DESIGN FOR WELDING
IN MECHANICAL ENGINEERING

To the memory of

my brother

WALTER

DESIGN FOR WELDING

IN MECHANICAL ENGINEERING

by

F. KOENIGSBERGER
DIPL.ING., M.I.MECH.E.
MEM.A.S.M.E.

*WITH DIAGRAMS
AND PHOTOGRAPHS*

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PREFACE

THE development of the fabrication of welded structures into an efficient manufacturing method is largely due to the considerable research work which has been carried out in this field. One of the main tools of the designer, however, is experience, and this can only be gained in the practical application of the results of scientific work. As in all other fields of engineering, the effective combination of theoretical knowledge and practical experience will be the best basis for producing good designs.

This book is an attempt to co-ordinate theory and research results with the experience gained in the actual design and manufacture of welded structures as used in mechanical engineering.

Details of specifications laid down by various authorities (British Standards Institution, Institute of Welding, British Welding Research Association, Lloyd's Register of Shipping, American Welding Society, etc.) have been omitted, as the book is not intended to replace the specifications which must be consulted wherever a design has to be carried out to a particular specification. Where such specifications exist in connection with a certain subject under discussion, therefore, due reference only is made.

An attempt has been made to stress the theoretical and practical factors which might assist the designer of welded steel constructions. Much has still to be learned in this branch of mechanical engineering, and the author is aware that the principles given in the book are far from complete. He hopes, however, that the book will be useful to students and to engineers who are entering this field for the first time. It may perhaps even assist designers who have already had experience of fabrication.

It will be noted that great weight is given to the use of a correct drawing practice in connection with designs for fabricated welded construction. The author is of the opinion that the drawing-office work is the true basis for any production process in general, and for the operations in the welding shop in particular.

Various points suggested in this connection may perhaps be considered exaggerated and over-cautious, but the damage done, for example, by a faulty weld in a heavy and complicated structure may far outweigh the savings gained by omitting such precautions. Some of the suggestions may appear too complicated at first reading ; but much suspicion and prejudice still persists among engineers with regard to welded products, not only so far as their technical reliability is concerned, but also with regard to their economy. If, however, fabricating shops achieve success by the application of principles such as those outlined in this book, the author ventures to suggest that it should not be difficult to destroy the suspicions and to produce technically efficient and economically competitive welded products.

The author is indebted to the firms who have assisted him by the provision of photographs and drawings of their products and who are named wherever their products are shown and discussed.

The author's own designs, which he executed when in charge of welding design at Messrs. Cooke and Ferguson Ltd., Manchester, are, of course, used extensively as examples to stress his points, and the author is indebted to the directors of that company for permission to publish them. He also wishes to thank Messrs. Cleaver Hume Press Ltd., London, for permission to use parts of his contribution to their book on *Modern Workshop Technology*, and Mr. H. C. Town and Messrs. Paul Elek (Publishers) Ltd., London, for permission to use illustrations from his contribution to their *Workshop Yearbook and Production Engineering Manual.*

Professor Dr.-Ing. h.c. M. Ros, Zürich, kindly provided photographs of typical weld failures.

Permission to quote from *British Standard Specifications*, *Lloyd's Rules and Regulations*, *Welding Memoranda of the Advisory Service on Welding* by permission of the Controller of H.M. Stationery Office, and from the *Handbook for Electric Welders*, published by Messrs. Murex Welding Processes Ltd., is also gratefully acknowledged.

The author is indebted to Professor H. Wright Baker, Manchester College of Technology, for his kind support and his able advice and suggestions. Valuable assistance has been generously given by Mr. E. J. Heeley and Mr. H. Diederich in connection with problems on the design of pressure vessels, and by Mr. E. Bill on matters relating to resistance welding procedures.

Mr. E. Johnson of the Manchester College of Technology has kindly read through the script.

Finally, the author is greatly indebted to his father for his valued assistance and encouragement.

F. KOENIGSBERGER

MANCHESTER, *Spring* 1947

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INTRODUCTION

ON THE TECHNICAL AND ECONOMIC CONSIDERATIONS WHICH DECIDE WHETHER OR NOT FABRICATED WELDED CONSTRUCTION IS ADVANTAGEOUS IN PARTICULAR CASES

THE design of fabricated welded structures is not a particular science which requires specialists to work continuously upon its development. It is, however, a fairly new departure in engineering design. If a designer intends to use fabrications by welding he must have a clear idea not only how, but also where, to apply this method of manufacture, and, to this end, must know the technical and economic advantages which it holds over other alternative methods. Consideration should be given to the mechanical properties of the material, the problems which arise in connection with the manufacturing processes which might be employed, and the cost of production.

(1) *Important properties of materials for mechanical construction:*

- (*a*) Tensile strength.
- (*b*) Compressive strength.
- (*c*) Strength against shock loads.
- (*d*) Stiffness against distortion.
- (*e*) Rigidity, and vibration damping qualities.
- (*f*) Sliding qualities.

(2) *Problems arising from the manufacturing process:*

- (*a*) Difficulties in maintaining wall thicknesses in castings.
- (*b*) Combination of various wall thicknesses.
- (*c*) Limitation of wall thicknesses.
- (*d*) Special additional material allowances.
- (*e*) Residual stresses.
- (*f*) Modifications during manufacture.
- (*g*) Available manufacturing facilities.

(3) *Economic considerations:*

- (*a*) Cost of patterns or fixtures for small or large quantity production.
- (*b*) The problem of saving material at the expense of additional labour cost or vice versa.

In general engineering practice the fabricated welded construction is usually an alternative to either fabricated riveted construction or the use of iron or steel castings. The material properties and manufacturing problems connected with these alternatives, therefore, may have to be considered when selecting the material and manufacturing method for a particular job.

1. Properties of Materials

(*a*) *Tensile strength.*

Due to the low elastic limit of cast iron, the permissible tensile stress in a cast-iron structure should be about one-third of that permissible in mild steel or in a steel casting. The cost of high-duty cast iron, with a tensile strength approaching that of mild steel, is higher than that of ordinary cast iron.

(*b*) *Compression strength.*

In the case of compressive stresses the comparison is less unfavourable for cast iron.

(*c*) *Strength against shock loads.*

With regard to shock-absorbing qualities, mild steel is the more favourable material.

(*d*) *Stiffness against distortion.*

The modulus of elasticity of mild or cast steel is about twice that of ordinary cast iron. Assuming equal sections and equal loading conditions, the former are, therefore, about twice as stiff as the latter against distortion under either static or pulsating loads.

(*e*) *Rigidity, and vibration damping qualities.*

Danger of resonance exists if the natural frequency of the structure is too close to the periodicity of load impulses such as may be provided by running machinery supported by the structure.

The natural frequency of a structure is higher the greater the stiffness and the lower the weight of the structure. Due to the higher modulus of elasticity of steel than of cast iron, it is possible to get higher natural structure frequencies with the former, as greater stiffness can be obtained with smaller sections, and therefore with lower weight. The vibration damping capacity of cast iron, however, is superior to that of steel.

(*f*) *Sliding qualities.*

Cast iron has better sliding qualities and greater resistance to wear than steel, unless the steel is case hardened and ground.

2. Manufacturing Problems

(*a*) *Maintenance of wall thicknesses.*

Core displacements which are unavoidable in a moulding may cause deviations from the nominal wall thickness of a casting.

To allow for possible reduction of wall thicknesses, the designer has to add to the theoretically required wall thickness an amount equal to the tolerance in the mould (Fig. 1). This has to be done in the case of all castings, whether steel or iron castings, and

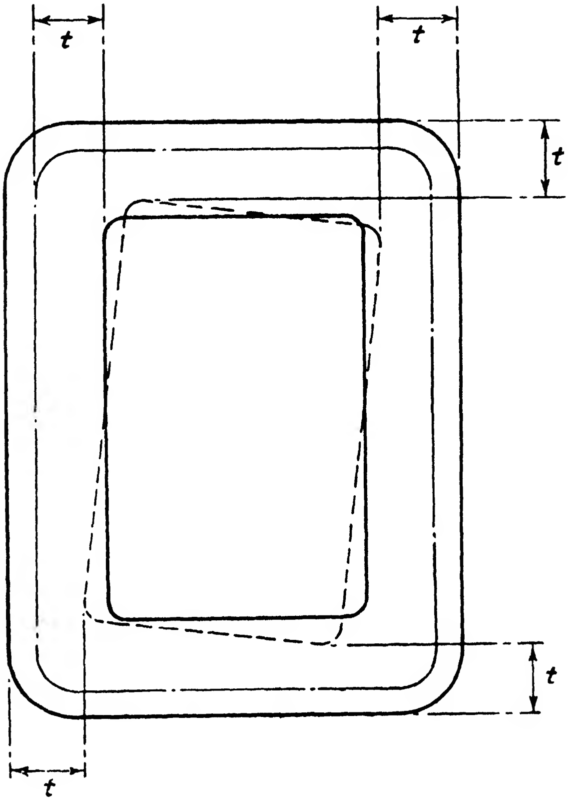


FIG. 1

— · — · — Theoretically required section (width t)
- - - - - - Core displacement
———— Contour of final design which maintains width t when allowing for core displacement

requires additional material over and above the theoretical minimum.

(*b*) *Combination of various wall thicknesses.*

Although with certain precautions it is possible to combine almost all sizes of steel plates in a welded structure, care must be taken in the case of castings, where variation of wall thicknesses may create difficulties in the flow of material due to different rates of cooling, and may also cause the formation of blow holes. It is, therefore, sometimes necessary to arrange for the walls of a cast structure to be thicker than actually necessary to satisfy the requirements of strength or stiffness. This, again, means additional material over and above the theoretical minimum.

(*c*) *Limitation of wall thicknesses.*

The minimum wall thickness in a cast structure is limited by such factors of the casting process as the flow of material, and the casting may easily become heavier than is theoretically required for strength reasons. Practically no such limitations are imposed in the case of the fabricated structure.

(*d*) *Special additional material allowances.*

For riveted or spot-welded assemblies additional flanges must often be provided which are not necessary in the case of arc-welded assemblies (Fig. 2). This may frequently mean a considerable

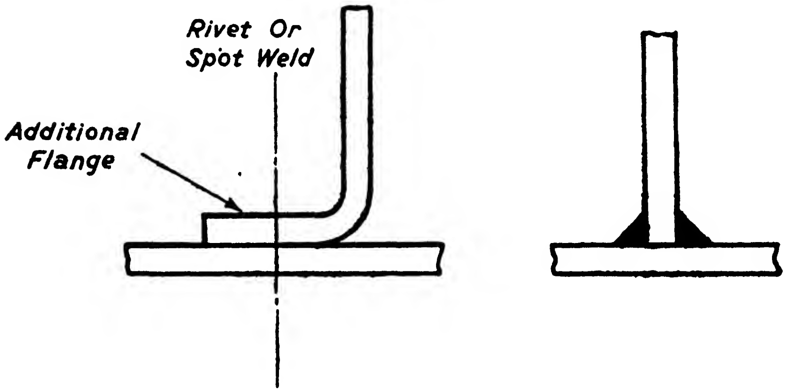


FIG. 2

increase in the weight of material. In the case of riveted structures, due allowance must also be made for the weakening of plates and rolled sections by rivet holes. No rivet holes are required in welded structures, and consequently smaller sections may be used. As an example of the considerable savings in weight thus obtained in a welded structure, the weight of one welded crane bridge 58 ft. long was only 70 per cent of that of the corresponding riveted crane (11 tons against 16 tons).

(*e*) *Locked-up stresses.*

Residual stresses may be created in castings if small sections and parts which are particularly exposed have a higher cooling rate than others, and cracks or creeping of the structure during its working life may occur as a consequence. Heat treatment or seasoning may be used for reducing or eliminating such stresses. In fabricated welded structures residual stresses may be caused during the welding process. They may create considerable difficulties in heavily stressed structures or in structures which have to maintain high dimensional accuracy. In the case of such structures heat treatment to an accurate specification is of vital importance.

(*f*) *Modifications during manufacture.*

In the case of prototype machines of which only one is required and where modifications may be necessary according to the experimental development of the job, welded construction has the advantage that no patterns are required and that modifications or rectifications can easily be carried out by flame cutting, adding parts to the structure or building up faces by welding.

(*g*) *Available manufacturing facilities.*

The available manufacturing facilities, such as a good foundry or an efficient fabricating shop, may turn the scale in favour of one of the processes, especially in cases where other considerations do not particularly favour the use of either. In this connection, a warning may be given not to make the mistake of overriding technical and economical considerations and favouring a particular method only because efficient manufacturing equipment is available. A prejudice which neglects the use of other manufacturing possibilities may result in the inefficient use of a valuable manufacturing method.

3. Economic Considerations

(*a*) *Cost of patterns or fixtures.*

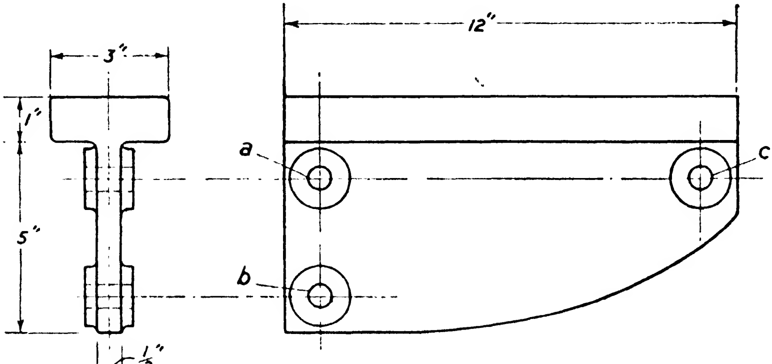
With small-quantity production of castings, the cost of patterns will influence considerably the total cost of the component. Though it is possible to build fabricated structures in small quantities without the help of fixtures, efficient welding fixtures are essential for quantity production and where interchangeability is required.

(*b*) *The problem of saving material at the expense of additional labour cost or vice versa.*

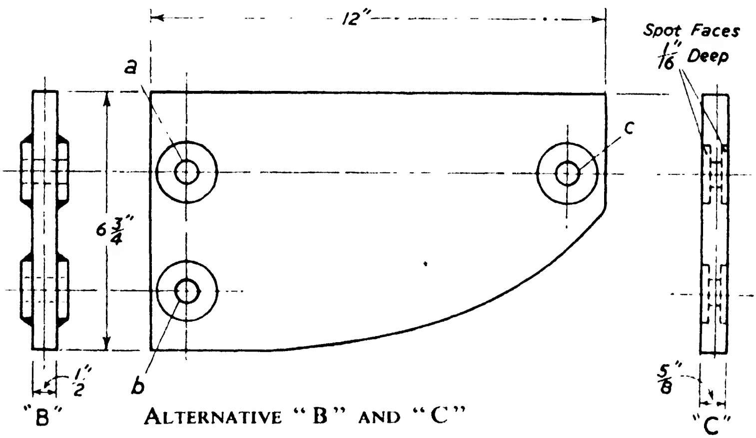
In the fabrication of a welded structure, welding time is usually the biggest factor on the labour cost side. It may be stated that in general the most economical fabricated welded structure is the one which requires the minimum amount of welding. The saving in material cost which is possible in a fabricated steel structure may often be lost or exceeded by the cost of the additional labour required. It is the task of the designer to find a solution for which the total of labour and material costs is a minimum.

As an example, the design of a bracket arm (Fig. 3) may be considered.

Under working conditions the arm is held by two bolts at *a* and *b*,



Alternative "A"



Alternative "B" and "C"

Fig. 3

and a load applied through the bolt *c* causes a bending stress in the arm. Three alternative designs will be compared :

A, for an iron casting.

B, for a steel plate with bosses similar to those on the casting for the fastening nuts and washers.

C, for a steel plate which is slightly thicker than actually required for strength reasons, to allow the machined faces, for the fastening nuts and washers, to be spot-faced into the surface of the material.

In the following comparison of costs, the labour cost for the machining operations of drilling, facing, or spot-facing will not be considered, as they will be practically identical in all three cases.

For the calculation of the section moduli the permissible bending stress in tension for steel has been assumed to be 2·5 to 3 times the stress permissible for cast iron.

Alternative A (*cast iron*).

The section modulus is 9·4 inch^3 on the tension side and 4·25 inch^3 on the compression side.

The weight of the casting is about 16 lb., its cost, at about 30s. a cwt., 4s. 4d.

Alternative B (*steel plate with bosses welded on*).

The section modulus is 3·75 inch^3 in both tension and compression. The weight of the steel plate is 10 lb., its cost, at 20s. per cwt., is 1s. 9d.

The labour cost for flame-cutting the plate, and for cutting and welding the bosses, plus 120 per cent overhead charges, will be about 1s. 8d. + 2s. = 3s. 8d.

The factory cost of alternative B would therefore be 1s. 9d. + 3s. 8d. = 5s. 5d.

Although there is a saving in material of about 37·5 per cent by weight or 60 per cent by value, the design which copies the bosses from the cast-iron construction proves to be 25 per cent more costly, due to the additional labour involved.

Alternative C (*bosses eliminated at the expense of using a plate $\frac{1}{8}''$ thicker, with the faces for the nuts and washers spot-faced $\frac{1}{16}''$ deep on either side of the bracket arm*).

The weight of the plate is 12·5 lb., its cost, again at 20s. per cwt., is 2s. 3d. Labour, plus 120 per cent overhead for flame cutting, is about 4d. + 5d. = 9d.

The factory cost of alternative C is, therefore, 2s. 3d. + 9d. = 3s.

The alternative which uses slightly more material than is theoretically required but saves the cost of welding, thus proves to be most economical. This applies, of course, only to the particular case mentioned in the example. If, on the other hand, the size of the plate is very large in relation to the number and size of the bosses, a slight increase in plate thickness might result in a large increase in weight and cost of material, and alternative C would then become less economical. In such a case it may even be possible that the casting proves more economical than both the other alternatives.

The example shows that no general rule can be given, and that each case must be checked individually.

COMPARISON OF SHAPE, LAYOUT, AND MATERIAL CONSUMPTION

It should be remembered that stiffness and strength considerations and the desirability of utilizing material to its full capacity, do not alone influence the quantity of metal required in a certain design. The shape of the structure and its layout have an important influence in the matter. Consider the example of a bending load on a simple cantilever of rectangular section (Fig. 4).

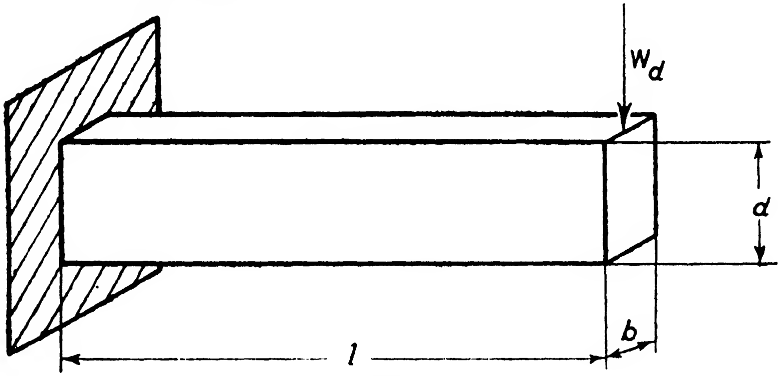


FIG. 4

In the following calculation only deflections and stresses due to bending are considered, though the influence of shearing stresses becomes considerable when d/l exceeds a certain value. The following symbols are used :

E = modulus of elasticity of material	lb./inch²
I = moment of inertia of beam section	inches⁴
V = volume of beam	cubic inches
W_d = static load on beam	lb.
Z = section modulus of beam section	inches³
b = width of beam	inches
d = depth of beam	inches
l = length of beam	inches
f_b = bending stress	lb./sq. inch
δ = deflection of beam under load	inches

The equation for the deflection of the cantilever is

$$\delta = \frac{W_d l^3}{3\,EI} = \frac{12\,W_d l^3}{3\,Ebd^3}.$$

The volume of material used for the cantilever is

given by $$V = bdl,$$

so that $$\delta = 4 \cdot \frac{1}{E} \cdot W_d \cdot \frac{1}{V} \cdot \left(\frac{l^2}{d}\right)^2$$

and $$V=4\,.\,\frac{W_d}{\delta}\,.\,\frac{1}{E}\,.\left(\frac{l^2}{d}\right)^2. \qquad (1)$$

The bending stress is given by

$$f_b=\frac{W_d l}{Z}=\frac{6W_d l}{bd^2}=6W_d\,.\,\frac{1}{V}\,.\,\frac{l^2}{d}$$

or $$V=6W_d\,.\,\frac{1}{f_b}\,.\,\frac{l^2}{d}. \qquad (2)$$

Equation (1) shows that for a certain permissible deflection the amount of material (volume V) used in the structure depends upon the modulus of elasticity of the material and the square of the ratio of the square of the length to the depth of the section of the cantilever, that is $\left(\frac{l^2}{d}\right)^2$.

With regard to strength, equation (2) shows that the volume of material required depends upon the permissible stress and again upon the ratio $\frac{l^2}{d}$.

As the modulus of elasticity and the permissible stress are constant factors for each material, the designer has the possibility of varying the ratio $\frac{l^2}{d}$ (see also page 94).

It will be obvious from the above equations that there is only one value of this ratio for which both the strength and stiffness of the material can be fully exploited.

It is very unlikely that general design considerations will allow this ideal ratio to be obtainable, but it is interesting to note that Krug* once worked out this most favourable condition in order to prove the advantage of applying fabricated steel construction to machine tool beds. In the case of the most favourable ratio $\frac{l^2}{d}$, and from equations (1) and (2), the volume of material required is

$$V=\frac{9W_d\,\delta\,E}{f_b^2}. \qquad (3)$$

Comparing this volume when using cast iron (permissible bending stress assumed to be 1500 lb. per sq. inch, modulus of elasticity 15,000,000 lb. per sq. inch) with the corresponding volume for mild steel (permissible bending stress 4500 lb. per sq. inch, modulus of elasticity 30,000,000 lb. per sq. inch), the following equation is obtained :

$$\frac{V_{\text{cast iron}}}{V_{\text{steel}}}=\frac{\dfrac{9\,W_d\,\delta\,E_{\text{cast iron}}}{f_b{}^2{}_{\text{cast iron}}}}{\dfrac{9\,W_d\,\delta\,E_{\text{steel}}}{f_b{}^2{}_{\text{steel}}}}=\frac{E_{\text{cast iron}}}{E_{\text{steel}}}\times\left(\frac{f_{b\ \text{steel}}}{f_{b\ \text{cast iron}}}\right)^2,$$

* *Maschinenbau*, 1931.

for which

$$\frac{V_{\text{cast iron}}}{V_{\text{steel}}} = \frac{15000000}{30000000}\left(\frac{4500}{1500}\right)^2 = 4 \cdot 5.$$

This would mean that, in the case of the most favourable conditions, the amount of material required for a cast-iron structure would be 4·5 times that required for a mild-steel structure of equal loading capacity and stiffness.

As the free length of a cantilever is usually determined by general design considerations, this would mean that the depth of the steel structure should be considerably different from that of the casting. In fact, in order to exploit each material to its fullest extent, a steel structure should be much deeper and of smaller wall thickness than one of cast iron. As already mentioned, this theoretically most favourable condition can very seldom, if ever, be obtained, but the calculations show that the whole shape and design of a structure should be laid out according to the properties of the material used.

Chapter 1

THE MATERIALS USED FOR FABRICATED WELDED STEEL STRUCTURES

ALTHOUGH most metals can be welded successfully, it is proposed to deal here only with welded structures built from ferrous metals. The materials available range from low carbon steels to high-tensile alloy steels. They may be obtained in the form of plates, rolled sections, drop forgings, or—with certain exceptions—steel castings.

When deciding upon the most suitable material for a particular job, the designer has to consider :

A. The mechanical properties of the material (strength, stiffness, and resistance to wear).
B. The weldability of the metal.
C. Such properties as are advantageous from the point of view of economical and efficient manufacture in the fabricating shop.

These will now be considered in turn.

A. The Mechanical Properties of Materials

Strength.

The tensile strength of the steels generally used varies from about 26 tons per sq. inch for ordinary mild steel to about 45 tons per sq. inch for high-tensile steel. Ordinary steel castings have a tensile strength similar to that of mild steel.

The influence of the welding process upon the strength characteristics of the material adjacent to the weld should not be overlooked, especially if it is not intended to apply heat treatment after welding. In case of critically stressed portions, it is advisable to base the strength calculations on the minimum tensile stress specified for the particular material rather than on the average stress. As an example, in the case of 28/32 ton steel, the safe stress should be based on the value of 28 tons per sq. inch. It is a well-known occurrence that, in correctly welded tensile test pieces, failure occurs in the parent metal close to the weld instead of in the weld itself. This is due to the ductility of the weld metal being usually different from that of the parent metal. Stress concentrations which occur in the welded zone will often be more easily " ironed out " in the weld than in the zone adjacent to it, and cracks may start in the latter. Furthermore, the welding process causes heating and cooling of the material adjacent to the weld, and consequently leads to the coarsening of the grain, thus influencing both structure and strength.

Stiffness.

When studying the mechanical requirements for a design, it is essential to determine carefully whether strength or stiffness alone is required, or a combination of these qualities, and then to select the material to suit the conditions.

The modulus of elasticity of all steels quoted in this chapter is within about $\pm$ 3 per cent of 30,000,000 lb. per sq. inch. Nothing can be gained, therefore, by using a material of high-tensile strength in a certain part of a welded structure if stiffness alone is the deciding factor.

Resistance to Wear; Sliding Qualities.

In the case of bearing surfaces, such as machine slides, the resistance to wear and the sliding qualities of the material are of prime importance. Whilst cast iron is highly suitable for such applications, most steels do not give the required result unless they are hardened and ground.

The designer has the following alternatives :

(1) *The application of cast-iron strips or blocks, which are bolted to the welded structure.*

This means additional machining operations, as both the welded structure and the casting have to be machined prior to the casting being bolted on, as well as an additional fitting operation, involved in the assembly of the casting to the welded structure.

(2) *The use of mild steel hard-surfacing by welding.*

This may involve additional heat treatment after the application of the harder metal.*

(3) *The use of steel which can be welded into the structure as an integral part and the surface of which can be hardened by local heat treatment.*

The Shorter flame-hardening process is effective and suitable in such cases. Ordinary mild steel will not give sufficient surface hardness when flame-hardened, as its carbon content is too low, but 0·4 per cent carbon steel has given good results. If appropriate precautions are taken, this material can be welded without great difficulties, and a surface hardness of 500–600 Brinell can be obtained with the Shorter process.

* See M. Riddihough, " Hard Surfacing by Welding," *Transactions of the Institute of Welding*, May 1945, page 58.

B. Weldability

In his paper, " Factors Controlling the Weldability of Steel,"* L. Reeve defines Weldability as follows :

" Weldability is a combined property of base metal and filler metal or electrode, the measure of which is the capacity to produce crack-free and mechanically satisfactory welded joints, by as many as possible of the known welding processes."

From the above definition it will be seen that three factors affect the weldability : the properties of the base metal, the quality of the electrode, and the welding process and procedure. All three factors have to be considered, and their influence must be known to the designer. The material he selects for his welded construction must not only fulfil the mechanical requirements which are determined by the working conditions of the completed structure, it must also be suitable for economic and efficient assembly by welding, *i.e.* it must be appropriately weldable.

It is not proposed to give here any detailed discussion of the problems involved. The results only of the various technical investigations will be given in so far as they have to be considered by the designer.

With regard to weldability, the following classification of materials may be made :

Group 1. Materials which can be welded by any of the known welding processes, with standard equipment (plant, electrodes) and without special precautions.

Group 2. Materials as under 1, but for which special measures have to be taken during the welding procedure (for example, preheating in the case of arc or gas welding ; increased welding times in the case of spot welding).

Group 3. Materials which can be welded by certain processes only and for which special equipment (special welding rods in the case of arc or gas welding, special control equipment in the case of spot welding) is necessary.

The designer must give consideration to the above, and should he decide to use one of the materials in groups 2 or 3, he should give definite instructions to the welding shop with regard to procedure and equipment to be used. This is necessary in order to obtain in the completed structure the mechanical properties required for his design. When high-tensile alloy steels and steel castings are used, knowledge of their weldability is of great importance.

In this connection it must be remembered that the probability of mistakes in the shop is higher, and the need for special precautions (and the expense caused by these precautions) greater, the more com-

* Presented to the Institute of Welding on April 12th, 1944, and published in *Welding*, November 1944, page 521, and December 1944, page 573.

plicated the procedure necessary to ensure satisfactory welds. From these considerations it is obvious that materials should be used, so far as possible, which fall under group 1.

For materials of group 2 no special equipment is required. Exact instructions with regard to correct procedure and careful supervision during the welding operations should be sufficient to ensure efficient welds. Materials of group 2 should, therefore, be considered as the next desirable after those of group 1.

It is for materials of group 3 that special precautions have to be taken.

The storage of the welding rods requires attention. To prevent incorrect selection the movement of equipment in the shop must be carefully checked, and appropriate control gear for resistance welding machines is required.

Furthermore, no mistakes must be made in the use and application of such special equipment. It is particularly important to prevent its use where it would be not only unnecessary but even detrimental to the job. The use of materials of group 3, therefore, apart from their high purchase price, will cause higher overhead expenses in the welding shop.

The mechanical soundness of a weld is affected by chemical and physical influences, among them being oxidation, vaporization, the tendency to cause inclusions of foreign matter and blow holes ; but all research workers seem to agree that the main problem in connection with weldability is the development of cracks either in the weld itself, in the junction zone between the weld and the parent metal, or in the parent metal adjacent to the weld. The loss of ductility, the " hardenability ", as Reeve calls it, is the most important factor.

This hardening effect is caused by the local heating created by the welding process and the quenching action created by the neighbouring cold metal. This may reduce the ductility to such an extent that the material will not stretch under the internal stresses created by the welding process.

From the *Memorandum on the Arc Welding of High Tensile (Low Alloy) Structural Steels** the following may be quoted :

" The weldability of a high-tensile steel decreases with an increase in the percentage of carbon and other alloying elements which it contains, the effect of carbon being the most important, and far greater than that of the other elements. The greater the amount of carbon, manganese, nickel, chromium, molybdenum, copper, etc., present in the steel, the harder will the weld junction tend to become and the greater will be the tendency for the formation of cracks in the hardened zone. . . .

" The hardening of the weld junction results from the sudden cooling of the metal, and since thick plate gives a quicker cooling rate, the hardness of the heat-affected zone increases with plate thickness.

* Report T/2, issued by the Welding Research Council of the Institute of Welding, February 1944.

" As the thickness increases, the potential difficulties increase, particularly the danger of cracking in the hardened zone when welding restrained joints.

" For a given steel, the hardness of the hardened zone can be kept down by reducing the cooling rate of the weld and adjacent plate. This can be done by increasing the size of the electrode and by increasing the size of the weld metal run, and also by preheating and post-heating, all of which increase the concentration of heat put into the plate. . . .

" With special types of mild steel electrodes the tendency to crack is reduced. . . .

" The deposit from . . . austenitic electrodes is immune from cracking whatever the size of the weld or composition of the high-tensile steel plate, within wide limits. The presence of the hard zone must, however, be taken account of in considering the use of the finished article."

In the case of spot welding, high carbon content may again result in a hardening effect producing hard and brittle welds. To overcome this difficulty, low-current settings and long welding times may be used to reduce the quenching effect. Long welding times, however, reduce the life of the spot welding electrodes, and it is preferable to apply a second current impulse when the job has cooled down after welding and whilst it is still between the spot welding electrodes. In his paper* Gardner draws attention to the fact that this technique, which requires special control gear on the spot welding machine, cannot generally be used, however, for projection welding processes.

The table (Fig. 5) gives the material specifications, weldability, and recommendations for special welding procedure requirements in connection with some typical steels as used in fabricated welded constructions. It should be noted that high-alloy, high-tensile steels are not included in the table, as they are not very widely used in welded constructions, and not much information is available about the best procedure for welding them. Research work on the subject is still going on, and in special cases, where the use of high-alloy steels is essential, information may be found in the Reports of the various Committees of the Institute of Welding, London, and of the American Welding Society.

C. Requirements for Economic and Efficient Fabrication in the Shop

The choice of materials is not dictated by technical matters alone. The consideration of economical and efficient production is often the factor which decides the type and form of the material to be used. Complicated structures are usually more economically produced as

* H. St. G. Gardner, " System of Indicating Welding Requirements on Engineering Drawings," *Transactions of the Institute of Welding*, February 1945, page 38.

ITEM No	SPECIFICATION No	TITLE	DESCRIPTION	MATERIAL THICKNESS	MECHANICAL PROPERTIES: ULTIMATE TENSILE STRESS TONS/INCH²	YIELD STRESS TONS/INCH²	MINIMUM ELONGATION %	ANALYSIS 1): CARBON % MAX	SILICON % MAX	MANGANESE % MAX	NICKEL % MAX	CHROMIUM % MAX	SULPHUR % MAX	PHOSPHORUS % MAX	WELDING REQUIREMENTS: ARC WELDING: MIN SIZE OF FILLET WELDS	MIN SECTIONAL AREA OF FIRST RUN OF BUTT WELDS (INCH²)	PREHEATING	RESISTANCE WELDING: SPOT WELDING	PROJECTION WELDING	BUTT WELDING
1	BSS 13	STRUCTURAL STEEL FOR SHIPBUILDING	PLATES (OTHER THAN FOR COLD FLANGING) BARS AND SECTIONS.	1/8"-3/16" INCL 1/4"-5/16" INCL 3/8"-5/8" INCL 3/4"-1" INCL 1 1/16"-1 3/8" INCL 1 1/2" AND ABOVE	28-33 OR 26-32	— 2)	16 20	— 3)	— 2)	— 2)	— 2)	— 2)	·06	·06	1/8" 4) 3/16" 4) 1/4" 4) 3/8" 4) 1/2" 4) 3/4" 4)	NOT SPECIFIED	ONLY WHEN JOINING 2 PLATES OF VERY UNEQUAL THICKNESSES SHOULD THE THICKER PLATE BE PREHEATED.	NO SPECIAL PRECAUTIONS NECESSARY	NO SPECIAL PRECAUTIONS NECESSARY.	NO SPECIAL PRECAUTIONS NECESSARY.
			PLATES FOR COLD FLANGING	1/8"-3/16" INCL 1/4"-5/16" INCL 3/8"-5/8" INCL 3/4"-1" INCL 1 1/16"-1 3/8" INCL 1 1/2" AND ABOVE	26-30	— 2)	19 23	— 3)	— 2)	— 2)	— 2)	— 2)	·06	·06	1/8" 4) 3/16" 4) 1/4" 4) 3/8" 4) 1/2" 4) 3/4" 4)	NOT SPECIFIED	ONLY WHEN JOINING 2 PLATES OF VERY UNEQUAL THICKNESSES SHOULD THE THICKER PLATE BE PREHEATED	NO SPECIAL PRECAUTIONS NECESSARY.	NO SPECIAL PRECAUTIONS NECESSARY	NO SPECIAL PRECAUTIONS NECESSARY
2	BSS 14	STRUCTURAL STEEL FOR PRESSURE PARTS OF MARINE BOILERS.	PLATES FOR FLANGING OR WELDING AND FOR COMBUSTION CHAMBERS AND FURNACES	— 2)	26-30	— 2)	23	— 3)	— 2)	— 2)	— 2)	— 2)	·05	·05	AS FOR ITEM No 1	NOT SPECIFIED	AS FOR ITEM No 1	AS FOR ITEM No 1	AS FOR ITEM No 1.	AS FOR ITEM No 1
			ANGLE AND TEE BARS.	— 2)	28-32	— 2)	20	— 3)	— 2)	— 2)	— 2)	— 2)	·05	·05	AS FOR ITEM No 1	NOT SPECIFIED	—	AS ABOVE BUT LONGER WELDING TIMES	AS ABOVE BUT LONGER WELDING TIMES	AS ABOVE BUT LONGER WELDING TIMES.
3	BSS 15	STRUCTURAL STEEL FOR BRIDGES ETC. AND GENERAL BUILDING CONSTRUCTION 5)	PLATES SECTIONS AND FLAT BARS	1/8"-5/16" INCL 3/8" AND ABOVE	28-33	— 2)	16 20	— 3)	— 2)	— 2)	— 2)	— 2)	·06	·06	AS FOR ITEM No 1	NOT SPECIFIED	AS FOR ITEM No 1	AS FOR ITEM No 2.	AS FOR ITEM No. 2	AS FOR ITEM No 2
4	BSS 487	FUSION WELDED STEEL AIR RECEIVERS.	PLATES, ETC	1/8"-1/4" INCL 1/4" AND ABOVE	24-30 MAX RANGE 4 TONS/INCH² IN ANY ONE CASE	— 2)	20 23	— 3)	— 2)	— 2)	— 2)	— 2)	·05	·05	AS FOR ITEM No. 1	NOT SPECIFIED	AS FOR ITEM No. 1.	AS FOR ITEM No 1.	AS FOR ITEM No 1.	AS FOR ITEM No 1
5	BSS 1099	SMALL FUSION WELDED STEEL AIR RECEIVERS.	COLD ROLLED, CLOSE ANNEALED STEEL PLATES	·02"-·16"	20 MIN	— 2)	20	— 3)	— 2)	— 2)	— 2)	— 2)	·05	·05	AS FOR ITEM No 1	NOT SPECIFIED	AS FOR ITEM No 1	AS FOR ITEM No.1	AS FOR ITEM No 1	AS FOR ITEM No 1
6	BSS 537	LANCASHIRE AND CORNISH BOILERS.	PLATES FOR SHELLS, BUTT STRAPS, GUSSET STAYS.	— 2)	26-32 MAX RANGE 4 TONS/INCH² IN ANY ONE CASE	— 2)	23 FOR 26-30 TONS/INCH² 20 FOR 28-32	— 3)	— 2)	— 2)	— 2)	— 2)	·05	·05	AS FOR ITEM No.1	NOT SPECIFIED	AS FOR ITEM No. 1	AS FOR ITEM No 1	AS FOR ITEM No. 1	AS FOR ITEM No. 1
			PLATES FOR FLANGING OR WELDING AND FOR FURNACES (BOILER PLATE)	— 2)	24-30 MAX RANGE 4 TONS/INCH² IN ANY ONE CASE	— 2)	23	— 3)	— 2)	— 2)	— 2)	— 2)	·05	05	AS FOR ITEM No 1	NOT SPECIFIED	AS FOR ITEM No 1	AS FOR ITEM No. 1	AS FOR ITEM No 1	AS FOR ITEM No 1
			SECTIONS AND BARS.	— 2)	26-32 MAX RANGE 4 TONS/INCH² IN ANY ONE CASE	— 2)	23 FOR 26-30 TONS/INCH² 20 FOR 28-32 TONS/INCH²	— 3)	— 2)	— 2)	— 2)	— 2)	·05	·05	AS FOR ITEM No. 1	NOT SPECIFIED	AS FOR ITEM No. 1	AS FOR ITEM No 2	AS FOR ITEM No 2	AS FOR ITEM No. 2

7	BSS 968	HIGH TENSILE (FUSION WELDING QUALITY) STRUCTURAL STEEL FOR BRIDGES, ETC. AND GENERAL BUILDING CONSTRUCTN 5)	PLATES.	$\frac{1}{4}$" $\frac{3}{8}$" $\frac{1}{2}$" 1" ABOVE 1"	37-43 35·41 33-39	23 21 19	14 18	·23	·35	·18 6)	·5 (OPTIONAL)	1·0 6) (OPTIONAL)	·06	·06	$\frac{3}{32}$" 7) $\frac{1}{4}$" 7) $\frac{11}{32}$" 7) $\frac{7}{16}$" 7) —	·018 7) ·027 7) ·040 7) ·075 7) —	AS FOR ITEM No. 1.	LONGER WELDING TIMES THAN FOR MILD STL. REQD OR REHEATED AFTER WELDING. SPECIAL CONTROL GEAR 8)	LONGER WELDING TIMES THAN FOR MILD STL. REQUIRED 8)	PREHEATING AND POST HEATING ADVISABLE
			SECTIONS AND FLATS AND ROUND AND SQUARE BARS.	UP TO $\frac{3}{8}$" $\frac{3}{8}$" - $\frac{3}{4}$" INCL ABOVE $\frac{3}{4}$	35·41 33-39	21 19	14 18	·23	·35	1·8 6)	·5 (OPTIONAL)	1·0 6) (OPTIONAL)	·06	·06	AS ABOVE (FOR PLATES)	AS ABOVE (FOR PLATES)	—	AS ABOVE (FOR PLATES)	AS ABOVE (FOR PLATES)	AS ABOVE (FOR PLATES)
8	BSS 548	HIGH TENSILE STRUCTURAL STEEL FOR BRIDGES, ETC AND GENERAL BUILDING CONSTRUCTION 5)	PLATES SECTIONS AND FLAT BARS	$\frac{1}{4}$" $\frac{3}{8}$" $\frac{1}{2}$" 1" $1\frac{1}{4}$" ABOVE $1\frac{1}{4}$"	37-43	23 19-22	14 18	·30	— 2)	— 2)	— 2)	— 2)	·05	·05	— — $\frac{3}{8}$" 7) $\frac{1}{2}$" 7) — —	— — ·053 7) 083 7) — —	AS FOR ITEM No. 1 PLATE MUST BE PREHEATED TO 200° C OR AUSTENITIC STL. ELECTRODES SHOULD BE USED. 7)	REHEATING AFTER WELDING REQUIRED (SPECIAL CONTROL GEAR) 8)	NOT WELDABLE 8)	PREHEATING AND POST HEATING REQUIRED 8)
			ROUND AND SQUARE BARS	UP TO $\frac{1}{2}$" $\frac{1}{2}$"-1" INCL ABOVE 1"	37-43	23 19-22	18 22	30	— 2)	— 2)	— 2)	— 2)	·05	·05	AS ABOVE (FOR PLATES)	AS ABOVE (FOR PLATES)	AS ABOVE (FOR PLATES)	—	—	AS ABOVE (FOR PLATES.)
9	En. 8	·40 CARBON STEEL 10)	PLATES SECTIONS AND BARS.	$\frac{1}{4}$" $\frac{3}{8}$" $\frac{1}{2}$" 1"	35-45	17-20	20	·45	·35	1·5	1·0 (OPTIONAL)	— 2)	·06	·06	$\frac{3}{16}$" 9) $\frac{1}{4}$" 9) $\frac{5}{16}$" 9) $\frac{3}{8}$" 9)	·014 9) ·025 9) ·036 9) 053 9)	PREHEATING REQUIRED OR AUSTENITIC STEEL ELECTRODES TO BE USED	NOT WELDABLE 8)	NOT WELDABLE 8)	PREHEATING AND POST HEATING REQUIRED 8)
10	BSS 592	CARBON STEEL CASTINGS FOR SHIPS AND FOR MARINE ENGINE AND GENERAL ENGINEERING PURPOSES	28-35 TON GRADE	—	28-35	14	20	·25 11)	— 2)	— 2)	— 2)	— 2)	·06	·06	—	—	PREHEATING OFTEN ADVISABLE	—	—	—

FIG. 5.—Typical Materials used in Welded Steel Construction.

For notes on Table see next page.

castings than built up as fabrications, and designers will often be unable to avoid small complicated portions in machine structures which in other respects could conveniently be fabricated. In such cases it is advantageous to use a compound method of construction and to weld steel castings into the fabricated structure.

Fig. 6 (Plate I) shows a marine reduction gear-case of fabricated construction in which the bearing housings and the thrust-block for the propeller shaft are steel castings. Factors, such as the need for numerous bosses for connecting lubrication piping and the special wall shapes which are required because of tenon-groves, are complications which make a fabrication less economical than a casting.

Sometimes even simple and plain-shaped component parts like the bearing housing in the milling machine headstock (*a*, Fig. 7) are not an economical proposition for fabrication. Here, again, the use of a steel casting is the best solution, since a fabricated tubular section of the

NOTES ON FIG. 5

(1) No molybdenum content specified in the respective specifications for any of the materials in this list.
(2) Not specified in B.S.S.
(3) Not specified in B.S.S., but usually supplied between ·12 and ·23 per cent.
(4) Quoted from C. H. Jennings, " Welding Design ", *Transactions A.S.M.E.*, October 1936.
The *Tentative Code for Arc and Gas Welding in Building Construction* of the American Welding Society (1941) specifies smaller weld sizes, as follows

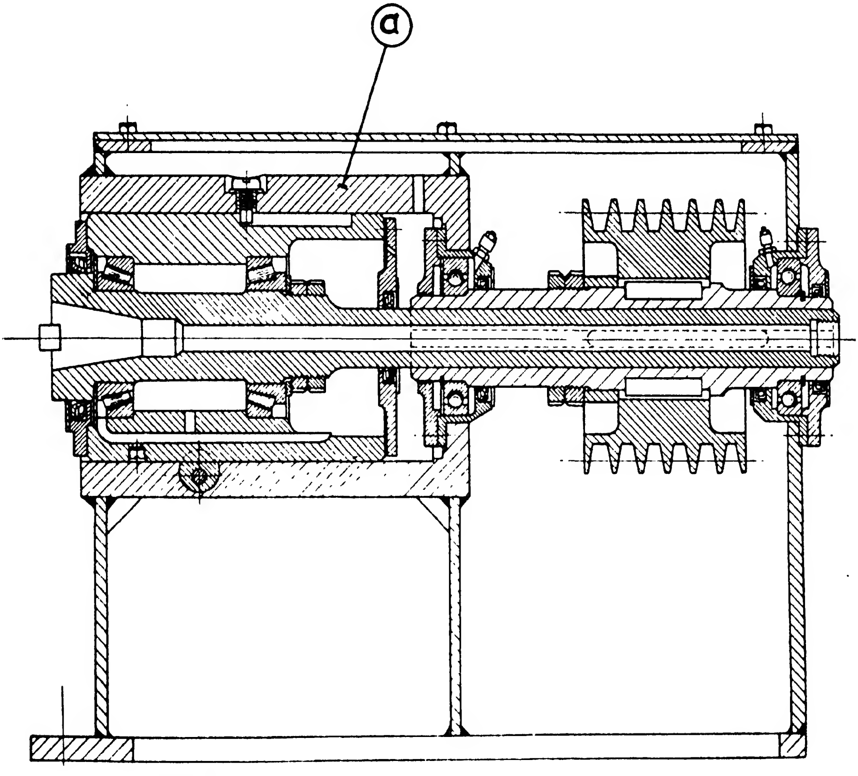
Size of Fillet Weld	*Maximum Thickness of Part*
$\frac{3}{16}''$	$\frac{1}{2}''$
$\frac{1}{4}''$	$\frac{3}{4}''$
$\frac{5}{16}''$	$1\frac{1}{4}''$
$\frac{3}{8}''$	2″
$\frac{1}{2}''$	6″
$\frac{5}{8}''$	Over 6″

(5) This steel is also used for general engineering purposes.
(6) Manganese and chromium together not to exceed 2·0 per cent.
(7) Sizes given are those for first run. " Where fillet sizes in excess of the above are required and a multi-run weld is found necessary, the remaining runs should be put down with the largest possible size of electrode and as soon after depositing the first run as is practicable." (Quoted from *The Arc Welding of High Tensile (Low Alloy) Structural Steels*, Memorandum T.2, published by the Welding Research Council of the Institute of Welding.)
(8) From H. St. G. Gardner, " System of Indicating Welding Requirements on Drawings ", *Transactions of the Institute of Welding*, February 1945, page 38.
(9) Sizes given for chromium-nickel austenitic steel electrodes. " When austenitic steel electrodes are used, the whole weld must be made with these electrodes. It is not permitted to deposit mild steel or high tensile steel weld metal over austenitic steel weld metal, as the resultant weld will contain cracks." (Quoted from *The Arc Welding of High Tensile (Low Alloy) Structural Steels*, Memorandum T.2, published by the Welding Research Council of the Institute of Welding.)
(10) Suitable for flame hardening.
(11) Maximum carbon content is not specified in B.S.S. The percentage as quoted here is a recommended maximum from the point of view of weldability.

size required in this particular case would be out of the question, and the production of a forging and its subsequent machining would be too expensive.

The material used for steel castings should be carefully specified in order to avoid difficulties of weldability (see Fig. 5).

Often, in the case of large-quantity production, the use of drop



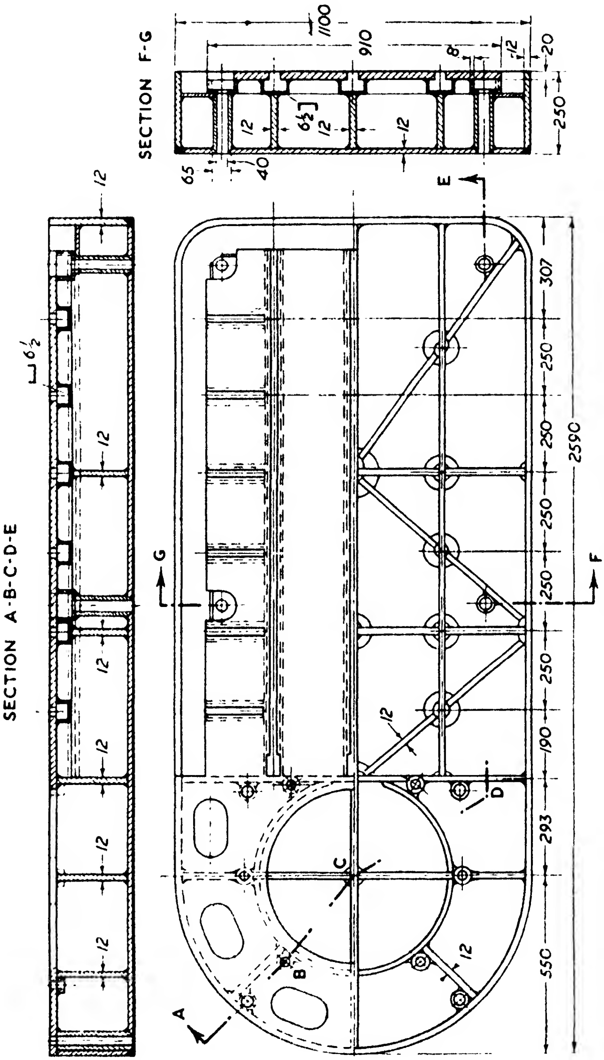
Milling Machine Headstock (Cooke & Ferguson, Ltd., Manchester)

FIG. 7

forgings, applied in a similar way to the steel castings in the previous examples, will be advantageous.

The use of rolled sections is always worth considering, as a certain amount of welding may be eliminated. An example of the application of rolled channel sections, which uses not only their strength but also their shape, is shown in the building up of Tee-slots in the base of a radial drill (Fig. 8) (Elin, Vienna).

The application of rolled standard sections, considered with due regard to the possibilities offered by the welding process as compared with that of riveting, may lead to valuable savings in material. Hill* compares two alternative schemes for a stanchion, one built from two



Dimensions in M.M

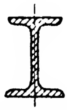
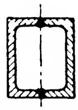
FIG. 8

standard channels which are welded together, the other a normal standard I section (Fig. 9). The stanchion built from two standard

* H. V. Hill, " Developments in the Design of Welded Steel Structures," *The Structural Engineer*, August 1945, page 357.

channels has about 120 per cent higher load-carrying capacity for equal weight, or at equal load-carrying capacity shows a saving in material of about 36 per cent (Fig. 9).

In this connection it should be pointed out that the rolled sections as standardized in the *British Standard Specifications* have been designed for riveted and not for welded construction, and are therefore not the ideal solution for use in connection with the latter. This point has been

SECTION	 6"x5"x25 lbs./Ft. R.S.J.	 2-6"x3"x12·41 lbs/Ft. Channels	 2-4"x2"x7·91 lbs/Ft. Channels
Weight Per Foot W	25 lbs/Ft.	24·82 lbs/Ft.	15·82 lbs/Ft.
Minimum Moment Of Inertia I	9·1 Inch^4	38·1 Inch^4	10·9 Inch^4
Section Area A	7·37 Inch^2	7·30 Inch^2	4·66 Inch^2
Least Radius Of Gyration k	1·11 Inch	2·28 Inch	1·52 Inch
Ratio $\frac{l}{k}$	130	63	95
※ Permissible Working Stress f	2·58 Tons/Inch^2	5·75 Tons/Inch^2	4·06 Tons/Inch^2
Permissible Load W_d	19·0 Tons	42·0 Tons	19·0 Tons

※ B.S.S. 449

Stanchion (effective length $l = 12$ ft.)

FIG. 9

discussed by Hill,* Braithwaite,† and others. Fig. 10, taken from Hill's paper, shows the difficulties created by the taper and the radius at the toe of the rolled standard sections which are available to-day in accordance with B.S.S. 4 and 6. It is shown that, for butt-welding two joists end to end, four cuts have to be made at the end of each joist.

The maximum possible fillet size which can be applied to the toe of a section is controlled by the toe radius (Fig. 11).

The advantages of having standard rolled sections with flanges which are not tapered and which have square ends are obvious.

Much could be gained by rolling standard sections with chamfers where butt welds are likely to be required. For building up broad-flanged joist sections Braithwaite suggests that the web end of the Tee should be rolled with a chamfer, thus avoiding the necessity for weld

* H. V. Hill, " Developments in the Design of Welded Steel Structures ", *The Structural Engineer*, August 1945, page 357.

† R. G. Braithwaite, " Application of Welding to Steel Structures ", *Transactions of the Institute of Welding*, May 1945, page 63.

preparation (Fig. 12).* For the same purpose other special flange sections have been produced† (Figs. 15-18), the design of which is

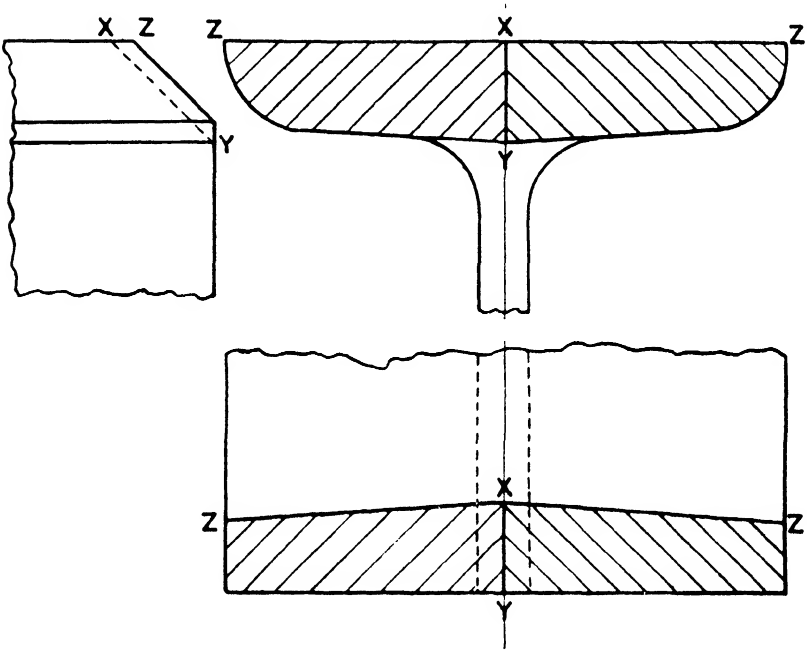
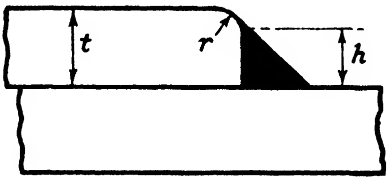


FIG. 10

based on the following principles. Whilst the usual design (Fig. 13) or the improved version (Fig. 14) use standard flats, the sections (Figs.



r	h
t	·40t
·9t	·46t
·8t	·52t
·7t	·58t
·6t	·64t
·5t	·70t
·4t	·76t
·3t	·82t

FIG. 11

15 and 16) give better continuity between flange, web, and connecting weld. The section (Fig. 17) moves the weld into a region of lower

* See R. G. Braithwaite, " Application of Welding to Steel Structures ", *Transactions of the Institute of Welding*, May 1945, page 63.

† K. Schaechterle, "General Considerations on Welding", *Proceedings of the 2nd International Congress for Bridge and Structural Engineering*, 1936.

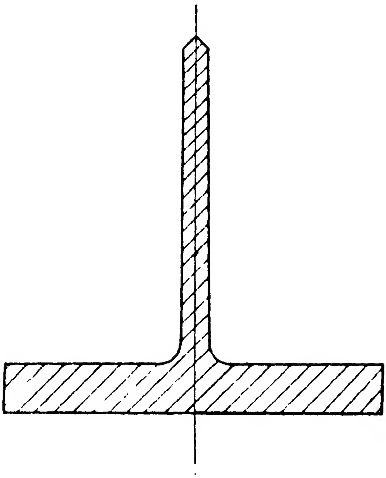


FIG. 12

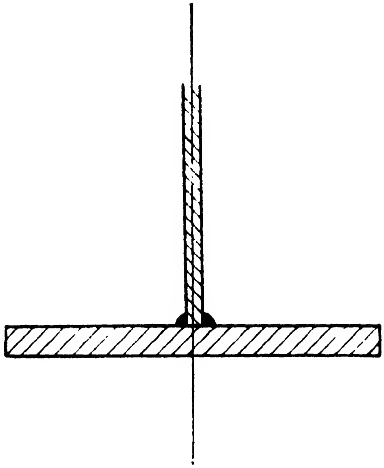


FIG. 13

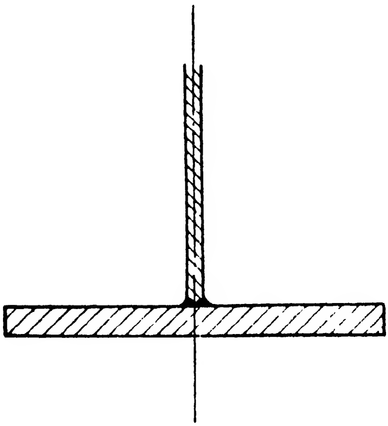


FIG. 14

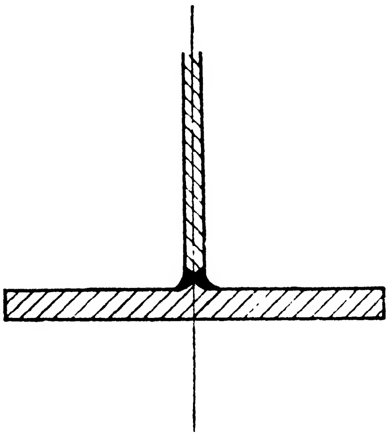


FIG. 15

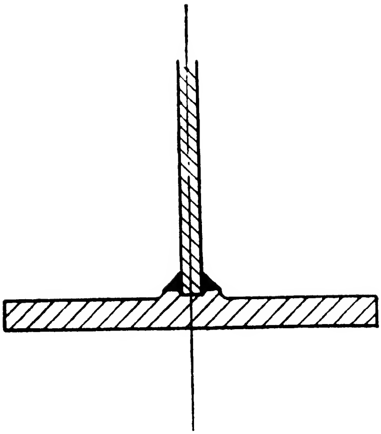


FIG. 16

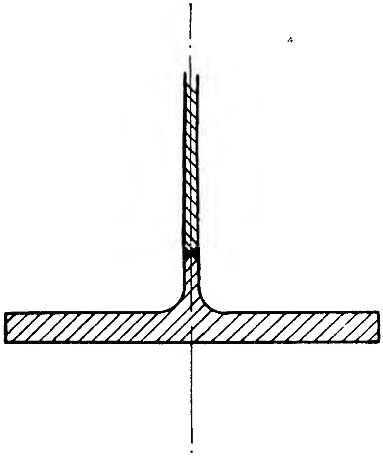


FIG. 17

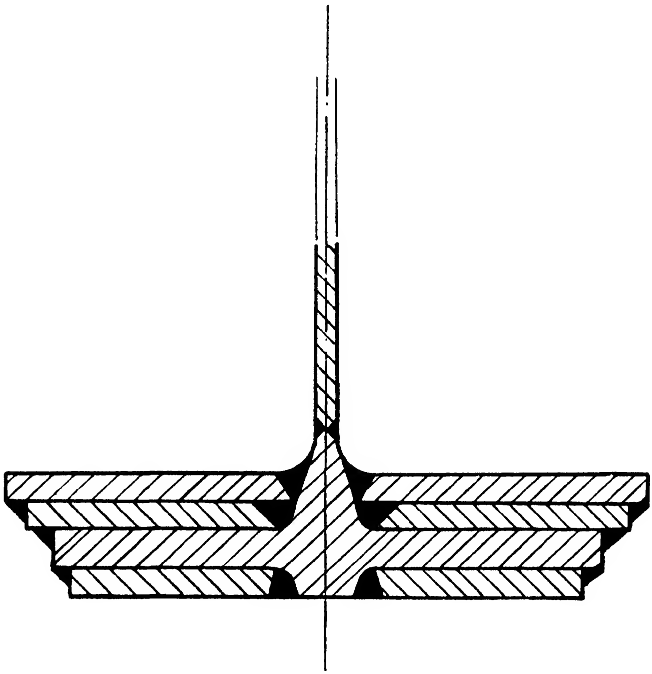


FIG. 18

stress than is the case for sections (Figs. 13, 14, 15, 16). More uniform transition between flange and web is obtained, notching effects in the flange are avoided, and the danger of flange distortion is reduced by moving away from the flange the zone which is heated during the welding process.

As the weld is removed farther from the flange it becomes more easily accessible for examination, *e.g.* by means of X-rays.

For particularly heavy structures, which require heavy-flanged joist sections, the design Fig. 18 is suggested.

It is often said that spot welding is similar to riveting from a designer's point of view. Although this similarity applies to the strength calculations for the weld itself as compared with that for rivets, the present rolled standard sections are not suitable for spot welding, due to the sloping surfaces of the flanges, which would necessitate specially shaped electrodes.

The above points show the importance of carrying out—in collaboration with the rolling mills—the standardization of rolled sections which are suitable for use in connection with the main welding processes.

When specifying materials for a particular job, consideration must also be given to the state in which the material is to be supplied to the welding shop. Whilst slight rust, mill scale, etc. on the material may not affect to a considerable degree the efficiency of arc welds, lack of cleanness will be fatal to resistance-welded joints. Pickling or shot blasting immediately prior to the resistance welding operation is essential for making this latter method a success.

Material up to $\frac{3}{16}''$ thickness which is to be used on resistance-welded jobs is usually purchased in a pickled and slightly oiled condition, and should be carefully stored in order to keep it clean. It can then be used without even removing the thin film of oil, as long as the oil is clean.

Material above $\frac{3}{16}''$ thickness should be shot-blasted before being taken to the resistance welding machines. No long delay should occur between the shot blasting and the resistance welding operations, in order to avoid new corrosion which might eliminate the advantage gained by the shot-blasting treatment. Sand-blasting is not recommended, as particles of the siliceous material may be embedded on the steel surface and may influence its electrical resistance.

Chapter 2

THE WELDED JOINTS

THE connecting element between the components of a fabricated welded structure is the weld, and its functions are :

(1) The location of the components in relation to each other, and
(2) The transmission of loads between the components.

It is obviously inefficient to use the welding method if a designer distrusts the weld and uses it for location purposes only. Designers must get used to the idea that, with modern welding equipment and with modern means of inspection and testing, the trustworthiness of welds is on an equality with that of any other means of load transmission.

The weld fulfils the same functions as other basic elements of construction such as screws, bolts, or rivets, but is different from these elements in that while they are ready and completed before they are used in the assembly shop, and can be taken from stock with their mechanical properties and load-carrying capacities guaranteed beforehand, the welded joint is produced on the particular job to which it belongs. Its production is part of the manufacturing process in the assembly shop, and it has, in fact, to be designed for each particular job.

In order to decide upon size, shape, and type of a welded joint it is necessary to know :

(*a*) The ultimate and permissible safe stresses for the various types of welds.
(*b*) The load to be transmitted by each weld, in order to calculate the stresses in the weld and to determine for each particular case the required type and size of weld.

It must be appreciated, however, that it is not only the size and shape of the weld which determine its mechanical properties, but also the procedure used to produce it. For example, to obtain good welds conditions of accessibility must be favourable and an endeavour should be made to enable the welder to work in the down-hand position. Points such as these contribute to the obtaining of a weld the mechanical properties of which are as close as possible to those which can theoretically be expected.

As on a bolted structure the materials, dimensions, and forms of the bolts have to be specified, so the size, shape, and welding procedure for the various welds must be indicated on the drawing for a welded construction, and no doubts must exist or any decisions be left to the judgment of the workman in the shop.

The welds generally used in fabricated constructions can be classified as follows :

A—WELDS PRODUCED BY THE OXY-ACETYLENE OR THE ARC WELDING PROCESS.

I. Butt welds (Fig. 19).
II. Fillet welds (Fig. 20).

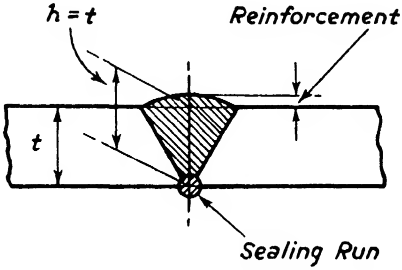


FIG. 19

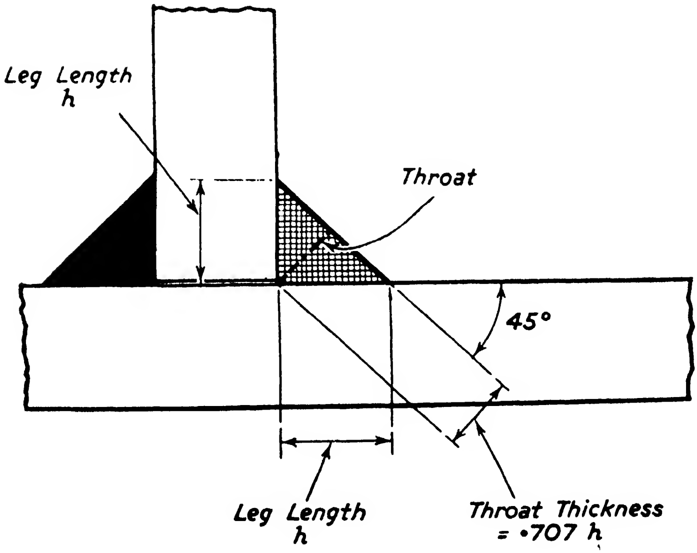


FIG. 20

B—WELDS PRODUCED BY RESISTANCE WELDING PROCESSES.

