TECHNICAL NOTES

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No. 296

BEARING STRENGTH OF WOOD UNDER STEEL AIRCRAFT BOLTS AND WASHERS AND OTHER FACTORS INFLUENCING FITTING DESIGN

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Preface

During the past four years, the Bureau of Aeronautics, Navy Department, has financed four investigations relative to the bearing strength of wood under steel aircraft bolts, each covering one particular phase of the problem. As each investigation was completed a report was prepared, with the result that there are four reports (Reference 1) on the subject. Furthermore, it so happened that the first two reports came out at a time when the Army Air Service and the Bureau of Aeronautics, Navy Department, had not yet come to a common standard of moisture content and duration of stress. Consequently, the charts in these reports are not based on the values now in use.

The purpose of this report is to correlate, bring up to date, and put under a single cover all the information thought to be essential to an understanding of the formulas. Greater detail, however, will be found in the former reports (loc. cit.).

Description of Test Material Used

The bolts used for the tests were steel aircraft bolts furnished by the Bureau of Aeronautics, Navy Department. The diameters and lengths of the solid bolts were as follows.

Diameter of Bolt	Length of Bolt
Inches	Inches
No. 8 (.16)	1, 2, $2\frac{1}{2}$
No. 10 (.19)	1, $1\frac{1}{2}$, 2, $2\frac{1}{2}$, 3
1/4	1, $1\frac{1}{2}$, 2, $2\frac{1}{2}$, 3, 4
5/16	$1\frac{1}{2}$, 2, $2\frac{1}{2}$, 3, 4, 5
3/8	$1\frac{1}{2}$, 2, $2\frac{1}{2}$, 3, 4, 5, 6
1/2	2, 3, 4, 6

The dimensions of the hollow bolts are shown in Figure 1.

The wood used for practically all of the tests was Sitka spruce selected in the state of Oregon. Part of it was shipped to the laboratory in plank form, but the most of it was shipped in log form, where it was cut into lumber, marked, and seasoned. In order to demonstrate that the results obtained with spruce were applicable to other species by adjusting for the difference in physical properties, some commercial white ash was used.

For the principal variables, such as bearing length, diameter, and type of bolt, the dimension of the test specimens perpendicular to the direction of the bolt holes was from 1 to

3 inches, according to the size of bolt tested. The length of the test specimens was such as to permit several tests on the same piece, the spacing of test holes being governed by the size of bolt tested. Test holes were drilled near one end. At the other end a hole was drilled at right angles to the axis of the test holes to take a pin of large diameter which supported the specimens during test.

For the secondary tests, such as washer tests, laminated block tests, etc., the sizes of the test specimens were such as to best suit the conditions of the particular test.

Method of Test

The axes of all bolts were perpendicular to the grain of the wood. Load was applied at a rate of from .03 to .06 inch per minute.

First Series - Pull Parallel to the Grain of the Wood

A large pin passed through the specimen at one end and rested at each end on a frame attached to the base of a testing machine. Suspended from this pin the specimen hung down over the center of the movable head of the testing machine. The loading straps were pin connected to the movable head and the bolts to be tested passed through the specimen and the loading straps. For eccentric (one-end loading) one strap was used and load was applied at one end of the bolt. Symmetrical loading was of two types: first, in which two straps were used and load was applied at each end of the bolt; and second, in which one strap inserted in a slot in the test specimen applied a load at the mid-section of the bolt. Load was applied by lowering the movable head of the machine. In the earlier tests one Ames dial was connected between one of the loading straps and the specimen. Later, two dials were used, making it possible to measure the movement of each end of the bolt. New bolts were used for each test.

Second Series - Pull Perpendicular to the Grain of the Wood

For this test, the specimen rested horizontally on a frame attached to the base of a testing machine. It was supported over a short span so that very little bending was introduced. The test procedure was the same as for pull parallel to the grain.

Third Series - Pull at an Acute Angle to the Axis of the Bolt and the Grain of the Wood and in a Plane Determined by the Axis of the Bolt and the Grain of the Wood

In this test the test specimen was supported at an angle in a frame attached to the base of a testing machine. A strap, pin connected to the movable head of the machine, was attached to a fitting, which in turn was attached to the specimen by two bolts, spaced 5-inch centers. Deflections parallel to the length of the test specimen and parallel to the bolt were read simultaneously for the same load increments.

Fourth Series - All Directions of Pull Not Covered by the First Three Series

This included both two-end and one-end loadings. In the two-end loading test the specimen was supported at an angle in a frame attached to the base of the testing machine. It was so placed that the bolt hole came directly above the center of the movable head of the machine. Two Ames dials were fastened to the upper surface of the specimen, one directly above each loading strap. A fine wire connected each dial with a pin in the adjacent loading strap. For one-end loading, one strap and one dial were used. The same bolt was never used for more than one test.

Secondary Tests

For the washer tests bearing perpendicular to the grain, a specimen was laid horizontally on a frame attached to the base of a testing machine. A hole the same size as the hole in the washer was drilled through the specimen and placed directly over the center of the movable head. A bolt connected to the movable head passed vertically through this hole and to its upper end a washer and nut were attached. Load was applied by lowering the head. An Ames dial, attached to the specimen and operated directly by the bolt, registered the movement of the washer in the wood.

For the tests to determine how bolts of different diameters acted together, how the bolt hole should be drilled, how pieces of hard dense wood glued to the surface of a member affected the loads, etc., the procedure was the same as that described above for pull parallel to the grain.

Discussion

The data on the bearing strength of wood under steel aircraft bolts are based on what may be termed bearing strength at the apparent elastic limit. This elastic limit was taken as that load at which the deflections ceased to be directly proportional to the loads. These values are probably not true elastic limit values, since the bolt did not return to the exact initial position when the load was reduced, due to a slight friction of the washers and to the embedding of the bolt into the frayed wood fibers. However, in design they have the same significance as elastic limit values in other strength properties. How the allowable design stresses were calculated from the elastic limit stresses will be discussed later.

For convenience, the stress in bearing parallel to the grain is discussed first; next, the bearing stress for loads perpendicular to the grain; and finally, the bearing stress for loads at any angle to the grain. The reason for this will become apparent as the discussion is followed. The study was primarily of solid bolts, but reference is made from time to time to the results of tests on hollow bolts.

Bearing Strength - Pull Parallel to Grain

The data assembled under this heading represent two loading conditions; symmetrical loading, which includes two-end and mid-section loading of the bolts, and nonsymmetrical loading, which is one-end loading. Load in both cases was applied perpendicular to the bolt and in a direction parallel to the general direction of the grain. It was found after running a quite complete series of tests on Sitka spruce and commercial white ash that the loads at elastic limit for one-end loading were just one-half of those for two-end loading for all lengths and diameters.

After considerable data had been assembled for bolts of various diameters and lengths, it became apparent that the results for all could be combined if bearing length were expressed in terms of diameter of bolt (L/D) rather than in inches of length only.

In Figure 2, bearing stress at elastic limit for solid bolts is plotted against L/D. The results are from data indicated by circles, on Sitka spruce and commercial white ash adjusted to a common basis. One-end and two-end loading values are both included. The one-end loading values were simply doubled and adjusted for differences in the quality of the test pieces. The values for each test piece were corrected in accordance with their maximum crushing strength as determined by minor tests.

The general trend of the points indicates a relation between bearing stress at elastic limit and L/D. The heavy line marked stress curve No. 1 approximating this general trend divides the points in half and is midway between the dotted line 15 per cent above it and the dotted line 15 per cent below it, which cut off the upper and lower 25 per cent of the points, respectively.

