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A DESIGN STUDY OF A NUMERICALLY CONTROLLED FRAME BENDING MACHINE

By M. SHAMA, B.Sc.,* (*Associate*), and N. S. MILLER, B.Sc.†

Summary

The impact on production procedure of recent developments in mathematical methods of defining the shape of a ship hull is briefly considered. It is shown that the logical outcome of these developments is the numerical control of all machine tool operations connected with forming parts of the hull structure. The traditional methods of frame forming are discussed and it is concluded that the cold-forming process using the 3-point bending method is the most practical for modern control systems.

The considerations which have to be borne in mind when designing a numerically controlled frame bender are discussed, emphasis being placed on the need to consider all frame operations in a shipyard with particular reference to those operations which could be performed in conjunction with the bending process.

A system of control for an automatic 3-point-bending machine is described together with the accuracies with which the various measurements have to be made. No attempt is made to design specific items of the equipment. Laboratory and full-scale tests have been carried out which have established the feasibility of application of the proposed control system.

Factors which are important to the successful operation of such a machine, e.g. length of frame between bends, minimum bend radius and buckling have been considered, and the practical limitations established. The problems arising from the residual stresses set up by the bending operation are discussed and it is shown that these may be serious from the point of view of stress and deformation of the frame when welded.

The text describes the problems involved in general terms and the mathematical statement of these problems is given in the Appendices.

Introduction

The wide-spread adoption of welding has enabled a ship to be constructed in large prefabricated units under cover. Only the actual assembly of the units takes place on the building berth. The construction of these units in indoor workshops has, in turn, enabled modern methods of production and control to be applied and this has recently received additional impetus from the advent of the electronic digital computer and associated control techniques.

Computers are now widely used for design calculations and attempts are being made to bridge the gap between design and construction. Methods of mathematically defining the hull shape have been developed,[1,2,3] and these greatly facilitate the introduction of numerically controlled machine tools.

Numerically controlled burners have already been developed by the British Oxygen Company in association with Ferranti Ltd., in Norway by the Central Institute of Industrial Research and in the U.S.A. by the Air Reduction Co.[4]

The British Oxygen Co.'s Eagle machine has already demonstrated its high accuracy resulting in savings in time when assembling the prefabricated sections and also in a substantial reduction (5–10 per cent)[5] in the amount of weld metal deposited.

Nevertheless, the total savings due to introducing numerically controlled burning are small, since a mould loft must still be retained in order to produce templates for bending the frames and plates to shape. This means that the accuracy attained at one stage is offset by errors introduced elsewhere, and the possibilities of improved production control cannot be achieved. Consequently, the chief need is to eliminate mould lofting and associated templates completely and to form or burn the parts of the hull from purely numerical data. Besides the direct savings in mould loft and scrieve board costs, this, in fact, would enable modern production planning and control techniques to be applied throughout the steel working shops and would eventually reduce costs, as a result of the greater accuracy of formed component parts.

This paper is concerned solely with the problems of the cold-forming of frames using numerical control. A discussion is given of the theoretical aspects of the problem and a summary of the tests which have been carried out, leading to the design of a numerically controlled machine. This machine, which is the subject of a patent application by Glasgow University, is now being developed by Hugh Smith Ltd. in association with Ferranti Ltd., who are receiving financial help from the British Ship Research Association.

Traditional Methods of Frame Forming

Three methods of frame forming are in common use in British shipyards:—

1. *Furnacing.*—Here the frame is heated to a sufficiently high temperature (about 650° C.) to enable it to be bent on heavy thick slabs using mainly hand tools.[6] There is no control on the temperature, the workman depending mainly on his experience to heat the frame until it reaches a certain colour. Overheating is possible with consequent deleterious effects on the mechanical properties of the steel. In addition the furnace is not always used to full capacity with resulting wastage of fuel.

* Research student, Naval Architecture Department, Glasgow University.

† Senior Lecturer, Naval Architecture Department, Glasgow University.

The process is costly in equipment and man-power and cannot easily be fitted into a modern material flow system, since it is not easy to shotblast and paint the bent frames automatically due to the large area occupied and the difficulty in handling bent frames efficiently. However, it has the advantage that spring-back problems are removed by the heating and there is no material wastage.

2. *Cold-Frame Bending.*—Here the frame is passed through a machine which applies a 3-point bending procedure in stages. In some machines, the central point is fixed, the load being applied by two moving arms on either side of it, whereas in others the reverse is the case.

The bending is carried out by the progressive method, i.e. the frames are clamped, bent, fed through, clamped, bent and fed through progressively. The bent shape is checked by templates prepared from the mould loft or scrieve board. At each point bending is carried out until a satisfactory shape has been achieved locally and then the frame is fed through for the next bend.

The operator determines from experience, knowing the required shape of curve, the intervals between each two bending operations. He depends on his judgment for the amount of deflection to be imposed at each point. In fact the whole operation depends on art rather than science.

Depending on the shipyard and the length of the template used, from two to five men are required for the operation. During most of the time only the machine operator is working, the remainder are waiting to lift the template, assist with feeding the frame and removing it from the machine, etc. Only seven to eight pairs of frames per day are produced by this process. However, the utilisation factor for this machine taken over a year is commonly about 30 per cent of an 8-hour day, in British shipyards.

A certain amount of scrap is unavoidable in the use of these machines since about 2ft, of frame at each end cannot be bent. This disadvantage can be reduced by the ship designer by eliminating the length of the frame to be bent to a sharp curvature. This, in effect, will reduce the number of frames to be bent since the majority of frames, in the parallel middle body, will no longer require any bending. From an examination of the drawings produced by yards possessing these machines few designers seem aware of this practical difficulty.

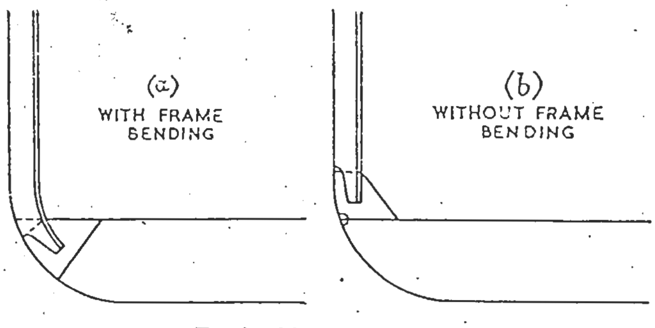


FIG. 1.—MIDSHIP SECTIONS

Fig. 1 shows two designs of bracket connection at the tank top. Design (*b*) involves no bending over the parallel middle body length and would appear to be a much cheaper connection than design (*a*), bearing in mind the handling and bending.

3. *Frames Manufactured from Plates.*—One or two yards construct their frames by burning the shape of the web out of plating and welding a flat bar stiffener to form a Tee section. This method is believed to be more expensive than the others due to the cost of the machine tools, labour and scrap involved. Difficulties are experienced due to thermal distortions caused by the burning and welding.

Other methods have been attempted, e.g. applying high lateral pressures on the convex side of the frame web so as to reduce the thickness and consequently induce bending in the desired direction, but none of these has proved sufficiently attractive to be used in shipyards for frame bending.

The most widely adopted method is cold frame bending, using the three point bending method which appeared to be the most efficient and it was decided to try and develop this method for numerical control.

Present Methods of Control

For all these methods, the shape of the frame has to be transferred from the mould loft (full scale or 1/10th scale) in the form of a template or of a full-scale drawing on the floor. If 1/10th scale lofting is used, a variety of techniques can be used to obtain the templates (automatic burning, projection from an optical tower and then hand cutting, etc.). Whatever the system used, several transferences of data are involved introducing errors which cannot be properly assessed since other factors, welding distortions, mechanical distortions of large sections during lifting, thermal effects, etc., all combine to produce a lack of fit between prefabricated sections which is common in shipyards. Further, it is general practice to store the templates so that they may be used in the future when a sister ship is to be built. The storage facilities will always introduce errors due either to distortion or mechanical damage.

No figures are available for actual accuracies achieved, but rectification costs for prefabricated assemblies on the berth are known to be high in most shipyards and part of these arise in the frame and plate bending operations.

General Frame Operations

It is important to consider the general frame operations before any design requirements for a numerically controlled bending machine are established, since its performance must be considered in relation to the planning and execution of the other frame operations from ordering to arrival in the prefabrication shed. There would be no great incentive to control and speed up the bending operation if some of the associated operations are still slow and inefficient through the use of templates. It is essential that the accuracy achieved by numerical control of the frame bending is not offset by errors introduced by manual operations.

The need for the integration of numerical control into the complete shipyard operation can be illustrated by the difficulties experienced in introducing numerically controlled burners to the shipbuilding industry. Although these were very well engineered and involved very extensive computer programming preparation, the fact that the mould loft and templates had to be retained for frame bending, plate bending and some other operations meant that few of the potential economic advantages could be realized in practice. It follows that, even with numerically controlled frame bending, it will be essential for plate bending and marking to be done by a numerical method before the complete benefits of automatic machine tools in planning, speed and accuracy can be obtained. If a numerically controlled frame bender can be made then steel templates can be produced by numerical control for plate bending and although this would only be a half-way stage to full automation it would obviate the need for a mould loft.

Recently, mould lofts have undergone considerable changes due to the conversion from full scale to 1/10th scale lofting with its consequent emphasis on drawing office type of work. It is therefore difficult to assess the precise savings that will accrue from the elimination of this department. In a medium shipyard producing ships up to about 70,000 tons deadweight, the savings in space might range from 20,000 $ft.^2$ to 4,000 $ft.^2$ depending on whether full scale or 1/10th scale lofting is used. There will be savings in material for scrieve boards, templates, etc., which will be of the order of £500 per annum. The problem of template storage and its associated cost will be obviated by all the data being stored on paper tape.

The savings in labour are difficult to estimate since, although fewer men need be employed, the personnel involved in planning and programming will require higher salaries. Eventually it is to be anticipated that a large part of the planning can also be programmed on a computer and it is possible that the staff reduction in the mould loft will be of the order of six to eight men.