I. Spot welds (Fig. 21).
II. Projection welds (Fig. 22).
III. Butt welds (Fig. 23).
IV. Flash Butt welds (Fig. 24).

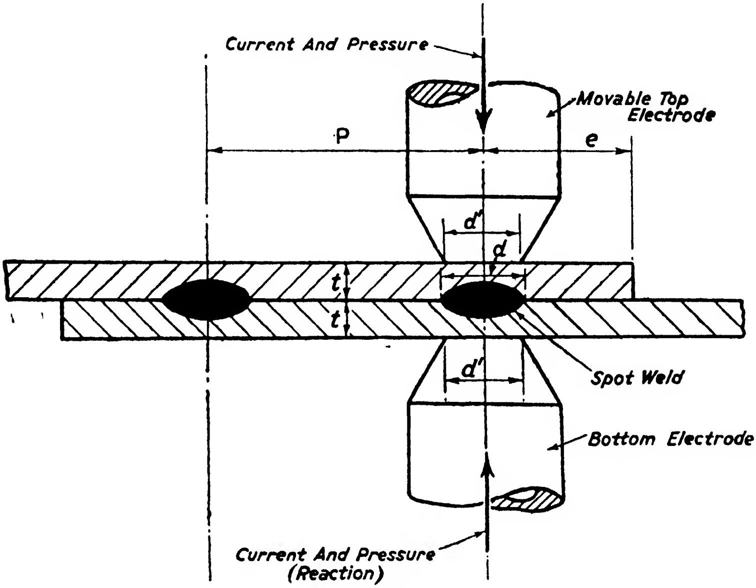


FIG. 21

In constructions which involve the heavier sizes of plates, the electric-arc welding process is generally used. In many cases resistance welding is very efficient, whilst oxy-acetylene welding is mainly used on work on light-gauge sheet metal, and particularly light alloys.

In the following chapters, arc-welded and resistance-welded joints will be considered in detail. Strength considerations of oxy-acetylene welded joints are similar to those produced by the arc-welding process.

A. Arc-Welded Joints

Some general considerations regarding the application of butt* and fillet welds are as follows :

Butt welds, with the exception of those used for joining thin plates

* The butt weld (Fig. 19) produced by the arc-welding process must not be confused with the weld produced by the butt-welding (resistance) process (Figs. 23 and 24).

or those produced by means of deep penetration electrodes, require plate preparation by either bevelling or gouging. This is often an additional operation after cutting a plate to the required shape. It can be carried out by flame cutting, flame gouging, or machining (milling or

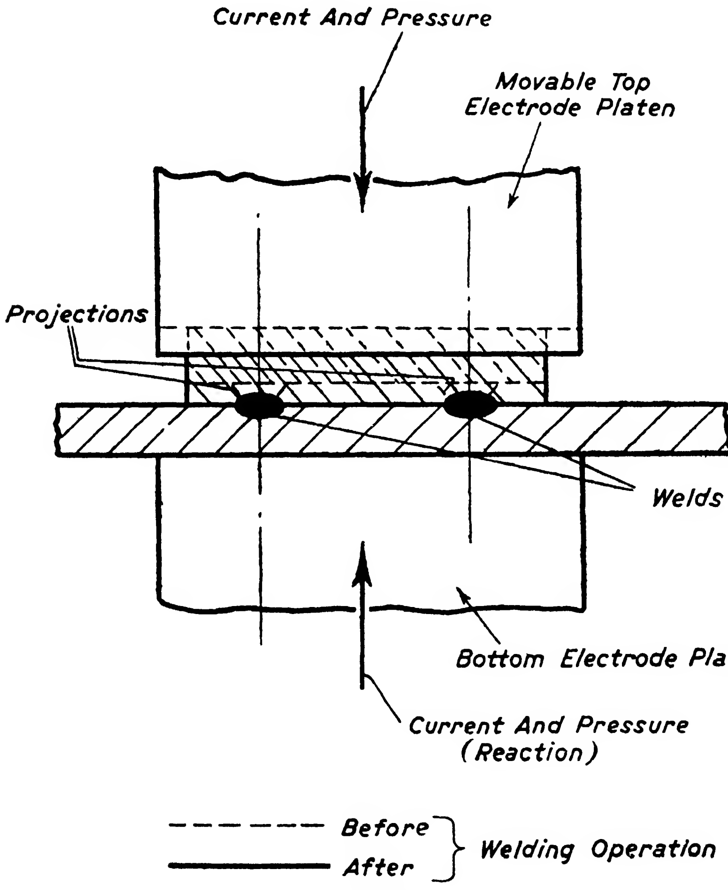


FIG. 22

planing). Usually no plate-edge preparation is required for the execution of fillet welds. It will be obvious, therefore, that, in general, joints using fillet welds are produced more economically than those using butt welds.

The more elaborate the required plate preparation the greater the danger of errors and inaccuracies. Inaccurate plate preparation, resulting in gaps between the plates, increases the amount of weld metal to be deposited to fill the gaps, and, therefore, increases the welding costs. It also favours the creation of internal stresses due to the higher heat input, and may even affect the effective strength of the weld.

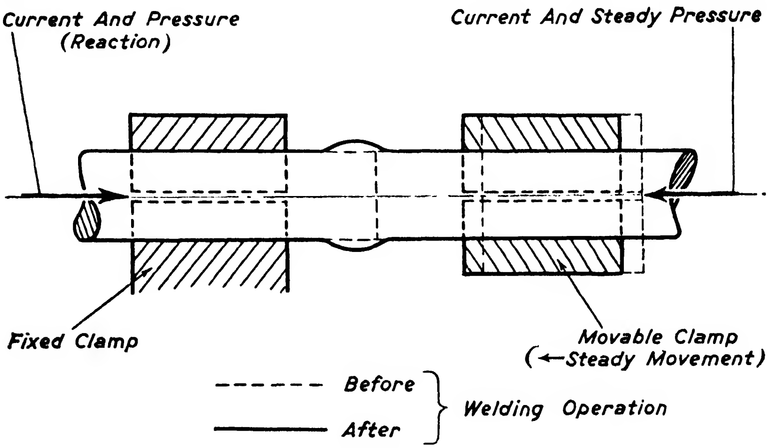


FIG. 23

With regard to the last point, it must be remembered that, in the case of a prepared butt weld joining two plates, a gap caused by bad plate preparation is bound to be filled with weld metal and the danger of an invisible reduction of the weld section is very remote. This statement

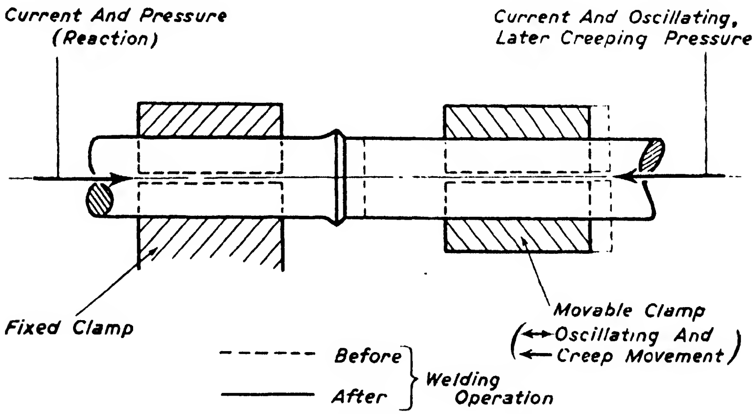


FIG. 24

only affects the actual weld size, it does not concern the problem of slag inclusions or blow holes. If, however, a gap exists between two abutting plates which are joined by fillet welds, this might remain nvisible, due to its being covered by the welds, and the effective strength of the weld itself will be reduced. A check-up as to whether or not the required effective weld size has been obtained (see Fig. 223) will only be possible by means of a destructive test or, if practicable, by an X-ray inspection. These points will be discussed in detail in connection with the problem of tolerances for plate preparation (see page 187).

The shape of fillet welds favours the creation of stress concentrations either at the toe or at the root of the weld. This point has to be taken into consideration particularly if the welds are subjected to dynamic loads. In butt welds the danger of stress concentrations is less pronounced, especially if reinforcements are removed ; on the other hand, it has been found that, in general, butt welds produce greater residual stresses.

These general considerations have to be kept in mind for guidance only. The final decision on the most suitable type of weld must be taken individually for each particular application, and after considering the general lay-out and the loads to be transmitted by the weld.

The welded joint must be designed in such a way as to keep the maximum working stress in the weld below the permissible value, this value allowing a satisfactory margin of safety between actual and ultimate working stresses. It is not sufficient, however, to base the design on a standard factor of safety alone. The permissible working stresses in welds depend upon a variety of factors :

(1) The loading conditions (static, dynamic, alternating, pulsating).
(2) The type of weld (butt weld, fillet weld).
(3) The shape of the weld (concave or convex weld, the existence of undercuts, and the blending of the weld surface with that of the parent plate).
(4) The surface of the weld (as welded, hand ground, or machined).
(5) The method of producing the weld, and the nature of any treatment applied afterwards (welding procedure, peening, heat treatment).

Although in the case of static loading the ultimate tensile stresses of the weld metal are close to those found for the corresponding parent metal, conditions are different in the case of dynamic or alternating stresses.

Photo-elastic investigations have shown that stress concentrations do occur in welds, for example at the heel and the toe of fillet welds, where stresses may be high even when the load is relatively small. As the load increases, the yield point will soon be reached at the point of stress concentration, where slight yielding will occur, thus transferring the maximum stress to another part of the weld. This process continues until the stress distribution over the whole load-carrying section is fairly uniform. Such balancing of the stress distribution takes place under static loads, and, for the purpose of stress calculations, does not considerably affect the final stress in the weld.

In the case of alternating stresses, however, sufficient time is not available for redistribution and ultimate balancing of stresses, and fractures occur under lower loads. It has been suggested* that stress-concentration factors should be used when calculating the maximum stresses in welds which are subjected to alternating loads. These stress concentration factors are intended to be used in the following way :

The stress in the weld is first calculated on the assumption that the load is static and is then multiplied by the stress concentration factor. The resulting stress must be below the value which is permissible for static loading.

The stress concentration factors suggested by C. H. Jennings† are :

Reinforced butt weld	1·2
Toe of transverse fillet weld	1·5
End of parallel fillet weld	2·7
T-butt joint with sharp corners	2·0

These are, of course, only average figures and the value selected must depend to a great extent upon the shape and the surface finish of the weld.

As an example, stress concentrations in a convex-fillet weld will be higher than those in a straight-fillet weld, and higher still than those in a concave-fillet weld the shape of which is blended in with the surfaces of the parent plates. Convex fillet-welds, however, have a greater throat section than the two other types of fillet welds. They will, therefore, be the strongest of the three under static loads, whilst concave-fillet welds will be capable of carrying higher alternating loads.

For the same reason, reinforced butt welds, subjected to static loads, will stand higher stresses than butt welds which are machined flush with the parent plate. The latter, on the other hand, will stand higher stresses when subjected to alternating loads.

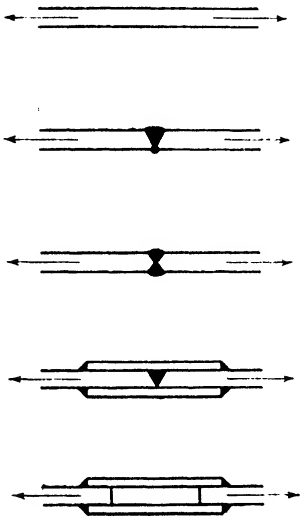
From the above it will be obvious that the permissible stresses under particular loading conditions should be determined by considerations based on the knowledge of the properties and the ultimate breaking stress of the form of weld in question. These must be found experimentally, therefore, under stressing conditions which are equivalent to those which will occur in the actual structure under working conditions.

Figs. 25 and 26 show the ultimate stresses for various cases of alternating and pulsating loads (2,000,000 cycles or more), as found by

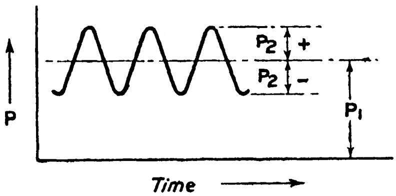
* See C. H. Jennings, " Welding Design ", *Transactions of the A.S.M.E.*, October 1936, page 497.

† *Ibid.*

different research workers and summarized by H. N. Pemberton in his paper " The Testing of Welds ".*



	Stress under Pre-load P_1 (Tons/Inch²)	Stress under Load P_2 (Tons/Inch²)	Remarks	Authority
(a)	+ 11·5	±5·1	$P_1 > P_2$	Thum
(b)	0	±5·5	$P_1 = 0$	Haigh
	+ 6·0	±4·0	$P_1 > P_2$	Haigh
	+ 11·5	±2·9	$P_1 > P_2$	Thum
	− 3·0	±8·0	$P_1 < P_2$	Haigh
	− 5·0	±6·0	$P_1 < P_2$	Haigh
	+ 6·0	±8·2	$P_1 < P_2$ Stress relieved	Haigh
(c)	+ 11·5	±2·5	$P_1 > P_2$	Thum
(d)	+ 11·5	±2·0	$P_1 > P_2$	Thum
(e)	+ 11·5	±1·0	$P_1 > P_2$	Thum



Alternating Load with or without Preload

FIG. 25

Examples of the appearance of weld failures under static and dynamic loads are shown on Figs. 27 to 30 (Plates I and II) : †

Fig. 27 (Plate I) shows the failure under static load of a butt-welded sample (material failed at 24 tons/sq. inch ; weld strength, 27 tons/sq. inch).

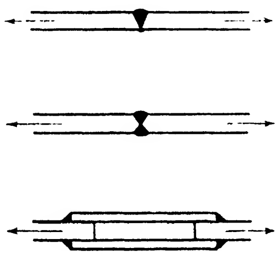
* *Transactions of the North East Coast Institution of Engineers and Shipbuilders*, vol. lviii, pages 205-40.

† M. Ros, *Die Festigkeit und Sicherheit der Schweissverbindungen* (*The Strength and Safety of Welded Joints*), Eidgenössische Materialprüfungs- und Versuchsanstalt für Industrie, Bauwesen und Gewerbe, Zürich, Bericht No. 156.

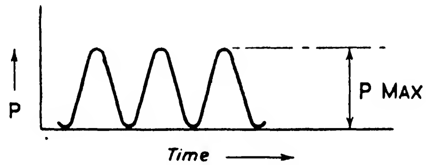
Fig. 28 (Plate II) shows the fatigue failure of a butt weld, the failing stress being 13·5 tons/sq. inch. The fracture started from a blow hole in the weld.

Fig. 29 (Plate II) shows the failure under static load of a fillet-welded T-connection, the failing stress being 28 tons/sq. inch.

Fig. 30 (Plate II) shows the fatigue failure of a fillet-welded T-connection, the failing stress being 6·4 tons/sq. inch. The white arrow shows the start of the failure.



	STRESS UNDER LOAD P MAX TONS/INCH²	REMARKS	AUTHORITY
(a)	8·3	MEAN RESULT	ROSS & EICHINGER
(b)	8·0	1" SQUARE SPECIMEN	HANKINS & THORPE
(c)	7·0		HANKINS & THORPE. GRAF



Pulsating Load

Fig. 26

Surface finish and treatment of welds have an influence upon the ultimate stresses of butt- and fillet-welded joints. Figs. 31 and 32* show the fatigue strength of butt- and fillet-welded joints under alternating bending stresses after the welds have been subjected to various treatments as follows :

(1) *Butt weld joints* (Fig. 31) :
  (*a*) Untreated.
  (*b*) Undercut superficially milled.†

* A. Thum and A. Erker, " Dauerfestigkeit von Kehl- und Stumpfnahtverbindungen " (" Fatigue Strength of Fillet and Butt Welded Joints "), *Z.d.V.D.I*, Sept. 1938.

† " Undercut " is a groove formed by melting away some parent metal and thus reducing the thickness of the parent plate at the boundary of the weld.

(*c*) Undercut smoothly milled. (Stress at cross-section of fracture.)
(*d*) Undercut superficially milled and cold peened.
(*e*) Undercut rolled and undercut cold peened.
(*f*) Undercut and weld reinforcement machined flush.
(*g*) Normalized, weld forged. (Stress at cross-section of fracture.)

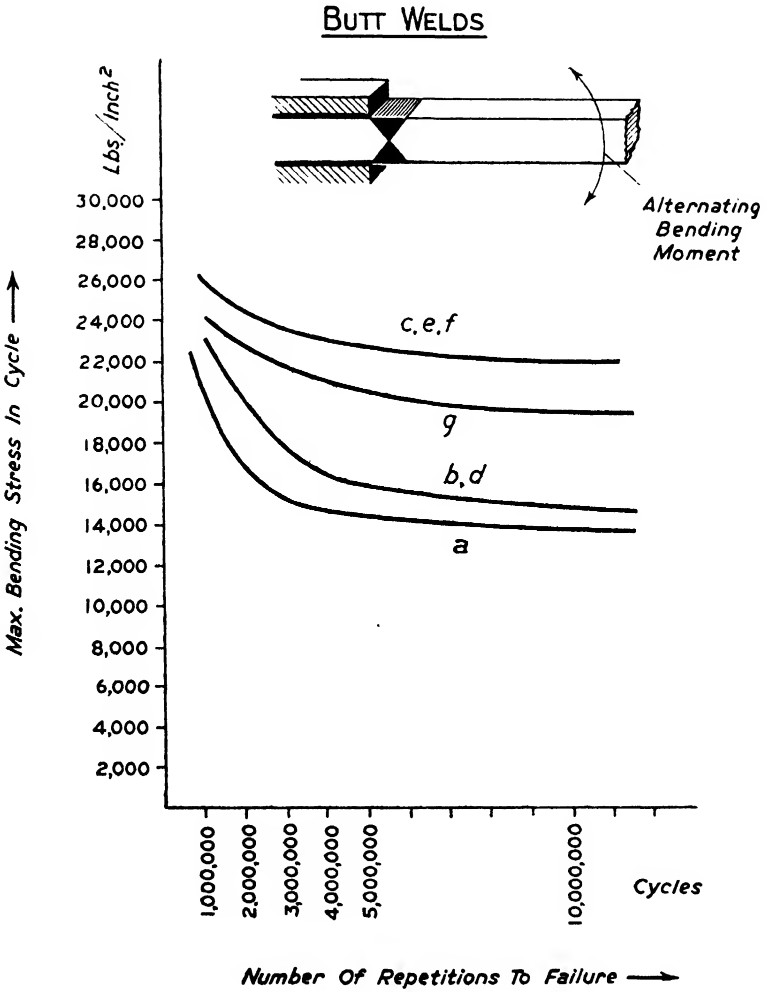


FIG. 31

(2) *Fillet weld joints* (Fig. 32) :

(*a*) Untreated.
(*b*) Undercut superficially milled.

(*c*) Undercut smoothly milled.
(*d*) Undercut superficially milled and cold peened.

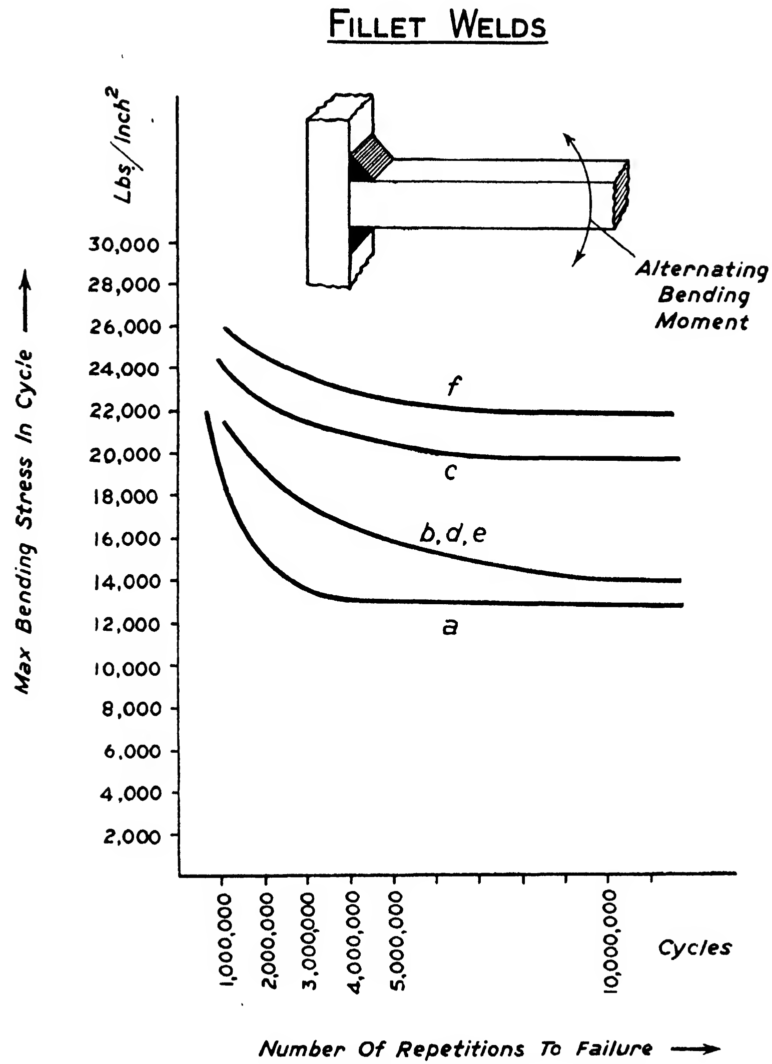


FIG. 32

(*e*) Undercut smoothly milled and cold peened.
(*f*) Undercut rolled. (Stress at cross-section of fracture.)

Curves *c*, *e*, *f* (Fig. 31) and *f* (Fig. 32) show the fatigue strength of the untreated base metal with mill scale, curve *g* (Fig. 31) of normalized base metal, because fracture did not occur in the weld or undercut zone.

The influence of the undercut upon the fatigue strength is clearly shown by the curves *a* and *c* in Figs. 31 and 32. With an undercut as resulting from the welding operation (curves *a*) the fatigue strength for 10,000,000 cycles is about 13,800 lb./sq. inch for the butt weld and about 13,000 lb./sq. inch for the fillet weld. By means of blending operations (milling, grinding, or cold rolling with a specially developed apparatus) the fatigue strength has been increased up to about 22,000 lb./sq. inch.

In view of the many different stress values in welds subject to alternating loads published by various research workers, the American Welding Research Council of the Engineering Foundation have carried out research work in connection with ultimate fatigue stresses for various types of welded joints.* As the result of this work, so called " dependable stresses " have been established. These can be assumed to apply in all cases of welded joints which are produced under normal workshop conditions with the care usually required for a high-quality welding job.

With the knowledge of the dependable ultimate stress of a particular welded joint under consideration, a designer can now determine the stress which he considers permissible. Figs. 33 to 35 show the dependable fatigue stresses in relation to the number of stress cycles, as published in the above Reports.

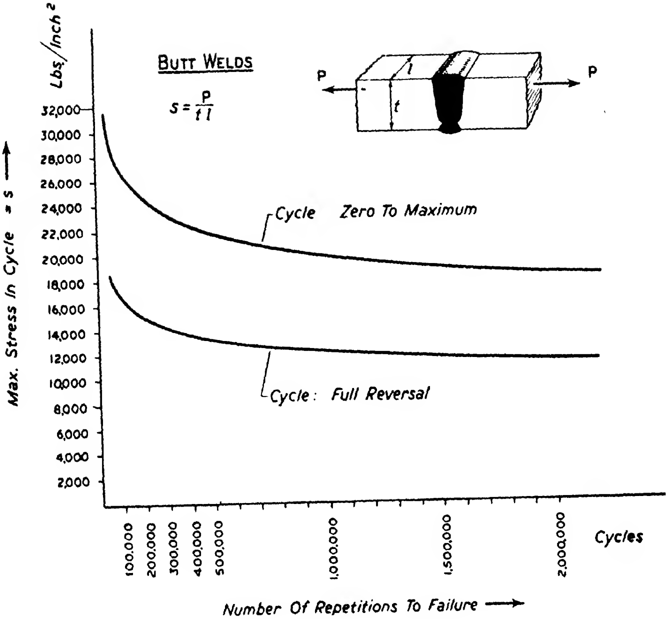
The values established for butt welds in tension (Fig. 33) are applicable to hand-made welds as well as those produced by automatic processes. In the case of fillet welds in shear (Fig. 34) the best results have been obtained by using a fillet across the free end continued into fillets along each side of the narrower connected part.

In fillet-welded Tee-joints (Fig. 35) a bending moment acts on the weld throat apart from the shear force. The magnitude of this moment depends upon the size of the weld itself, as this determines the leverage.

In general practice the stress calculation of fillet-welded Tee-joints involves only the shear action and neglects the bending moment (see page 59). This assumption was also made when the above-mentioned research results were explored, and it is obvious, therefore, that the dependable fatigue stresses obtained by this calculation (Fig. 35) are lower than those obtained for the case of pure shear (Fig. 34).

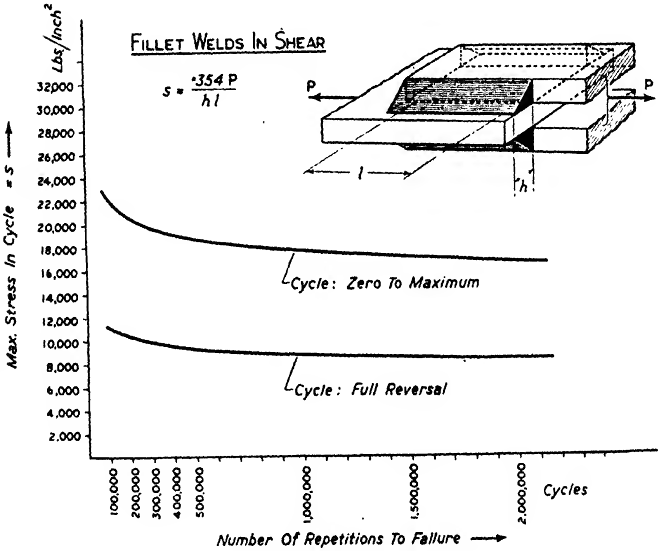
Permissible working stresses for dynamic loading conditions have not yet been established, but B.S.S. 538 specifies the following permissible stresses for static loading conditions of welds used on mild

* *Fatigue Strength of Butt Welds in Ordinary Bridge Steel*, Report No. 3 of Committee on Fatigue Testing (Structural) ; *Fatigue Strength of Fillet, Plug, and Slot Welds in Ordinary Bridge Steel*, Report No. 4 of Committee on Fatigue Testing (Structural).



Butt Welds
$s = \frac{P}{t\,l}$
P
P
t
l
Cycle Zero To Maximum
Cycle: Full Reversal
Lbs/Inch²
Max. Stress In Cycle = s
32,000
30,000
28,000
26,000
24,000
22,000
20,000
18,000
16,000
14,000
12,000
10,000
8,000
6,000
4,000
2,000
100,000
200,000
300,000
400,000
500,000
1,000,000
1,500,000
2,000,000
Cycles
Number Of Repetitions To Failure

FIG. 33



Fillet Welds In Shear
$s = \frac{.354\,P}{h\,l}$
P
P
l
h
Cycle: Zero To Maximum
Cycle: Full Reversal
Lbs/Inch²
Max. Stress In Cycle = s
32,000
30,000
28,000
26,000
24,000
22,000
20,000
18,000
16,000
14,000
12,000
10,000
8,000
6,000
4,000
2,000
100,000
200,000
300,000
400,000
500,000
1,000,000
1,500,000
2,000,000
Cycles
Number Of Repetitions To Failure

FIG. 34

steel (28/32 tons/sq. inch).* In these values it has been assumed that under static loading conditions the ultimate tensile strength of the weld metal equals that of the parent metal within close limits, that the shape of fillet welds creates less favourable conditions (see page 31) and that the stress distribution in fillet welds is less uniform than is the case for butt welds.

(*a*) *Butt welds :*

Permissible tensile and compression stress : 8 tons/sq. inch.
Permissible shear stress : 5 tons/sq. inch.

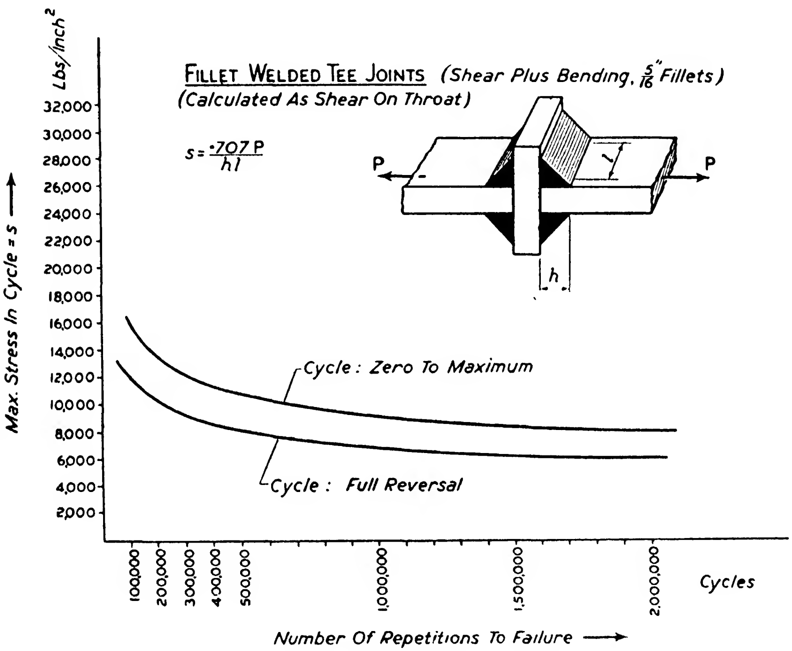


FIG. 35

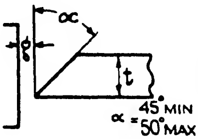
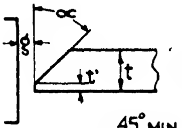
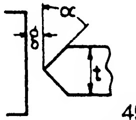
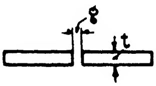
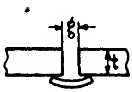
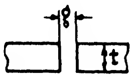
(*b*) *Fillet welds:*

End fillets in lap joints : 7 tons/sq. inch.
All other fillet welds : 5 tons/sq. inch.

(*c*) For J- or bevel-butt welds (see Figs. 36 and 37), either single or double, only 75 per cent of the stresses mentioned under (*a*) are permissible.

Plate preparations for various types of butt welds are also specified in B.S.S. 538 (Figs. 36-37). Butt welds without preparation are only

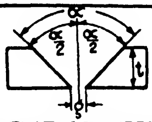
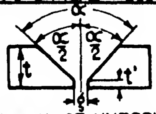
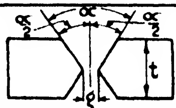
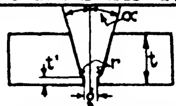
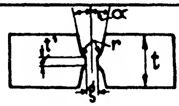
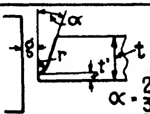
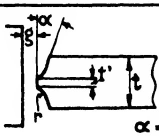
* From B.S.S. 538, *Metal Arc Welding in Mild Steel as applied to General Building Construction*, quoted by permission of the British Standards Institution, 28 Victoria Street, London, S.W.1, from whom official copies of the specification can be obtained, price 2s. post free.

TYPE OF BUTT WELD.	DETAILS OF WELD	PLATE THICKNESS t	ROOT FACE t'	GAP g
		IN	IN	IN
SINGLE BEVEL...	 $\alpha = 45°$ MIN $50°$ MAX	NOT MORE THAN $\frac{3}{8}''$ OVER $\frac{3}{8}''$	— —	$\frac{1}{8}''$ MIN. $\frac{1}{4}''$ MAX $\frac{3}{16}''$ MIN. $\frac{5}{16}''$ MAX
SINGLE BEVEL...	 $\alpha = 45°$ MIN. $50°$ MAX	NOT MORE THAN $\frac{3}{8}''$ OVER $\frac{3}{8}''$	$\frac{1}{16}''$ MAX $\frac{1}{8}''$ MAX	$\frac{1}{8}''$ MIN $\frac{1}{4}''$ MAX $\frac{3}{16}''$ MIN $\frac{5}{16}''$ MAX.
DOUBLE BEVEL...	 $\alpha = 45°$ MIN $50°$ MAX	NOT MORE THAN $\frac{3}{8}$ OVER $\frac{3}{8}''$	NIL NIL	$\frac{1}{8}''$ MIN. $\frac{1}{4}''$ MAX $\frac{3}{16}''$ MIN. $\frac{5}{16}''$ MAX.
SQUARE BUTT...		NOT MORE THAN $\frac{1}{8}''$	—	0 TO $\frac{1}{16}''$
SQUARE BUTT...		NOT MORE THAN $\frac{3}{16}''$	—	$\frac{1}{16}''$ MIN $\frac{1}{8}''$ MAX
SQUARE BUTT...		NOT MORE THAN $\frac{1}{2}''$	—	1·4" x OVERALL DIA OF ELECTRODE* FOR HORIZONTAL WELDING. 1·8" x OVERALL DIA OF ELECTRODE* FOR VERTICAL WELDING
SQUARE BUTT...		$\frac{3}{16}''$ TO $\frac{1}{2}''$ FOR VERTICAL WELDING ONLY.	—	1·2 x OVERALL DIA OF ELECTRODE* FOR VERTICAL WELDING ONLY.

*NOTE· OVERALL DIAMETER OF ELECTRODE TO BE MEASURED OVER COATING

Butt Weld Preparations (B.S.S. 538)

Fig. 36

TYPE OF BUTT WELD	DETAIL OF WELD.	PLATE THICKNESS t.	ROOT FACE t'.	GAP g.
		IN.	IN.	IN.
SINGLE V ...	 FOR FLAT OR UNDERH° WELDING ··· α ≮ 60° FOR VERTICAL & UPRIGHT HORIZONTAL WELD° .. α ≮ 70° FOR OVERHEAD WELD° α ≮ 80°	NOT MORE THAN 3/8" OVER 3/8"	— —	1/16" MIN. 3/16" MAX. 1/8" MIN. 1/4" MAX.
SINGLE V ...	 FOR FLAT OR UNDERH° WELDING α ≮ 60° FOR VERTICAL & UPRIGHT HORIZONTAL WELD° α ≮ 70° FOR OVERHEAD WELD° α ≮ 80°	NOT MORE THAN 3/8" OVER 3/8"	1/16" MAX 1/8" MAX.	1/16" MIN. 3/16" MAX. 1/8" MIN 1/4" MAX
DOUBLE V ...	 FOR FLAT OR UNDERHAND WELDING .. α ≮ 60° FOR VERTICAL & UPRIGHT HORIZONTAL WELD° ... α ≮ 70° FOR OVERHEAD WELD° .. α ≮ 80°	NOT MORE THAN 3/8" OVER 3/8"	— —	1/16" MIN. 3/16" MAX. 1/8" MIN. 1/4" MAX.
SINGLE U...	 r = 1/8 IN α = 15° MIN. r = 3/16 IN OR MORE α = 10° MIN	NOT MORE THAN 1/2" OVER 1/2"	1/16" MAX 1/8" MAX.	1/8" MIN. 1/4" MAX. 1/8" MIN. 1/4" MAX.
DOUBLE U ...	 r = 1/8 IN. α = 15° MIN r = 3/16 IN. OR MORE α = 10° MIN	NOT MORE THAN 1/2" OVER 1/2"	1/16" MAX. 1/8" MAX.	1/8" MIN. 1/4" MAX. 1/8" MIN. 1/4" MAX.
SINGLE J ...	 α = 20° MIN. 30° MAX r = 3/16 IN. MIN	NOT MORE THAN 1/2" OVER 1/2"	1/16" MAX. 1/8" MAX.	1/8" MIN. 1/4" MAX 1/8" MIN. 1/4" MAX.
DOUBLE J ...	 α = 20° MIN. 30° MAX r = 3/16 IN. MIN	NOT MORE THAN 1/2" OVER 1/2"	1/16" MAX. 1/16" MAX.	1/8" MIN 1/4" MAX 1/8" MIN. 1/4" MAX.

Butt Weld Preparations (B.S.S. 538)

FIG. 37

used for relatively small plate thicknesses unless special deep-penetration electrodes are used. The V-preparation of thick plates has the limitation that, unless a sufficiently large value of angle α is used, difficulties will be encountered in depositing the first runs right at the root of the weld preparation. Large angles, however, have the disadvantage that an uneconomically large amount of weld metal has to be deposited at the top of the weld, due to the wide gap created by the large angle.

For such cases, the J- or U-preparation is recommended, which combines good accessibility at the bottom of the weld with an economical width at the top. Whilst V-preparations can be produced by flame cutting, J- or U-preparations have to be machined, unless special flame-gouging equipment is available.*

If the plates to be welded are of relatively large thickness and accessible from both sides, double V- or double U-preparation may be used. They produce less distortion than single V- or U-welds and require less weld metal.

Stress Calculations for Welds.

The critical dimension with regard to the strength of a fillet or a butt weld is the throat thickness. This is the minimum thickness of the

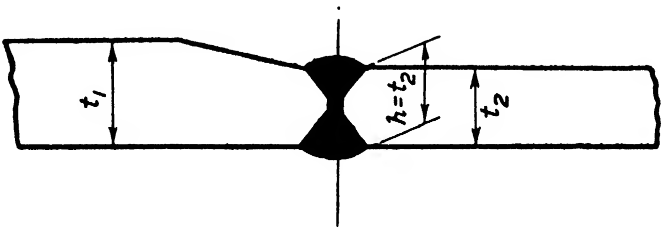


FIG. 38

weld on a straight line passing through its root. The throat thickness h of a butt weld connecting two plates of unequal thicknesses t_1 and t_2 is, therefore, equal to the thickness t_2 of the thinner plate (Fig. 38). In such cases the taper of the thicker plate should have a slope of 1 inch in at least 5 inches (B.S.S. 538). For stress calculations of 45° fillet welds with the leg length h, the theoretical value of the throat thickness, $\frac{h}{\sqrt{2}}$ or $\cdot707h$ (see Fig. 20), is used. In actual practice the throat thickness of fillet welds has to be kept within the limits of $\cdot6h$ and $\cdot9h$.

The reinforcement of a butt weld and the additional thickness created by the sealing run (see Fig. 19) should not be taken into account when calculating the stress-carrying section of the weld, because the reinforcement is zero at the side of the weld, and the sharp corner at this point might even produce stress concentrations (see page 32). In fact, the most perfect butt weld is the one which is produced by de-

* See *Welding Journal*, July 1941.

positing a reinforcement and a sealing run and afterwards machining these down to the thickness of the plate. This procedure is prescribed by B.S.S. 538 for cases where the surface of a butt weld has to be flush with the surfaces of the plates which are welded together. The minimum height of reinforcements is also specified by B.S.S. 538. In the centre of the weld it should be :

Up to and including $1\frac{1}{4}''$ butt weld size—not less than 10 per cent of the weld size.
Above $1\frac{1}{4}''$ butt weld size—not less than $\frac{1}{8}''$.

The throat section which carries the maximum stress is determined by the throat thickness and the effective length, l, of the weld.

In the case of fillet welds, the full section is usually not maintained up to the end of the welding run. B.S.S. 538, therefore, specifies that, for the purpose of stress calculations for fillet welds, the actual length, l', of the weld less twice the weld size (leg length h) should be considered as the effective length, l, of the fillet weld (see Fig. 79). B.S.S. 538 further specifies that the effective length of fillet welds as specified above should be not less than 2″ or 6 times the weld size (leg length h). The effective length of fillet welds $l = l' - 2h$ should, therefore, be as follows :

$$\text{For } h < \cdot 333'' \ldots\ldots l \gtreqless 2'',$$
$$\text{For } h \gtreqless \cdot 333'' \ldots\ldots l \gtreqless 6h.$$

Whilst the throat sectional area of a fillet weld is the product of its throat thickness and effective length, the amount of deposited weld metal grows with the square of the throat thickness and only with the first power of the length. For equal throat sectional area, therefore, long welds with small throat thickness require less weld metal than short welds with large throat thickness. Consequently, the former are more economical than the latter.

A certain minimum size of fillet weld is, however, required for each material thickness, in order to avoid cracking in the weld due to differential contraction between weld and parent metal. Minimum fillet weld sizes, as determined by various specifications, are shown in the table (Fig. 5).

Although the stress distribution in welds is very rarely uniform and not as simple and accessible to accurate calculation as is the case for stresses in the parent metal, experiments have shown that, for the sake of design calculations,* uniform stress distribution may be assumed in most cases. It is important, however, to consider carefully those cases where uniformity of stress distribution is seriously influenced by conditions of design or loading.

In the case of welds generally used in engineering structures, the lack of uniform stress distribution is usually taken into account by the values of the recommended permissible stresses (see page 39).

* See C. H. Jennings, "Welding Design", *Transactions of the A.S.M.E.*, October 1936, page 497.

The symbols used in the following stress calculations are as shown below :

A_w =	throat section of weld	sq. inches
I_w =	moment of inertia of weld	inch[4]
L =	length of leverage applied	inches
M_b =	bending moment	lb. inches
M_t =	torque	lb. inches
P =	force applied	lb.
Z_w =	section modulus of weld	inch[3]
f =	total stress	lb./sq. inch
f_b =	bending stress	lb./sq. inch
f_n =	normal stress	lb./sq. inch
f_s =	shear stress	lb./sq. inch
h =	size of weld (In the case of butt welds h = depth of throat ; in the case of fillet welds h = leg length of weld.)	inches
l =	effective length of weld	inches
t =	thickness of plate	inches

Stresses due to the weight of beams are neglected in the following calculations.

I. Butt Welds

As a butt weld by its shape and arrangement forms an approximate continuation of the parent metal, the stress calculations are relatively simple.

The following 17 cases will now be considered :

Full Butt Welds.

(1) (Figs. 39-42) :

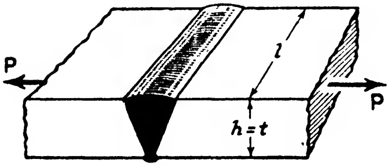


Fig. 39

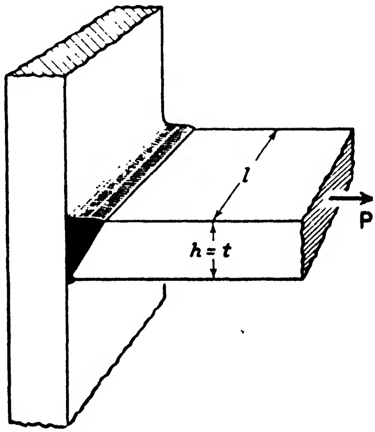


Fig. 40

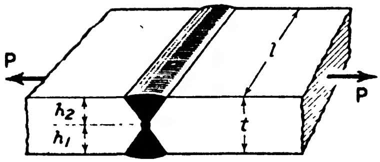


FIG. 41

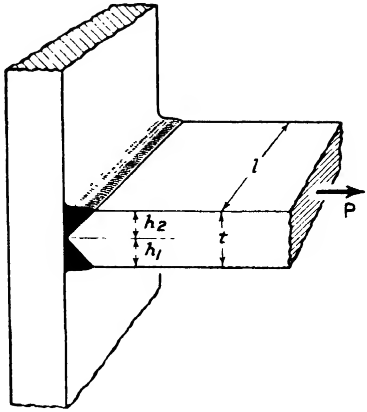


FIG. 42

$$f_{\max} = f_{n\ \max} = \frac{P}{A_w}.$$

(a) (Figs. 39, 40) $A_w = lh = lt$.

(b) (Figs. 41, 42) $A_w = l(h_1 + h_2) = lt$.

$$f_{\max} = \frac{P}{lt}. \qquad (4)$$

(2) (Figs. 43-46) :

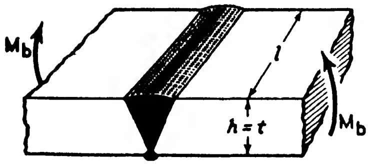


FIG. 43

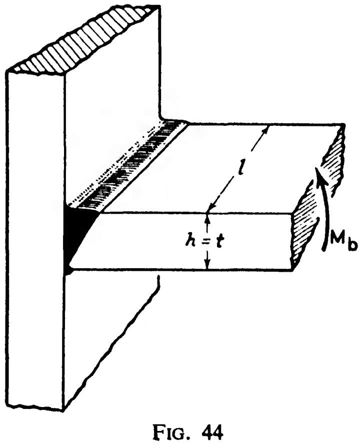


FIG. 44

$$f_{\max} = f_{b\ \max} = \frac{M_b}{Z_w}.$$

(a) (Figs. 43, 44) $Z_w = \dfrac{lh^2}{6} = \dfrac{lt^2}{6}$.

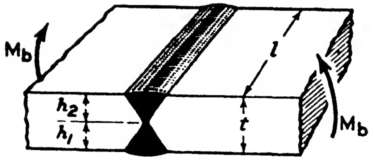


FIG. 45

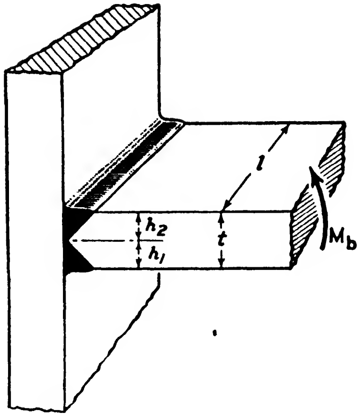


FIG. 46

(*b*) (Figs. 45, 46) $Z_w = \frac{l(h_1 + h_2)^2}{6} = \frac{lt^2}{6}$.

$$f_{\max} = \frac{6\ M_b}{lt^2}. \qquad (5)$$

(3) (Figs. 47, 48) :

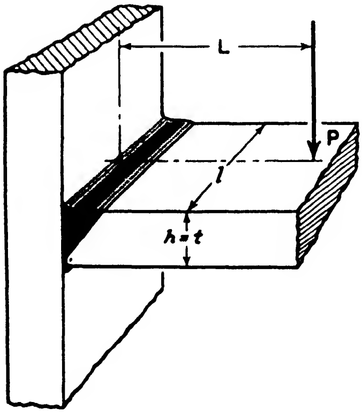


FIG. 47

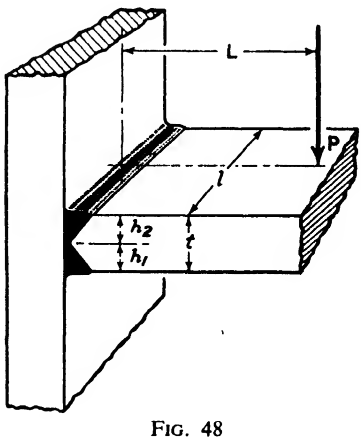


FIG. 48

$$f_{b\ \max} = \frac{PL}{Z_w},$$

$$f_{s\ \max} = \frac{3}{2}\frac{P}{A_w}.*$$

* Whilst the maximum bending stress occurs at the extreme fibre where the shear stress is zero, the maximum shear stress occurs at the neutral axis and is 50 per cent greater than the average shear stress $\frac{P}{A_w}$. See Timoshenko, *Strength of Materials*, D. Van Nostrand Company, New York, 1943–4.

(a) (Fig. 47) :

$$Z_w = \frac{lh^2}{6} = \frac{lt^2}{6},$$
$$A_w = lh = lt.$$

(b) (Fig. 48) :

$$Z_w = \frac{l(h_1 + h_2)^2}{6} = \frac{lt^2}{6},$$
$$A_w = l(h_1 + h_2) = lt,$$
$$f_{b\ max} = \frac{6\ PL}{lt^2}, \quad . \quad . \quad . \quad . \quad (6)$$
$$f_{s\ max} = \frac{3\ P}{2\ lt}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

(4) (Figs. 49, 50) :

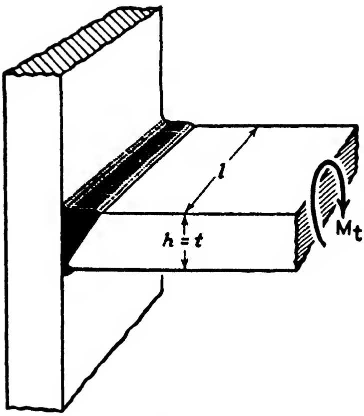


FIG. 49

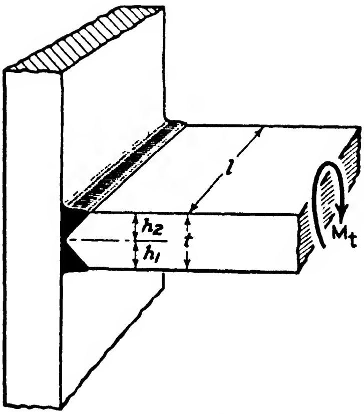


FIG. 50

(a) (Fig. 49) :

$$f_{max} = f_{s\ max} = \frac{M_t}{lh^2}\left(3 + 1{\cdot}8\ \frac{h}{l}\right)^*$$
$$= \frac{M_t}{l^2h^2}(3l + 1{\cdot}8h)$$

and $h = t$.

(b) (Fig. 50) :

$$f_{max} = f_{s\ max} = \frac{Mt}{l(h_1 + h_2)^2}\left[3 + 1{\cdot}8\ \frac{h_1 + h_2}{l}\right]$$
$$= \frac{M_t}{l^2(h_1 + h_2)^2}\left[3l + 1{\cdot}8\ (h_1 + h_2)\right]$$

$h_1 + h_2 = t,$

* This approximate but sufficiently accurate solution is given by C. Weber, " Die Lehre der Drehungsfestigkeit " (Theory of torsional strength "), *V.D.I.*, Berlin, 1921.

hence $$f_{\max} = \frac{M_t}{l^2t^2}(3l + 1{\cdot}8t). \quad . \quad . \quad . \quad (8)$$

(5) (Figs. 51, 52) :

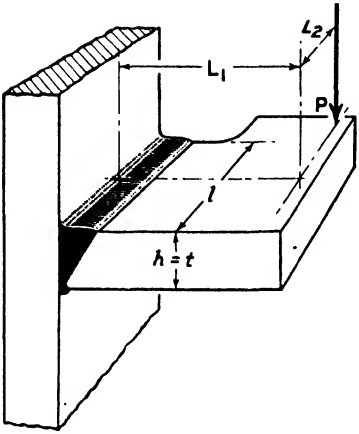


FIG. 51

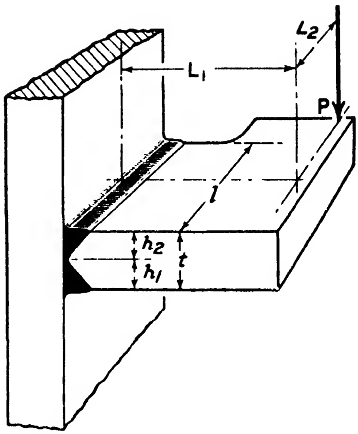


FIG. 52

$$f_{b\ \max} = \frac{PL_1}{Z_w} \quad \text{(see case 3)},$$

$$f_{\substack{s1\ \max \\ \text{shear}}} = \frac{3P}{2A_w} \quad \text{(see case 3)}.$$

(*a*) (Fig. 51) :

$$Z_w = \frac{lh^2}{6} = \frac{lt^2}{6},$$

$$A_w = lh = lt.$$

(*b*) (Fig. 52) :

$$Z_w = \frac{l(h_1 + h_2)^2}{6} = \frac{lt^2}{6},$$

$$A_w = l(h_1 + h_2) = lt,$$

$$f_{b\ \max} = \frac{6\ PL_1}{lt^2}, \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$f_{\substack{s1\ \max \\ \text{shear}}} = \frac{3\ P}{2lt} \quad \text{as (7)},$$

and, according to equation (8), substituting PL_2 for M_t,

$$f_{\substack{s2\ \max \\ \text{torsion}}} = \frac{PL_2}{l^2t^2}(3l + 1{\cdot}8t). \quad . \quad . \quad . \quad (10)$$

As the maximum value of f_{s1} occurs at the neutral axis where f_b is zero, the maximum combined bending and shear stress will usually occur at the extreme fibre where f_b and f_{s2} have their maximum value.* The combined stress at the extreme fibre is :

$$f_{\max} = \frac{f_{b\ \max}}{2} + \frac{1}{2}\sqrt{f^2_{b\ \max} + 4f^2_{s2\ \max}} \quad . \quad . \quad (11)$$

where $f_{b\ \max}$ and $f_{s2\ \max}$ are obtained from equations (9) and (10).

(6) (Figs. 53-56) :

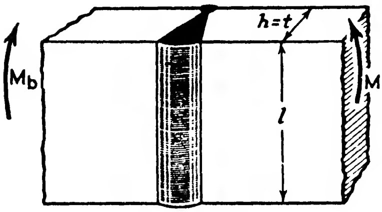


FIG. 53

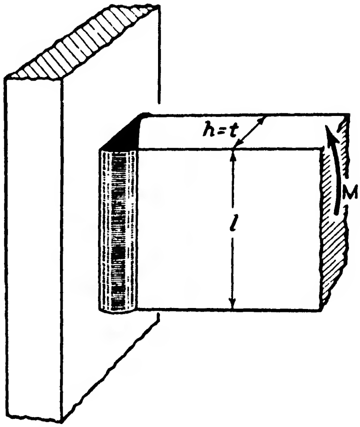


FIG. 54

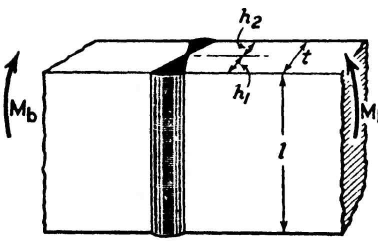


FIG. 55

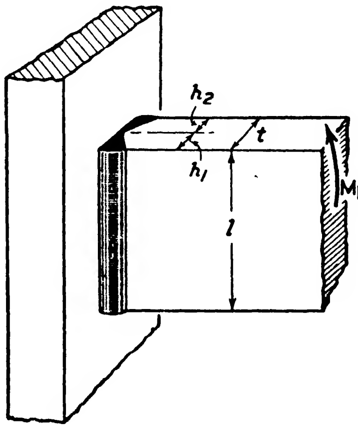


FIG. 56

* See Timoshenko, *Strength of Materials*, D. Van Nostrand Company, New York, 1943–4.

$$f_{\max} = f_{b\,\max} = \frac{M_b}{Z_w}.$$

(*a*) (Figs. 53, 54) :

$$Z_w = \frac{l^2h}{6} = \frac{l^2t}{6}.$$

(*b*) (Figs. 55, 56) :

$$Z_w = \frac{l^2(h_1 + h_2)}{6} = \frac{l^2t}{6},$$

$$f_{\max} = \frac{6\,M_b}{l^2t} \qquad \qquad (12)$$

(7) (Figs. 57, 58) :

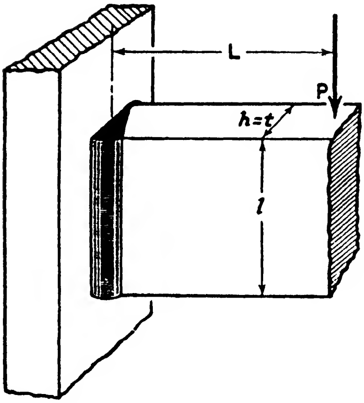


Fig. 57

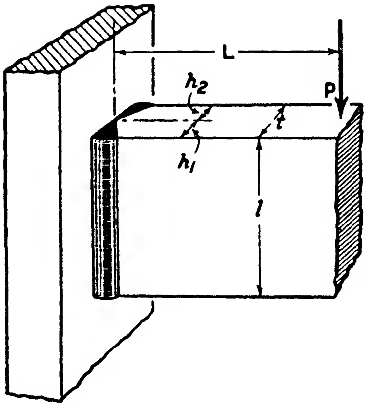


Fig. 58

$$f_{b\,\max} = \frac{PL}{Z_w},$$

$$f_{s\,\max} = \frac{3}{2}\frac{P}{A_w} \quad \text{(see case 3).}$$

(*a*) (Fig. 57) :

$$Z_w = \frac{l^2h}{6} = \frac{l^2t}{6},$$

$$A_w = lh = lt.$$

(*b*) (Fig. 58) :

$$Z_w = \frac{l^2(h_1 + h_2)}{6} = \frac{l^2t}{6},$$

$$A_w = l(h_1 + h_2) = lt,$$

$$f_{b\ \max} = \frac{6\ PL}{l^2 t}, \qquad (13)$$

$$f_{s\ \max} = \frac{3\ P}{2lt},\ \text{as (7)}.$$

(8) (Figs. 59, 60) :

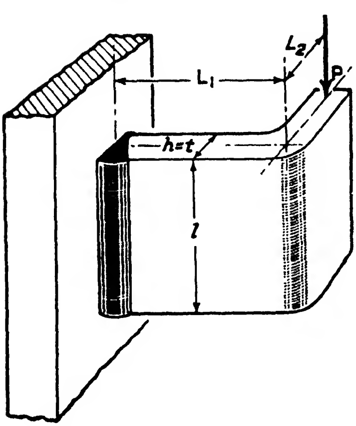


FIG. 59

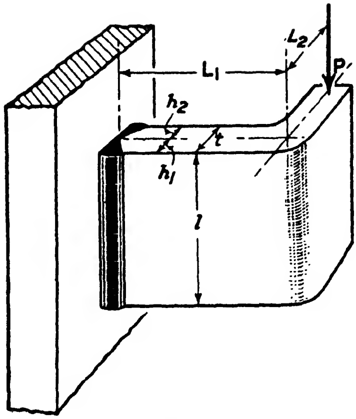


FIG. 60

$$f_{b\ \max} = \frac{PL_1}{Z_w} \quad \text{(see case 7)},$$

$$\underset{\text{(shear)}}{f_{s1\ \max}} = \frac{3\ P}{2\ A_w} \quad \text{(see case 7)}.$$

(*a*) (Fig. 59) :

$$Z_w = \frac{l^2 h}{6} = \frac{l^2 t}{6},$$

$$A_w = lh = lt.$$

(*b*) (Fig. 60) :

$$Z_w = \frac{l^2(h_1 + h_2)}{6} = \frac{l^2 t}{6},$$

$$A_w = l(h_1 + h_2) = lt,$$

$$f_{b\ \max} = \frac{6\ PL_1}{l^2 t}, \qquad (14)$$

$$f_{s1\ \max} = \frac{3\ P}{2lt}, \qquad (7)$$

$$\underset{\text{(torsion)}}{f_{s2\ \max}} = \frac{PL_2}{l^2 t^2}(3l + 1{\cdot}8t). \qquad (10)$$

Similar to case 5 :

$$f_{\max} = \frac{f_{b\ \max}}{2} + \frac{1}{2}\sqrt{f^2_{b\ \max} + 4 f^2_{s2\ \max}} \qquad (11)$$

where in this case $f_{b\ \max}$ is obtained from equation (14), $f_{s2\ \max}$ from equation (10).

Partial Butt Welds.

(9) (Figs. 61, 62) :

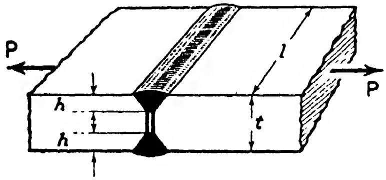


Fig. 61

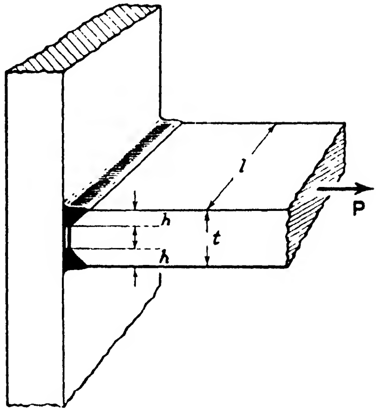


Fig. 62

$$f_{\max} = f_{n\ \max} = \frac{P}{A_w},$$

$$A_w = l(h+h) = 2lh,$$

$$f_{\max} = \frac{P}{2lh}. \qquad (15)$$

(10) (Figs. 63, 64) :

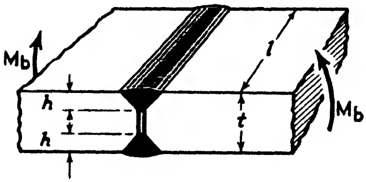


Fig. 63

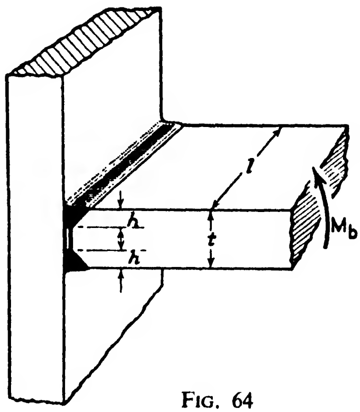


Fig. 64

$$f_{\max} = f_{b\ \max} = \frac{M_b}{Z_w},$$

$$Z_w = \frac{\dfrac{lt^3}{12} - \dfrac{l(t-2h)^3}{12}}{\dfrac{t}{2}}$$

$$= \frac{lh(3t^2 - 6th + 4h^2)}{3t},$$

$$f_{\max} = \frac{3\ M_b t}{lh(3t^2 - 6th + 4h^2)}. \qquad (16)$$

(11) (Fig. 65) :

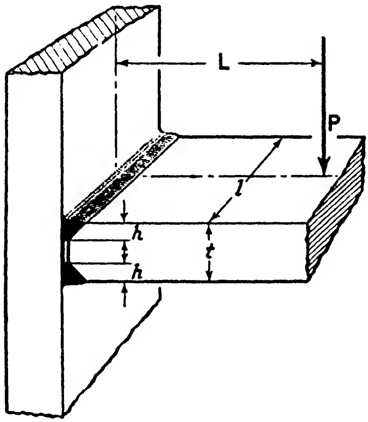


Fig. 65

$$f_{b\ \max} = \frac{PL}{Z_w},$$

$$f_{s\ \max} = \frac{3}{2}\frac{P}{A_w} \quad \text{(see case 3)},$$

$$Z_w = \frac{lh(3t^2 - 6th + 4h^2)}{3t} \quad \text{(see case 10)},$$

$$A_w = 2lh,$$

$$f_{b\ \max} = \frac{3\ PLt}{lh(3t^2 - 6th + 4h^2)}, \qquad (17)$$

$$f_{s\ \max} = \frac{3\ P}{4lh}. \qquad (18)$$

(12) (Figs. 66, 67) :

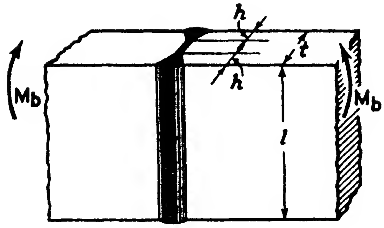


FIG. 66

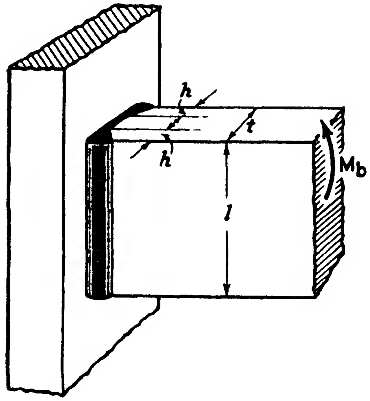


FIG. 67

$$f_{\max} = f_{b\max} = \frac{M_b}{Z_w},$$

$$Z_w = \frac{2l^2h}{6} = \frac{l^2h}{3},$$

$$f_{\max} = \frac{3\,M_b}{l^2h}. \qquad (19)$$

(13) (Fig. 68) :

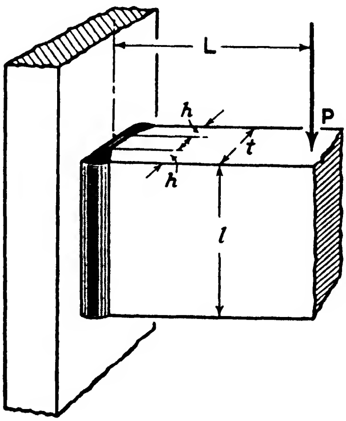


FIG. 68

$$f_{b\ \max}=\frac{PL}{Z_w},$$

$$f_{s\ \max}=\frac{3\ P}{2\ A_w}\quad \text{(see case 3)},$$

$$Z_w=\frac{2l^2h}{6}=\frac{l^2h}{3},$$

$$A_w=2lh,$$

$$f_{b\ \max}=\frac{3\ PL}{l^2h}, \qquad (20)$$

$$f_{s\ \max}=\frac{3\ P}{4lh}. \qquad (18)$$

(14) (Fig. 69) :

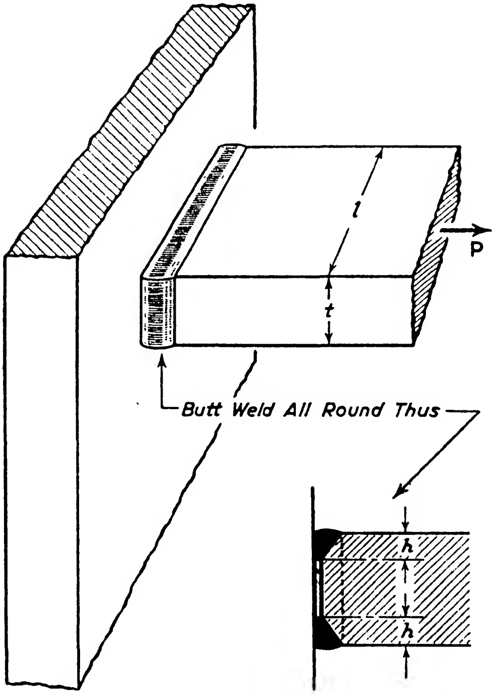


Fig. 69

$$f_{\max}=f_{n\ \max}=\frac{P}{A_w},$$

$$\begin{aligned}A_w&=lt-(l-2h)(t-2h)\\&=2h(l+t-2h),\end{aligned}$$

$$f_{\max}=\frac{P}{2h(l+t-2h)}. \qquad (21)$$

(15) (Fig. 70):

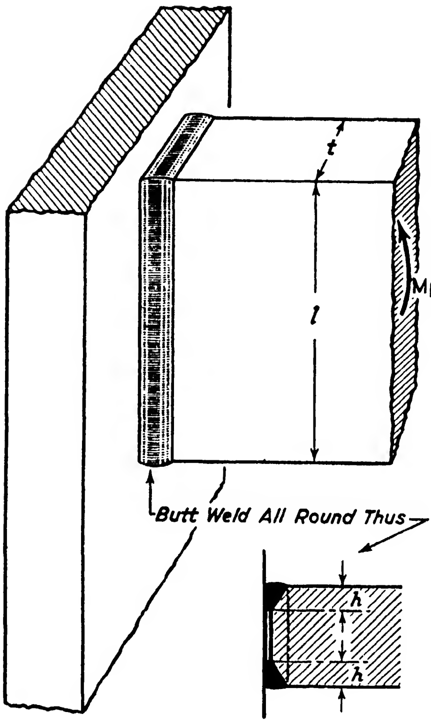


Fig. 70

$$f_{\max}=f_{b\max}=\frac{M_b}{Z_w},$$

$$Z_w=\frac{2}{l}\times\frac{l^3t-(l-2h)^3(t-2h)}{12},$$

$$Z_w=\frac{l^3t-(l-2h)^3(t-2h)}{6l},$$

$$f_{\max}=\frac{6\,M_bl}{l^3t-(l-2h)^3(t-2h)}. \qquad (22)$$

(16) (Fig. 71) :

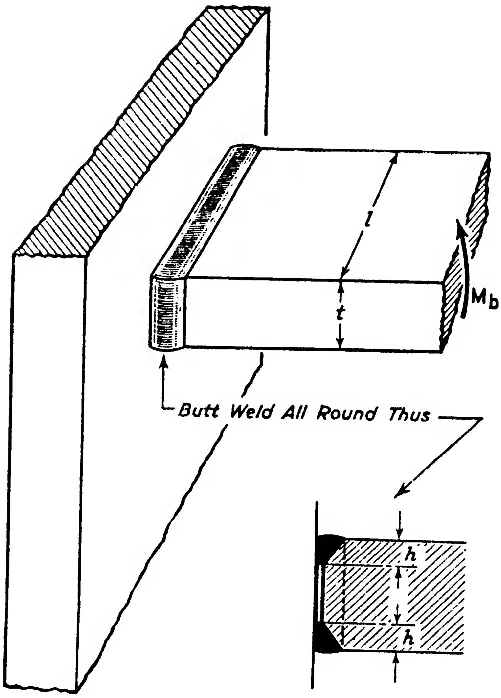


FIG. 71

$$f_{\max} = f_{b\ \max} = \frac{M_b}{Z_w},$$

$$Z_w = \frac{2}{t} \times \frac{lt^3 - (l-2h)(t-2h)^3}{12}$$

$$= \frac{lt^3 - (l-2h)(t-2h)^3}{6t},$$

$$f_{\max} = \frac{6\ M_b t}{lt^3 - (l-2h)(t-2h)^3}. \qquad (23)$$

(17) (Fig. 72) :

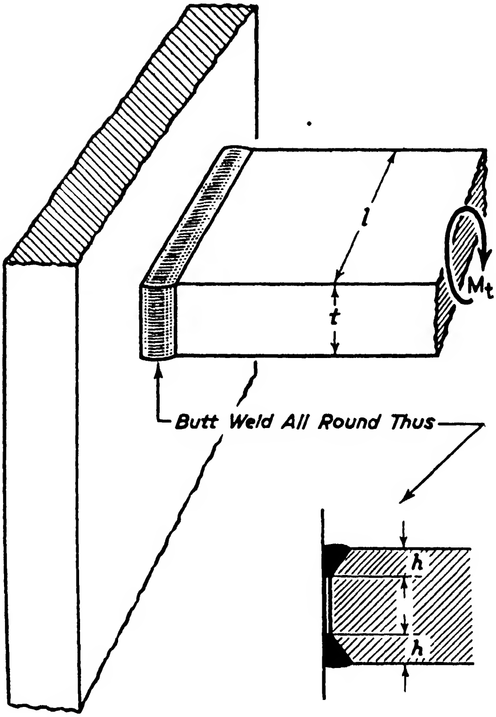


FIG. 72

$$f_{s\ max} = \frac{M_t}{2\,Ah}\ *$$

where A is the area which lies inside the centre line of the weld section :

$$A = (l-h)(t-h),$$

$$f_{s\ max} = \frac{M_t}{2(l-h)(t-h)h}. \qquad (24)$$

II. Fillet Welds

The accuracy attained in stress calculations of fillet welds is influenced by two complicating factors :

(*a*) The stress distribution is not uniform.