As this curve was sketched in, an efficiency curve (No. 1), based on length of bolt being constant, was constructed simultaneously. It is evident that an efficiency curve must be a smooth curve. Consequently, the stress curve was altered until the efficiency curve fulfilled that condition. Though, with the bolts furnished, no tests for very large L/D ratios could be made, it was assumed that the load at elastic limit for a given diameter of bolt remained constant after a certain L/D ratio was passed. This L/D ratio was taken as 13 and the efficiency curve based on weight of stem only would become horizontal at an L/D of 13 and remain horizontal for greater ratios. After repeated trials to obtain a smooth efficiency curve, a stress curve was obtained which fit the data very well. It will be noted that from 0 to 1 L/D the stress was assumed constant. It was impossible to verify this by test, but the assumption seems logical. The efficiency curve for this range would be a straight line. Between the ratios 1 and 13 it became a parabolic curve of the power shown on the curve, tangent to the straight line at an L/D of 1 and to the horizontal line at an L/D of 13. Efficiency was taken as the load per unit weight of stem or $E = C \dfrac{\text{Load}}{\text{Volume of stem}}$. In order to express efficiency in terms of L/D, the expression $E = \dfrac{C \text{ load}}{\text{Volume of stem}}$ may be reduced to $E = \dfrac{C_1 \text{ load}}{LD^2}$ and since LD = projected area, $\text{load} / LD = \text{stress}$ and $E = \dfrac{C_1 \text{ stress}}{D}$. Assuming as we did above that L is constant and writing $LE = C$ stress L/D, efficiency may be expressed as $E = C_2$ stress L/D.

Incidentally, it may be noted that if the length of bolt, that is, thickness of member, were not a constant, but could be varied at will, the stress curve would become the efficiency curve.

In the early tests the nuts were drawn quite snug and the loads were accordingly raised. On a percentage basis this friction element became less important as the length of bolt was increased. Because of this friction, the stress curve meets the stress axis at 92.6 per cent of the maximum crushing strength, whereas we know that for Sitka spruce the elastic limit stress in compression parallel to the grain is only 80 per cent of the maximum crushing strength. Correcting for friction, we get the lower stress curve No. 2 and a corresponding efficiency curve No. 2. Later it will be found convenient to put this curve on a percentage basis so that the bearing stress for any species can be calculated therefrom. (See Figure 7, curve A.)

Another set of efficiency curves are shown on Figure 2, which have been calculated on a different basis. These are based on stress curve No. 2 and the weights of head, stem, and nut have been included in the weight of the bolt. On the assumption made above regarding a constant load beyond an L/D of 13 for a given diameter, these curves do not reach a maximum. Since the weight of bolt is only one element which must be considered in the efficiency of a fitting, little can be said regarding these efficiency curves except that as far as the bolts are concerned, efficiency increases as diameter of bolt decreases when length is a constant.

In general, the same type of stress curves are obtained for hollow bolts. Due to a flattening of such bolts under load, the stress curves begin with a less stress at 0 L/D than solid bolts and then dip somewhat more rapidly for increasing values of L/D. The stress at 0 L/D also decreases as the ratio of shell thickness to radius decreases. Consequently, with different shell thicknesses, we have a family of stress curves. Only three diameters of hollow bolts were available for these tests, namely, 5/8 inch, 3/4 inch, and 1 inch. All had a shell thickness of 1/8 inch. Guided by the information obtained from the tests of solid bolts, it was possible to establish stress curves with a relatively few tests. A family of such curves are shown in Figure 3. With the bolts available, it was possible to determine only three of these hollow bolt stress curves; the 1 : $2\frac{1}{2}$, 1 : 3, and 1 : 4, corresponding to the 5/8-inch, 3/4-inch, and 1-inch bolts, respectively. The other curves were interpolated. As for the solid bolts, stress curves and efficiency curves were drawn simultaneously and trials made until the stress curve fit the data fairly well and gave a smooth efficiency curve. The efficiency curves are also shown in the figure. The dimensions of the bolts were taken from Aircraft Specifications 95 A of March, 1919.

As a usual thing, hollow bolts are used in much larger sizes than solid bolts and in many cases they are likely to be less efficient than the solid. It would be well, therefore, to check the efficiency based on total weight of bolt before completing the design of the fitting. In some cases, however, a sacrifice of efficiency as regards weight of bolt may result in a net saving when the weight of fitting is taken into account. In the following cases it will, in general, be found that the hollow bolts are more efficient than the solid bolts:

1. When the diameters are equal.
2. When the weight of the hollow bolt per unit of length is equal to or less than that of the solid bolt.
3. When the number of bolts used are equal.

One Component of Lead Parallel to Grain and the Other Taken by the Washer or Fitting

Drift stresses carried through a drift wire to the bolt which holds the wire to the spar apply to that bolt a load which has one component parallel to the grain and another which is taken by the washer on the back of the spar. When ample bearing at the head of the bolt is provided by the fitting or washer, the stress in the wood due to the component parallel to the grain may be made equal to the permissible stress parallel to the grain for a one-end loading. As pointed out above, the permissible stress for a one-end load is half that for a symmetrical load. So far we have dealt with elastic limit stresses only, but the same applies to design stresses, which will be discussed later.

Bearing Strength Perpendicular to the Grain

It has been known for some time that the elastic limit stress in bearing perpendicular to the grain was influenced by the size of bolt or plate bearing on a wooden member, due to the fact that as load is applied, the fibers, which are bent down along the edges of the plate, are thrown into bending and help to resist the load. Lowest values will be obtained, therefore, when the entire test specimen is covered, and as the size of the bolt or plate is decreased, stress values will increase. Tests were made and the relation shown in Figure 7, curve D, was found to exist for spruce. Fiber stress is plotted against width of plate with stress for a 2-inch plate as 100 per cent. This was done because all standard tests are made with a 2-inch plate and strength values now in use are based on these tests. While this curve was made from tests on spruce, it may be generally applied to other species.

It is apparent that we will get a different curve representing the relation between bearing stress perpendicular to the grain and L/D ratio for each size of bolt. The general relation, however, may be plotted on a percentage basis and is so shown in Figure 7, curve C. Elastic limit bearing stresses perpendicular to the grain can be determined from Figure 7 (curves D and C) by first multiplying the elastic limit stress perpendicular to the grain recommended for the species by the percentage for bolt size and the product by the percentage for L/D ratio.

While no tests were made on hollow bolts bearing perpendicular to the grain, it is thought that the relation shown in Figure 7, curve C, may be applied to them without appreciable error. The loads at elastic limit perpendicular to the grain are much smaller than those parallel to the grain and would not produce much flattening of the bolts and consequent reduction in load.

Bearing Strength at an Angle to the Grain

We have discussed the case of bearing parallel to the grain or where one component was parallel to the grain and the other was taken by a washer or fitting under the head of the bolt. We have also discussed the condition of bearing perpendicular to the grain. There remains the condition in which the bolt to wood reaction may be reduced to two components, one acting parallel to the fibers and one acting perpendicular to them. A lift stress might throw such a load on a bolt.

A two-end load was applied to bolts, the axes of which were perpendicular to the grain of the wood. Tests were made with the line of action of the force acting at 30, 45, and 60 degrees to the grain. In Figures 4, 5, and 6 are shown the results of tests on 1/4-inch, 3/8-inch, and 1/2-inch solid bolts thus tested. Bearing values parallel (at 0°) and perpendicular (at 90°) to the grain are also included. Elastic limit stresses are plotted against the angle of the line of action with the grain. All tests of a given size of bolt and bearing length were made in the same piece of spruce, but the results for one diameter and bearing length are not directly comparable with those for another size and bearing length, since the separate pieces varied somewhat in quality.

It is apparent that the points in each series bear a relation to each other fixed by some law. A formula for bearing at an angle to the grain advocated by Mr. Hankinson for rigid plates fitted the data better than any other formula proposed when modified to take care of size of bolt. The full line curves of Figures 4, 5, and 6 are the values computed by this formula.

Mr. Hankinson's formula is

$$N = \frac{PQ}{P\sin^2\theta + Q\cos^2\theta}$$

in which N = the unit bearing stress in a direction at inclination θ with the direction of the grain.

P = the unit stress in compression parallel to the grain.

Q = the unit stress in compression perpendicular to the grain.

In using this formula, it would be necessary to modify it or to modify the values of P for L/D and Q for L/D and diameter of bolt. The latter appeared to be the simpler of the two. The modifications of P and Q for L/D are shown in Figure 7, curves A and C, and of Q for diameter of bolt in curve D.

