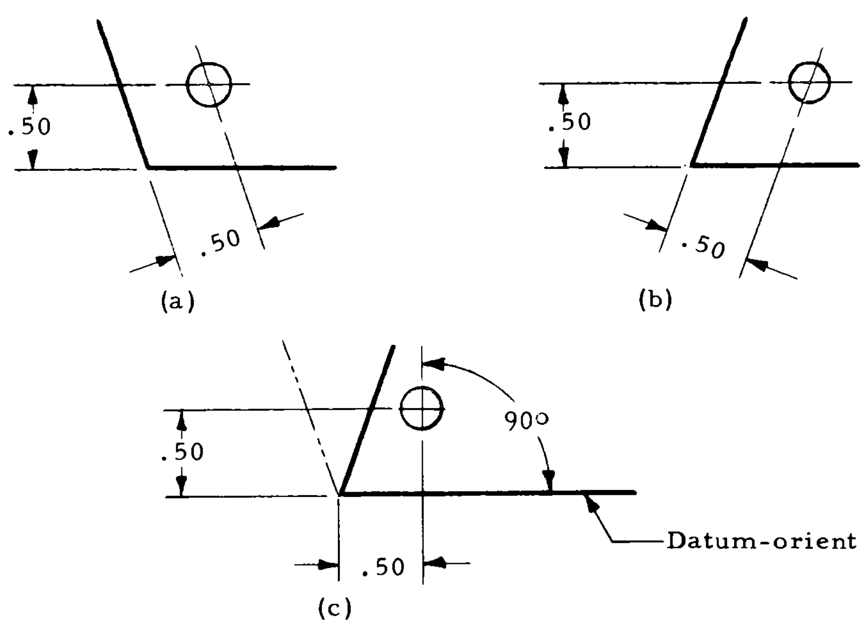


Figure 3

In Figure 4 we have given a ±.03 tolerance for the location of the center of hole X. This tolerance forms a .06 square zone, within which the center of hole X must lie. Once it is established that the center of hole X does in fact fall within this zone, the ±.03 ceases to have importance in locating the hole pattern, and the question of adding this tolerance to the .010 tolerance from the center of hole X does not arise. We should remember, though, that regardless of where the center of the hole falls within the .06 tolerance, it is still the datum.

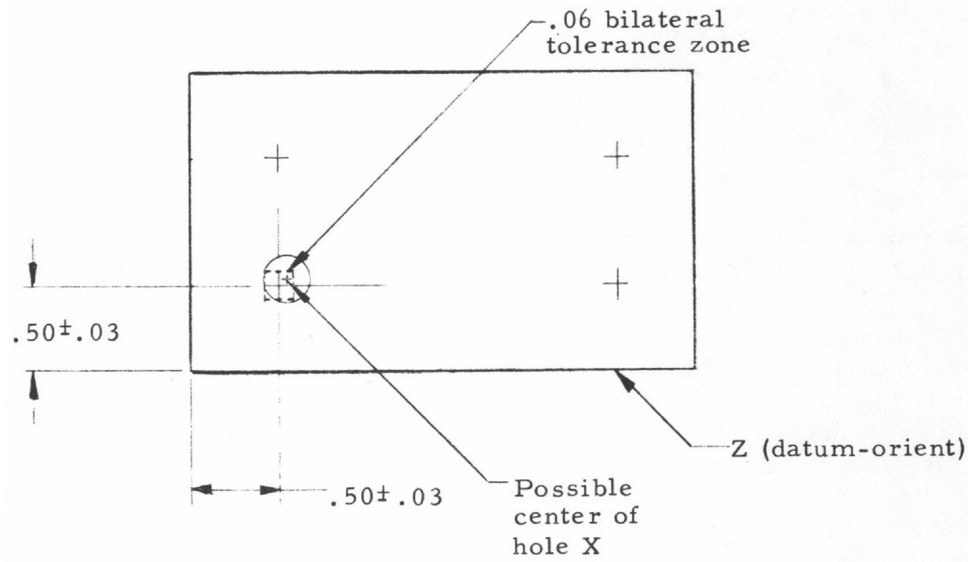


Figure 4

In Figure 5, below, we introduce a new callout "Posn Tol .028 Dia 3 Places." Posn Tol means, clearly enough, positional tolerance. It is defined as a circular tolerance zone, concentric about true position, within which the center of the hole must lie. The callout ".028 Dia" thus describes the tolerance zone.

The change from the square bilateral tolerance zone to the circular positional tolerance zone requires some explanation.

During the re-examination of the bilateral dimensioning method it was realized that the square tolerance zone given by ± callouts actually allowed tolerances greater than the stated tolerance, namely, at or near the corners. It therefore appeared logical that if a part would function properly when a feature fell on a corner it would work equally well if it fell anywhere within the distance determined by the diagonal to the corner, that is, within a circle. In our example of a ±.010 tolerance, that diagonal is .014 ($\sqrt{.010^2 + .010^2} = .014$).

The diagonal distance from the center of the square tolerance zone to the corner thus becomes the radius of a circle circumscribed about the square. The diameter of that circle (.028) becomes the tolerance callout. (See Figure 6.)

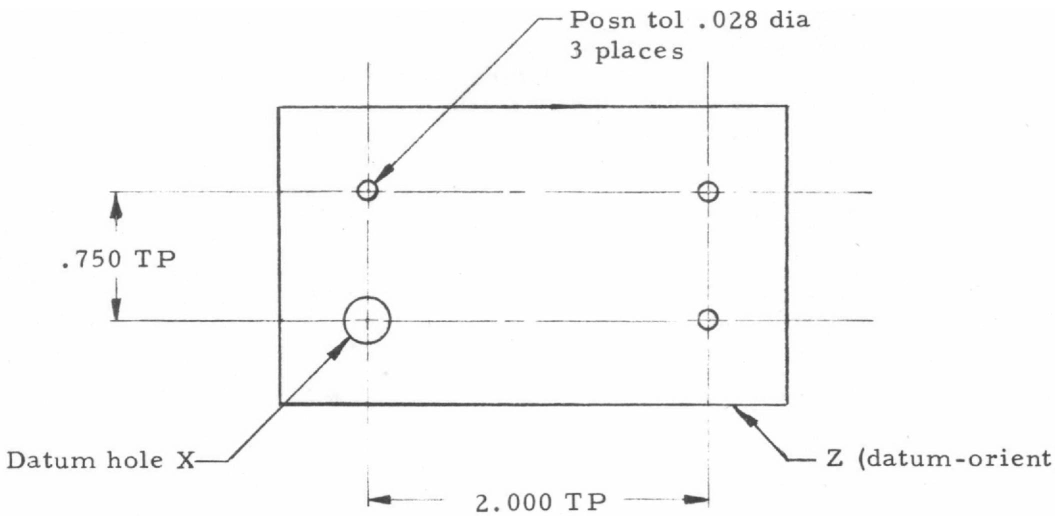


Figure 5

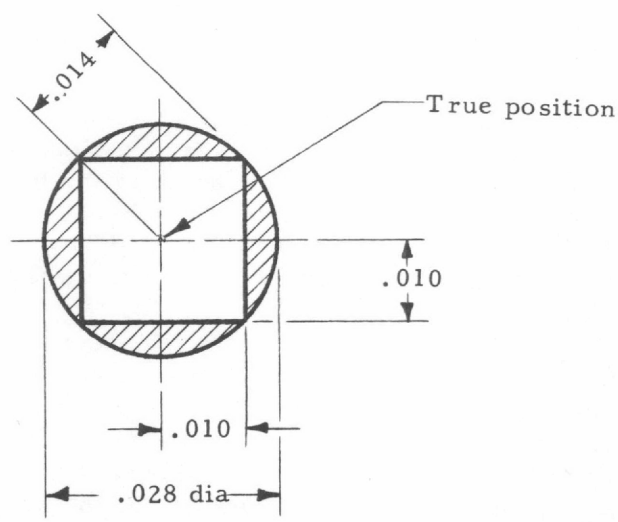


Figure 6

By changing our tolerance zone from a square to a circle we have increased the tolerance about 57 percent and at no cost to function or design intent. Rejection of parts is reduced at the same time by the number of parts in which a feature falls in the segments between the square and the circle.

Clearance Holes -- The assembly of the plates in Figure 2 is dependent upon the size of the clearance holes through each of the two plates. How do we determine the clearance hole size?

Since the plates are dimensioned the same, the datum and true position points of each plate can be aligned when placed one upon the other.

Figure 7a shows hole 2 in both mating parts. These holes are placed to their extreme tolerance limits and in an opposite direction from each other (.014 radius from true position or .028 distance apart.) At this worst condition the design requirement must be met. A .250 diameter bolt must pass through both holes as illustrated in Figure 7b.

With the holes in their worst condition as shown, a .139 radius (.278 diameter) is computed as minimum acceptance of a .250 bolt through both parts. The derivation is as follows.

The maximum deviation allowed from TP for each plate is the radius, that is, one-half the positional tolerance diameter assigned. If the diameter is the same for both plates, as in our example, we have only to add the positional tolerance diameter to the maximum bolt diameter to obtain the minimum clearance hole size.

A more general formula for determining the minimum clearance hole size, since the positional tolerance assigned to the holes in each plate may not be the same, is as follows: 1/2 Posn Tol Dia. of plate 1 plus 1/2 Posn Tol Dia. of plate 2 plus the maximum bolt diameter equals the minimum clearance hole size

Since the positional tolerance diameter (.028) corresponds to the distance across the corners of the bilateral tolerance (±.010), as explained above, the same size minimum clearance hole is indicated for either the bilateral or the positional method.

The method stated above for determining clearance hole sizes is based on 100 percent interchangeability; that is, when the holes are located in their worst possible condition with respect to one another the parts are completely interchangeable at assembly.

Inspection of Parts Dimensioned by Positioned Tolerances -- If holes 1, 2, and 3 in Figure 1 (which, we recall, was dimensioned bilaterally) are measured from hole X and each is within a ±.010 tolerance zone, then, in order to inspect the pattern of holes with a simple fixed-pin gage, the pins for checking the holes must be .020 smaller in diameter than the size of the holes.

The parts accepted by this inspection gage would be those parts whose centers fall in a circular zone inscribed in the .020 square bilateral zone as illustrated in Figure 8. This type of gage covers only 78 percent of the tolerance allowable.

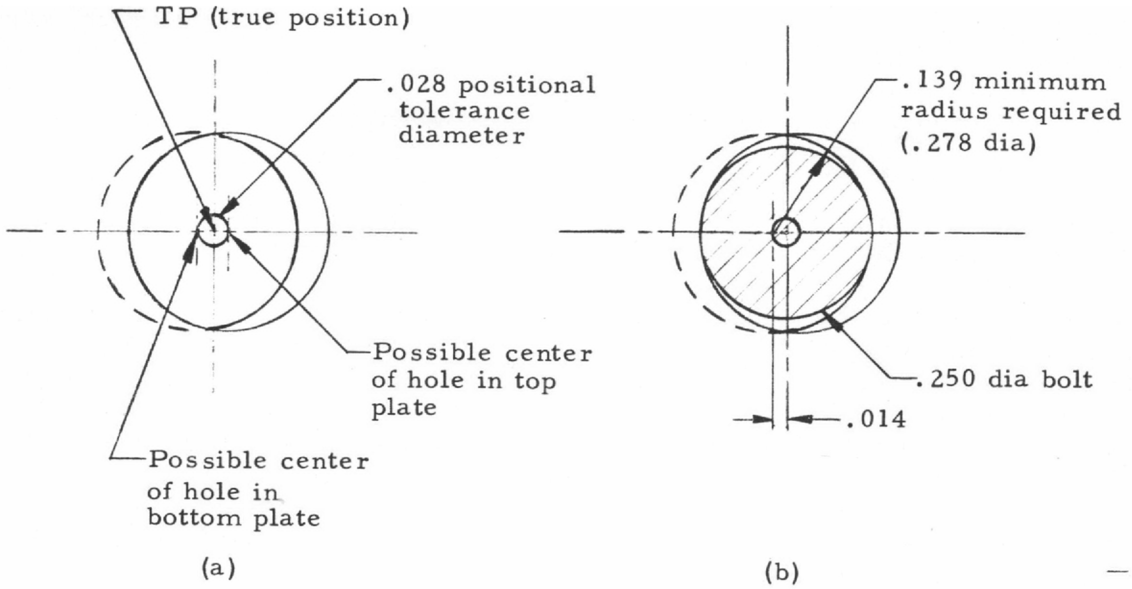


Figure 7

Parts not passing this acceptance criterion should not be rejected until they are reinspected by open setup. This method of inspection can determine whether or not the hole centers fall within the corners of the square bilateral zone, as indicated by the shaded area in Figure 8.