The above are all direct savings but the indirect savings due to the greater accuracy possible with numerical control and the reduced time between receiving an order and being able to start production may produce greater economic benefits. McIver[5] has stated that computer controlled burning has resulted in a substantial reduction (5–10 per cent) in the amount of weld metal deposited. This type of saving would be even greater in the case of frames if the flat portions between bends could be eliminated. However, it is in the reduction of rectification costs and fairing on the berth that the biggest savings should come. No costs have been published for these.

At present one of the bottlenecks in ship production is the need to obtain a great amount of offset and other data from the mould loft to enable material ordering and drawings to be started. Great speed is not possible with manual drawing methods and development must be along the line of extended use of mathematical methods of defining the hull shape. Frame lengths can then be calculated with great precision and through use of the memory capacity of a computer they can be ordered in the most economic lengths. Thus 'tween deck frames which may be of the order of 10 ft. long could be ordered in approximately 50 ft. lengths, thereby reducing the handling cost and obviating the need to order an extra length on each frame to allow for the end scrap caused by the unbent portion. Such reduction in actual frame numbers will lead to a reduction in the area of the stockyard and simplification of ordering and stock control. Proper planning should ensure that the minimum capital is employed in the stockyard. It might be argued that these savings could be realized now with manual frame forming methods, but the incentive to go to this degree of computer planning is greatly reduced when one has to draw the numerical data full scale and produce a template. Normally it only leads to a fresh incidence of cost with no attendant benefit elsewhere.

After the stockyard the frames pass through the shot blasting unit, followed by the painting unit. These machines operate at a speed of about 12–18 ft./min. and since this is much faster than present bending equipment, a buffer area has to be formed between the painting unit and the bending machine. Such a buffer area will occupy approximately 1,000 $ft.^2$ of floor space and the frame has to be handled into and out of it, generally using overhead cranes.

After passing through the bending machine, the frame is transferred to the marking area where the necessary markings are painted or punched on to it using templates. It is then transferred to the bay where drilling, burning and punching operations are carried out before being passed to the prefabrication unit assembly bay. Between each one of these operations buffer areas have to be formed to overcome bottlenecks arising from the varying speeds of operation. Thus the floor area involved in the complete frame operations may be of the order of 12,000–15,000 $ft.^2$ and all of this area has to be served by overhead crane systems, since as many as six handling operations may be required.

It will have been noted that marking involves templates and, therefore, it must be included in the operation to be performed by a numerically controlled bending machine if the savings associated with the elimination of the mould loft are to be realized. This is a logical step from the point of view that any numerically controlled machine tool must involve an accurate measuring system and thus all operations involving precise measurement should be done with one machine where this is possible. The same argument applies to the drilling, burning and punching operations so that ideally one measuring system should control all these functions and they should be designed to operate at speeds compatible with one another.

If this could be achieved only two buffer areas would be required, one between the painting unit and the bending marking unit and the other between the latter and the fabrication shed. This would lead to savings in floor area of the order of 6,000 ft^2, savings in labour of two to four men on bending, marking, drilling, burning, etc., and savings in handling costs. The latter cannot be accurately assessed from published data, but Ref. (7) suggests that it may be as much as 50–70 per cent of the total labour cost. A reduction in the handling operations could mean that overhead cranage could be eliminated completely through the use of conveyors and this will lead to much cheaper buildings.

Some of the frames need to be twisted so that the web is always normal to the shell plating (about 5 per cent depending on the ship shape and type). The accuracy required is not very high (± 2 deg.) and it is considered that at this stage this operation (bearing in mind the small number of frames involved) should be done by manual control and not included in the operations of an automatic bending machine.

The above discussion attempts to set out the savings which might be set against the cost of a machine which could perform all frame operations involving measurement. Various intermediate designs are possible, but it is considered that only the full development of automatic control can lead to appreciable savings and the retention of certain existing departments in order to maintain employment for the men at present engaged in them may lead to increased rather than reduced costs. The case for automatic machine tools must be judged against the continued upward trend in wage rates and the growing scarcity of skilled labour in the shipbuilding industry.

Design Requirements of Bending Machine

A bending machine must be capable of dealing with a wide range of frame sizes and profiles. It is assumed that every frame will only have one flange, i.e. only welded ship construction. Frame depths range from 5 to 21 in. and in profile can be angles, bulb plate, offset bulb plates or Tee bars. The length of individual frames may exceed 50 ft. in future, e.g. on tankers of the order of 100,000 tons deadweight or more.

The shape of the frame will be specified as a series of coordinates at discrete intervals on the outer edge of the frame with reference to rectangular axes (see Fig. 2—x, y axes). Ideally, these coordinates, together with the correct slope and curvature at each point, should be reproduced exactly. In practice at present errors of as much as 1·5 in. in the Y-coordinate for a X of 20 ft. (frame length about 30 ft.), see Fig. 2, have been measured and in many cases the bending operations result in a series of flats connected by short curved lengths. These errors are corrected by elastically deforming the frame (in some cases the ship just becomes wider or narrower than it should be) and by depositing excess weld metal in way of the flat portions.

Calculations, based on the errors in feed, slope, etc. (see Appendix I) suggested that a maximum error of $\pm 0 \cdot 25$ in. in the Y-coordinate for a radial distance of 35 ft. (frame length about 40 ft.) was a practical figure to aim at. It is believed that the positions where bending actions must be applied (i.e. feed

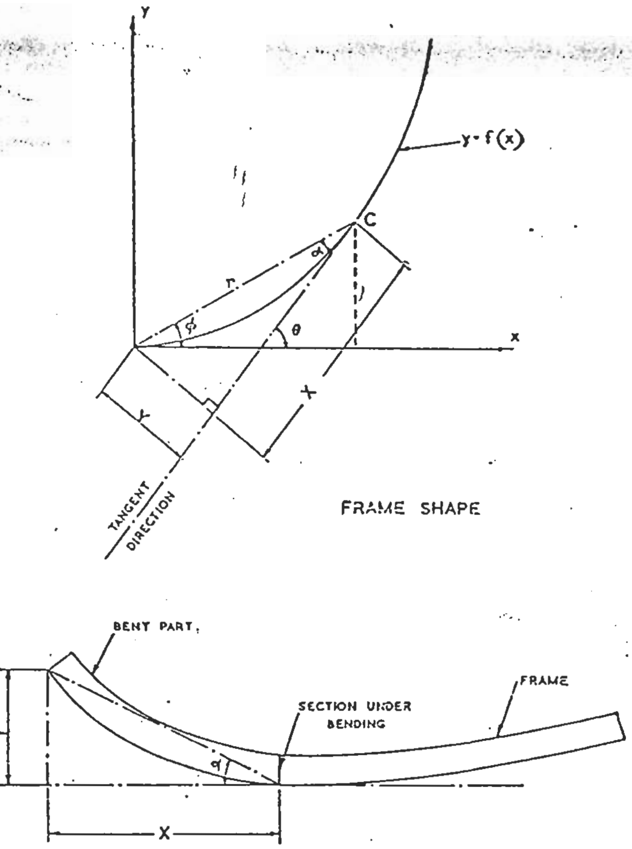


FIG. 2.—LOCUS OF END POINT

distances) can be computed so that the bent frame may have a continuous smooth curvature and the flats found under the present system largely eliminated. The length of the frame along its neutral axis must be correct to within $\pm 0{\cdot}25$ in. over a length of 40 ft. The coordinates of all marking operations must be accurate to within $\pm 0{\cdot}05$ in.

It is assumed that the new machine must have a productivity rate at least comparable with existing practice. In Britain this is, broadly speaking, eight pairs of large frames (about 30 ft. long) per 8-hour day, i.e. one pair of frames per hour. In the new machine it will be necessary, for reasons of accuracy (to be explained later), to bend each frame separately and the rate of the new machine will be one frame per 30 min. Due to the need to produce continuous curvature it is expected that as the frame depth decreases more bending operations per frame will be carried out. If it is assumed that the handling time will be 4 min. per frame and that for a 30 ft. frame 10 in. deep 30 bending operations will be required, then the time per operation, i.e. feeding through and applying the bending operation repeatedly until the right shape is produced, will be approximately 50sec. This figure will be increased when deep frames are considered since the feed length will be relatively high and curvatures will be less.

The above argument does not take account of the time spent, at present, on marking, burning and drilling together with the associated frame handling. These vary greatly from yard to yard, but this time will not be less than 30 min. per frame, i.e. it would double the time allowable if these operations are performed in conjunction with bending operations. However, since most of this time is taken up in handling it would seem best to aim at an operation time of around 50 sec.

A large number of frames, about 50 per cent, will not require bending, but are subjected to the other operations. Therefore the drive mechanism of the machine must be designed to provide different speeds depending on whether bending is required or not. This must be programmed in the control data of the machine. It must be emphasized that frames and longitudinals should go through the machine so that even if they require no bending the marking and other operations are carried out. Provision must be made for straightening frames which have been bent through bad handling, etc.

When not bending the feeding speed will be governed by the speed of the shot blasting machine (about 16 ft./min.) and marking operations should not require interruption of the feed process.

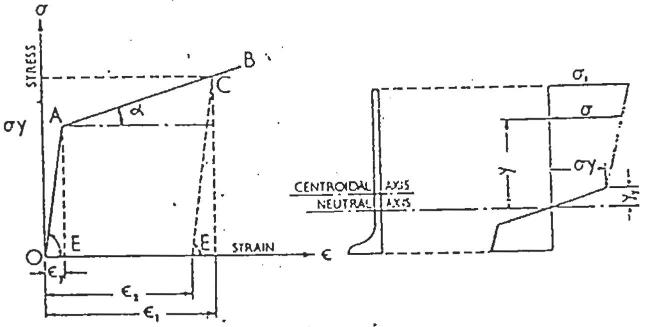
All operations of this machine should originate from signals from a control tape (paper tape in preference to magnetic tape) with the exception of certain handling features and possibly drilling, burning operations, etc., as mentioned earlier. The machine should be designed for continuous operation. This implies that the control must be integrated with stockyard control of frame sections and might require some dimensioning device to ensure that the correct frame has been withdrawn from stock.

The measuring system and control equipment must be of a robust nature which will not be affected by the conditions in steel working shops.

Choice of Control System

In existing numerically controlled machine tools the accuracy of the workpiece is achieved by precise positioning of the cutting tool or burner. However, in forming operations there will always be a tendency for the material to spring back after the load is released and the shape produced after spring back must be measured. If it is incorrect, further bending must be applied until, by an iterative process, an acceptable shape is achieved. There is thus a fundamental difference between the two types of machine tools.