One-End Loading at an Angle to the Grain

Tests were made with a one-end load acting not only at an angle to the grain but at an angle to the bolt as well. It was found that a one-end load at elastic limit acting perpendicular to the bolt was just one-half a two-end load. With a one-end load acting at an angle to the bolt, it was found that the component perpendicular to the bolt could be made equal to the permissible one-end load.

Allowable Design Stresses

1. Bearing parallel to the grain.- The elastic limit stress parallel to the grain is approximately 80 per cent of the maximum crushing strength for conifers (soft woods) and 75 per cent for hard woods. Therefore, for really short bolts (0 to 1 L/D), the allowable design stress is fixed by the ultimate, i.e., a factor of 1.25 for conifers and 1.33 for hard woods can be applied to elastic limit to obtain design values. As the L/D ratio increases, the ratio between fiber stress at elastic limit

and the ultimate decreases until it is soon less than 50 per cent of the ultimate. Here the allowable design load may be fixed by the elastic limit, since the maximum probable load (one-half the design load) should not exceed the elastic limit. Because of the splitting which takes place near the elastic limit for large L/D ratios, it is recommended that maximum probable load should be fixed at 85 per cent of the elastic limit load and consequently, design load will be 1.70 times the elastic limit load. For L/D ratios above 12 the same factor should be used. For L/D ratios from 1 to 12 the factors should increase lineally from the 1.25 or 1.33 mentioned above to the 1.70 recommended for an L/D of 12. These factors are shown graphically in Figure 7 (B curves).

2. Bearing perpendicular to the grain.- Allowable design stresses for bearing perpendicular to the grain should be one-third greater than elastic limit stresses for all L/D ratios as represented by the elastic limit L/D curve. Of course, modification for size of bolt must also be made.

Since the recommended design stress in bearing perpendicular to the grain given in the table of strength properties is 33-1/3 per cent more than the elastic limit stress (see Table I of strength properties, footnote 4), the design value for a given diameter and L/D can be obtained by multiplying the table value by the diameter factor and the L/D factor.

3. Bearing at an angle to the grain.- When calculating the allowable design stress for bearing at an angle to the grain, it is only necessary to calculate the allowable design stress parallel to the grain (P) for the L/D given, and the allowable design stress perpendicular to the grain (Q) for the diameter of bolt and L/D given and substitute them in formula

$$N = \frac{PQ}{P\sin^2\theta + Q\cos^2\theta}$$

Allowable Distance between Bolts on a Line

It is often convenient or essential to place two or more bolts on a line to resist a load acting in the direction of the grain. When bolts are so placed, it is important that they be sufficiently spaced to prevent shear in the plugs between them. Several questions arise in this connection. First, can the calculation be based on double shear? Second, should center to center of bolts, edge to edge, or some other distance be used in calculating shear area? Third, can the same design stresses in shear be used for all L/D ratios?

Our tests show that double shear may be used provided that a distance intermediate between the center to center and edge to edge distance is used for calculating shear area and that design stress is reduced for L/D ratio. It is apparent that with large L/D ratios the shear failure is a progressive one starting at the edges of the piece. Several series of tests with various L/D ratios indicated that the allowable design shear stress should be the shear strength of the material reduced for L/D ratio in accordance with the A curve of Figure 7.

Cross Bolts

When load parallel to the grain is applied to a bolt, splitting to the end is likely to occur, no matter what the margin or length of bolt, if the capacity load is continued long enough. A little wedge of fibers forms under the bolt and produces a cleavage action. Furthermore, with long bolts as the angle of the line of action with the grain approaches 90 degrees, splitting occurs nearer and nearer to the elastic limit. The most logical and effective way of preventing this splitting is by tying the member together with small cross bolts. This is especially effective when there is a combination of forces on a fitting and not simply a direct tension or compression.

A few tests were made which merely point out the importance of using cross bolts, but which were not extensive enough to furnish rules for the spacing of bolts, size to be used, etc. They do show, however, that it becomes necessary to use cross bolts when the cross-sectional dimension of the member perpendicular to the bolt is less than three times the diameter of the bolt, and in all cases the joints with cross bolts showed greater reliability than those without.

Bolts of Different Diameters in the Same Fitting

When bolts of different diameters are used in the same fitting, it is logical to suppose that the load which they will sustain collectively is not the sum of what they will resist singly. While no extensive series of tests was made, the little data which we have would indicate that where the ratio of diameters is not greater than two to one, the sum of their individual loads may be used without appreciable error.

Allowable Bearing Stress for Washers

The size relation shown in Figure 7, curve D, applies to washers as well as bolts and plates. The diameter of the washer may be taken as the width of plate.

Making the Bolt Hole

It is recommended that the bolt holes be carefully matched with the fitting and of such size that the bolts can be driven by light taps. When the holes were made slightly smaller than the bolts so that considerable force was required to drive them, initial stress was put into the wood and the elastic limit and ultimate were both lowered. It was also found necessary to use freely clearing bits with side cutter lips to insure freedom from initial splits.

Bolts in Laminated Members

Unless a bolt in a wooden member is very short, it will bend under loads applied at or near the ends. Consequently, the fibers at the edges of the piece are stressed beyond the elastic limit and reach their ultimate before the other fibers. It is possible, therefore, to increase the loads by gluing pieces of denser wood to the surface of the member in which the bolts bear. Fitting blocks for box beams have been made of spruce with thin pieces of birch glued to the sides. A few tests were made to throw some light on how thick the dense lamination should be and on what increase in load might be expected. Three sizes of bolts were used. The members were 2-3/4 inches wide in the direction of the bolt, except those with plywood on the outside of birch which were 3 inches. This latter construction simulated a filler block with the beam webs glued to it. The following table shows the results expressed in per cent. Tests were made with load applied parallel to the grain.

Diameter of bolt inches	$\frac{L}{D}$	Spruce	Outside lamination			
			1/8-inch birch	1/4-inch birch	1/2-inch birch	1/8-inch birch 2-ply spruce
1/4	11.0	100	110	139	155	123
5/16	8.8	100	109	135	156	
1/2	5.5	100	112	128	147	123

The maximum increase that would normally be expected is the ratio of crushing strength of the birch to that of the spruce.

It required about 1/2 inch of birch on the outside of spruce for a maximum increase in load with the size of member used. This applies only to the specimens with the outer surface of birch. If we put about 1/8 inch of two-ply 45-degree spruce plywood on the outside of the birch to correspond to the web of the box beam, about 20 to 25 per cent is all the increase we can expect.

Use of Bushings

The tests on spruce blocks with denser wood glued to the surface seem to indicate that light, hard bushings might be used very effectively. As a matter of fact, a sort of washer 1/2 inch or so thick countersunk at the ends of the bolts may be even more efficient than full bushings. Light, metal alloys or such material as bakelite, offer possibilities as bushing material. No tests were made along this line since time and funds were not available for a series of tests complete enough to be of any real value to designers.

Tightness of Nut

In preparing the charts which accompany this report, the friction of the washers was practically eliminated. It is apparent that the calculated apparent bearing stress for a short bolt with the nut drawn tight could be almost any value, depending upon how tight the nut was drawn. Tightening of the nuts so as to crush the wood does not insure that they will remain tight.

Such tightening will cause the permanent set of a member if much swelling takes place due to changes in atmospheric conditions, with the result that the nuts will no longer be tight if the piece shrinks again later on.

Conclusions and Recommendations

With the thickness of member (bearing length of bolt) fixed, the efficiency of solid bolts based on load divided by volume increases as the diameter is decreased. The weight and compactness of the fitting, the tendency to corrode, and the labor involved, will usually limit the minimum diameter which can be used effectively.

In general, the efficiency L/D relation for hollow bolts with bearing length fixed shows the same trend as that for solid bolts. There is, of course, an efficiency curve for each ratio of shell thickness to radius, the bolts with the thinner walls having the greater relative efficiency.

The substitution of hollow bolts for solid ones, the quality of the metal being the same for both, will lead to a reduction in efficiency if there is a great increase in diameter. If the difference in diameters is not too great, the hollow bolt will usually be more efficient. In the following cases the hollow bolt will ordinarily be found to be more efficient than the solid:

1. When the diameters are equal.
2. When the weight per unit length of the hollow bolt is equal to or less than that of the solid.
3. When the number of bolts are equal.