(*b*) Due to their shape and the positions occupied by fillet welds, a combination of stressing conditions will be encountered. The forces acting will usually be eccentric as the plates are overlapped, and bending moments in the fillet welds are thus created.

For those reasons, stress calculations of fillet welds can only be approxim-

* See Timoshenko, *Strength of Materials*, D. Van Nostrand Company, New York, 1943–4.

ate, but experiments and tests have proved that, from the designer's point of view, satisfactory results can be obtained.

Parallel Fillet Welds.

The simplest conditions occur in the case of parallel fillet welds, *i.e.* in welds which are parallel to the line of action of the forces applied to them (Fig. 73). The bending stress in such welds will be small in comparison with the shear stress. The leverage is proportional to the thickness of the plates, whilst the section modulus resisting the bending moment grows with the square of the effective length of the weld, which should be at least equal to the width of the connected plate.

In general practice the bending stress is neglected in such welds, and calculations are carried out as if they were only stressed in pure shear (see page 37).

In the previous calculations, butt welds were treated as if they were part of the parent metal (see page 44). With regard to side fillet welds, it has been confirmed by a large number of experiments, according to Jennings,* that the shear stress at the ends is considerably higher than it is in the centre, but that the shear strength of such welds is directly proportional to the throat area, of the weld. Consequently, the practice used in stress calculations, which considers uniform stress distribution throughout the weld throat area and which assumes equal stress distribution between symmetrically arranged welds, has proved satisfactory from the designer's point of view. The inaccuracy in these calculations is allowed for in the permissible stress values as specified for such cases (see page 39).

The following 18 cases will now be considered :

(1) (Fig. 73) :

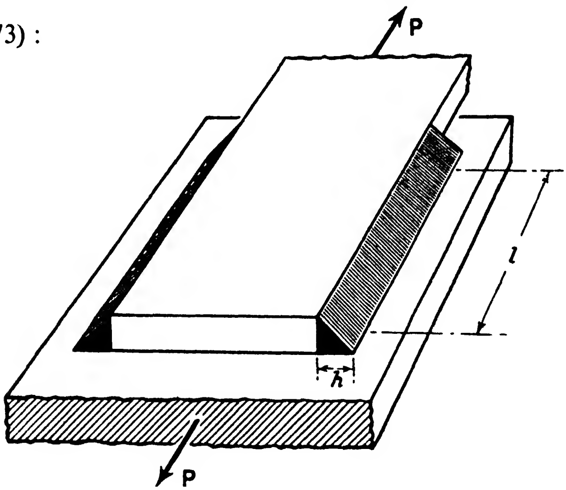


FIG. 73

* C. H. Jennings, " Welding Design ", *Transactions of the A.S.M.E.*, October 1936, page 497.

The section of highest stress is the throat area of the weld, which is equal to the throat depth ($\cdot 707h$) multiplied by the effective length l (see page 43) of the two welds :

$$f_{\max} = f_s = \frac{P}{A_w},$$

$$A_w = 2l \times \cdot 707h = 1 \cdot 414lh,$$

or

$$f_{\max} = \frac{P}{1 \cdot 414lh}$$

$$= \frac{\cdot 707\ P}{lh} \quad . \quad . \quad . \quad . \quad . \quad . \quad (25a)$$

(2) (Fig. 74) :

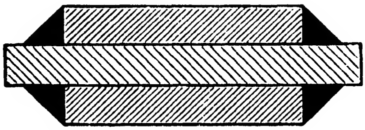


FIG. 74

If three plates are connected, the application of the load is more symmetrical still and is distributed over four welds instead of two, so that the stress is half that obtained in equation (25a) ; thus

$$f_{\max} = \frac{\cdot 354\ P}{lh} \quad . \quad . \quad . \quad . \quad (25b)$$

(3) (Fig. 75) :

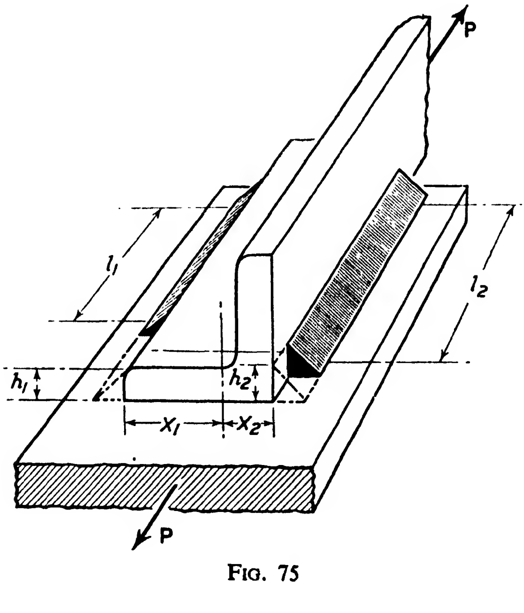


FIG. 75

When connecting unsymmetrical rolled sections, fillet welds should be arranged so as to make the centre of gravity of the welds coincide with the centre of gravity of the rolled section, *i.e.* the throat areas of the fillet welds on either side of the neutral axis of the rolled section must be arranged in proportion to their distances from that axis. The reduction of effective leg length of the fillet due to the radius at the toe of the rolled section (see Fig. 11) has to be taken into consideration.

The stress in the welds is

$$f_{\max}=f_s=\frac{P}{A_w},$$

where

$$A_w=A_{w1}+A_{w2}$$

or

$$f_{\max}=\frac{P}{\cdot707l_1h_1+\cdot707l_2h_2}$$

$$=\frac{1\cdot414\ P}{l_1h_1+l_2h_2}. \qquad (26)$$

The distribution of throat areas of the fillet welds in relation to the neutral axis of the rolled section can be obtained through the following equations :

$$\frac{A_{w1}}{A_{w2}}=\frac{x_2}{x_1},$$

$$\frac{A_{w1}}{A_{w1}+A_{w2}}=\frac{A_{w1}}{A_w}=\frac{x_2}{x_1+x_2},$$

$$\frac{A_{w2}}{A_{w1}+A_{w2}}=\frac{A_{w2}}{A_w}=\frac{x_1}{x_1+x_2},$$

$$A_{w1}=A_w\frac{x_2}{x_1+x_2}, \qquad (27)$$

$$A_{w2}=A_w\frac{x_1}{x_1+x_2}. \qquad (28)$$

With given sizes (leg lengths) of the fillets, and a given permissible maximum shear stress, the effective lengths of the fillet welds on either side of the angle section are determined as follows :

$$A_w=\frac{P}{f_{\max}},$$

$$A_{w1}=\frac{P}{f_{\max}}\times\frac{x_2}{x_1+x_2},$$

$$A_{w2}=\frac{P}{f_{\max}}\times\frac{x_1}{x_1+x_2},$$

$$l_1=\frac{A_{w1}}{\cdot707h_1}=\frac{1\cdot414A_{w1}}{h_1},$$

and

$$l_2=\frac{A_{w2}}{\cdot707h_2}=\frac{1\cdot414A_{w2}}{h_2}.$$

Substituting for A_{w1} and A_{w2}, we get

$$l_1 = \frac{1 \cdot 414\ P}{h_1 f_{max}} \times \frac{x_2}{x_1 + x_2} \qquad (29)$$

and

$$l_2 = \frac{1 \cdot 414\ P}{h_2 f_{max}} \times \frac{x_1}{x_1 + x_2} \qquad (30)$$

The bending stress in parallel fillet welds must, however, be considered when a bending moment (Fig. 76) or a force acting on a long leverage (Fig. 77) is applied.

Although the throat area of each weld lies at an angle of 45° to the main supporting surface, it is generally an accepted practice to treat it as if it were in one plane with that surface.

(4) (Fig. 76) :

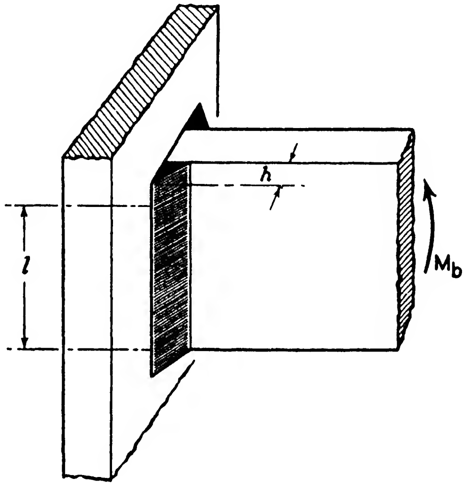


FIG. 76

$$f_{max} = f_{b\ max} = \frac{M_b}{Z_w},$$

$$Z_w = \frac{2l^2 \times \cdot 707h}{6}$$

$$= \cdot 236 l^2 h,$$

$$f_{max} = \frac{M_b}{\cdot 236 l^2 h}$$

$$= \frac{4 \cdot 24 M_b}{l^2 h} \qquad (31)$$

(5) (Fig. 77) :

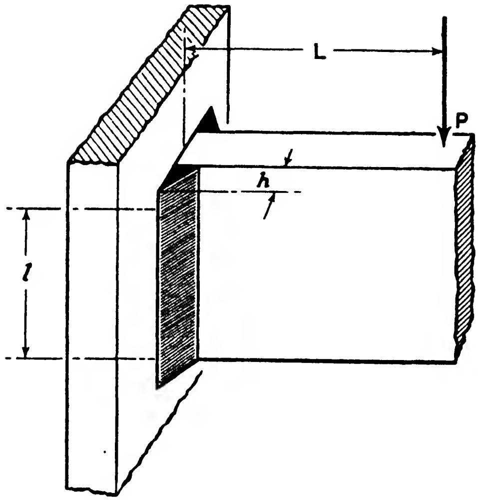


FIG. 77

Bending $f_{b\ \max}=\frac{\mathrm{PL}}{\mathrm{Z}_w}$.

Shear $f_{s\ \max}=\frac{3\ \mathrm{P}}{2\mathrm{A}_w}$.

See butt welds, case 3 (page 46).

Since $\mathrm{Z}_w=\cdot 236 l^2 h$

and $\mathrm{A}_w=1\cdot 414 lh,$

we get $f_{b\ \max}=\frac{\mathrm{PL}}{\cdot 236 l^2 h}$

or $$f_{b\ \max}=\frac{4\cdot 24\ \mathrm{PL}}{l^2 h}, \qquad (32)$$

$$f_{s\ \max}=\frac{3}{2}\times\frac{\mathrm{P}}{1\cdot 414 lh}$$

$$=\frac{1\cdot 06\ \mathrm{P}}{lh} \qquad (33)$$

This maximum shear stress occurs in the centre of the weld, whilst the maximum bending stress occurs at the ends of the weld. From equations (32) and (33) it can be seen that the bending stress is greater than the shear stress, if $\mathrm{L}>\frac{l}{4}$. As l is at least $6h$ (see page 43), $f_b>f_s$ if $\mathrm{L}>1\cdot 5h$. For this reason the shear stress need not be considered generally in case 5, and the bending stress need not be considered in cases such as cases 1-3 on pages 59-61.

(6) (Fig. 78) :

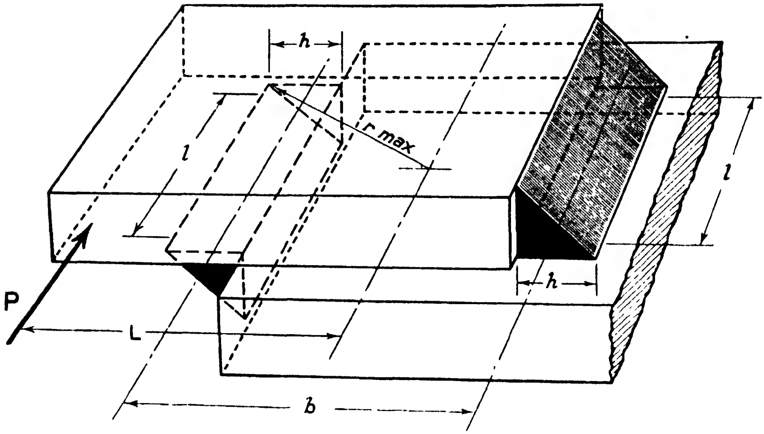


FIG. 78

See Combined Side and End Fillet Welds (case 18).

End Fillet Welds.

The stress in an end fillet weld with a face angle of 45° which connects two overlapping plates in tension will now be considered (Fig. 79).

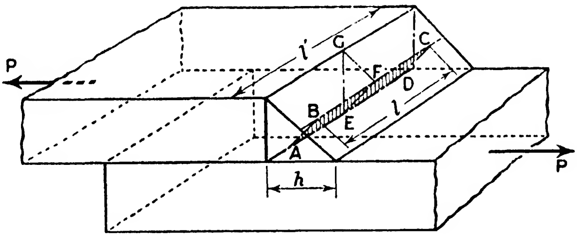


FIG. 79

The most unfavourable condition occurs when the plates are not resting flat against each other, the weld being then stressed in tension, shear, and bending. The effective throat area A–B–C–D is the product of the effective length B–C or l (see page 43) and the throat thickness A–B which equals ·707h (see Fig. 20).

In the resulting load at the centre portion E–F–G of the weld (Fig. 80) it may be assumed that the force P acting to the left is distributed over the leg-length h (E–G), and acts at the centre of gravity of the top plate, *i.e.* at $\frac{h}{2}$ above the bottom of the weld.

The reaction (of value P acting to the right) is distributed over the throat area, its resultant acting through the centre of gravity of the latter, *i.e.* $h/4$ below the line of action of the force P acting to the left.

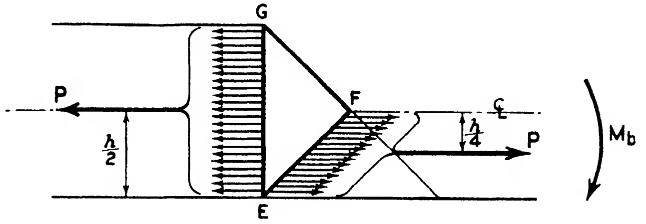


FIG. 80

If these forces (Fig. 80) are to be in equilibrium, a bending moment or couple $= M_b = \frac{Ph}{4}$ must be exerted.

If the two connected plates are sufficiently stiff, this bending moment could be balanced by the supporting forces P_M (Fig. 81) acting on a

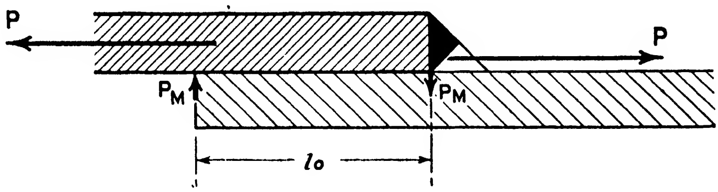


FIG. 81

leverage which is equal to the overlap l_0 ($P_M = \frac{M_b}{l_0}$). If, however, the overlap is small and P_M becomes large enough to stress the plates above their yield point, deformation (Fig. 82) will occur and the bending stress will have to be carried by the weld.

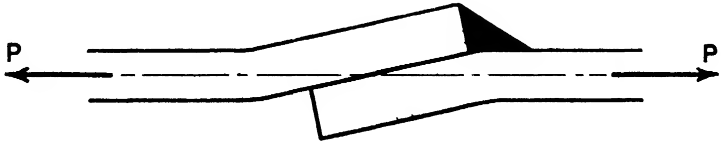


FIG. 82

If the axis parallel to the weld is called X, the axis transverse to the weld Y, and the axis normal to the weld throat Z (Fig. 83), the force P can be divided into the component forces

$$P_y = P \cos 45°, \text{ and}$$
$$P_z = P \sin 45°,$$
$$P_y = \cdot 707P \quad \text{(shear)},$$
$$P_z = \cdot 707P \quad \text{(tension)}.$$

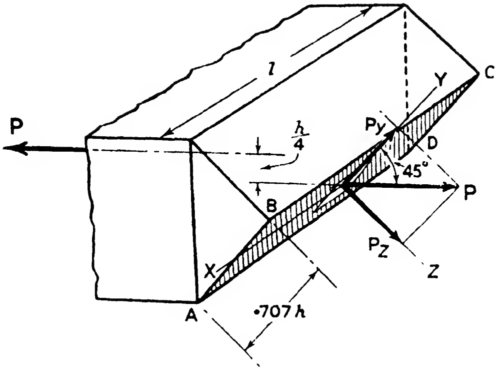


FIG. 83

The bending moment acting on the throat section is then

$$M_b = \frac{Ph}{4}.$$

Assuming uniform stress distribution over the throat area (see page 59), the shear stress is

$$f_s = \frac{P_y}{\cdot 707lh} = \frac{\cdot 707P}{\cdot 707lh}$$
$$= \frac{P}{lh};$$

the tensile stress is

$$f_t = \frac{P_z}{\cdot 707lh} = \frac{\cdot 707P}{\cdot 707lh}$$
$$= \frac{P}{lh};$$

the bending stress is

$$f_b = \frac{M_b}{Z_w},$$

and

$$M_b = \frac{Ph}{4},$$
$$Z_w = \frac{l(\cdot 707h)^2}{6} = \frac{lh^2}{12},$$
$$f_b = \frac{\frac{Ph}{4}}{\frac{lh^2}{12}} = \frac{3P}{lh}.$$

The maximum normal stress in the extreme fibre, where f_b and f_t act in the same direction, is

$$f_n = f_t + f_b = \frac{P}{lh} + \frac{3P}{lh}$$

$$= \frac{4P}{lh}.$$

The maximum stress is then

$$f_{max} = \frac{f_n}{2} + \frac{1}{2}\sqrt{f_n^2 + 4f_s^2},$$

$$f_{max} = \frac{2P}{lh} + \frac{1}{2}\sqrt{\frac{16P^2}{l^2h^2} + \frac{4P^2}{l^2h^2}}$$

$$= \frac{2P}{lh} + \frac{P\sqrt{5}}{lh}$$

$$= \frac{4{\cdot}236P}{lh} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

In order to relieve the weld from heavy bending stresses, single fillet joints should be avoided, if loading conditions are likely to create bending stresses, and the welds should be arranged in pairs (see Figs. 85 and 86), making the overlap as large as possible and not less than four times the thickness of the thinner plate.

If the bending stress is not carried by the weld, the normal stress in a fillet weld is

$$f_n = f_t = \frac{P}{lh},$$

and the maximum stress is

$$f_{max} = \frac{P}{2lh} + \frac{1}{2}\sqrt{\frac{P^2}{l^2h^2} + 4\frac{P^2}{l^2h^2}}$$

$$= \frac{P}{2lh} + \frac{P\sqrt{5}}{2lh}$$

$$= \frac{1{\cdot}618P}{lh} \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

Equation (35) gives a slightly higher stress than the formula generally used, which assumes the weld throat to be stressed in direct tension by the full force P, and without shear, *i.e.*

$$f_{max} = \frac{P}{{\cdot}707lh}$$

$$= \frac{1{\cdot}414P}{lh} \quad . \quad . \quad . \quad . \quad . \quad (36)$$

C. H. Jennings quotes experimental results * which are in close agreement with equation (35), and this equation has also been suggested by L. W. Schuster † and M. F. Spotts.‡

It is, however, general practice, and suggested by standards and regulations in use in the welding industry,§ to neglect the shear in the weld throat and to use formula (36).

The following arrangements of end fillet welds are generally used :

(7) (Figs. 84, 85) :

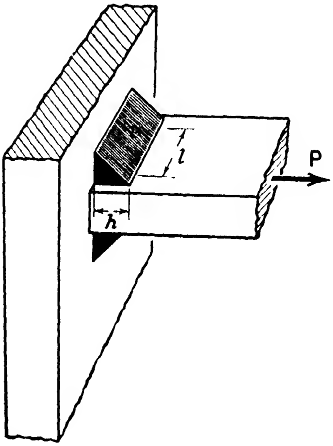


FIG. 84

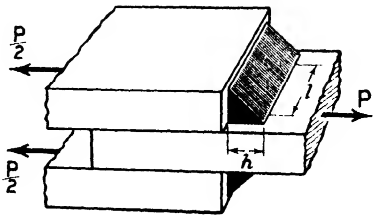


FIG. 85

$$f=\frac{P}{A_w},$$

$$A_w=2\times(\cdot707lh)$$
$$=1\cdot414lh,$$

$$f=\frac{P}{1\cdot414lh},$$

$$f=\frac{\cdot707P}{lh} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

* Discussion on the paper "Welding Design," by C. H. Jennings, *Transactions of the A.S.M.E.*, 1937, page 462.

† British Engine, Boiler and Electrical Ins. Co., Technical Report for 1928.

‡ " Stresses in Fillet Welds with Excentric Loads ", *Welding Journal*, 1944, page 692.

§ See *Transactions of the A.S.M.E.*, 1936 and 1937, C. H. Jennings ; *Handbook for Welded Structural Steelwork*, The Institute of Welding, 1943 ; *Welding Memoranda*, issued by the Advisory Service on Welding, Ministry of Supply, London ; etc.

(8) (Fig. 86) :

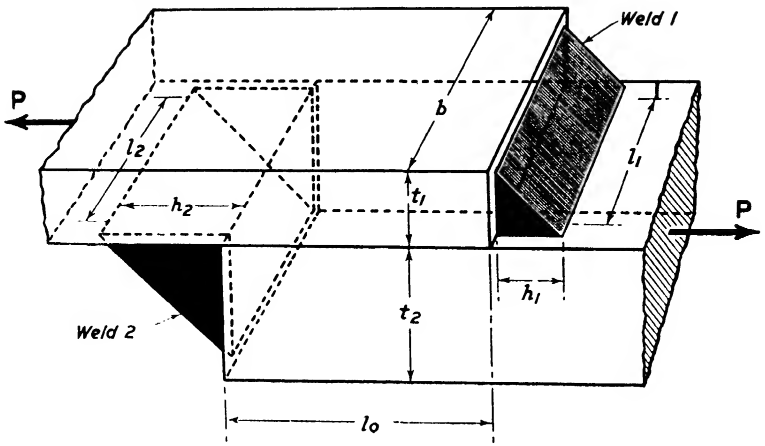


FIG. 86

(*a*) If plates of unequal thicknesses (t_1 and t_2) are connected in a lap joint, the load transmitted by each weld depends upon the ratio of the plate thicknesses.

The force P_1 carried by weld 1 is transmitted by the overlapping part of the thinner plate (thickness t_1), whilst the force P_2 carried by weld 2 is transmitted by the overlapping part of the thicker plate (thickness t_2). The sum of P_1 and P_2 must be equal to the applied total force P, or

$$P_1 + P_2 = P.$$

The elongations e_1 and e_2 of the overlapping parts of the two plates must be equal, as the distance between the welds cannot vary. If E is the modulus of elasticity of the plate material,

$$e_1 = \frac{P_1}{bt_1E}l_0$$

and $$e_2 = \frac{P_2}{bt_2E}l_0\,;$$

now $$e_1 = e_2,$$

hence $$\frac{P_1}{P_2} = \frac{t_1}{t_2}$$

and $$\frac{P_1}{P_1+P_2} = \frac{t_1}{t_1+t_2};$$

but since $$P_1 + P_2 = P,$$

$$P_1 = P\frac{t_1}{t_1+t_2},$$

$$P_2 = P\frac{t_2}{t_1+t_2}.$$

According to equation (36) the stress in weld 1 is

$$f_1 = \frac{1{\cdot}414P_1}{l_1 h_1},$$

and in weld 2 is

$$f_2 = \frac{1{\cdot}414P_2}{l_2 h_2};$$

or on substitution

$$f_1 = \frac{1{\cdot}414Pt_1}{l_1 h_1(t_1 + t_2)} \quad . \quad . \quad . \quad (38)$$

and

$$f_2 = \frac{1{\cdot}414Pt_2}{l_2 h_2(t_1 + t_2)} \quad . \quad . \quad . \quad (39)$$

(*b*) In case of the weld sizes being equal to the corresponding plate thicknesses ($h_1 = t_1$; $h_2 = t_2$), the stress in weld 1 is

$$f_1 = \frac{1{\cdot}414P}{l_1(t_1 + t_2)}; \quad . \quad . \quad . \quad (40)$$

the stress in weld 2 is

$$f_2 = \frac{1{\cdot}414P}{l_2(t_1 + t_2)}. \quad . \quad . \quad . \quad (41)$$

(*c*) In case of the two plates and the two welds being of equal size ($h_1 = h_2 = h$; $l_1 = l_2 = l$; $t_1 = t_2 = t$), the stress in each weld is

$$f = \frac{{\cdot}707P}{lh} \quad . \quad . \quad . \quad . \quad . \quad (42)$$

(9) (Fig. 87) :

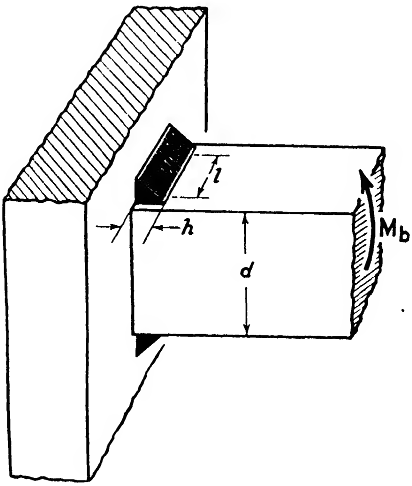


FIG. 87

$$f_b = \frac{M_b}{Z_w},$$

$$Z_w = \frac{2}{(d+1\cdot414h)} \times \left[\frac{l(d+1\cdot414h)^3}{12} - \frac{ld^3}{12}\right] \quad \text{(Fig. 88)}$$

$$= \frac{l}{6(d+1\cdot414h)}(4\cdot242d^2h + 6dh^2 + 2\cdot828h^3)$$

$$= \frac{l}{6(d+1\cdot414h)} 4\cdot242h^2d\left[\frac{d}{h} + 1\cdot414 + \cdot667\frac{h}{d}\right].$$

Usually d will be not less than $2h$. Therefore, if the value $\cdot667\frac{h}{d}$ is

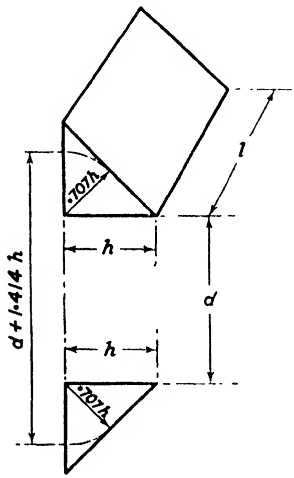


FIG. 88

neglected, the error will not be greater than 10 per cent, reducing Z_w, and correspondingly raising the calculated stress and thus erring on the safe side. Then

$$Z_w = \frac{\cdot707ldh^2}{d+1\cdot414h} \times \left(\frac{d}{h} + 1\cdot414\right)$$

$$= \frac{\cdot707ldh^2}{d+1\cdot414h} \times \left(\frac{d+1\cdot414h}{h}\right)$$

$$= \cdot707ldh\ ;$$

hence $$f_b = \frac{M_b}{\cdot707ldh}$$

and $$f_b = \frac{1\cdot414M_b}{ldh} \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

(10) (Fig. 89):

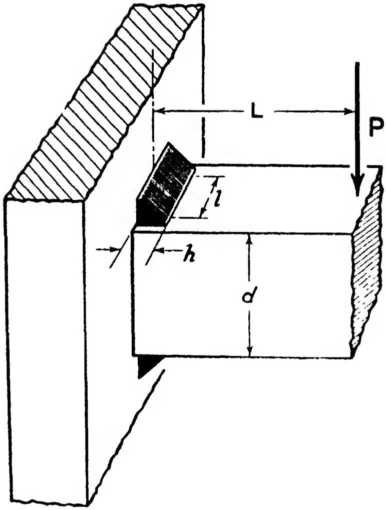


FIG. 89

Assuming that each weld takes 50 per cent of the vertical force,

$$f_t = \frac{P}{2A_w},$$

$$A_w = \cdot 707lh,$$

and

$$f_t = \frac{P}{1\cdot 414lh}$$

$$= \frac{\cdot 707P}{lh} \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

The bending stress, according to case 9, where $M_b = PL$, is

$$f_b = \frac{1\cdot 414PL}{ldh} \quad . \quad . \quad . \quad . \quad (45)$$

These stresses can be added geometrically to obtain the resultant total stress (Fig. 90), or

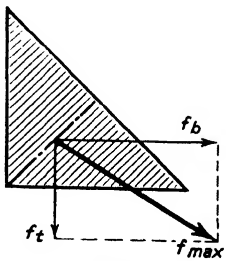


FIG. 90

$$f_{max} = \sqrt{f_t^2 + f_b^2}$$

$$= \sqrt{\frac{P^2}{2h^2} + \frac{2P^2L^2}{l^2d^2h^2}}$$

$$f_{max} = \frac{\cdot 707P}{lh} \times \sqrt{1 + \frac{4L^2}{d^2}} \quad . \quad . \quad . \quad . \qquad (46)$$

(11) (Fig. 91) :

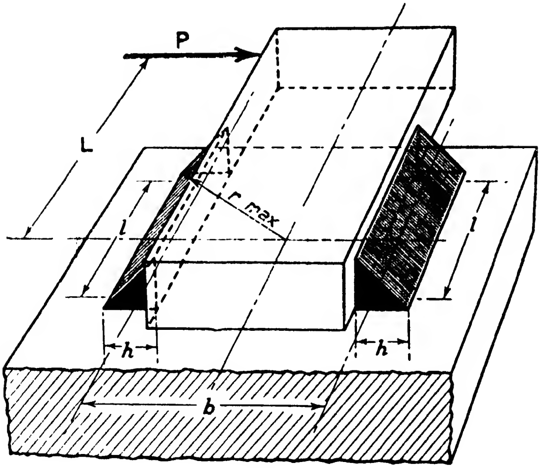


FIG. 91

See Combined Side and End Fillet Welds (case 18).

Combined Side and End Fillet Welds.

It has been explained before (see page 43) that the full length of a fillet weld cannot be considered as effective. It will often occur, however, that weld lengths equal to the full length or width of a lap joint are required in order to keep the stresses below the permissible maximum. In such cases the ends of the welds are returned round the corner of the lap by not less than the leg length of the fillet or not less than ½″, whichever is the lower value * (Fig. 92).

* *Welding Memorandum*, No. 11, issued by the Advisory Service on Welding, Ministry of Supply.

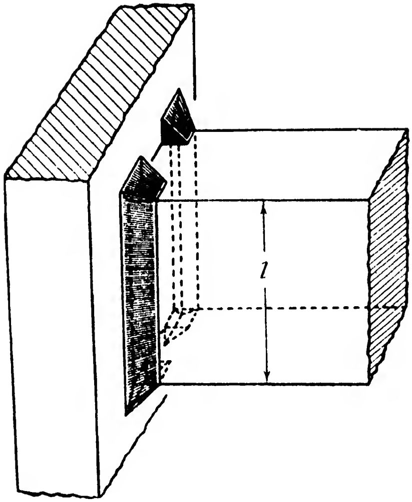


FIG. 92

This procedure can be applied in the case of side or end fillet welds. The full length of the fillet can then be considered as effective, but it is not permissible to consider, in the stress calculation, any additional strength obtained by the length of the weld return.

The total length of a fillet weld can only be considered in stress calculations if the weld is continuous, as in Fig. 93, where the effective length would be taken as $2l_1 + l_2$.

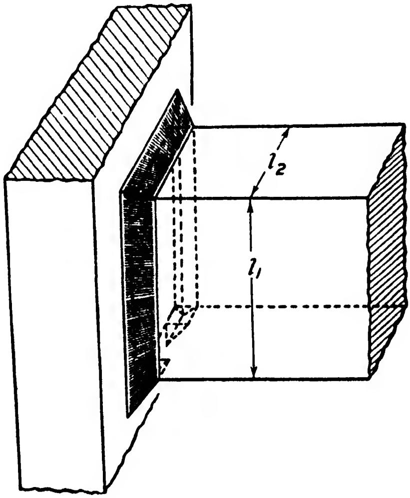


FIG. 93

Typical endless fillet weld connections are used when welding wheels or brake drums to shafts, or when welding fixed shafts to supporting brackets. Bending or torsional shear stresses will occur in such cases.

(12) (Fig. 94) :

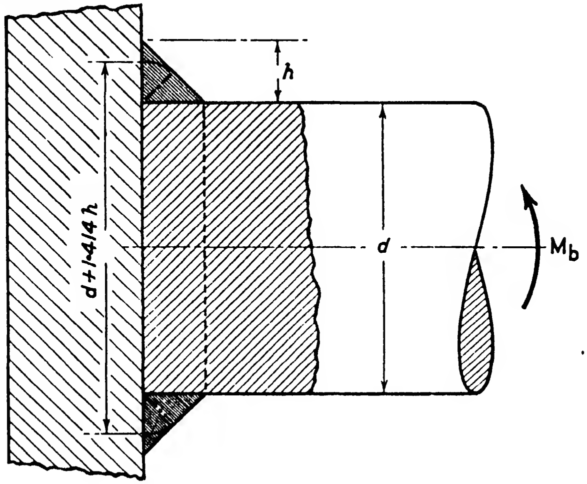


Fig. 94

Considering bending forces only,

$$f_b = \frac{M_b}{Z_w},$$

$$Z_w = \frac{2}{(d+1{\cdot}414h)} \times \left[\frac{\pi(d+1{\cdot}414h)^4}{64} - \frac{\pi d^4}{64}\right]$$

$$= \frac{\pi}{32(d+1{\cdot}414h)} \times \left[(d+1{\cdot}414h)^4 - d^4\right]$$

$$= \frac{(d+1{\cdot}414h)^4 - d^4}{10{\cdot}2(d+1{\cdot}414h)}$$

or

$$f_b = \frac{10{\cdot}2M_b(d+1{\cdot}414h)}{(d+1{\cdot}414h)^4 - d^4} \quad . \quad . \quad . \quad . \quad (47)$$

In case of the value $\frac{h}{d}$ being small, the section modulus can be calculated approximately as

$$Z_{w\ \text{approx.}} = {\cdot}8(d+{\cdot}707h)^2({\cdot}707h)$$

$$= {\cdot}566h(d+{\cdot}707h)^2$$

or

$$f_{b\ \text{approx.}} = \frac{M_b}{{\cdot}566h(d+{\cdot}707h)^2}$$

$$= \frac{1{\cdot}76M_b}{h(d+{\cdot}707h)^2}. \quad . \quad . \quad . \quad . \quad (48)$$

(13) (Fig. 95) : Considering shear forces only,

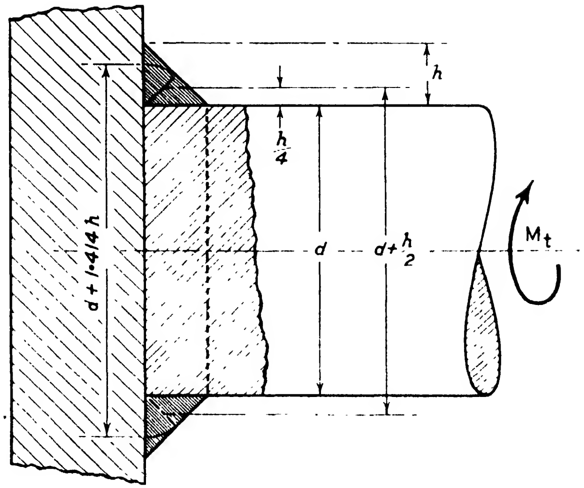


FIG. 95

$$f_s = \frac{M_t}{Z_{pw}},$$

$$Z_{pw} = \frac{2}{(d+1{\cdot}414h)}\left[\frac{\pi(d+1{\cdot}414h)^4}{32} - \frac{\pi d^4}{32}\right]$$

$$= \frac{\pi}{16(d+1{\cdot}414h)}\left[(d+1{\cdot}414h)^4 - d^4\right]$$

or $$f_s = \frac{5{\cdot}1 M_t(d+1{\cdot}414h)}{(d+1{\cdot}414h)^4 - d^4}. \qquad (49)$$

As an approximation, it can be assumed that the average shear force transmitted at the centre of gravity of the weld throat is

$$P_s = \frac{2M_t}{d+\frac{h}{2}}.$$

The throat area is

$$A_w = \pi\left(d+\frac{h}{2}\right)(\cdot707h)$$

$$= 2{\cdot}23h\left(d+\frac{h}{2}\right);$$

also $$f_{s\ \text{approx.}} = \frac{P_s}{A_w},$$

and $$f_{s\text{ approx}} = \frac{2M_t}{\left(d+\frac{h}{2}\right)2{\cdot}23h\left(d+\frac{h}{2}\right)}$$

or $$f_{s\text{ approx}} = \frac{{\cdot}9M_t}{h\left(d+\frac{h}{2}\right)^2} \quad . \quad . \quad . \quad . \quad (50)$$

(14) (Fig. 96) :

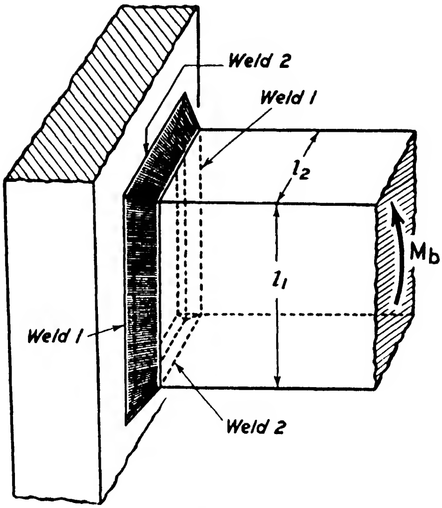


FIG. 96

For a rectangular shaft subject to bending, assuming that the maximum stress in all welds is the same, the following approximate calculation can be used : *

If welds 1 take part moment M_{b1} and welds 2 take part moment M_{b2} of the total moment M_b, the maximum stress in welds 1 is given by

$$f_{1\text{ max}} = \frac{4{\cdot}24M_{b1}}{l_1^2h} \quad \text{(see equation 31),}$$

and the maximum stress in welds 2 by

$$f_{2\text{ max}} = \frac{1{\cdot}414M_{b2}}{l_1l_2h} \quad \text{(see equation 43).}$$

As assumed above,

$$f_{1\text{ max}} = f_{2\text{ max}}$$

or $$\frac{4{\cdot}24M_{b1}}{l_1^2h} = \frac{1{\cdot}414M_{b2}}{l_1l_2h}.$$

* This method has been suggested by C. H. Jennings, "Welding Design", *Transactions of the A.S.M.E.*, October 1936, page 497.

Hence $$M_{b2}=M_{b1}\frac{4{\cdot}24}{1{\cdot}414}\frac{l_2}{l_1}$$
$$=M_{b1}\frac{3l_2}{l_1}.$$

But $$M_b=M_{b1}+M_{b2}$$
$$=M_{b1}+M_{b1}\times\frac{3l_2}{l_1}$$
$$=M_{b1}\times\left[1+\frac{3l_2}{l_1}\right]$$

or $$M_{b1}=\frac{M_b}{1+\frac{3l_2}{l_1}}.$$

Also $$f_{max}=f_{1\,max}=\frac{4{\cdot}24M_b}{\left(1+\frac{3l_2}{l_1}\right)l_1^2h}$$

or $$f_{max}=\frac{4{\cdot}24M_b}{h[l_1^2+3l_1l_2]}. \qquad (51)$$

(15) (Fig. 97) :

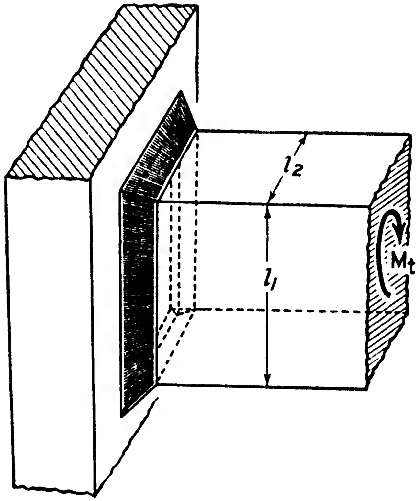


FIG. 97

For a rectangular shaft subject to torsion a calculation similar to that for a circumferential butt weld (see butt welds case 17, page 58) can be applied :

$$f_{max}=f_s=\frac{M_t}{2A(\cdot707h)},$$

where, however,

$$A = (l_1 + \cdot707h) \times (l_2 + \cdot707h) \quad \text{(Fig. 98).}$$

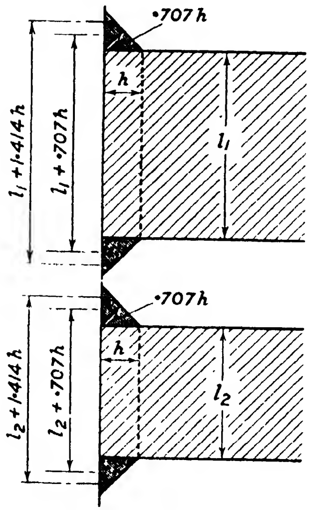


FIG. 98

Therefore

$$f_{\max} = \frac{M_t}{2(l_1 + \cdot707h) \times (l_2 + \cdot707h) \times (\cdot707h)}$$

$$= \frac{\cdot707M_t}{h(l_1 + \cdot707h) \times (l_2 + \cdot707h)} \cdot \qquad (52)$$

(16) (Fig. 99)

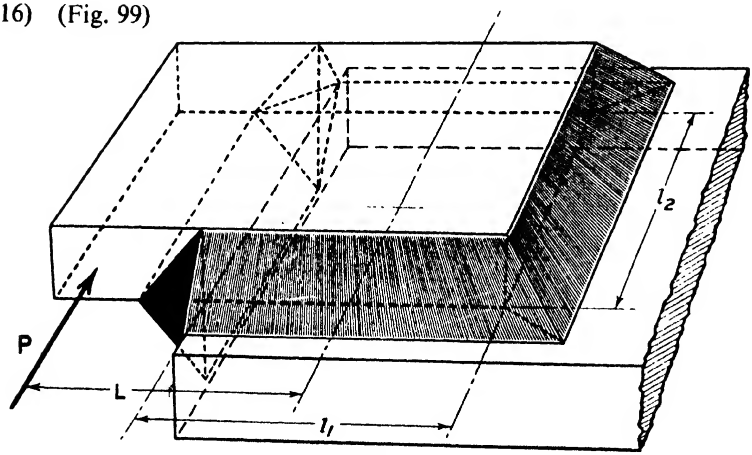


FIG. 99

In the case shown here, for determining the torsional stress, the method used in case 15 can be applied by substituting PL for M_t.

Hence
$$f_{s\ (\text{torque})\ \max} = \frac{\cdot 707PL}{h(l_1 + \cdot 707h) \times (l_2 + \cdot 707h)} \qquad (53)$$

The maximum shear stress in the centre of the welds parallel to P is: *

$$f_{s\ \max} = \frac{P}{2 \times (\cdot 707 l_2 h)}$$
$$= \frac{\cdot 707P}{l_2 h} \qquad (54)$$

(17) (Fig. 100):

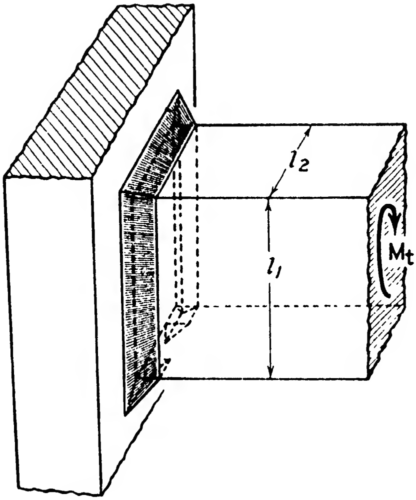


FIG. 100

If the continuous weld is only carried round three sides of the rectangular shaft, the torsional stress is: †

$$f_{s\ (\text{torque})} = \frac{3M_t}{l_o \times (\cdot 707h)^2}$$

where
$$l_o \approx 2l_1 + l_2 + h,$$

hence
$$f_{s\ (\text{torque})} = \frac{6M_t}{(2l_1 + l_2 + h) \times h^2} \qquad (55)$$

(18) (Figs. 78 and 91):

In cases where only two parallel fillet welds carry a torque, various

* Neglecting the area of the welds normal to P (see Timoshenko, *Strength of Materials*).

† See C. Weber, "Die Lehre der Drehungsfestigkeit" ("Theory of Torsional Strength"), *V.D.I.*, Berlin, 1921.

handbooks suggest the calculation of the maximum stress on the basis that the polar moment of inertia I_p of the weld and the distance r_{max} of the extreme fibre from the centre of gravity of the weld are related by the equation

$$f_{max} = \frac{PLr_{max}}{I_p}.$$

It is suggested that such a calculation should be used with great reserve, as it is correct only for the case of circular and similar sections.

B. Resistance Welded Joints

The outstanding feature of the resistance welding processes is the speed with which the joints can be produced once the welding machines have been correctly set. Furthermore, with modern equipment which automatically controls all the factors (current, pressure, and timing) which influence the weld, unskilled labour can be used to operate the welding machines.

The following points, however, must not be overlooked :

Variations in the composition of the material, discrepancies in the shape of the components, as well as inaccuracies in the setting of the welding machine and carelessness in the maintenance of jigs and welding electrodes, will influence the quality and soundness of the welded joints and may eliminate the possible advantages.

Resistance welded joints can usually only be checked by destructive tests, and it is more or less standard practice, prior to putting a batch of components through production, to break a typical test piece which is similar to the component to be produced and which has been welded under conditions identical, so far as possible, with those occurring on the actual job. It is especially important that the material of the test piece be identical with that of the actual component and that the welding settings and the amount of metal in the throat of the welding machine are the same as will occur on the actual job.

The necessity for preparing and breaking test pieces periodically during the production process, the need for close control over the quality and state of the tools and over the working material and components, are all reasons for resistance welding being most economical when used for large quantity production.

The use of the various types of resistance welding processes is limited by the capacity of the available plant : the pressure and current available at the spot welding machine limit the thickness of plate which can be welded, the capacity of the projection welding machine or of the butt welding machine determines the maximum weld area which can be produced, and the capacity of presses available in the shop may limit the size of material into which projections can be pressed.

Some general considerations with regard to the application of the various resistance welding processes are as follows :

I. Spot Welding (see Fig. 21)

This is applicable to the joining of components made from plate material, the plate thickness being limited by the pressure and current capacity of the available spot welding machine. This method has an advantage over projection welding in cases where the number of welds required is relatively small, when the size of the job is too large for the capacity of projection welders, or where the design allows the necessary spacing and arrangement of the welds to be obtained. The lower cost of the equipment makes spot welding economically advantageous for the production of smaller quantities.

Seam welding is a continuous spot welding operation and creates a series of overlapping spots when used for pressure-tight joints.

II. Projection Welding (see Fig. 22)

This allows more than one spot weld to be produced in one operation, the number being determined by the number of projections raised in one plate. The projection welding process is advantageous, therefore, for joining, in large quantities, relatively small components made of plate material. As all welds are produced at the same time, the shunting effect between neighbouring welds is not detrimental (the current supply being sufficient for all welds), and the spacing between the spots is not critical, as in the case of spot welding. The process should, therefore, be used where size limitations make it impossible to space spot welds sufficiently to avoid one weld being electrically shunted by the one previously made.

The tooling cost is obviously higher for projection welding than it is for spot welding.

Projection welding is invaluable for welding studs, nuts, or bosses to plates, the speed of operation exceeding that of any other welding method.

III. Butt Welding (see Fig. 23)

This is mainly confined to the welding of wires or thin flat material.

IV. Flash Butt Welding (see Fig. 24)

This process creates a joint which has practically the full strength of the parent metal under static loading conditions and a strength slightly less than that of the parent metal under dynamic loading. It is particularly useful in cases of large-quantity production where large forgings can be replaced by smaller forgings welded to standard rolled sections. With some special precautions, such as preheating and heat treatment after welding, alloy steels can be welded.

When considering the use of resistance-welded joints, the designer must be aware of their specific requirements with regard to design lay-out (see page 84), weld specifications (see page 199), material specification and preparation (see page 25).

In order to obtain reliable welds with the spot-welding process it is necessary, apart from the correct welding procedure, to have the plates lying flat against each other to ensure good contact. Any distortion or inaccuracy in the plate surface would have to be overcome by a portion of the welding pressure, which portion would then be lost to the actual welding process.

The material surfaces to be welded must be free from mill scale or corrosion (see page 25).

All these points can be ensured only if they are specified and indicated in the instructions to the workshop. They are discussed in detail in the appropriate chapters, but it has been considered necessary to mention them here, because, if they are not carefully observed, it will be impossible to obtain welds which fulfil their required task in accordance with the designer's calculations and expectations.

Spot- and projection-welded joints should be arranged in such a way as to be stressed in shear. Tensile loads create a tearing effect on the welded joint and do not give the reliable strength characteristics required. Experiments have proved that, for welds in mild steel (28–32 tons/sq. inch, with less than ·2 per cent C), 20 tons/sq. inch is a reliable ultimate shear stress value for spot and projection welds under static load. With a recommended safety factor of 4, the permissible shear stress is 5 tons/sq. inch.*

The size and arrangement of spot welds depends not only upon strength considerations but also upon the requirements of the welding process.

The weld diameter d, which is approximately equal to the electrode tip diameter, is usually determined by the equation $d=\sqrt{t}$, where t is the thickness of the plate material to be welded.

The spacing between two welds should be not less than $3 \times d$ nor less than $\frac{1}{2}''$, whichever is the higher value, in order to avoid an electric shunting effect between neighbouring welds. In order to maintain plate stability, it is recommended † that a spacing of 12 times the plate thickness in the case of single row spot welds should not be exceeded, or 18 times the plate thickness in the case of staggered row spot welds.

* *Welding Memorandum*, No. 4c, Advisory Service on Welding, Ministry of Supply, London. † *Ibid.*

This does not apply to the case of sheets below 16 S.W.G., where the consideration of an electric shunting effect requires larger spacings. It is pointed out, however,* that designs using such thin sheets are not likely to be affected by questions of plate stability.

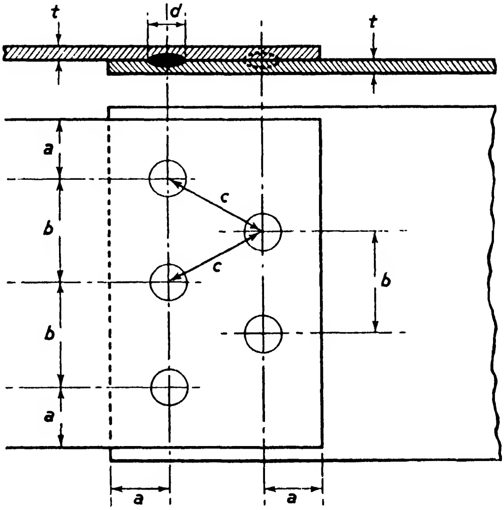


FIG. 101

t		$d_{min.}$	$a_{min.}$	$b_{min.}$	$c_{min.}$	$b_{max.}$ Single row	$b_{max.}$ 2 or more rows
S.W.G.	In.	In.	In.	In.	In.	In.	In.
24	·022	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
22	·028	$\frac{3}{16}$	$\frac{4}{16}$	$\frac{9}{16}$	$\frac{9}{10}$	$\frac{9}{16}$	$\frac{5}{8}$
20	·036	$\frac{3}{16}$	$\frac{5}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$
18	·048	$\frac{3}{16}$	$\frac{5}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$
16	·064	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{8}$
14	·080	$\frac{5}{16}$	$\frac{1}{2}$	$\frac{15}{16}$	$\frac{15}{16}$	$1\frac{1}{4}$	$1\frac{7}{8}$
12	·104	$\frac{5}{16}$	$\frac{1}{2}$	$\frac{15}{16}$	$\frac{15}{16}$	$1\frac{1}{4}$	$1\frac{7}{8}$
10	·128	$\frac{3}{8}$	$\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{2}$	$2\frac{1}{4}$
	$\frac{3}{16}$	$\frac{7}{16}$	$\frac{11}{16}$	$1\frac{5}{16}$	$1\frac{5}{16}$	$2\frac{1}{4}$	$3\frac{3}{8}$
	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{2}$	3	$4\frac{1}{2}$
	$\frac{3}{8}$	$\frac{5}{8}$	$\frac{15}{16}$	$1\frac{7}{8}$	$1\frac{7}{8}$	$4\frac{1}{2}$	$6\frac{3}{4}$

The edge distance should be at least 1·5 times the diameter. This will result in the welds shearing prior to the plates being torn through to their edges. The table (Fig. 101) shows the main dimensions to be observed in spot-welded designs.

* *Welding Memorandum*, No. 4c.

Figs. 102–5 show typical projections as used on sheet material (Fig. 102), studs (Fig. 103), bosses (Fig. 104), and hexagon nuts (Fig. 105).

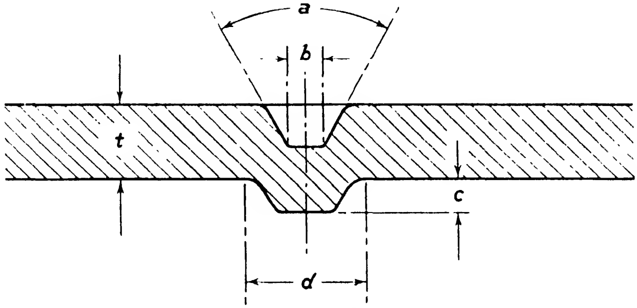


FIG. 102

t min.	t max.	a	b	c	d
In.	In.	Degrees	In.	In.	In.
·020	·040	90	·030	·030	·125
·030	·045	90	·040	·034	·144
·045	·070	90	·050	·040	·170
·070	·100	90	·080	·046	·192
·100	·140	40	·080	·055	·172
·140	·190	40	·080	·065	·185
·190	·250	40	·080	·075	·250

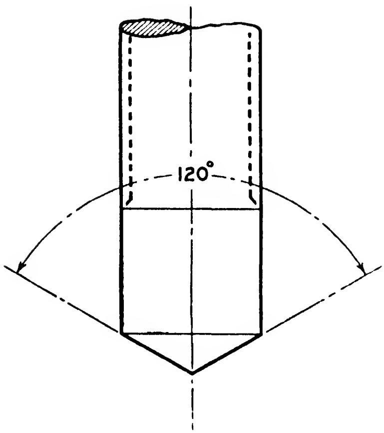
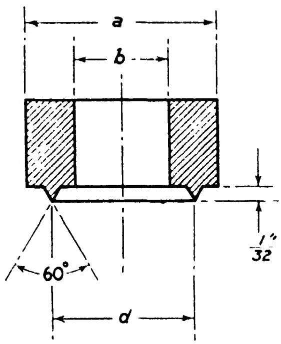


FIG. 103



$$d=\frac{a+b}{2}.$$

FIG. 104

Projection for bosses up to 1" diameter

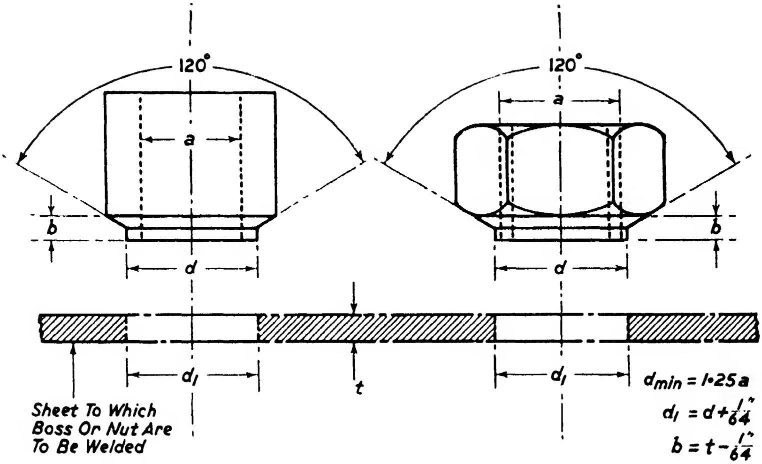


FIG. 105

In the case of flash butt-welded joints it must be remembered that the shape and area of the two components must be identical at the point to be joined, and that the required upset allowance must be made (Fig. 106).

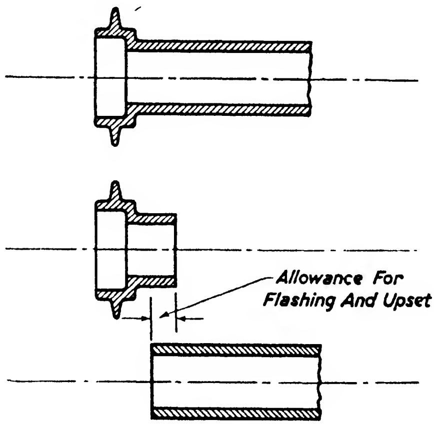


FIG. 106

Stress Calculation for Spot Welds.

As previously mentioned, spot welds should be arranged to carry loads in shear (see page 83). The shear stress in a spot weld of diameter d under a load P (Fig. 107) is

$$f_s = \frac{P}{\frac{\pi d^2}{4}} = \frac{4P}{\pi d^2}$$

$$= \frac{1{\cdot}28P}{d^2} \quad . \quad . \quad . \quad . \quad . \quad (56)$$

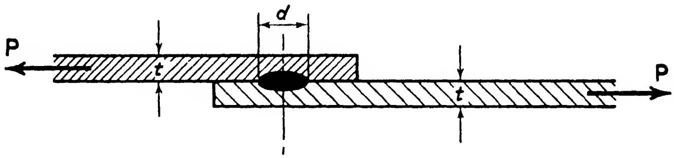


FIG. 107

As the forces P are applied through the centre of gravity of the plates, and not in the plane of the contacting surfaces, a bending couple is exerted, and unless the plates are stiffened and held firmly in position. bending will occur until the forces P in both plates have one line of action and are in equilibrium (Fig. 108). A tensile stress, in addition to shear, will then occur in the weld, its amount being determined by angle α.

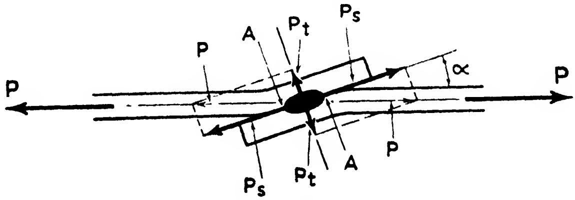


FIG. 108

The shear force applied to the weld will be

$$P_s = P \cos \alpha,$$

the tensile force

$$P_t = P \sin \alpha,$$

the shear stress in the weld

$$f_s = \frac{1{\cdot}28P_s}{d^2} = \frac{1{\cdot}28P}{d^2} \times \cos \alpha,$$

and the tensile stress

$$f_t = \frac{1{\cdot}28P_t}{d^2} = \frac{1{\cdot}28P}{d^2} \times \sin \alpha.$$

The ratio p of the tensile stress to the shear stress expressed in per cent is given by

$$p=\frac{f_t}{f_s}\times 100$$

$$=\frac{\dfrac{1{\cdot}28\mathrm{P}}{d^2}\times\sin\alpha}{\dfrac{1{\cdot}28\mathrm{P}}{d^2}\times\cos\alpha}\times 100$$

or $$p=100\tan\alpha,$$

so that to keep p as small as possible, α must be kept to a minimum.

The bending moment acting on each plate is (Fig. 109)

$$\mathrm{M}_b=\frac{\mathrm{P}t}{2}.$$

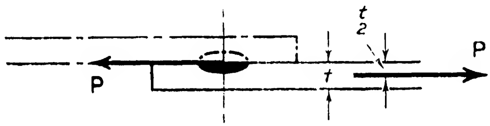


FIG. 109

If the moment $\frac{\mathrm{P}t}{2}$ stresses the material beyond its yield point, a sharp bend will occur near points A (Fig. 108) where the plates are not held together by the welds, and the plates will take the forms indicated.

In a spot-welded structure which is fully stressed, the bending moment $\frac{\mathrm{P}t}{2}$ is limited by the maximum shear force P_{max} which can be transmitted by the weld without stressing it above the permissible limit $f_{s\ max}$,

or $$\mathrm{P}_{max}=\frac{f_{s\ max}\,d^2}{1{\cdot}28}\text{ (see equation 56).}$$

Considering a plate of minimum width in relation to the spot weld (plate edge distance from centre of weld $=1{\cdot}5d$ on either side), the section modulus of the plate section is given by

$$\mathrm{Z}_p=\frac{3dt^2}{6}=\frac{dt^2}{2}$$

and the bending stress in the plate material by

$$f_b=\frac{\mathrm{M}_{b\ max}}{\mathrm{Z}_p}=\frac{\dfrac{\mathrm{P}_{max}\,t}{2}}{\dfrac{dt^2}{2}}=\frac{\mathrm{P}_{max}}{dt}$$

$$=\frac{f_{s\ max}\,d}{1{\cdot}28t}.$$

When $$f_{s\,max} = 5 \text{ tons/sq. inch,}$$
the maximum bending stress in the plate, is
$$f_b = 3{\cdot}9\frac{d}{t}.$$
The plate material will be stressed beyond its yield point (16 tons/sq inch for low carbon sheet steel as used in spot-welded assemblies) if
$$3{\cdot}9\frac{d}{t} > 16,$$
that is $$\frac{d}{t} > 4{\cdot}1,$$
and since $$d = \sqrt{t}$$
then $$\frac{1}{\sqrt{t}} > 4{\cdot}1$$
and $$t < {\cdot}06.$$
For the material below 16 S.W.G., therefore, bending as shown in Fig. 108 will occur, and α will then be given by
$$\tan \alpha = \frac{t}{\sqrt{l^2 - t^2}},$$
when l is the length of the overlap area held together by the weld. The percentage of tensile to shear stress in the spot weld will be given by
$$p = \frac{100t}{\sqrt{l^2 - t^2}}.$$
In the case of single-row welds, l is equal to the diameter of the weld (see Fig. 108) and
$$p_1 = \frac{100t}{\sqrt{d^2 - t^2}}.$$
In case of two rows of spot welds
$$\left(l = 3d + \frac{d}{2} + \frac{d}{2} = 4d\right)$$
and $$p_2 = \frac{100t}{\sqrt{16d^2 - t^2}}.$$
In Fig. 110 p_1 and p_2 are plotted in relation to t, and for the minimum diameter of the weld given by $d = \sqrt{t}$. It will be seen that for two rows of welds the percentage of tensile stress will not be more than 6·5 per cent in the most unfavourable case, which can be considered satisfactory.