Open-setup methods are, however, relatively slow and expensive. A single inspection with a fixed pin gage is much simpler and less expensive and, for the example used, can guarantee 100 percent interchangeability of mating parts. One of the significant advantages of the positional tolerance method is simplification of design for such gages. In our example, gage pin diameter is the specified minimum hole size less the specified positional tolerance diameter.

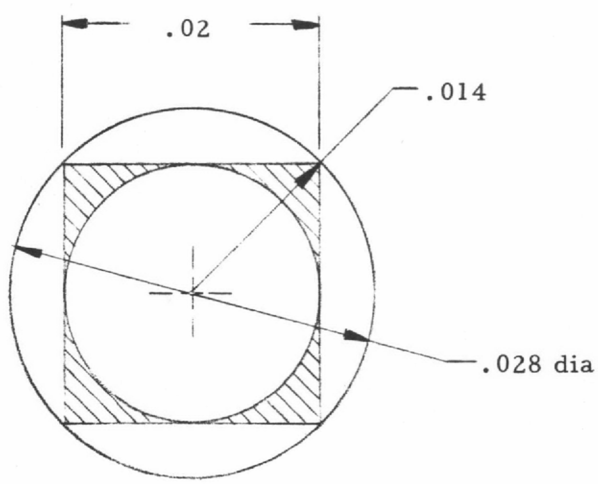


Figure 8

MAXIMUM MATERIAL CONDITION

Maximum Material Condition (MMC) is defined as the maximum limit of size of an external feature and the minimum size of an internal feature.

For example, the 1/4 inch bolt shown in Figure 7 may vary in size from .2435 diameter to .2500 diameter. The bolt will be at Maximum Material Condition when, since it is an external feature, it is .2500 diameter.

Since they are internal features, the clearance holes will be at Maximum Material Condition, when they are .278 diameter.

In Figure 7a we saw that when the two holes were placed in their worst condition for assembly they still allowed a .250 bolt to go through them. It is obvious that if the holes in either plate were increased in size, their relative positions could deviate even more from true position and still allow assembly.

As the clearance holes increase in size from MMC, their position relative to true position may increase proportionately and still ensure assembly. This benefit is incorporated in positional tolerance dimensioning with the callout: Posn Tol .028 dia (MMC).

The callout indicates that the circular tolerance zone of .028 diameter is the maximum tolerance zone allowable at the Maximum Material Condition (.278 diameter hole size) of the clearance holes; however, if the holes are larger by some amount than their minimum specified size, the positional tolerance zone may be increased in diameter by the same amount.

The table below illustrates this point.

TABLE II

Hole Size	Positional Tolerance Zone Diameter
.278	.028
.2785	.0285
.279	.029
.2791	.0291
.280	.030
.288	.038

Allowing the positional tolerance to increase as the hole size increases is, we think you will agree, a much more realistic approach than was taken with the bilateral method. In the bilateral method the holes had to be within tolerance regardless of their size.

A further increase in pattern tolerance may be given by applying the positional tolerance and MMC concepts to the datum itself, rather than, as in the discussions above, simply to the center of hole X. That is, we permit the datum to lie somewhere in a datum tolerance zone around the center of the feature that establishes the datum. The datum tolerance zone may be defined (1) by applying MMC to the datum, (2) applying datum tolerance to datum, or (3) applying both tolerances to datum. It must, however, be understood that the relaxations of tolerance are not automatic; they must be called out only after possible effects on design intent have been considered.

MMC to Datum -- Let us look first at the MMC concept applied to datum as in Figure 9.

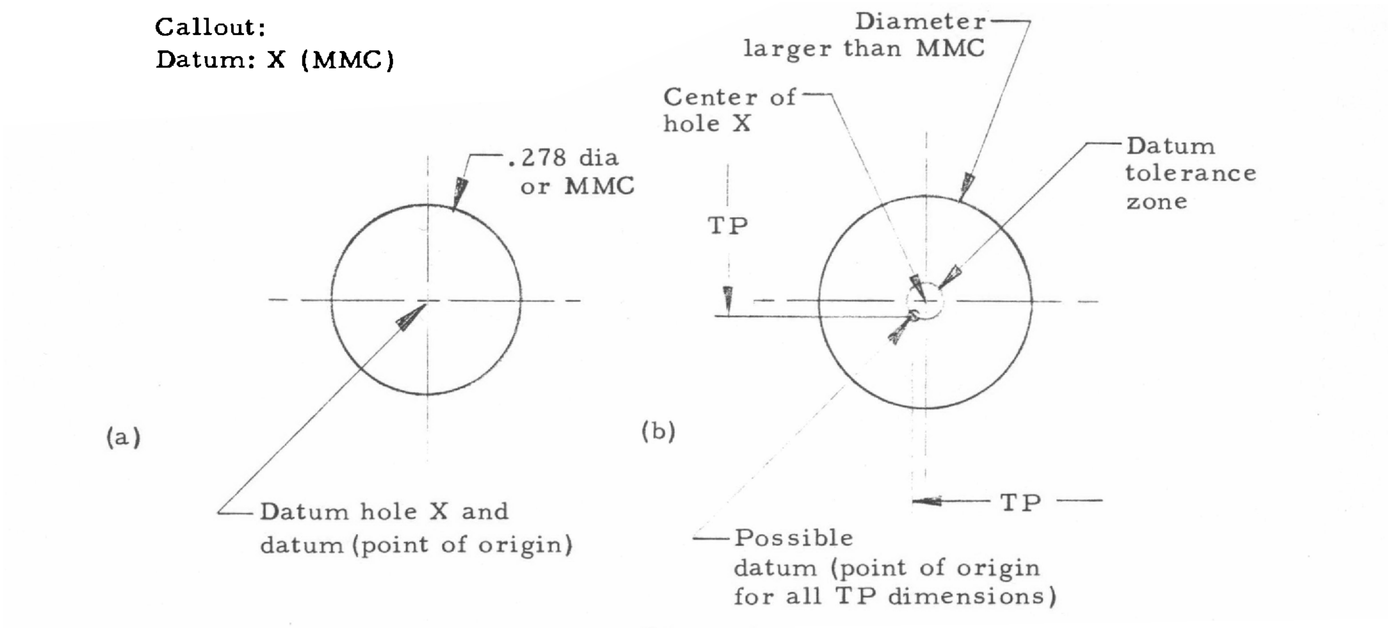


Figure 9

The callout indicates, by definition, that the datum is at the center of hole X when hole X is at its specified minimum size. But when hole X is larger than MMC, the datum may deviate from the center of hole X within a tolerance zone concentric about the center of hole X and equal in diameter to the difference between the actual hole size of X and its specified minimum size. As indicated by the callout .278/.288, the hole may be as large as .288, and the datum may lie anywhere within a .010 diameter zone—or as far as .005 from the center of hole X.

The difference between MMC and actual hole size thus determines a datum tolerance zone anywhere within which the datum may lie. This datum must, however, remain the same for every measurement made from it if interchangeability is to be ensured.

Datum Tolerance to Datum -- A tolerance may be applied to datum just as to TP dimensioned features. The callout would be Datum:X, Datum Tol .028 Dia. Figure 10 illustrates the application.

The center of hole X must fall within the bilateral tolerance zone specified, i.e., ±.03. The center of hole X now acts as a true position concentric about which the .028 dia. datum tolerance zone is formed. The datum (point of origin, common for the TP dimensions) may fall anywhere within this zone.

Datum Tolerance and MMC to Datum -- Finally, we may apply MMC to the datum tolerance zone, that is, let the datum deviate from its position as defined above still further by virtue of MMC. The effect is to increase the datum tolerance zone by the difference between MMC and the actual feature size. The callout is Datum:X

Datum Tol .028 Dia (MMC). The total possible tolerance for the example we have been using is .038 diameter, .028 for datum tolerance and .010 for maximum deviation from MMC.

The use of the callout Datum Tol .028 Dia (MMC) gives us a part description for inspection with which we can use a truly functional gage. For the example, the gage would have four fixed pins of .250 diameter, each pin being located at true position.

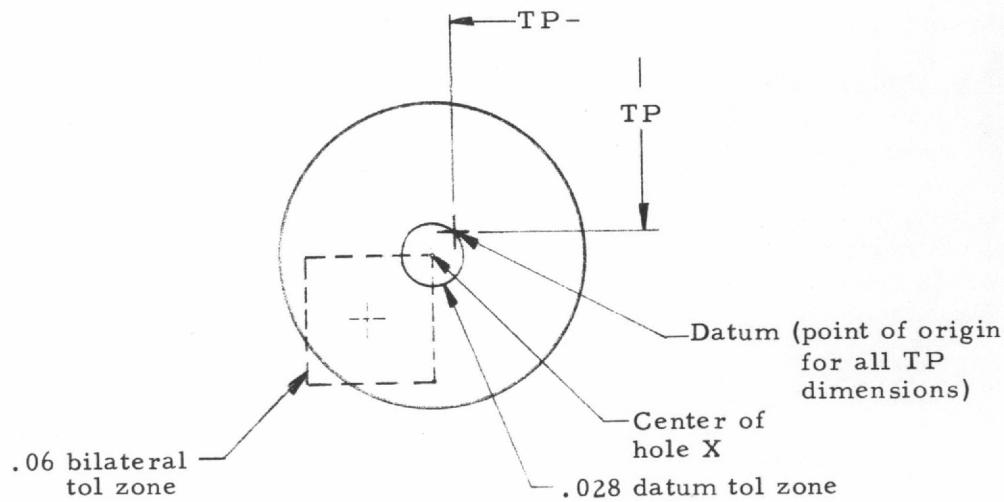


Figure 10

Bilateral and Positional Tolerances Compared -- At Sandia Corporation, positional tolerance dimensioning has been found to define a part more precisely and at the same time to provide wider tolerances than bilateral tolerance dimensioning. The reaction of suppliers, however, has often been one of surprise and dismay at a supposed tightening of tolerances.

Nothing could be further from the truth. In order to show the error in this view let us outline the increases over bilateral tolerancing that positional tolerances allows. The discussion will also serve as a brief review. Our example will, as before, be the hole pattern in a plate, considering this time, though, holes X and 1 only. The example appears as Figure 11.

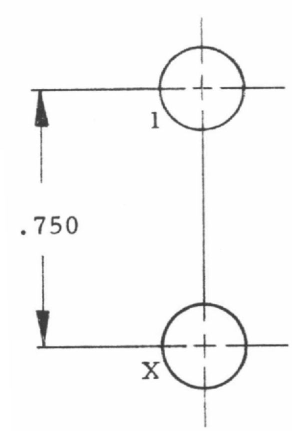


Figure 11

First, in the bilateral method. The tolerance of ±.010 allows the center of hole 1 to lie not more than .760 from the center of hole X (stated dimension .750 plus maximum deviation from desired position, or .010).

We have a number of possible tolerances to be applied if our drawing is dimensioned by positional tolerancing. First, by converting the square tolerance into a circular tolerance we obtain a maximum distance from desired position of .014.* Added to the stated dimensions, maximum separation is .764. Assuming that the hole is finished to .288, or .010 over MMC, the MMC callout permits us to add half the difference between MMC and actual hole size, and the distance increases to .769 to the distance between centers.

By assigning a datum tolerance of .028 diameter to the datum (hole X), the maximum distance between the holes becomes .783 (.769 + .014). Finally, adding MMC to the datum tolerance callout results in an allowable total separation of the holes of .788.

Comparison of the two maximum tolerances should clearly dispel the belief that positional tolerancing means a tightening of tolerances. The bilateral tolerance method allows a +.010 deviation; the positional tolerance method allows a +.038 deviation from the nominal dimension.

---

*In earlier discussions we added tolerance increases to the diameter of the tolerance zone. For the example here, maximum deviation of a point (center of the hole) from the center of the tolerance zone to the circumference is, of course, the radius. Tolerance increases are therefore one-half the increases in diameter discussed above.

THREE-DIMENSIONAL TOLERANCE ZONES

In the explanations we have so far made of tolerance of position and MMC, the tolerance zone was considered to be a circle about a true position point.

The actual part however, has a third dimension, depth, and we must establish a third dimension for the tolerance zone as well. By adding a dimension of height to the circular zone, usually equal to the length or depth of the feature, we establish a cylindrical tolerance zone.

Similarly, the true position is not a point but, rather is a line which extends through the length or depth of the feature.

This concept is illustrated in Figure 12a. Hole 3 is shown as viewed from the side of the plate.