The bearing stress at elastic limit for solid bolts subjected to a symmetrical pull varies with L/D ratio in accordance with curve a, (Fig. 7) when the pull is parallel to the grain, and with curve C when the pull is perpendicular to the grain. For elastic limit bearing stresses under bolts subject to symmetrical loads acting at an angle (θ) to the grain, the following formula is recommended:

$$N = \frac{PQ}{P\sin^2\theta + Q\cos^2\theta}$$

provided P, the bearing stress parallel to the grain, has been reduced for L/D ratio and Q, the bearing stress perpendicular to the grain, has been modified for both diameter of bolt and L/D ratio in accordance with the relations shown in Figure 7, curves A, C, and D.

Allowable design stress must be based on both elastic limit and ultimate values. For long bolts, it is based on the former (because the elastic limit values are less than 50 per cent of the ultimate) and for short bolts on the latter (because the elastic limit values are over 50 per cent of the ultimate). For design purposes, it is recommended that the elastic limit values of P be multiplied by the factors given in Figure 7, curve B, for the various L/D ratios, and the elastic limit value of Q increased 33-1/3 per cent for all L/D ratios. When the elastic limit values of P and Q are thus further modified, the above formula may be used to calculate design loads.

A one-end load perpendicular to the bolt is equal to one-half of the symmetrical two-end load. For a one-end load acting at an angle to the bolt, the component perpendicular to the bolt may be made equal to a normal one-end load. These statements apply to loads acting in any direction with respect to the grain.

In calculating the allowable distance between bolts on a line for a pull parallel to the grain, our tests show that double shear may be used provided that the calculation is based on an edge to center distance and that the allowable shear strength of the material is reduced for L/D ratio by the same percentage that the elastic limit bearing stress is reduced, that is, in accordance with curve A (Fig. 7).

Cross bolts are exceedingly important in obtaining reliable joints. This is especially true if the cross-sectional dimension of the member perpendicular to the bolt is less than three times the diameter of the bolt (under these conditions we were unable to get the expected load with recommended stress without them).

It is concluded that for all practical purposes it may be assumed that several bolts of unequal size in the same fitting will sustain collectively the sum of what they will resist singly when the ratio of diameters is not greater than two to one.

It is recommended that holes be drilled with a wood bit which has side cutter lips. The holes should also be of such size and so spaced that the bolts can be driven by light taps.

The use of fitting blocks with hard dense wood glued to the surfaces, or the use of light, hard bushings, will lead to greater joint strength.

B i b l i o g r a p h y

The following reports are not available for distribution but may be had on loan for a limited period.

Reference 1. Grenoble: Bearing Strength of Bolts in Wood. (B.A.T.N. No. 147) (1925)

" Bearing Strength of Hollow Bolts in Wood. (T.M.-1 F.P.L.) (1925) 1166.5 26

" Bearing Strength of Bolts in Wood (Line of Force not Parallel to Grain of Wood). (T.M.-7 F.P.L.) (1926) 1166.5 23

Trayer : Some Factors Influencing Fitting Design with Special Reference to the Bearing Strength of Wood Under Steel Bolts. (T.M.-9 F.P.L.) (1927) 1166.5 27

Forest Products Laboratory,

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3—8494

TABLE A-4

Strength Values of Various Woods for Use in Airplane Design

(Based on 15 per cent moisture content)

COMMON AND BOTANICAL NAMES	SPECIFIC GRAVITY BASED ON VOL. AND WT. WHEN OVEN-DRY		WEIGHT AT 15% MOISTURE	SHRINKAGE FROM GREEN TO OVEN-DRY CONDITION		STATIC BENDING				COMPRESSION PARALLEL TO GRAIN		COMPRESSION PERPENDICULAR TO GRAIN [4]	SHEARING STRENGTH PARALLEL TO GRAIN [4]	HARDNESS, SIDE LOAD REQ'D TO IMBED 0.444-IN. BALL TO HALF ITS DIAMETER
	Average	Minimum permitted		Radial	Tangential	Fiber stress at elastic limit [1]	Modulus of rupture [1]	Modulus of elasticity [1]	Work to maximum load	Fiber stress at elastic limit 1-3	Max. crushing strength [1]			
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
			Lbs. per cu. ft.	*Per cent*	*Per cent*	*Lbs. per sq. in.*	*Lbs. per sq. in.*	*1,000 lbs. per sq. in.*	*In.-lbs. per cu. in.*	*Lbs. per sq. in.*	*Lbs. per sq. in.*	*Lbs. per sq. in.*	*Lbs. per sq. in.*	*Lbs.*
HARDWOODS														
Ash, black (Fraxinus nigra)	0.53	0.48	35	5.0	7.8	6,400	11,900	1,340	14.8	4,050	5,400	1,260	1,050	760
Ash, commercial white (Fraxinus sp.) [6]	.62	.56	41	4.3	6.9	8,900	14,800	1,460	14.2	5,250	7,000	2,250	1,380	1,180
Basswood (Tilia Americana)	.40	.36	26	6.6	9.3	5,600	8,600	1,250	6.6	3,370	4,500	620	720	870
Beech (Fagus atropunicea)	.66	.60	44	4.8	10.6	8,200	14,200	1,440	13.5	4,800	6,500	1,670	1,300	1,060
Birch (Betula sp.) [7]	.68	.58	44	7.0	8.5	9,500	15,500	1,780	18.2	5,480	7,300	1,590	1,300	1,100
Cherry, black (Prunus serotina)	.53	.48	36	3.7	7.1	8,500	12,500	1,330	11.7	5,100	6,800	1,170	1,180	900
Cottonwood (Populus deltoides)	.43	.39	29	3.9	9.2	5,600	8,600	1,190	7.4	3,520	4,700	650	660	410
Elm, cork (Ulmus racemosa)	.66	.60	45	4.8	8.1	7,900	15,000	1,340	19.3	5,180	6,900	2,090	1,360	1,230
Gum, red (Liquidambar styraciflua)	.53	.48	34	5.2	9.9	7,500	11,600	1,290	10.9	4,050	5,400	1,190	1,100	650
Hickory (true hickories) (Hicoria sp.) [8]	.79	.71	51	---	---	10,600	19,300	1,860	27.5	6,520	8,700	3,100	1,440	---
Mahogany, African (Khaya sp.)	.47	.42	32	4.8	5.5	7,900	10,800	1,280	8.0	4,280	5,700	1,400	980	720
Mahogany, true (Swietenia sp.) [9]	.51	.46	34	3.4	4.7	8,800	11,600	1,260	7.3	4,880	6,500	1,760	860	790
Maple, sugar (Acer saccharum)	.67	.60	44	4.8	9.2	9,500	15,000	1,600	13.7	5,620	7,500	2,170	1,520	1,270
Oak, commercial white and red (Quercus sp.) [10]	.69	.62	45	4.6	9.0	7,800	13,800	1,490	13.6	4,950	6,600	1,870	1,300	1,240
Poplar, yellow (Liriodendron tulipifera)	.43	.38	28	4.0	7.1	6,000	9,100	1,300	6.5	3,750	5,000	810	800	420
Walnut, black (Juglans nigra)	.56	.52	39	5.2	7.1	10,200	15,100	1,490	11.4	5,700	7,600	1,730	1,000	990
CONIFERS														
Cedar, incense (Libocedrus decurrens)	.36	.32	25	3.8	5.7	6,000	8,700	1,020	5.6	4,320	5,400	900	650	450
Cedar, Port Orford (Chamaecyparis lawsoniana)	.44	.40	30	4.6	6.9	7,400	11,000	1,520	8.7	4,880	6,100	1,030	760	520
Cedar, western red (Thuja plicata)	.34	.31	23	2.5	5.1	5,100	7,800	1,030	5.8	4,000	5,000	800	630	320
Cedar, white (northern) (Thuja occidentalis)	.32	.29	22	2.1	4.9	4,700	6,600	700	4.9	3,040	3,800	560	610	300
Cypress, bald (Taxodium distichum)	.48	.43	32	3.9	6.1	7,100	10,500	1,270	7.7	4,960	6,200	1,230	720	480
Douglas fir (Pseudotsuga taxifolia)	.51	.45	34	5.0	7.8	8,000	11,500	1,700	8.1	5,600	7,000	1,300	810	620
Pine, Norway (Pinus resinosa)	.51	.46	34	4.6	7.2	8,500	11,900	1,560	8.9	5,280	6,600	1,080	870	520
Pine, sugar (Pinus lambertiana)	.38	.34	26	2.9	5.6	5,600	8,000	1,040	5.4	3,680	4,600	810	730	370
Pine, western white (Pinus monticola)	.42	.38	27	4.1	7.4	6,000	9,300	1,310	7.9	4,240	5,300	750	640	360
Pine, white (Pinus strobus)	.38	.34	26	2.2	6.0	5,900	8,700	1,140	6.3	3,840	4,800	780	640	380
Spruce (Picea sp.) [11]	.40	.36	27	4.1	7.4	6,200	9,400	1,300	7.8	4,000	5,000	840	750	440