In the case of material above 16 S.W.G., where the yield stress will not be reached under the greatest possible bending moment, the plates under the most unfavourable conditions will take the shapes shown in Fig. 111. Here
$$\tan \alpha = \frac{\sqrt{\left(R + \frac{t}{2}\right)^2 - R^2}}{R}$$

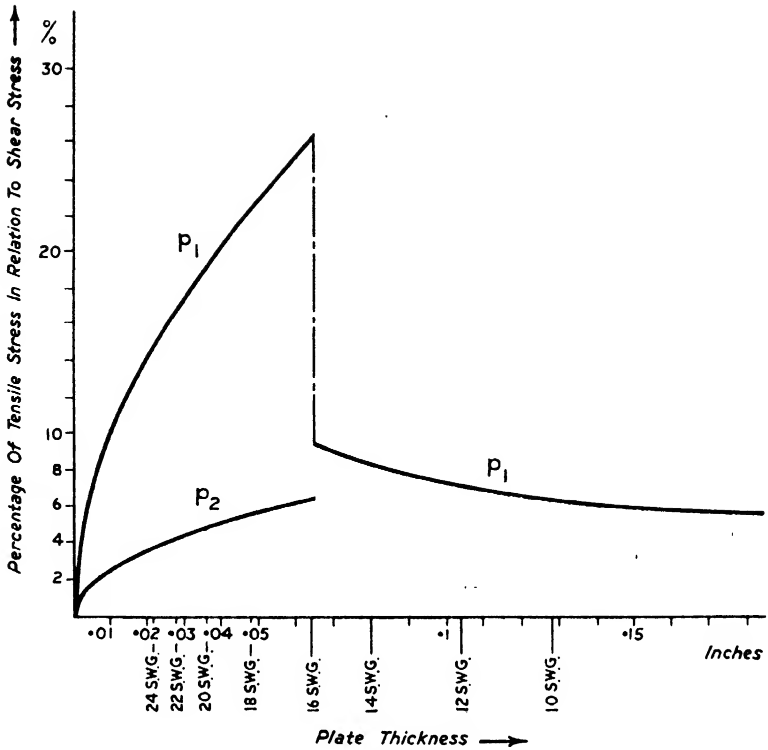


FIG. 110

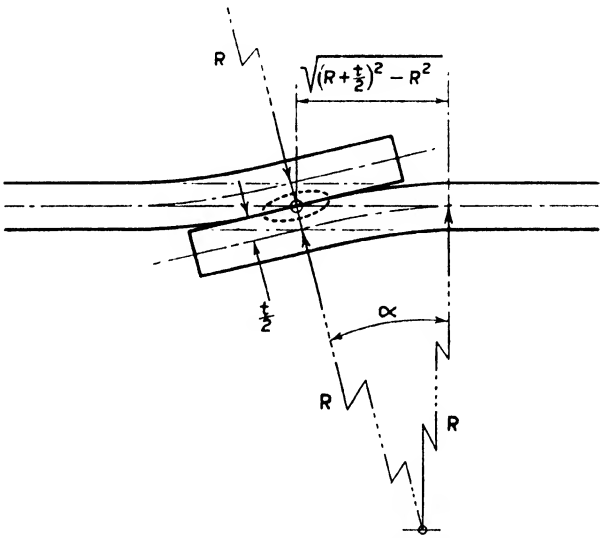


FIG. 111

$$=\frac{\sqrt{Rt+\frac{t^2}{4}}}{R}=\sqrt{\frac{t}{R}+\frac{t^2}{4R^2}}$$

and
$$\frac{1}{R}=\frac{M_b}{EI},$$

in which expression
$$I=\frac{3dt^3}{12}=\frac{dt^3}{4}.$$

Again,
$$M_{b\ \max}=\frac{f_{s\ \max}\,d^2t}{2\cdot 56}$$

and
$$\frac{1}{R}=\frac{1\cdot 56 f_{s\ \max}\,d}{Et^2}=\frac{1\cdot 56\times 11200d}{30000000t^2}$$

$$=\cdot 00058\frac{d}{t^2}.$$

Hence
$$\tan\alpha=\sqrt{\cdot 00058\frac{d}{t}+\cdot 000000084\frac{d^2}{t^2}},$$

and
$$p=100\sqrt{\cdot 00058\frac{d}{t}+\cdot 000000084\frac{d^2}{t^2}},$$

$$p\simeq 2\cdot 4\sqrt{\frac{d}{t}}.$$

This applies in the case of single joints (p_1) and is also plotted in relation to t in Fig. 110. It will be seen that, in the case of plates above 16 S.W.G., the percentage of tensile to shear stress will be below 10 per cent. For these thicker plates, therefore, it is not necessary to use more than one row of welds in order to keep the tensile stress down to a reasonable value. If minimum deformation is required, however, multi-row welds will lengthen the connected portion of the plates and reduce α.

Chapter 3

GENERAL DESIGN PRINCIPLES FOR WELDED CONSTRUCTION

IN the design of steel plates and other component parts for welded fabrications, due consideration must be given to the best use of the properties of the material to be used, and to the necessities and possibilities created by the welding process itself.

The calculations of stresses, deformations, rigidity, and stability of the component parts and of the completed structure are identical, in principle, with those in normal use. If it is intended to obtain a high degree of accuracy in strength calculations for such articles as machines, however, theoretical calculations will be rather complicated, as most machine bodies, gear boxes, bases, and bedplates are not as simple as the majority of cases which are published in hand-books as examples of calculations of strength and stiffness.

To overcome this difficulty in a manner which is acceptable in practice, it is probably best to assume an extreme case of such unfavourable conditions as may arise, and which at the same time is susceptible to a more or less accurate—and not too complex—theoretical calculation.

Such an assumption will not give accurate results but will afford valuable assistance in estimating the order of magnitude of distortions and stresses to which the part or the structure in question may be subjected.

If such fundamental calculations can be condensed into graphic form they will give the designer a ready guide as to the influence which wall thicknesses, working length, depth, and other factors of the shape of the part under consideration will have upon the strength, stiffness, and probable material consumption, even if every design problem which might be encountered is not covered.

The influence of the properties of the material upon the shape of a structure for which strength and stiffness are to be used to the fullest extent, has been mentioned on page 8.

The graph * (Fig. 112) shows these relations for the case of a beam simply supported at the ends and loaded at the centre. As before (page 8), only deflection and stresses due to bending have been considered in the calculations.

* From F. Koenigsberger, " The Application of Fabricated Construction to Machine Design ", *Proceedings of the Institution of Mechanical Engineers*, 1945, vol. 152, page 245.

The following symbols are used :

A	= area of beam section	sq. inches
E	= modulus of elasticity of material	lb./sq. inch
I	= moment of inertia of beam section	inches^4
S	= specific gravity of material	lb./cu. inch
U	= energy of shock load	inch. lb.
V	= volume of beam	cu. inches
W	= weight of beam	lb.
W_d	= static load on beam	lb.
Z	= section modulus of beam section	inches^3
a	= velocity of sound in beam material	inch./sec.
b	= width of beam	inches
d	= depth of beam	inches
f_b	= bending stress	lb./sq. inch
f_o	= natural frequency of transverse vibration of beam structure	(minutes)$^{-1}$
g	= acceleration due to gravity	inches per sec. per sec.
k	= radius of gyration of beam section	inches
l	= length of beam	inches
δ	= deflection of beam under load	inches

In this case

$$\delta = \frac{W_d l^3}{48EI}$$

or as

$$I = \frac{bd^3}{12},$$

$$\delta = \frac{W_d l^3}{4Ebd^3}.$$

But since $V = bdl$, we can write

$$\delta = \frac{W_d l^4}{4EVd^2}.$$

Hence

$$\frac{W_d}{\delta} = 4EV\left(\frac{d}{l^2}\right)^2. \quad . \quad . \quad . \quad . \quad (57)$$

Now

$$f_b = \frac{W_d l}{4Z},$$

or as

$$Z = \frac{bd^2}{6};$$

$$f_b = \frac{3W_d l}{2bd^2}.$$

Thus

$$f_b = \frac{3W_d l^2}{2Vd}$$

and

$$W_d = \frac{2}{3} f_b V \frac{d}{l^2}. \quad . \quad . \quad . \quad . \quad (58)$$

Furthermore, with $a^2 = \frac{Eg}{S}$ and $k^2 = \frac{I}{bd}$,

the minimum natural frequency per minute becomes

$$f_0 = \frac{60}{2\pi} \frac{\pi^2}{l^2} ak = \frac{30\pi}{l^2} \times \sqrt{\frac{EIg}{bdS}},$$

and substituting the value $I = \frac{bd^3}{12}$,

$$f_0 = \sqrt{\frac{900\pi^2 d^2 Eg}{12 S l^4}}$$

$$= \sqrt{285100 \frac{E}{S} \left(\frac{d}{l^2}\right)^2}$$

$$= 534 \sqrt{\frac{E}{S}} \times \frac{d}{l^2}. \qquad (59)$$

The importance of the factor $\frac{d}{l^2}$, which might be called the " degree of compactness ", will be clearly seen from equations (57), (58), and (59). The stiffness $\frac{W_d}{\delta}$ depends upon a material factor (the modulus of elasticity), the volume of material employed, and the square of the degree of compactness. The strength (the maximum permissible load determined by the permissible bending stress) depends upon a material factor (the permissible maximum bending stress), the volume of material employed, and the first power of the " degree of compactness ". The natural frequency depends upon a material factor (the square root of the modulus of elasticity divided by the specific weight) and the degree of compactness.

It should not be overlooked that stiffness is not always the decisive factor in a machine structure. In cases where shock loads are likely to occur, high stiffness against deformation may be dangerous, as the energy of the shock has then to be absorbed over a short distance, and the stressing of the structure must be correspondingly high.

If shock energy U is applied to a beam, causing a deflection δ, the equivalent static load to cause the same deflection is given by W_d where

$$\frac{W_d \delta}{2} = U \quad \text{or} \quad W_d = \frac{2U}{\delta}. \qquad (60)$$

In comparing the shock loading of similar structures of, say, mild steel and cast iron, though the permissible stress for the former is higher than for the latter, its higher modulus of elasticity, and consequent smaller deformation for a given load, will to some extent offset its initial advantage. The permissible shock load U for the beam can be found from equations (58) and (60) in the following way. The permissible equivalent static load is

$$W_d = \frac{2}{3} f_b V \frac{d}{l^2} \quad . \quad . \quad \text{(from equation 58)}$$

$$= \frac{2U}{\delta} \quad . \quad . \quad . \quad . \quad . \quad (60)$$

or

$$\frac{2U}{\delta} = \frac{2}{3} f_b V \frac{d}{l^2}.$$

Hence

$$U = \frac{f_b}{3} \delta V \frac{d}{l^2}, \quad . \quad . \quad . \quad . \quad (61)$$

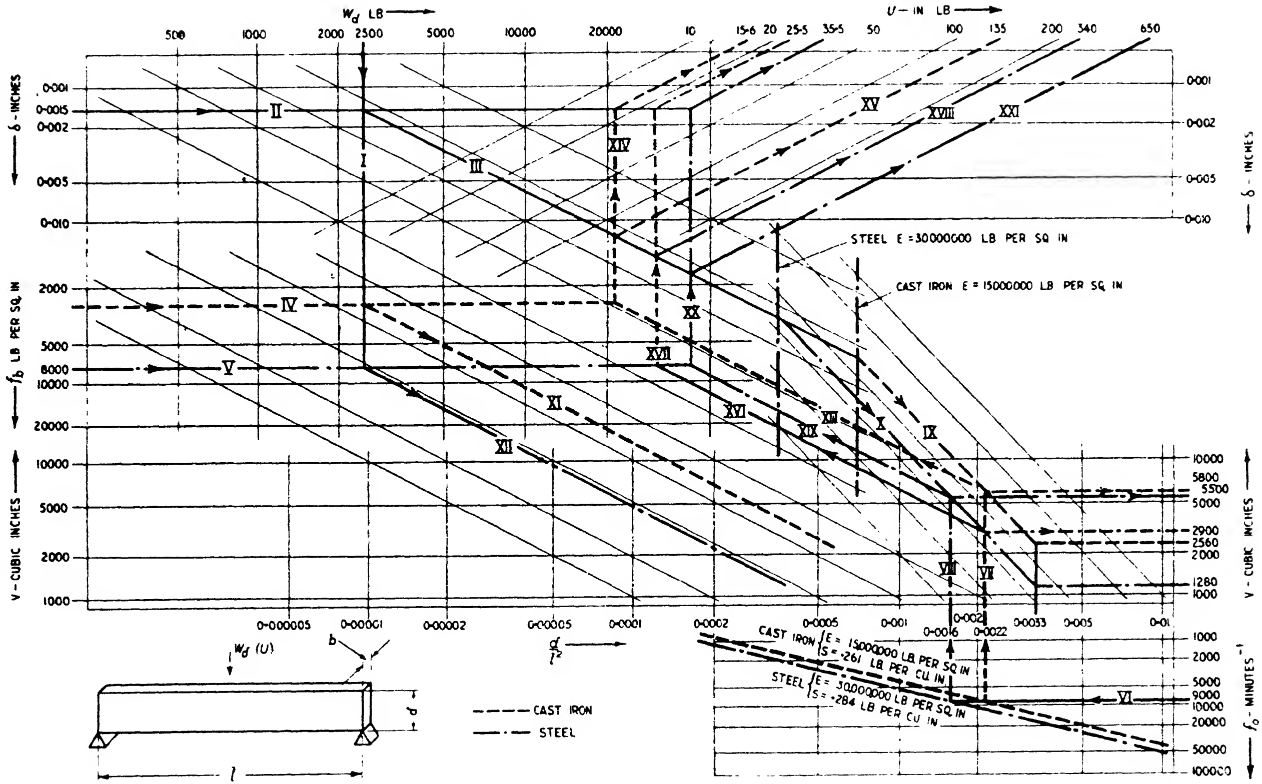
showing that the permissible shock load is greater the greater the permissible stress and the greater the deflection of the beam under the load.

The graph (Fig. 112), which has been based on the foregoing formulae, not only gives an idea of the relations between the various factors to be considered, but also affords a comparison between the application of steel and cast iron.

The graph has been drawn in the double logarithmic system in order to obtain straight lines for all equations, and has been developed in the following manner. In the top left-hand corner W_d and δ give the scales of the $\frac{W_d}{\delta}$ relation (stiffness) on the lines sloping thence to the bottom right-hand corner. These lines intersect with the two thick vertical lines for $E = 30{,}000{,}000$ lb. per sq. inch (steel) and $E = 15{,}000{,}000$ lb. per sq. inch (cast iron), these figures having been considered sufficiently accurate for the purpose of this comparison. The lines sloping downwards at an angle of 45° to these vertical lines towards the bottom right-hand corner, give the values for $\frac{W_d}{4E\delta}$, and therefore constitute the factor $V\left(\frac{d}{l^2}\right)^2$, and from them can be found the corresponding values for V (vertical scale) and $\frac{d}{l^2}$ (horizontal scale). Where lines corresponding to the values of W_d and f_b intersect on the left-hand side of the diagram, the lines sloping down from left to right give the values $\frac{3W_d}{2f_b}$. These lines, therefore, represent the value $V \times \frac{d}{l^2}$, and from them, again, the corresponding values of V and $\frac{d}{l^2}$ can be found, which are required in connection with the estimation of the permissible stress.

In the right-hand bottom corner the relation between f_o and $\frac{d}{l^2}$ is given according to the values of $\frac{E}{S}$ for the two materials.

An example will show the use of the chart for the case of a machine-bed, which may be regarded as a form of freely supported beam. The



W_d LB
U — IN LB
δ - INCHES
f_b LB PER SQ IN
V - CUBIC INCHES
f_0 - MINUTES^{-1}
$\frac{d}{l^2}$
STEEL E = 30000000 LB PER SQ IN
CAST IRON E = 15000000 LB PER SQ IN
CAST IRON {E = 15000000 LB PER SQ IN, S = ·261 LB PER CU IN}
STEEL {E = 30000000 LB PER SQ IN, S = ·284 LB PER CU IN}
CAST IRON
STEEL

FIG 112

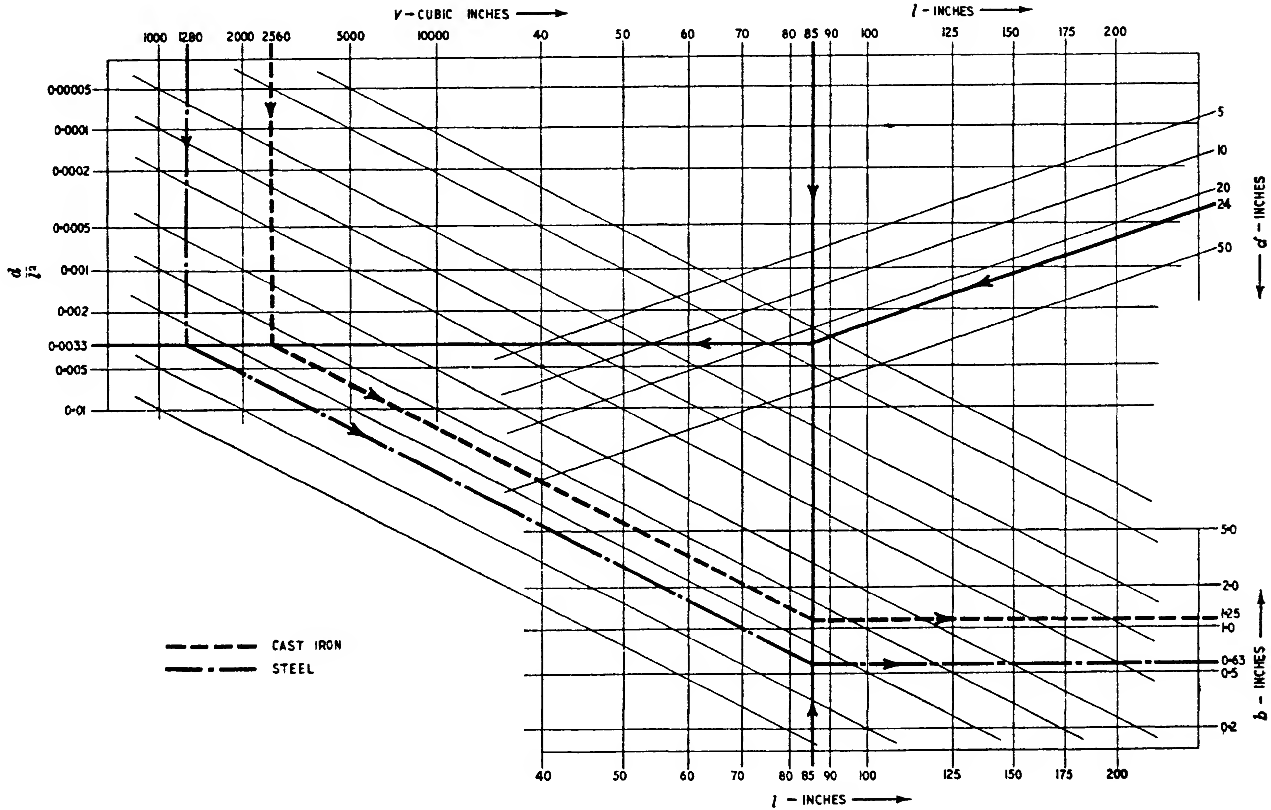

V – CUBIC INCHES
l – INCHES
1000
1280
2000
2560
5000
10000
40
50
60
70
80
85
90
100
125
150
175
200
0·00005
0·0001
0·0002
0·0005
0·001
0·002
0·0033
0·005
0·01
d – INCHES
5
10
20
24
50
b – INCHES
5·0
2·0
1·25
1·0
0·63
0·5
0·2
CAST IRON
STEEL

FIG. 113

H

arrangement of the machine may require a free length of 85 inches. For a maximum load applied in the centre, of 2,500 lb. (line I), the permissible deflection under load may be 0·0015 inch (line II). Line III represents $\frac{W_d}{\delta}$. In this case the permissible stresses may be assumed to be 2500 lb. per sq. inch for cast iron (line IV) and 8000 lb. per sq. inch for steel (line V). If the minimum natural frequency is to be 9000 min^{-1} (line VI), the minimum value of $\frac{d}{l^2}$ is 0·0022 for cast iron (line VII) and 0·0016 for steel (line VIII). IX and X are the $V\left(\frac{d}{l^2}\right)^2$ lines as determined by the deflections of the cast-iron and steel beams respectively. The intersection of lines VII and IX shows that for cast iron a volume of 5800 cu. inches will be required, whilst the intersection of lines VIII and X show a volume of 5500 cu. inches is needed for steel.

Within the range of $\frac{d}{l^2}$ shown on the graph, the lines (XI and XII), determined by the stress, lie well to the left of the above-mentioned points, indicating that the fullest use of all the material properties could only be obtained in the case of very large values of $\frac{d}{l^2}$. If the permissible height of the machine-bed allows a value as great as 0·0022 for $\frac{d}{l^2}$, then the steel structure could be built with a volume of only 2900 cu. inches (obtained from the intersection of lines VII and X).

With the degree of stiffness $\frac{W_d}{\delta}$ (in this case elasticity) which can be allowed in the structure, the permissible shock loads are found to be 135 inch lb. for cast iron (lines XIII, XIV, XV), or 340 or 650 inch lb. respectively for steel (lines XVI, XVII, XVIII, or XIX, XX, XXI). If the deflection is not to exceed 0·0015 inch, even under shock loads, then the permissible shock energy would be very small, *i.e.* 15·6 inch lb. for cast iron, or 25·5 or 35·5 inch lb. for steel.

Fig. 113 gives the relation between $\frac{d}{l^2}$, V, l, b, and d.

Taking the example of a beam 85 inches long and assuming the maximum permissible height within the structure of the machine to be 24 inches, Fig. 113 gives a value of 0·0033 for the quantity $\frac{d}{l^2}$. Both for steel and cast iron this value is well above the least figures demanded by considerations of minimum permissible natural frequency and maximum permissible stresses. For the requisite stiffness, the volume of material required is 2560 cu. inch for the cast-iron beam and 1280 cu. inch for the steel beam (Fig. 112).

Using these values in Fig. 113, the necessary width b of the beam is found to be 1·25 inches for cast iron and 0·63 inch for steel. If the beam is composed of two side walls, each would require to be about $\frac{5}{16}''$ thick if of steel and $\frac{5}{8}''$ if of cast iron. Whilst a wall thickness of $\frac{5}{16}''$ is easily obtainable with a steel structure, a value of $\frac{5}{8}''$ in a cast-iron structure, 85 inches in length and 24 inches in height, may create casting difficulties, thus making necessary a further increase in the volume of material, on account of the increased wall thickness.

A problem frequently encountered is that of a bracket with variable cross-section, *i.e.* a cantilever beam of uniform strength against bending (Fig. 114). If d is the depth, I the moment of inertia, and Z the section modulus of the beam section at the fixed end, the other symbols being used as before (see page 93), we have

$$\delta = \frac{2W_d l^3}{3EI} = \frac{8W_d l^3}{Ebd^3}$$

and
$$f_b = \frac{W_d l}{Z} = \frac{6W_d l}{bd^2}.$$

The shape of the beam follows a parabolic law, the depth y at any point at distance X from the load following the equation

$$y^2 = \frac{d^2}{l} \times X,$$

and the amount of material required for the beam is

$$V = \frac{2}{3}bdl.$$

Hence
$$\delta = \frac{16}{3} \times \frac{W_d l^4}{EVd^2}$$

and
$$\frac{W_d}{\delta} = \frac{3}{16} \times EV\left(\frac{d}{l^2}\right)^2.$$

Considering structures of steel only and taking the modulus of elasticity as practically constant (E = 30000000 lb./sq. inch),

$$\frac{W_d}{\delta} = 5625000V\left(\frac{d}{l^2}\right)^2 \quad . \quad . \quad . \quad (62)$$

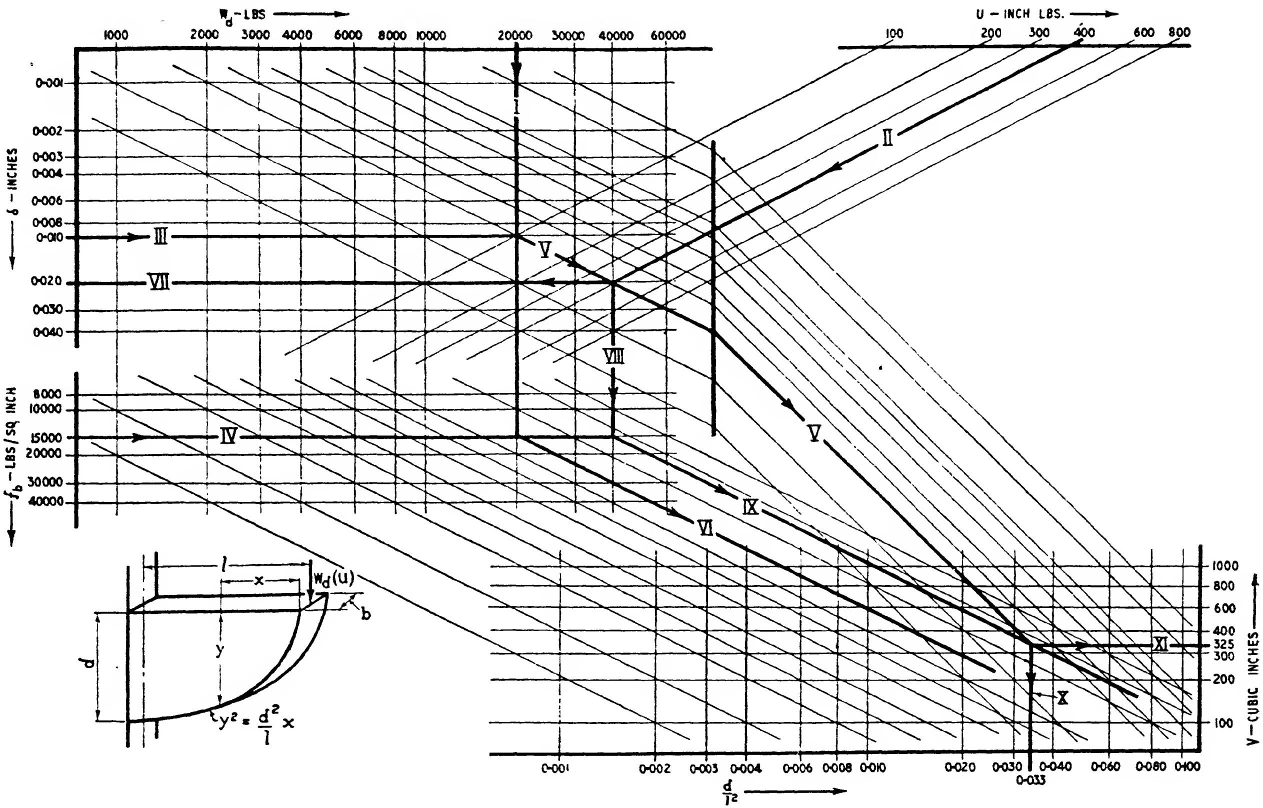
and
$$f_b = \frac{4W_d l^2}{Vd},$$

$$\frac{W_d}{f_b} = \frac{1}{4}V\frac{d}{l^2}, \quad . \quad . \quad . \quad . \quad (63)$$

$$U = \frac{W_d \delta}{2},$$

and
$$U = \frac{f_b}{8}\delta V\frac{d}{l^2}. \quad . \quad . \quad . \quad . \quad (64)$$

With equations (62), (63), and (64), graph Fig. 114 has been developed in a similar way as graph Fig. 112. The use of the graph Fig. 114 is shown by the following example :



W_d - LBS
U - INCH LBS.
δ - INCHES
f_b - LBS/SQ INCH
V - CUBIC INCHES
$\frac{d}{l^2}$
$y^2 = \frac{d^2}{l}x$
$W_d(u)$

FIG. 114

A double plate bracket of a shape that gives uniform bending stress over its whole length (Fig. 115) is to carry a load of 20,000 lb. (line I, Fig. 114) at a distance l of 24 inches from the fixed end. It is assumed

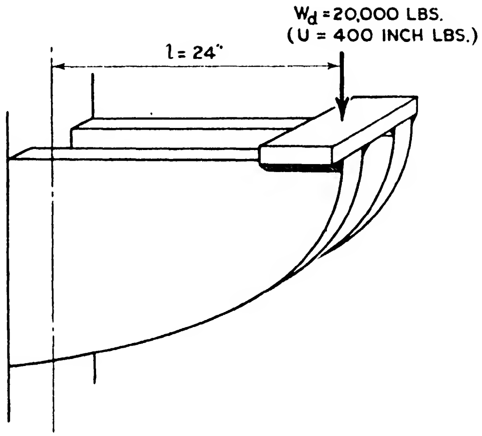


FIG. 115

that, due to vibrations in the structure, this load may also be applied by dropping it on to the bracket from a height of about ·020 inch, the shock energy being, therefore,

$$U = 20000 \times \cdot 020 = 400 \text{ inch lb.} \quad \text{(line II).}$$

The maximum permissible deflection at the loaded end of the bracket under the static load is ·010 inch (line III), the maximum permissible bending stress is 15,000 lb./sq. inch (line IV).

The intersection of lines I and III gives the required point $\frac{W_d}{\delta}$ on line V, which at the bottom half of the graph gives the required relation between V and $\frac{d}{l^2}$ from the point of view of stiffness under static load.

The intersection of lines I and IV gives the point $\frac{W_d}{f_b}$ on line VI, which at the bottom half of the graph gives the required relation between V and $\frac{d}{l^2}$ from the static stress point of view.

As the deflection is only limited for the static loading condition, we can allow greater deflection under the impact of the shock load U. The value of this deflection is obtained from the intersection of lines II and V, and is found to be ·020 inch (line VII). Going downwards (line VIII) from the intersection of lines II and V until line IV is reached, we find, from the intersection of lines IV and VIII, the line IX giving the relation between V and $\frac{d}{l^2}$ which is required to meet the stressing condition created by the shock load. It is obvious that the shock loading condition determines the minimum values of V $\frac{d}{l^2}$ with regard to stressing, and from the intersection of lines V and IX we find the values of $\frac{d}{l^2}$ and V at which full use is made of strength and stiffness of the material.

Hence	$\frac{d}{l^2} = \cdot 033$ (line X),
	$V = 325$ cu. inches (line XI).
As	$l = 24$ inches
we obtain	$d = 19$ inches,
and with	$V = 325$ cu. inches,
as	$l = 24$ inches
and	$d = 19$ inches,
we find that	$b = 1 \cdot 08$ inches,

which means that two walls of $\frac{9}{16}$ inch thickness each will be required.

The graphs (Figs. 112-14) refer to simple cases of beams with rectangular cross-section. More complex conditions are, however, likely to occur in general engineering and particularly in machine structures, and here the two factors which are generally of greatest importance are deformation and stress. The following equations apply to the more general case of beams under different conditions of support and of different cross-sections, with the only restriction that the cross-sections are constant over the length and symmetrical to the horizontal (neutral) axis of the beam :

$$\delta = C_1 \frac{W_d l^3}{EI},$$

$$f_b = C_2 \frac{W_d l}{Z},$$

C_1 and C_2 being constants which vary according to the conditions of support of the beam (see table, Fig. 116).

Since

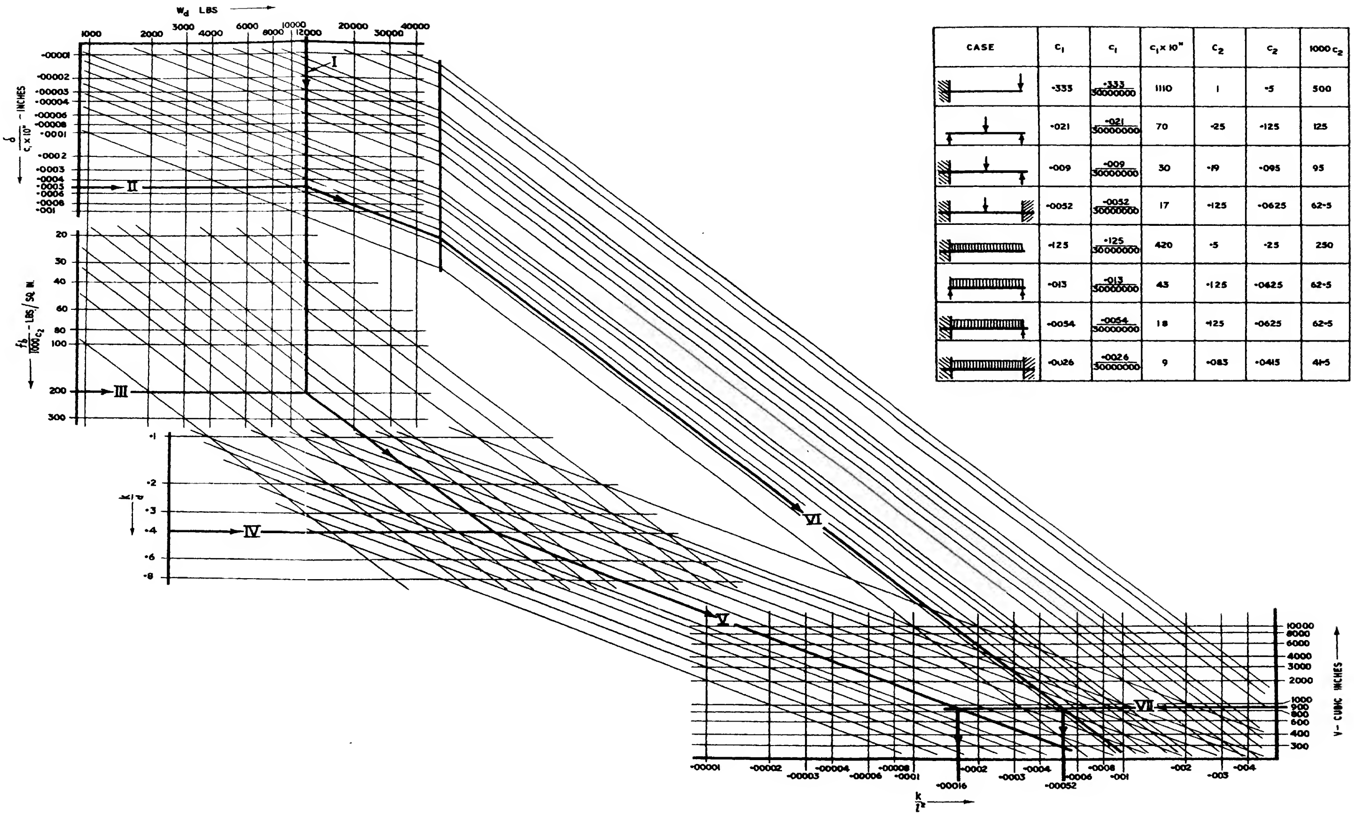
$$Z = \frac{I}{\frac{d}{2}} = \frac{2I}{d},$$

$$f_b = \frac{C_2}{2} \frac{W_d l d}{I}.$$

As the area of the cross-section of the beam is given by

$$A = \frac{V}{l},$$

$$I = k^2 A = k^2 \times \frac{V}{l}.$$



CASE	c_1	c_1	$c_1 \times 10^{11}$	c_2	c_2	$1000 c_2$
	·333	$\frac{·333}{30000000}$	1110	1	·5	500
	·021	$\frac{·021}{30000000}$	70	·25	·125	125
	·009	$\frac{·009}{30000000}$	30	·19	·095	95
	·0052	$\frac{·0052}{30000000}$	17	·125	·0625	62·5
	·125	$\frac{·125}{30000000}$	420	·5	·25	250
	·013	$\frac{·013}{30000000}$	43	·125	·0625	62·5
	·0054	$\frac{·0054}{30000000}$	18	·125	·0625	62·5
	·0026	$\frac{·0026}{30000000}$	9	·083	·0415	41·5

Fig. 116

Hence
$$\delta = C_1 \frac{W_d l^4}{EVk^2}$$
$$= \frac{C_1}{E} \times \frac{W_d}{V} \times \left(\frac{l^2}{k}\right)^2$$

and
$$f_b = \frac{C_2}{2} \times \frac{W_d l^2 d}{k^2 V}$$
$$= \frac{C_2}{2} \times \frac{W_d}{V} \times \frac{l^2}{k} \times \frac{d}{k}.$$

If we consider only structures of steel with practically constant modulus of elasticity (E = 30000000 lb./sq. inch), and we can substitute

$$c_1 = \frac{C_1}{E} = \frac{C_1}{30000000},$$
$$c_2 = \frac{C_2}{2},$$

c_1 and c_2 being again constants which vary according to the conditions of support of the beam. Hence we obtain

$$\delta = c_1 \times \frac{W_d}{V} \times \frac{1}{\left(\frac{k}{l^2}\right)^2}, \qquad (65)$$

$$\frac{W_d}{\delta} = \frac{1}{c_1} \times V \times \left(\frac{k}{l^2}\right)^2, \qquad (66)$$

$$f_b = c_2 \times \frac{W_d}{V} \times \frac{1}{\frac{k}{l^2} \times \frac{k}{d}} \qquad (67)$$

$$W_d = \frac{1}{c_2} \times f_b \times V \times \frac{k}{l^2} \times \frac{k}{d}. \qquad (68)$$

The graph (Fig. 116) has been developed in the same way as graph Fig. 112 for values

$$c_1 = 10^{-11} \qquad c_2 = 10^{-3}.$$

In order to use the graph (Fig. 116) for various loading conditions, the scales of δ and f_b must be adjusted, by multiplying the δ– scale by $c_1 \times 10^{11}$ and the f_b– scale by $1000c_2$, the values of c_1 and c_2 being applied in accordance with the particular case under consideration. Values of c_1 and c_2 for various loading conditions are shown in the table (Fig. 116). It may be repeated that the graph (Fig. 116) applies to all kinds of cross-sections, full sections, box sections, channel sections, I sections, etc., as long as the section is constant over the length and symmetrical in relation to the (horizontal) neutral axis of the beam.

A short example may demonstrate the use of the graph (Fig. 116). A joist rigidly clamped (or for the sake of the example, welded) at one end, and freely supported at the other, is to carry an equally distributed load (Fig. 117) of $W_d = 12000$ lb. (line I, Fig. 116) over a length of 96 inches. The maximum permissible deflection is $\delta_{max} = \cdot 009''$, and the maximum permissible bending stress is $f_{b\ max} = 12{,}500$ lb./sq. inch.

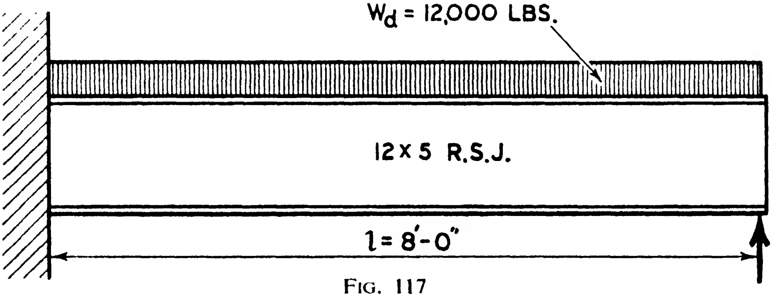


FIG. 117

With $c_1 \times 10^{11} = 18$ and $1000c_2 = 62{\cdot}5$, for this case (sec table Fig. 116) we have to plot the maximum deflection at $\frac{\cdot 009}{18} = \cdot 0005''$ (line II) and the maximum bending stress at $\frac{12500}{62{\cdot}5} = 200$ lb./sq. inch (line III). An average value for $\frac{k}{d}$ of standard joists is $\frac{k}{d} = \cdot 4$ (line IV).

The progress of the train of lines in the graph is shown by arrows, as in the previous cases, and we find that line V gives the relation $V\frac{k}{l^2}$ from the stress point of view and line VI gives the relation $V\left(\frac{k}{l^2}\right)^2$ from the deflection point of view.

A standard 12×5 joist has a radius of gyration of $k = 4{\cdot}84''$ and an area of $9{\cdot}45$ sq. inches. Hence $V = 9{\cdot}45 \times 96 \simeq 900$ cu. inches (line VII), which requires a minimum value of $\frac{k}{l^2}$ of $\cdot 00016$ from the stress point of view and of $\cdot 00052$ from the deflection point of view. For the 12×5 joist

$$\frac{k}{l^2} = \frac{4{\cdot}84}{9200} = \cdot 00053,$$

which is satisfactory.

Ribs

The design of ribs in a structure deserves special consideration.* Designers often overlook the fact that, if ribs are unsuitably proportioned, an increase in stiffness alone may give rise to an increase in

* See A. Thum and S. Berg, "Über die Festigkeit von Rippen bei ruhender und stossartiger Belastung" ("The Strength of Ribs under Static and Shock Loads"), *Z.V.D.I.*, 1933, page 281.

stress at the same time. Whilst stiffness grows with an increasing moment of inertia, the bending stress in the extreme fibre depends upon the section modulus, which, of course, is dependent on the distance of the extreme fibre from the neutral axis, and the moment of inertia.

If, therefore, in unfavourable circumstances, the moment of inertia does not increase in proportion to the distance of the extreme fibre from the neutral axis, the maximum stress may become higher than is permissible in the structure, although the stiffness may have been increased.

This state of affairs is particularly dangerous if it occurs in a structure subjected to alternating loads which may produce such cases of over-stressing. The " Bauschinger effect ", to be described below, may then cause failure of the structure.

By exceeding a certain limit (for instance, elastic limit under tension) in a material, the corresponding limit for the opposite loading condition (in this example, the elastic limit under compression) will be reduced in proportion to the former overstress.

If, in a structure subjected to an alternating load, one fibre has been over-stressed in one direction, it will be the first to be over-stressed when load is applied in the opposite direction, and vice versa ; and this will go on until, eventually, failure due to fatigue will occur in this fibre.

The conditions of stiffness and strength, for various designs of stiffening ribs on beams which are simply supported at the ends and loaded at the centre, have been studied by Thum and Berg, and the following calculations, as well as the graphs given in Figs. 118-22, have been deduced.*

In the formulae given below the suffix 1 always refers to the main plate, the suffix 2 to the rib. Symbols without suffix refer to the plate without rib, and those with suffix R to the structure with rib.

The ratio of the thickness of the rib to the width of the main plate is denoted by m, $\left(m = \frac{b_2}{b_1}\right)$. The ratio of the thickness of the plate to the height of the plate plus rib is denoted by n, $\left(n = \frac{d_1}{d_2}\right)$. The graphs show the influence of the above factors upon—

(1) The ratio p_Z of the strengths of the plate, with and without the rib, or $p_Z = \frac{Z_R}{Z}$ (Z being the section modulus).

(2) The ratio p_I of stiffnesses of the plate, with and without the rib, or $p_I = \frac{I_R}{I}$ (I being the moment of inertia).

* A. Thum and S. Berg, " Über die Festigkeit von Rippen bei ruhender und stossartiger Belastung " (" The Strength of Ribs under Static and Shock Loads "), *Z.V.D.I.*, 1933, p. 281.

For a plate with two ribs symmetrically arranged as shown in Fig. 118,

$$\mathrm{I}=\frac{b_1d_1^3}{12};\quad \text{and}\quad \mathrm{I_R}=\frac{(b_1-b_2)d_1^3+b_2d_2^3}{12}$$

or

$$p_\mathrm{I}=\frac{\mathrm{I_R}}{\mathrm{I}}=\frac{(b_1-b_2)d_1^3+b_2d_2^3}{b_1d_1^3}$$

$$=\frac{b_1d_1^3-b_2d_1^3+b_2d_2^3}{b_1d_1^3}$$

$$=1-m+\frac{m}{n^3}.$$

Also

$$\mathrm{Z}=\frac{b_1d_1^2}{6};\quad \text{and}\quad \mathrm{Z_R}=\frac{(b_1-b_2)d_1^3+b_2d_2^3}{6d_2}$$

or

$$p_\mathrm{Z}=\frac{\mathrm{Z_R}}{\mathrm{Z}}=\frac{(b_1-b_2)d_1^3+b_2d_2^3}{b_1d_1^2d_2}$$

$$=\frac{b_1d_1^3-b_2d_1^3+b_2d_2^3}{b_1d_1^2d_2}.$$

Hence

$$p_\mathrm{Z}=n-mn+\frac{m}{n^2}$$

$$=n(1-m)+\frac{m}{n^2}.$$

If p_Z is less than unity, the increase in stiffness due to the addition of the rib means a corresponding loss in strength. It will be seen from the graph (Fig. 118) that, with the exception of the cases for $m=1{\cdot}0$ (no rib) and $m={\cdot}5$ and $m={\cdot}33$ (ribs of a thickness of more than one-third of the total plate width), the strength is decreased when ribs of moderate height are used, and only with increasing height of the ribs, *i.e.* with decreasing n, can the strength of the main plate be regained and then increased.

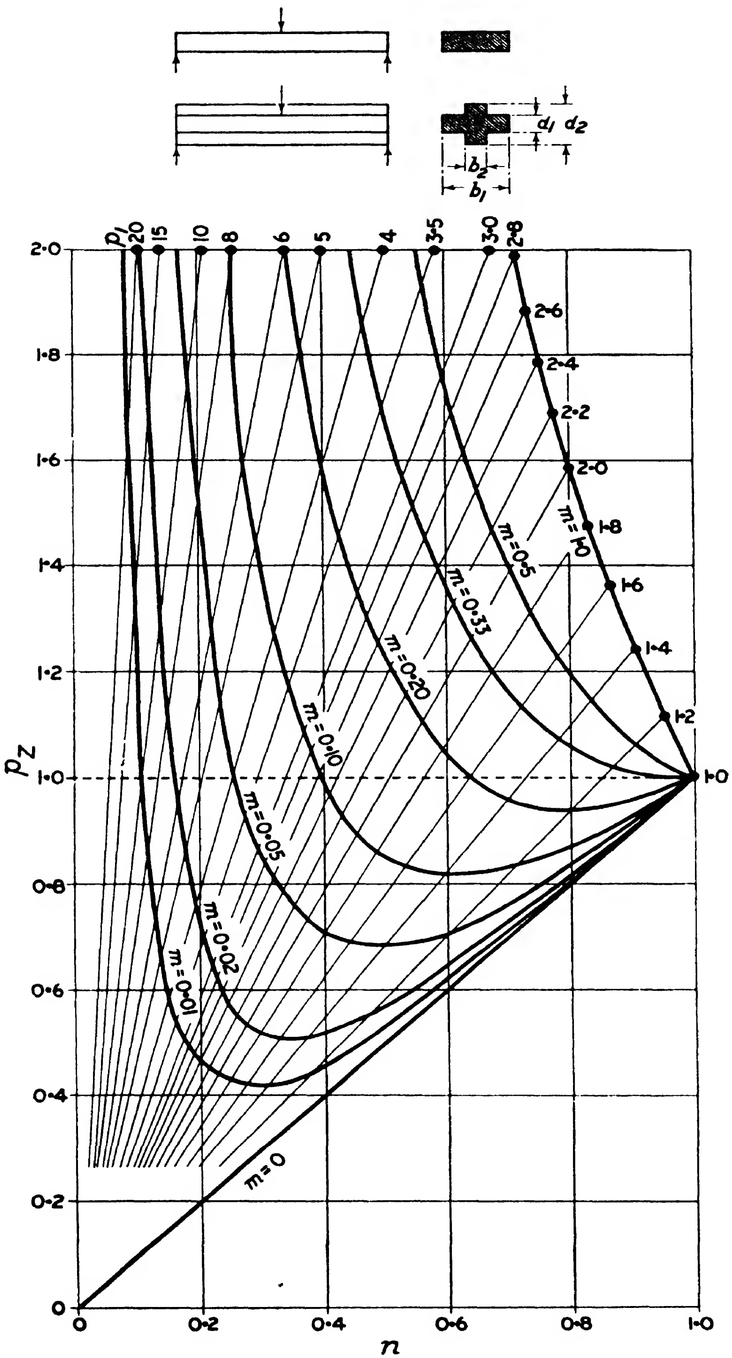
In the case of a simple unsymmetrical rib (Fig. 119), only ribs of a thickness of more than half the plate width ($m>{\cdot}5$) show no initial decrease in strength with increasing height of the rib.

The danger of faulty rib design is particularly great when dealing with brackets stiffened by a rib (Fig. 120), because at constant thickness of the rib ($m=$const.) the height of the rib grows (*i.e.* n decreases) from the point of application of the load towards the fixed end of the bracket. With decreasing n, p_Z may decrease, whilst at the same time the bending moment grows. Graphs, Fig. 121, show the bending moment M_b the section moduli Z and $\mathrm{Z_R}$, and the bending stresses f_b and $f_{b\mathrm{R}}$ for the various points of the bracket without or with rib. The values of Z and $\mathrm{Z_R}$ have been obtained from the equation

$$\mathrm{Z}=\frac{bd^2}{6},$$

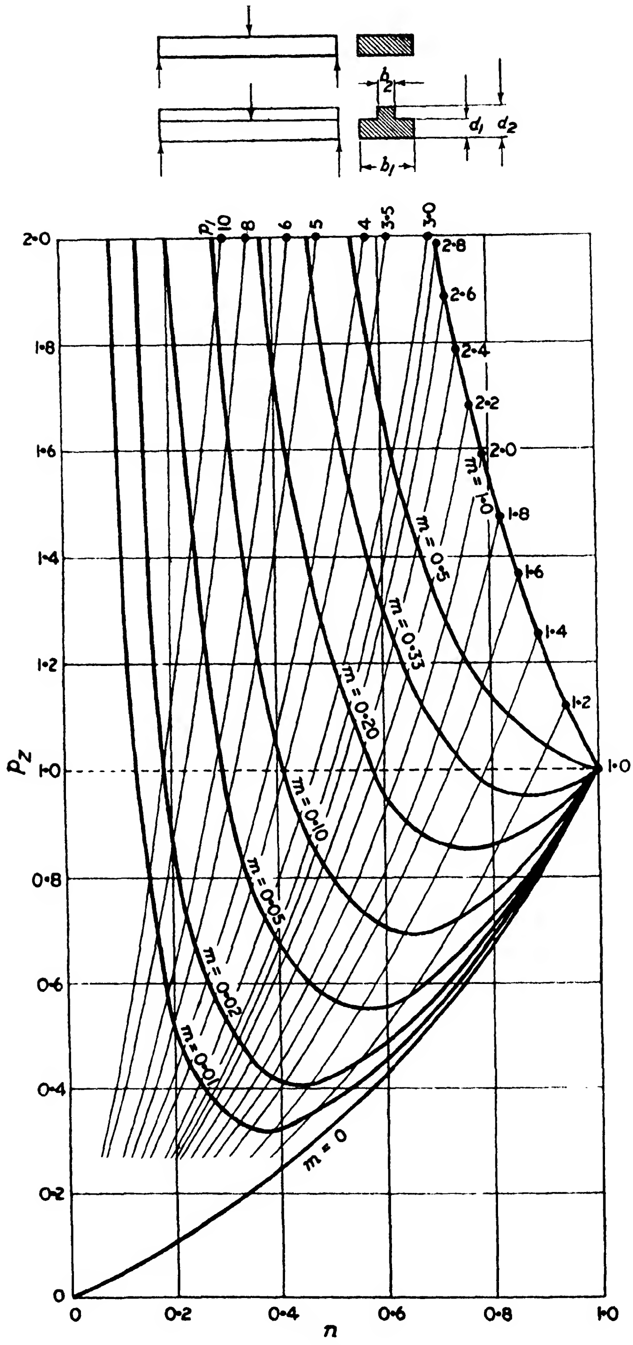
$$\mathrm{Z_R}=p_\mathrm{Z}\mathrm{Z},$$

p_Z for the various points being taken from graph (Fig. 119).



d_1
d_2
b_2
b_1
p_1
20
15
10
8
6
5
4
3·5
3·0
2·8
2·6
2·4
2·2
2·0
1·8
1·6
1·4
1·2
1·0
$m = 1{\cdot}0$
$m = 0{\cdot}5$
$m = 0{\cdot}33$
$m = 0{\cdot}20$
$m = 0{\cdot}10$
$m = 0{\cdot}05$
$m = 0{\cdot}02$
$m = 0{\cdot}01$
$m = 0$
p_Z
2·0
1·8
1·6
1·4
1·2
1·0
0·8
0·6
0·4
0·2
0
0
0·2
0·4
0·6
0·8
1·0
n

FIG. 118



$\frac{b_2}{2}$
b_1
d_1
d_2
p_1
10
8
6
5
4
3·5
3·0
2·8
2·6
2·4
2·2
2·0
1·8
1·6
1·4
1·2
1·0
$m = 1·0$
$m = 0·5$
$m = 0·33$
$m = 0·20$
$m = 0·10$
$m = 0·05$
$m = 0·02$
$m = 0·01$
$m = 0$
p_z
n
0
0·2
0·4
0·6
0·8
1·0
1·2
1·4
1·6
1·8
2·0

FIG. 119

It will be seen that, apart from increasing the stiffness of the bracket, the rib reduces the bending stress at the fixed end, but that, between the fixed end and point of load application, a zone exists where the stress is higher than it would have been if there had been no rib at all. When designing such brackets which are highly stressed, therefore, the designer has to proportion the bracket accordingly.

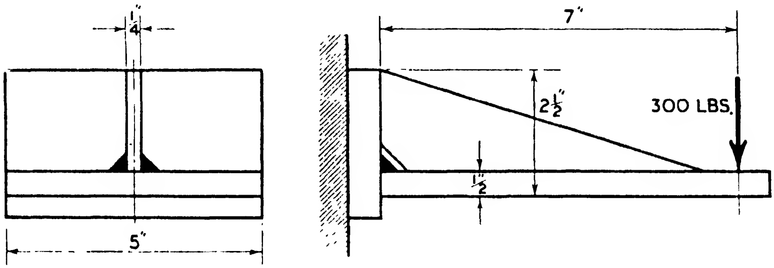


FIG. 120

If shock loads occur on a beam simply supported at the ends and loaded in the centre, the maximum stress can be obtained by the equation

$$f_b = \frac{W_d l}{4Z},$$

W_d being the equivalent static load which, under a shock energy U and at a deflection δ, can again be found from the equation

$$U = \frac{W_d \delta}{2}.$$

For a beam without a rib the following equation is obtained :

$$\delta = \frac{W_d l^3}{48EI}.$$

Hence $$U = \frac{W_d^2 l^3}{96EI}.$$

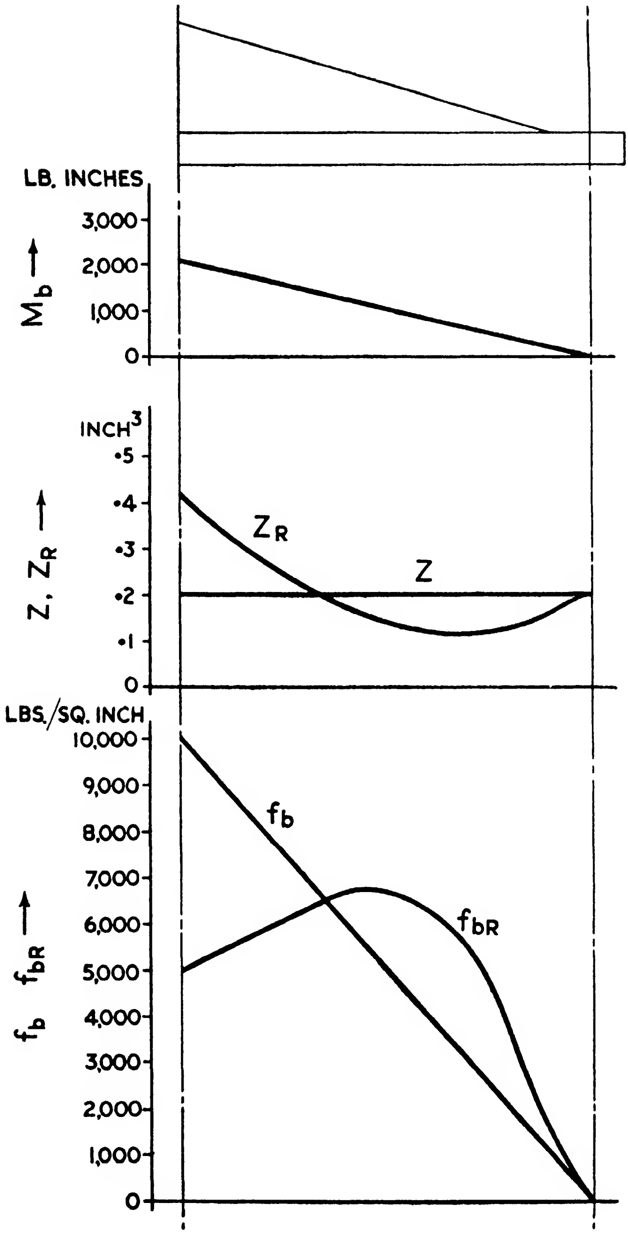
And as $$W_d = \sqrt{\frac{96UEI}{l^3}},$$

$$f_b = \frac{l}{4Z} \times \sqrt{\frac{96UEI}{l^3}}$$

$$= \frac{1}{Z} \times \sqrt{\frac{6UEI}{l}}.$$

For a ribbed beam the stress would therefore be

$$f_{bR} = \frac{1}{Z_R} \times \sqrt{\frac{6UEI_R}{l}}.$$



LB. INCHES
3,000
2,000
1,000
0
M_b
INCH^3
·5
·4
·3
·2
·1
0
Z, Z_R
Z_R
Z
LBS./SQ. INCH
10,000
9,000
8,000
7,000
6,000
5,000
4,000
3,000
2,000
1,000
0
$f_b \quad f_{bR}$
f_b
f_{bR}

FIG. 121

The ratio p_f between the two stresses is obtained as follows:

$$p_f = \frac{f_{bR}}{f_b} = \frac{Z}{Z_R}\sqrt{\frac{I_R}{I}}$$

$$= \frac{Z}{Z_R} \times \sqrt{p_I},$$

$$\frac{Z}{Z_R} = \frac{1}{p_Z}.$$

Hence the required ratio is

$$p_f = \frac{\sqrt{p_I}}{p_Z}.$$

In order to compare the strength under shock load in the same way as for static loads, Thum and Berg introduced the value $\frac{1}{p_f}$ and called it p_{Zs} (the ratio between the strengths as calculated against shock). This ratio can be shown in relation to the values of m and n.

Hence
$$p_{Zs} = \frac{p_Z}{\sqrt{p_I}}$$

$$= \sqrt{m\left(\frac{1}{n} - n^2\right) + n^2}.$$

The graph (Fig. 122) resembles that in Fig. 118 but the p_{Zs} values are still smaller than the p_Z values in the case of static loads. This proves that in the case of shock loads special care is necessary in the arrangement of ribs.

The cases for which p_{Zs} is less than unity are the most frequent, as they cover most of the ribs of medium height.

A shock resistance which is higher than that obtained with medium-sized ribs can be obtained by omitting ribs altogether, or, similar to the cases of static loads yet more pronounced, highest resistance will be obtained when very high ribs are used. It is usually difficult to produce ribs of great height by the casting process, but there should be no difficulty in producing them by welding. This may be considered another reason in favour of fabricated construction where shock loads are likely to occur.

Welds

The foregoing observations referred to conditions in the main component parts (plates) of a welded fabrication. The designer must, however, always bear in mind that these component parts have to be joined together, and that the welding operation imposes certain requirements and limitations upon the design.

The arrangement and lay-out of the welded joints in the structure

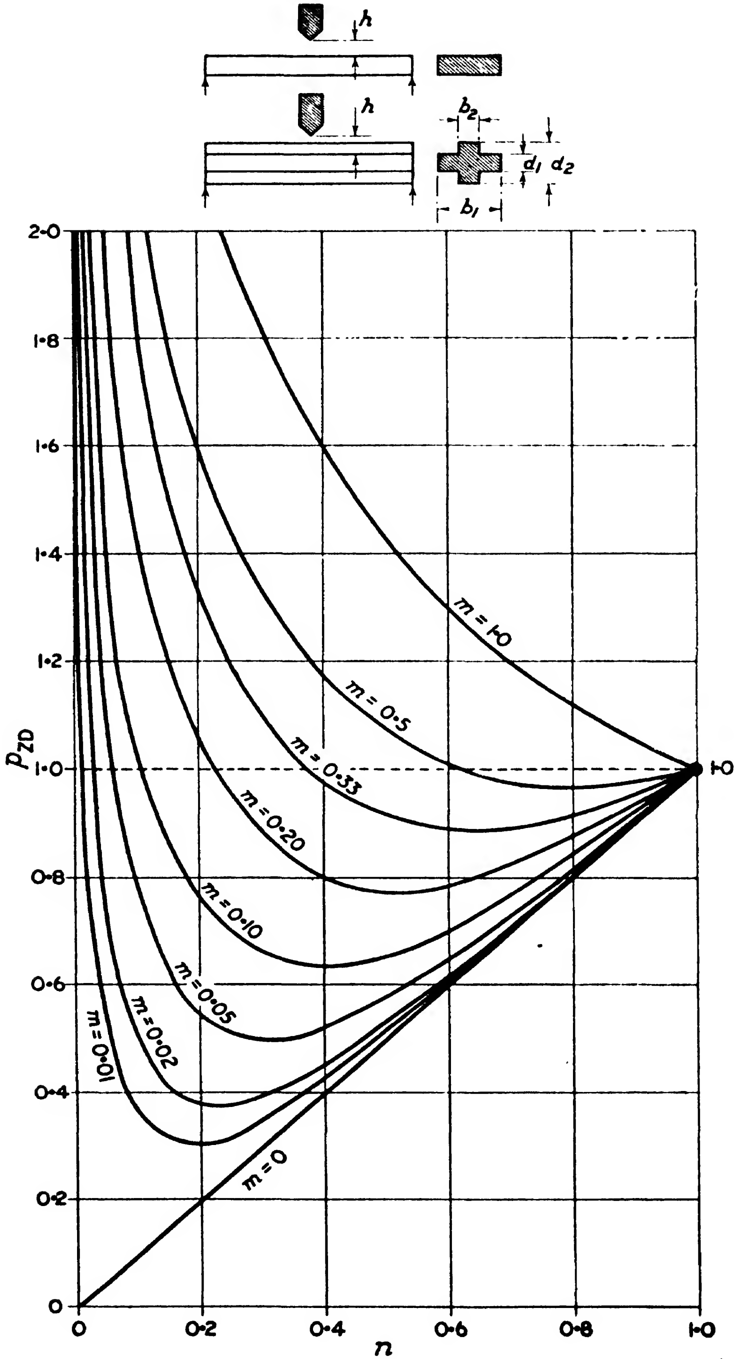

h
h
b_2
d_1 d_2
b_1
2·0
1·8
1·6
1·4
1·2
1·0
0·8
0·6
0·4
0·2
0
p_{ZD}
m = 1·0
m = 0·5
m = 0·33
m = 0·20
m = 0·10
m = 0·05
m = 0·02
m = 0·01
m = 0
1·0
0
0·2
0·4
0·6
0·8
1·0
n

FIG. 122

must be carried out with due consideration for the welding process and the subsequent manufacturing processes, like machining and fitting operations.

Some of the points to be observed, the application of which will be shown later in the various examples, are given below.

The juxtaposition of several heavy welds will cause internal stresses, and should be reduced to a minimum. Symmetrical arrangement of

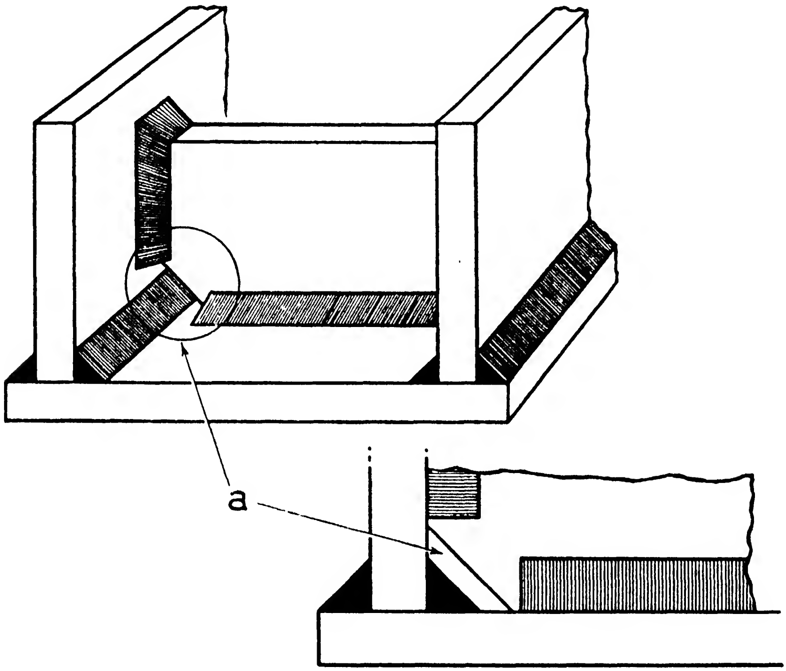


FIG. 123

welds, like double V- or U-butts instead of single ones, will avoid unilateral heating and the tendency of plates to lift on either side of the weld.

Mechanical clearances between welds and adjacent components which have to be added to the structure later, should be generous in order not to tighten down the necessary tolerances on weld sizes beyond economical limits. This point is particularly important when designing stiffener webs. In such cases it is also important to ensure that interference of various welded seams does not occur. A corner (*a*) cut off from a web will clear a weld which is crossing the joints connecting the web to the main plates (Fig. 123).

Where machining operations have to cover weld metal, it should be remembered that differences may exist between the hardness of the weld metal, the parent metal adjacent to the welds, and the parent metal which has not been affected by the welding process. This applies to arc-welded as well as to spot- and projection-welded joints, especially if no heat treatment has been carried out after welding. Differences in the machined surfaces of these three portions may, therefore, be encountered. If such differences are not permissible, an alternative arrangement of the welded joint must be found (see Fig. 154).

It has been mentioned before (see page 5) that the labour cost in laying down a welded seam should be reduced to a minimum. It is

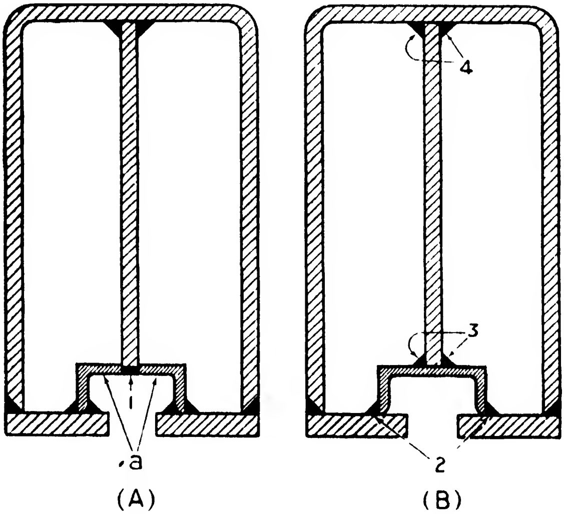


FIG. 124

probably cheaper to bend one plate rather than weld two plates at right angles. When designing bent plates and determining bending radii, care must be taken, however, that overstressing of the plate material due to the bending operation does not reduce the strength of the component. If at all possible, bending radii should not be less than 1·25 times the thickness of the plate. Under no circumstances should a bending radius be less than the thickness of the plate to be bent.