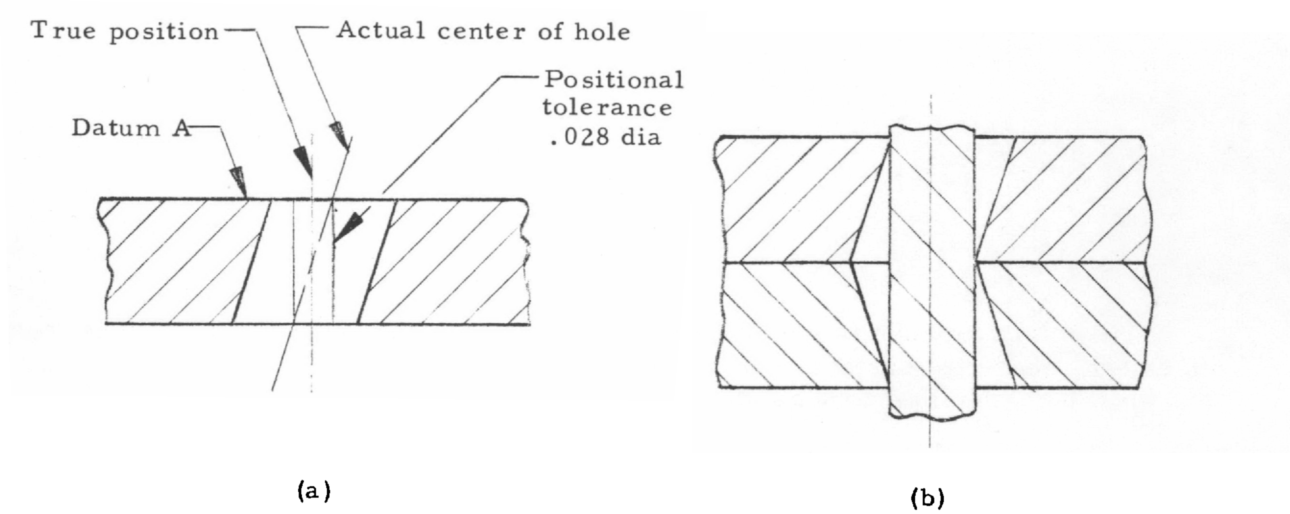


Figure 12

The true position line is perpendicular to the datum surface of the plate. The .028 positional tolerance zone, since it is concentric about the TP line, is perpendicular to the surface also.

The feature, in this case a hole, must be within the specified tolerance throughout its entire length. We have shown the actual center of the hole falling at the extreme tolerance limit to the right at the top surface and at the extreme tolerance limit to the left at the bottom surface. Although the hole is not square to the datum surface, it is within tolerance through the depth of the plate and will accept a .250 dia. gage pin that is at true position and perpendicular to the datum surface.

Figure 12b shows hole 3 in both plates seen from the side. The upper hole has angled, from the mating surfaces, upward to the right; the lower hole has angled, from the mating surfaces, downward to the right. Both holes are within tolerance through the depth of their respective plates. A .250 dia bolt can be placed through them at the worst condition of assembly.

When positional tolerance is specified to fixed pins or studs the tolerance applies from the mating surface to the height of the pin or stud extending beyond that surface.

Figure 13a illustrates the positional tolerance method of callout.

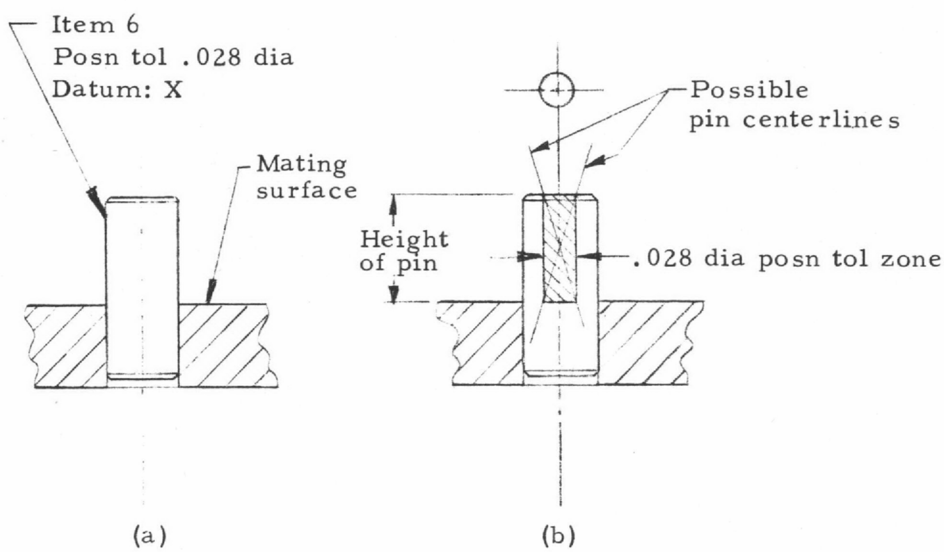


Figure 13

The tolerance zone is a cylinder .028 in diameter and equal in length to the height the pin extends above the surface. The centerline of the pin that extends beyond the mating surface must be within this zone as shown in Figure 13b.

The portion of the pin that falls below the mating surface has no function. Its location, then, is not important; only the portion that extends from the mating surface need be considered critical.

When there are two mating parts, one with threaded or tapped holes and one with clearance holes, the size of the clearance holes is determined by the following formula: positional tolerance of tapped holes plus positional tolerance of clearance holes plus maximum bolt diameter equals the minimum clearance hole diameter.

Assuming that the tapped holes and clearance holes both have a positional tolerance of .028 diameter and the bolt is 1/4 inch diameter, the minimum size of the clearance holes is .306 diameter (.028 + .028 + .250 = .306).

Figure 14 shows the true position location and tolerance for a tapped hole and a clearance hole in two mating plates. By computing the figures given the formula stated above can be confirmed.

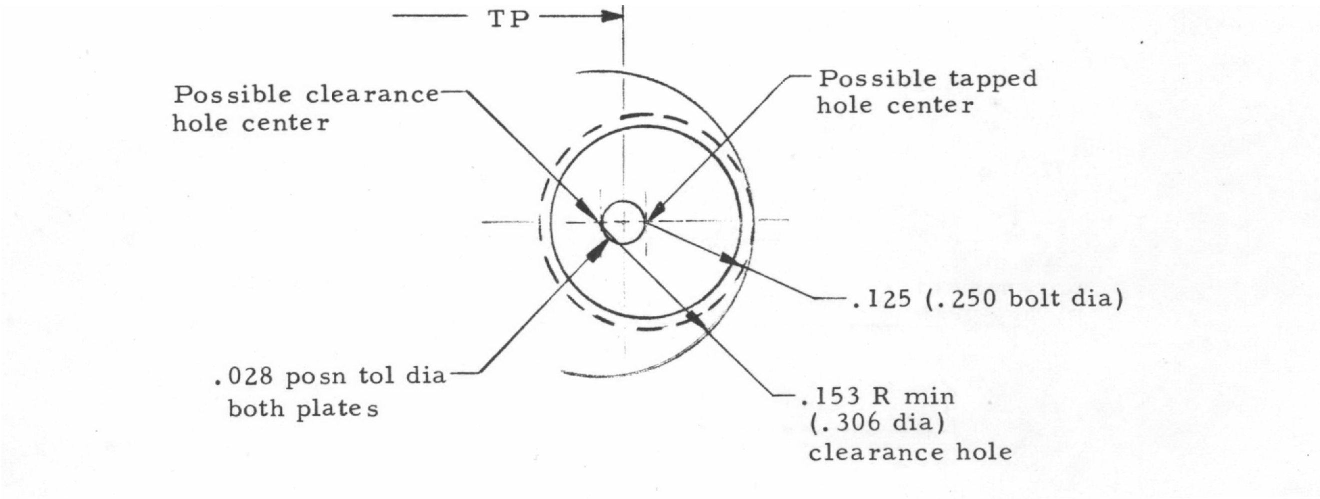


Figure 14

It is understood that a bolt must pass completely through a clearance hole for its entire depth and thread into a tapped hole. The location of the tapped hole below the mating surface is of no importance. The length of the clearance hole in the mating part is recognized as the critical element. Therefore we specify a projected tolerance zone which is equivalent to the length of the clearance hole or maximum thickness of the mating part as shown in Figure 15.

This positional tolerance is a cylindrical zone of .028 diameter, perpendicular to the datum surface. This zone originates and projects .500 from the datum surface.

If the principle of projected tolerance were not applied, it can be seen by Figure 16a that interference would occur if the part features went to their worst condition. This application of the formula and principle stated above will assure 100 percent interchangeability as shown in Figure 16b.

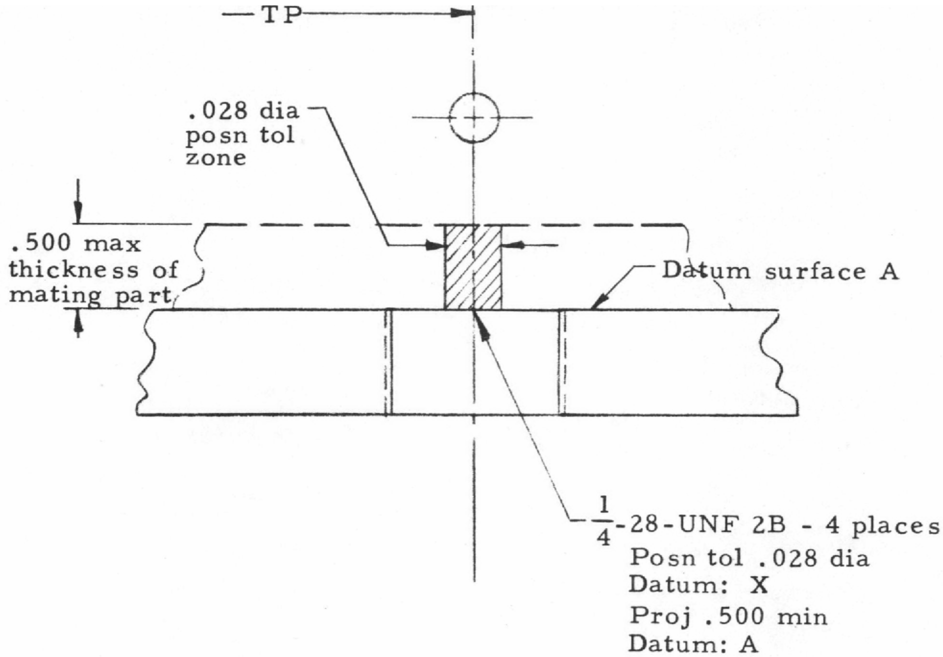


Figure 15

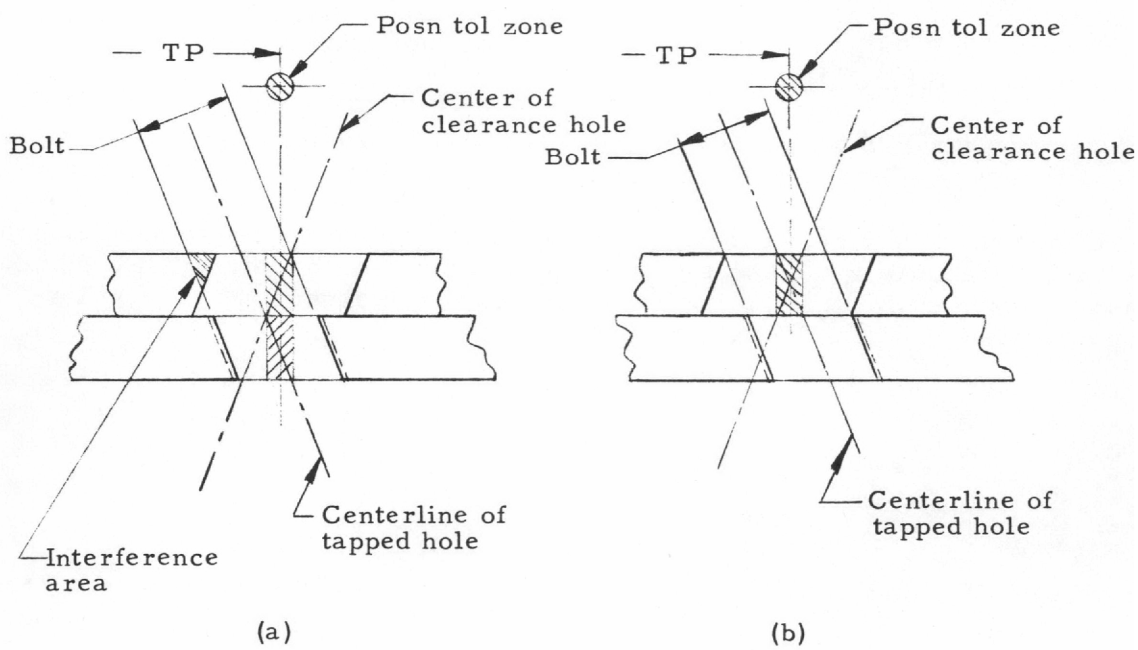


Figure 16

APPENDIX

This appendix presents the methods used by Sandia Corporation in making open-setup inspections of parts dimensioned with positional tolerances. In the absence of functional-type gages these methods have proved valuable in reducing manufacturing and inspection time and expense.