In order to achieve permanent deformation of the frame it must be bent into the plastic range of the material (see Fig. 3). The amount of spring back, which is an elastic recovery, depends on the yield stress and the shape of the stress/strain diagram. There is a wide scatter in the yield stress for shipyard grades of mild steel (12–20 tons/in.^{2}) and also in the shape of the stress/strain diagrams. It is thus impossible to calculate the spring back for a given beam with any degree of precision.



(a) Modified stress-strain diagram

(b) Stress distribution

FIG. 3

Tests were conducted on small specimens in the Engineering laboratories at Glasgow University and on ship frames in a local shipyard to determine the magnitude and scatter of spring back. Fig. 4 gives typical results for the relationships between the central deflection over a span b before and after spring back. It can be seen that they all fall in a fairly broad band which is increased in breadth if there exists any residual stress

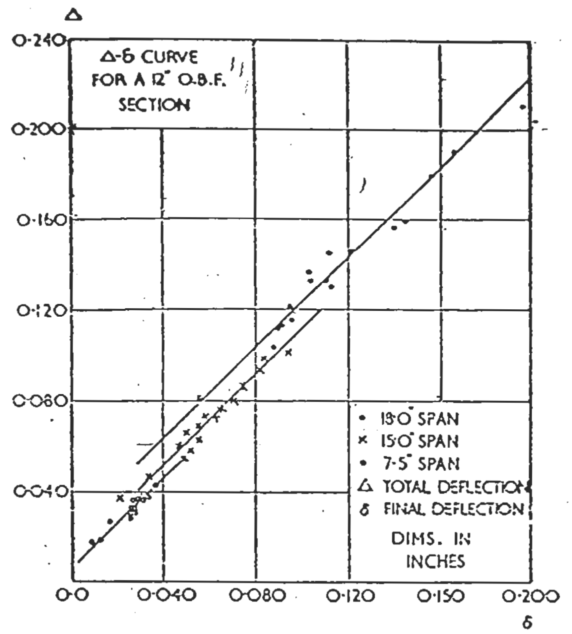


FIG. 4.—SPRING BACK DATA

in the frame. Some of the data were obtained from a 12 in. O.B.F. section which had been cut from a 15 in. O.B.F. section by burning, and it was clear that the resulting residual stresses caused a considerable variation in spring back. It was noted that when frames are bent back to back (port and starboard) they rarely had the same shape when bending was completed, even though they were both initially straight and were clamped together. Differences of as much as 1·5 in. in an offset for a frame 30 ft. long were measured, and only rarely were these differences corrected before the frames passed to the fabrication shop.

For this reason it was clear that if an accurate result was to be obtained the frames must be bent one at a time. The experimental data for springback indicated that it was possible to calculate the amount of bend required so that the frame would spring back to approximately the correct shape. When such a machine is used in practice it will be important to continuously monitor the spring back so that this initial calculation can be made with the maximum probability of success and thus will cut down the number of subsequent hunting operations, i.e. the machine will be continuously learning from its own operations.

The variable nature of the mechanical properties of ship frames also makes it impossible to calculate the force required to produce a given deformation, so that the method of control must rest entirely on measurement of the shape achieved after each bending operation.

This can be done in a number of ways:—measurement of curvature, central deflection over a fixed base length, slope of the portions outside the plastic zone and the locus of the end of the beam on the bent side (see Fig. 2). The first three methods are attractive from the point of view that all the measurements would be made close to the point of application of the load and would involve a compact piece of apparatus. It was found from laboratory and full-scale tests that the shape and extent of the curved part (plastic region) at each bend depended on the properties of the steel and could not be predicted. This caused small errors in curvature which tended to accumulate and caused unacceptable errors in the over-all frame shape. These errors also affect the computation of central deflection and slope. A further difficulty arises due to the influence of the curved portion from the previous bend on the shape of the plastic zone on the bent side of the central ram. These sources of error were disclosed by laboratory tests, but full-scale tests showed that even with shot-blasted frames, the degree of roughness of the surface and the variation in rolling tolerance in frame depth added further errors which made these methods unacceptable.

It was concluded that the only acceptable method was to ensure that the locus of the end of the frame on the bent side and its bent length were correct at each point of bending. The locus must be measured relative to an axis along the tangent to

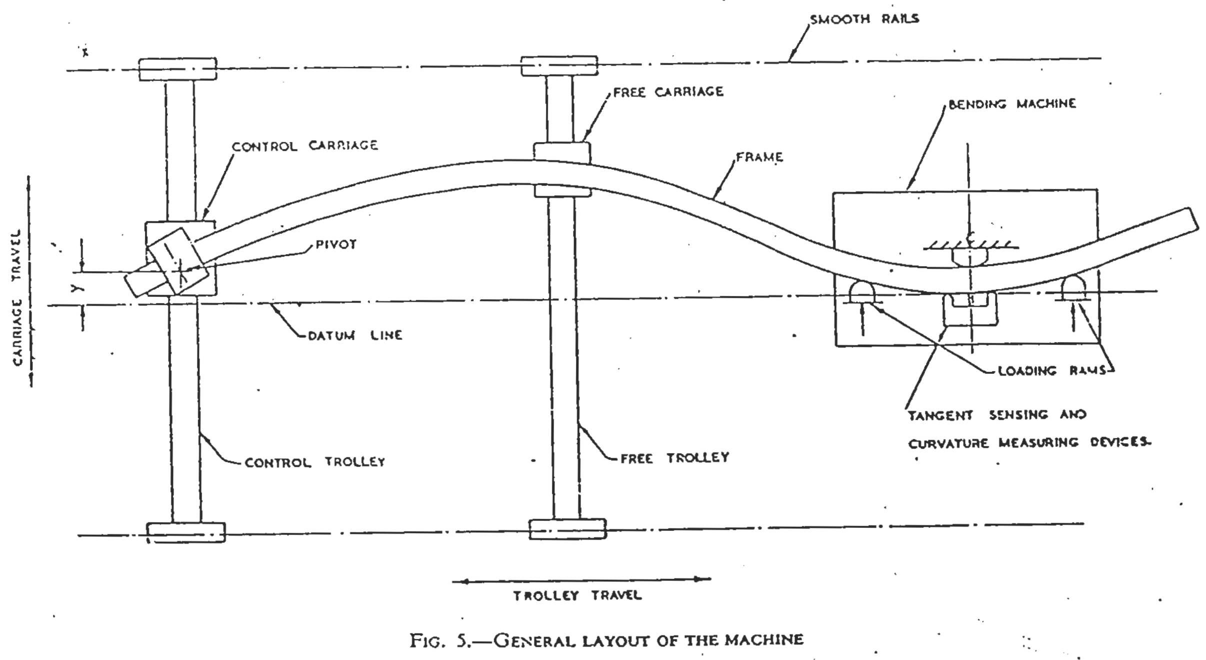


FIG. 5.—GENERAL LAYOUT OF THE MACHINE

the frame at the point of application of the central load. (See Fig. 2 and Appendix 1 for the calculation of this quantity.) Thus the control system would ensure the geometrical accuracy of the frame and rely in no way on the properties of the material. Unfortunately this involves measurements over considerable distances (up to 60 ft.) in length and ± 20 ft. in the lateral direction (see Fig. 5). These distances are based on a study of a 100,000 tons deadweight tanker.

To carry out the measurements it was suggested that the end of the frame be supported on a carriage which was free to travel laterally along a beam and which itself would be able to travel freely in the direction of feed through the machine (see Fig. 5).

The motion of these two carriages measures the end locus. Obviously friction in this trolley system must be kept to a minimum if the frame is not to be distorted. Measurements were carried out in a shipyard which had a carriage system of similar design and it was concluded that this was quite practicable—in fact the friction forces could have been doubled in this instance without causing appreciable errors. A discussion of all these errors is given in Appendix (I).

An alternative system of locus measurement was suggested by Mr. Taylor of Messrs. Swan Hunter and Wigham Richardson Ltd., whereby the end of the frame would move along a straight line and the bending machine rotate until the correct locus with respect to the tangent at the point of bending had been achieved. This does involve some reduction in floor area and might be desirable where this quantity is limited, but it involves an increased cost in the machine control system.

Buckling Problems

In the early stages of the investigations buckling of the web when undergoing compression was mentioned by both the manufacturers and shipyards as one of the chief problems in cold-frame bending. This only arises in the bow and stern regions of the ship where the flange of the section is in tension and the free edge of the web is in compression.

Existing designs of cold-frame benders all attempt to deal with this problem by applying high lateral clamping pressure to the web, a force of approximately 78 tons spread over a 12 in. diameter ram being employed on a 400-ton capacity bending machine. Even with this force, buckling is still experienced and it is overcome by

(1) Carrying out the three point bending operation at very small distances along the frame (3–4 in. against the normal 9–14 in.);

or by

(2) Passing the frame two or three times through the frame bender using the normal feed distances (9–14 in.), but reducing the radius of curvature through each pass.

In a numerically controlled frame-bending machine, it is believed that, for ease of control, the feed distance should be kept constant along any one frame, although it may differ from one frame to another depending on the frame size and the required curvature. Further, since all the control data will be obtained from the equation to the frame shape, the frame can only be bent by a single pass through the machine.

Since the problem was of such vital importance to the successful operation of a numerically controlled machine a theoretical study of the plastic deformation of an element in the plastic compression zone was made. The element is subjected to tangential stresses due to bending, radial stresses and transverse stresses due to clamping, as shown in Fig. 6.