One of the most important points to be watched, and one which is easily overlooked in the design of welded structures, is the provision of adequate accessibility for welding. Proper accessibility enables the welder to produce sound welds which fulfil the requirements of the designer's specifications. Inadequate provision may lead to unsound welds or to improvisations in the welding shop which are undesirable from a design and manufacturing point of view. As a very simple example, Fig. 124A shows a section through the outer stay of a production milling machine. By splitting the T-slot base (*a*) into two angles weld 1 can be laid down in a final welding operation. If the T-slot base had been made from a channel (Fig. 124B) one of the weld groups 2, 3, or 4 could not have been made.

The above is a case of the total inaccessibility of a weld position. Poor accessibility, however, may be very dangerous, as it may not be discovered officially until, due to a weld failure, it is brought forward as an excuse for bad workmanship on the welder's part. The most careful stress calculation of welds will be worthless if the welds are not of the quality for which the permissible working stresses have been established.

The designer of a welded fabrication, while considering the shape of plates and other component parts from the point of view of their required mechanical properties, should, therefore, continuously keep before his eyes the arrangement of his design as a whole, and particularly the lay-out of the welds which form the vital joints of his structure.

Chapter 4

DETAIL DESIGN OF TYPICAL PART-ASSEMBLIES USED IN FABRICATED WELDED CONSTRUCTION

THE various parts, plates, rolled sections, or castings, when connected by welded joints, form the final assemblies, the sub-assemblies, or the part-assemblies of the welded structure. The terms may be defined as follows :

(1) *The final assembly* is the structure completed so far as the welding shop is concerned, and ready, for instance, for delivery to the machine shop for final machining operations. It is a complete unit which, after further operations in the fitting and assembly shop, may become, or may be incorporated in, a complete machine or other engineering structure.

(2) *The sub-assembly* is a part of the complete welded structure which, for reasons of manufacturing convenience, is assembled separately before being incorporated in the final product. The division of a welded structure into sub-assemblies is, therefore, dictated by considerations of fabricating and welding procedure and has to be determined by the designer with these considerations in view.

(3) *The part-assembly* is a portion of the structure which may form the subject of a particular stress or stiffness calculation. It is, therefore, the result of a purely imaginary subdivision of the structure, and a part-assembly as such may never appear as an independent structure during the actual assembly procedure. It is a portion of the structure which may be considered separately, in order to determine certain dimensions, stresses, or distortions.

A simple example of these definitions is shown in the case of a worm reduction gear-box with motor bracket (Fig. 125). Sub-assemblies which should be completed before final assembly of the gear-box are shown in Fig. 126. A part-assembly, for which a stress calculation might have to be carried out, is the motor bracket (Fig. 127), which has to be strong enough to carry the weight of the driving motor. The drawings (Figs. 125-7) should be self-explanatory. It will be appreciated that a number of typical part-assemblies will recur frequently in fabricated constructions, whilst sub-assemblies will, of course, be different for each particular design.

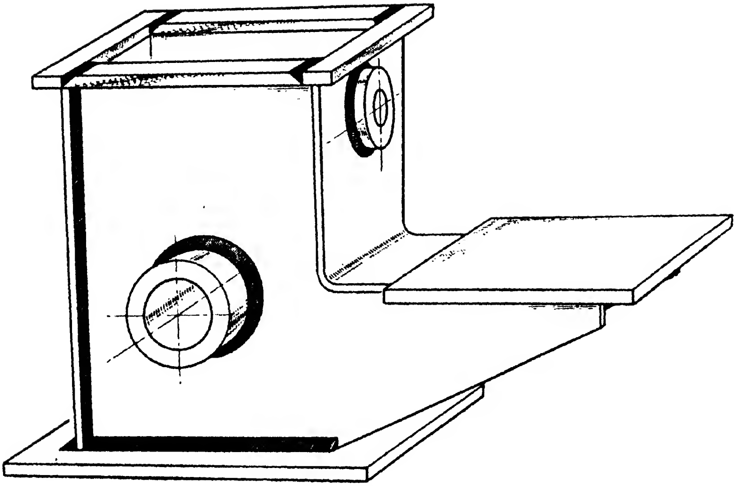


FIG. 125

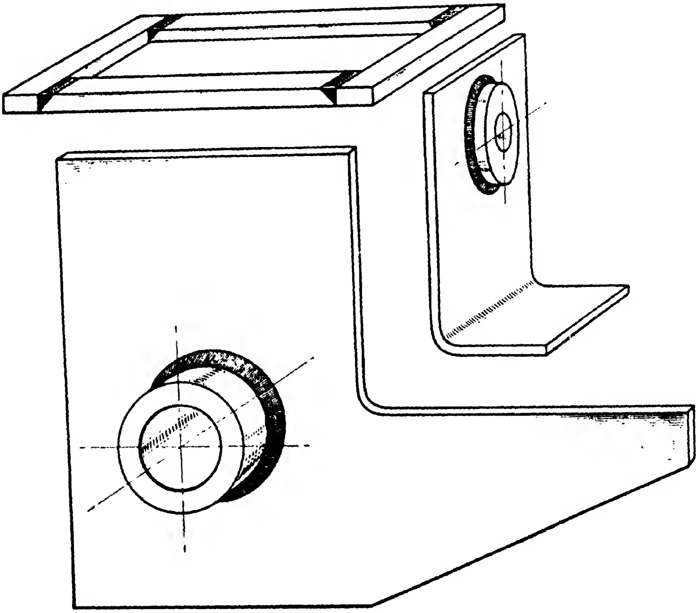


FIG. 126

Some typical part-assemblies, their stress calculations, and design principles shall now be discussed.

In the previous chapter the strength and stiffness of ribs and ribbed sections have been discussed in principle. Various additional points have to be considered, however, when laying out ribbed constructions. Ribs are usually provided when the stiffness of the bare main structure would not be adequate for the duty required, but the designer has to consider not only the shape and arrangement of the ribbed structure as a whole, but also the strength of the welded joints which connect the ribs to the main plates. Furthermore, the arrangement of ribs often creates abrupt changes of sections and it may be necessary to allow for possible stress concentrations if alternating loading conditions exist

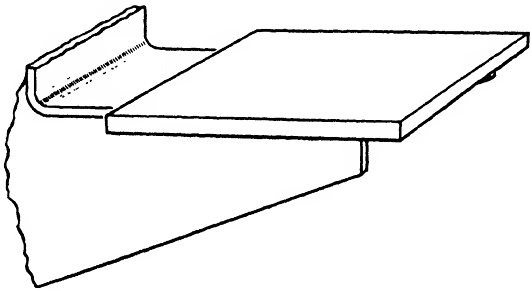


FIG. 127

(see page 32). Considering the example of a fabricated T-section subjected to a load which creates bending stresses (Fig. 128(a)), it is necessary—

(*a*) To determine the maximum stress in the complete section.
(*b*) To find the shear stresses in the welds connecting the rib to the top plate.

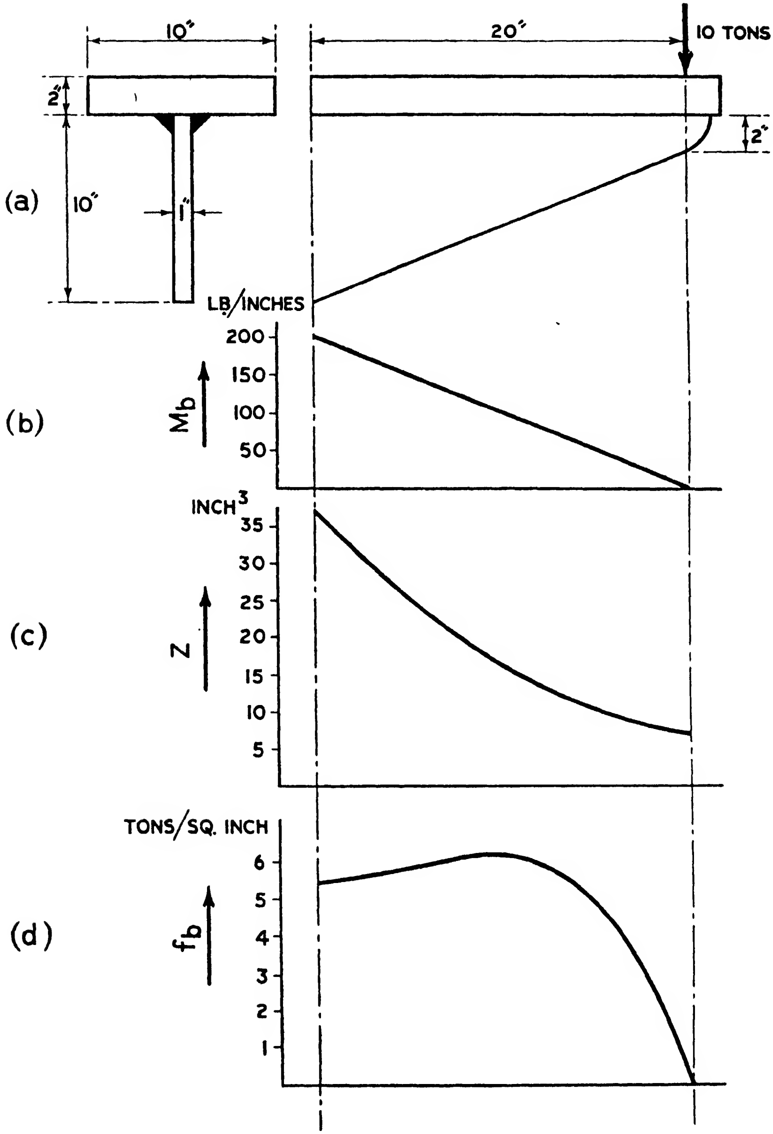
Fig. 128(b) shows the values of the bending moment, and Fig. 128(c) the section moduli, for the various sections along the length of the bracket.

The stress for any section is given by $f_b = \frac{M_b}{Z}$ (see also page 110), and is shown in Fig. 128(d). The maximum bending stress in the parent metal is 6·25 tons/sq. inch.

The shear stresses in the welds are determined in the following way :

The horizontal shear force P_{s1} per inch length of weld is

$$P_{s1} = \frac{P_v G}{I},$$



10"
20"
10 TONS
2"
2"
(a)
10"
1"
LB/INCHES
200
150
100
50
(b)
M_b
INCH³
35
30
25
20
15
10
5
(c)
Z
TONS/SQ. INCH
6
5
4
3
2
1
(d)
f_b

FIG. 128

where P_v is the vertical shear force for the section in question.
G is the first moment about the neutral axis of that part of the section in question which lies beyond the weld.
I is the moment of inertia of the section.

The shear stress in each of the two fillet welds is then

$$f_{sw}=\frac{1}{2}\frac{P_{s1}l}{\cdot 707hl}=\frac{\cdot 707P_{s1}}{h},$$

where h is the leg length of the welds, and l is the over-all length of the welds.

$$f_{sw}=\frac{\cdot 707P_vG}{Ih}.$$

In our case $P_v=\text{const.}=10$ tons, and $h=\text{const.}=\cdot 375$ inch. G and I vary over the length of the bracket according to the changes of section and the varying position of the neutral axis, which is shown in Fig. 129(a). The values of G for the various sections are shown in Fig. 129(b), those of I in Fig. 129(c), and Fig. 129(d) shows f_{sw} for the various portions of the fillet welds over the length of the bracket. The maximum shear stress in the weld is found to be 4·4 tons/sq. inch.

The calculations for a beam having a constant section over the whole of its length is less complicated, as I, Z, and G do not vary. For the case of a fabricated I-section (Fig. 130) we find the maximum bending stress in the extreme fibre of the section, and having a value given by

$$f_b=\frac{M_bd}{2I},$$

where $$I=\frac{bd^3}{12}-\frac{(b-t_2)(d-2t_1)^3}{12}.$$

Furthermore $$P_{s1}=\frac{P_vG}{I}$$

and $$G=bt_1\left(\frac{d-t_1}{2}\right).$$

The shear stress in each fillet weld is

$$f_{sw}=\frac{\cdot 707P_{s1}l}{hl}=\frac{\cdot 707P_{s1}}{h},$$

$$f_{sw}=\frac{\cdot 707P_vG}{Ih}.$$

When designing built-up sections similar to those shown in Figs. 128-30 it will often occur that the weld size (h) required for strength reasons will be below the minimum value recommended in connection with the plate thicknesses used (see Fig. 5). In such cases intermittent welds may be used (Fig. 131).

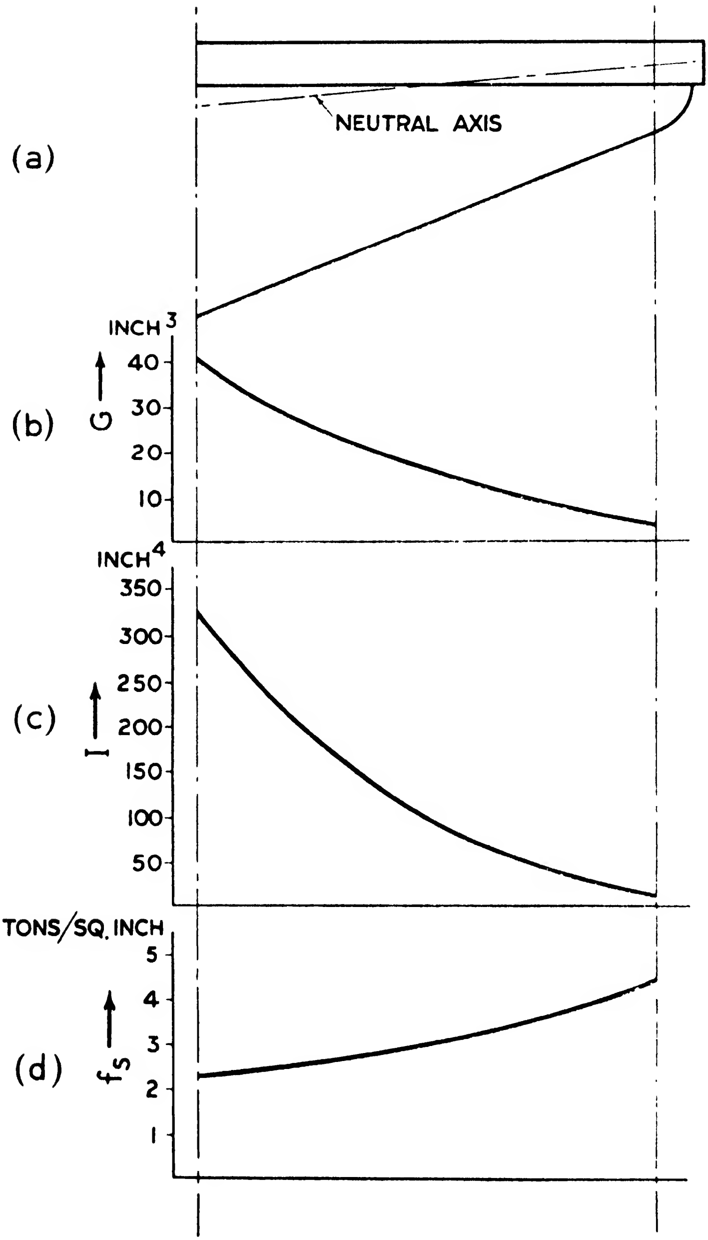

NEUTRAL AXIS
(a)
INCH 3
G
40
30
20
10
(b)
INCH4
350
300
250
200
150
100
50
(c)
I
TONS/SQ. INCH
5
4
3
2
1
(d)
f_s

FIG. 129

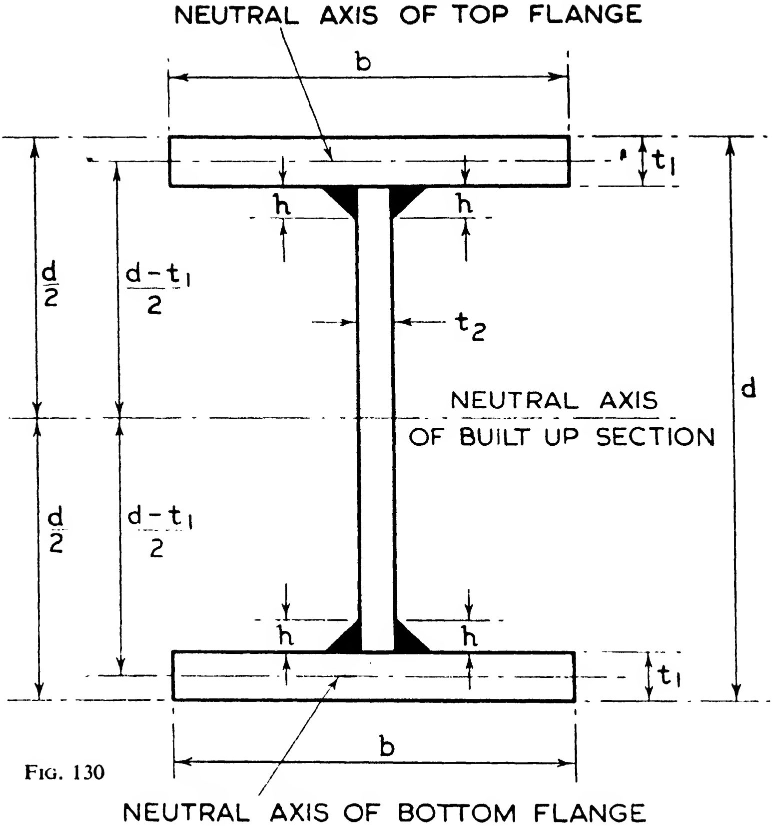


FIG. 130

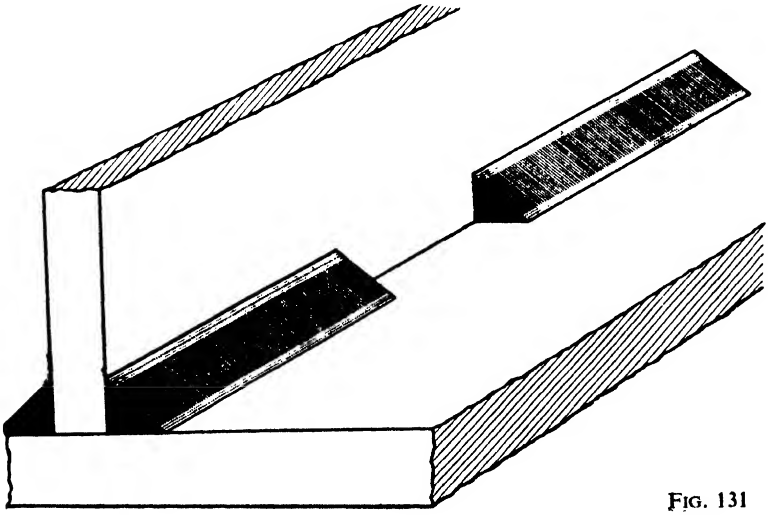


FIG. 131

The shear force per inch length of the built-up section is given by

$$P_{s1} = \frac{P_v G}{I}.$$

If the permissible load per inch length of the minimum weld size recommended for the particular plate thicknesses is P_{w1}, the length of the built-up section is l, and the total effective weld length (total of the effective lengths of the part welds) is l_w, then

$$P_{w1} l_w = P_{s1} l.$$

The percentage of effective weld length to full length of the built-up section is then

$$100\frac{l_w}{l} = 100\frac{P_{s1}}{P_{w1}},$$

or, considering the required length of weld per foot length of built-up section,

$$l_w \text{ per foot length of built-up section} = \frac{12P_{s1}}{P_{w1}}$$

$$= \frac{12P_v G}{IP_{w1}}. \qquad (69)$$

Intermittent welds on both sides of a stiffener or rib can be laid out as chain welds (Fig. 132), or as staggered welds (Fig. 133). Staggered

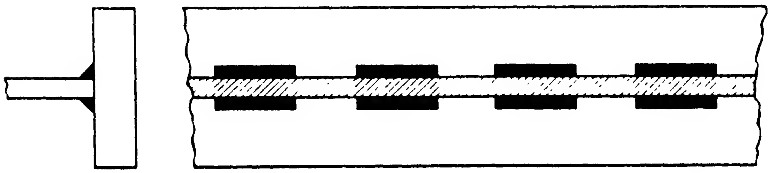


FIG. 132

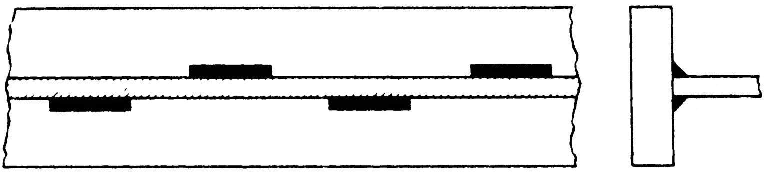


FIG. 133

welds have the advantage of saving weld metal while giving approximately the same joint stiffness as chain welds. The effective length of each part weld is, of course, determined as in the case of continuous welds (see page 43). It is advisable * to limit the maximum space between welds on the compression side to 12 times the thickness of

* *Welding Memorandum*, No. 11, issued by the Advisory Service on Welding, Ministry of Supply, London.

the thinner member of the built-up section, and to 16 times the thickness of the thinner member on the tension side.

When fixing beam type structures to a fabricated construction, the designer has to consider that the welds create rigid joints and should be calculated as such. They have to carry, therefore, the forces and bending moments which are to be transmitted to a rigidly clamped beam. If, for example, a horizontal joist section (*a*) is loaded vertically and connects two vertical plates (*b*) in a structure (Fig. 134), the maximum bending moment $M_{b\ max}$ which occurs at the joints has to be taken

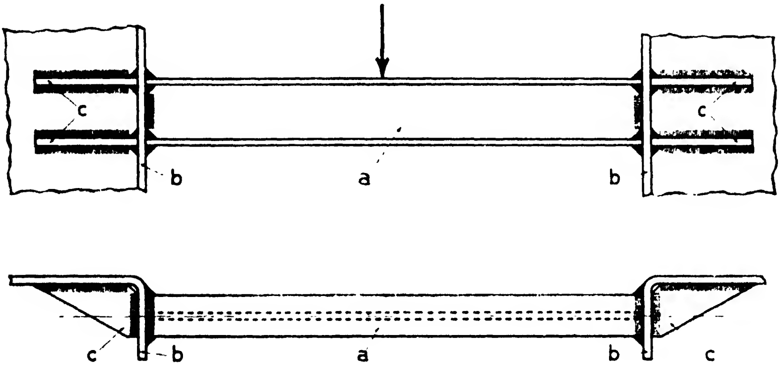


FIG. 134

by the welds as well as by the joist itself. The section modulus of an $8'' \times 5''$ standard joist (Fig. 135) is given by

$$Z_{Joist} = 22{\cdot}42 \text{ inches}^3.$$

With the permissible bending stress

$$f_{b\ Joist}\ 8 \text{ tons/sq. inch}$$

the joist can transmit a maximum bending moment given by

$$\begin{aligned} M_{b\ max\ (Joist)} &= f_{b\ Joist} \times Z_{Joist} \\ &= 8 \times 22{\cdot}42 \\ &= 179{\cdot}4 \text{ ton inches.} \end{aligned}$$

The connecting welds must be capable of transmitting the same bending moment in addition to the vertical supporting force which will create a shear stress (see page 63).

With $\frac{3}{4}''$ fillet welds round the flanges and $\frac{3}{8}''$ fillet welds along the web, we obtain for the welds (see Fig. 135):

$$I_{Welds\ (Web)} = 2 \times \frac{{\cdot}265 \times 6{\cdot}85^3}{12} = 14 \text{ inch}^4,$$

$$I_{Welds\ (Flanges)} \approx 4 \times {\cdot}53 \times 5 \times 3{\cdot}71^2 = 146 \text{ inch}^4,$$

$$I_{Welds\ (total)} = 160 \text{ inch}^4,$$

$$Z_{Welds} = \frac{160}{4.53} = 35{\cdot}5 \text{ inch}^3.$$

If the permissible bending stress in the weld (see page 39) is given by

$$f_{b\ (\mathrm{Weld})} = 5 \text{ tons/sq. inch},$$

then

$$\begin{aligned} M_{b\ \max\ (\mathrm{Weld})} &= f_{b\ (\mathrm{Weld})} \times Z_{\mathrm{Welds}} \\ &= 5 \times 35{\cdot}5 \\ &= 177{\cdot}5 \text{ ton inches,} \end{aligned}$$

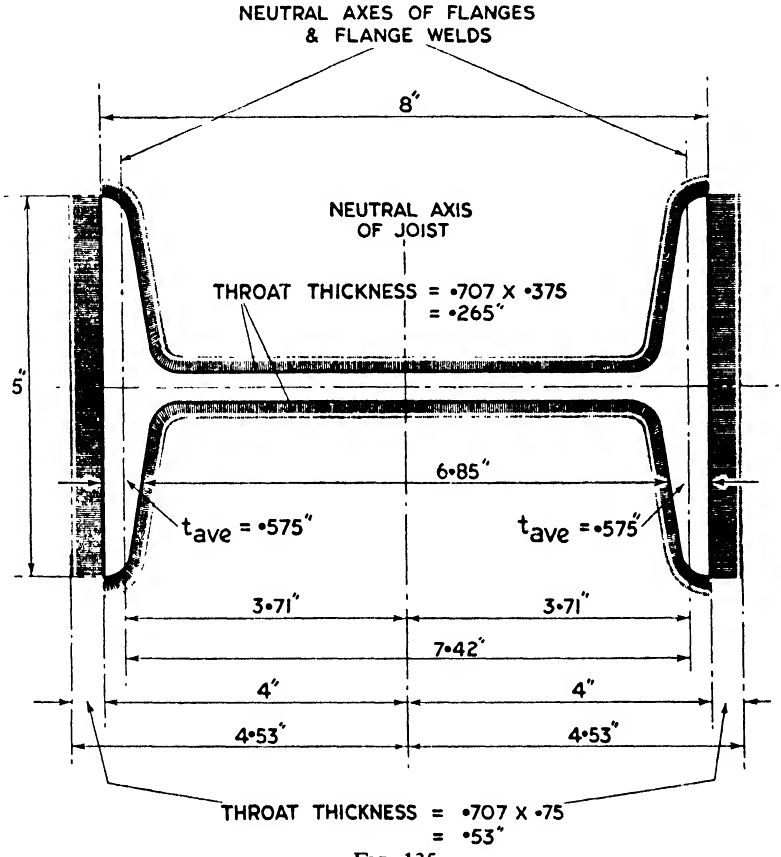


FIG. 135

which is sufficiently close to the maximum bending moment for the joist, which is given by

$$M_{b\ \max\ (\mathrm{Joist})} = 179{\cdot}4 \text{ ton inches.}$$

In order to transmit this bending moment without buckling in the vertical side walls, special stiffeners (c) (see Fig. 134) should be arranged, unless the remaining structure itself provides the necessary stiffness for these plates. Such stiffeners are best calculated to take the full horizontal force P_H which is exerted on the side walls through the bending moment (Fig. 136).

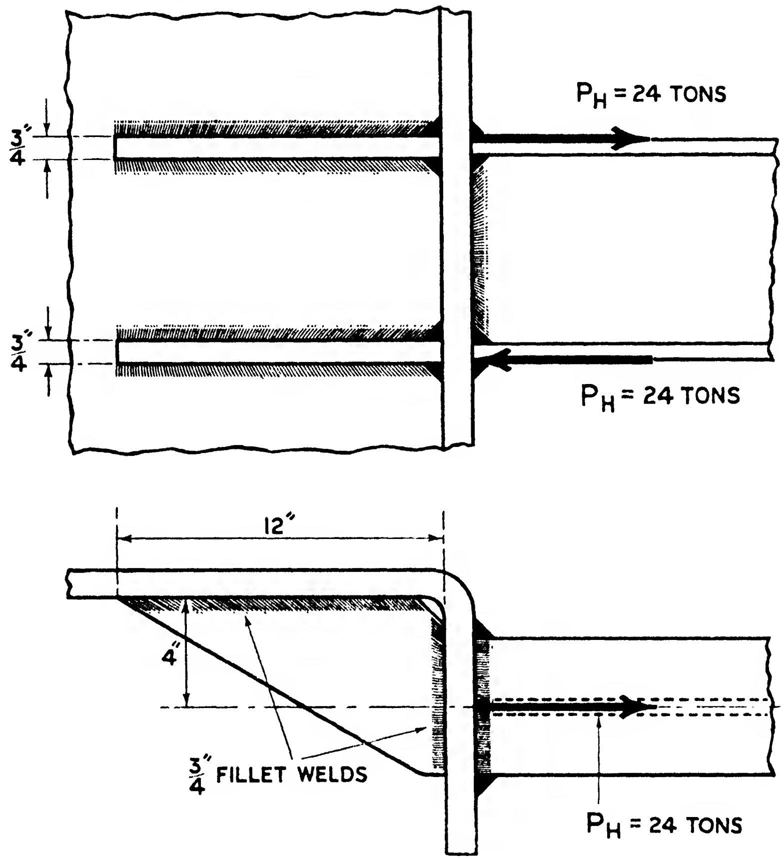


Fig. 136

Here
$$P_H = \frac{M_{b\ max}}{d - t_{ave}}$$
$$= \frac{179{\cdot}4}{7{\cdot}425}$$
$$= 24 \text{ tons.}$$

Each stiffener should, therefore, be calculated to transmit a force of 24 tons. The stiffener is stressed along the 12″ side by a shear stress given by

$$f_s = \frac{1{\cdot}5 \times 24}{12 \times {\cdot}75}$$
$$= 4 \text{ tons/sq. inch,}$$

and by a bending stress

$$f_b = \frac{24 \times 4 \times 6}{\cdot 75 \times 144}$$

$$= 5{\cdot}33 \text{ tons/sq. inch.}$$

The maximum stress

$$f_{\max} = \frac{f_b}{2} + \frac{1}{2}\sqrt{f_b^2 + 4f_s^2}$$

$$= 2{\cdot}67 + \frac{1}{2}\sqrt{28{\cdot}5 + 4 \times 16}$$

$$= 7{\cdot}5 \text{ tons/sq. inch.}$$

For the calculation of the $\frac{3}{4}''$ fillet welds connecting the stiffeners to the

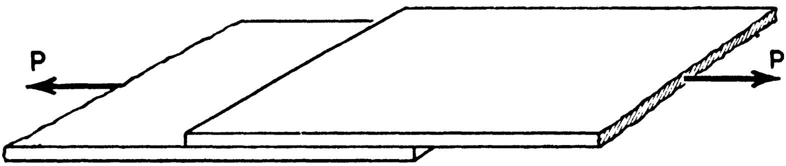


FIG. 137

wall along their 12″ edges (effective length of the welds being 12″ − 1·5″ = 10·5″), we use equation 32 (page 63):

$$f_{b\ \max} = \frac{4{\cdot}24 \times 24 \times 4}{10{\cdot}5^2 \times \cdot 75}$$

$$= 4{\cdot}95 \text{ tons/sq. inch.}$$

In part-assemblies which consist of relatively thin plates—usually $\frac{1}{4}''$ or below—it may be considered advantageous to use resistance-welded instead of arc-welded joints (thicker plates require more powerful resistance welding equipment than is in general use at present). As pointed out previously (see page 83), spot welds should not be highly stressed in tension, and spot welding should, therefore, be applied in such a way that the joints in question are stressed mainly in shear. The problem amounts, then, to the joining of two plates which are loaded as shown in Fig. 137. As an example, the spot-welded joint between two mild steel plates $\frac{1}{4}''$ thick and 5″ wide may have to transmit P = 4·5 tons.

At first the tensile stress in the plates themselves is checked. It is

$$f_{\text{Plate}} = \frac{4{\cdot}5}{5 \times \cdot 25} = 3{\cdot}6 \text{ tons/sq. inch.}$$

Material $\frac{1}{4}''$ thick requires a minimum spot weld diameter d of $\frac{1}{2}''$ (see Fig. 101).

With a permissible shear stress in spot welds given by $f_s = 5$

tons/sq. inch, we find the maximum load which can be transmitted by one weld, from equation (56),

or $$P_{\text{Weld}} = \frac{f_s d^2}{1{\cdot}28} = \frac{5 \times {\cdot}25}{1{\cdot}28}$$
$$= {\cdot}98 \text{ ton.}$$

The number of spot welds required to transmit 4·5 tons is

$$\frac{4{\cdot}5}{{\cdot}98} = 4{\cdot}6$$

and the number to be provided is 5.

The minimum spacing between ½″ diameter spot welds is 1·5″, and

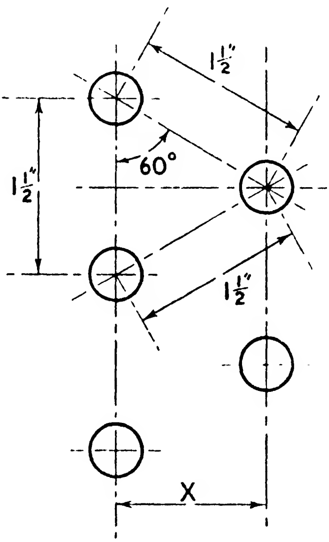


FIG. 138

the minimum edge distance is ·75″ (see Fig. 101). On a plate width of 5″, no more than three welds can be accommodated in a row, and we have therefore to provide for two rows. In order to save space, the welds may be laid out in a staggered arrangement. The overlap required is dependent on the spacing between the rows (Fig. 138). This is given by

$$X = 1{\cdot}5 \times \sin 60^\circ$$
$$= 1{\cdot}3''.$$

The edge distance on either end being again ·75, the overlap will be

$${\cdot}75 + 1{\cdot}3 + {\cdot}75 \quad \text{or} \quad 2{\cdot}8 \text{ inches.}$$

The welds will, therefore, be arranged as shown in Fig. 139.

In cases where torsional loading occurs, the closed box or tubular section will give the greatest strength and resistance to distortion. There are many cases, however, where the closed-box arrangement cannot be applied, due to other design considerations which require openings or completely open sections. The application of fully closed sections may also be prohibited by manufacturing difficulties. This difficulty is encountered in the design of lathe beds where torque is the dominant stressing condition, and the free fall of cuttings must not be interfered with by a closed-bed section. It was in the design of lathe beds, therefore, that the arrangement of " Zig-Zag Ribs " was first introduced, an arrangement of diagonal ribbing which gives much higher resistance to torque than parallel longitudinal or transverse

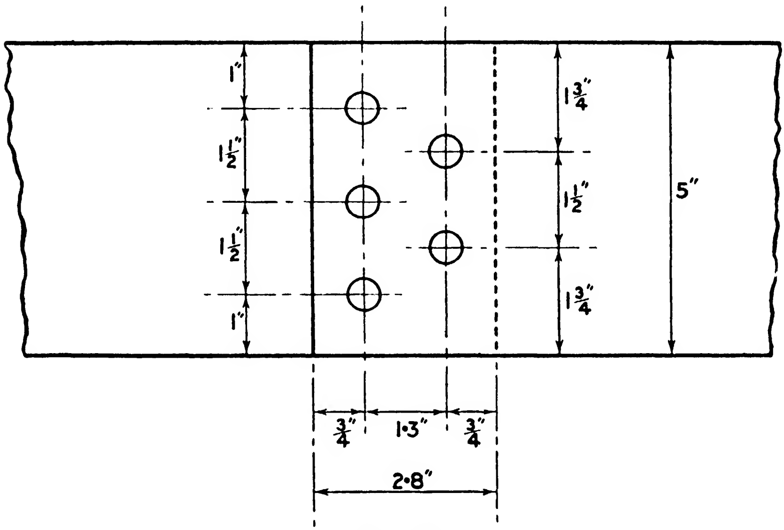


FIG. 139

ribbing. Whilst diagonal ribbing sometimes causes casting difficulties, especially in structures of large dimensions where different cooling contraction may produce cracking, its application is relatively simple in welded constructions.

The stressing conditions under torque of a channel section reinforced by diagonal ribs (Fig. 140) have been studied by A. Wolff.*

Wolff considers first the action of a torsional moment M_{t1} on the channel profile without ribs. Under the action of M_{t1} the cross sections of the channel do not remain plane and different elongations occur at its various points. The elongation at the top of the open part may be called δ_1, that at the bottom corner δ_2. If the small angle through which

* A. Wolff, " Beitrag zur Berechnung der Verdrehungssteifigkeit von Gusskörpern " (" A Contribution to the Calculation of Torsional Stiffness in Castings "), *Werkstattstechnik* (Julius Springer, Berlin), 1925, page 356.

the channel is twisted is $\Delta\phi$, the strain energy created by M_{t1} is given by

$$U_1 = \frac{M_{t1}\Delta\phi}{2},$$

and as

$$\Delta\phi = \frac{3M_{t1}l}{t^3(b+2d)C},\ *$$

where C is the modulus of shear of the material,

$$U_1 = \frac{3M_{t1}{}^2 l}{2t^3(b+2d)C}.$$

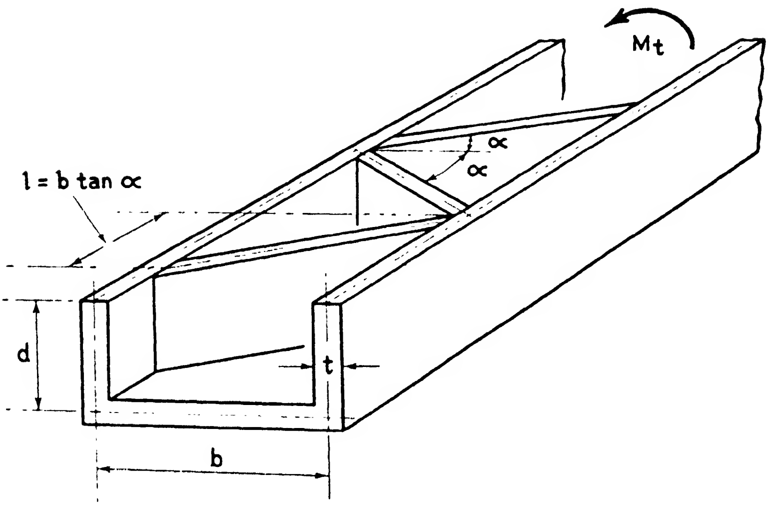


FIG. 140

If the same distortion is to be caused in the channel profile reinforced by diagonal ribs, an additional torsional moment M_{t2} must be added to M_{t1}, the total torque being

$$M_t = M_{t1} + M_{t2}.$$

The strain energy created by M_{t2} is

$$U_2 = \frac{M_{t2}\Delta\phi}{2}.$$

The increase in stiffness created by the ribs as compared with that of the structure without ribs is

$$\eta = \frac{M_{t1}+M_{t2}}{M_{t1}} = \frac{U_1+U_2}{U_1}$$

$$= 1 + \frac{U_2}{U_1}.$$

* See C. Weber, "Die Lehre der Drehungsfestigkeit" ("Theory of Torsional Strength"), *V.D.I.*, Berlin, 1921.

The different elongations at the top and bottom of the section cause the rib to be strained, as indicated in Fig. 141, in tension, torsion, and bending. Only the bending stress will be considered ; the tensile stress will be small because of the relatively great sectional area, and the torsional stress will be low because of the low torsional stiffness of the rib.

To calculate the bending strain energy, consider the bending strain in the rib which is caused by the different elongations δ_1 and δ_2 at the top and bottom corners of the profile (see Fig. 141). Considering that α varies very little during the distortion, the relative elongation between

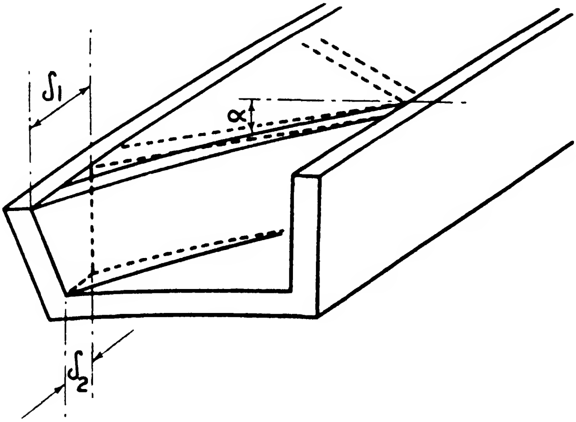


FIG. 141

the top and bottom fibres of the rib may be determined as follows :

$$\delta = (\delta_1 - \delta_2) \sin \alpha \quad \text{(Fig. 142).}$$

With
$$\delta_1 - \delta_2 = f_t \times \frac{d}{C}*$$

and
$$f_t = \frac{9M_{t1}}{2t^2(2d+b)},\dagger \qquad (70)$$

we obtain
$$\delta = \frac{9M_{t1}}{2t^2(2d+b)} \times \frac{d}{C} \times \sin \alpha.$$

The bending moment to create this deformation is calculated from (1) :

$$\delta = \frac{f_b}{E} \times \frac{b}{\cos \alpha},$$

where $\frac{b}{\cos \alpha}$ is the length of the rib, and (2) the bending stress in the rib

$$f_b = \frac{M_b d}{2I_{Rib}}. \qquad (71)$$

* A. Föppl, *Festigkeitslehre (Strength of Materials)*, iii (Teubner), 1927.
† See A. Wolff, *Werkstattstechnik*, 1925, page 356.

Hence $$\delta = \frac{M_b db}{2EI_{Rib} \cos \alpha},$$

$$M_b = \frac{2EI_{Rib} \cos \alpha}{db} \times \delta,$$

and with $$\delta = \frac{9M_{t1}}{2t^2(2d+b)} \times \frac{d}{C} \times \sin \alpha,$$

$$M_b = \frac{2EI_{Rib} \cos \alpha}{db} \times \frac{9M_{t1}}{2t^2(2d+b)} \times \frac{d}{C} \times \sin \alpha$$

$$= \frac{9M_{t1}}{2t^2(2d+b)b} \times \frac{EI_{Rib}}{C} \times \sin 2\alpha. \quad . \quad . \quad . \quad (72)$$

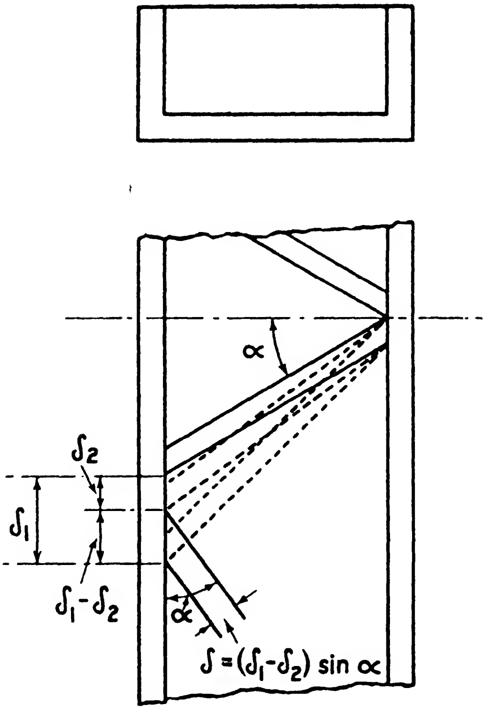


FIG. 142

The strain energy exerted by M_b is

$$U_2 = \frac{1}{2} \frac{M_b^2 \dfrac{b}{\cos \alpha}}{EI_{Rib}}$$

$$= 10{\cdot}1 \times \frac{M_{t1}^2 I_{Rib}}{t^4(2d+b)^2 b} \times \frac{E}{C^2} \times \frac{\sin^2 2\alpha}{\cos \alpha}.$$

In the case of steel we can write

$$\frac{E}{C}=2{\cdot}6,$$

hence $$U_2=26{\cdot}3\frac{M_{t1}^2}{t^4(2d+b)^2b}\times\frac{I_{Rib}}{C}\times\frac{\sin^2 2a}{\cos a}.$$

Since $$U_1=\frac{3M_{t1}^2l}{2t^3(b+2d)C},$$

and as the length covered by one rib is given by

$$l=b\tan a \text{ (see Fig. 140)},$$

$$U_1=\frac{3M_{t1}^2b\tan a}{2t^3(b+2d)C},$$

we obtain, therefore,

$$\frac{U_2}{U_1}=17{\cdot}5\times\frac{\sin^2 2a}{\sin a}\times\frac{I_{Rib}}{tb^2(2d+b)}$$

and $$\eta=1+17{\cdot}5\times\frac{\sin^2 2a}{\sin a}\times\frac{I_{Rib}}{tb^2(2d+b)} \quad . \quad . \quad (73)$$

But $$\eta=\frac{M_{t1}+M_{t2}}{M_{t1}}=\frac{M_t}{M_{t1}},$$

hence $$M_{t1}=\frac{M_t}{\eta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (74)$$

For the case of a torque M_t and with the knowledge of η (equation 73) we can determine M_{t1} (equation 74) and M_b (equation 72), and thus calculate the torsional stress f_t in the channel profile (equation 70) and the bending stress f_b in the rib (equation 71).

Bearings and Bearing Housings.

In the design of engineering structures the arrangement of bearings and bearing housings requires special consideration.

Large reduction gear-cases containing large gears and heavy shafts require split bearings and bearing housings so that the shafts and gears may be dropped into position. If such bearing housings are relatively simple they can be made by bending plates as shown in Fig. 143, or, if they are not too wide, they may be flame-cut from plate. In this case the thickness of the plate equals the length of the bearing housing (Fig. 144, Plate III). If, however, they are more complicated, steel castings may be used (see Fig. 6, Plate I).

In the arrangement (Fig. 143) it is necessary to check the stiffness of the wall portions which support the two sets of smaller top bearings. For this purpose it is assumed that the tooth pressure is equally distributed on the two bearings supporting each shaft.

An approximate check on the following lines is recommended. The " part assembly " in question (Fig. 143) can be considered as a cantilever (Fig. 145) subjected to deflection through bending and shear.

If the maximum horizontal component of the tooth pressure is 1300 lb. per bearing, the maximum side deflection * will be

$$\delta = \frac{1300 \times 24^3 \times 12}{3 \times 30000000 \times \cdot 5 \times 15^3}\left[1 + \cdot 98\frac{15^2}{24^2}\right]$$
$$= \cdot 002'',$$

which can be considered satisfactory in view of the fact that the

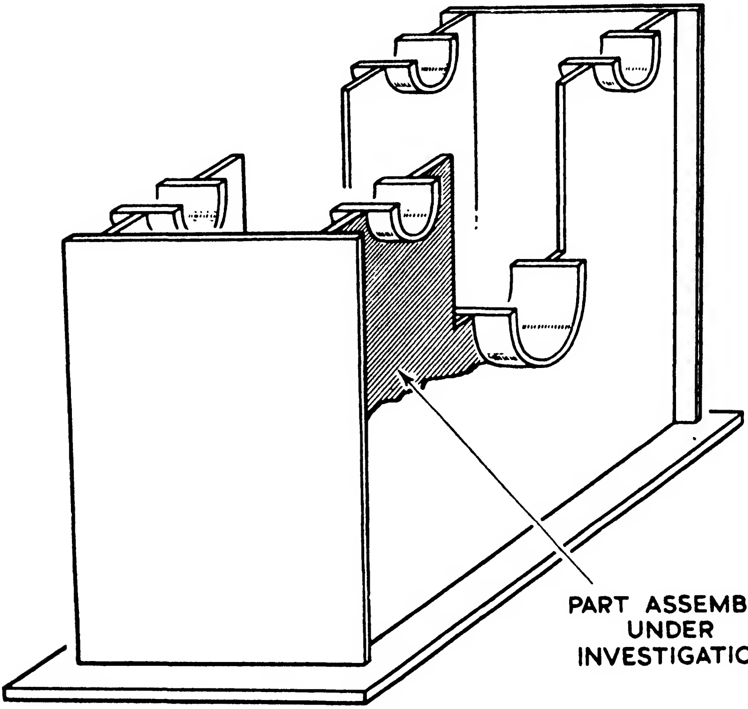


FIG. 143

influence of the sidewalls has not been considered, which, if anything, will still further reduce the above result.

When using ball or roller bearings, the width of which is small in relation to their diameter, it will often be cheaper to avoid additional housing bosses in the structure and to arrange for increased thickness of the main supporting walls (see page 6). In fabricated constructions the arrangement (Fig. 146) will, therefore, often be found instead of the casting-like arrangement of special bosses (Fig. 147).

If bosses are required, they can be arranged in different ways. The

* See Timoshenko, *Strength of Materials.*

alternatives (a) to (e) (Fig. 148) show various arrangements in order of cost of preparation, (a) being the cheapest, (e) the most expensive.

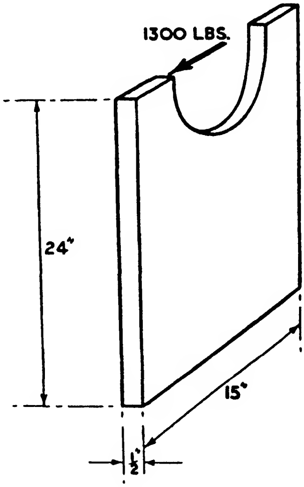


FIG. 145

Considering that the boring operation will be carried out after the welding operation, (a) requires no special preparation apart from parting

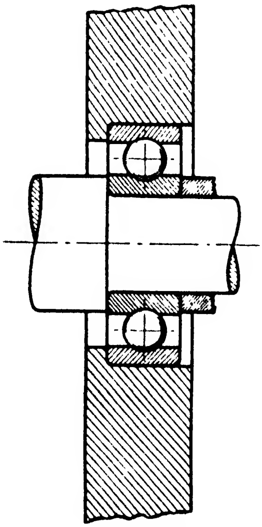


FIG. 146

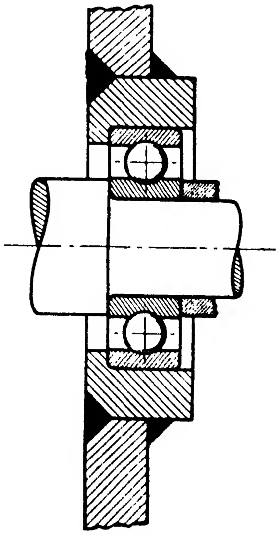


FIG. 147

the boss off a bar or cutting it from plate. For (b) a preliminary hole has to be cut in the plate, and in order to make this arrangement serve its purpose of locating the boss accurately in the plate, the diameter of the hole must be equal to that of the boss within the accuracy required for the welded assembly. Greater expense in plate and boss preparation has to be incurred if the boss has to be flush with the plate at one side, as in (c). Chamfering of boss and plate has to be carried out in order to obtain an efficient butt weld preparation on the left-hand side. If accurate axial location of the boss is desired, a shoulder may be turned on the boss. Alternative (d) shows the fillet-welded version of this arrangement, and (e) gives the alternative using flush butt welds.

Whilst reduction in fabricating cost is a very important factor in the design of welded structures, the advantage of accurate location as

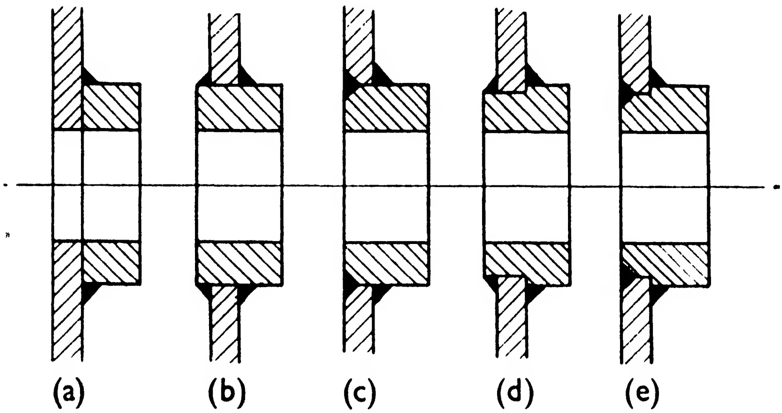


FIG. 148

shown in alternatives (b) to (e) can often outweigh the increase in cost of preparation. Locating holes or shoulders will eliminate the necessity of locating fixtures, and valuable assembly time can be saved through avoiding the marking-out operation for determining the position of the bosses.

If a boss has to carry heavy loads, alternative (a), with only one fillet weld in shear and sometimes bending, will be inferior in strength to alternatives (b) to (e), where not only two welds carry the load, but where the welds will be assisted by the circumferential support given to the boss by the plate itself.

Alternative (a) is also undesirable from the point of view of machining quality, as the boring tool may tend to jump at the joint face and a gap may develop if the plate is not flat or has warped during the welding operations.

The problem of bosses is somewhat different in the design of levers and mechanism links. In such cases it is usually desirable to make the boss flush with the profile of the lever, so that alternatives (b) and

(c) (Fig. 148) cannot be applied, This leaves available for the purpose an arrangement similar to alternative (a) but which has a prepared butt weld instead of the fillet weld (Fig. 149), or (d) or (e) (Fig. 148) may be used, but without the fillet weld on the right-hand side.

Good location can be obtained in the case of the arrangement Fig. 149 by placing the outer profile of the lever and the boss against a V-block or a similar fixture.

Projection welding of bosses to levers is a very favourable alternative. When using spigoted bosses (see Fig. 105) accurate location can be obtained without special fixtures. The use of this method is limited mainly—as mentioned previously—by the capacity of the welding plant.

As an example of a lever arrangement, the lever (Fig. 150) should

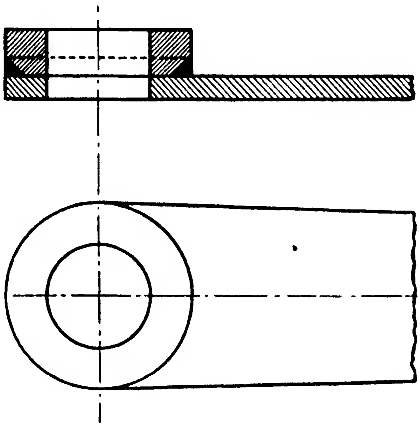


FIG. 149

be designed to carry a torque equal to that which can be transmitted by the shaft. With a permissible torsional stress of 6 tons/sq. inch, the shaft of $\frac{7}{8}''$ diameter will transmit a torque given by

$$M_t = \frac{\pi \times \cdot 875^3}{16} \times 6$$
$$= \cdot 78 \text{ ton inches.}$$

The shear stress in the $\frac{5}{16}''$ diameter high tensile steel taper pin is

$$f_{s\ \text{Pin}} = \frac{\cdot 78 \times 4}{\cdot 875 \times \pi \times \cdot 3125^2}$$
$$= 11 \cdot 6 \text{ tons/sq. inch.}$$

The $\frac{1}{8}''$ deep butt weld is calculated as a ring section (1·5″ outside diameter and 1·25″ inside diameter),

or
$$f_{s\ \text{Weld}} = \frac{\cdot 78 \times 32 \times 1 \cdot 5}{\pi \times (1 \cdot 5^4 - 1 \cdot 25^4) \times 2}$$
$$= 2 \cdot 3 \text{ tons/sq. inch.}$$

The weld will, therefore, easily transmit the required torque.

When using projection-welded spigoted bosses, the shear stress in the projected area must be below the permissible value. If sound welds are obtained the welded area can be assumed equal to that covered by the full boss ($1\frac{1}{2}''$ diameter) less that covered by the spigot ($1\frac{1}{4}''$ diameter) (Fig. 151).

If, instead of using a boss and a taper pin or a key, the lever is

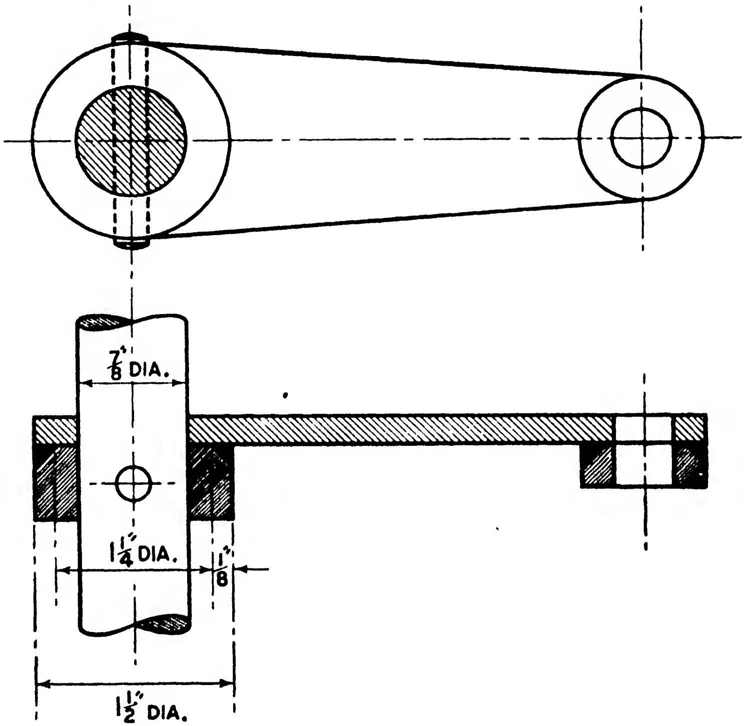


FIG. 150

fillet-welded directly to a shaft, the shear stress in the fillet weld may be determined by using equation 50.

In the crankshaft arrangement (Fig. 152) the shaft of 2″ diameter will transmit—at 6 tons/sq. inch permissible torsional stress—a torque of

$$M_t = \frac{\pi \times 2^3}{16} \times 6$$

$$= 9{\cdot}4 \text{ ton inches,}$$

which corresponds to a crank force

$$P = \frac{9{\cdot}4}{2{\cdot}25} = 4{\cdot}2 \text{ tons.}$$

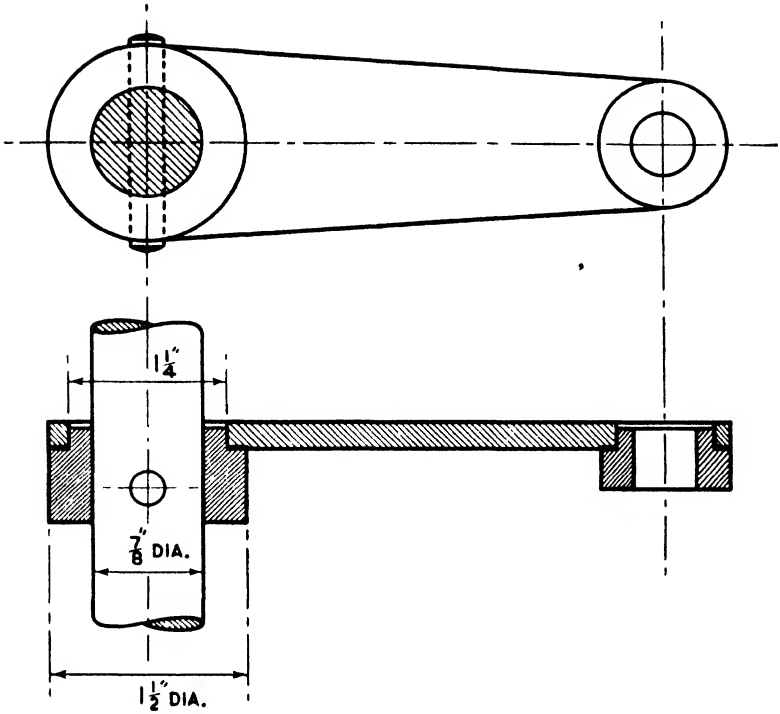


FIG. 151

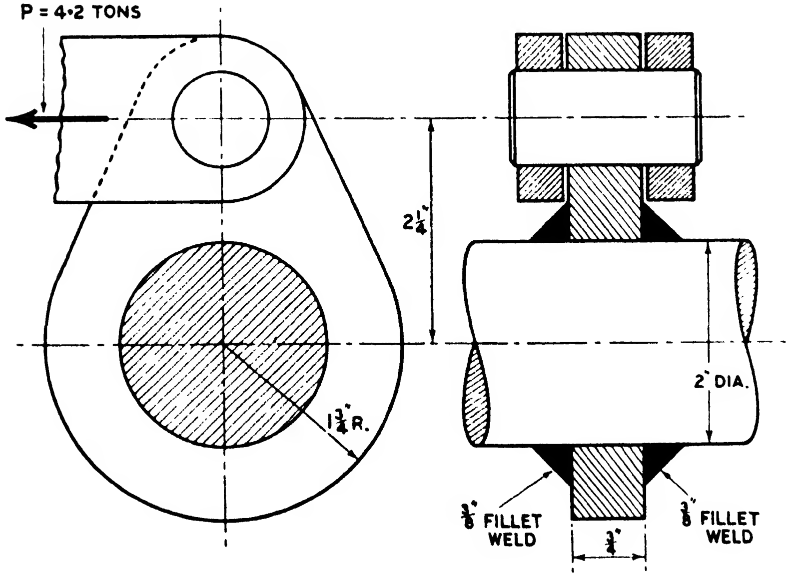


FIG. 152

If the permissible bending stress in the crank arm is $f_b = 8$ tons/sq. inch, its minimum thickness t_{min} is calculated from

$$\frac{M_t}{f_b} = Z$$

or

$$\frac{9 \cdot 4}{8} = \frac{(3 \cdot 5^3 - 2^3)t_{min}}{12} \times \frac{1}{1 \cdot 75},$$

giving

$$t_{min} = \cdot 71''.$$

Because of the connecting links which transmit the force of 4·2 tons, the size of a full circumferential fillet weld is limited to $\frac{3}{8}''$.

In case of only one circumferential $\frac{3}{8}''$ fillet weld the stress in the weld would, according to equation (50), be given by

$$f_s = \frac{\cdot 9 \times 9 \cdot 4}{\cdot 375(2 + \cdot 19)^2}$$
$$= 4 \cdot 75 \text{ tons/sq. inch.}$$

If it is feared that the lever may get out of square through inaccurate fitting on the shaft or through one-sided welding, a symmetrical arrangement of fillet welds on both sides (Fig. 152) can be used. The stress in each weld will then be halved, or

$$f_{s\,(2\ \mathrm{Welds})} = \frac{4 \cdot 75}{2} = 2 \cdot 375 \text{ tons/sq. inch.}$$

Facings and Pads.

The question of warping which creates gaps between two abutting faces has been mentioned in connection with the arrangement of bosses (see page 136). The problem becomes more important still in the case of facings and pads for machining.

If facings, other than very small ones, are welded to a plate (Fig. 153(a)), buckling and warping may occur under the welding heat (Fig. 153(b)). Subsequently, when machining the face to its required height, excessive thinning of the facing plate may occur where the warping is greatest, thus weakening the part of the structure which is to carry the load of another component fastened to it. Vibrations and drumming of the supporting plate may result. It is recommended, therefore, that the arrangement shown in Fig. 153(c) should be used, which, moreover, requires less welding than the one shown in Fig. 153(a).

It will be appreciated that the danger of warping, and its detrimental effects on superimposed plates, is small in the case of narrow facing strips, and the problem can generally be considered as unimportant if the width of the superimposed plate is less than twelve times its thickness.

If this width is exceeded, and other design considerations prevent the use of one thick plate instead of superimposed thinner plates, a plug weld in the middle portion of the plates (Fig. 153(d)) may be employed. The diameter of the plug weld should be two to four times the thickness of the superimposed plate.

It should be borne in mind, however, that plug welding requires extra labour for drilling holes and welding, and that it creates additional heat, thus giving rise to further distortions and stresses. Plug welds

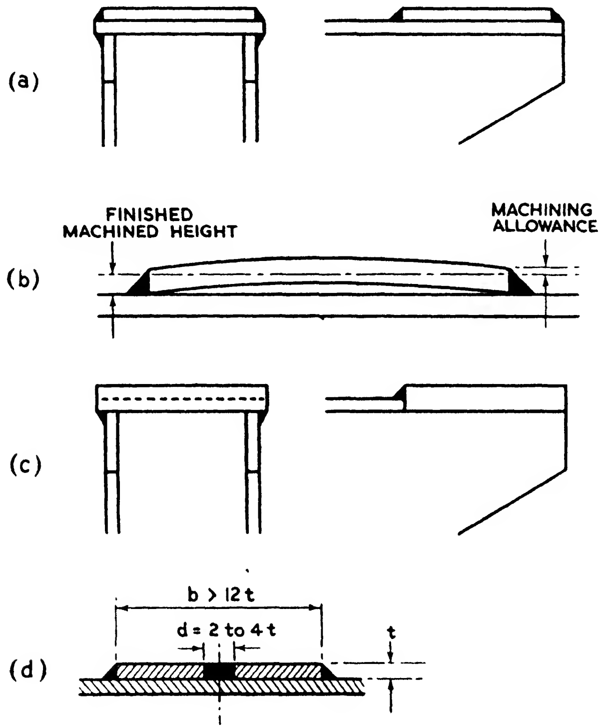


FIG. 153

should, therefore, be used only if no alternative method of design can be found.

Plates.

When attaching plates which have to be machined, to webs or supporting plates at right angles, various alternatives may be considered (Fig. 154).

The cheapest way, requiring no special plate-edge preparation, is alternative (a). As the horizontal plate will usually be under load, the fillet welds should be strong enough to carry the load entirely, as, with the tolerances used in the fabricating shop, a gap will usually exist between the two abutting plates. The reduction in effective weld size caused by such a gap must also be taken into account (see Fig. 223).

In case of a highly stressed beam structure, the abutting face of the vertical plate must be carefully smoothed after flame cutting, in order to eliminate little notches which might act as stress raisers (see page 185). These difficulties are avoided in the arrangement (b) (Fig. 154), where

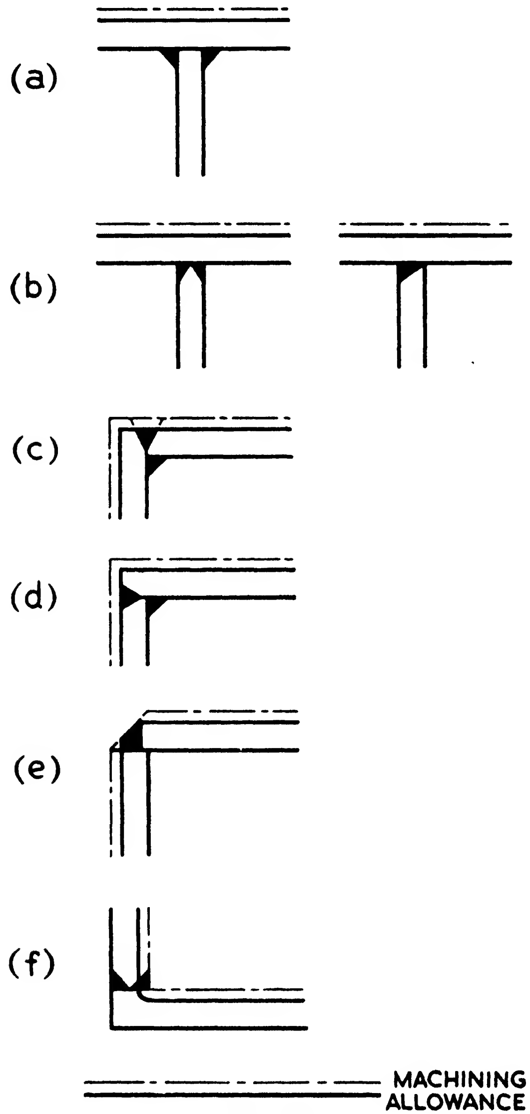


FIG. 154

the weld covers any possible gap and also melts the metal at the flame-cut edge, thus eliminating any stress raisers automatically.