COMPARISON CHARTS

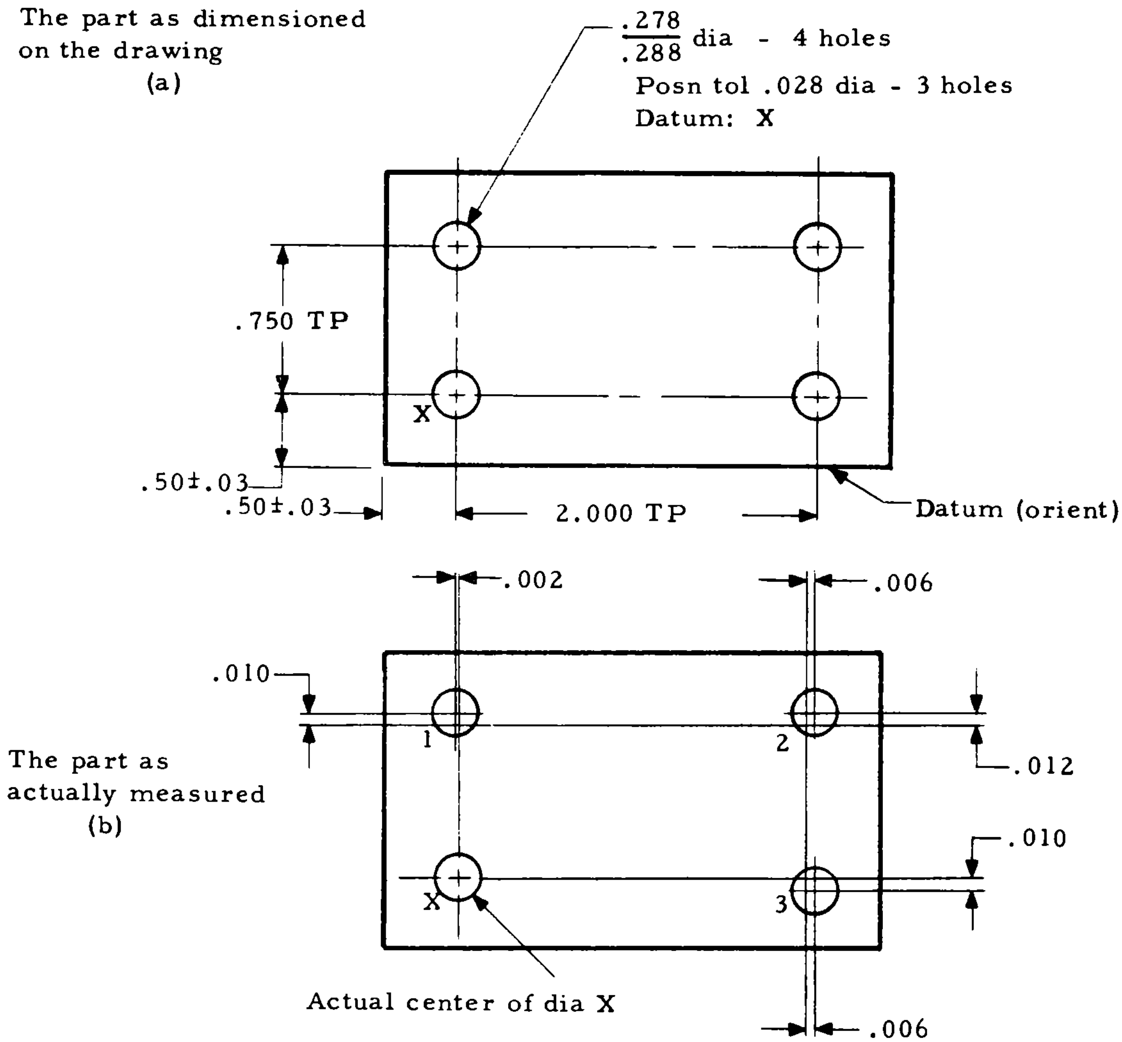


Figure A-1

Figure A-1 shows the part as dimensioned and the part as produced, with the actual hole centers plotted as directions along X and Y axes from true position. On the Comparison Chart (Figure A-2) find the intersection "Z" of the respective X and Y coordinates for holes 1, 2, and 3. This reading is the actual location expressed as a diameter (positional tolerance diameter). The actual radial deviation from true position is Z divided by 2.

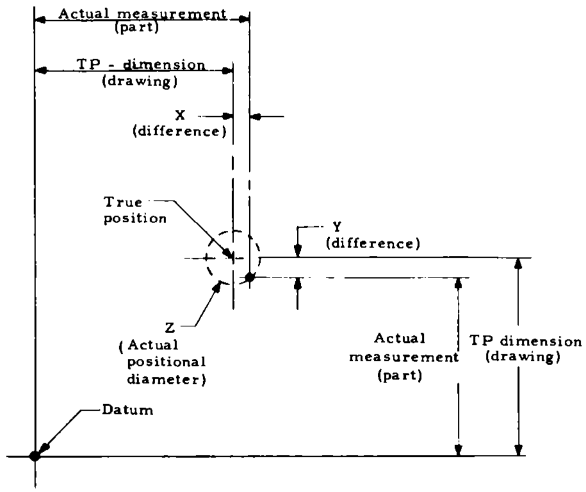
The Z dimension on the chart for holes 1, 2, and 3 is .0204, .0268 and .0233, respectively. Since these Z diameters are less than the positional tolerance of .028 diameter stated on the drawing, the holes are centered within tolerance.

The Tolerance Comparison Chart (Figure A-3) is provided for manufacturing purposes to permit conversion of a given positional tolerance diameter to a permissible bilateral tolerance.

COMPARISON CHART

CO-ORDINATE TO POSITIONAL TOLERANCE DIAMETER

X																				
.020	.0400	.0402	.0404	.0408	.0412	.0418	.0424	.0431	.0439	.0447	.0456	.0466	.0477	.0488	.0500	.0512	.0525	.0538	.0552	.0566
.019	.0380	.0382	.0385	.0388	.0393	.0398	.0405	.0412	.0420	.0429	.0439	.0449	.0460	.0472	.0484	.0497	.0510	.0523	.0537	.0552
.018	.0360	.0362	.0365	.0369	.0374	.0379	.0386	.0394	Z	.0412	.0422	.0433	.0444	.0456	.0469	.0482	.0495	.0509	.0523	.0538
.017	.0340	.0342	.0345	.0349	.0354	.0360	.0368	.0376		.0394	.0405	.0416	.0428	.0440	.0453	.0467	.0481	.0495	.0510	.0525
.016	.0321	.0322	.0325	.0330	.0335	.0342	.0349	.0358	.0367	.0377	.0388	.0400	.0412	.0425	.0439	.0452	.0467	.0482	.0497	.0512
.015	.0301	.0303	.0306	.0310	.0316	.0323	.0331	.0340	.0350	.0360	.0372	.0384	.0397	.0410	.0424	.0439	.0453	.0469	.0484	.0500
.014	.0281	.0283	.0286	.0291	.0297	.0305	.0313	.0322	.0333	.0344	.0356	.0369	.0382	.0396	.0410	.0425	.0440	.0456	.0472	.0488
.013	.0261	.0263	.0267	.0272	.0278	.0286	.0295	.0305	.0316	.0328	.0340	.0354	.0368	.0382	.0397	.0412	.0428	.0444	.0460	.0477
.012	.0241	.0243	.0247	.0253	.0260	.0268	.0278	.0288	.0300	.0312	.0325	.0339	.0354	.0369	.0384	.0400	.0416	.0433	.0449	.0466
.011	.0221	.0224	.0228	.0234	.0242	.0250	.0261	.0272	.0284	.0297	.0311	.0325	.0340	.0356	.0372	.0388	.0405	.0422	.0439	.0456
.010	.0201	.0204	.0209	.0215	.0224	.0233	.0244	.0256	.0269	.0283	.0297	.0312	.0328	.0344	.0360	.0377	.0394	.0412	.0429	.0447
.009	.0181	.0184	.0190	.0197	.0206	.0216	.0228	.0241	.0254	.0269	.0284	.0300	.0316	.0333	.0350	.0367	.0385	.0402	.0420	.0439
.008	.0161	.0165	.0171	.0179	.0189	.0200	.0213	.0226	.0241	.0256	.0272	.0288	.0305	.0322	.0340	.0358	.0376	.0394	.0412	.0431
.007	.0141	.0146	.0152	.0161	.0172	.0184	.0198	.0213	.0228	.0244	.0261	.0278	.0295	.0313	.0331	.0349	.0368	.0386	.0405	.0424
.006	.0122	.0126	.0134	.0144	.0156	.0170	.0184	.0200	.0216	.0233	.0250	.0268	.0286	.0305	.0323	.0342	.0360	.0379	.0398	.0418
.005	.0102	.0108	.0117	.0128	.0141	.0156	.0172	.0189	.0206	.0224	.0242	.0260	.0278	.0297	.0316	.0335	.0354	.0374	.0393	.0412
.004	.0082	.0089	.0100	.0113	.0128	.0144	.0161	.0179	.0197	.0215	.0234	.0253	.0272	.0291	.0310	.0330	.0349	.0369	.0388	.0408
.003	.0063	.0072	.0085	.0100	.0117	.0134	.0152	.0171	.0190	.0209	.0228	.0247	.0267	.0286	.0306	.0325	.0345	.0365	.0385	.0404
.002	.0045	.0056	.0072	.0089	.0108	.0126	.0146	.0165	.0184	.0204	.0224	.0243	.0263	.0283	.0303	.0322	.0342	.0362	.0382	.0402
.001	.0028	.0045	.0063	.0082	.0102	.0122	.0141	.0161	.0181	.0201	.0221	.0241	.0261	.0281	.0301	.0321	.0340	.0360	.0380	.0400
Y	.001	.002	.003	.004	.005	.006	.007	.008	.009	.010	.011	.012	.013	.014	.015	.016	.017	.018	.019	.020



EQUATION

$$Z = 2\sqrt{X^2 + Y^2}$$

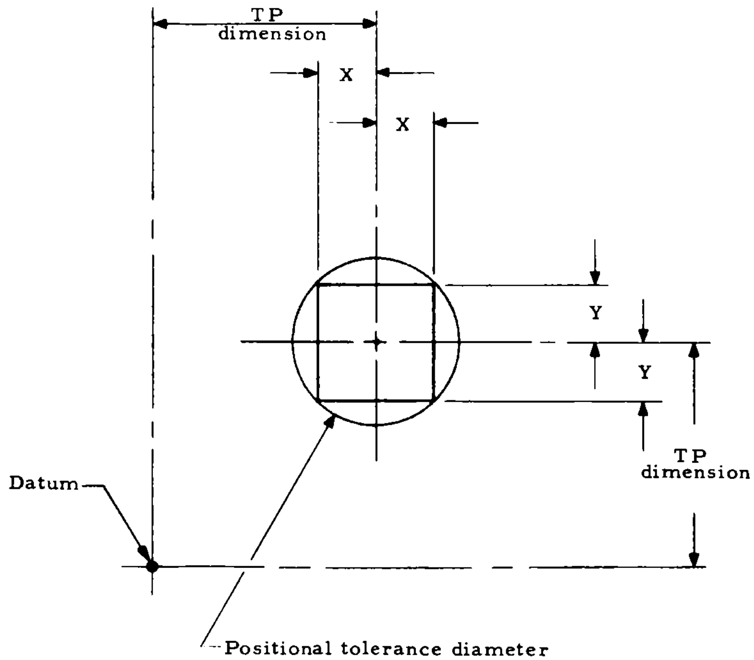
Figure A-2

TOLERANCE COMPARISON CHART

POSITIONAL TOLERANCE DIAMETER TO BILATERAL

Z =	.001	.002	.003	.004	.005	.006	.007	.008	.009	.010	.011	.012	.013	.014	.016	.018	.020	.022	.024
X = Y =	.0003	.0007	.0011	.0014	.0018	.0021	.0025	.0028	.0032	.0035	.0039	.0042	.0046	.0049	.0057	.0064	.0071	.0078	.0085

Z =	.026	.028	.030	.032	.034	.036	.038	.040	.042	.044	.046	.048	.050	.052	.054	.056	.058	.060	.062
X = Y =	.0092	.0099	.0106	.0113	.0120	.0122	.0134	.0141	.0148	.0156	.0163	.0170	.0177	.0184	.0191	.0198	.0205	.0212	.0219



INSTRUCTIONS:

TO OBTAIN THE BILATERAL TOLERANCE, FIND THE POSITIONAL TOLERANCE GIVEN, THEN READ THE EQUIVALENT BILATERAL ($\pm$) DIRECTLY BELOW.