If we consider that the frame material follows the ideal plastic stress strain diagram, see Fig. 6, in the plastic zone, it follows that the stress system acting on an element in this zone would

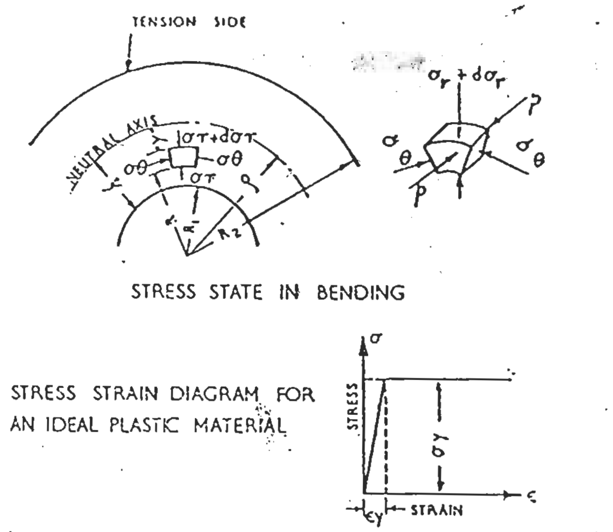


Fig. 6

be the principal stresses. However, even if we consider the actual case, when strain hardening is taken into consideration, the effect of the shear stresses on the stress system, in both magnitude and direction, will be small and can be neglected so that these stresses can be considered as the principal stresses. The error resulting from this approximation decreases as we go further from the neutral axis and becomes zero at the extreme fibres where the actual value of the shear stress is zero (since the reaction force is applied to the flanges of the frame section). Since the buckling zone is remote from the neutral axis, the previous approximation to the stress system is justified.

Under this triple compression, it is difficult for an element to deform, and permanent deformations of the element would require certain stress conditions for yielding to start. In order to find these conditions, the Von-Mises yield criterion was used. It was shown, see Appendix II, that the existence of the clamping pressure p, in effect, reduces the formability of the material in the compression zone by preventing the plastic flow from taking place in the lateral direction. Also, its existence would generate frictional forces between the clamping head and the surface of the frame. These frictional forces would oppose the plastic flow in the tangential direction. Their values depend on the clamping pressure and the coefficient of friction between the two surfaces.

From Appendix II it can be seen that the part under compressive bending stresses will increase in thickness, having zero change at the neutral axis and maximum change at the outermost fibres. This increase is of the order of 0·03–0·06 in. for a 17 in. O.B.F. section. Due to this increase in thickness, the clamping pressure p will increase and will attain its maximum value at the outer edge of the frame. The pressure distribution is very complex and is difficult to determine precisely.

From the previous discussion it is obvious that, due to all these factors, clamping leads to a reduction of formability of the compression side under the clamping head and thereafter the initiation of buckling zones just outside the clamped area where instability is liable to take place.

Therefore, to achieve the best conditions for the material to flow and form the required shape without any buckling taking place, the clamping pressure should be reduced to zero. This would improve the plastic flow in both directions, laterally and tangentially.

In order to check these theoretical conclusions tests were carried out on a standard 400 ton cold-frame bender using an

8 in. O.B.F. section, 0·4 in. thick. It was found that using the full clamping pressure over a 12 in. diameter ram (56 in.^{2} contact area) the frame could be bent to an arc of a circle of 20 ft. radius before buckling commenced. When a thick plate was inserted under the ram to spread the full clamping pressure over a rectangular area 26 in. × 6 in. (156 in.^{2} contact area), the frame could be bent to a radius of 10 ft. before buckling commenced. When no pressure was applied and the ram was merely used to prevent any lateral distortion in excess of the normal thickening of the section, the frame could be bent to a radius of approximately 5 ft.

It is thus clear that in future designs of these machines no clamping pressure should be applied, but a ram should be positioned over the web surface for at least half of the machine span. Ideally it should be positioned so as to allow the increase in thickness induced by the plastic bending process. It will then be possible to use equal feed distances and to bend the frame with a single pass through the machine.

The Feed Through the Machine

The feed is defined as the distance between two successive loading points.

Theoretically, the feed should depend on the dimensions of the frame, the curvature and the precise stress-strain curve obtained at the loading point, i.e. it depends on the extent of the plastic zone. Basically it is desirable to feed the frame through so that the summation of curvature, produced at each point by successive bends, adds up to the desired value. This is represented graphically in Fig. 7. The calculation is complex and the result depends entirely on the physical properties of the steel. In this respect, the optimum feed is defined as the distance between two successive loading points which will not leave straight portions or cause excessive overlapping of two successive plastic zones. Theoretically, the optimum feed could be determined using the simple theory of plastic bending, which is based on an ideal stress strain diagram. However, since strain

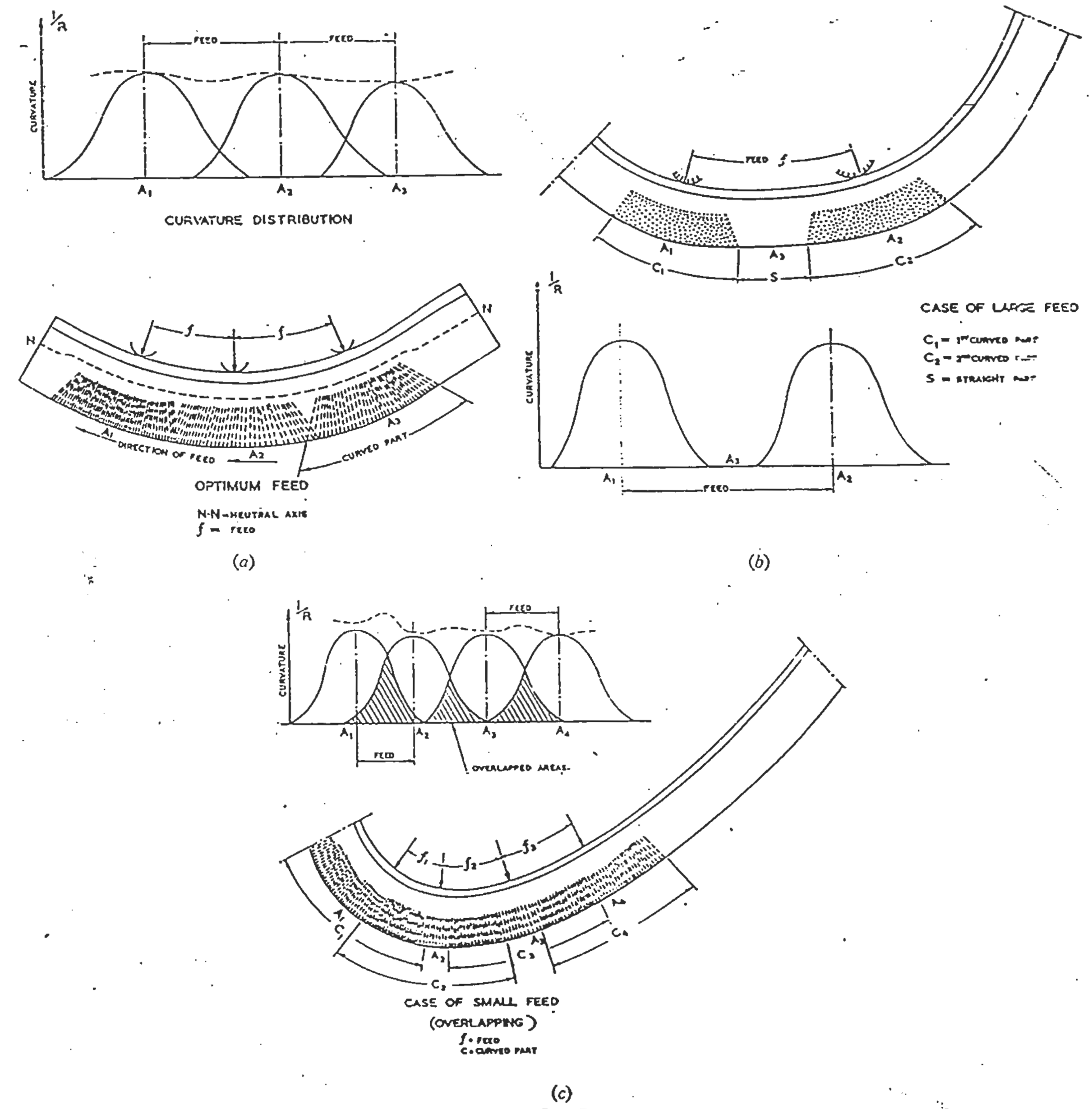


FIG. 7

hardening is inevitable, in the case of rolled steel sections, these theoretical values of the optimum feeds are not strictly correct. The existence of strain hardening tends to increase the length of the plastic zone and thus increase the corresponding amount of feed.

The optimum feed for a rectangular section, as calculated according to the simple theory of plastic bending is equal to span/6. For an O.B.F. section, 12 in. deep and 0·5 in. thickness (shape factor $\alpha = 1\cdot 710$), the optimum feed $= 0\cdot 207\,L = 1\cdot 035$ ft., for a 5·0 ft. machine span.

A number of laboratory and full-scale tests were carried out and it was found that the optimum feed is in the range of d to $1\cdot 5\,d$ (where d is frame depth for O.B.F. section) depending on the curvature and frame depth.

This amount of feed must be taken on the neutral axis of the frame since, by definition, it does not change in length. The neutral axis referred to is not the centroidal axis used in elastic bending, but the neutral axis when the fully plastic state is attained. This quantity could easily be computed for each frame and will be constant during bending. However, to calculate the exact location of the neutral axis, the actual strain hardening of the material should be taken into account. The calculations involved are complex and the increase in accuracy would be of the order of the manufacturing tolerances on the frame so it is not considered that these refinements should be applied. Further, due to bending in the plastic range, the cross-section dimensions will vary across the depth (a reduction occurring in the tension side and an increase in the compression side), see Appendix II. This will certainly affect the position of the neutral axis.

The total number of operations is governed by the size of the feed and, therefore, it is essential to increase the feed length to the maximum tolerable from the point of view of welding cost at the flat portions. It is anticipated that experience with the new machine will establish the optimum feed distance for each frame size. Once this distance has been determined, it is vital that the bending machine be capable of feeding the frame through by the correct amount to within a tolerance of approximately $\pm 0\cdot 2$ per cent of the feed distance. All subsequent calculations and control data are computed for these loading points, see Appendix I, and it is essential that they be correctly obtained.

Some Features of the Proposed Design of Machine

It was pointed out earlier that the best medium to control the bending process is the locus of the end of the frame with respect to the tangent at the point of application of the load. In order to make possible the application of this control system the bending machine must have the following features:—

1. *Ram Arrangement.*—The machine must have a central fixed support and two independently moving side rams so that the tangent at the point of loading can always be brought into a given plane. This would be impossible with either of the existing designs of machine, since in the case of the fixed side arms—movable central ram the tangent position at each bend is influenced by the curvature at the preceding bend. In the other machine the side arms are controlled by a single ram so as to have equal movements at all times and again the tangent direction is variable due to the effect of the previous bends.