TABLE A-4

Strength Values of Various Woods for Use in Airplane Design—Continued

[1] The average values for fiber stress at elastic limit and modulus of rupture in static bending, and fiber stress at elastic limit and maximum crushing strength in compression parallel to grain have been multiplied by two factors to obtain values for use in design. A statement of these factors and of the reasons for their use follow: It was thought best, in fixing upon strength values for use in design, to give some influence to the variability of wood and to the fact that a greater number of values are below the average than above it, and the most probable value (as represented by the mode of the frequency curve) was accordingly decided upon as the basis for design figures. From a study of the ratios of most probable to average values for three species (Sitka spruce, Douglas fir, and white ash), .94 was adopted as the best value of this ratio for general application to the properties in question.

The stress that wooden members can carry depends on its duration. A factor of 1.17 has been applied to test results to get values of the stress which can be sustained for a period of three seconds, it being assumed that the maximum load will not be maintained for a longer period.

[2] The values given are the most probable values (92 per cent of the average) of the apparent modulus of elasticity (E_a) as obtained by substituting results from tests of 2 by 2 inch beams on a 28-inch span with load at the center in the formula $E_a = Pl^3/48\Delta I$. The use of these values of E_a in the usual formulas will give the deflection of beams of ordinary length with but small error. For exactness in the computation of deflections of I and box beams, particularly for short spans, the formula which takes into account shear deformations (see National Advisory Committee for Aeronautics Report No. 180, "Deflection of Beams with Special Reference to Shear Deformations") should be used. This formula involves E_t the true modulus of elasticity in bending, and F, the modulus of rigidity in shear. Values of E_t may be obtained by adding 10 per cent to the values of E_o as given in the table. If the I or box beam has the grain of the web parallel to the axis of the beam or parallel and perpendicular thereto, as in some plywood webs, the value of F may be taken as $E_t/16$ or $E_o/14.5$. If the web is of plywood with the grain at 45 degrees to the axis of the beam F may be taken as $E_t/5$ or $F_o/4.5$.

[3] Design values for fiber stress at elastic limit in compression parallel to grain were obtained by multiplying the values of maximum crushing strength as given in the next column by factors as follows: 0.75 for hardwoods—0.80 for conifers. Values as given are to nearest 10 pounds.

[4] Wood does not exhibit a definite ultimate strength in compression perpendicular to grain, particularly when the load is applied over only a part of the surface, as at fittings. Beyond the elastic limit the load continues to increase slowly until the deformation and crushing become so severe as to seriously damage the wood in other properties. Figures in this column were obtained by applying a duration of stress factor of 1.17 (see Note 1) to the average elastic limit stress and then adding 33⅓ per cent to get design values comparable to those for bending, compression parallel to grain, and shear as listed in the table.

[5] Values in this column are for use in computing resistance of beams to longitudinal shear. They are obtained by multiplying average values by .75. This factor is used because of the variability in strength and in order that failure by shear may be less probable than failure from other causes. Furthermore, tests have shown that because of the favorable influence upon the distribution of stresses resulting from limiting shearing deformations, the maximum strength weight ratio and minimum variability in strength are attained when I and box beams are so proportioned that the ultimate shearing strength is not developed and failure by shear does not occur.

[6] Includes white ash (F. Americana), green ash (F. lanceolata), and blue ash (F. quadrangulata).

[7] Includes sweet birch (B. lenta) and yellow birch (B. lutea).

[8] Includes big shellbark hickory (H. laciniosa), mockernut hickory (H. alba), pignut hickory (H. glabra), and shagbark hickory (H. ovata).

[9] Includes material from Central America and Cuba.

[10] Includes white oak (Q. alba), bur oak (Q. macrocarpa), cow oak (Q. michauxii), post oak (Q. minor), red oak (Q. rubra), Spanish (highland) oak (Q. digitata), laurel oak (Q. laurifolia), water oak (Q. nigra), Spanish (lowland) oak (Q. pagodaefolia), willow oak (Q. phellos), yellow oak (Q. velutina).

[11] Includes red spruce (P. rubens), white spruce (P. canadensis), and Sitka spruce (P. sitchensis).

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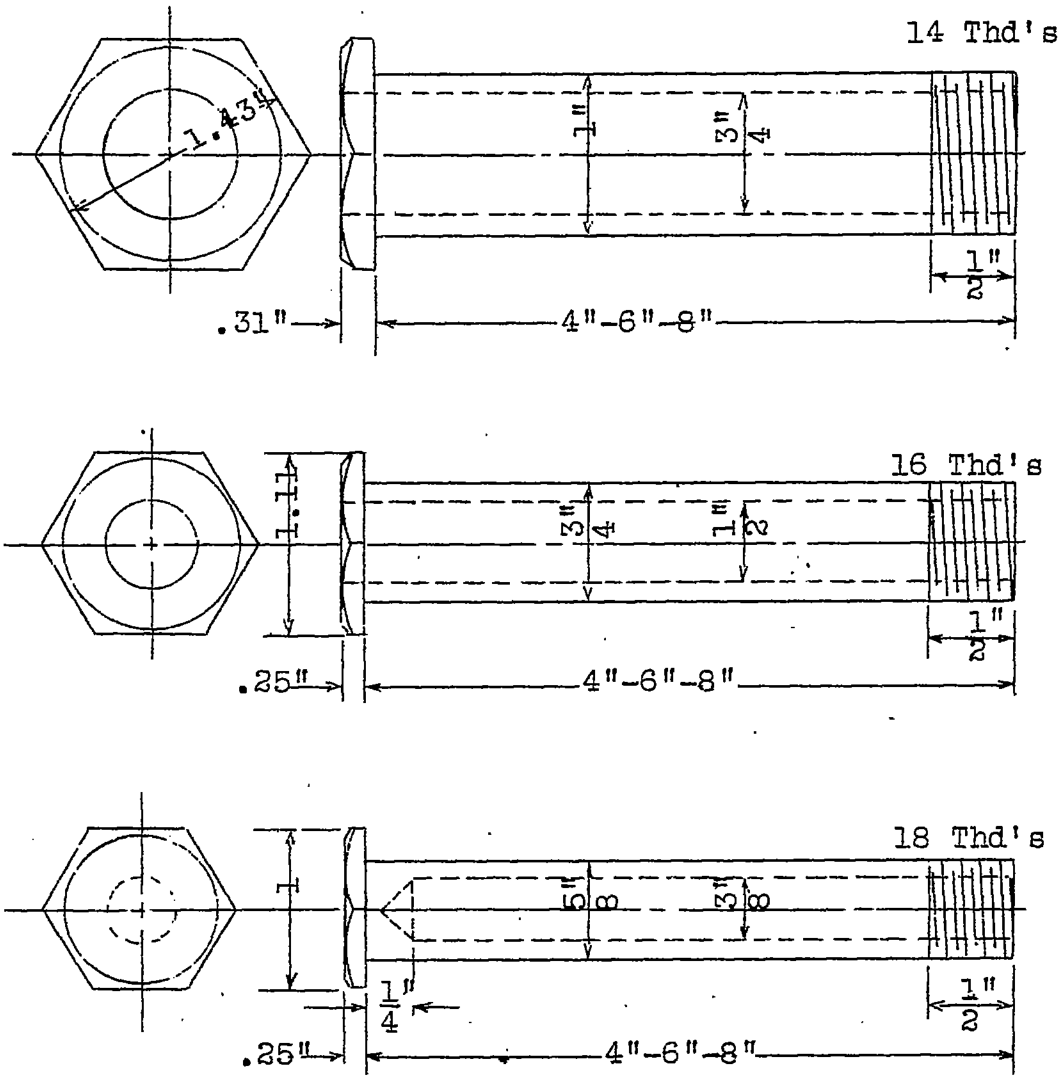

Fig.1 Dimensions of hollow aircraft bolts.