EQUATION

X = Y = .3535Z

Figure A-3

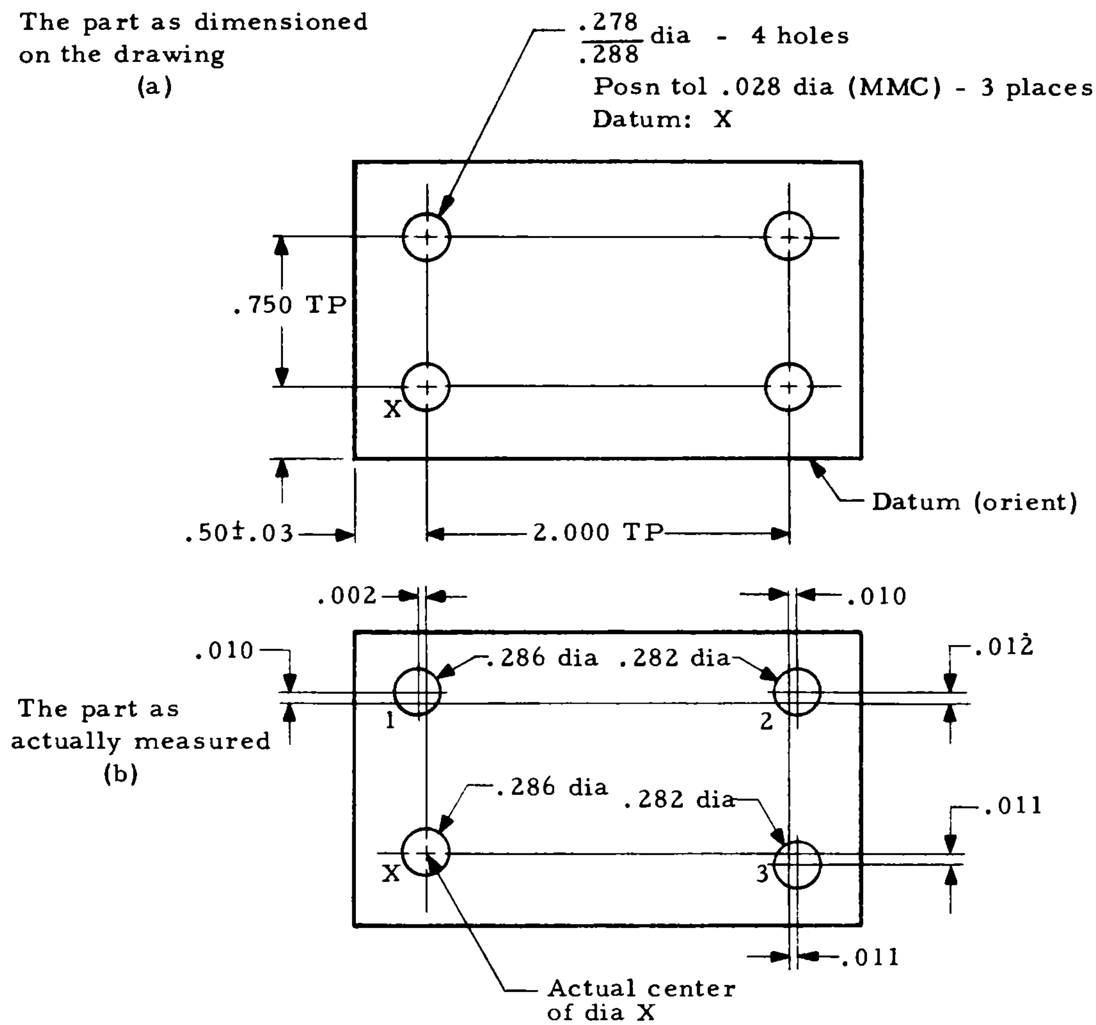


Figure A-4

When MMC appears after tolerance of position as in Figure A-4a, the minimum specified diameter of the hole (.278) can be subtracted from the actual diameter of the hole and the difference then added to the position tolerance diameter. Thus an additional tolerance is allowed when the hole is larger than its minimum specified size.

Using the Comparison Chart as described on the preceding page, hole 1 (Figure A-4b) proves to be centered at .0204 diameter and is within tolerance regardless of size.

Since holes 2 and 3 are .004 over minimum specified size, .004 can be added to .028. Thus the acceptable positional tolerance for these holes is .032 diameter. From the X and Y coordinates, the comparison chart gives Z's of .0312 and .0311 for holes 2 and 3, respectively. The holes are therefore centered within the tolerance as expressed by the MMC callout.

OPEN-SETUP INSPECTION AID

To facilitate open-setup inspection of parts, a graphical method is employed at Sandia Corporation. This method, or aid, uses a Coordinate Measurement Plotting Sheet (Figure A-5 and a Positional Tolerance Selection Sheet (Figure A-6).

The purpose of the aid is to show graphically for either the maximum material condition of a datum feature or for the datum tolerance zone plus maximum material condition whether parts as manufactured are within specified tolerances. With the aid and in the absence of functional gages, open-setup inspection will be able to determine whether parts will meet product drawing requirements of "Datum (MMC)" or "Datum Tolerance plus (MMC)."

The aid may also be used to analyze an outside-of-limits condition of position in order to recommend possible rework of parts or tooling by considering any unused tolerance of size when maximum material condition is specified.

The principle of the aid is to superimpose the positional tolerance system (tolerance zones), a transparent overlay, on the coordinate measurement system (actual measured positions) of a group of related features where "Datum (MMC)" or "Datum Tolerance plus (MMC)" is specified and the line of orientation is known. All positional tolerance zones and coordinate measurements are resolved about a single point regardless of feature pattern configuration.

The aid in employed as follows (see Figure A-5). The intersection of the X and Y axes of Figure A-5 represents the true position of each feature and also the actual center of the datum feature. Use coordinate measurements to plot the actual deviation of each feature (other than datum) from its true position. Care must be taken that the plotted point falls within the proper quadrant.

The center of the concentric circles of Figure A-6 represents both the true position (TP) of all positional tolerance zones and the center of the datum tolerance zone. Measure each feature size to determine if it is within specified limits and calculate the additional amount of positional tolerance allowed by MMC. Select the specified positional tolerance (MMC) zone of each feature (other than datum).

Measure the datum feature size to determine if it is within its specified limits. Select the diameter of its datum tolerance zone allowed by "Datum (MMC)" or "Datum Tolerance plus (MMC)."

Next, superimpose Figure A-6 on Figure A-5. Manipulate Figure A-6 in any manner that will (1) permit the actual center of the datum feature (Figure A-5) to remain within its datum tolerance zone (Figure A-6) and, concurrently, (2) allow plotted feature centers (Figure A-5) to remain within their respective positional tolerance zones (Figure A-6).

When a position of the two sheets is found which will satisfy the above conditions, the related features are considered to be within the functional limits of "Datum (MMC)" or "Datum Tolerance Plus (MMC)."

Examples of the use of the aid are given for both maximum material condition of a datum feature (pages 30 to 34) and datum tolerance plus maximum material condition pages 35 to 36).