The alignment of the frame can be achieved by a tangent sensing device which operates on both rams in such a way as to keep the tangent in the desired direction. This device need consist only of a linear transducer which measures the relative displacement from a fixed plane of two points on the frame equally spaced from the central ram and at a short distance (6·9 in.) away from it. An error in the tangent direction of the order of one minute will lead to an error of 0·1047 in. at the end of a frame 30 ft. long.

Further, this design will facilitate feeding the bent frame through the machine, since the position of each ram can be adjusted separately to suit the shape of the bent frame.

2. *Control of Buckling.*—Before bending is started very low clamping pressure (about 20 lbs./in^2) should be applied so that the web may be flattened and any initial buckling should be removed. After this the clamping device should be used only to prevent lateral distortion and must be designed to withstand such forces.

3. *Side Clamps.*—These should be designed so that they allow any frame shape (O.B.F. section, Tee section, etc.) to be fed through the machine without fouling. Further, ample clearance should be provided in these side clamps to allow for the variation in flange thickness and for spring back after unloading.

For automatic operation of the bending machine the frames are liable to foul these side clamps, especially on the bent side or when feeding is in progress. It was found difficult to calculate for each frame shape the exact position of the side ram so that fouling would not occur. However, this problem could be overcome by a transducer which allows a certain gap between the flange of the frame and the slot in the side clamp and operates a servo system to maintain this gap.

4. *Power.*—The bending force will be supplied by two hydraulic rams in order to ensure the smooth running and safety of the machine.

The bending force per ram could be calculated from the assumption that it must at least exert a bending moment, at midspan, of the order of the fully plastic moment for the deepest section to be bent as calculated from the simple theory of plastic bending.

It was found that a 100 tons capacity ram on each side of the central fixed support will be sufficient to bend ship frames up to 17·0 in. O.B.F. section and 21·0 in. × 1·5 in. rectangular section, since these deep sections are normally bent to small curvatures.

5. *Machine Span.*—The choice of a suitable machine span depends on the following factors:—

(*a*) Range of frame sizes (between 5 in. and 21 in.).
(*b*) Amount of scrap left at each end.
(*c*) Effect of the concentrated load on the surface of the frame at the three loading points.

As the machine span increases, the amount of scrap at both ends of the frame will increase. Further, feeding the frame through the machine will become a difficult problem since the bent length will be increased. On the other hand, as the span is reduced the amount of scrap will be reduced, but severe indentations will take place at the loading points due to the high bearing pressure applied.

However, in order to form a beam by the 3-point method, the span/depth ratio should not be less than 3·0, and since the maximum frame depth required to be bent on this machine is 21 in. a machine span of the order of 60 in. will give a reasonable compromise for all practical sizes of ship frames. This figure i.e. 60 in. is, in fact, the general practice adopted at present in the design of these machines.

In some machines, the span is made variable so that for every frame section there is a corresponding span. This is a desirable feature since it reduces the end scrap, particularly for small sections. However, it was found that the complications involved in making this process automatic (i.e. the span varies for every frame size) is not justifiable since, in any case, a certain amount of scrap is inevitable.

6. *Method of Control*

(*a*) *The Feed Through the Machine.*—Since the amount of feed will have to be measured on the neutral axis of the frame, it is believed that a feeding mechanism, possibly feed rollers, working on the unbent side only (which is assumed to be straight) would give the best solution having regard to the machine operation and feed distance measurements. The accuracy achieved by this system is expected to be adequate, although it may be necessary to incorporate a transducer to check the feed and to correct it if in error.

(*b*) *Locus Measurement.*—It is proposed that the ordinate be measured as indicated diagramatically in Fig. 5. The end of the frame on the bent side will be supported on a carriage which can move freely transversely on a trolley which itself moves longitudinally parallel to the plane of the tangent at the central loading point. Both carriage and trolley must be designed for minimum friction. The maximum transverse force should not exceed 25 lb and the maximum longitudinal force 50 lb. These figures are based on a 21 in. × 1·5 in. rectangular section and should be reduced for lighter sections.

Further, it is important that the edge of the frame at the locus point should pivot about the measuring point, and Fig. 8 shows

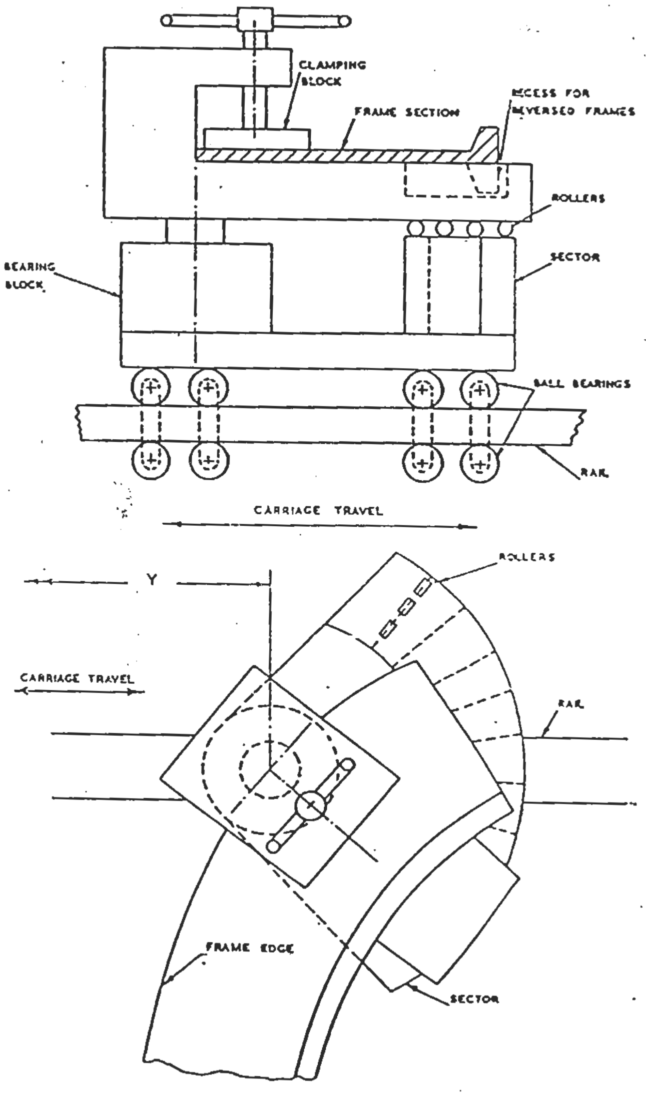


Fig. 8.—Carriage arrangement

one design of clamping arrangement ... actual measurement of the transverse position of the trolley can be achieved by several methods and the required accuracy of ±0·05 per cent (for a frame length of 40 ft., the accuracy is ±0·24 in.) is less than many existing machine tool systems so that no difficulty is anticipated.

To reduce sag of the frame, on the bent side, a second trolley system may be required, see Fig. 5, but it would be solely for support purposes and no measurement would be involved.

An alternative measuring system could be based on the angle made by the radial line to the frame end point and the tangent, as shown in Fig. 2. The final choice of system must rest with the firm designing the control equipment.

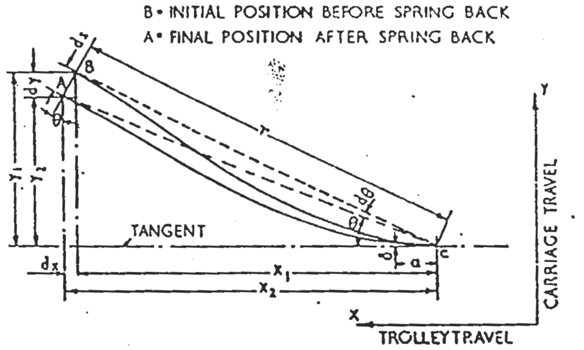


Fig. 9.—Spring back at the free end

Initially the frame will be overbent by an amount calculated (see Appendix 1) to overcome spring back. It will then be released and once spring back has taken place and the tangent direction has been adjusted, the ordinate will be measured and compared with the computed value for the correct end position. If the error exceeds ±0·05 per cent (minimum value is $\frac{1}{8}$ in.), the frame will be rebent in the appropriate direction, see Fig. 9, by an amount sufficient to correct the error and the checking process is repeated until the end locus is within the acceptable limit. This error correction process can be purely a "hunting" operation or the appropriate correction can be computed each time using the formula given in Appendix (1). Some experimental data will be necessary to determine which of these is likely to be the cheaper solution.

From the full scale test data, the spring back in curvature is of the order of 0·01–0·035 in. through a span of 15 in. Therefore, for a frame 40 ft. long the end point would spring back by an amount of the order of 1·0–2·0 in. Consequently the hunting mechanism must cover a range of about 3·0 in.

(*c*) *Curvature at the Central Point.*—Since it will not be possible to mount the frame end on the trolley until it has cleared the bending machine, it is necessary, in order to avoid excessive scrap, for the control for the first one or two bends to be carried out using curvature measurements as the control medium.

The imposed curvature, applied before spring back, could be calculated from the desired value after taking account of spring back. When the initial curvature is reached, the frame is released and the curvature measured again. If the measured value differs from the computed value, an error signal will be sent to the hunting device and the operation repeated until the error is within acceptable limits (±2 thousand through a 10 in. span).

Experimental Verification of the Proposed Design

In order to examine the feasibility of the above design a model which incorporated most of the features mentioned above was built in the University workshops (Figs. 10, 11). It can bend flat bars up to 4 in. × $\frac{1}{2}$ in. using hand operated hydraulic jacks of 10 tons capacity. The tangent sensing device consisted of two arms placed 1·5 in. on either side of the central ram (see Fig. 10), the rotation of the arms being measured on a 6 in.-diameter protractor. This limited the accuracy with which the tangent could be aligned to ±0·25 deg. The locus was recorded by the movement of the end of the bar over graph paper.