The problem becomes more involved if both the horizontal top face and the vertical side face have to be machined, as in alternatives (c), (d), (e) (Fig. 154). It will be seen that, while arrangements (c) and (d) require similar V or J preparation, in the arrangement (d) the vertical plate will support the horizontal one during assembly and thus locate it automatically. There is the possibility, however, that different hardnesses of the weld and parent plate may cause a mark on the machined surface, and the weld may have to be arranged in whichever surface such a mark will be of least importance.

Alternative (e) (Fig. 154) shows an arrangement which requires no plate edge preparation, but which, however, does not provide for a sharp edge between the two machined faces.

A more intricate problem is raised by an arrangement where two internal faces have to be machined at right angles. In such a case care must be taken that the machining operation does not remove the inner weld, thus considerably weakening the whole assembly. The arrangement (f) (Fig. 154) avoids this danger.

Pressure Vessels.

In the design of pressure vessel part-assemblies, due consideration must be given to the stresses created by static and pulsating loads, the possible influence of corrosion and heat, and the danger to human life in case of failures. Various authorities have laid down specifications with regard to design principles, permissible stresses, and inspection procedure in order to ensure soundness of the parent material (see Fig. 5) and of the welds.* The permissible stresses as specified by the various authorities differ considerably in some cases.‡

For a certain permissible stress at the main welded joints, where this is the dominating factor, the thickness of the plate material (and the weld sizes) will depend upon the size of the vessel, the maximum internal working pressure which has to be withstood, and the allowance for wastage during the operating life of the vessel.

With regard to the welded joints, several authorities use an efficiency factor based upon riveted joint practice, and which is intended to allow for inconsistencies in the welds. It has been pointed out, however, by Heeley † and Dorey ‡ that in the case of class 1 pressure vessels—with the present precautions (*i.e.* supervision of welding procedure, stress relieving heat treatment, X-ray inspection, and mechanical testing of welds)—there should be no reason against assuming the strength of the butt welds to be equal to that of parent metal containing the usual surface discontinuities, such as blemishes, stamp marks, etc.

* See B.S.S. 487, 1099, 1113 ; Rules issued by Lloyd's Register of Shipping, the Associated Offices Technical Committee, the British Corporation, the American Society of Mechanical Engineers (Power Boiler Code), the American Petroleum Institute (the American Society of Mechanical Engineers).

† E. J. Heeley, " Some Considerations in the Design of Class 1 Pressure Vessels ", *Proceedings of the Institution of Mechanical Engineers*, June 1946, page 22.

‡ S. F. Dorey, " Note on Design Stresses in Class 1 Welded Pressure Vessels ", *Proceedings of the Institution of Mechanical Engineers*, June 1946, page 32.

There are in this country several published formulae for calculating the wall thickness of cylindrical pressure vessels where butt welds are used for longitudinal or circumferential seams. Examples of these are the formulae given in B.S.S. 487, 1099, 1113, Lloyd's Register, A.O.T.C., and British Corporation Register of Aircraft. In the following notes, some of these formulae have been modified by the author for the purpose of unifying the symbols and units used by the different authorities.

The following symbols should be noted :

C_1 = constant, used in the calculation of the thickness of cylindrical shell plates (B.S.S. 487).
- $C_1 = 12{\cdot}5$ for a longitudinal butt joint welded from one side only.
- $C_1 = 17{\cdot}5$ for a longitudinal butt joint welded from both sides.

C_2 = constant, used in the calculation of the thickness of cylindrical shell plates for small air receivers (B.S.S. 1099).
- $C_2 = 400$ for a longitudinal butt joint welded from one side only.
- $C_2 = 500$ for a longitudinal butt joint welded from both sides.

C_3 = allowances for corrosion wastage or erosion (recommended by Heeley).

C_4 = constant, used in the calculation of the thickness of outwardly dished end plates (B.S.S. 487 and 1099).
- $C_4 = 17{\cdot}5$ for receivers up to and including 6 ft. diameter.
- $C_4 = 15{\cdot}0$ for receivers over 6 ft. and up to and including 9 ft. diameter.
- $C_4 = 12{\cdot}0$ for receivers over 9 ft. diameter.

C_5 = constant, used in the calculation of the thickness of inwardly dished end plates (B.S.S. 487 and 1099).
- $C_5 = 0$, if $\frac{R}{T}$ is under 30.
- $C_5 = 15$, if $\frac{R}{T}$ is under 40 but not under 30.
- $C_5 = 20$, if $\frac{R}{T}$ is under 50 but not under 40.
- $C_5 = 25$, if $\frac{R}{T}$ is under 160 (maximum permissible) but not under 50.

C_6 = constant, used in the calculation of the thickness of flat end plates (Lloyd's Register of Shipping, Rules and Regulations).
- $C_6 = 72{\cdot}5$, when the end plate is exposed to flame.
- $C_6 = 100$, when the end plate is not exposed to flame.

C_7 = constant, used in the calculation of the thickness of inwardly dished end plates (Lloyd's Register of Shipping, Rules and Regulations).

$$\left.\begin{array}{l} C_7 = 8000 \text{ for T not exceeding } \frac{9}{16}'' \\ C_7 = 8800 \text{ for T exceeding } \frac{9}{16}'' \end{array}\right\} \text{for class 1 pressure vessels.}$$

D = internal diameter of the shell in inches.

D_e = external diameter of the shell in inches.

E = efficiency of ligaments between tube holes or other openings in the shell, or of longitudinal joints.

F = permissible stress in lb./sq. inch for plate material at temperatures up to 260° C.

K = factor dependent upon the ratio between the external height of dishing and the outside diameter of the drum end.

R = inner radius of curvature of the end plate in inches (R must not exceed D).

S = minimum ultimate tensile stress of the shell plate material in tons/sq. inch.

T = minimum thickness of shell plate in inches (in case of small fusion welded steel air receivers, T must not be less than ·04″ or $\frac{D_e}{100}$, whichever is greater).

WP = maximum permissible working pressure in lb./sq. inch.

d = diameter of flat portion of end plate in inches.

f = permissible working stress in lb./sq. inch at working temperature of the metal.

h = external height of dishing.

r = internal blending radius between the curved and straight parts of dishing.

Shell Plate Thickness.

B.S.S.—Formulae :

(*a*) For Fusion-Welded Steel Air Receivers : *

$$T = \frac{WP \times D}{32C_1 \times S} + \cdot 06. \qquad (75)$$

(*b*) For Small Fusion-Welded Steel Air Receivers : †

$$T = \frac{WP \times D_e}{C_2 \times S} + \cdot 03. \qquad (76)$$

(*c*) For Water Tube Boilers : ‡

$$T = \frac{D}{2}\left(\sqrt{\frac{fE + WP}{fE - WP}} - 1\right). \qquad (77)$$

* Extracted from B.S. 487, Fusion-Welded Steel Air Receivers, by permission of the British Standards Institution, 28 Victoria Street, London, S.W.1, from whom official copies of the specification can be obtained, price 2s. post free.

† Extracted from B.S. 1099, Small Fusion-Welded Steel Air Receivers, by permission of the British Standards Institution, 28 Victoria Street, London, S.W.1, from whom official copies of the specification can be obtained, price 2s. post free.

‡ Extracted from B.S. 1113, Water Tube Boilers and their Integral Superheaters, by permission of the British Standards Institution, 28 Victoria Street, London, S.W.1, from whom official copies of the specification can be obtained, price 7s. 6d. post free.

Lloyd's Register of Shipping Formula : *

$$T = \frac{WP \times D}{2f} + \cdot 06, \qquad (78)$$

where, in the case of class 1 pressure vessels, f can be obtained from the following table :

	Minimum Tensile Stress of Plate,	
	26 tons/sq. inch	28 tons/sq. inch
Unfired vessels	12,900 lb./sq. inch	13,900 lb./sq. inch
Fired vessels	12,200 lb./sq. inch	13,100 lb./sq. inch

These formulae cover vessels of class 1, class 2, and class 3 qualities, in which the main difference is the safe stress-carrying capacity of the welded seams. The values to be ascribed to the various factors are not readily apparent from the formulae employed, even for vessels of "class 1" quality, where the welded seams are of the highest quality.

Against this variety of formulae, Heeley † has suggested the use, in the design of class 1 vessels working at ordinary temperatures, of the following formula, which excludes the doubtful efficiency factor :

$$T = \frac{WP \times D}{29120 - WP} + C_3. \qquad (79)$$

This is the "mean diameter" formula with a stress of 6·5 tons per sq. inch on the main seams. In the case of vessels of a lower standard or quality, a suitable value of maximum permissible working stress should be used, 4·5 and 3·5 tons respectively being considered appropriate.

End Plates

Various methods of attaching end plates to the cylindrical shell are recommended by British Standard Specifications, by Lloyd's Rules, and by other authorities.

Dished ends are frequently used (Figs. 155, 156, 157), butt welds between the dished end and the shell (Fig. 155) being the only joint allowed by Lloyd's for use in class 1 pressure vessels.

In the case of flat end plates, particular care must be taken in the arrangement of the welds, in order to avoid any abrupt change in the stress flow which might create stress raisers.

The arrangements shown in Figs. 158 and 159 are recommended by B.S.S., while Figs. 160 and 161 show some of those recommended by Lloyd's. Similar considerations to those used for flat end plates apply

* Lloyd's Register of Shipping, Rules.

† E. J. Heeley, "Some Considerations in the Design of Class 1 Pressure Vessels", *Proceedings of the Institution of Mechanical Engineers*, June 1946, page 22.

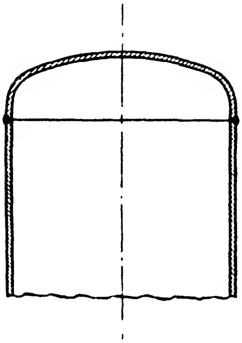


FIG. 155

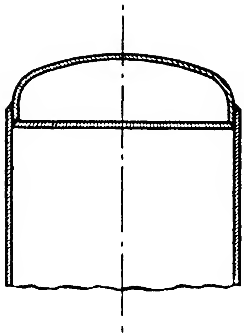


FIG. 156

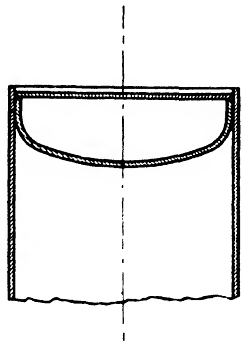


FIG. 157

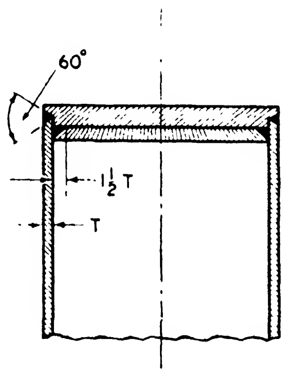


FIG. 158

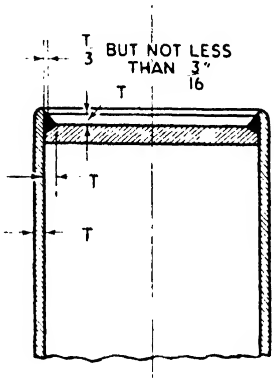


FIG. 159

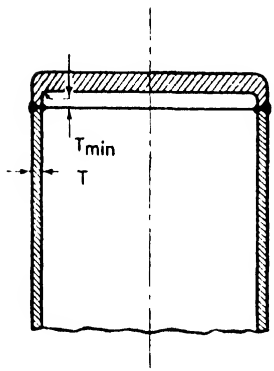


FIG. 160

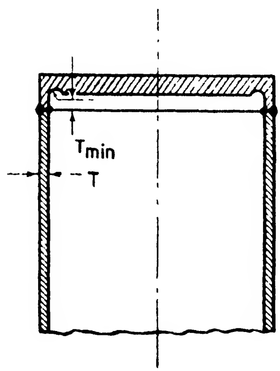


FIG. 161

in the case of end flanges to which end covers are bolted (Fig. 162).* Formulae for the calculation of the wall thickness of end plates are :

B.S.S.—Formulae:

(*a*) For Fusion-Welded Steel Air Receivers : †

$$T=\sqrt{\frac{WP\times D^2}{3072S}} \quad \text{(unflanged flat end plates)} \qquad (80)$$

$$T=\sqrt{\frac{WP\times d^2}{4300S}}+\cdot 03 \quad \text{(flanged flat end plates)} \qquad (81)$$

$$T=\frac{WP\times R}{32C_4\times S} \quad \text{(outwardly dished plates)} \qquad (82)$$

$$T=\frac{WP\times R}{384S} \quad \text{(inwardly dished plates for pressures 200 lb./sq. inch and over)} \qquad (83)$$

$$T=\frac{(WP+C_5)\times R}{384S} \quad \text{(inwardly dished plates for pressures under 200 lb./sq. inch)} \qquad (84)$$

(*b*) For Water Tube Boilers : ‡

$$T=\frac{F}{f}\left(\frac{WP\times D_e\times K}{1184\times S}+\cdot 1\right) \quad \text{(dished drum ends)} \qquad (85)$$

Lloyd's Register of Shipping Formulae : §

$$T=\sqrt{\frac{WP\times D^2}{1024C_6}}+\cdot 03 \quad \text{(flat end plates)} \qquad (86)$$

$$T=\frac{WP\times D\times K}{2363S}\left[\frac{\left(20\frac{r}{R}\right)+3}{\left(20\frac{r}{R}\right)+1}\right]+\cdot 03 \quad \text{(outwardly dished end plates for class 1 pressure vessels)} \qquad (87)$$

$$T=\frac{WP\times R}{C_7} \quad \text{(inwardly dished end plates)} \qquad (88)$$

For the attachment of flanges, B.S.S. 487 recommends arrangements (Figs. 163, 164, 165) and Lloyd's requirements are on similar lines, with certain limitations in pressure and temperature. For the attachment of branches and flanges, Heeley || has put forward recommendations (Figs. 166 and 167) which give full consideration to the important question of accessibility and the resulting degree of certainty of the soundness of the welds.

An interesting detail problem, similar to that of welding flat end plates to cylindrical vessels, has occurred in the welding of a distribution

* This drawing has been put at the author's disposal by H. Diederich, Esq.

† B.S.S. 487 and B.S.S. 1099 (see before).

‡ B.S.S. 1113 (see before).

§ Lloyd's Register of Shipping, Rules.

|| E. J. Heeley, "Some Considerations in the Design of Class 1 Pressure Vessels", *Proceedings of the Institution of Mechanical Engineers*, June 1946, page 22.

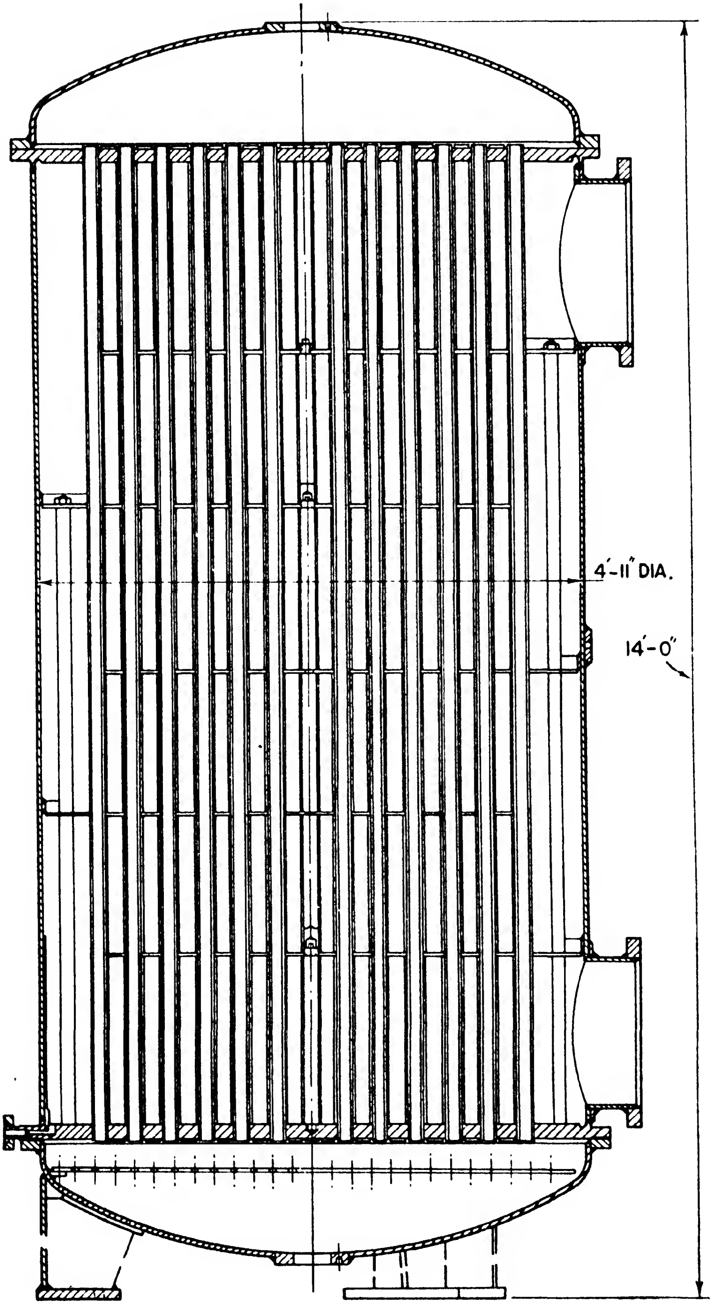


Fig. 162

Welded Cooler for Compressor Plant (40 lb./sq. inch)

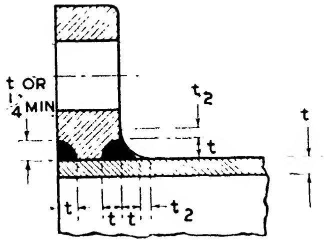


FIG. 163

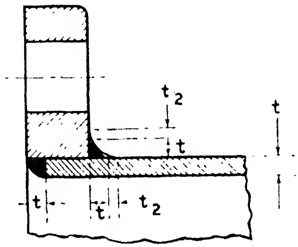


FIG. 164

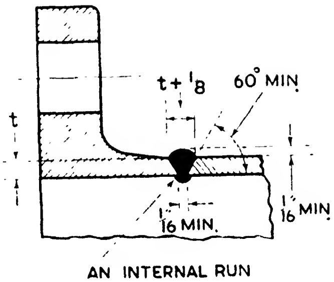


FIG. 165

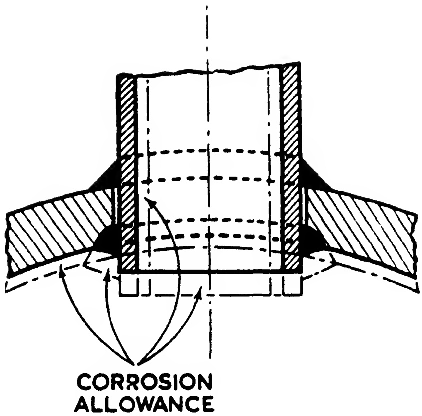


FIG. 166

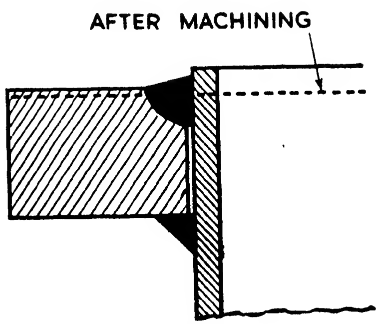


FIG. 167

head to a forged high pressure vessel.* With the original design (Fig. 168) cracking was experienced in the weld, due to the different

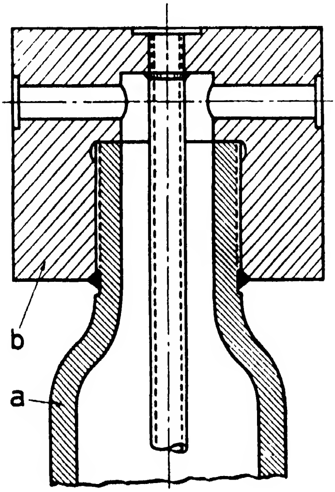


FIG. 168

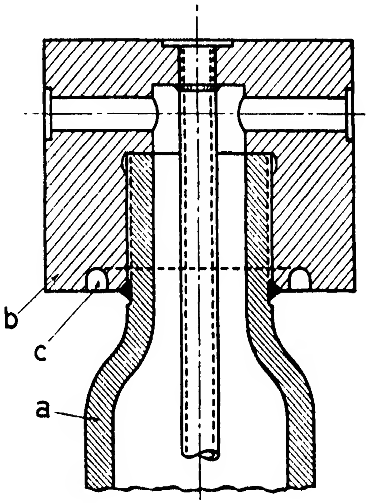


FIG. 169

contractions of the main body of the vessel *a* and the connecting piece *b*. A groove *c* turned into the connecting piece (Fig. 169) gave the latter sufficient elasticity to avoid cracking, and subsequent heat treatment eliminated any dangerous residual stresses.

* This has been communicated to the author by H. Diederich, Esq.

Chapter 5

TYPICAL WELDED STRUCTURES

(*Practical application of the principles laid down in the previous chapters*)

THREE different types of structure will be considered :

A. Structures without special requirements of accuracy.
B. Structures which have to preserve their accuracy under working conditions, the degree of accuracy depending upon the special requirements of the job.
C. Vessels which have to be pressure-tight and safe and, therefore, require the highest welding quality, as leakages or failures may endanger human life.

A. Welded Structures without Special Accuracy Requirements

It frequently occurs that high accuracy or limitation of distortions under load is only of secondary importance when compared with the load-carrying capacity of a component. This is often the case in building structures, but crane bridges and bed plates are typical examples which arise in general engineering practice. If, in the design of such constructions, full economic use of the material is to be made, careful stress calculation is of great importance.

Crane Bridge.

The arrangement of the rolled sections in the lattice construction of the crane bridge (Fig. 170) is practically identical with that in a riveted construction, and the forces acting on the various members are determined by the usual graphical method.

The nominal loading capacity of this crane is 10 tons. With regard to the loading conditions of the crane bridge itself (allowance for shock loads and overloads), the specifications laid down in B.S.S. 466 must be observed.

Table (Fig. 171) shows the procedure for calculating the required sections and the resulting stresses under the wheel pressure of the crab and the weight of the bridge itself, with due allowance for the most unfavourable loading conditions.

Only the calculation of complete members of the lattice girder is

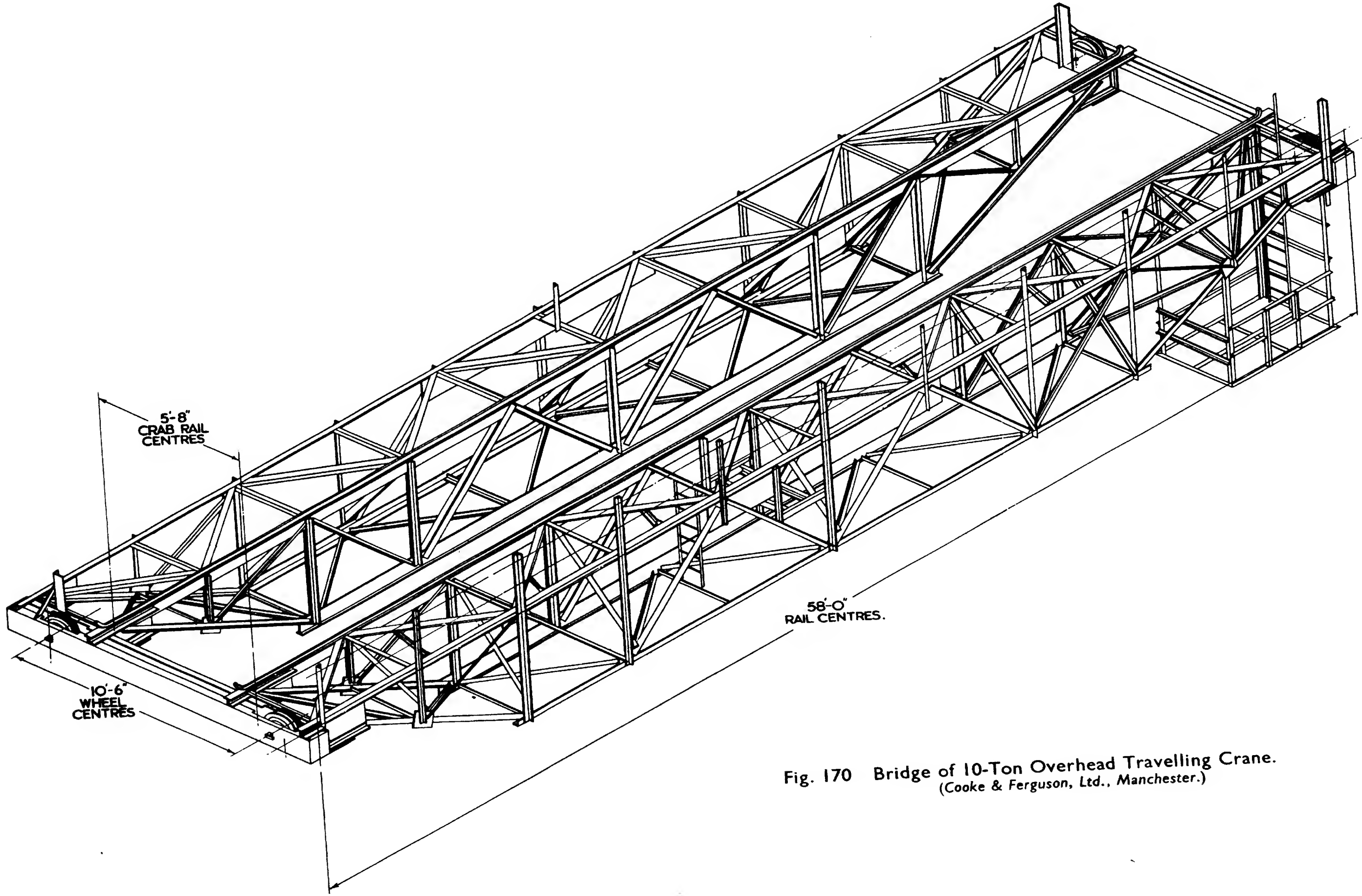
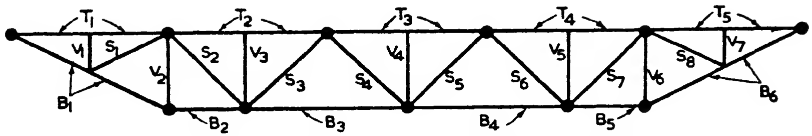


Fig. 170 Bridge of 10-Ton Overhead Travelling Crane.
(Cooke & Ferguson, Ltd., Manchester.)



1	2	3	4	5	6	7	8	9	10							
MEMBER	FORCE P (+=TENSION −=COMP.R)	FREE BUCKLING LENGTH l	SECTION	AREA A	MOMENT OF INERTIA I	RADIUS OF GYRATION k	$\frac{l}{k}$	PERMISSIBLE STRESS f_1 x)	ACTUAL STRESS $f_2 = P/A$	WELDS CONNECTING ANGLE SECTIONS xx) X_1	X_2	l_2	h_2	l_1	h_1	STRESS f_w
	LBS.	INCHES		SQ.INCHES	INCHES⁴	INCHES		LB/SQ.INCH	LB/SQ.INCH	INCHES	INCHES	INCHES	INCHES	INCHES	INCHES	LB/SQ.INCH
$T_1 = T_5$	−41560	68	2·66″; 4″ × 5/8″; 8″ × 3/8″	5·5	41·5	2·75	25	11800	7600	—	—	—	—	—	—	—
$T_2 = T_4$	−47180	68		5·5	41·5	2·75	25	11800	8600	—	—	—	—	—	—	—
T_3	−53880	68		5·5	41·5	2·75	25	11800	9800	—	—	—	—	—	—	—
$B_1 = B_6$	+46200	—	2 − 2½″ × 2½″ × 3/8″ L; ·75″	3·46	—	—	—	13400	13350	1·75	·75	7	·375	4·5	·25	8700
$B_2 = B_5$	+36620	—	4″ × 3/8″; 4″ × 5/8″	4·0	—	—	—	13400	9200	—	—	—	—	—	—	—
$B_3 = B_4$	+52980	—		4·0	—	—	—	13400	13250	—	—	—	—	—	—	—
$V_1 = V_7$	−11172	34	2 − 1¾″ × 1¾″ × ¼″ L	1·62	·44	·52	66	9450	6900	1·23	·52	2·5	·25	2·2	·125	8700
$V_2 = V_6$	−18020	68	2 − 3″ × 3″ × ¼″ L	2·88	2·4	·91	75	8600	6300	2·17	·83	4·3	·25	3·3	·125	8600
$V_3 = V_4 = V_5$	−11172	68	2 − 2½″ × 2½″ × 5/16″ L	2·92	1·66	·75	91	7100	3850	1·77	·73	2·6	·25	2·1	·125	8700
$S_1 = S_8$	+ 744	—	2 − 1¾″ × 1¾″ × ¼″ L	1·62	—	—	—	13400	460	1·23	·52	2·5	·25	2·2	·125	590
S_2 (S_7)	+21630	—	2 − 2¼″ × 2¼″ × ¼″ L	2·12	—	—	—	13400	10300	1·61	·64	5	·25	4	·125	8800
S_3 (S_6)	−17180	96	2 − 3½″ × 3½″ × 5/16″ L	4·18	4·76	1·07	90	7300	4150	2·53	·97	4·1	·25	3·1	·125	8600
S_4 (S_5)	+13243	—	2 − 3½″ × 3½″ × ¼″ L	3·38	—	—	—	13400	3950	2·55	·95	3·2	·25	2·4	·125	8600
S_5 (S_4)	−9325	96	2 − 3½″ × 3½″ × ¼″ L	3·38	3·88	1·07	90	7300	2800	2·55	·95	3·2	·25	2·4	·125	6000
S_6 (S_3)	+5395	—	2 − 3½″ × 3½″ × 5/16″ L	4·18	—	—	—	13400	1290	2·53	·97	4·1	·25	3·1	·125	2700
S_7 (S_2)	−1484	96	2 − 2¼″ × 2¼″ × ¼″ L	2·12	·98	·68	141	3700	700	1·61	·64	5	·25	4	·125	600

NOTES: x) THE PERMISSIBLE STRESS IN M S CRANE STRUCTURES IS 6 TONS/SQ IN THE SAFE WORKING STRESS UNDER COMPRESSION (BUCKLING) HAS BEEN BASED ON THAT FIGURE AND HAS BEEN WORKED OUT IN ACCORDANCE WITH BSS 449

xx) SEE FIG. 75—AS EACH OF THE MEMBERS CONSISTS OF TWO ANGLES EACH SET OF WELDS $(l_1 h_1 - l_2 h_2)$ PER ANGLE HAS TO CARRY $P/2$

FIG. 171

shown. In the case of compression members which are built of two or more angles, $\frac{l'}{k_1}$ for each part member must not be greater than $\frac{l}{k}$ for the complete member and not greater than 30. In order to fulfil this condition, intermediate connections, *a*, between the part members have to be arranged (Fig. 172) at intervals equal to the permissible free buckling length, l', of the part members.

The calculation of the sizes of the transverse members which have

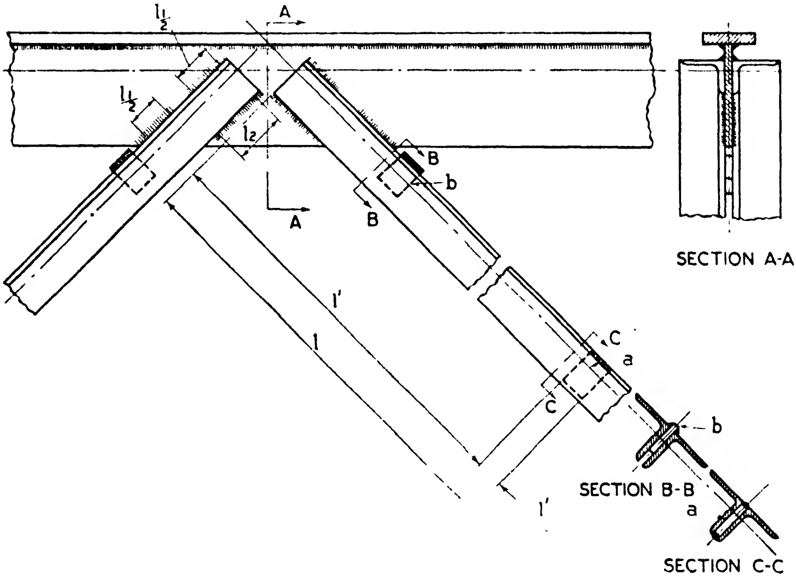


FIG. 172

to carry the load created by the braking forces is carried out in a similar manner.

As there is no need to allow for rivet holes in the rolled sections, he full value of their sectional area and moment of inertia can be used in the calculations. This results not only in a reduction in weight as compared with the riveted construction (the saving in weight of the bridge, Fig. 170, was about 30 per cent) but in an improvement of the load transmission between the various members.

The welds connecting the angles are calculated in accordance with the principle shown in Fig. 75 (see page 60).

In order to prevent the development of additional bending stresses in the welds, reinforcement strips, *b*, are welded to the compression members near the welds connecting the vertical and sloping members to the top and bottom beams (Fig. 172). These welds are stressed in shear.

The arrangement shown in Fig. 173 is often valuable with regard to space saving, while obtaining good accessibility of all welds.

When designing a welded crane bridge for outdoor use, care should be taken to prevent corrosion of the welds from the inside, and a good sealing of the gap underneath the welds, or continuous runs, even of small size, are advisable.

Bed Plates.

In the design of bed plates the two fundamental requirements are

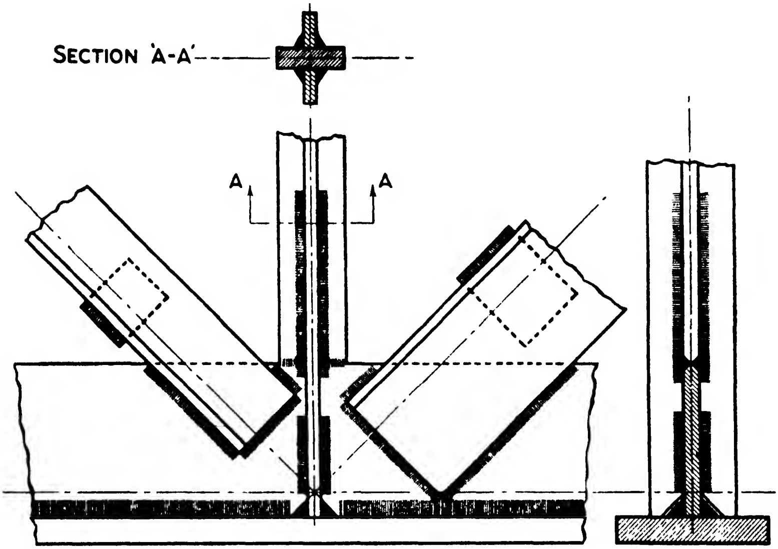


FIG. 173

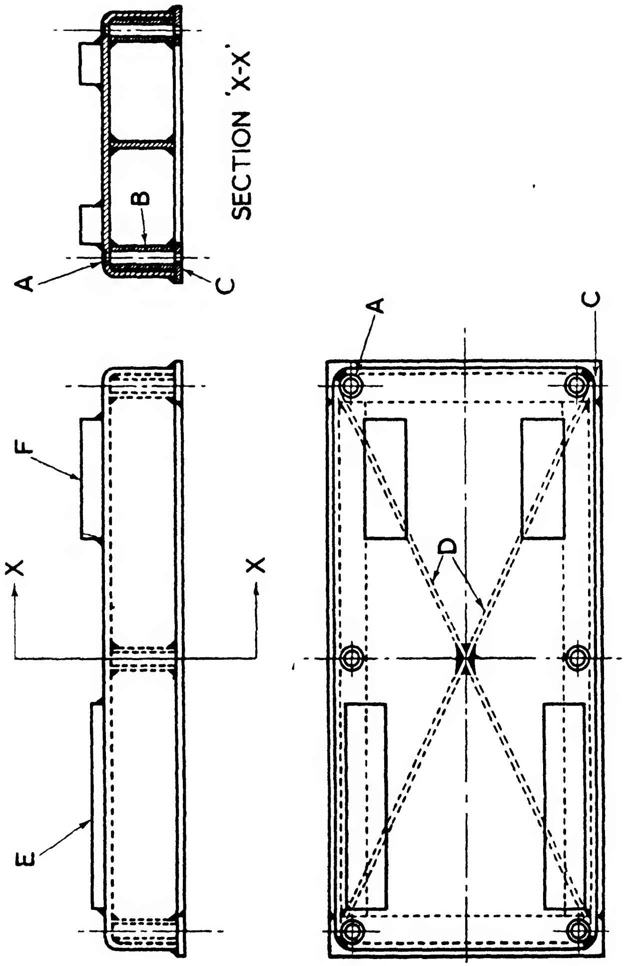
the capacity to support the weight of the various units (machines, motors, etc.) fastened to the bed, and the strength to transmit the forces between the units. Furthermore, the location of these units must be such as to guarantee, within the required limits of accuracy, the alignment of corresponding shafts.

The bed plate (Fig. 174) has been designed for a motor compressor unit. The usual design procedure is as follows :

(1) The determination of the positions of the machined faces which are to carry the bases of the various units in question, by laying out the whole assembly on a common centre line.
(2) The determination of the height of machining pads, if required, and from there—
(3) Designing the complete bed plate.

The bed plate (Fig. 174) is built from one main plate only, which is bent to the desired shape. Thus a minimum amount of welding is required, the total length of weld being four times the depth of the bed.

Spot faces at A replace the machining bosses usually found on a casting. The load exerted by the foundation bolts is supported by



Bed Plate for Motor-Compressor Unit (Cooke & Ferguson, Ltd., Manchester)

Fig. 174

tubular sections B, and the whole structure is carried on a base frame C, built of flats. The bed plate itself has to carry the torque which is transmitted between the driving shaft of the motor and the driven shaft of the compressor, the reactions of which are transmitted to the facings E and F through the bases of the motor and the compressor respectively. The diagonal stiffeners D give the structure added strength against that torque.

A large-size bed plate used for supporting a group of special-purpose electrical machinery, weighing about 30 tons, is shown in Fig. 175. The central portion of this bed plate had to be open, in order to clear the bottom portion of the stator frame of one of the machines. The design is based on a frame built of 24″ × 7½″ joists which at intervals are stiffened by stiffener webs ¾″ thick. In order to clear fillet welds which cross the webs, the latter have their adjacent corners cut off at 45°, as described previously (see Fig. 123).

This special-purpose bed plate, of which only one will ever be required, is a typical example of the economically advantageous use of a welded construction. If a casting had been used, the cost and manufacturing time of a pattern alone would probably have exceeded that of the complete fabrication.

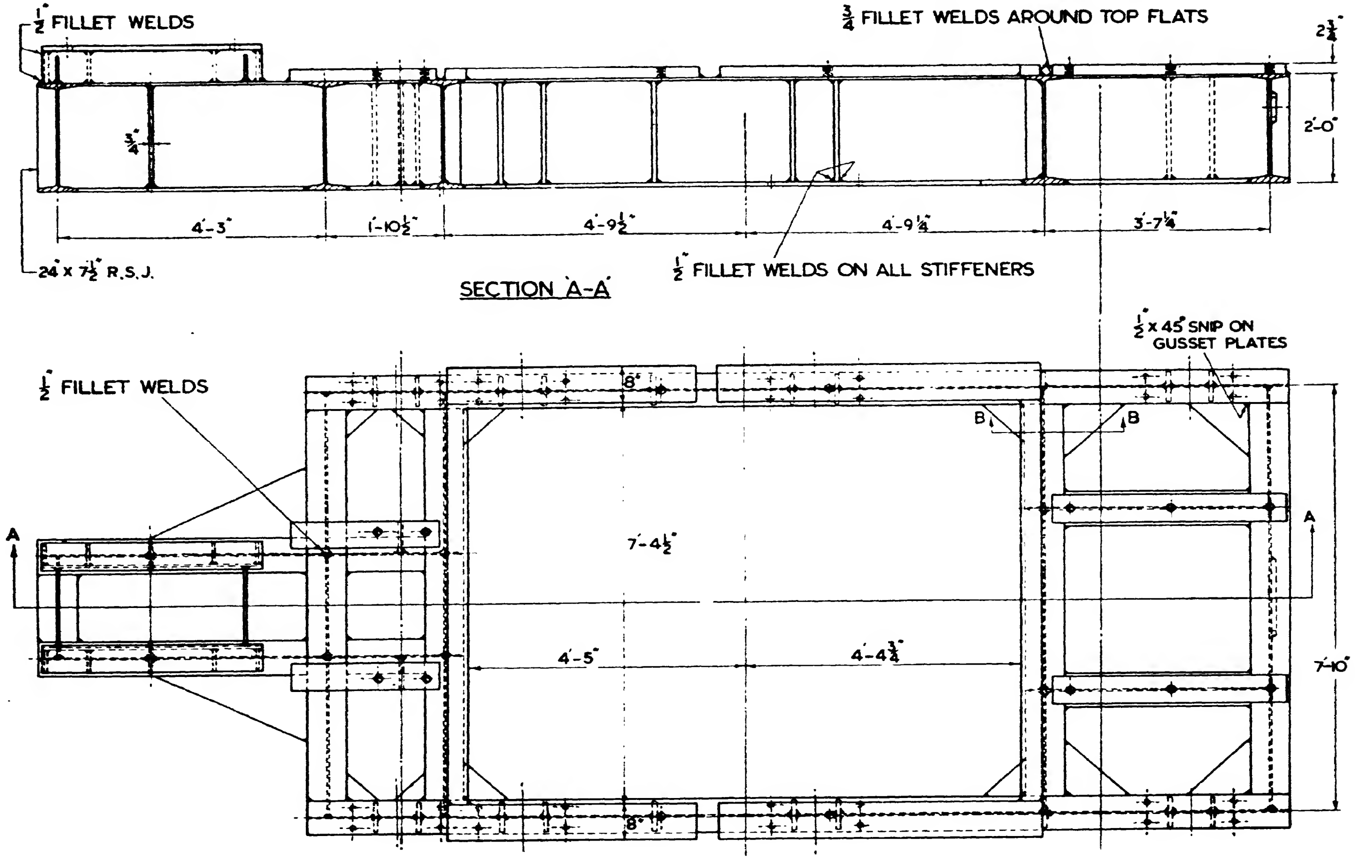
Saving in labour can sometimes be obtained by the use of spot welding instead of arc welding, but the designer should consider carefully whether or not the size, shape, and arrangement of the bed plate are suitable for this method of manufacture.

The bed plate (Fig. 176, Plate III) has been designed for spot welding, as the flanges of the channels and the flats used as supporting faces for the machinery lend themselves to that method. As the flat faces are machined after welding, care must be taken with regard to the specification of the welding procedure in order to avoid hardening at the points where the electrode tips are applied. Prolonged welding times at reduced amperage were the solution to this problem. Whilst the stressing of the spot welds under working conditions of the bed plate is negligibly low, they might be highly stressed during the machining operations of the facing flats. This point should be considered carefully when deciding upon the arrangement of the welds.

Some typical applications of resistance welding to components of smaller gauge material are shown in Figs. 177 to 181 (Plates IV and V).*

Figs. 177 and 178 (Plate IV) show the sub-assembly of a bomb tail. The parts are projection-welded in order to reduce the welding time and to obtain closer pitch between the welded points (see page 82). The outer ring is welded in a fixture (Fig. 177, Plate IV) which holds it truly circular. The inner webbing is composed of three identical pressings suitably designed for the resistance welding operation and in such a way as to allow easy access to the top and bottom electrode (Fig. 178, Plate IV). For the final assembly the parts are slipped on a simple angular fixture (Fig. 179, Plate IV) and welded in three operations. The simple component design providing ample area for the contact surfaces should be noted.

* These photographs of resistance-welded components have been kindly provided by Messrs. Metropolitan-Vickers Electrical Co. Ltd., Trafford Park, Manchester. The tank (Fig. 180) is included, although it is made of non-ferrous material, as it is an excellent example of using spot welding for the fixing of stiffeners.



½" FILLET WELDS
¾" FILLET WELDS AROUND TOP FLATS
2¾"
2'-0"
¾"
4'-3"
1'-10½"
4'-9½"
4'-9¼"
3'-7¼"
24" x 7½" R.S.J.
½" FILLET WELDS ON ALL STIFFENERS
SECTION 'A-A'
½" x 45° SNIP ON GUSSET PLATES
½" FILLET WELDS
8"
B
B
A
A
7'-4½"
4'-5"
4'-4¾"
7'-10"
8"

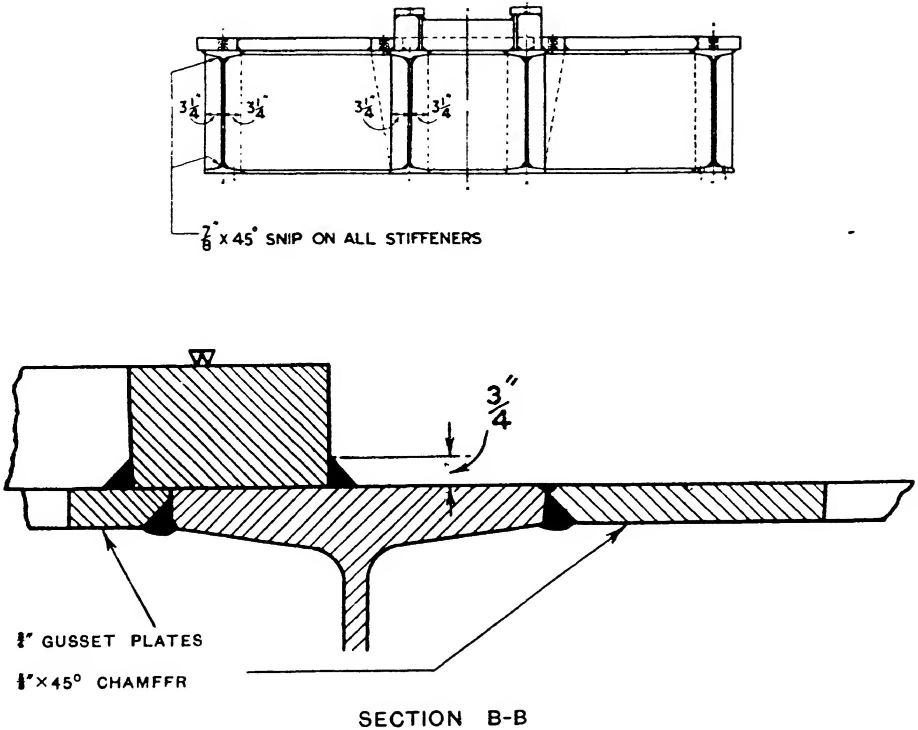


FIG. 75—Heavy Bed Plate for a Group of Electrical Machinery (Cooke & Ferguson, Ltd., Manchester

The U-shaped stiffeners for the " Lancaster " tanks are spot-welded to the panels (Fig. 180, Plate V) before these are closed by gas welding. The arrangement of the spots can be seen clearly on the right-hand side of the panels (see also Fig. 218, Plate XV). The foot brackets of the tank (Fig. 181, Plate V) are offset, as the fastening holes must be in line with the rounded corners of the tank. In order to reduce the size of the brackets, projection welding has been used, as spot welding would have required too great a spacing between the spots, and correspondingly longer flanges on the brackets. The dimpled projections can be clearly seen on the loose brackets.

The destructive test of one of the welds shows that the weld is not breaking, but that a nugget is taken out of the parent metal of the bracket.

B. Welded Structures which must preserve Accuracy

In general mechanical engineering, stiffness against distortion is usually at least as important as strength, though the necessary degree of stiffness depends upon the nature of the structure.

The variety of these requirements can be illustrated in connection with the design of presses : punching, deep drawing, and moulding presses require the parallelism between table and ram to be maintained to a high degree of accuracy, while for stretching or straightening presses a greater latitude in the degree of stiffness is permissible. When designing the main frames of such presses, therefore, the designer bases his first calculations for the lay-out on stresses only ; subsequently, when the general design has been established, he will check whether the deflection under load is within permissible limits.

The two main frames of the 300-ton straightening press * (Fig. 182, Plate VI) were originally designed as steel castings (Fig. 183a). Due to manufacturing difficulties, it was decided to use fabricated frames instead of castings, and the design (Fig. 183b) was adopted and produced † (Fig. 184, Plate VI).

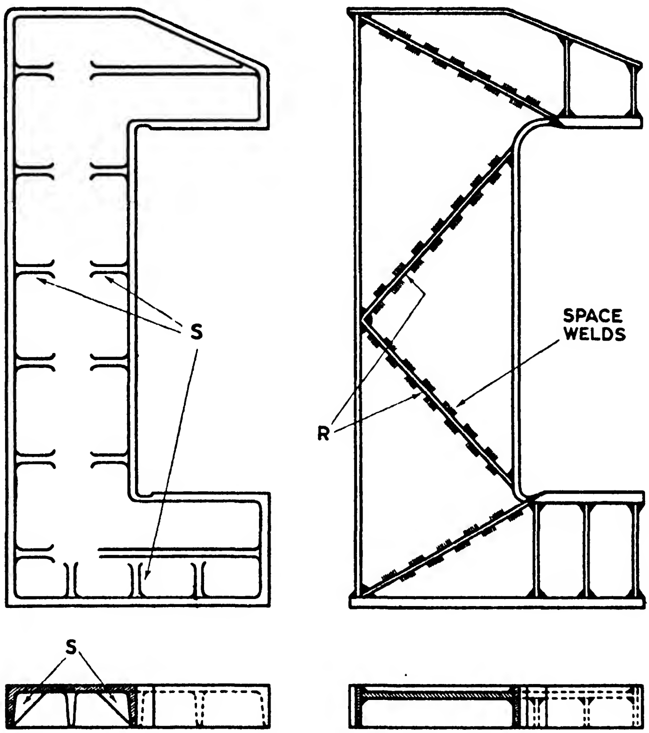
The design is based on the fact that it is possible to fabricate a section which is both deeper and of smaller wall thickness than a casting, with the result that considerable saving in material can be achieved. Instead of the small triangular stiffeners (S) required on the casting, diagonal ribs (R) are space-welded across the free area of the main plate in order to stiffen the frames against any possible twist during machining, handling, or fitting operations.

The load of 300 tons is applied by means of a hydraulic cylinder

* This press has been designed and built by the Hydraulic Engineering Co., Ltd., Chester.

† Cooke & Ferguson, Ltd., Manchester.

and ram, the cylinder being bolted between facings (1) on the inner sides of the two frames, the vertical thrust being taken on facings (2) (Fig. 185). Facings (3), (4), (5) serve for the fastening of distance pieces between the frames, while facings (6) support the long table carrying the work-piece which is to be straightened. An upward force



(a) CAST STEEL FRAME (b) FABRICATED FRAME

FIG. 183

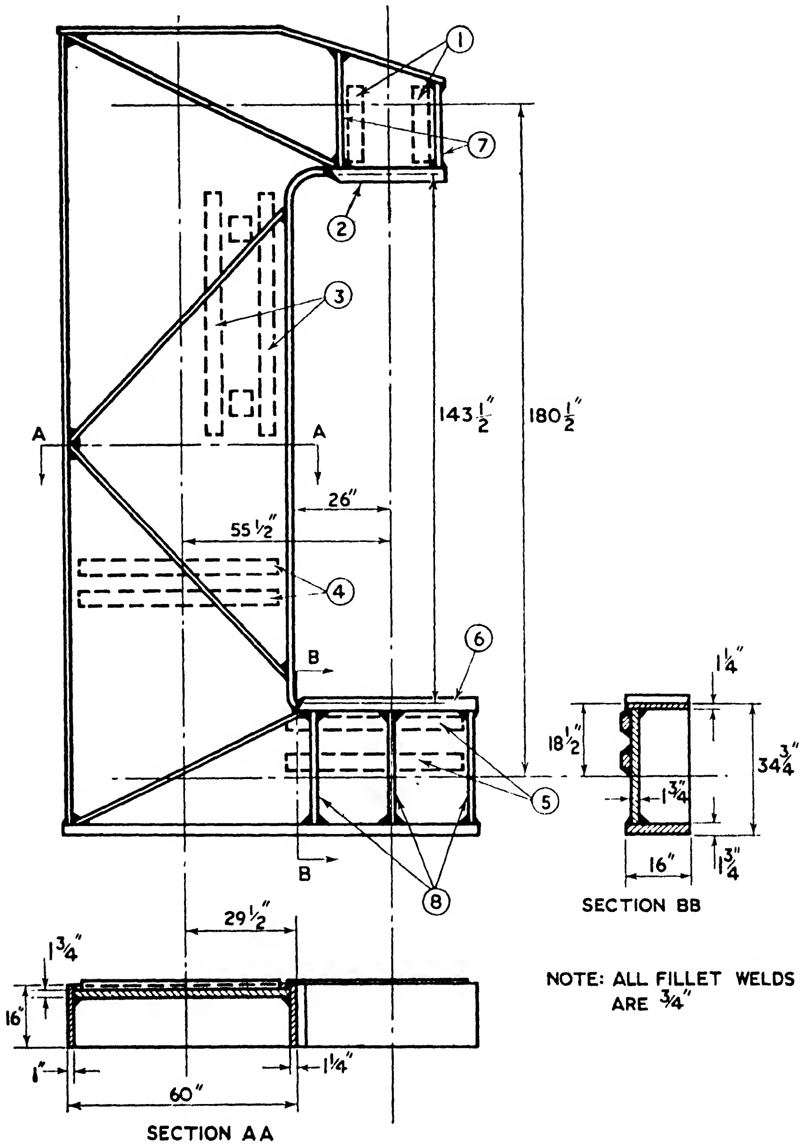
of 150 tons is thus transmitted to each frame, through the cylinder facings (2), the reaction being downwards through the table facings (6). Both these frames are supported against side distortion by stiffeners (7), (8), which are to prevent any bending under the application of the load which may have slight sideways eccentricity.

Neglecting the diagonal ribs, the cross-section (A-A) in the middle of the frame (Fig. 185) has a sectional area

$$A_1 = 135 \text{ sq. inch}$$

and a moment of inertia

$$I_1 = 63500 \text{ inch}^4.$$



1
7
2
3
143 1/2″
180 1/2″
A
A
26″
55 1/2″
4
6
B
B
5
8
1 1/4″
18 1/2″
34 3/4″
1 3/4″
16″
1 3/4″
SECTION BB
29 1/2″
1 3/4″
16″
1″
1 1/4″
60″
SECTION AA
NOTE: ALL FILLET WELDS ARE 3/4″

FIG. 185

The bending stress on the tension side of this section is

$$f_{b1} = \frac{150 \times 55\cdot5 \times 29\cdot5}{63500}$$

$$= 3\cdot88 \text{ tons/sq. inch ;}$$

the tensile stress is $$f_{t1} = \frac{150}{135} = 1\cdot12 \text{ tons/sq. inch ;}$$

and the total stress on the tension side

$$f_1 \text{ total} = 3\cdot88 + 1\cdot12$$

$$= 5 \text{ tons/sq. inch} \quad \text{(tension).}$$

The total stress on the compression side is smaller, because f_{t1} has in this case to be deducted from the bending stress on the compression side, which is

$$f'_{b1} = \frac{150 \times 55\cdot5 \times 30\cdot5}{63500}$$

$$= 4 \text{ tons/sq. inch,}$$

or $$f'_1 \text{ total} = 4 - 1\cdot12$$

$$= 2\cdot88 \text{ tons/sq. inch} \quad \text{(compression).}$$

The most highly stressed vertical section (B–B) has a sectional area

$$A_2 = 102 \text{ sq. inch}$$

and a moment of inertia

$$I_2 = 16800 \text{ inch}^4.$$

The bending stress on the tension side is

$$f_{b2} = \frac{150 \times 26 \times 18\cdot5}{16800}$$

$$= 4\cdot3 \text{ tons/sq. inch,}$$

and the shear stress * $$f_{s2} = \frac{150}{56}$$

$$= 2\cdot68 \text{ tons/sq. inch.}$$

This shear stress occurs in the middle of the section where the bending stress is approaching zero.

The greatest shear stress in a weld occurs in the weld connecting the $1\frac{3}{4}''$ thick flange to the web, and depends upon the ratio $\frac{G}{I}$ (see page 118).

For section B–B

$$\frac{G_2}{I_2} = \frac{28 \times 15\cdot375}{16800}$$

$$= \cdot026.$$

* As a good approximation the maximum shear stress at the centre of the section can be found by dividing the shearing force by the cross-sectional area of the web alone. (See Timoshenko, *Strength of Materials*.)

The shear force per inch length, in this section, is therefore

$$P_{s2} = 150 \times \cdot 026$$
$$= 3{\cdot}9 \text{ tons.}$$

The shear stress in the two $\frac{3}{4}''$ fillet welds connecting the flange to the main plate is therefore (equation 25a)

$$f_{sw} = \frac{\cdot 707 \times 3{\cdot}9}{1 \times \cdot 75}$$
$$= 3{\cdot}7 \text{ tons/sq. inch.}$$

The calculation of this press frame is completed by making an approxi-

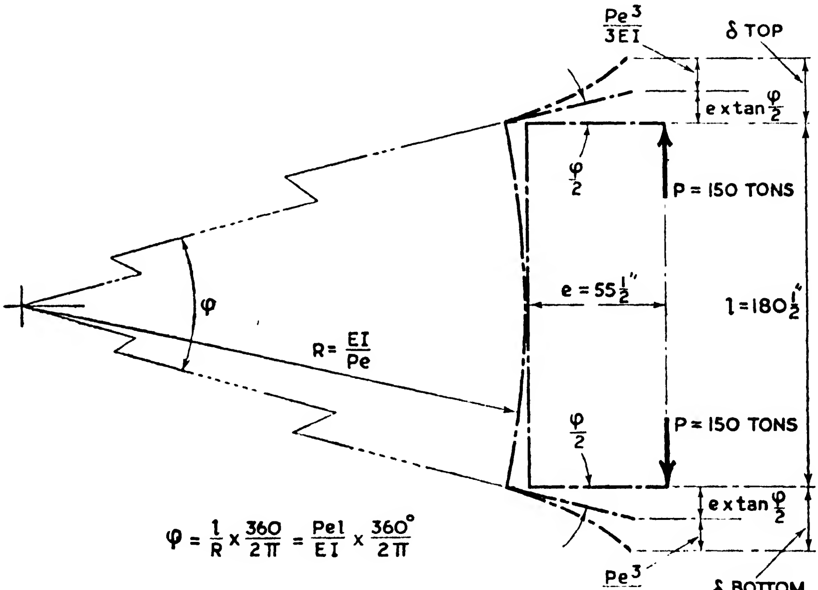


FIG. 186

mate calculation of the total deflection at the centre-line of the press ram (Fig. 186), considering the corners stiff enough to maintain the right angle between the vertical portion and the two horizontal portions of the frame, and neglecting the stiffening influence of the 3″ thick facing plates (2) and (6).

The deflection angle of the vertical portion is (Fig. 186)

$$\Phi = \frac{Pel}{EI} \times \frac{360}{2\pi}$$
$$= \frac{150 \times 2240 \times 55{\cdot}5 \times 180{\cdot}5}{30000000 \times 63500} \times \frac{360}{2\pi}$$
$$= \cdot 1^\circ$$
$$= 0^\circ 6'.$$

Assuming that the average moment of inertia of the top horizontal portion is equal to that of the bottom horizontal portion, we obtain

$$\delta_{\text{Top}} = \delta_{\text{Bottom}} = 55{\cdot}5 \times \tan 0°3' + \frac{150 \times 2240 \times 55{\cdot}5^3}{3 \times 30000000 \times 16800}$$

$$= {\cdot}05 + {\cdot}038$$

$$= {\cdot}088'',$$

$$\delta = \delta_{\text{Top}} + \delta_{\text{Bottom}} = {\cdot}176'',$$

which is high, but still permissible for this type of press.

Conditions are different if parallelism and flatness of the table and top plates of a press have to be accurate to within small limits. In the case of a column type press the stiffness of the cross-head, base, and table must be such as to prevent any considerable deflecting moment being transmitted to the column itself.

The parts of the 750-ton press (Fig. 187, Plate VI, designed and built by Fielding and Platt Ltd., Gloucester) have been designed to give very great strength and stiffness, at the same time exploiting the possibilities of the fabrication method to the full. The design of the press head (Fig. 188), which carries the three hydraulic cylinders, uses a reinforced box section (1–2–3–4) to take the cylinder bodies, and this section is connected by diagonal stiffeners (5) to the corners in which the reaction of the pressing force is transmitted to the columns. In this way strength and stiffness of the head are obtained in just those parts where they are most required, and no unnecessary weight is used. Smaller stiffeners (6) support the main plates (7, 8) at the points where the force of the return cylinders is applied.

The design of the moving table (Fig. 189) is of particular interest in so far as the accessibility of the welds is concerned. It will be noted that the stiffeners have to be assembled between the main top and bottom plates (1, 2) starting from the centre rib (3). Whilst the longitudinal ribs are bevel-butt welded from the outside, the cross ribs can only be welded on three sides, gaps (4) being left between each cross rib and the succeeding longitudinal rib. This does not impair the stiffness of the table, however, as it is the welded connections between the cross ribs and the top and bottom plates which transmit the shearing forces created by the bending action of the pressing force.

The machining allowance for the face taking the ram pressure is obtained by inserting a thicker plate (5), thus avoiding the difficulties which would be caused by a superimposed plate (see page 140). Steel castings are used for the guide bearings which locate the table on the columns and which would be rather difficult to fabricate. On both cross-head and moving table, the relieving of those corners of ribs which cross fillet welds (see page 113) will be noted.

In the design of fabricated structures for precision machine tools the requirements of the machines must be carefully considered when laying

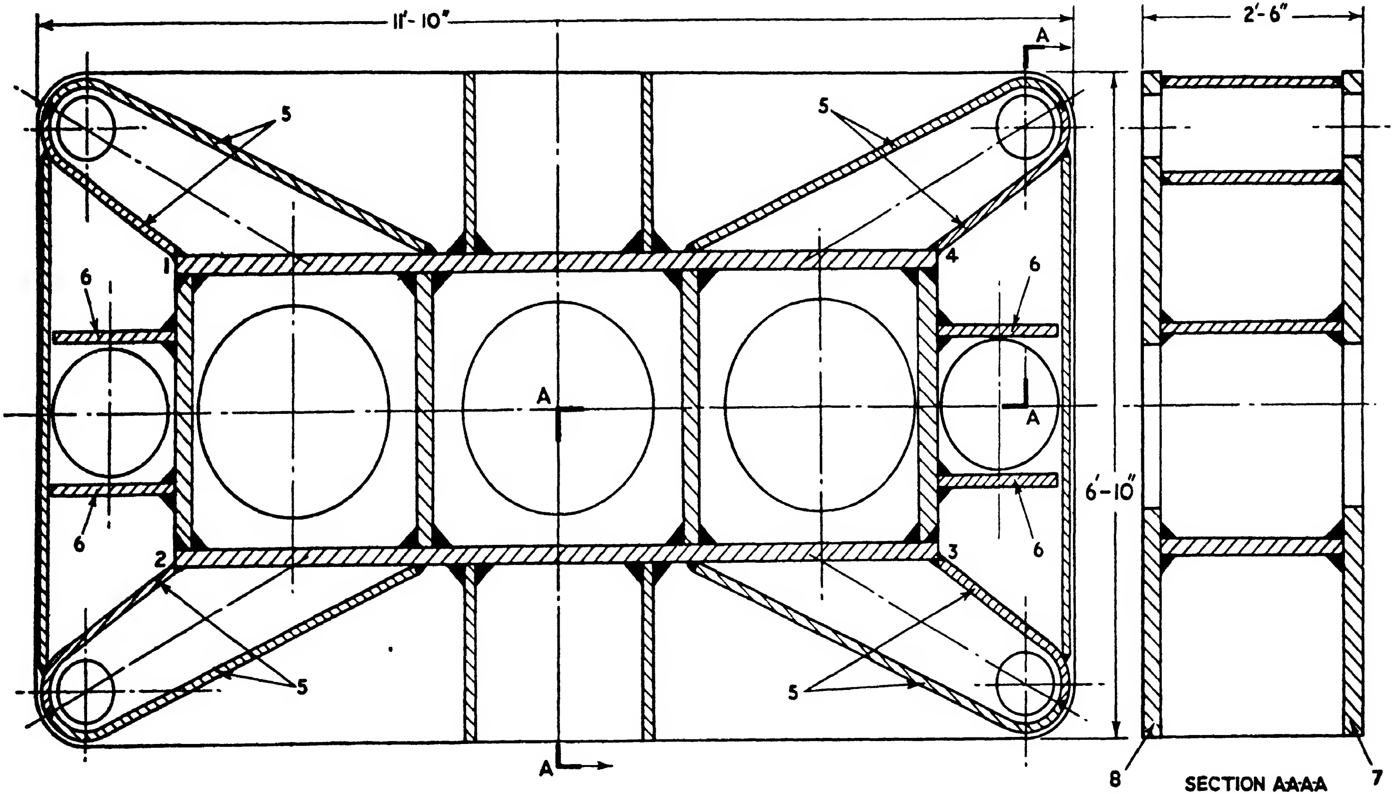


FIG. 188.—Head of Press (Fig. 187, Plate VI).