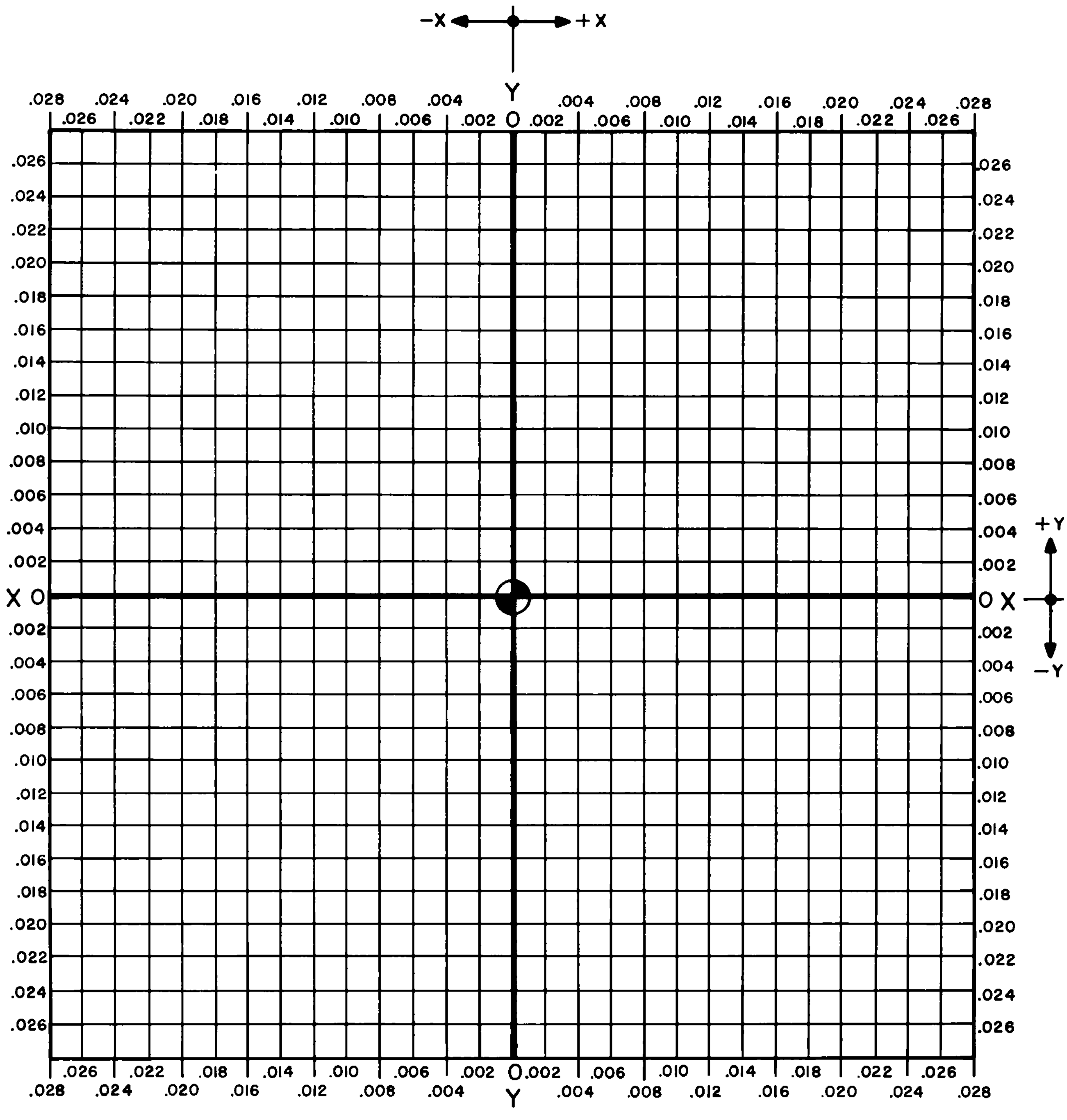

-X
+X
Y
.028
.026
.024
.022
.020
.018
.016
.014
.012
.010
.008
.006
.004
.002
O
X O
O X
+Y
-Y

Figure A-5

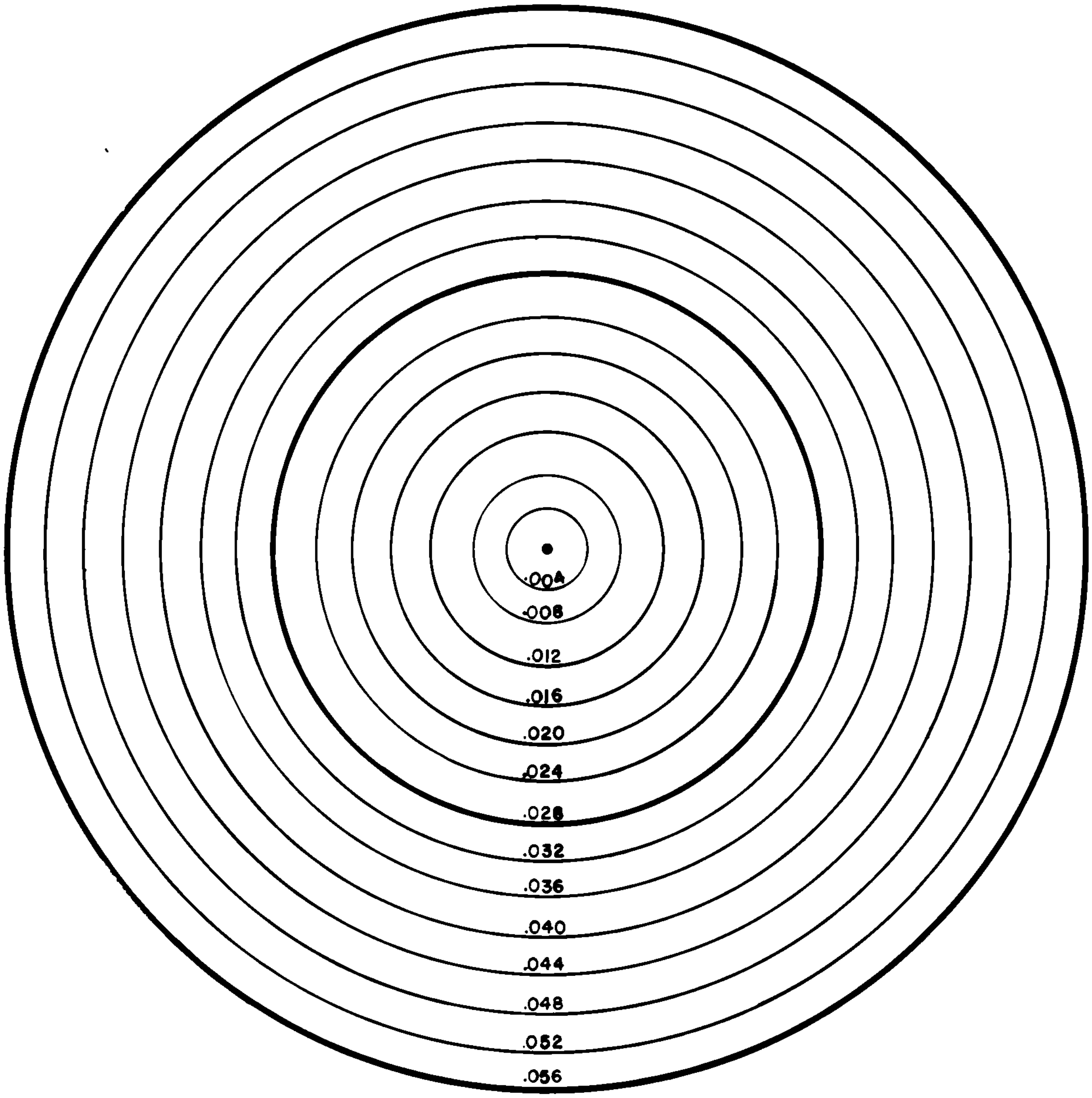


Figure A-6

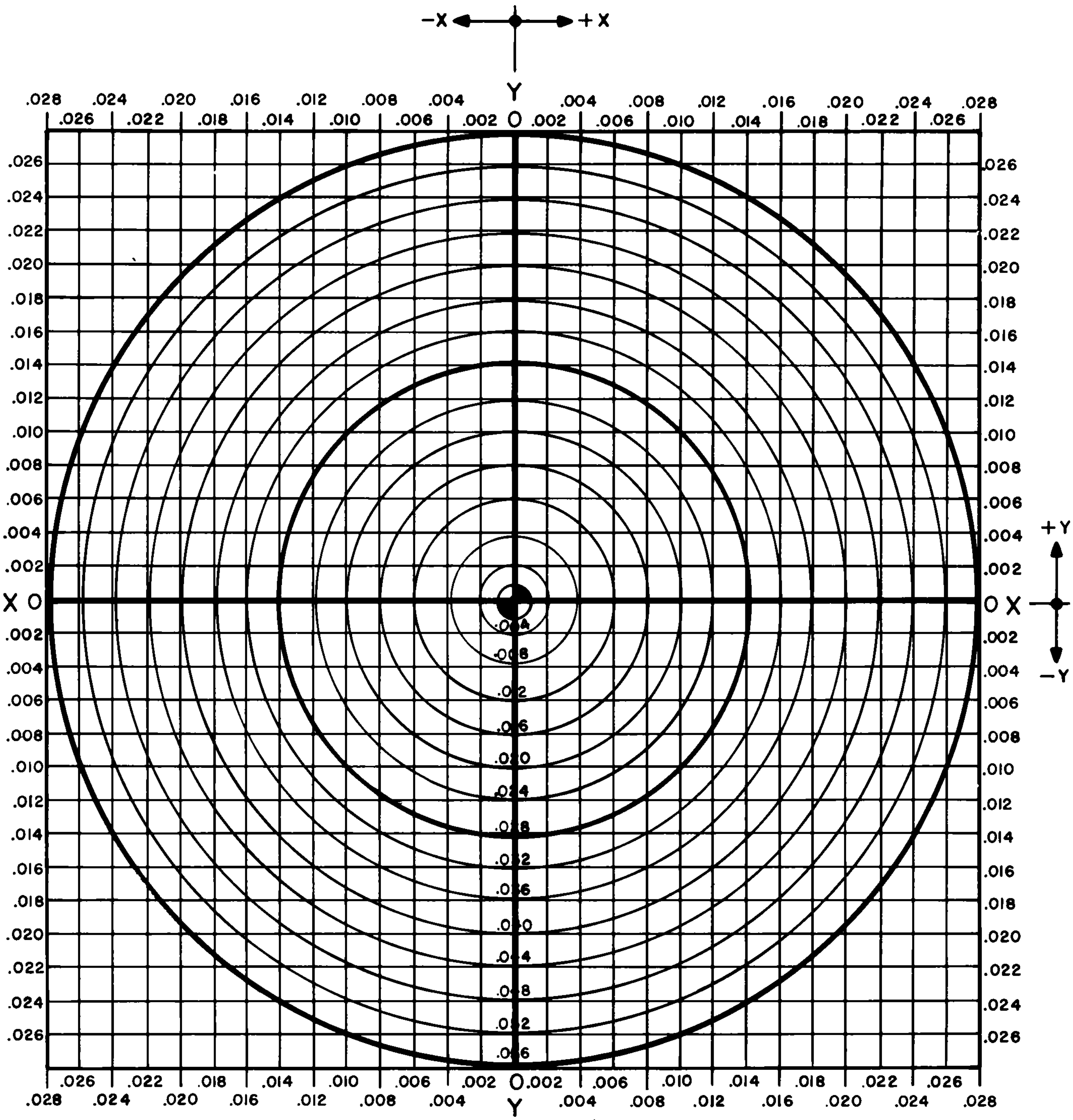

-X
+X
Y
.028 .024 .020 .016 .012 .008 .004 .004 .008 .012 .016 .020 .024 .028
.026 .022 .018 .014 .010 .006 .002 0 .002 .006 .010 .014 .018 .022 .026
X 0
0 X
+Y
-Y
.004 .008 .012 .016 .020 .024 .028 .032 .036 .040 .044 .048 .052 .056

Figure A-6

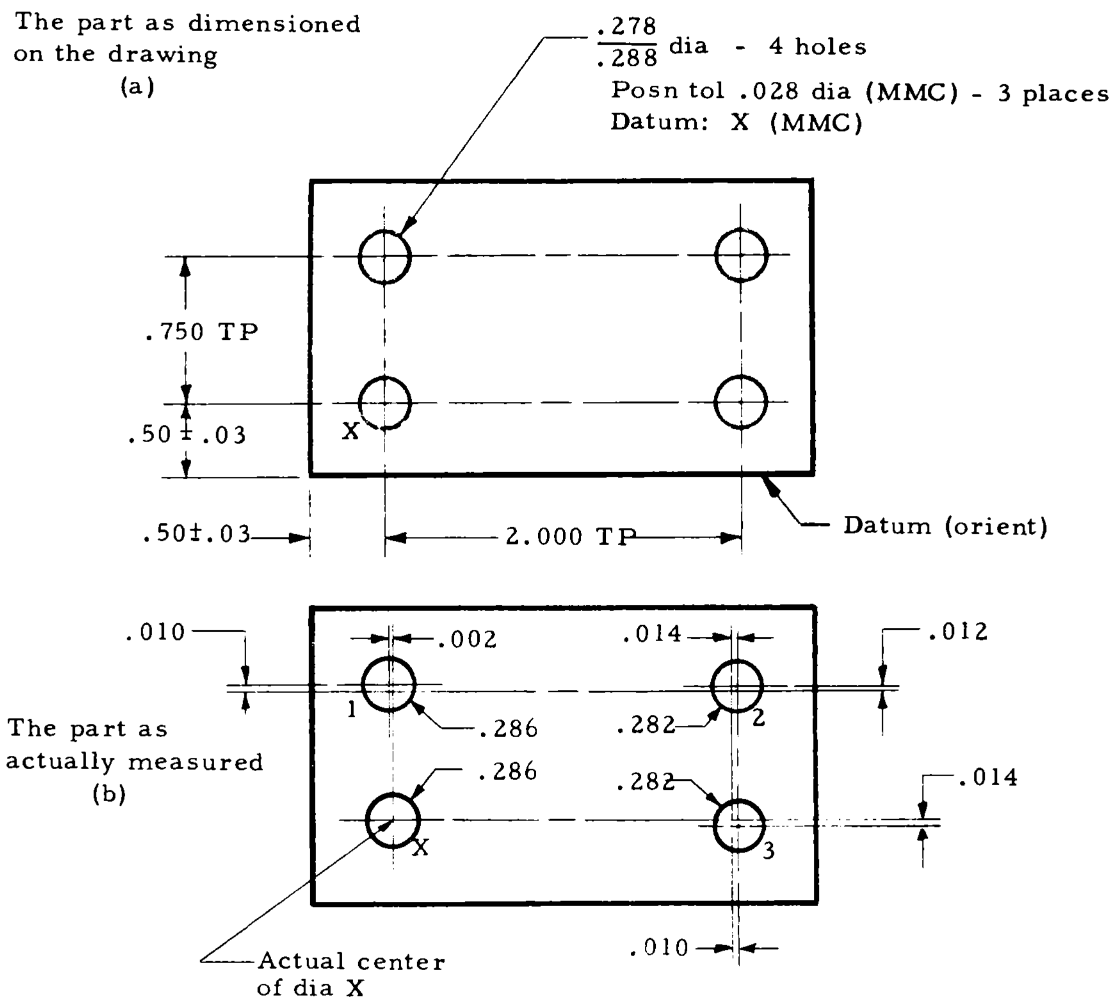


Figure A-7

When (MMC) appears after Datum, as in Figure A-7a, the minimum specified diameter of the datum hole (.278) can be subtracted from the actual diameter of the datum hole. The difference is the amount that the "point of origin" (common for all TP dimensions) may vary from the actual center of the datum hole.

The amount allowable on Datum hole X, as shown in Figure A-7b is .008 diameter (.286 - .278 = .008). Hole 1, .286 diameter with (MMC), is allowed a positional tolerance of .036 diameter (.028 + .008). Holes 2 and 3, .282 diameter with (MMC), are allowed a positional tolerance of .032 diameter (.282 - .278 + .028).

Holes 1, 2, and 3 are plotted on the square graph (Figure A-8), at their respective X and Y axes.

The illustration shows Figure A-6 superimposed and manipulated on Figure A-5 so that holes 2 and 3 are within their .032 diameter tolerance zones. Hole 1 is well within .036 diameter tolerance. The Datum hole X lies within the .008 diameter tolerance allowed by (MMC) relative to the common point of origin.