A number of bars were bent to arcs of circles using a range of feed distances. Typical results for a 3 in. × 0·375 in. beam bent to a radius of 30 in. are given in Table I, where it can be seen that the variation in radius is of the order of ±0·30 per cent. Bearing in mind the limitations in accuracy of the equipment this is considered satisfactory.

At the request of the British Ship Research Association full-scale tests were carried out using a conventional frame-bending machine. Since the tangent cannot be aligned along a given datum in such a machine a tangent sensing device was designed such that the locus could be measured relative to the actual position of the tangent at each bend, see Fig. 12. In this case the locus was measured in radial coordinates (angle α, radius r), a Wild theodolite being used to measure the angle of the end of the frame to the tangent at the point of loading. The frame shapes selected for the test were a bow and a stern frame for a 100,000-ton deadweight tanker and a frame of S shape made up of a circular arc of 8 ft. radius followed by a straight portion to allow the frame to pass through the machine and then a bend of 6 ft. radius. An 8 in. O.B.F. section was used for the latter test, the free edge of the web being in compression during the 8-ft. bend and a 12 in. O.B.F. section for the ship frames.

In these tests the approximate angle to which the frame had to be bent, so as to spring back to the correct position, was calculated for a range of feed distances using a computer programme developed by Mr. Wellman of the B.S.R.A. staff using the method given in Appendix I. The frame was bent to this angle and the locus position after spring back measured and compared with the required value to ascertain the extent and direction of further bending. This was repeated until the locus

Fig. 10.—A model for bending rectangular sections

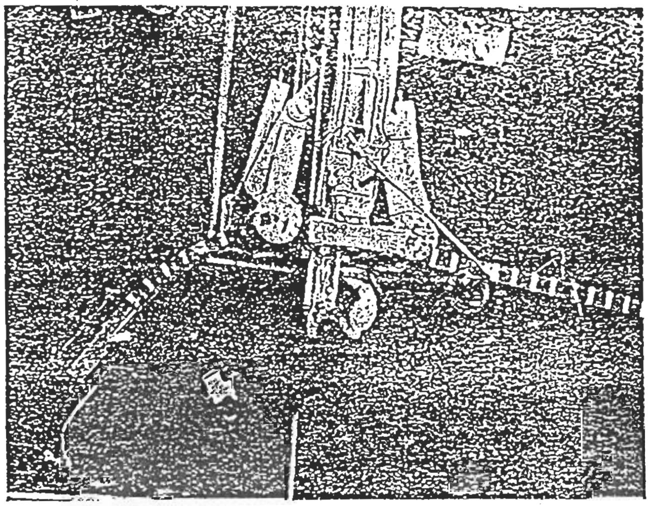


Fig. 12.—An aerial view of the full-scale tests

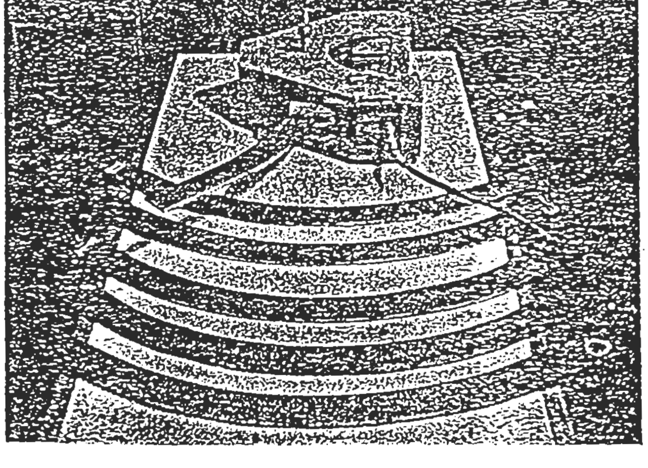


Fig. 11.—The bending model and test beams

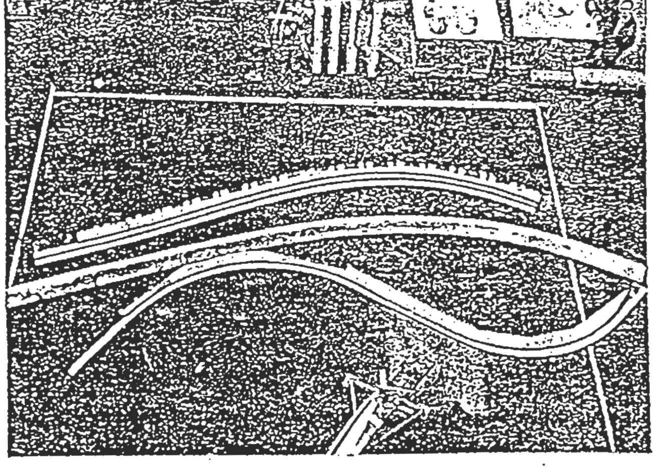


Fig. 13.—Repeatability of the bending process

was within ±3 minutes of the required angle. The shape of the ship frames was checked against templates prepared by Messrs. Swan Hunter and Wigham Richardson Ltd.

These tests disclosed the importance of the side rams applying their loads at equal distances on either side of the central ram, since unsymmetrical distribution of the load causes errors in the tangent measuring system. After initial troubles due to this cause had been overcome, it was found that, even with this relatively crude measuring system, the frame could be produced to within the accuracy of the templates and was well within present accuracies accepted in shipyards. The maximum deviation of the frame, consisting of two circular arcs from the desired shape, was ±0·1 in. and the average deviation was of the order of ±0·05 in.

Several beams were bent to the shape of one of the frame lines (see Fig. 13) to ensure that the same degree of accuracy could be reproduced.

As a result of these tests it was considered that if the accuracy of measurement asked for in the prototype machine could be achieved in practice these frames could be produced by this method to within the accuracy specified earlier.

TABLE I

(Feed = 3·50 in.)

Loading point	1	2	3	4	5	6	7	8
Radius ..	29·95	29·42	30·7	29·71	30·3	30·3	30·43	29·91

Marking of Ship Frames

As pointed out earlier, marking must be carried out by a numerical control process if templates are to be eliminated and the accuracy of bending maintained.

It is believed that the following markings will cover most of the British shipyards' requirements:

1. Ship No. 4 digits
2. Frame No. 3 digits
3. Top or Bottom T or B
4. Port or Starboard P or S.
5. Waterline marks (including the bottom waterline), at least one waterline per plate.
6. Distance from top of frame to nearest waterline.
7. Erection holes.
8. Notches for shell seam.

Numbers 1, 2, 3, 4, and 6 could be made in digital form and stamped on one face of the frame at its lowest part. Number 3 could be eliminated if the marks are always placed on the same end of frame.

Numbers 5, 7, and 8 could be indicated by an arrow head and should be made on both sides of the frame.

The marking unit should be placed beside the bending machine on the unbent side of the frames. As the frame passes through the unit, markings will be stamped on at the required positions.

Since the feed for each frame will be kept constant and the frame will always be fed through parallel to the datum line, the marking positions, in the longitudinal direction, could be calculated with reference to the neutral axis and the loading points. Transverse positions will be referred to the outer edge of the frame.

The marking head should be capable of scanning an area 2·5 ft. by 2·5 ft., longitudinally and transversely to cover any amount of feed and also any frame size.

If an x–y control system is used for the marking device, the same control unit could operate a numerically controlled burner. Alternatively, the feed to the bending machine could position the frame longitudinally and only a transverse carriage need be provided for the marking device.

Effect of Residual Stresses

So far we have been endeavouring to ensure that the frame has an acceptable degree of accuracy when it enters the prefabrication shed. However, if it has been bent into shape in the cold state it will contain residual stresses, in addition to any it may have possessed when it entered the shipyard and any subsequent work on the frame such as welding, punching, etc., is liable to release these stresses and cause further deformation of the frame.

Appendix III gives an approximate calculation of the residual stresses from which it can be seen that in the region of the bilge radius they may reach the order of ±10 ton/in.^{2}. Furthermore, when the frame is welded to plating it is possible that the spring back at the end, due to the release of these residual stresses, may be of the order of ±0·3 in. in a 12 in. frame 30 ft. long. This calculation is necessarily approximate since the precise effect of the welding on the residual stresses is a function of many factors, not least of which is the behaviour of the welder himself and is therefore not exactly predictable.

It is clear, however, that this may cause serious problems and it will have to be studied carefully when numerically controlled frames are produced. Theoretically it might be possible to calculate an over or under bend allowance which would take account of this residual stress effect, but such a method must involve empirical factors which could only be obtained from full-scale tests of considerable complexity.

Conclusions

1. The availability of methods of numerically defining the hull shape using computers has replaced the work of fairing in the mould loft. However, this department has to be retained to provide templates for frame and plate bending.

2. These templates are of limited accuracy and present methods of bending frames and plates lead to errors in individual parts of as much as $\pm 1\frac{1}{2}$ in. in a 40·0 ft. frame. These errors are reduced by elastic deformation and by excess deposit of weld metal which result in considerable internal stresses. They also give rise to considerable rectification costs through loss of time.

3. These practical difficulties could be avoided if frame bending and plate bending were numerically performed to within a tolerance of $\pm\frac{1}{4}$ in. for a frame of 40·0 ft. length.

The mould loft as used at present could be dispensed with completely, giving a saving of up to six men and a floor space of between 2,500 and 20,000 sq. ft., depending on the scale of lofting.

4. The operations carried out on the frame (shotblasting, painting, bending, marking, drilling, etc.) require 6 to 8 men, entail the frame being handled by overhead cranage as many as six times and necessitate buffer areas between the different operations. The total area involved may be as high as 12,000 to 15,000 sq. ft. in a medium size shipyard (excluding the mould loft area).

5. The provision of a numerically controlled machine tool requires a precise measuring system and it is therefore logical to carry out all operations involving measurements on one numerically controlled machine. Thus in the case of frame operations, bending, marking and punching, drilling and burning should ideally be associated with the same measuring system. This would obviate the need for much of the handling and give a greatly reduced floor space.

Since all frames and longitudinals require some of these operations, it is assumed that they will all pass through the same machine. Frames or longitudinals which are intended to be straight will be straightened by the bending machine, if they were initially bent.

6. A machine tool control system for forming materials which possess spring back (elastic recovery) characteristics must include a servo system to measure the error at the end of each operation and decide on the magnitude and direction of the correction if it is required. This is due to the unpredictable nature of spring back in materials in the as-delivered condition (particularly shipyard quality mild steel). To obtain accurate results and repeatability, frames must be bent one at a time and not port and starboard as at present.