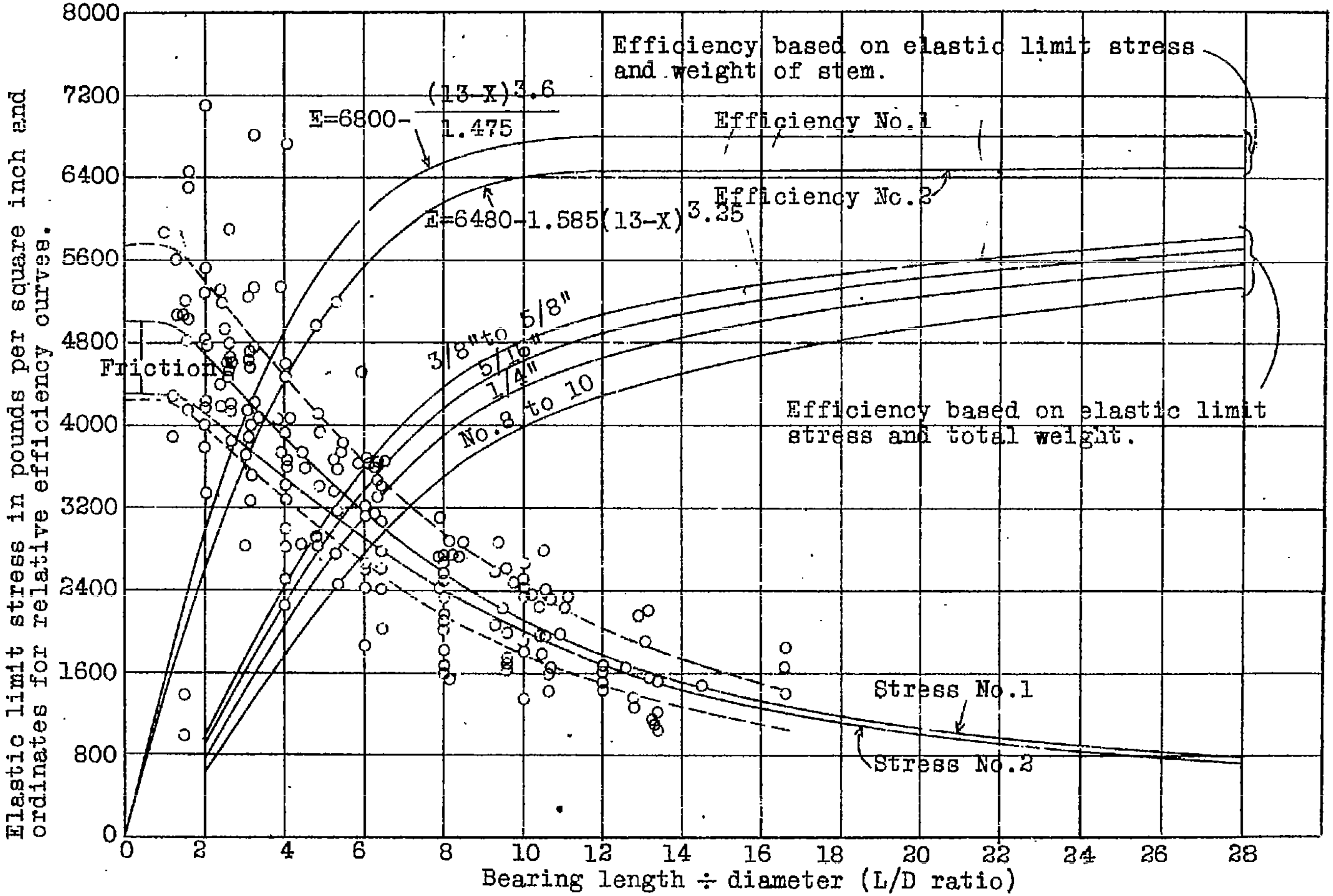

Fig.2 Stress and efficiency curves for solid aircraft bolts.

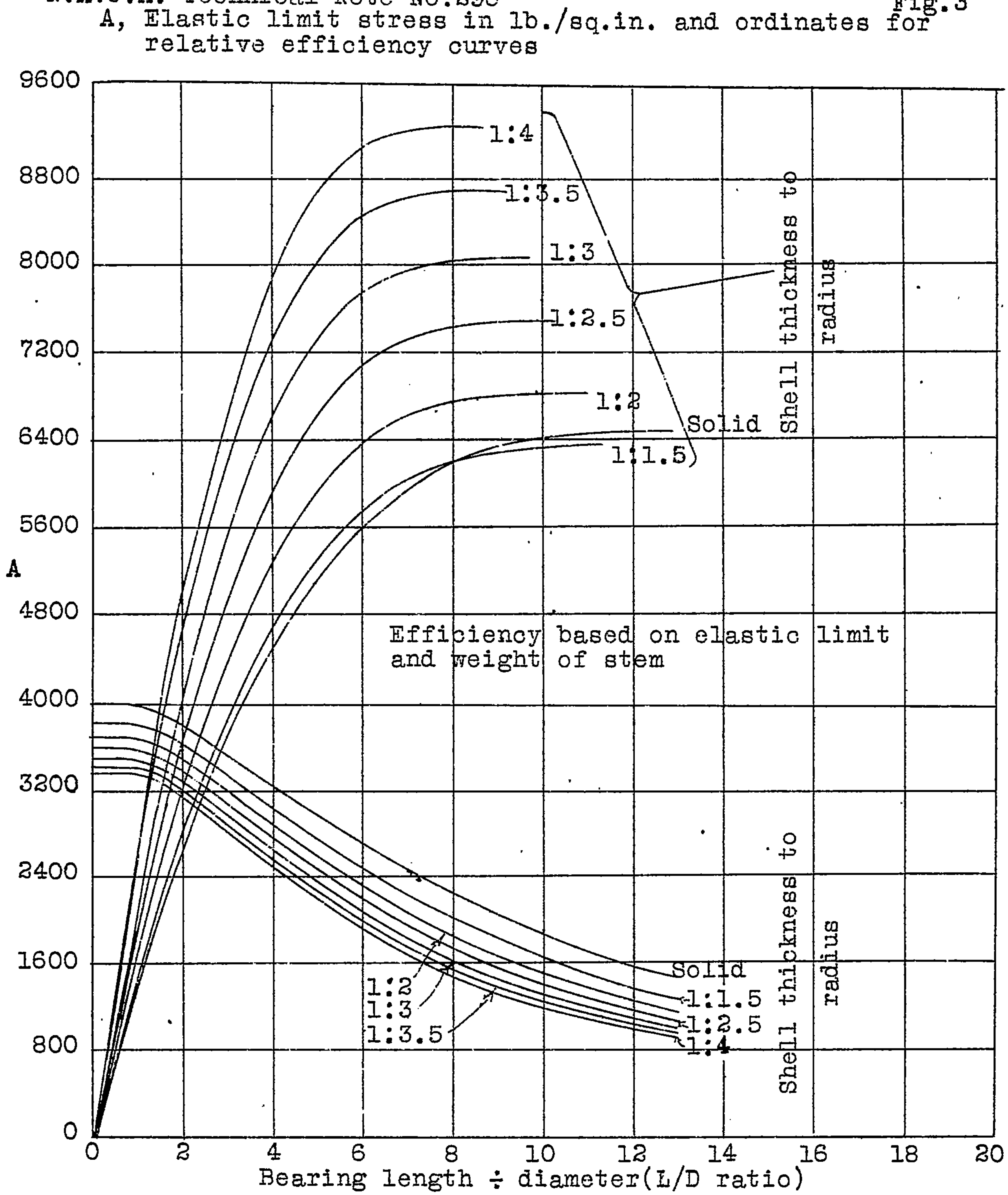

Fig.3 Stress and efficiency curves for hollow aircraft bolts.