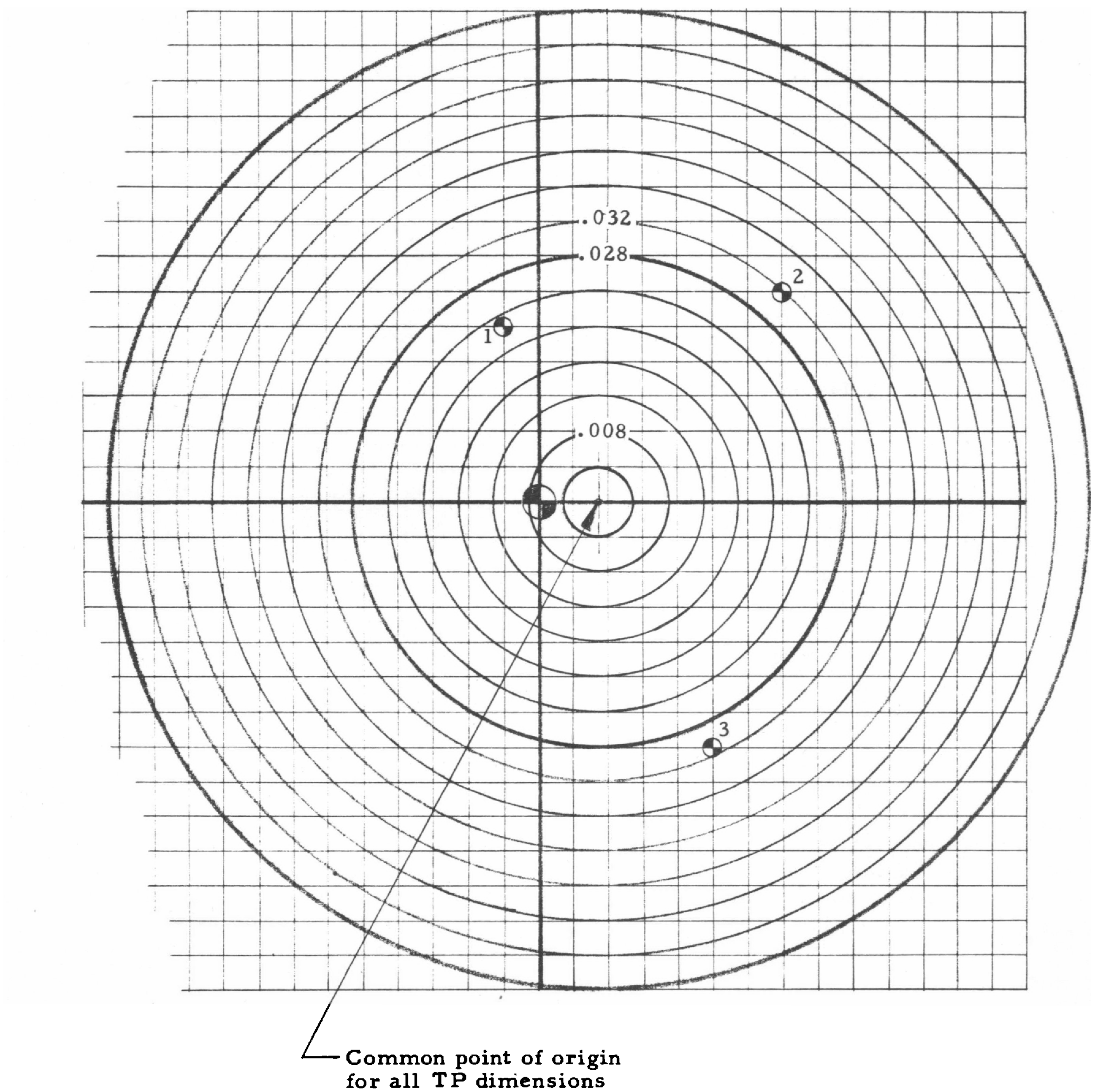


Figure A-8

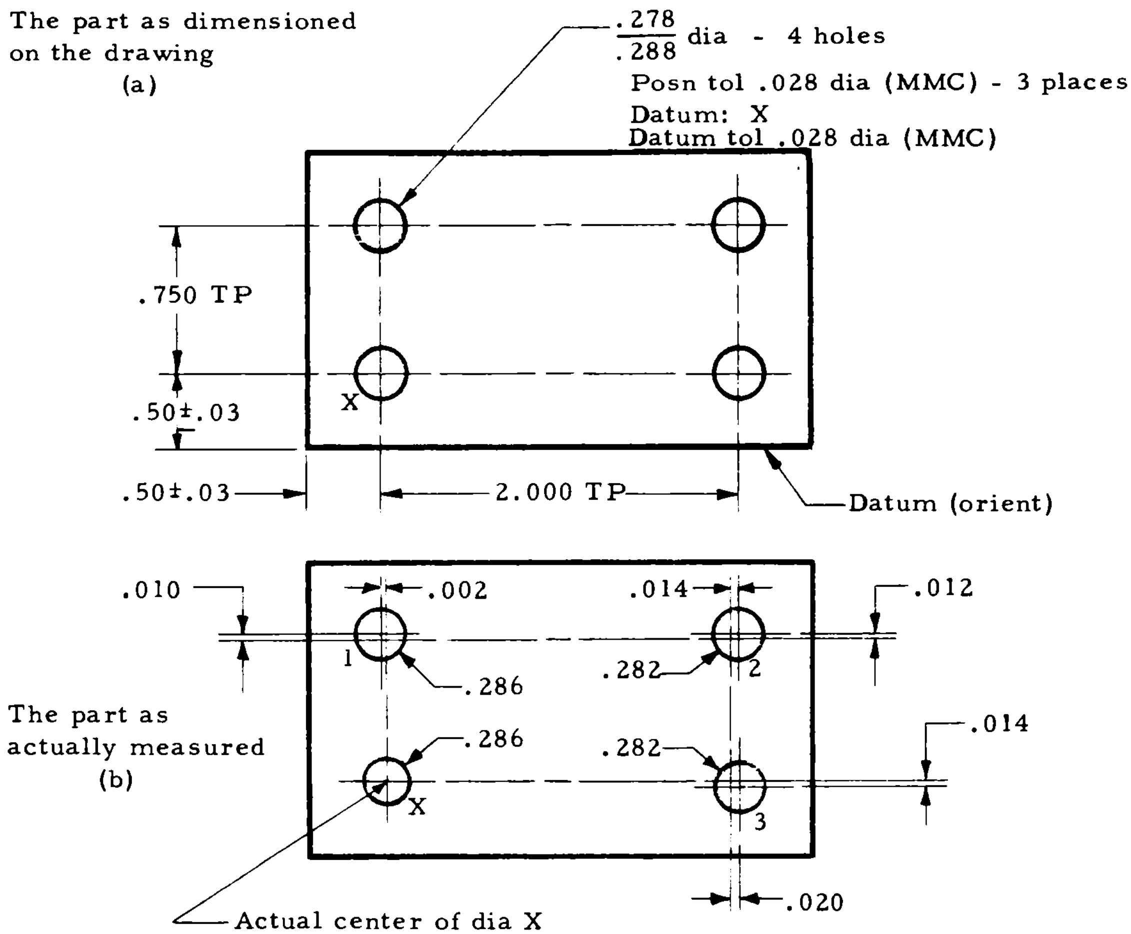


Figure A-9

When "Datum Tolerance plus (MMC)" follows the datum callout, as in Figure A-9a, the datum tolerance specified and any tolerance allowed by MMC to datum can be added. The sum is the amount that the "common point of origin" may vary from the actual center of the datum hole.

The amount allowable on datum hole X as shown in Figure A-9b is .028 datum tolerance plus .008 (.286 - .278). Hole 1 with MMC is therefore allowed a positional tolerance of .036 diameter. Holes 2 and 3 with MMC are allowed a positional tolerance of .032 diameter.

As in the previous example, holes 1, 2, and 3 are plotted on the square graph of Figure A-10 at their respective X and Y axes.

The illustration shows Figure A-6 superimposed and manipulated on Figure A-5 so that holes 2 and 3 are within their .032 diameter tolerance zones. Hole 1 is well within the .036 diameter tolerance.

The datum hole X lies within the .036 diameter tolerance zone allowed by "Datum Tolerance plus (MMC)" relative to the common point of origin.

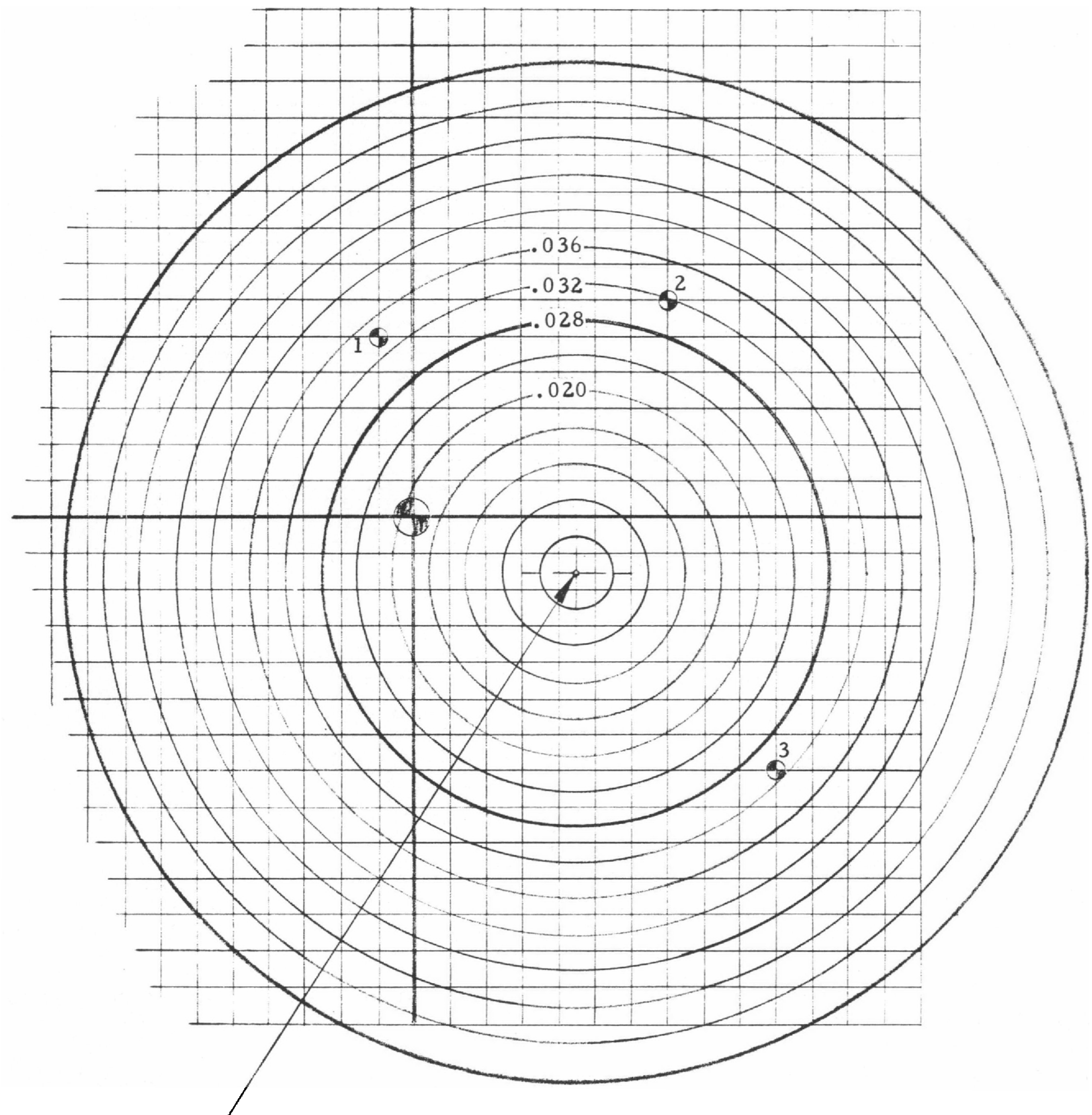

.036
.032
.028
.020
1
2
3
Common point of origin
for all TP dimensions

Figure A-10

ALTERNATE METHOD OF USING THE OPEN-SETUP INSPECTION AID

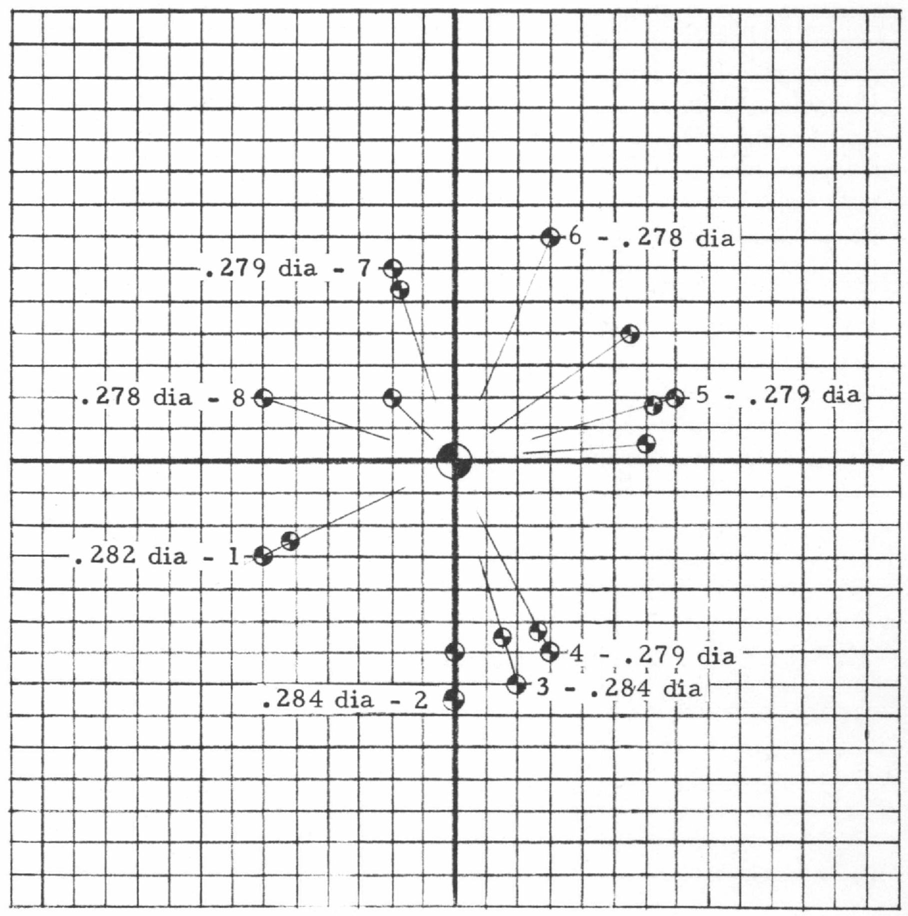


Figure A-11

An alternate method can be used to particular advantage when the number of holes is large and the holes vary in diameter.