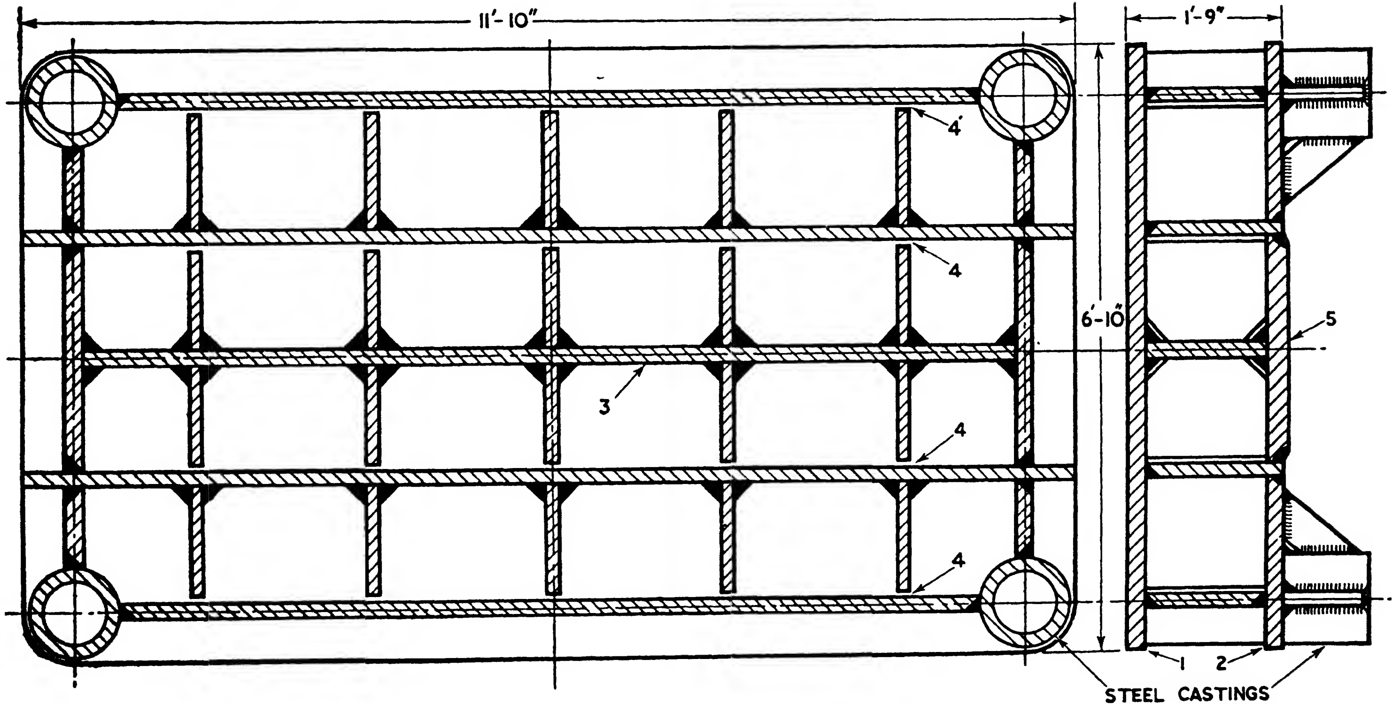


FIG. 189.—Moving Table of Press (Fig. 187, Plate VI).

out the design of the various parts. Machine tools have to produce articles which have one, or several, of the following qualities :

(*a*) a specified accuracy of shape ;
(*b*) a specified accuracy of dimensions ;
(*c*) a specified degree of surface finish.

For this purpose machine tools have to be designed and built in such a way as to

(1) Give—within the limits required to satisfy points (*a*), (*b*), and (*c*)—accurate alignment and guidance to the tool and/or work-piece in every position during their various movements.
(2) Maintain the accuracy mentioned under (1) during the operation of the machine, *i.e.* when the whole structure is under load created by the weight of the work and by the cutting forces.

Whilst point (1) depends mainly upon the quality of workmanship applied during the manufacture of the machine tool, point (2) is determined by the design, the term design comprising the lay-out of the structure as well as the choice of the material to be used. It is, again, the designer's skill in using the properties of steel to the fullest extent, when designing a fabricated machine-tool structure, which will produce success or failure, technically and economically, of the design.

From the start of laying out the machine, the designer must decide which parts may suitably be made in fabricated steel construction, and arrange his plans accordingly. It is difficult to give any definite recommendations as to which parts of machine tools are suitable for fabricated construction, each case having its own problems, but some typical suggestions, incorporated in the table * (Fig. 190), may give general guidance.

An example of a machine tool in which fabricated construction has been used to a great extent is the production milling machine shown in Figs. 191 (Plate VII) and 192.† It should be noted that this type of construction has not been used indiscriminately, but only where it has been considered the most suitable for the purpose. Whilst the main body, the upright, the outer stay, the headstock, and the overarm are fabricated structures, the saddle is a casting, as it is rather complicated, owing to its incorporation of the control gear and the various devices for driving the table feeds.

The machine body (Fig. 193), which consists mainly of three $\frac{3}{8}''$ thick plates (1, 2, 3) bent to the required shape, is supported by a frame of five $3\frac{1}{2}''$ by $1''$ flats (4). Whilst, in general, the stiff closed box section is maintained, a sloping plate (5) allows cuttings and coolant to get freely through openings (6) into the tray provided for the purpose (see

* F. Koenigsberger, " Fabrication of Machine Tools by Welding ", *The Workshop Yearbook and Production Engineering Manual* (I), Paul Elek, London.

† See F. Koenigsberger, " Design and Construction of Machine Tool Parts in Fabricated Steel ", *Machinery* (May 8th, 1941), page 149.

TYPE OF MACHINE	PART OF MACHINE	QUALITIES INFLUENCED BY MATERIAL PROPERTIES					STABILITY (WEIGHT)	STEEL OR CAST IRON CONSTRUCTION SUGGESTED
		TENSILE STRENGTH	STIFFNESS AGAINST DISTORTIONS	STRENGTH AGAINST SHOCK LOADS	RIGIDITY AGAINST VIBRATIONS: DUE TO PULSATING LOADS	RIGIDITY AGAINST VIBRATIONS: DUE TO RESONANCE		
CENTRE LATHE TURRET LATHE CAPSTAN	BED	NOT FULLY EXPLOITED AS RIGIDITY AND STIFFNESS ARE OF FIRST IMPORTANCE	VERY IMPORTANT	SHOCK LOADS ARE NOT LIKELY TO OCCUR	LOADS ARE USUALLY FAIRLY UNIFORM	IMPORTANT FOR HIGH SPEED MACHINES	NOT IMPORTANT IF ROTATING MASSES ARE WELL BALANCED	VERY SUITABLE FOR STEEL CONSTRUCTION
DRILLING MACHINES	BASE-PLATE							
	COLUMN							
	RADIAL ARM (for RADIAL DRILLS)							
GRINDING MACHINES	BED					IMPORTANT AS SPEEDS ARE HIGH FOR THE GRINDING WHEEL DRIVE.	GRINDING WHEEL HAS TO BE WELL BALANCED FOR SATISFACTORY WORK	IF REQUIREMENTS DO NOT NECESSITATE TOO COMPLICATED A DESIGN, STEEL CONSTRUCTION SHOULD BE FAVOURABLE
	GRINDING HEAD							
MILLING MACHINE	BASE and UPRIGHT				IMPORTANT AS MILLING OPERATIONS CAUSE PULSATING-LOADS	IMPORTANT BECAUSE OF INTERMITTENT CUTTING ACTION	NOT IMPORTANT IF ROTATING-MASSES ARE WELL BALANCED	STEEL CONSTRUCTION FAVOURABLE EXCEPT FOR UNIVERSAL MACHINES WITH COMPLICATED GEARING BUILT IN THE UPRIGHT
	KNEE (for KNEE TYPE MACHINES)							USUALLY TOO COMPLICATED, SO THAT STEEL CONSTRUCTION COULD NOT COMPETE ECONOMICALLY WITH CASTING
PLANING MACHINE	BED			IMPORTANT AS SHOCK LOADS OCCUR AT THE BEGINNING OF EACH CUTTING STROKE	LOADS ARE USUALLY FAIRLY UNIFORM	NOT IMPORTANT	IMPORTANT BECAUSE OF THE MOMENTUM OF REVERSING MASSES	MECHANICAL STRENGTH AND STIFFNESS CAN BE OBTAINED EASILY WITH THE NECESSARY HEAVY WEIGHT THEREFORE CASTING ADVISABLE
	UPRIGHTS						NOT IMPORTANT AS NOT AFFECTED BY REVERSING ACTION	VERY SUITABLE FOR STEEL CONSTRUCTION
	CROSS RAIL							
	TABLE						LOW WEIGHT WILL REDUCE THE MOMENTUM DURING THE REVERSING ACTION.	STEEL CONSTRUCTION WILL USUALLY NOT BE COMPETITIVE ECONOMICALLY WITH CASTING
PRESS	UPRIGHT	IMPORTANT FOR CLOSED FRAME UPRIGHTS	IMPORTANT FOR OPEN (C FRAME) UPRIGHTS	VERY IMPORTANT	NOT IMPORTANT		IMPORTANT FOR PRESSES WITH FLY WHEEL ACTION	VERY SUITABLE FOR STEEL CONSTRUCTION (ONLY IF NECESSARY - BASE CASTING SUGGESTED FOR HAMMERS, AND PRESSES WITH FLY WHEEL ACTION)
HAMMER	UPRIGHT						VERY IMPORTANT.	

FIG. 190.—Considerations relevant to the application of Fabricated Construction for various types of Machine Tools

Fig. 191). The slideways (7, 8) for the saddle are made from ·4 per cent carbon steel and flame hardened and ground. The flame hardening operation carried out after heat treatment (see page 193) did not create

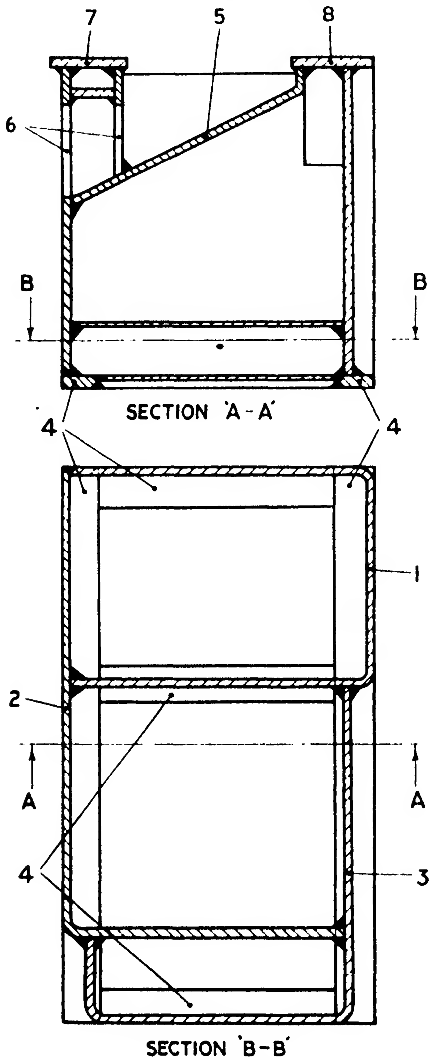


Fig. 193

new residual stresses to an extent detrimental to the accuracy of the machine.

Two openings in the side walls for the reception of the driving motors are closed by doors which are made by using the material cut out of the

body walls when forming the openings. Thus no extra material is required for the doors, and the method of production ensures a good fit of the doors in the openings.

The main structure of the upright (Fig. 194) is made from one piece of $\frac{3}{8}''$ steel plate (1) bent to shape, with zigzag ribs (2) in the portions

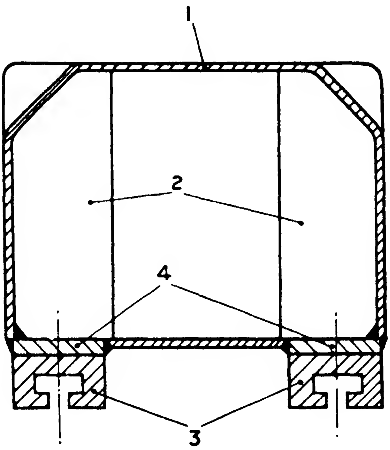


FIG. 194

carrying the highest stresses. The top and bottom are made from 1" thick plate. The vertical slideways (3) are of cast iron bolted against machining facings made from inserted flats (4). This design seems to be the most economical one in this case.

The headstock (Fig. 195 a) is divided into three parts : the box-sectioned portion carrying the spindle (Fig. 195 b), the housing for the driving gears (Fig. 195 c), and the outer skin (Fig. 195 d). These three parts are made as sub-assemblies and then finally assembled.

In most cases a universal machine tool must be arranged for quick changes of speeds and feeds and for universal movements, and must therefore be provided with rather complicated controls. On single-purpose or production machines, however, where changes of speeds and feeds are only infrequently required, the use of pick-off gears is satisfactory. The controls on such machines are comparatively simple, and the gear-boxes can be built up economically as fabricated constructions.

The feed-drive gear-box (Fig. 196) is divided into two box sections, one (1) containing the reduction gears, control mechanism, and clutches, and the other (2) containing the change wheels. The bores in the thick walls (3) and (4) house the ball-races. The thin wall (5), with a large opening (6) for access to the change wheels, provides at the same time the flange for attaching the whole box to the body. The various

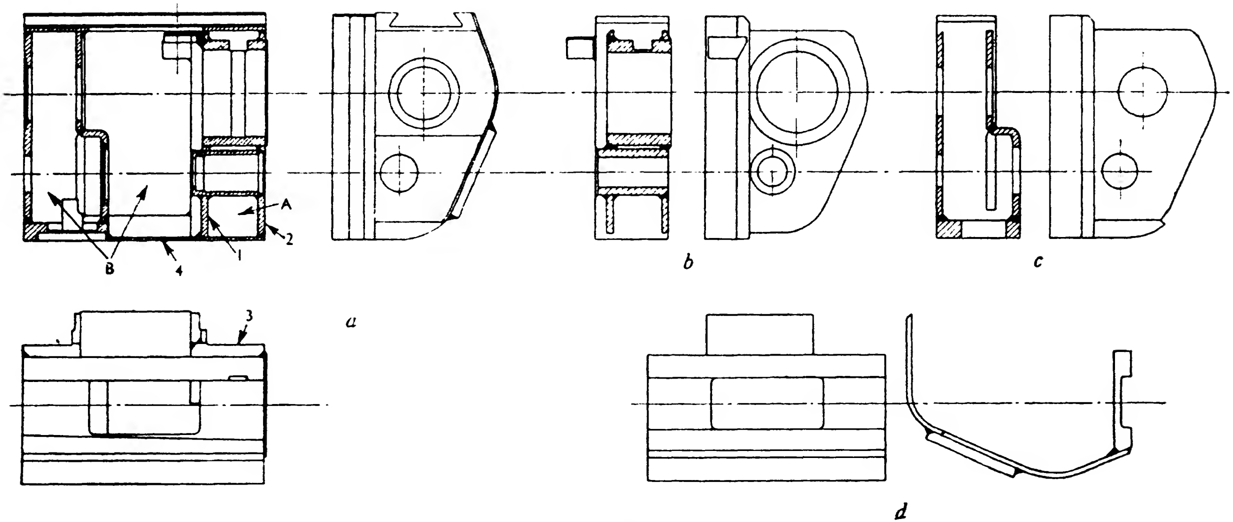


FIG. 195

members are held together by the outer skin cover plate (7), which is bent to the required shape.

An interesting example of the application of spot welding to machine-tool construction is the Discus surface grinding machine (Fig. 197). The main body of this machine, which has an effective table length of 5 feet, is built of $\frac{5}{32}$" thick material. Stiffness against distortion and freedom from vibration are obtained by the so-called " cell " construction, which is patented. This method not only

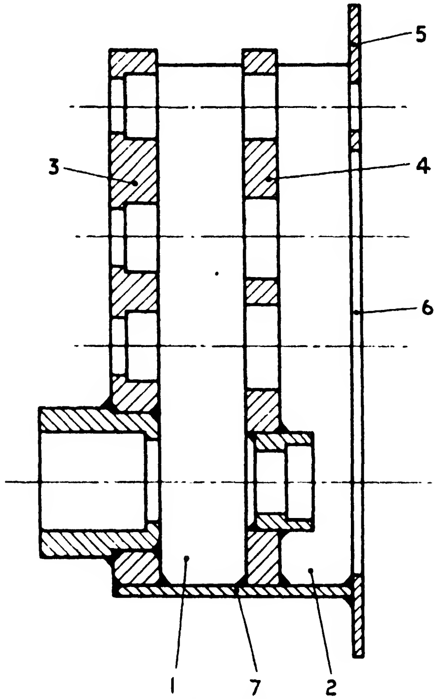


FIG. 196

eliminates big uninterrupted surfaces with their danger of drumming, and gives the necessary all-round stiffness to the body, but also makes possible the easy and economic assembly by means of spot welding. The horizontal section (Fig. 198) through the body shows the " cell " arrangement and the ingenious method of bending and connecting the several component plates into one rigid body, the spot welds being indicated by short centre lines.

From time to time a high degree of accuracy is required from a welded structure before any machining operations have been carried out. This problem arises particularly in cases of large-sized and complicated structures where inaccuracy during the initial welding opera-

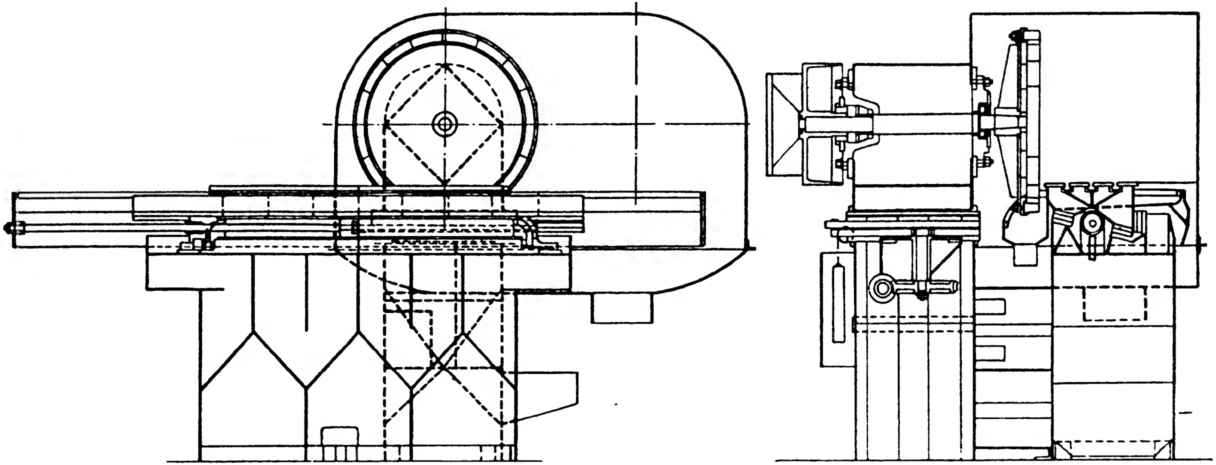


FIG. 197

tions may upset the whole assembly procedure, and where any saving in machining cost has a great effect upon the cost of the completed article.

Accuracy in welding assembly does not depend exclusively on workmanship or on procedure and general supervision in the shop; a great deal can be achieved by appropriate provisions in the design.

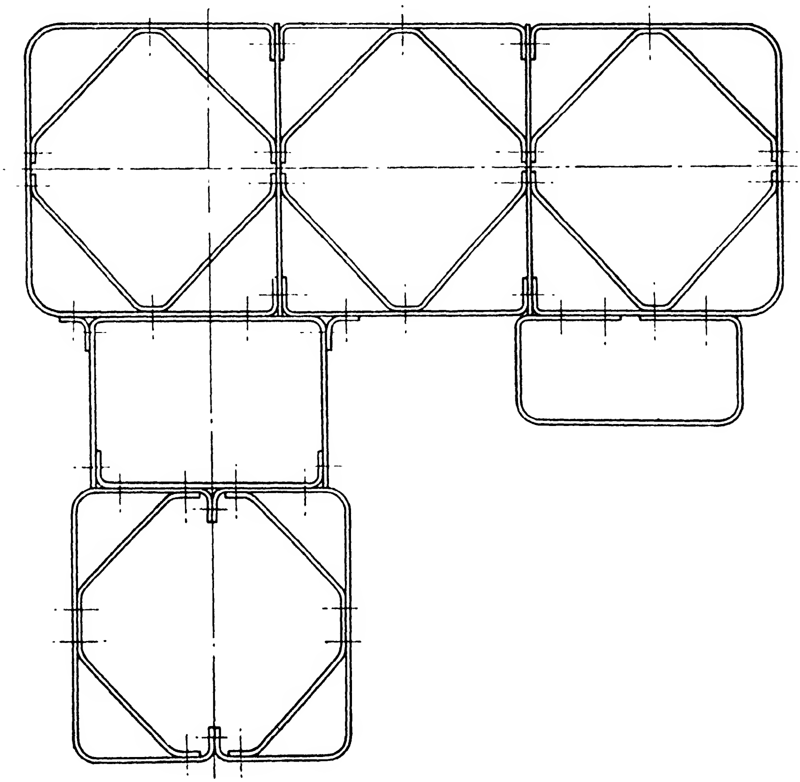


FIG. 198

An interesting example of the combination of careful design and efficient manufacture is the stator body of a 100,000 kW alternator * (Fig. 199, Plate VII). Fig. 200 (Plate VIII) shows the location of the longitudinal square bars in slots which are cut into the skeleton frame plates. Fig. 201 (Plate VIII) shows the stator body during manufacture.

* Figs. 199, 200, and 201 have been kindly provided by Metropolitan-Vickers Electrical Co., Ltd., Trafford Park, Manchester.

The skeleton frame plates are tack-welded into position before the longitudinal bars and the covering plates are welded on to the structure.

The fitting and welding procedure for this alternator stator has been brought to such perfection that no machining of the rotor tunnel is necessary after fabrication. The tolerance on the tunnel to which the fabricating shop works is $\frac{1}{32}''$ over the length and diameter.

For the fabricated steel parts of their diesel engines, Messrs. Davey, Paxman and Co. Ltd., Colchester, have developed very interesting methods of design and manufacture. The use of fabricated steel in place of cast iron for diesel engine frames, crank cases, and bed plates offers increased shock resistance, the capacity to withstand distortion, and a saving in weight ; three points which are considered particularly important.

Fig. 202 (Plate IX) shows the fabricated cylinder block for a Paxman V-type diesel engine (max. output 1125 H.P. at 750 r.p.m.) with some of the cylinder jackets in place. Each cylinder jacket consists of a top of cast steel which is welded to the lower portion formed of mild steel, the assembly being finally galvanized.

The required accuracy of assembly is obtained through the mechanical interlocking of the component plates, the patents of which are held by Mr. C. H. Stevens, C.B.E. The photograph (Fig. 203, Plate IX) taken during the early stages of fabrication of the cylinder block (Fig. 202, Plate IX) shows clearly the principles of the Stevens method. The small triangular stiffeners at the bottom of the side walls (Fig. 202, Plate IX) are actually part of the cross walls and are tenoned into the side walls. The top plates which carry the cylinder jackets are again tenoned into both the cross and side walls (Fig. 203, Plate IX), and by careful preparation of the component plates accurate assembly of the whole structure can be achieved.

Fig. 204 (Plate X) shows the bed plate for the engine (Fig. 202, Plate IX), with bearing housings cut from plate material.

The frame for a six-cylinder Paxman diesel engine (max. output 562 H.P. at 750 r.p.m.) also built to Stevens' patents is shown in Fig. 205 (Plate X).

C. Pressure-tight Welded Vessels

The necessity of avoiding manufacturing difficulties created by typical features of the welding process is very important in the case of pressure vessels, where failures might have very serious effects.

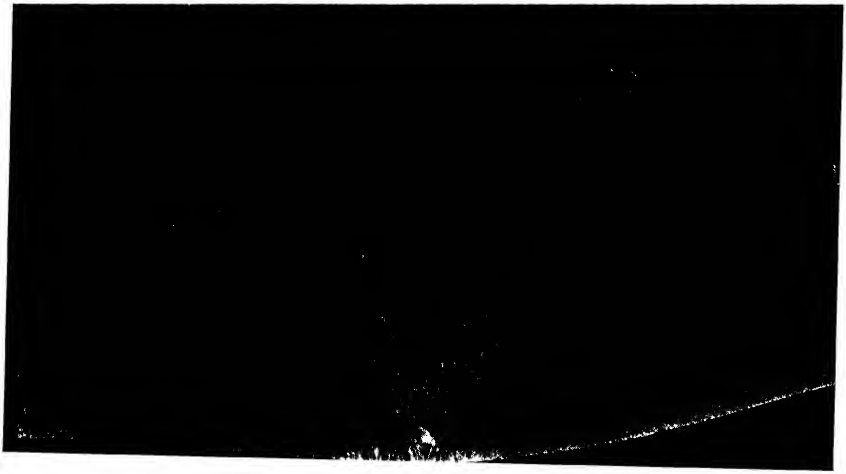
Fig. 206 * shows the shell of a cooler for a compressor plant (see Fig. 162), the details of the welds for connecting branches and flanges, as used by Continental designers, being of special interest.

* This drawing has been put at the author's disposal by H. Diederich, Esq.

Fig. 6 Fabricated Gear Case *(Cooke & Ferguson Ltd., Manchester)*

PLATE I

Fig. 27 Butt Weld Failure under Static Load



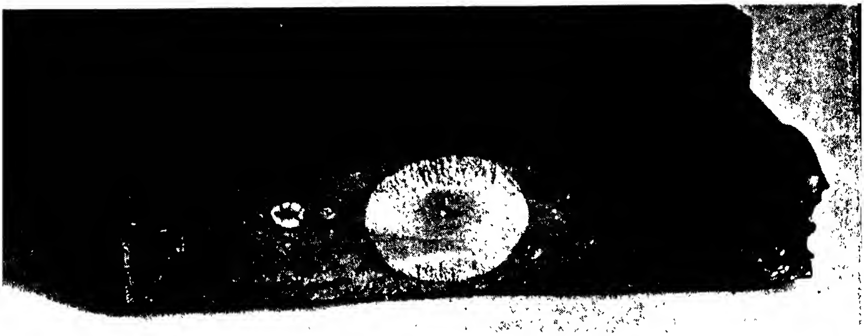
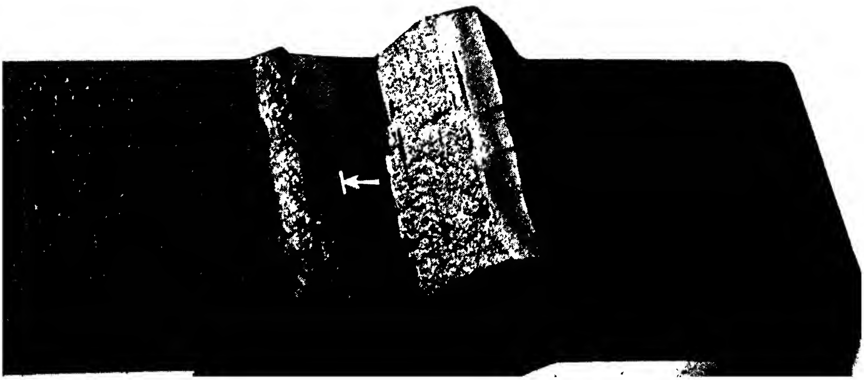


Fig. 28 Butt Weld Failure under Fatigue Load

Fig. 29 Failure of Welded T–Connection under Static Load

Fig. 30 Failure of Fillet Welded T–Connection under Fatigue Load



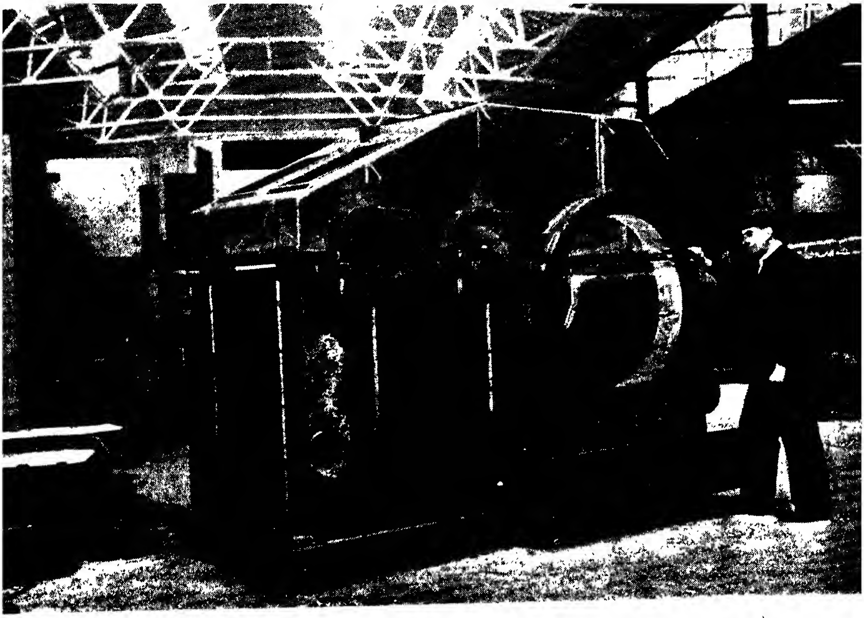


Fig. 144 Reduction Gear Case *(Cooke & Ferguson Ltd., Manchester)*

PLATE III

Fig. 176 Small Spot-Welded Bed-Plate, overall size about 21″ × 12″ × 5″
(Cooke & Ferguson Ltd., Manchester)

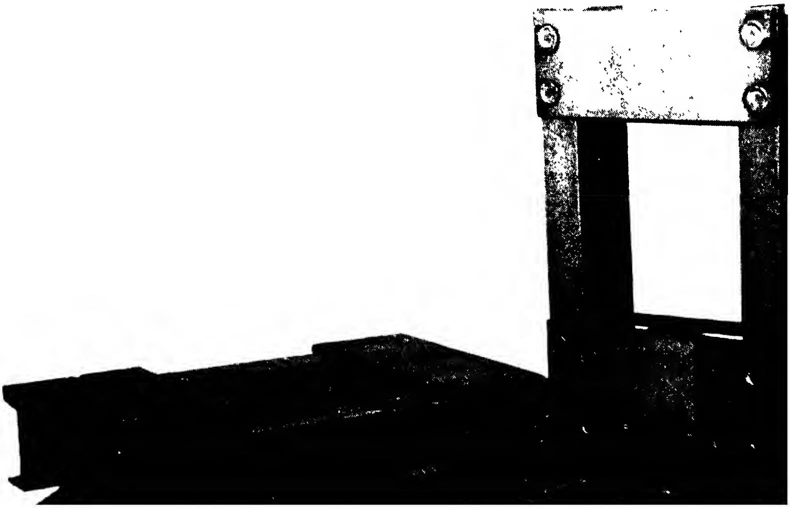
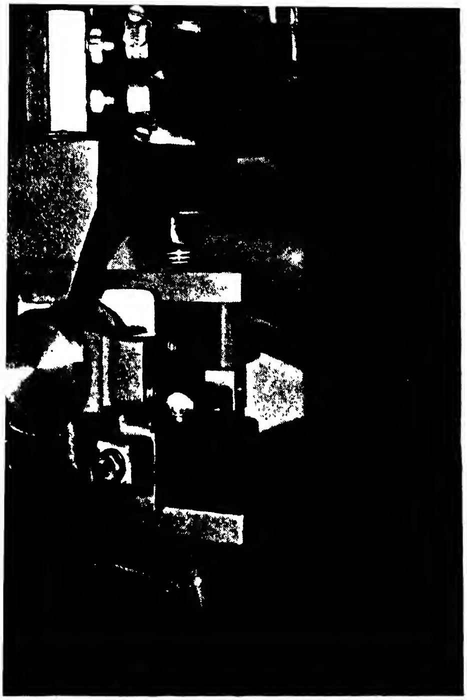
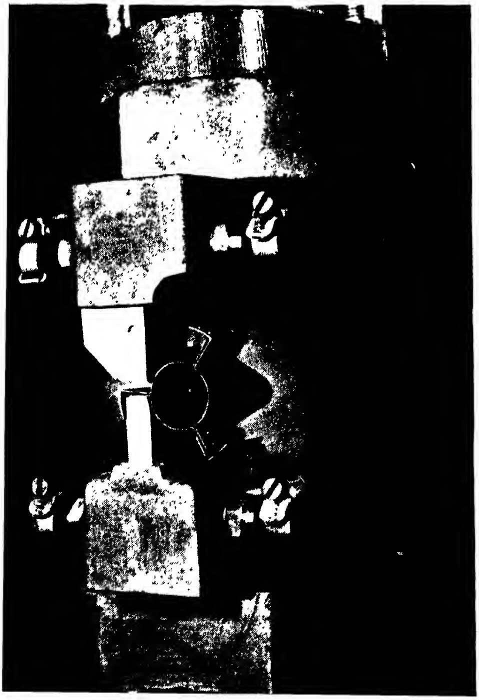


PLATE IV

Figs. 177-179 Projection-Welded Bomb Tail *(Metropolitan-Vickers Electrical Co., Ltd., Manchester)*





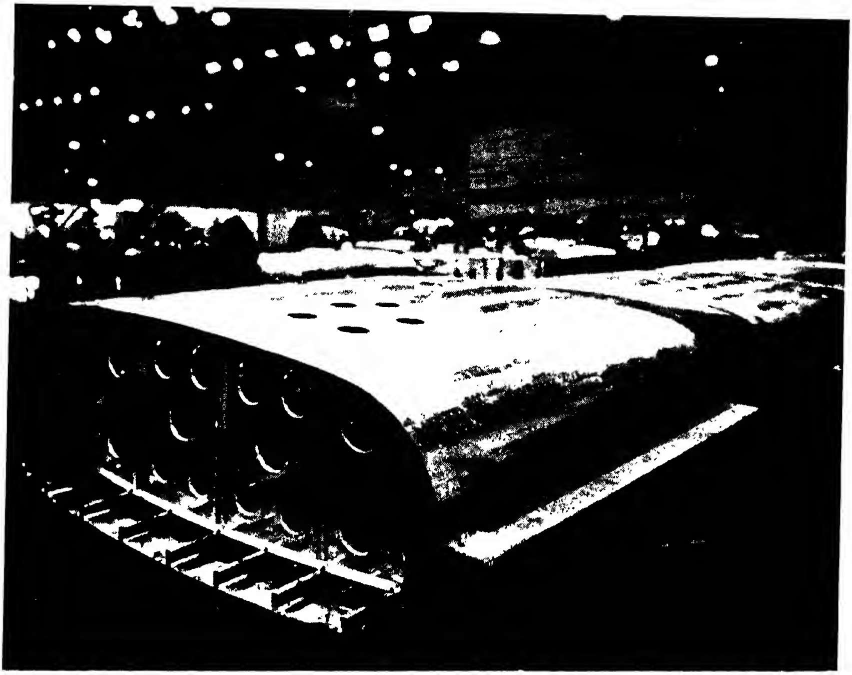
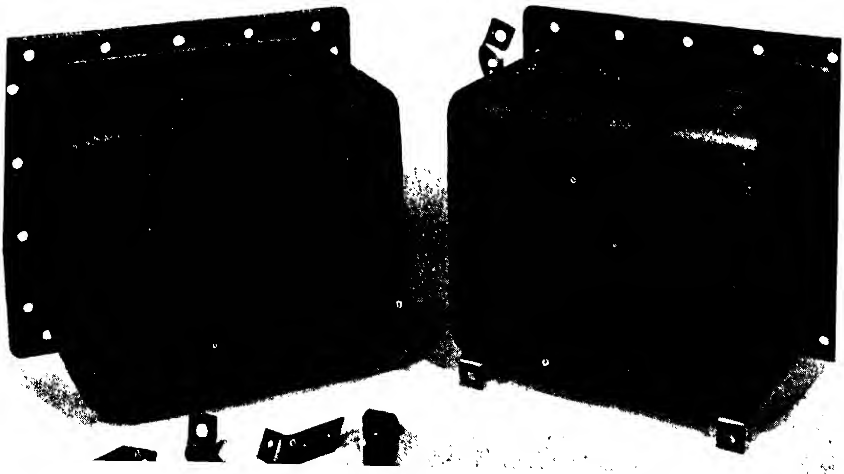


Fig. 180 Panel for "LANCASTER" Tank with Stiffeners
(Metropolitan-Vickers Electrical Co., Ltd., Manchester)

PLATE V

Fig. 181 Projection-Welding Feet to a Tank
(Metropolitan-Vickers Electrical Co., Ltd., Manchester)



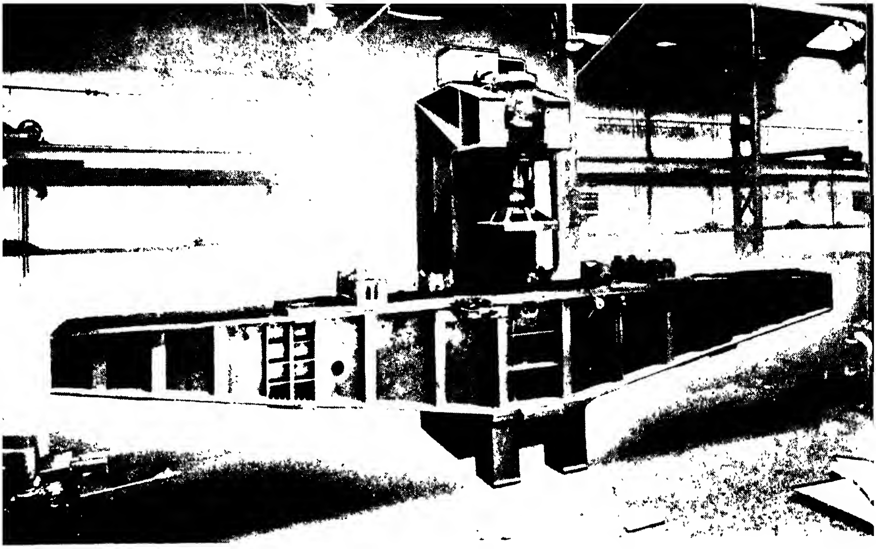
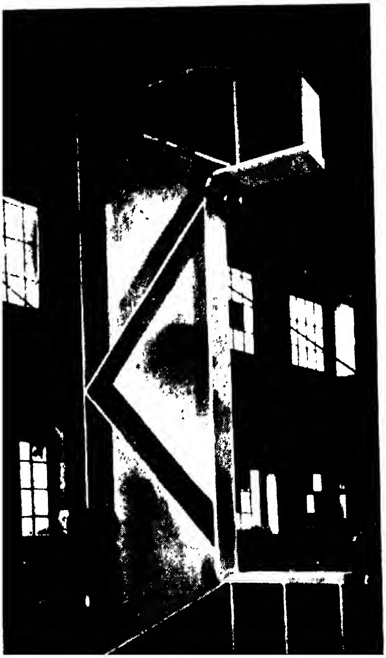


Fig. 182 300 Ton Straightening Press *(Hydraulic Engineering Co., Ltd., Chester)*

PLATE VI

Fig. 184 Frame for Straightening Press (Fig. 182)

Fig. 187 750 Ton Press at the Works of the Cambrian Wagon Company, South Wales *(Fielding & Platt Ltd., Gloucester)*



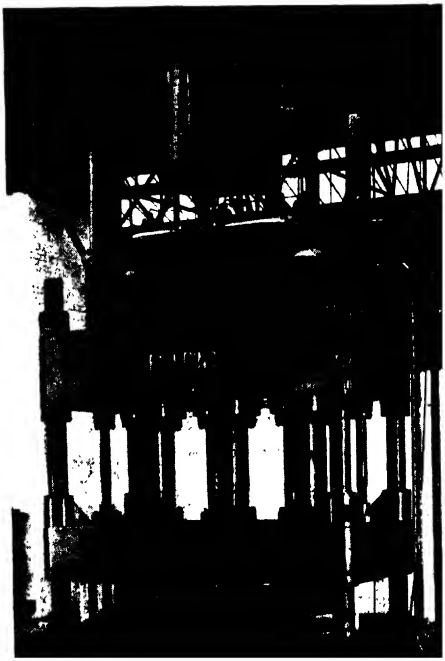
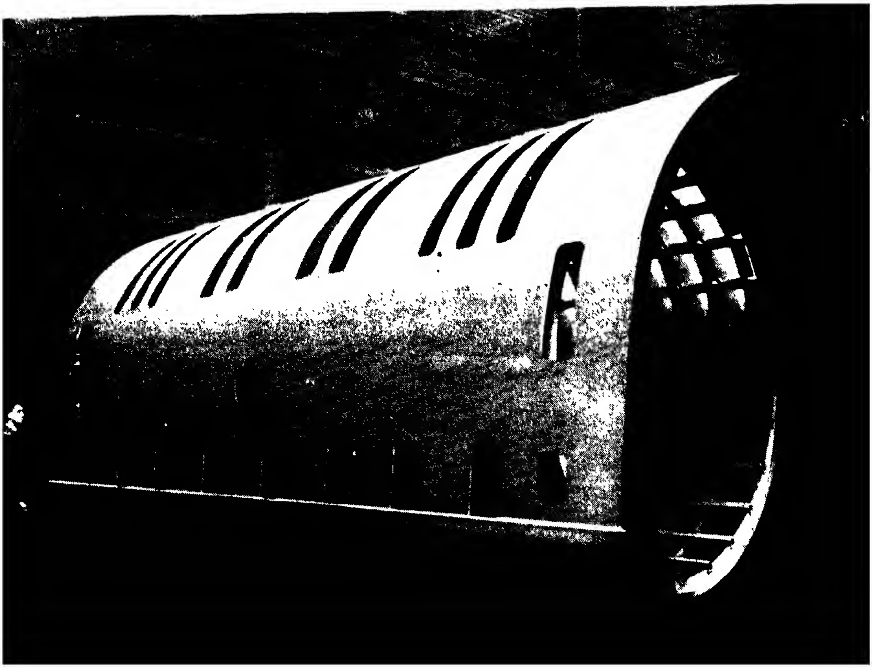


Fig. 191 Production Milling Machine
(Cooke and Ferguson Ltd., Manchester)

Fig. 199 100,000 KW Alternator Body *(Metropolitan-Vickers Electrical Co., Ltd., Manchester)*



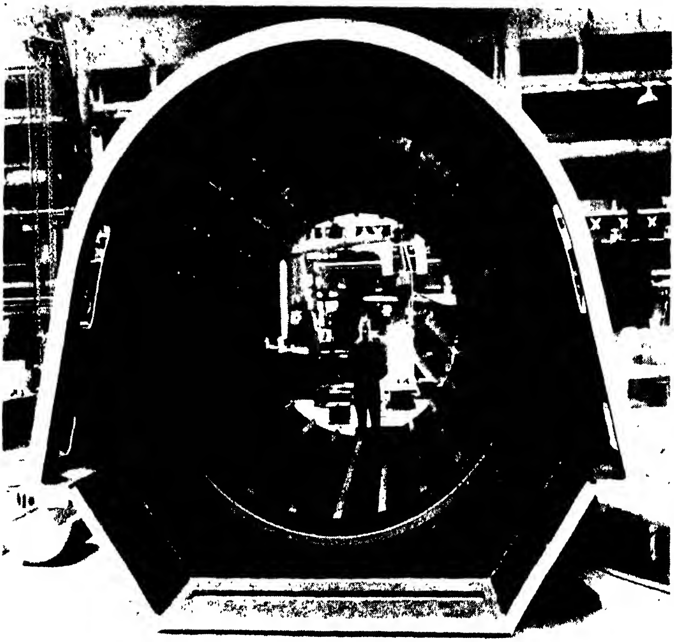
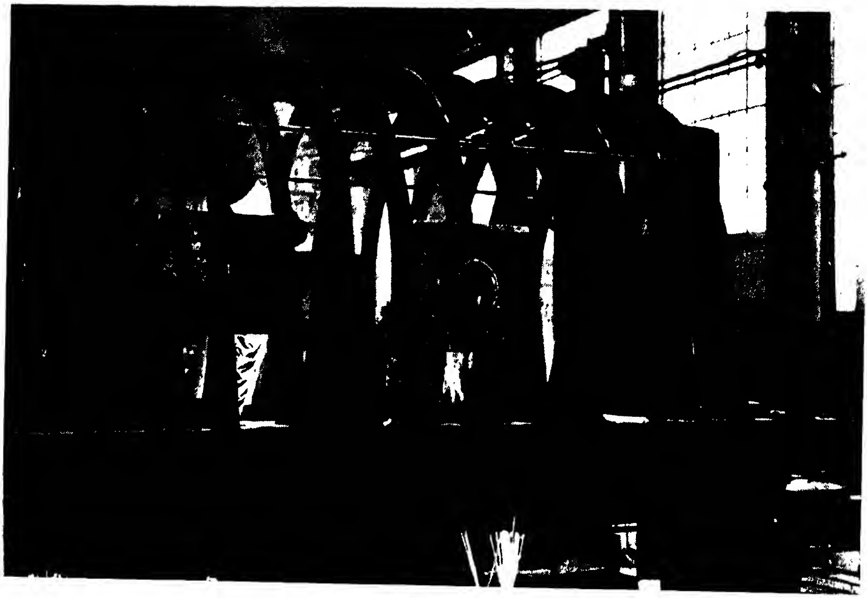
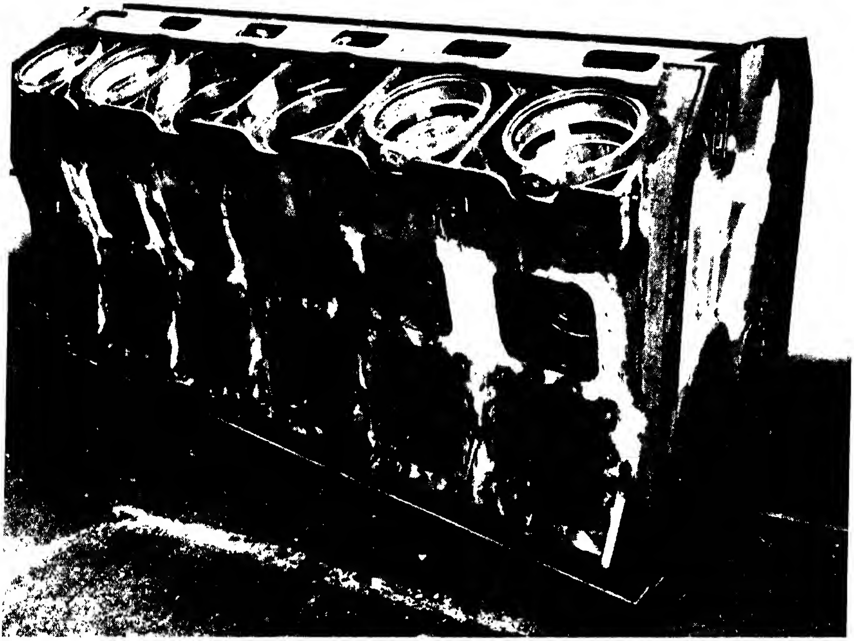


Fig. 200 Stator Body of Alternator (Fig. 199)

PLATE VIII

Fig. 201 Stator Body (Fig. 200) during Manufacture



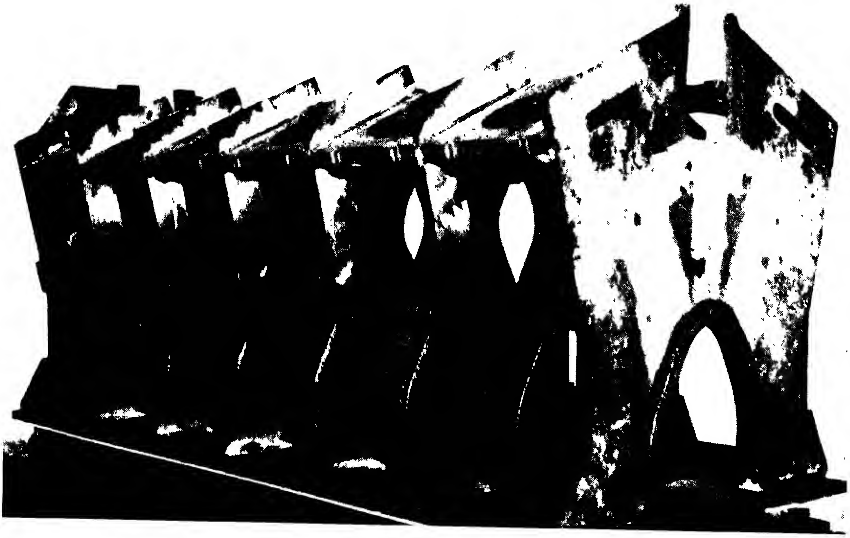


. 202 Fabricated Cylinder Block for Paxman V–type Diesel Engine, fully machined and fitted with some cylinder jackets, built to patents of C. H. Stevens

(Davey, Paxman & Co., Ltd., Colchester)

PLATE IX

Fig. 203 Cylinder Block (Fig. 202) in early Stage of Fabrication



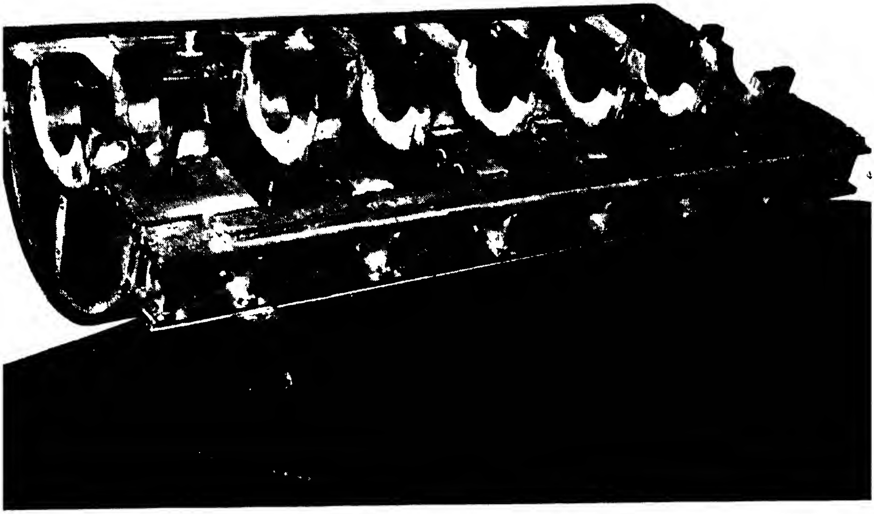


Fig. 204 Bed Plate for Cylinder Block (Fig. 202) ready for Machining

PLATE X

Fig. 205 Paxman Steel Frame for Vertical 6–Cylinder Diesel Engine, built to patents of C. H. Stevens *(Davey, Paxman and Co., Ltd., Colchester)*

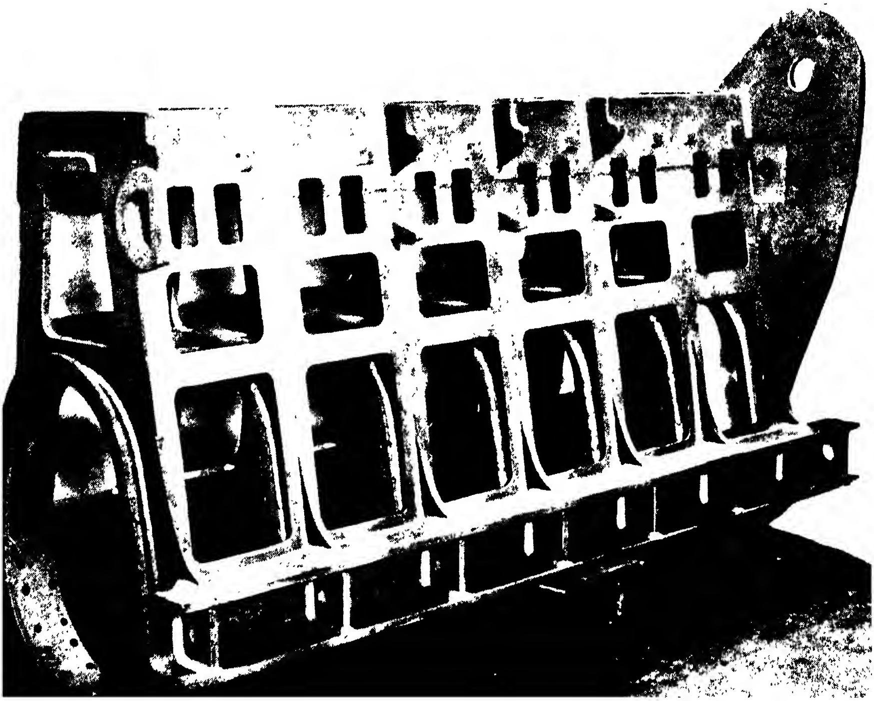


PLATE XI

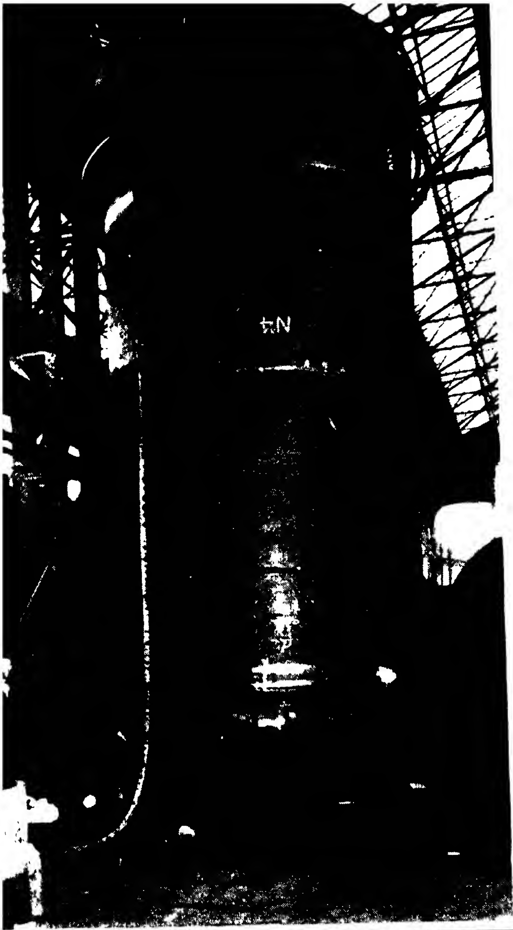


Fig. 207 Mild Steel Evaporator manufactured by G. A. Harvey and Co., Ltd., London (*Tate and Lyle Ltd.*)

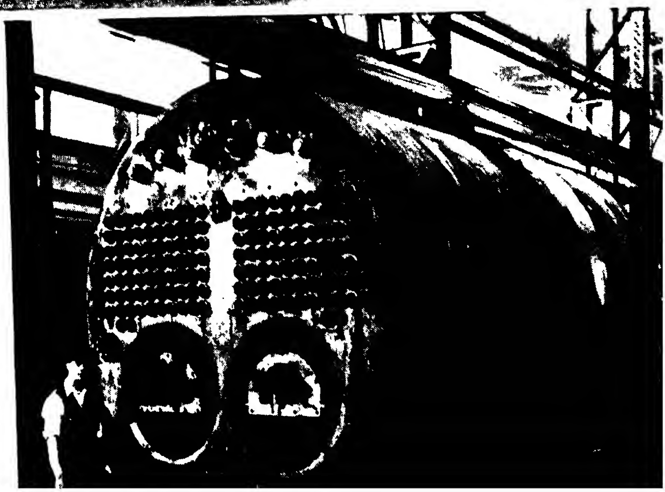


Fig. 209 Mild Steel All-welded Economic Boiler, Dryback Type (*G. A. Harvey and Co., Ltd., London*)

PLATE XII

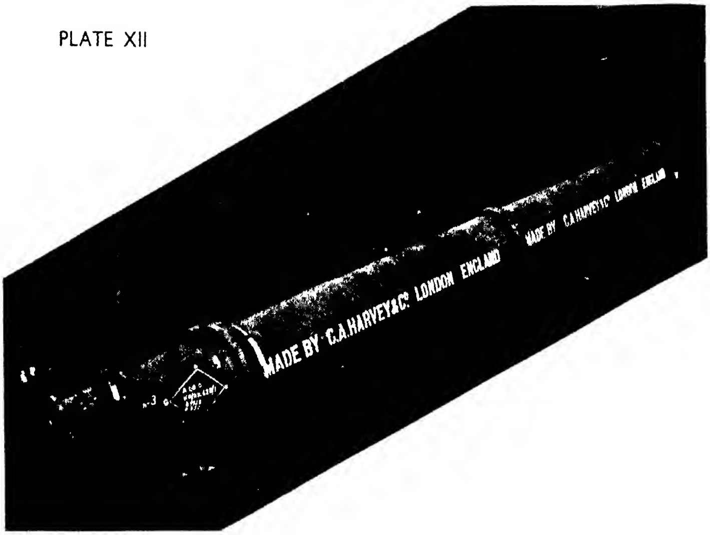
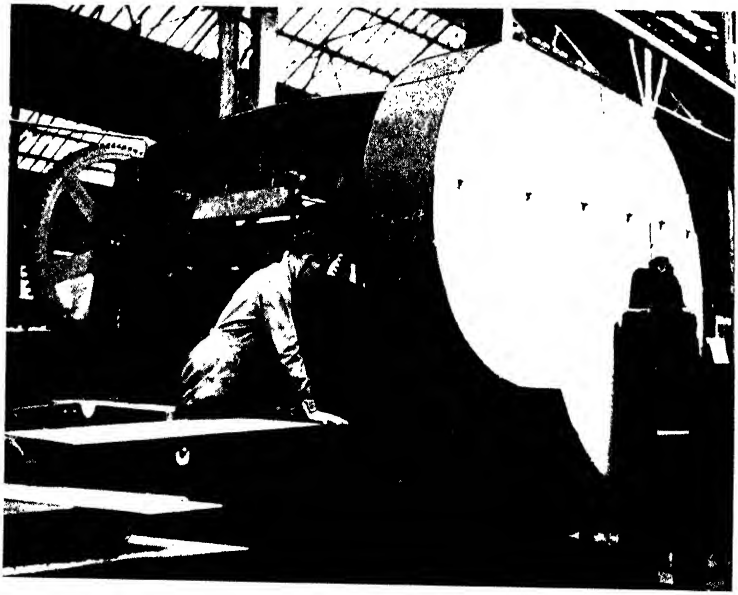


Fig. 210 Mild Steel Wellhead Separator (Anglo-Iranian Oil Co.), manufactured by G. A. Harvey and Co., Ltd., London

Fig. 212 Guillotine at the Works of Cooke and Ferguson Ltd., Manchester



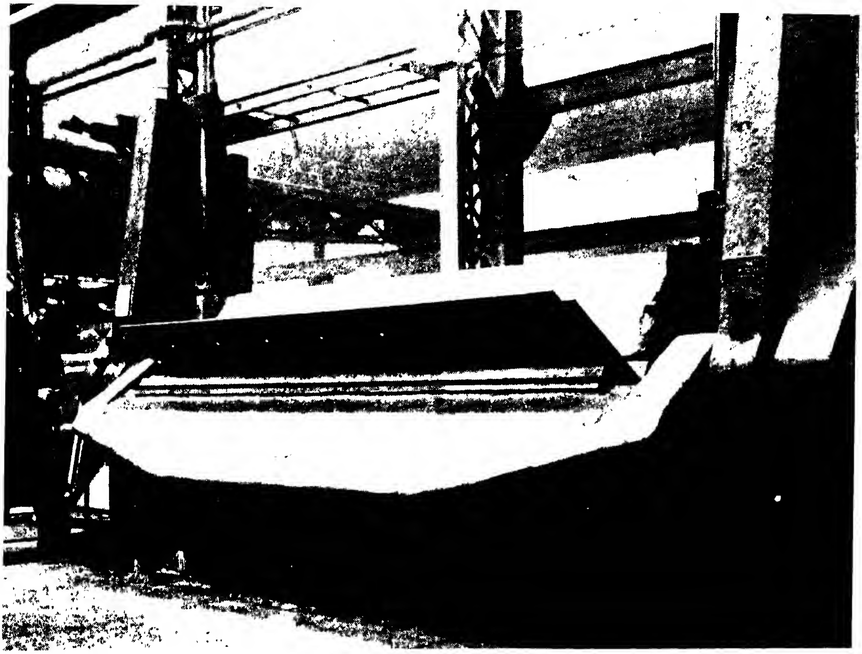


Fig. 213 Plate Bending Machine at the Works of Cooke and Ferguson, Ltd., Manchester

PLATE XIII

Fig. 214 Plate Rolling Machine at the Works of Cooke and Ferguson, Ltd., Manchester

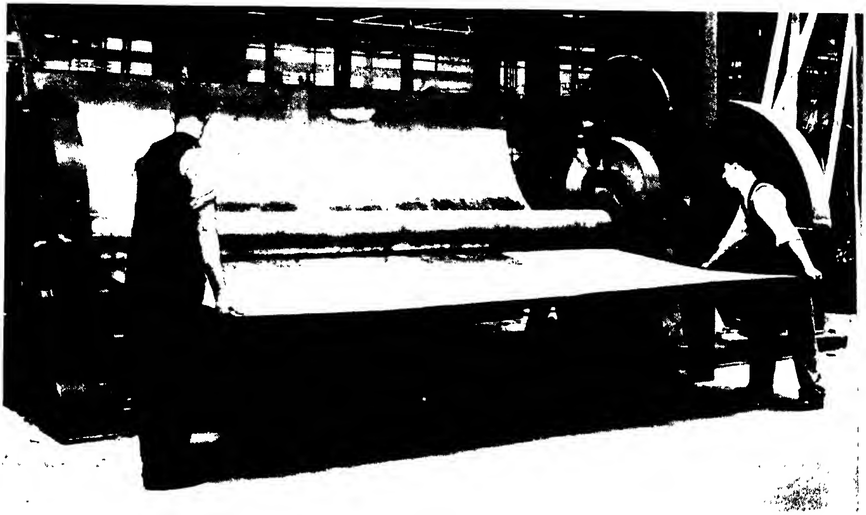


Fig. 215 Light Plate Rolling Machine at the Works of Cooke and Ferguson Ltd., Manchester

PLATE XIV

Fig. 216 Press Brake at the Works of Cooke and Ferguson Ltd., Manchester

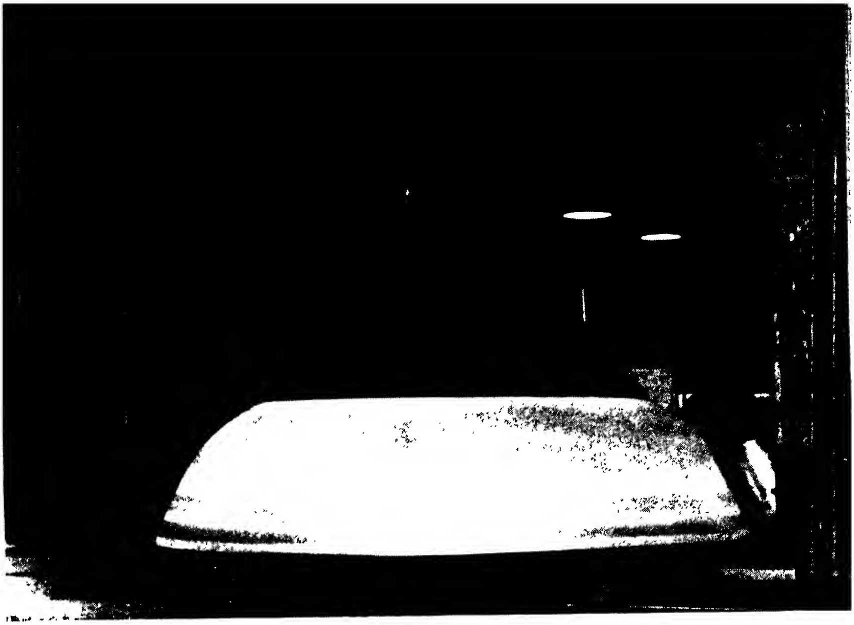


Fig. 217 Birdsboro Spinning Press, Spinning 3″ thick plate of Manganese-Molybdenum Steel to a 9 ft. Diameter Dishing, at the Works of G. A. Harvey and Co., Ltd., London

PLATE XV

Fig. 218 Spot Welding Machine Welding Stiffeners to "LANCASTER" Tank Panels (see Fig. 180) *(Metropolitan-Vickers Electrical Co., Ltd., Manchester)*

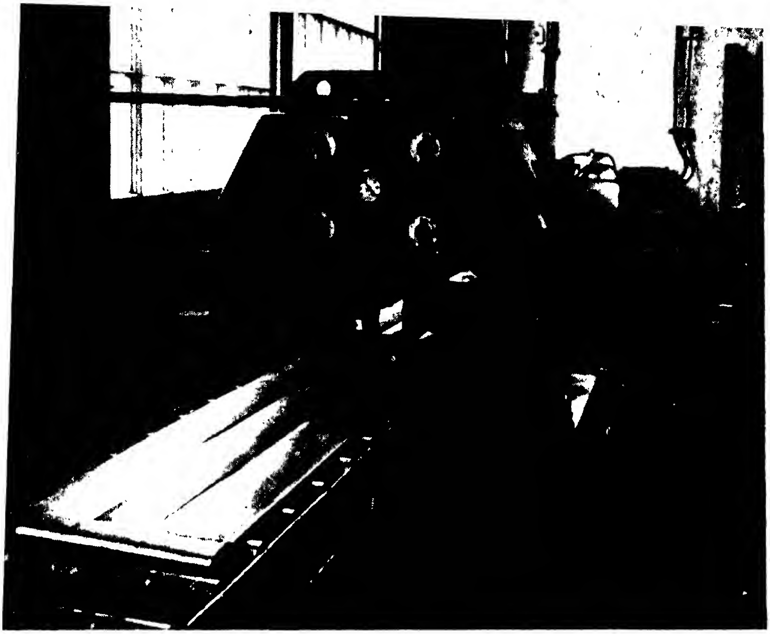


Fig. 219 Special Double Roll Seam Welding Machine Welding Radiator Parts
(Metropolitan-Vickers Electrical Co., Ltd., Manchester)

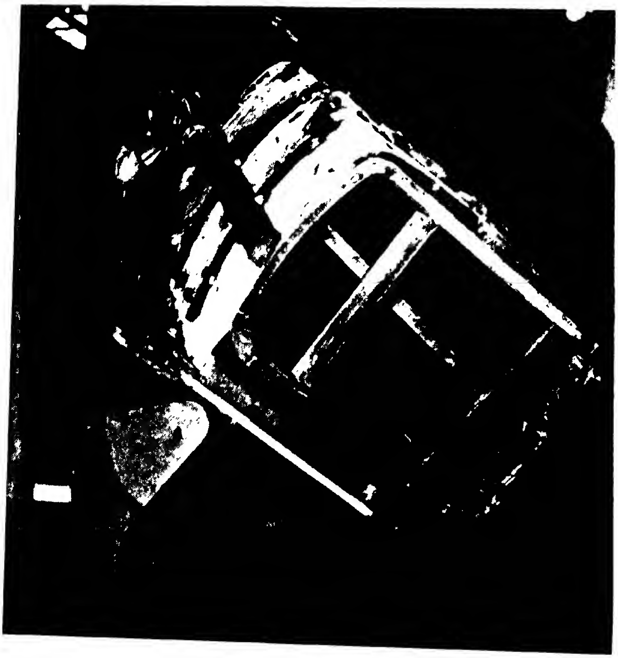


PLATE XVI

Fig. 220 Welding Manipulator (Metropolitan-Vickers) carrying a Condenser Shell, at the Works of Cooke and Ferguson Ltd., Manchester

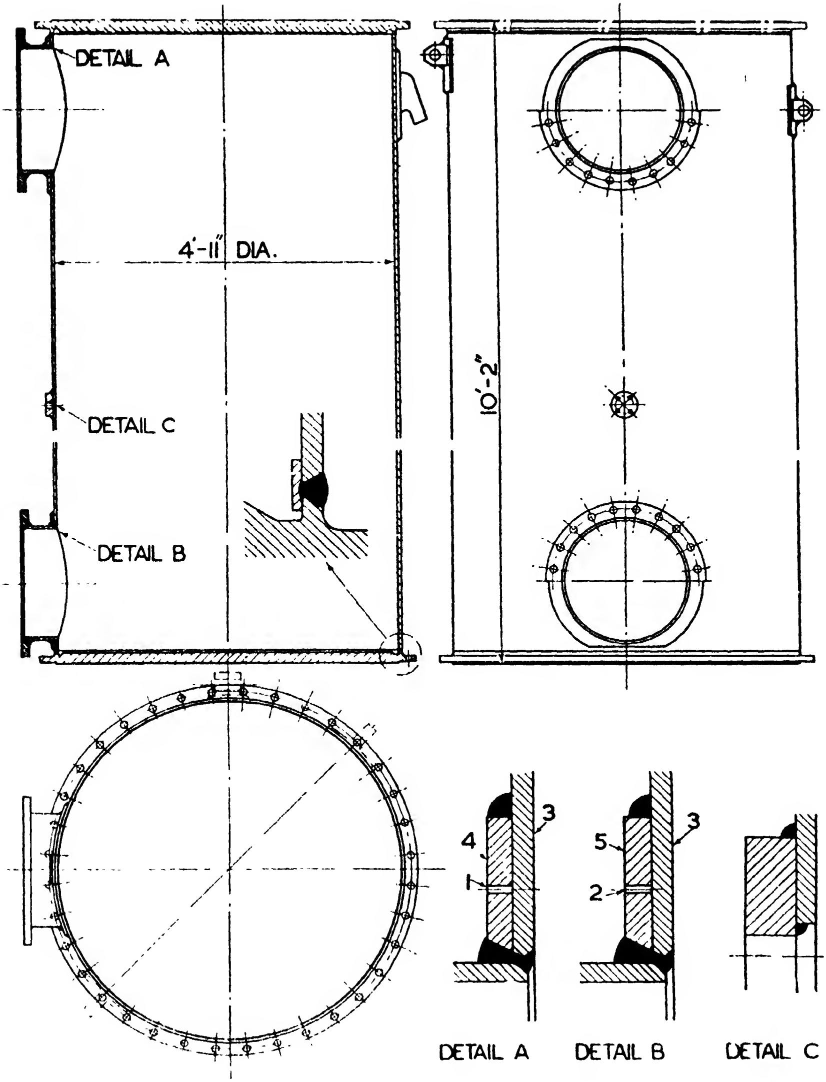


FIG. 206.—Shell of a Cooler for Compressor Plant (Continental design)

The small air holes (1, 2) in weld details (A, B) are provided in order to permit the escape of any gases which may be formed during the welding process and which might otherwise build up a pressure between the shell plate (3) and the reinforcing plates (4, 5).