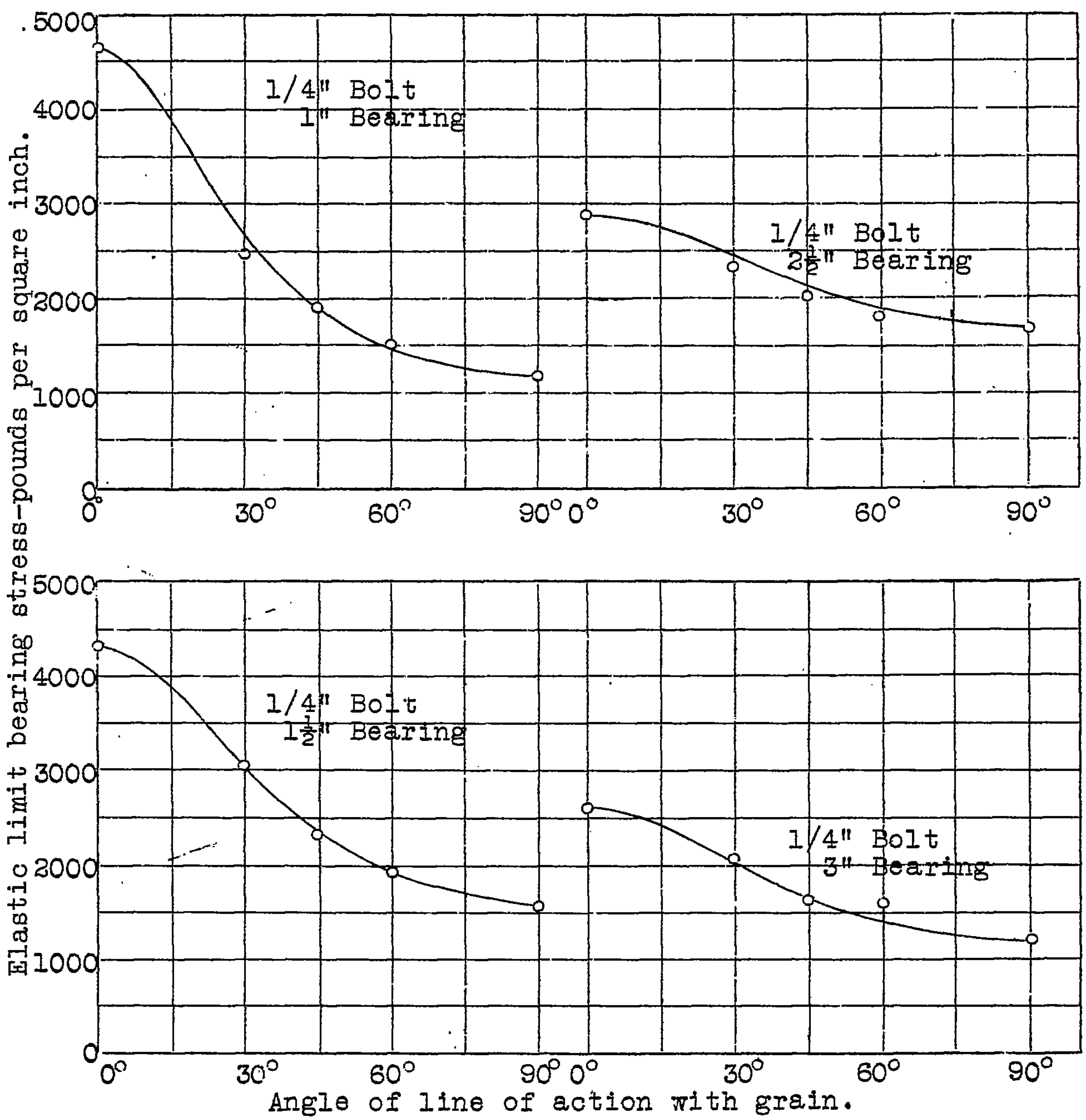

Fig.4 Curves showing bearing strength of spruce under bolts subjected to symmetrical loads acting at various angles to the grain of the wood. Each point is the average of three tests. Full line calculated by Hankinson Formula.

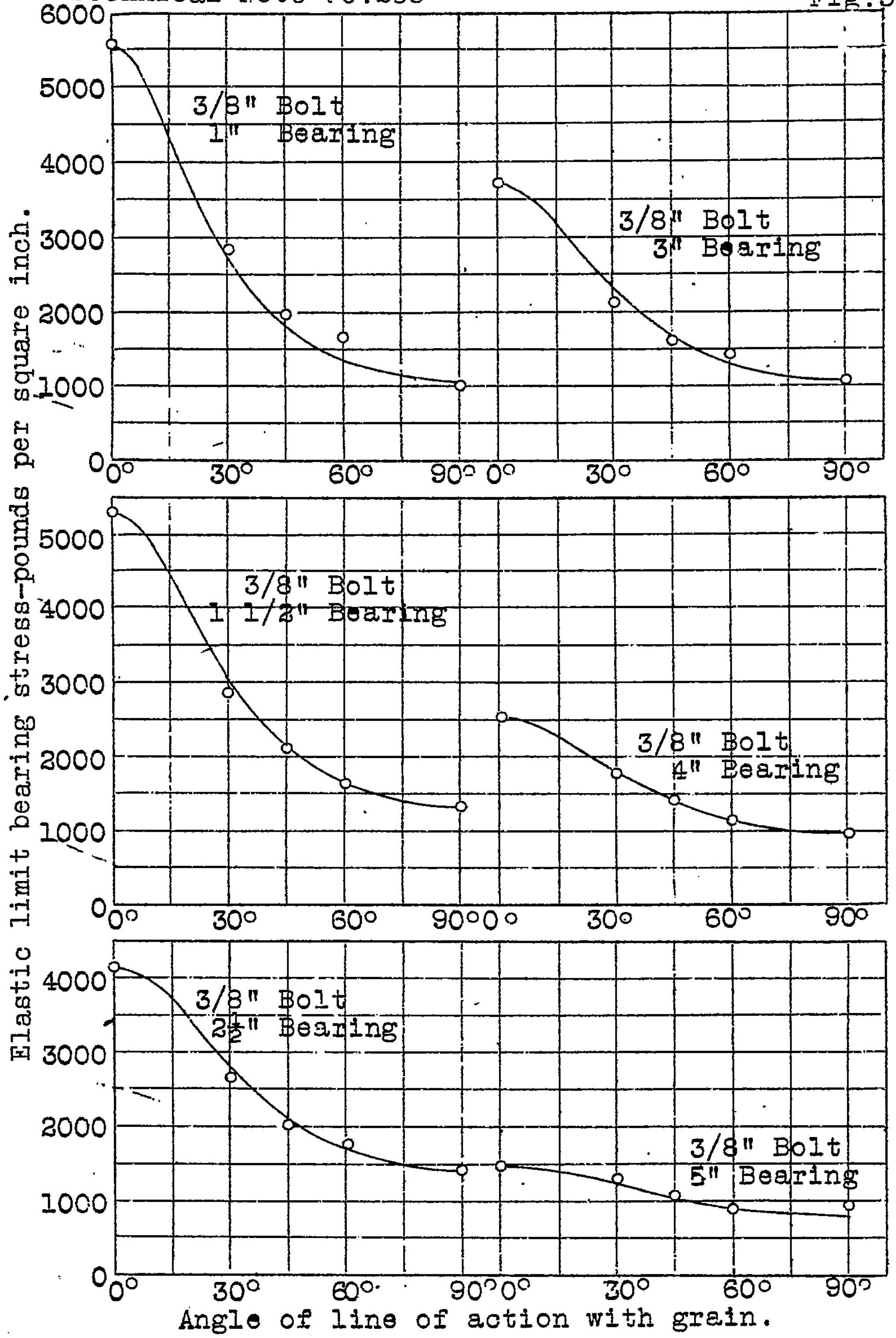

Fig.5 Curves showing bearing strength of spruce under bolts subjected to symmetrical loads acting at various angles to the grain of the wood. Each point is the average of three tests. Full line calculated by Hankinson formula.