7. Further, due to the variation in physical properties of mild steel (approximately ± 20 per cent variation in the yield strength), it is not possible from the stress strain diagram to predict the force required to achieve a precise deflection and a control system must be based on the measurement of the shape achieved.

8. The best method of controlling the frame shape was found to be the measurement of the locus of the end on the bent side relative to the tangent through the point of bending, having regard to the tolerance in the frame dimensions, the nature of the surface finish on the frame, the accuracies required from the transducer elements and the general design of the machine.

Since all measurements are made relative to the end point and not with respect to the previous bending point, this system ensures that there can be no accumulation of errors and frame marking, drilling, etc., can be carried out using the same measuring system.

9. This system can be applied to a three-point bending machine of conventional design, but it is much easier if the side rams are separately controlled.

10. In such machines, buckling of the compression web of the frame is best prevented by a large guiding area covering the web depth and half the span at the point of application of load. Ideally no clamping pressure should be applied to the web since the material must be allowed to flow in the transverse and tangential directions under the action of the compression forces. Pressure must be available to resist buckling when it occurs and also to flatten initially buckled sections.

These modifications in the clamping system will make possible bending offset bulb flats (O.B.F.) having thickness/depth ratio from 0·04 to 0·06 into arcs of circles (the web under compression) given by depth/radius ratio from 0·12–0·15.

These conclusions (from 8 to 10) have been confirmed by tests on model scale and on full scale. These tests have confirmed the possibility of producing a numerically controlled frame-bending machine which can bend frames to within less than $\pm 0 \cdot 1$ in. of the correct offset at all points.

11. The three-point cold-bending process gives rise to residual stresses which may be of the order of $\pm 10 \cdot 0$ tons/in.^{2} for normal ship curvatures. During fabrication, these residual stresses cause distortions due to the release of part of these stresses by welding, punching, . . ., etc. Eventually it will be necessary to calculate the locus of the end point so that when the residual stresses have been released, the frame will have the correct shape, i.e. the frame will be in error when leaving the bending machine and will be very close to the correct shape after the subsequent operations it undergoes.

Experimental data on the effect of releasing part or all the residual stresses as a result of welding after bending will be required before such calculations can be performed.

12. The efficient use of a numerically controlled frame-bending machine requires a fairly elaborate computer programme which will carry out all frame calculations from the faired offsets, combine the frames into suitable lengths to be bent from a single section, provide the material order and stock control data and all the machine tool control data. Complete integration of frame work in this manner should result in considerable economic savings in shipyards.

Acknowledgments

This work was carried out in the Naval Architecture Department and Engineering Laboratories at Glasgow University under the direction of Professor J. F. C. Conn. The shipyard tests were carried out at the Scotstoun shipyard of Messrs. Charles Connell & Co. Ltd., who supplied all the material and labour. The British Ship Research Association paid for the full-scale tests at the works of Messrs. Hugh Smith Ltd., and assisted in the tests. We are grateful for the help of all these firms and for the advice of Mr. J. Usherwood, lately chief designer to Messrs. Hugh Smith.

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APPENDIX I

A. Locus of End Point

The position of the end point with respect to point C, at any bending operation, see Fig. 2, will be called the locus of the end point.

(a) *Locus in Radial Coordinates*

The end point position could be defined by:—

The radial distance "r" and

The angle between the radial line and the tangent at the point at midspan (i.e.)

$$r = \sqrt{x^2 + f(x)^2} \quad (1)$$

$$\alpha = \theta - \phi \quad (2)$$

where $$\theta = \tan^{-1} f'(x) \quad (3)$$

and $$\tan \phi = y/x = f(x)/x \quad (4)$$

giving $$\alpha = \tan^{-1} f'(x) - \tan^{-1} \frac{f(x)}{x} \quad (5)$$

(b) *Locus in Cartesian Coordinates*

The end point position could also be defined by the distance X along the tangent at midspan and the distance Y along the normal from the end point.

$$X = r\cos\alpha$$

$$= \sqrt{x^2 + [f(x)]^2}\cos\left[\tan^{-1} f'(x) - \tan^{-1}\frac{f(x)}{x}\right]$$

$$Y = \sqrt{x^2 + [f(x)]^2}\sin\left[\tan^{-1} f'(x) - \tan^{-1}\frac{f(x)}{x}\right]$$

(c) Locus of End Point with respect to a Point at Midspan of the Machine and when a Flat Portion is considered

The calculation of point 0 relative to the first loading point (i.e. point E) is based on the following assumptions, see Fig. 14.

(*a*) The shape of the frame edge at the first bend—after spring back—is a parabola.

(*b*) The curved length = 1·5d.

The equation to a parabola at point E, where the curvature is $1/R_E$ with respect to a set of rectangular axes, $\bar{x}, \bar{y}$ is as follows:

$$\bar{y} = \bar{x}^2/2\,R_E$$

The coordinates of point 0 relative to point E becomes:—

$$\bar{Y} = \left(3\cdot5 - \frac{0\cdot75\,d}{R_E}\right)\sin\bar{\theta} + \frac{(0\cdot75\,d)^2}{2\,R_E}$$

$$\bar{X} = 0\cdot75\,d + \left(3\cdot5 - \frac{0\cdot75\,d}{R_E}\right)\cos\bar{\theta}$$

where $\tan\bar{\theta} = d\,\bar{y}/d\,\bar{x}$.

The length of the flat portion allowed at the frame end = 3·5 ft.

Let
$$R_0 = \sqrt{\bar{Y}^2 + \bar{X}^2}$$
$$\tan\alpha_0 = \bar{Y}/\bar{X}$$
$$\tan\theta_E = \left(\frac{d\,y}{d\,x}\right)_E$$

From Fig. 14, the coordinates of point 0 are as follows:—

$$y_0 = y_E - R_0\sin(\theta_E - \alpha_0)$$
$$x_0 = x_E - R_0\cos(\theta_E - \alpha_0)$$

The position of point 0 relative to point P is defined by X_1, Y_1, as follows

$$X_1 = r_1\cos\alpha_1$$
$$Y_1 = r_1\sin\alpha_1$$

where:
$$r_1 = \sqrt{(y_p - y_o)^2 + (x_p - x_o)^2}$$
$$\alpha_1 = \tan^{-1} y' - \tan^{-1}\frac{y_p - y_o}{x_p - x_o}$$

B. Calculation of Spring Back at the End Point of the Frame

Due to spring back of the frame at the portion under bending, the end point will move on an arc of a circle of radius r, see Fig. 9. The spring back is calculated according to the following notation:—

B = Position of the end point before spring back.
A = Position of the end point after spring back.
$1/R_1$ and $1/R_2$ = curvature at "C" before and after spring back.
$d\theta$ = angular spring back.
dy = spring back of the end point measured on the normal to the Datum Line.
x = trolley travel.
y = carriage travel.

The spring back in curvature at point C

$$= 1/R_1 - 1/R_2 \quad \dots \quad (1)$$
$$= \left(\frac{2}{b^2}\right)\delta_1$$

giving
$$\delta_1 = \frac{b^2}{2}\left(\frac{1}{R_1} - \frac{1}{R_2}\right)$$

where δ_1 = spring back through a span $2b$
$2b$ = span of the curvature measuring device

The values of δ_1 could be supplied either as a graph or in tabular form as obtained from some full-scale tests.

Let δ_2 = spring back through a span $2a$

$$\delta_2 = \delta_1\left(\frac{2\,a}{2\,b}\right)^2 \quad \dots \quad (2)$$

Since $(2a)$ is small compared to r, see Fig. 9

$$d\theta = \delta_2/a \quad \dots \quad (3)$$
$$d\,S = r\,d\,\theta = r\,\delta_2/a$$
$$\cos\theta = \frac{X_2}{r} = d\,y/d\,s.$$

therefore
$$d\,y = \frac{X_2}{r}\,d\,s = \frac{\delta_2}{a}\,.\,X_2 \quad \dots \quad (4)$$

X_2 is given before in (A)

therefore
$$d\,y = \frac{\delta_2}{a}\,r\cos\left[\tan^{-1} y' - \tan^{-1}\frac{y}{x}\right] \quad \dots \quad (5)$$

$d\,y$ will be calculated at the values of x corresponding to the points of equal feed on the neutral axis.

C. Calculation of Points on the Frame Edge corresponding to the Points on the Neutral Axis, at Equal Distances

It is to be expected that the frame shape will be derived directly from an equation to the whole surface, but for the present it is assumed that the information will be supplied in the form of mould loft offsets.

These offsets will contain errors due to the misreading of scales, dimensional inaccuracy of mould loft floor and scales, etc., and undoubtedly a smoothing programme to eliminate these errors will be required. This can be done in a number of ways and it will be assumed that the frame shape can be expressed in the form of an equation

$$y = f(x) \quad \dots \quad (1)$$

where $f(x)$ will probably be some form of polynomial. Once this equation has been obtained, it is then a simple matter to evaluate.

The slope $\quad d\,y/d\,x = f'(x) \quad \dots \quad (2)$

The rate of change of slope $\quad \frac{d^2 y}{d\,x^2} = f''(x) \quad \dots \quad (3)$

The curvature $\quad \frac{1}{R} = \frac{d^2 y/d\,x^2}{\left[1 + \left(\frac{d\,y}{d\,x}\right)^2\right]^{3/2}} \quad \dots \quad (4)$

The curved length $\quad S = \int_0^x \sqrt{1 + [f'(x)]^2}\,d\,x \quad \dots \quad (5)$

For purposes of feeding the frame through (and for ordering purposes), it is the length along the neutral axis, the position of which can easily be determined using the simple theory of plastic bending when full plasticity is attained.