Figs. 207 to 210 (Plates XI and XII) * show some typical examples of welded pressure vessels made to Lloyd's class 1 specification.

The evaporator (Fig. 207, Plate XI), constructed by Messrs. G. A. Harvey and Co. Ltd. to the order of Tate and Lyle Ltd., is 11 ft. in diameter.† The bottom shell is subject to a pressure of 250 lb./sq. inch and is $1\frac{15}{32}''$ thick ; the top shell works at 190 lb./sq. inch pressure and has a thickness of $1\frac{1}{8}''$. The dished ends are made from $1\frac{3}{8}''$ thick plate and have a dish radius of 8 ft. and a 12″ corner radius (Fig. 208).

Fig. 209 (Plate XI) shows the Economic Dryback Boiler (G. A. Harvey & Co. Ltd.), which is 15 ft. long and 9′ 9″ diameter.‡ The wall thickness of shell and end plates is $\frac{15}{16}''$ and the test pressure 320 lb./sq. inch.

A vessel designed and built to withstand very high pressures is the wellhead separator (Fig. 210, Plate XII) built by the same company for the Anglo-Iranian Oil Company. The diameter of the main section is 4′ 6″ and the over-all length 75 ft. The wall thickness of the shell is $1\frac{1}{4}''$ and the wall thickness of the cones $1\frac{5}{16}''$ and $\frac{3}{4}''$ respectively. The working pressure is 600 lb./sq. inch and the test pressure 950 lb./sq. inch.

The calculations for pressure vessels follow exactly the lines laid down in the appropriate specifications (see page 143). The air receiver (Fig. 211) is designed for a working pressure (WP) of 300 lb./sq. inch (600 lb./sq. inch test pressure) in accordance with B.S.S. 487. The values used in the calculation are :

$C_1 = 17{\cdot}5$ (longitudinal seam welded from both sides).
$C_4 = 17{\cdot}5$ (receiver under 6 ft. diameter).
$D = 20$ inches.
$R = 20$ inches.
$S = 26$ tons/sq. inch (material 26/30 tons/sq. inch, see Fig. 5).

For the calculation of shell thickness :

$$T = \frac{WP \times D}{32C_1 \times S} + {\cdot}06 \quad \text{(equation 75)}$$
$$= \frac{300 \times 20}{32 \times 17{\cdot}5 \times 26} + {\cdot}06$$
$$= {\cdot}473''.$$

Shell thickness selected :

$$T = \tfrac{1}{2}''.$$

* These illustrations have been kindly provided by the manufacturers of the pressure vessels, Messrs. G. A. Harvey and Co. (London) Ltd., London, S.E.7.

† See description in *The Engineer*, September 18th, 1936.

‡ See paper by Fergusson and Burford in *Arc Welding in Design, Manufacture and Construction*, Lincoln, 1939, page 828.

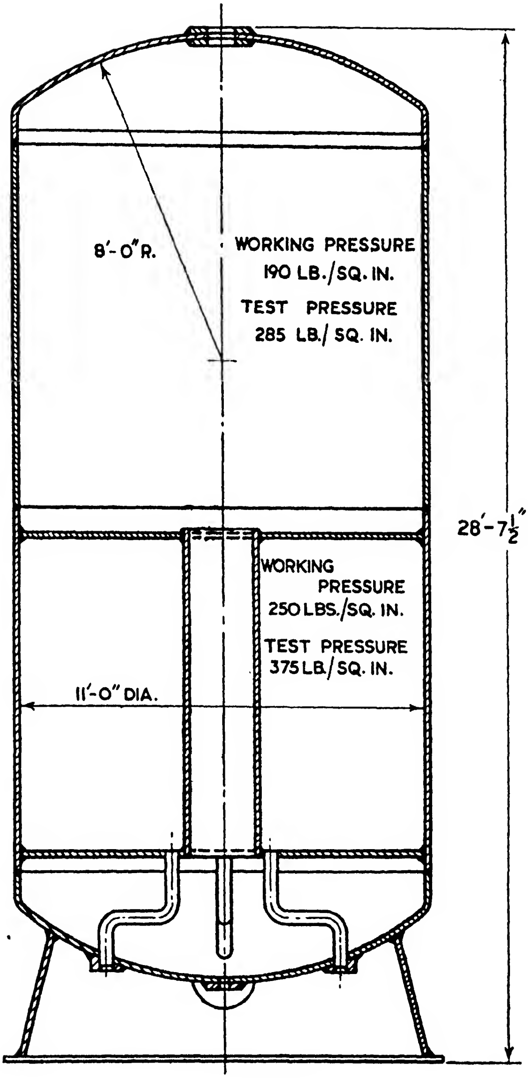
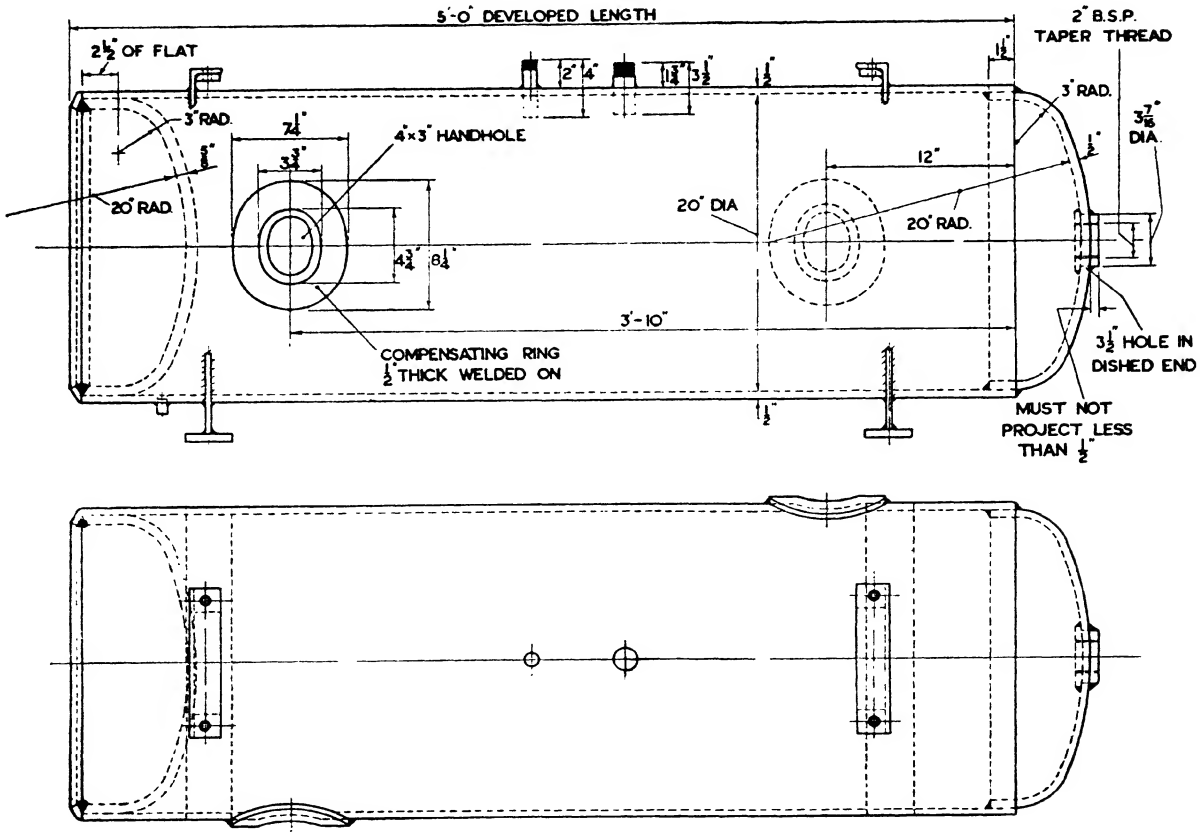
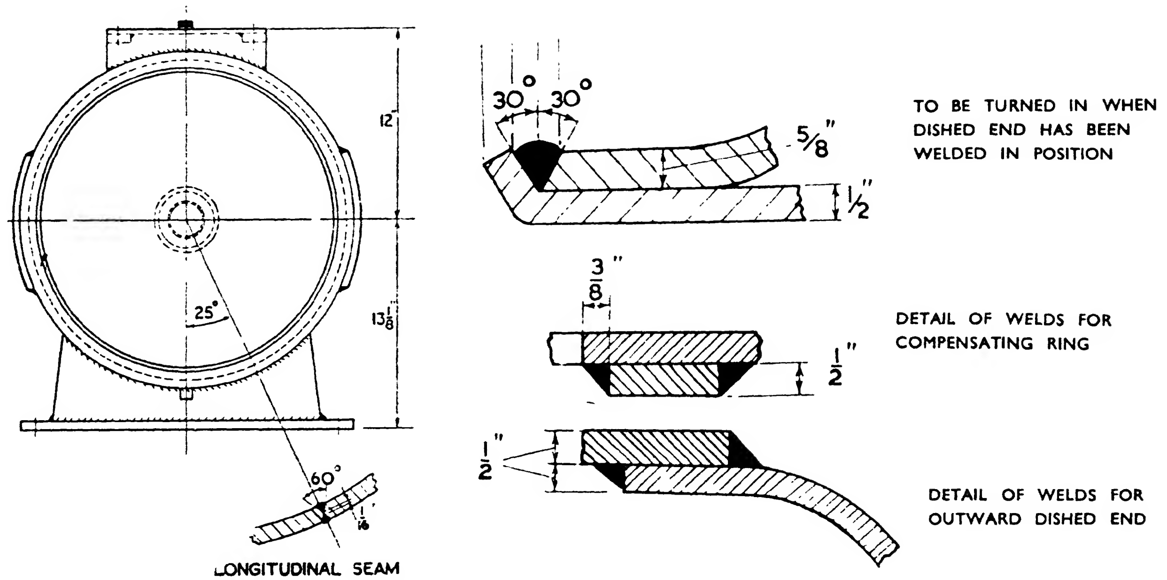

8'-0" R.
WORKING PRESSURE
190 LB./SQ. IN.
TEST PRESSURE
285 LB./SQ. IN.
28'-7½"
WORKING
PRESSURE
250 LBS./SQ. IN.
TEST PRESSURE
375 LB./SQ. IN.
11'-0" DIA.

Fig. 208


5'-0" DEVELOPED LENGTH
2" B.S.P. TAPER THREAD
2½" OF FLAT
3" RAD.
5/8"
20" RAD.
7¼"
3¾"
4"×3" HANDHOLE
2"
4"
1¾"
3½"
½"
1½"
3" RAD.
3 7/16" DIA.
12"
20" DIA
20" RAD.
4¾"
8¼"
3'-10"
COMPENSATING RING ½" THICK WELDED ON
3½" HOLE IN DISHED END
MUST NOT PROJECT LESS THAN ½"



NOTE.—*All fillet welds ¼" if not stated otherwies.*

FIG. 211.—Welded Air Receiver (Cooke & Ferguson, Ltd., Manchester)

For the calculation of plate thickness for outwardly dished end :

$$T = \frac{WP \times R}{32C_4 \times S} \quad \text{(equation 82)}$$

$$= \frac{300 \times 20}{32 \times 17{\cdot}5 \times 26}$$

$$= {\cdot}413''.$$

It is general practice to make the thickness of the outwardly dished end not less than that of the shell plate, therefore

$$T = \tfrac{1}{2}''.$$

For the calculation of plate thickness for inwardly dished end :

$$T = \frac{WP \times R}{384S} \quad \text{(equation 83)}$$

$$= \frac{300 \times 20}{384 \times 26}$$

$$= {\cdot}6''.$$

Thickness selected :

$$T = \tfrac{5}{8}''.$$

According to B.S.S. 487 a compensating ring has to be provided round hand-holes (or manholes) in such a way that " the cross-sectional area of the compensating ring plus the cross-sectional area of the manhole frame (if any) within two inches of the outside of the shell be not less than 80 per cent of the cross-sectional area of the shell plate cut-out ". Calling b the width of the $\frac{1}{2}''$ thick compensating ring on either side of the hand-hole, we have

$$2 \times {\cdot}5 \times b = {\cdot}8 \times {\cdot}5 \times 4,$$

$$b = 1{\cdot}6'',$$

and the size selected is $\quad b = 1\tfrac{3}{4}''.$

In order to be on the safe side, the longer axis of the hand-hole (4") has been considered in the calculation, although the highest stressing condition occurs in the section parallel to the axis of the shell.

Chapter 6

DRAWING PRACTICE TO ENSURE EFFICIENT FABRICATION BY WELDING

To obtain efficiency in the application of welding methods to construction it is necessary to stop the old practice, still used in many welding shops, by which the foreman is the technical authority who will " make the job do " up to the point of rectifying, at his own discretion, any faults which may be found when the job is completed.

The danger of such a practice is particularly evident in the welding shop, because of the great temptation to fill gaps with weld metal, to cut out faulty parts and re-weld corrected ones in position, to cover up faulty weld preparations, etc.

To eliminate this danger a definite specification is required to form the technical basis of the whole production process. This specification is the drawing, and the fundamental responsibility for efficient production rests, therefore, in the first place with the Design and Drawing Office.

To be able to issue to the works a drawing which fulfils the requirements of such a responsibility, the Design Department must not only know the technical requirements of the job itself, the loads and stresses expected under working conditions, etc., but also the quality of workmanship obtainable in the shop, the materials in stock or available from material suppliers, the manufacturing processes at the disposal of the shop, and the available plant and its special characteristics.

It is not possible to produce welded fabrication economically by just installing an arc welding set.

For preparing the plate material the efficient fabricating shop must have at its disposal flame cutting machines, guillotines (Fig. 212, Plate XII), bending machines (Fig. 213, Plate XIII), plate rolling machines (Figs. 214, Plate XIII, 215, Plate XIV), press brakes (Fig. 216, Plate XIV), hydraulic presses, and heavy spinning machines for dishing shell ends (Fig. 217, Plate XV).*

In the case of pressure vessels with long continuous seams, automatic welding machines are economically justified.

Resistance welding machines (Figs. 218, Plate XV, and 219, Plate XVI) † should be available, if required on work for which this method is favourable.

To handle the components, especially in the arc welding shop where down-hand welding gives the best-quality welds, welding positioners and manipulators (Fig. 220, Plate XVI) should be used.‡

The Drawing Office must know the capacities and limitations of this plant and the accuracy obtainable on the various machines, in order to be able to judge what can and should be expected from the shop, and to arrange the designs accordingly.

Armed with this knowledge, the Drawing Office must issue on the drawings the instructions which, in addition to forming the specification of the article as finally required, contain all the information necessary to ensure satisfactory manufacture. The drawing must guide, in a

* This illustration has been kindly provided by Messrs. G. A. Harvey and Co. Ltd., London.

† These illustrations have been kindly provided by Messrs. Metropolitan-Vickers Electrical Co. Ltd., Trafford Park, Manchester.

‡ See *Welding Memorandum*, No. 3, Advisory Service on Welding, Ministry of Supply, London. See also R. Drucker, Manipulators for Arc Welding with special reference to a New Design of a Heavy All-purpose Faceplate Type of 10 tons rated Capacity, Institute of Welding, Sir William Larke Prize Paper, 1944.

foolproof way, the Planning Department, the Tooling Department, the Shop Supervisors, and the Inspection Department in the carrying out of their respective work.

The work of the Drawing Office, therefore, covers the following matters :

(*a*) The development of the design itself, including the calculations of stress, size and shape of components, sizes and types of welds, plate preparations, etc.
(*b*) The estimation of the most efficient and economical way of cutting the material to the required shapes.
(*c*) The establishment of the required assembly and welding procedure.
(*d*) The specification of the heat treatment, if required.
(*e*) The laying-out of an Inspection specification, if required.

The actual design work has been dealt with in the previous chapters ; the additional work which is so vital for obtaining high efficiency will now be considered.

When designing component parts, especially plate parts of a welded structure, the efficient exploitation of standard plate sizes should be kept in mind. Plate parts, sections, forgings, castings, and all the components of the welded structure should be detailed in the same way as is customary for the machined items of general fitting and assembly jobs.

Numerous advantages can be obtained by determining on the drawing board the best possible arrangement of the components to be cut from plate material on available stock plates, instead of leaving the lay-out to the discretion of the platers in the shop. Apart from the fact that it is easier to try out various combinations on the drawing board than on the actual plate, it is often possible to save a considerable amount of material by slight modifications in design. If repeat orders are to be executed, the drawing becomes a permanent record, to be used as required, to avoid the plater having to start all over again. Even if—in a most favourable case—the same plater as before is allocated to the job, he will not have such a good memory as to remember every detail of his previous lay-out. Furthermore, the plate lay-out is the only means on which an accurate estimate of the required material can be based.

In the case of quantity production, flame cutting templates may be provided, and a drawing of the lay-out will in this case be required by the Tool Department.

To reduce the material expenses, only a small variety of plate thicknesses should be used, and the detail parts should be arranged on the stock plates in such a way as to reduce the amount of remaining scrap material to a minimum.

Due consideration should be given—especially in the case of thinner plates—to the possibility of guillotine cutting, instead of flame cutting, which is a slower process.

As a typical example Fig. 221 shows the drawing of a fabricated reduction gear-box.* It should be noted that ribbing, etc., has been omitted on this drawing, in order to make it more schematical and less complicated, as only the principle of the method is to be shown. The size of the motor base plate (2) is small enough to be obtained from the material cut-out of the base plate (1), and the plate components have been arranged (Fig. 222) in such a way as to necessitate only one flame cut on the thinner plate.

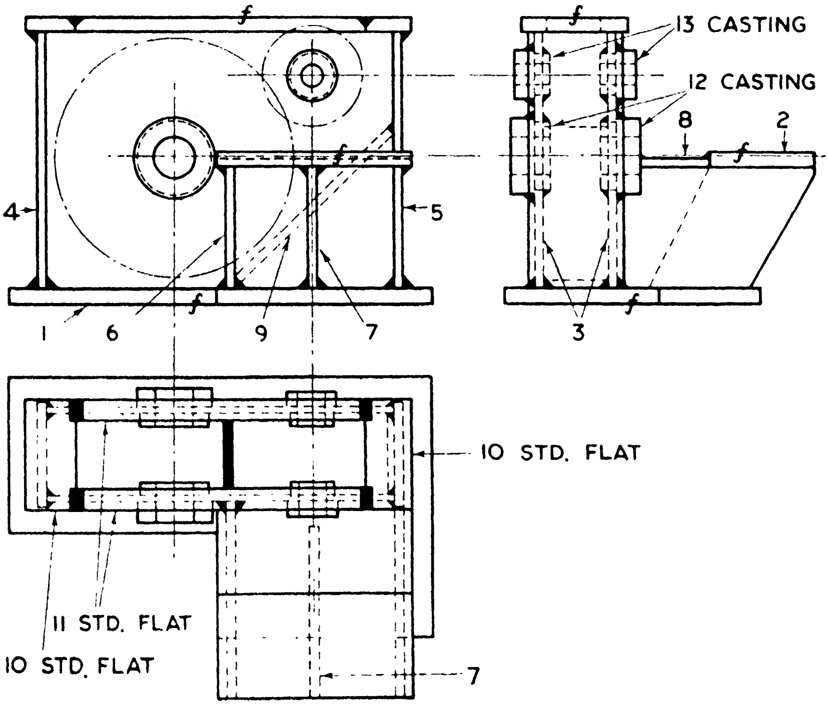


FIG. 221

In addition to being a more expensive operation in itself, flame cutting has the further disadvantage of making the plate material less resistant to fatigue stresses. This is due to the fine serrations left on the flame-cut edge by the cutting process and not, as is often thought, to a metallurgical change in the zone affected by the heat of cutting.† Smoothing the surfaces of the flame-cut edges by means of hand grinding is usually sufficient to eliminate this weakness. This point does not

* F. Koenigsberger, "The Application of Fabricated Construction to Machine Design", *Proceedings of the Institution of Mechanical Engineers*, December 1945, page 245.

† See O. Graf, "Über Dauerzugversuche und Dauerbiegeversuche an Stahlstäben mit brenngeschnittenen Flächen" ("Fatigue Tensile and Bending Tests on Steel Bars with Flame-cut Edges"), *Forschungsarbeiten auf dem Gebiete des Schweissens und Schneidens*, Marhold, Halle, 1936.

arise in the case of flame-cut weld preparations, as the welding operation melts the flame-cut edge and thus automatically eliminates the notches.

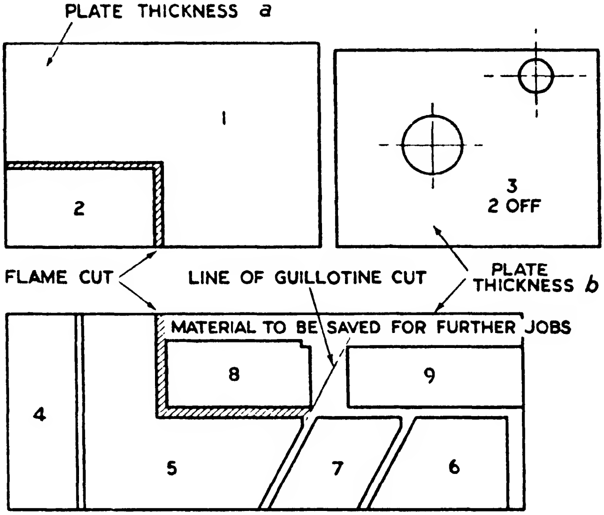


FIG. 222

Weld preparation by flame cutting is economically superior to machining, and with modern flame-cutting equipment a high degree of accuracy may be obtained.*

The accuracy of the plate preparation not only plays an important

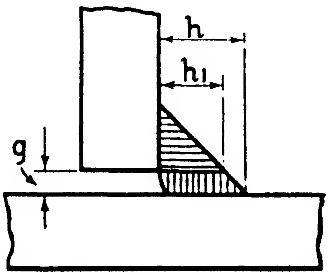


FIG. 223

part in facilitating the assembly work † but may also affect the efficiency of the welded joints. If a fillet-weld has to join two plates at right angles and the abutting plate is cut too short (Fig. 223), a gap, *g*, will

* See E. S. Semper and L. J. Hancock, "Plate Edge Preparation for Welding", *Transactions of the Institute of Welding*, April 1946, page 54.

† See F. Koenigsberger, "Production in the Fabricating Shop, Inspection Procedure", *Welding*, December 1944.

result, which reduces the actual size, h, of the weld to an effective size given by

$$h_1 = h - g,$$

necessitating an increase in leg length from the necessary h_1 to $h = h_1 + g$.

If the tolerance allowed for the plate preparation may result in such a gap being formed, due allowance must be made when determining the measurable weld size h.

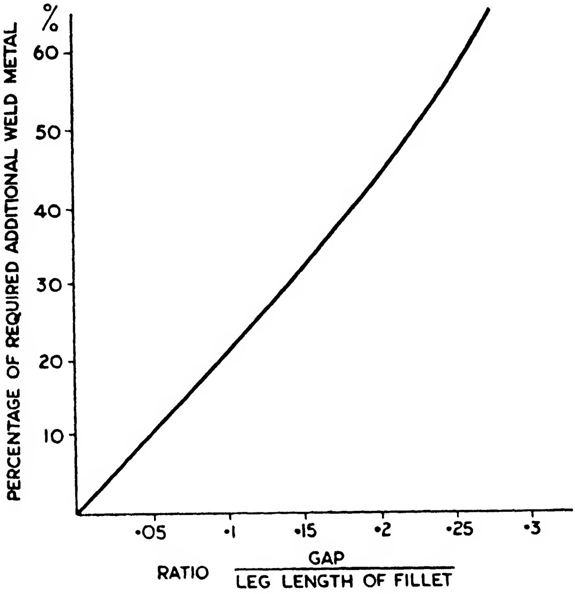


Fig. 224

The proportion of additional weld area thus required may be determined as follows :

If A is the weld sectional area required for strength reasons, and A_{add} the additional area required because of the gap g,

$$A = \frac{h_1^2}{2},$$

$$A_{add} = gh_1 + \frac{g^2}{2},$$

and

$$\frac{A_{add}}{A} = \frac{2g}{h_1} + \left(\frac{g}{h_1}\right)^2.$$

The graph (Fig. 224) shows this relation as a percentage of the minimum amount of weld metal required for strength reasons.

If, for example, the gap is ·125″ and the weld size theoretically required is " 1 ", 26·6 per cent more weld metal must be deposited. The importance of keeping the plate preparation tolerances to an economic minimum will be obvious.

Not only must the shapes of plate and other component parts and the preparations for the various welds be established and specified on the detail drawing, but due consideration must be given to the assembly procedure and the welded joints themselves.

The specification of the assembly procedure includes the sequence of operations to which the assembly plater should work and the way in which the welder should put down the welded seams. There are, of course, cases where these points are not of vital importance and the designer need not then specify detailed instructions. On the other hand, there are cases where correct assembly will only be possible if one particular sequence of operations is adhered to. An example is provided by the milling machine body (see Fig. 193), which must be assembled upside down, that is, by laying down the two slides first and building up the body over the slides, as otherwise it would be impossible to complete the very important internal welds under the slideways. These would become inaccessible if the fabricating shop, due to lack of information, proceeded in the more " logical " way by building up the body starting from the base frame.

The sequence and procedure of the welding operations may have an influence on the distortions of the component, and, if this is the case, definite instructions as to which welds have to be laid down first must again be given on the drawing. If, for example, part of a structure is likely to distort more than permissible under the heat of certain welding operations, these operations should be delayed until other stiffening components have been welded in position.

The argument, that it is better to permit distortion rather than to risk the danger of internal residual stresses and cracks, is only valid up to a point. In a machine structure, distortion might mean that a particular assembly operation could not be executed because the distortions due to a preceding operation were too great. In such cases prevention of distortion through struts or through a suitable welding and assembly procedure is imperative, as residual stresses can be eliminated subsequently by heat treatment (see page 201).

The welding-assembly drawing itself need not necessarily be identical with the actual design drawing of the job in question. It should not give finished machined dimensions, but should show the welded structure in its unmachined state, in order to relieve the assembly fitter or plater from the additional work of making calculations and taking machining allowances into account when positioning the components. Such a welding-assembly drawing will also prove helpful to the tool designer in cases where quantity production requires the use of assembly jigs.

Whilst plate preparations such as chamfering or gouging for butt welds are specified on the respective detail drawings, the actual welding procedure should be shown on the welding drawing.

The welder does not require any dimensions except the weld sizes. It is unnecessary, and may even be confusing, to indicate on the welding drawing anything but the types of welds required, the weld sizes, the recommended procedure of executing them, and probably the sequence in which the various welds must be carried out. The details of the welds themselves may be indicated by means of symbols such as those suggested in B.S.S. 499 (Figs. 225 and 226).*

In certain cases it may be helpful if the welding drawing shows the manipulator position in which welds can be put down most efficiently (Fig. 227). This drawing could at the same time contain the weld specifications, which may be indicated for arc welding as shown in Fig. 228, and for spot welding as shown in Fig. 229.

It may be advisable to determine at this point how far the Drawing Office should go in its specifications and where the task of the Planning Department begins and should not be interfered with.

Whilst the drawing calls, for instance, for a certain pressure, amperage, and welding time on a spot-welded job, or for a certain electrode type, size, and amperage to be used in arc welding, the Planning Department must indicate the welding machine to which the job is allocated, and the tap number on the arc welding transformer, or, in the case of spot welding, tap number, pressure setting, and time adjustment which give the required values on the respective resistance welding machine.

This is a very important point to consider, and friction between the two departments may easily occur through a misunderstanding of their respective duties. To repeat, therefore : the Drawing Office establishes requirements in technical measurements—in pounds, amperes, and seconds, for spot welding, for instance. The Planning Department then tells the shop which numerical machine settings correspond to these technical measurements on the particular machine allocated to the job.

Recommended welding procedure specifications for the welding of ordinary mild steel (maximum carbon content up to about ·2–·23 per cent) are shown for arc welding in Figs. 230, 231, 232,† for spot welding in Fig. 233, and for projection welding in Figs. 234, 235, 236.

With regard to the number of runs to be laid down for a particular arc weld, it should be remembered that for economic reasons it is generally desirable to lay down a weld with a minimum number of runs, thus depositing the greatest volume of metal in the shortest possible

* Extracted from B.S. 499, Nomenclature, Definitions, and Symbols for Welding and Cutting, by permission of the British Standards Institution, 28 Victoria Street, London, S.W.1, from whom official copies can be obtained, price 2s. post free.

† Figs. 230-32 have been taken from *Handbook for Electric Welders*, published by Murex Welding Processes Limited, Waltham Cross, Herts., 1945. The specifications refer to Murex Fastex 5 Electrodes.

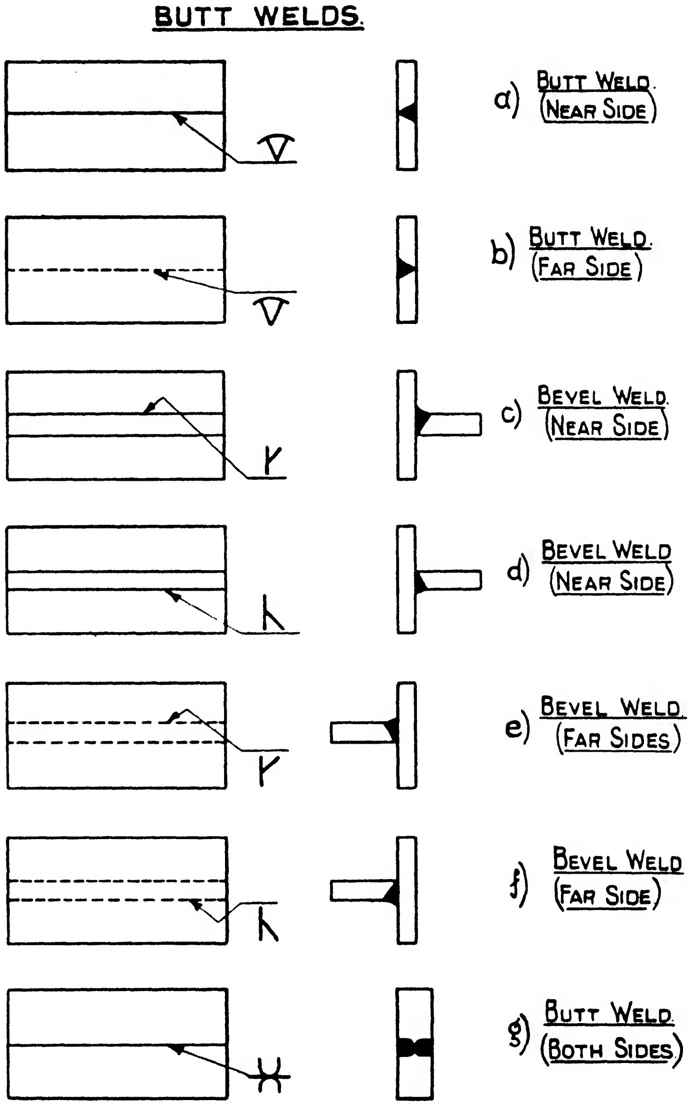

BUTT WELDS.
a) Butt Weld. (Near Side)
b) Butt Weld. (Far Side)
c) Bevel Weld. (Near Side)
d) Bevel Weld (Near Side)
e) Bevel Weld. (Far Sides)
f) Bevel Weld (Far Side)
g) Butt Weld. (Both Sides)

Fig. 225

time. On the other hand, it has been proved that, in the case of multi-run welds, each run anneals the preceding one, thus giving increased strength to the multi-run weld, especially against impact loads. Experiments have shown * that the impact strength of a ⅝″ butt weld produced

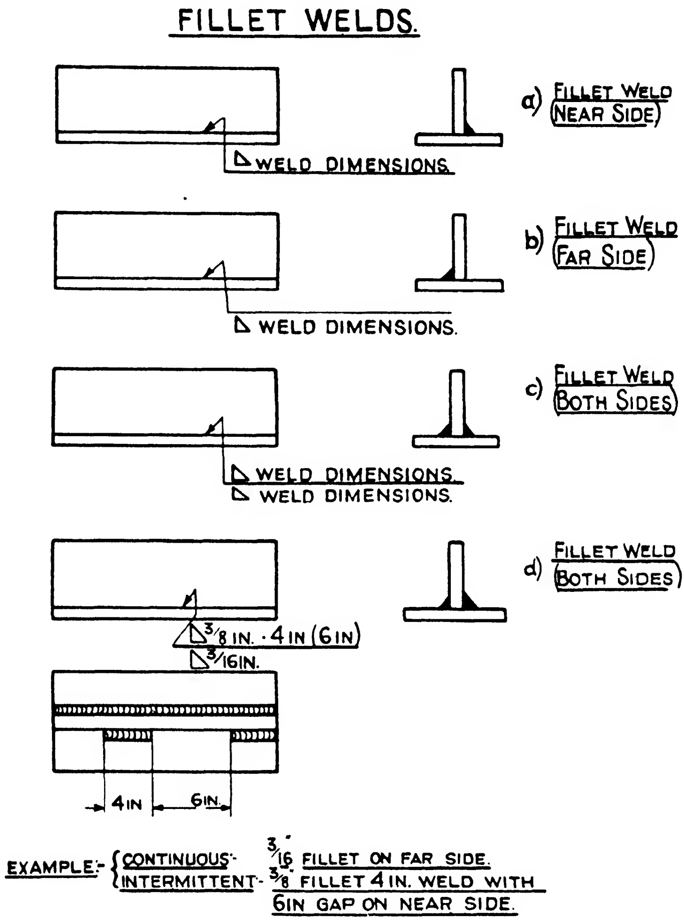


FIG. 226

with 7 runs was about 4 times as great as that of a ⅝″ weld produced with one single run. Multi-run welds may, therefore, be usefully applied when welds are likely to be heavily stressed by impact, or on structures for which heat treatment is difficult or impossible (see the crane bridge, Fig. 170).

* U. Guerrera, *L' Industria Meccanica*, September 1936.

Whilst careful specification of weld sizes and welding procedure is of great importance, especially with regard to highly stressed welds, it

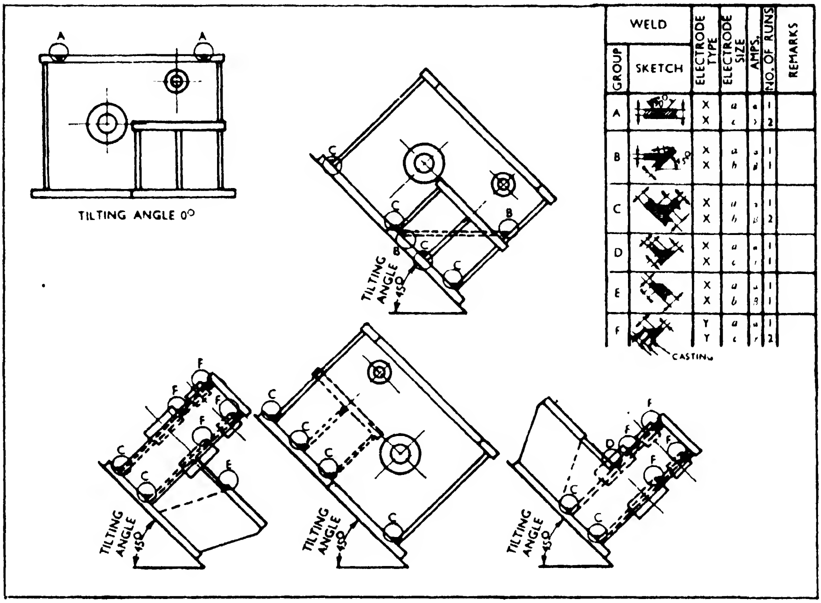


Fig. 227

should not be forgotten that in nearly every welded construction there will be welds which are only used for location or similar purposes. These welds should be clearly indicated as such on the drawing, in order

Arc Weld Settings						
Weld		Electrode		Amps	No. of Runs	Remarks
Group	Sketch	Type	Size			

Fig. 228

to prevent the workshop, and later the inspection, wasting time by applying more than the necessary care to them.

In case of singly produced structures which do not warrant too much Drawing Office work, general welding and assembly drawings only will be necessary. Careful indication, however, at least of assembly procedure and sequence of welding operations, will save many disappointments and low efficiency in the shop.

A point which needs particular attention is the creation of internal residual stresses, and their reduction or removal by heat treatment.

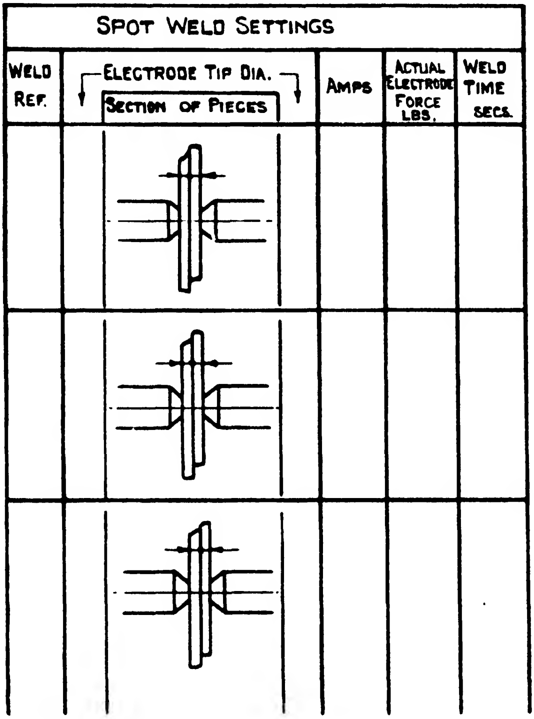


FIG. 229

Internal stresses may be caused during the welding process by the different expansions and contractions of the various component parts, especially if free movement of the parts in question is prevented by the rigidity of the structure itself, or by struts or stiffeners which are applied in order to keep it in its correct shape during the welding operations.

Fig. 237 shows the front of the milling machine body (see Fig. 193). During the welding operations stiffener A is left in the structure in order to prevent the two legs B and C of the U-shaped opening from collapsing. Stiffener A is removed (mechanically, not by flame cutting which would introduce new stresses) only after the residual stresses have been reduced by heat treatment.

WELDING PROCEDURE

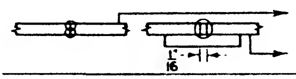
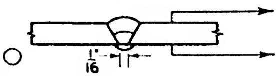
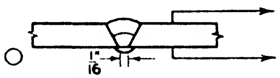
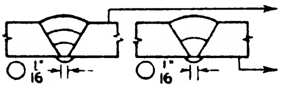
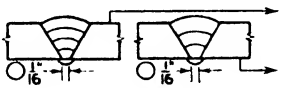
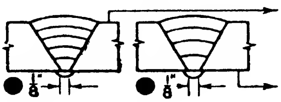
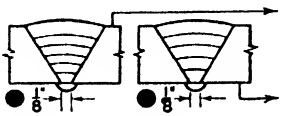
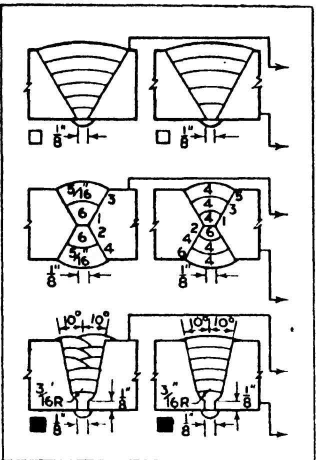
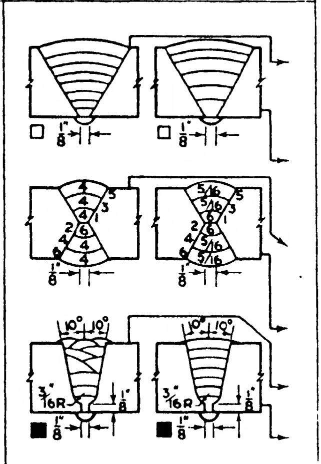
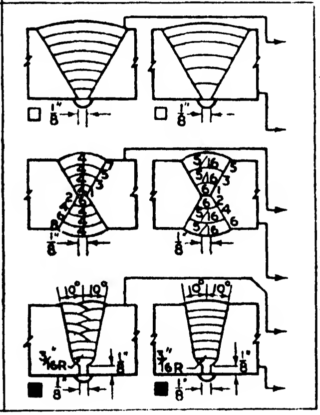
DIMENSIONS OF WELD — Plate Thickness (In.)	60° Vee Preparations for Downhand Welding unless shown otherwise — Preparation	Number of Runs (No.)	Gauge of Electrode (S.W.G.)	Length of Weld per Electrode (Gauge/In.)	Welding Current (Amps.)
1/8		2	8	8/21	170
		1	8	8/14½	170
3/16		1	10	10/15	100
		1	10	10/11	120
		1	10	10/15	100
		1	8	8/16½	170
1/4		1	10	10/15	100
		1	8	8/9	170
		1	10	10/15	100
		1	6	6/12	210
5/16		1	10	10/12	100
		2	8	8/12	170
		1	10	10/12	100
		1	6	6/8	210
3/8		1	10	10/12	100
		3	8	8/12	170
		1	10	10/12	100
		2	6	6/12	210
1/2		3	8	8/12	150
		2	8	8/7	170
		1	8	8/12	150
		1	6	6/12 }	210
		2	6	6/9 }	
5/8		1	8	8/12	150
		3	8	8/9 }	170
		1	8	8/6 }	
		1	8	8/5 }	
		1	8	8/12	150
		1	6	6/12 }	210
		1	6	6/10 }	
		1	6	6/6 }	
		1	6	6/5½ }	
○	KEY TO SEALING RUNS	1	10	10/16½	120
●		1	8	8/16½	170
□		1	6	6/16½	210
■		1	4	4/12	340

Fig.

3/4		1	8	8/12	150
		1	6	6/12	210
		1	6	6/10	
		1	6	6/9	
		2	6	6/6	
		1	8	8/12	150
		2	4	4/12	250
		2	4	4/6	
		2	6	6/16½	200
		2	5/16	5/16/12	560
		1	4	4/16½	340
		1	6	6/16½	240
		2	4	4/12	340
		2	4	4/9	340
		1	4	4/20	240
		8	4	4/12	340
		1	4	4/20	240
		5	5/16	5/16/12	560
7/8		1	6	6/12	190
		3	4	4/12	250
		1	4	4/9	
		2	4	4/6	
		1	4	4/4	
		1	6	6/12	190
		2	4	4/12	300
		3	4	4/8	
		1	4	4/6	
		1	4	4/19	340
		1	6	6/16½	240
		2	4	4/12	340
		2	4	4/8	340
		2	6	6/15	200
		2	5/16	5/16/12	560
		2	5/16	5/16/16½	560
		1	4	4/20	240
		9	4	4/12	340
		1	4	4/20	240
		6	5/16	5/16/12	560
1		1	6	6/12	190
		5	4	4/12	300
		2	4	4/6	
		1	6	6/12	190
		2	5/16	5/16/12	430
		1	5/16	5/16/11	
		2	5/16	5/16/8	
		1	4	4/16½	240
		1	6	6/20	240
		4	4	4/12	340
		2	4	4/9	340
		2	6	6/15	200
		4	5/16	5/16/12	560
		1	4	4/20	240
		10	4	4/12	340
		1	4	4/20	240
		7	5/16	5/16/12	560

Dimensions of Weld		Welding Procedure			
Size of Fillet	Throat Thickness	Number of Runs	Gauge of Electrode	Length of Weld per Electrode	Welding Current
In.	In.	No.	S.W.G.	Gauge/In.	Amps.
$\frac{1}{8}$	0·088	1	10	10/16½	120
		1	8	8/26½	170
$\frac{3}{16}$	0·133	1	10	10/8	120
		1	8	8/10	170
		1	6	6/22	210
$\frac{1}{4}$	0·176	1 1	10 10	10/12 10/6	120 120
		2	8	8/12	170
		1	6	6/12	210
		1	4	4/18	250
$\frac{5}{16}$	0·221	2 1	10 10	10/12 10/6	120 120
		2 1	8 8	8/12 8/22	170 170
		2	6	6/12	210
		1	4	4/13	250
$\frac{3}{8}$	0·265	1 2	10 10	10/12 10/6	120 120
		1 1	8 8	8/12 8/6	170 170
		1 1	6 6	6/12 6/9	210 210
		1	4	4/8	250

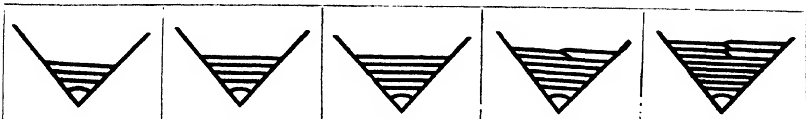
KEY TO

SKETCH			
No. of Runs	1	2	3

FIG.

Dimensions of Weld		Welding Procedure			
Size of Fillet	Throat Thickness	Number of Runs	Gauge of Electrode	Length of Weld per Electrode	Welding Current
In.	In.	No.	S.W.G.	Gauge/ In.	Amps.
1/2	0·354	1	8	8/12	170
		2	8	8/6	170
		3	6	6/12	210
		1	6	6/19	210
		2	4	4/12	250
		1	4	4/25	250
5/8	0·441	1	8	8/9	170
		3	8	8/6	170
		1	6	6/9	210
		2	6	6/6	210
		2	4	4/12	250
		1	4	4/8	250
3/4	0·530	1	8	8/12	170
		3	8	8/6	170
		1	8	8/4	170
		1	6	6/9	210
		3	6	6/6	210
		1	4	4/9	250
		2	4	4/6	250
7/8	0·618	1	8	8/12	170
		7	8	8/6	170
		2	6	6/12	210
		4	6	6/6	210
		1	4	4/12	250
		3	4	4/6	250
1	0·707	1	8	8/12	170
		10	8	8/6	170
		2	6	6/12	210
		6	6	6/6	210
		2	4	4/12	250
		4	4	4/6	250

WELDING PROCEDURE



4 5 6 8 11

Dimensions of Weld		Welding Procedure			
Size of Fillet	Throat Thickness	Number of Runs	Gauge of Electrode	Length of Weld per Electrode	Welding Current
In.	In.	No.	S.W.G. In.	Gauge/ In.	Amps.
$\frac{3}{16}$	·133	1	$\frac{1}{4}$	$\frac{1}{4}$/28	320
$\frac{1}{4}$	·176	1	$\frac{1}{4}$	$\frac{1}{4}$/20	320
$\frac{5}{16}$	·221	1	$\frac{5}{16}$	$\frac{5}{16}$/21	450
$\frac{3}{8}$	·265	1	$\frac{5}{16}$	$\frac{5}{16}$/14	450
$\frac{3}{8}$	·265	1	$\frac{3}{8}$	$\frac{3}{8}$/18	580
$\frac{1}{2}$	·354	1	$\frac{5}{16}$	$\frac{5}{16}$/9	480
$\frac{1}{2}$	·354	1	$\frac{3}{8}$	$\frac{3}{8}$/14	580
$\frac{5}{8}$	·441	2	$\frac{5}{16}$	$\frac{5}{16}$/10	480
$\frac{5}{8}$	·441	2	$\frac{3}{8}$	$\frac{3}{8}$/14	580
$\frac{3}{4}$	·530	3	$\frac{5}{16}$	$\frac{5}{16}$/9	480
$\frac{3}{4}$	·530	3	$\frac{3}{8}$	$\frac{3}{8}$/13	580

KEY TO WELDING PROCEDURE

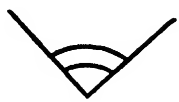
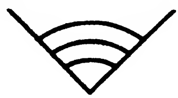
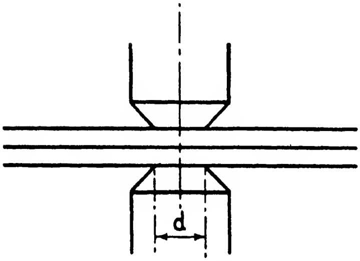
SKETCH			
No. of Runs	1	2	3

FIG. 232

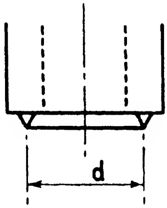
Residual stresses may be detrimental—

(*a*) In the case of highly stressed structures, particularly when stress raisers (notches, sharp corners, etc.) may start cracks.



d	WELDING CURRENT	ELECTRODE PRESSURE	WELDING TIME
INCHES	AMPS.	LBS.	SECS.
$\frac{3}{16}$	8000	450	•1
$\frac{1}{4}$	10000	750	•2
$\frac{5}{16}$	13000	1200	•4
$\frac{3}{8}$	16000	1700	•6
$\frac{7}{16}$	17500	2300	•8
$\frac{1}{2}$	20000	3000	1•4
$\frac{9}{16}$	28000	3800	1•4
$\frac{5}{8}$	31000	4700	2•0

FIG. 233



d	WELDING CURRENT	WELDING PRESSURE	WELDING TIME
INCHES	AMPS.	LBS.	SECS.
$\frac{1}{4}$	23000	1200	0•10
$\frac{5}{16}$	29000	1300	0•10
$\frac{3}{8}$	29000	1600	0•25
$\frac{7}{16}$	29000	1900	0•38
$\frac{1}{2}$	31500	2200	0•10
$\frac{9}{16}$	31500	2400	0•25
$\frac{5}{8}$	31500	2650	0•50
$\frac{11}{16}$	39500	2900	0•25
$\frac{3}{4}$	39500	3200	0•38
$\frac{13}{16}$	39500	3500	0•75
$\frac{7}{8}$	45000	3850	1•25
$\frac{15}{16}$	45000	4250	1•75
1	52000	5000	•75

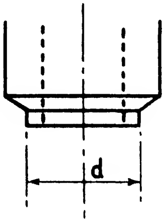
FIG. 234

(*b*) In the case of precision structures, where "creeping" of the structure may occur during its working life and the dimensional stability may be jeopardized.

(*c*) In the case of parts which are exposed to chemical agents and where corrosion embrittlement may be caused.

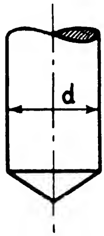
Heat treatment of such structures will remove or at least reduce internal stresses.

The heat treatment procedure should be carefully specified by the



d	WELDING CURRENT	WELDING PRESSURE	WELDING TIME
INCHES	AMPS.	LBS.	SECS.
$\frac{1}{4}$	23000	1200	·1
$\frac{5}{16}$	29000	1300	·1
$\frac{3}{8}$	29000	1600	·25
$\frac{7}{16}$	29000	1900	·38
$\frac{1}{2}$	31500	2200	·1
$\frac{9}{16}$	31500	2400	·25
$\frac{5}{8}$	31500	2650	·5
$\frac{11}{16}$	39500	2900	·25
$\frac{3}{4}$	39500	3200	·38
$\frac{13}{16}$	39500	3500	·75
$\frac{7}{8}$	39500	3850	1·25
$\frac{15}{16}$	39500	4250	1·5
1	50000	5000	·5

FIG. 235



d	WELDING CURRENT	WELDING PRESSURE	WELDING TIME
INCHES	AMPS.	LBS.	SECS.
$\frac{1}{8}$	15500	180	·1
$\frac{3}{16}$	18000	420	·1
$\frac{1}{4}$	23000	750	·1
$\frac{5}{16}$	29000	1160	·1
$\frac{3}{8}$	31500	1650	·1
$\frac{7}{16}$	31500	2250	·2
$\frac{1}{2}$	39500	2950	·1
$\frac{9}{16}$	39500	3750	·2
$\frac{5}{8}$	45000	4500	·1
$\frac{3}{4}$	45000	6600	·2

FIG. 236

designer in accordance with the requirements of the job. The main factors to be noted are :

(1) The heat treatment temperature.

(2) The soaking time, *i.e.* the time during which the heat treatment temperature in the furnace is to be held constant.

(3) The cooling process.

(1) The influence of the heat-treatment temperature on the reduction of residual stresses can be seen from the graph* (Fig. 238). A tempera-

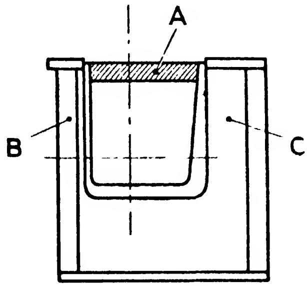


FIG. 237

ture of 600°–650° C. is usually satisfactory for relieving the stresses in general engineering structures, whilst the author has successfully used slightly higher temperatures (780°–800° C.) in the case of high-precision frames for machine tools and fixtures.

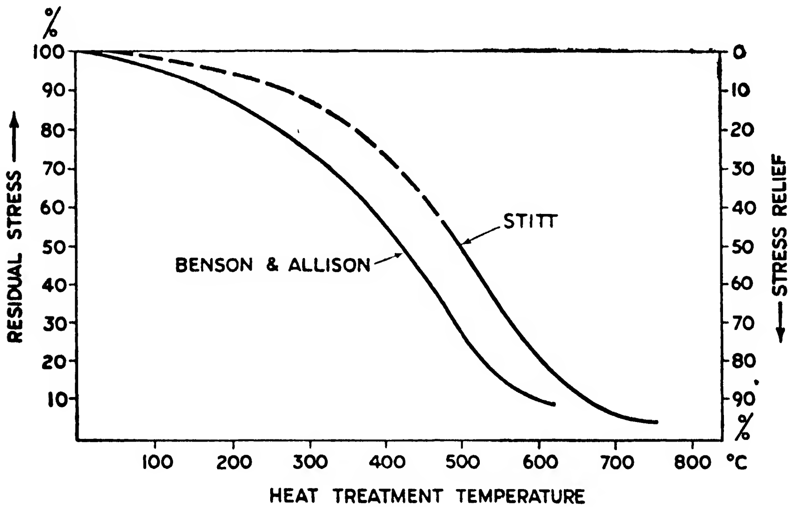


FIG. 238

(2) The soaking time should be long enough to enable every part of the structure to reach the full heat treatment temperature. This

* L. E. Benson and H. Allison, " Low-Temperature Annealing of Welded Mild Steel Structures to relieve Internal Stresses ", Iron and Steel Institute, *Symposium on the Welding of Iron and Steel*, 1935, vol. 2, page 79 ; I. R. Stitt, " Stress Relief of Weldments for Machining Stability ", *American Welding Journal*, June 1945, page 331 s, and January 1946, p. 30 s.

should be checked by thermocouples which are placed at various points of the charge in the furnace.

(3) The cooling process should be such as to avoid temperature gradients, and the furnace doors should not be opened before the temperature has dropped to about 200° C. in the case of general engineering structures, or 100° C. in the case of high-precision work. With general structures, if good screening provision is made, it may be permissible to open the furnace doors at 400° C., but not above that temperature.

A special committee of the British Welding Research Association has considered in detail the problem of the heat treatment of welded structures, and, as the result of this work, detailed recommendations have been issued.*

The final step in connection with the welding production process is a careful inspection. If properly applied, this may prevent subsequent manufacturing difficulties and the damage which may be done by allowing faulty parts to be carried through the whole production procedure.

In his paper " The Testing of Welds,"† H. N. Pemberton makes the statement that " the greatest safeguard of weld quality is to be found in works supervision of all the practical details comprising welding procedure ". This applies to the welded seam alone. In the work of the fabricating shop as a whole, however, rigid inspection procedure of plate preparations and welding operations, and of each operational step throughout the whole process, from drawing raw material from the stores to the completion of the finished product, is of vital importance.

In general machine-shop procedure, inspection after the completion of each section of the work is more or less standard practice, but in the worst case, a faulty component which is bolted or fitted to an assembly can still be removed if the fault is only detected later on. In a fabricated welded structure, faults in size, shape, or weld preparation of detail parts are difficult to detect or to repair after the parts have been welded together. It is therefore far more important to detect such faults while the parts can still be easily separated.

Careful inspection may prevent the shop from wasting time and labour on work which is already faulty, due to errors in previous operations. As the production personnel have an interest in carrying their jobs forward without losing output due to faulty operations, their judgment may be biased, when considering whether or not a repair or rectification is permissible. They must not be allowed, therefore, to cover or rectify errors at their own discretion. The decision as to whether or not a concession can be granted for a faulty job, or if or how rectification should be carried out, must be left to the technical authority,

* *Recommendations for Heat Treatment of Welded Constructions in Mild Steel.*

† H. N. Pemberton, " The Testing of Welds ", North East Coast Institution of Engineers and Shipbuilders, 1942, page 205.

the Design Engineer, who will inform the Chief Inspector accordingly. Otherwise the Inspection Department, which should be directly responsible to the technical authority, must only pass a job which is strictly in accordance with the instruction specified on the drawing.

In order to guide the inspectors in the making of decisions with regard to dimensional accuracy, tolerances should be specified by the Drawing Office and gauges should be provided wherever possible. Such gauges should cover plate preparations, chamfers, radii, etc., for detail inspection of plates and shapes of finished welds, and the inspec-

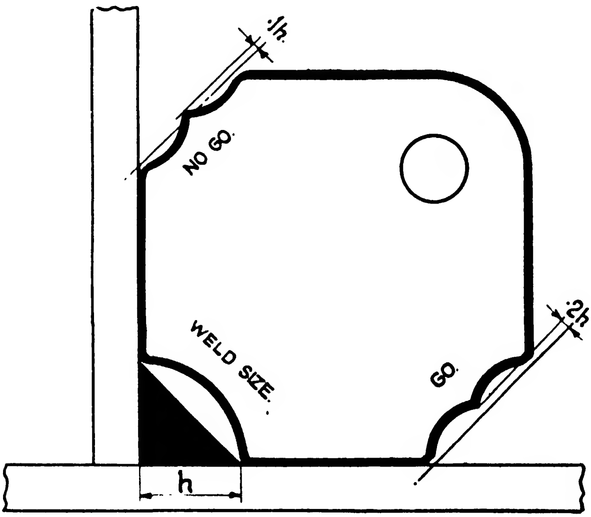


FIG. 239

tion of completed assemblies. The "go" and "no go" fillet-weld gauge (Fig. 239) allows the tolerance of the throat thickness $(\cdot 707h^{+\cdot 2h}_{-\cdot 1h})$* to be checked easily.

The Inspection Department should also be provided with a drawing of the finished machined structure, in order to be able to decide, in border-line cases, whether or not a job will be machinable.

Magnetic crack detection and X-ray inspection lie within the field of the Inspection Department.

The inspection processes are especially important in structures where welds are highly stressed, and X-ray inspection of every weld is a condition for the acceptance of Lloyd's class 1 pressure vessels. Instructions to this effect must appear clearly on the drawing.

If crack testing is applied, this should be carried out after heat treatment, as it may happen that welds or component parts crack

* B.S.S. 538.

during heat treatment. Here the importance of a graph recording the heat treatment process will be obvious (Fig. 240), as with it the Engineering Department will be able to determine whether incorrect heat treatment has had a part in crack formation.

It is often advisable to check whether or not the main dimensions of a structure have been affected by the heat treatment process. Here

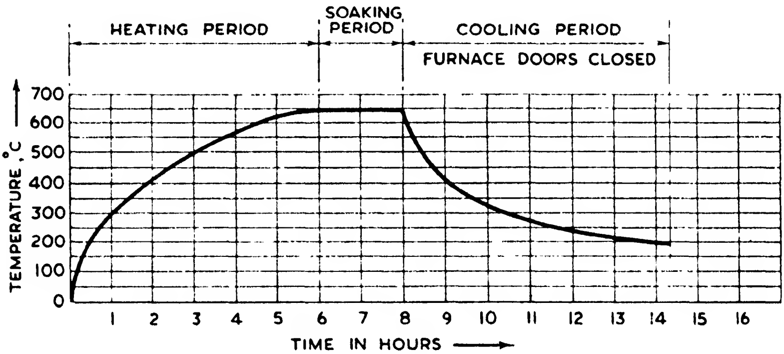
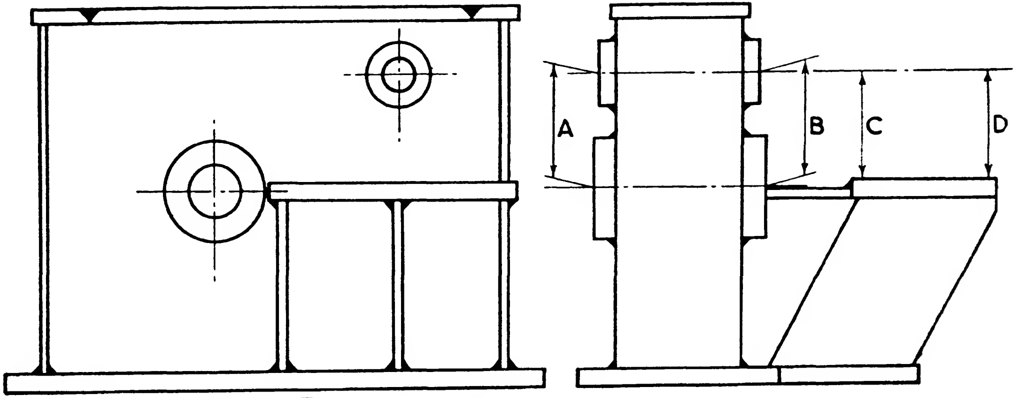
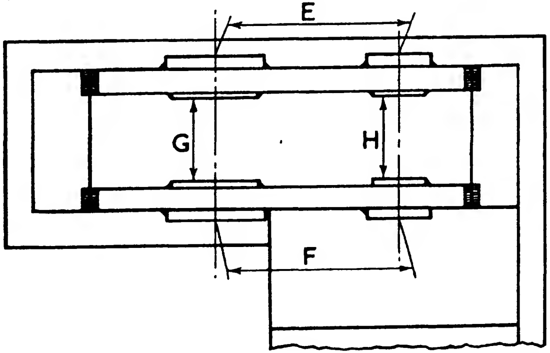


FIG. 240

again it should not be left to the inspector to decide which dimensions are to be considered of importance. On the other hand it would be uneconomical to inspect the whole structure twice. An inspection specification issued by the Drawing Office, as shown in Fig. 241 for the case of the reduction gear-box (Fig. 221), will show the inspector which dimensions are to be checked. If required, this sheet may also be used for record purposes of the Inspection Department, and can be filed away together with the graph record of the heat treatment.





MEASUREMENT	DIMENSION: DESCRIPTION	DIMENSION: TO DRAWING	DIMENSION: ACTUALLY MEASURED BEFORE HEAT TREATMENT	DIMENSION: ACTUALLY MEASURED AFTER HEAT TREATMENT	REMARKS
PARALLELISM OF AXES VERTICAL	A				
DITTO	B				
PARALLELISM OF AXES AND MOTOR BASE	C				
DITTO	D				
PARALLELISM OF AXES HORIZONTAL	E				
DITTO	F				
DISTANCE BETWEEN BIG BEARING BOSSES	G				
DISTANCE BETWEEN SMALL BEARING BOSSES	H				

FIG. 241

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