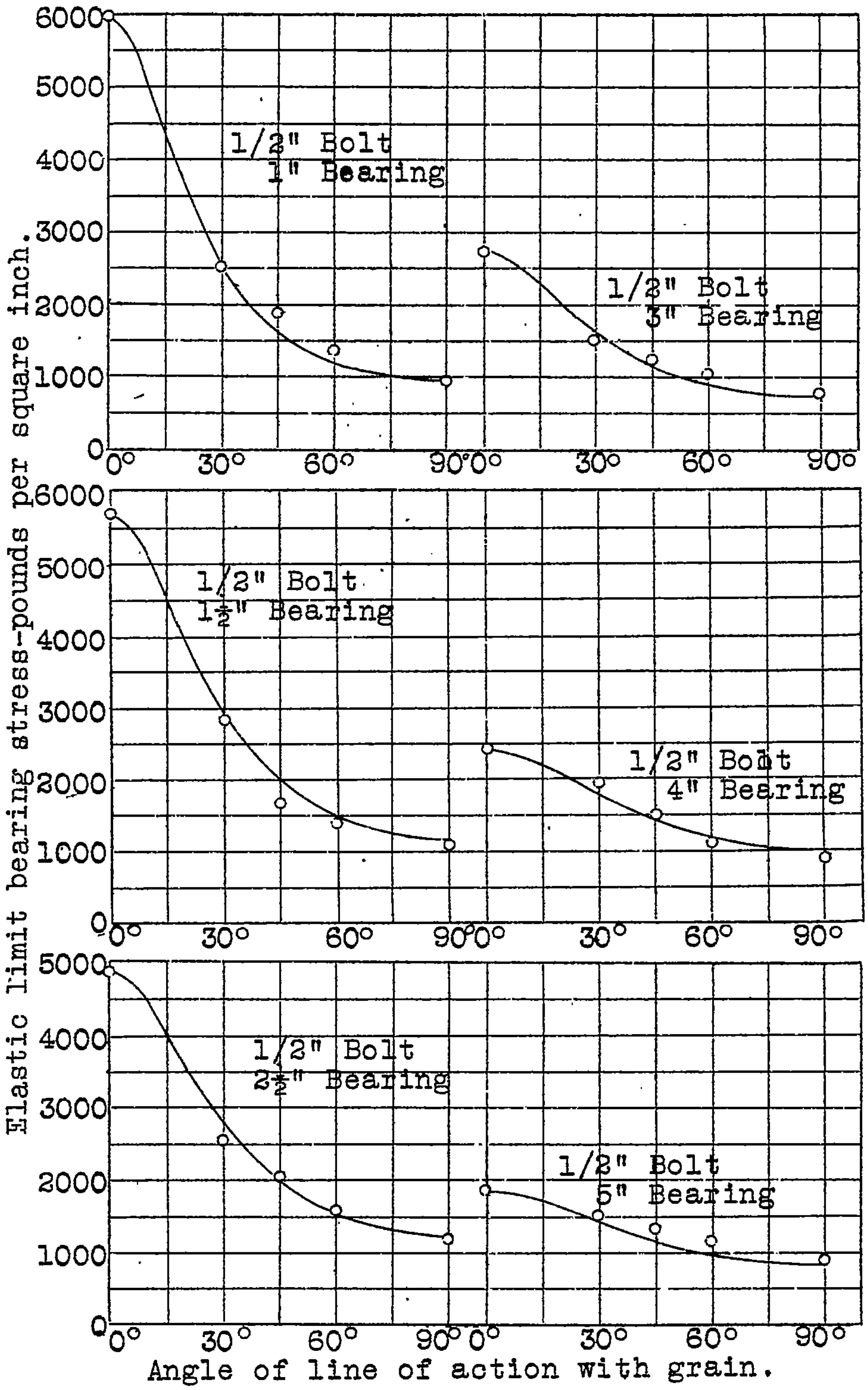

Fig.6 Curves showing bearing strength of spruce under bolts subjected to symmetrical loads acting at various angles to the grain of the wood. Each point is the average of three tests. Full line calculated by Hankinson Formula.

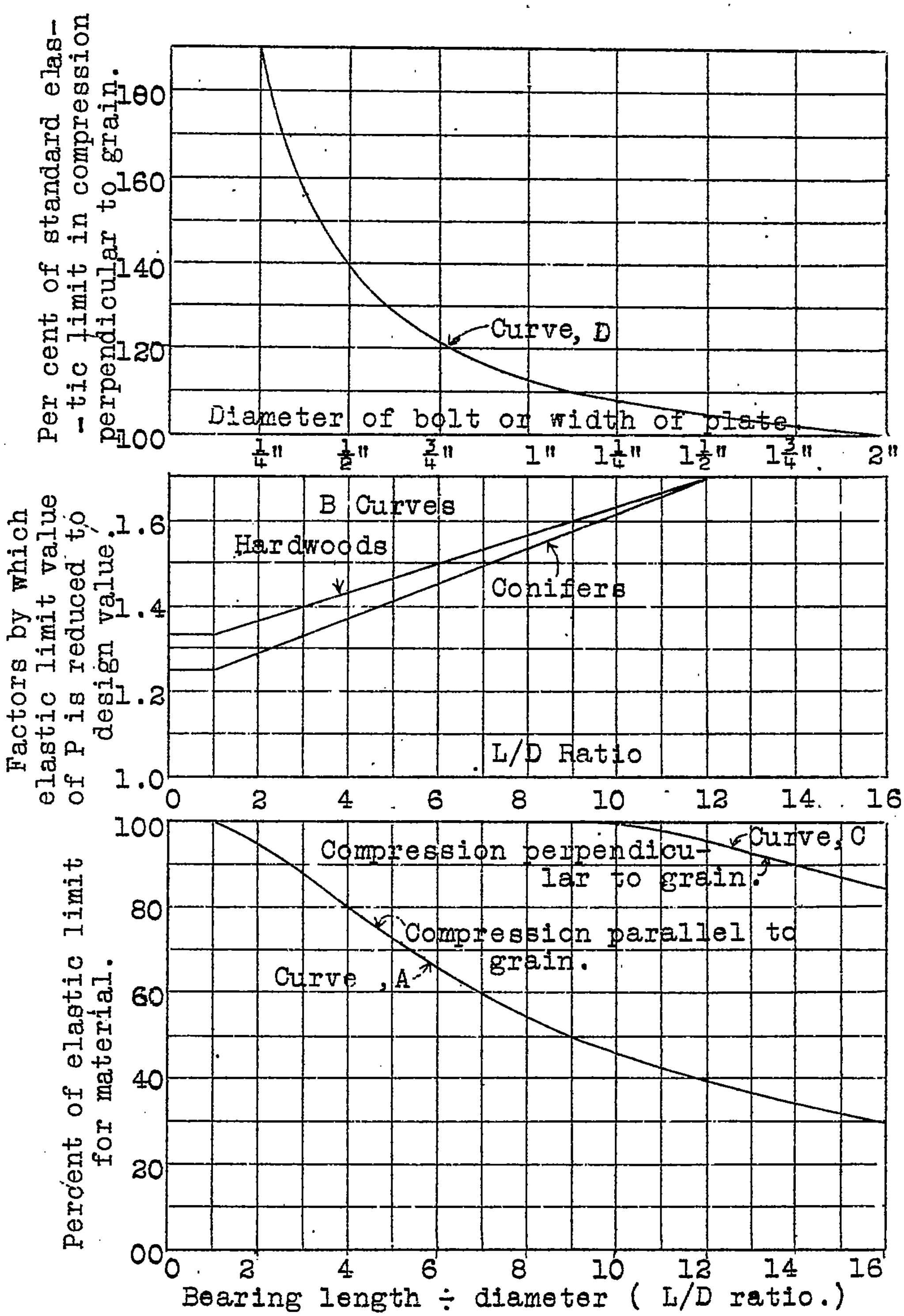

Fig.7 Curves showing relations essential to the calculation of bolt bearing values.