In Figure A-11, the holes were plotted from inspection measurement. Below is the applicable callout:

.278/.288 dia. - 12 holes
Posn tol .028 dia. (MMC) - 11 holes
Datum: X (MMC)

A temporary check by superimposing Figure A-6 directly over the center target will indicate that holes 1 through 8 are lying in the fringe area. After checking these actual hole diameters re-plot the holes as follows. Draw a radial line thru the center target and each plotted hole point. By scaling, move the plotted point on the drawn line towards the center by one-half the difference of the actual hole diameter and its minimum specified size. This compensates for (MMC) to tolerance of position.

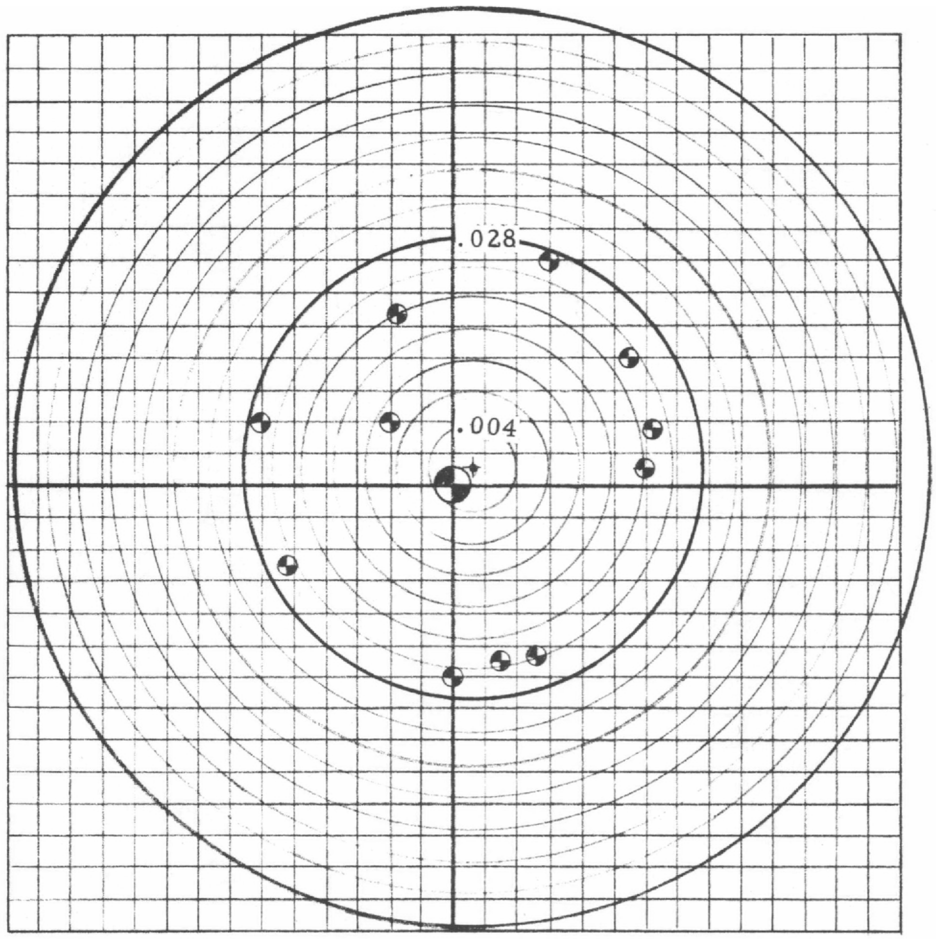


Figure A-12

As shown in Figure A-12, Figure A-6 is superimposed on Figure A-11; the .028-diameter circle can then be used for all holes. The datum, or center target of Figure A-11 is well within the .004 diameter circle of Figure A-6. Therefore, since MMC is specified to datum, the datum hole must be .282 minimum diameter to be within the tatum tolerance allowable.

No. of copies	Distribution
3	Commanding General, Aberdeen Proving Ground, Maryland
1	Nuclear Products, Erco, Division of ACF Industries, Inc., Riverdale, Maryland
1	Aerojet-General Corporation, Azusa, California
1	Aerojet-General Nucleonics, San Ramon, California
1	Commander, Air Force Ballistic Missile Division, USAF, Inglewood, California
1	Air Force Cambridge Research Center, Bedford, Massachusetts
1	Commandant, Air Force Institute of Technology, Wright-Patterson Air Force Base, Ohio
1	AFPR, Boeing Airplane Company, Plant II, Seattle 14, Washington
2	AFPR, Lockheed Aircraft Corporation, Marietta, Georgia
1	AFPR, Rocketdyne Division, North American Aviation, Canoga Park, California
2	Commander, Air Force Special Weapons Center, Kirtland Air Force Base, New Mexico
1	Commander, Air Technical Intelligence Center, Wright-Patterson Air Force Base, Ohio
1	Director, Air University Library, Maxwell Air Force Base, Alabama
1	ANP Project Office, Convair, Fort Worth, Texas
1	Alco Products, Inc., Schenectady, New York
1	American University AT(40-1)-2387, Washington 16, D.C.
1	Argonne National Laboratory, Lemont, Illinois
2	Commander, Field Command, DASA, Sandia Base, Albuquerque, New Mexico
1	Chief, DASA, Washington 25, D.C.
5	Commander, Armed Services Technical Information Agency, Arlington 12, Virginia
1	Army Ballistic Missile Agency, Huntsville, Alabama
1	Commanding General, Army Rocket and Guided Missile Agency, Redstone Arsenal, Alabama
1	Commanding Officer U.S. Army Signal Research and Development Laboratory, Fort Monmouth, New Jersey
3	Atomic Energy Commission, (TL), Washington 25, D.C.
4	Babcock and Wilcox Company, Lynchburg, Virginia
2	Battelle Memorial Institute, Columbus 1, Ohio
1	Bettis Plant, U. S. Atomic Energy Commission, Pittsburg 30, Pennsylvania
1	Brookhaven National Laboratory, Upton, Long Island, New York
1	Brush Beryllium Company, Cleveland 3, Ohio
1	Chief, Bureau of Aeronautics, Washington 25, D.C.
1	BAR, Aerojet-General, Azusa, California
1	BAR, Goodyear Aircraft, Akron, Ohio
1	BAR, Grumman Aircraft, Bethpage, Long Island, New York
1	Bureau of Ships (CODE 1500), Washington 25, D.C.
1	Bureau of Yards and Docks, Washington 25, D.C.
1	Carnegie Institute of Technology, Pittsburgh 13, Pennsylvania
1	Columbia University (SOO-187), New York 27, New York
1	Combustion Engineering, Inc., Nuclear Division, Windsor, Connecticut
1	Convair, General Dynamics Corporation, San Diego 12, California
1	Curtiss-Wright Corporation, Research Division, Quehanna, Pennsylvania
1	Defence Research Member, Washington 8, D.C.
2	Office of Assistant Chief of Staff, G-2, Department of the Army, Washington 25, D.C.
1	du Pont Company, Aiken, South Carolina
1	du Pont Company, Wilmington 98, Delaware
1	Edgerton, Germeshausen and Grier, Inc., Boston 15, Massachusetts
1	Edgerton, Germeshausen and Grier, Inc., Las Vegas, Nevada
1	Engineer Research and Development Laboratories, Fort Belvoir, Virginia
1	Frankford Arsenal, Philadelphia 37, Pennsylvania
1	General Atomic Division of General Dynamics Corporation, San Diego 12, California
1	General Electric Company (ANPD), Cincinnati 15, Ohio
1	General Electric Company, Richland, Washington
1	General Electric Company (AT(30-3)-502), San Jose, California
1	General Nuclear Engineering Corporation, Dunedin, Florida
2	Goodyear Atomic Corporation, Portsmouth, Ohio
1	Iowa State College, Ames, Iowa
1	Commanding Officer, Jet Propulsion Laboratory, Pasadena 2, California
1	Knolls Atomic Power Laboratory, Schenectady, New York

No. of copies	Distribution
2	Los Alamos Scientific Laboratory, Los Alamos, New Mexico
1	Commander, Lowry Air Force Base, Colorado
1	The Maritime Administration, Washington 25, D.C.
1	Martin Company, Baltimore 20, Maryland
1	Massachusetts Institute of Technology (HAYDEN), Cambridge 39, Massachusetts
1	Massachusetts Institute of Technology (GAUDIN), Cambridge 39, Massachusetts
1	Massachusetts Institute of Technology (HARDY), Cambridge 39, Massachusetts
1	Midwestern Universities Research Association, Madison, Wisconsin
1	Mound Laboratory, Miamisburg, Ohio
1	National Academy of Sciences, Washington 25, D.C.
1	National Advisory Committee for Aeronautics, Cleveland 11, Ohio
1	National Bureau of Standards, Atomic Energy Project, Washington 25, D.C.
1	National Bureau of Standards (LIBRARY), Washington 25, D.C.
1	Naval Civil Engineering Research and Evaluation Laboratory, Port Hueneme, California
3	The Director, Naval Research Laboratory, Washington 25, D.C.
1	Nuclear Development Corporation of America, White Plains, New York
1	Oak Ridge Institute of Nuclear Studies, Oak Ridge, Tennessee
15	Commanding Officer, Office of Naval Research, F.P.O., New York, New York
1	Department of Navy (CODE 422), Washington 25, D.C.
3	Commanding Officer, Office of Ordnance Research, Durham, North Carolina
1	Office of Quartermaster General, Department of the Army, Washington 25, D.C.
75	Office of Technical Services, Department of Commerce, Washington 25, D.C.
1	Office of the Chief of Naval Operations, Department of the Navy, Washington 25, D.C.
1	Commanding Officer, Ordnance Materials Research Office, Watertown 72, Massachusetts
1	Commanding General, Ordnance Tank-Automotive Command, Detroit 9, Michigan
1	Commanding Officer, Picatinny Arsenal, Dover, New Jersey
1	Power Reactor Development Company, Detroit 26, Michigan
3	Pratt and Whitney Aircraft Division, Middletown, Connecticut
1	Princeton University (SPITZER), Princeton, New Jersey
1	Rensselaer Polytechnic Institute, Troy, New York
1	U.S. Atomic Energy Commission, Schenectady Naval Reactors Operations Office, Schenectady, New York
1	South Dakota School of Mines and Technology, Rapid City, South Dakota
1	Stevens Institute of Technology, Hoboken, New Jersey
1	Sylvania Electric Products, Inc., Bayside, New York
325	U.S. Atomic Energy Commission, Technical Information Service Extension, Oak Ridge, Tennessee
1	Technical Research Group, New York 3, New York
1	Tennessee Valley Authority, Chattanooga, Tennessee
1	Texas Nuclear Corporation, Austin 17, Texas
1	Commander, U.S. Naval Ordnance Laboratory, Silver Spring, Maryland
1	U.S. Naval Postgraduate School, Monterey, California
1	University of California Radiation Laboratory, Berkeley, California
4	University of California Radiation Laboratory, Livermore, California
1	University of Chicago, USAF Radiation Laboratory, Chicago 37, Illinois
1	University of Michigan, Ann Arbor, Michigan
1	University of Puerto Rico, Mayaguez, Puerto Rico
1	University of Rochester, Atomic Energy Project, Rochester 20, New York
2	University of Washington (GEBALLE), Seattle 5, Washington
1	Vitro Engineering Division, New York 3, New York
1	Commanding Officer, Watertown Arsenal, Watertown 72, Massachusetts
1	Westinghouse Electric Corporation (BIONDI), Pittsburgh 25, Pennsylvania
1	Westinghouse Electric Corporation (SCHAFER), Pittsburgh 30, Pennsylvania
6	Commander, Wright Air Development Center, Wright-Patterson Air Force Base, Ohio
1	Yale University (BREIT), New Haven 11, Connecticut
1	Yale University (SCHULTZ), New Haven 11, Connecticut