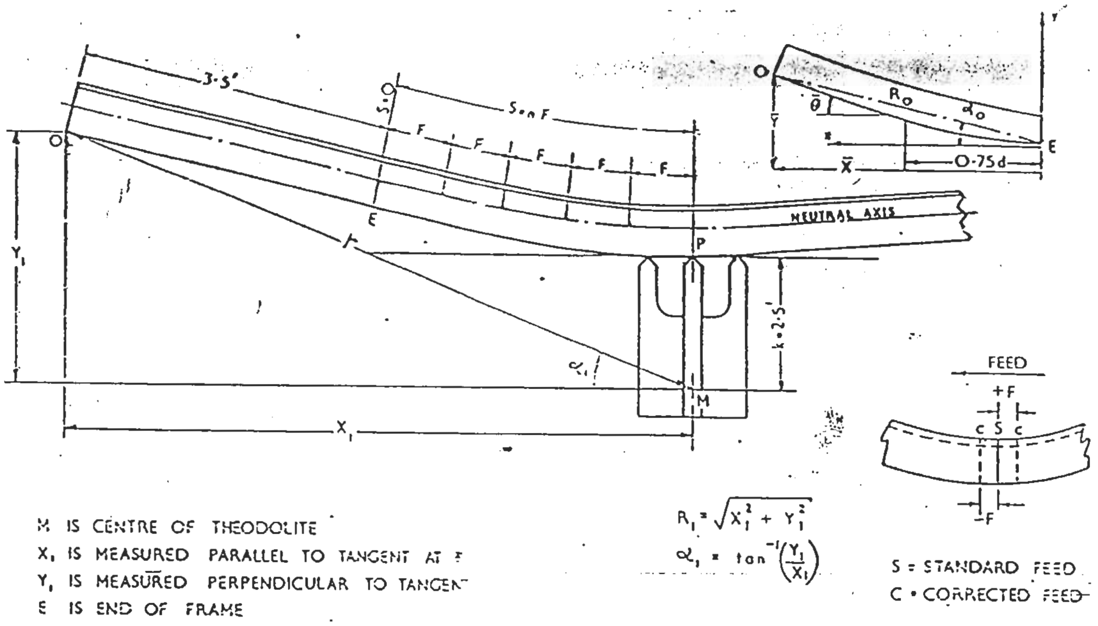

FIG. 14.—GENERAL ARRANGEMENT OF FULL-SCALE TESTS

The length along the outer edge of the frame can be calculated from (5) above at a number of discrete points and using the curve fitting routine again, an equation relating S and x is produced:

i.e. $x = \psi(S)$ . . . . (6)

It is now required to determine the points on the outer edge, which correspond to the loading points on the neutral axis. Suppose that the feed is taken constant and equal $= l$ from symmetry

$$\frac{R_i}{R_0} = l/S \quad . . . . (7)$$

where R_i and R_0 = radii of curvature of neutral axis and outer edge respectively.

S = curved length along the outer edge.

giving $$S = l\frac{(R_0)}{R_i} = \frac{l}{1 - h/R_0} \quad . . . . (8)$$

where h = distance of outer edge from neutral axis.

Using the frame equation $y = f(x)$, the curvature $\frac{1}{r_1}$ could be calculated. The value of h could be determined from the simple theory of plastic bending.

For O.B.F. sections

$$h \doteq \frac{A_T}{2t}$$

where A_T = total area of section.

t = thickness of web.

The feed is chosen according to frame size and by using equations (8) and (6) the x corresponding to each feed position can be determined.

These calculations are repeated at every loading point along the frame length.

In this way, all points on the outer edge, which correspond to the loading points on the neutral axis, can be determined. At these points, using the frame equation, the curvature and the locus of the end point can be calculated.

It is obvious that the calculations involved are very simple and therefore their programming is straightforward.

D. Calculation of the Allowable Error at the Free End on the Bent Side

Errors at the free end will be mainly due to:—

(1) *Error in the Tangent Direction*

Any error in the tangent direction will create an error in the end locus position and is given by:—

$$\text{error in } Y = dY = \pm X\,d\theta$$

an error of one minute will give an error of $\pm$ 0·1047 in. at the end of a frame of X = 30 ft.

(2) *Error Due to Tolerances in Frame Manufacture*

Tolerances of the order of $\pm 2\frac{1}{2}$ per cent of the weight of rolled steel sections are expected, as given in the manufacturers' handbooks. This error is chiefly due to tolerances on the frame depth which is of the order of $\pm\frac{1}{8}$ in. for any standard frame size.

Since all measurements will be taken relative to a datum line, which is chosen according to the frame size, there will be an error in the chosen datum caused by these tolerances. This error is of the order of $\pm\frac{1}{8}$ in.

(3) *Error Due to Reduction in Depth and Indentation*

Since bending will be done in the plastic range of the material, the section will experience a variation in thickness as well as a reduction in depth. This reduction is believed to be of the order of 0·10 in. Further, due to the high bearing pressure existing between the frame and the machine at the loading points, indentations of the order of 0·05 in. are expected to take place. Consequently, an error of −0·15 in. is expected to take place at the central support.

(4) *Error in Feed*

An error in the feed means an error in the bent length and can be interpreted as an error in the slope.

It was shown before that an error of $\pm$1·0 min. in the slope will lead to an error of $\pm$0·1047 in. in the locus. If this error accumulates along the frame length, a serious error at the frame end will be expected. As a result, it will be necessary to measure the feed so as to limit this source of error.

(5) Error caused by Friction Forces in supporting Trolley and Carriage

Any friction forces in either the transverse or longitudinal directions will give rise to bending of the frame about the central support. On the assumption that the lever arm to the frame is 30 ft., the moment of inertia of the frame section is about 50 in.[4] (based on 9·0 in. O.B.F., 0·58 in. thick), Young's Modulus 13,500 tons/in.^{2}, a transverse force of 25 lb at the frame end will give a deflection of:—

$$\delta = \frac{w\,l^3}{3\,E\,I} = 0{\cdot}257 \text{ in.}$$

The lever for the longitudinal force will rarely exceed 15 ft. and therefore the error is expected to be much less than the above figure.

Summation of Errors

If all the above errors were to sum up, then a serious error in the locus might occur. The probability of this occurrence has not been studied since it is thought that some of the errors will be systematic, while others will be random, and a series of tests will have to be carried out on a prototype machine to establish the true frequency distribution.

APPENDIX II

A. Effect of Plastic Bending on Thickness Variation

As a result of plastic bending, sections experience a reduction in thickness in the tension side and an increase in the compression side. In order to find the variation in thickness when the web is under compression, it will be assumed that yielding will take place without strain hardening.

i.e. $\sigma_\theta = \sigma_y =$ yield stress

where σ_θ = bending stress.

From the equilibrium condition in the radial direction we obtain the following equation, see Fig. 6

$$(\sigma_r - \sigma_\theta)\, d\,R + R\, d\,\sigma_r = 0 \quad . \quad . \quad . \quad (1)$$

which could be solved as follows:—

$$\sigma_r = \sigma_\theta\,(1 - R_1/R) \quad . \quad . \quad . \quad . \quad . \quad (2)$$

which shows that the radial compressive stress σ_r is a function of the tangential stress σ_θ and the curvature attained.

where:

σ_r = radial stress at a radius R,

R_1 = minimum radius of curvature in the compression side.

To obtain the variation of thickness across the frame depth, the following equations are made use of:—

(i) The relationship between the principal stresses and strains, in the plastic range, is as given by St. Venant:—

$$\frac{\sigma_\theta - \sigma_r}{\epsilon_\theta - \epsilon_r} = \frac{\sigma_r - \sigma_z}{\epsilon_r - \epsilon_z} = \frac{\sigma_z - \sigma_\theta}{\epsilon_z - \epsilon_\theta} \quad . \quad . \quad . \quad . \quad (3)$$

where: σ_θ, σ_r, σ_z are the three principal stresses and ϵ_θ, ϵ_r and ϵ_z are the corresponding natural strains.

In our case $\sigma_z = p$, the clamping pressure

(ii) The condition of constancy of volume

i.e. $$\epsilon_\theta + \epsilon_r + \epsilon_z = 0 \quad . \quad . \quad . \quad . \quad . \quad (4)$$

When the clamping pressure $\sigma_z = p$ is made zero in equation (3), and by using equations (2) and (4), the following expression is obtained:—

$$b = b_0\,[R/\rho]^{\frac{R_1 - 2R}{R_1 + R}} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where:

b = thickness at any radius R in the compression zone

b_0 = original thickness

R_1 = minimum radius

ρ = radius of curvature attained by neutral axis.

The maximum value of "b" occurs when:

$$R = R_1$$

giving $$b_{max} = b_0 \sqrt{\rho/R_1} \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Increase in thickness at the outermost fibres

$$= b_0\,[\sqrt{\rho/R_1} - 1] \quad . \quad . \quad . \quad . \quad . \quad (7)$$

For a 17·0 in. O.B.F. section, 0·78 in. thickness, bent to an arc of 100 in. radius (at the extreme fibres on the compression side), the increase in thickness is 0·0413 in.

B. Effect of the Clamping Pressure on Buckling

In order to investigate the effect of clamping pressure on the plastic deformation and buckling of ship frames, the Von–Mises yield criterion was used, i.e. yielding was assumed to start when the stress state is given by the following expression (8):

$$(\sigma_1 - \sigma_3)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_1 - \sigma_2)^2 = 2\,\sigma_y^2 \quad . \quad (8)$$

where $\sigma_1 > \sigma_2$ and $\sigma_1 > \sigma_3$ are the three principal stresses.

σ_y = yield stress of the material.

In the case of bending into the plastic range of the material, the principal stresses are σ_θ; σ_r and p. This is based on the assumption that shear stresses in the plastic range are zero.

In this case we have:

$\sigma_\theta = \sigma_1$ = bending stresses

$\sigma_r = \sigma_2$ = radial stresses

$p = \sigma_3$ = clamping pressure

Therefore, expression (1) could be modified as follows:—

$$(\sigma_\theta^2 + \sigma_r^2 + p^2) - (\sigma_\theta \,.\, p + \sigma_\theta \,.\, \sigma_r + p \,.\, \sigma_r) = \sigma_y^2 \quad . \quad (9)$$

when $\sigma_\theta = \sigma_r = p$.

i.e. hydrostatic stress condition, since the three stresses are compressive, no plastic deformation will take place, see Fig. 6.

To allow for the plastic deformations to take place, the following conditions must exist:—

$$\sigma_\theta \gg \sigma_r$$

and $$\sigma_\theta \gg p$$

i.e. to keep σ_r and p as low as possible.

Equation (2) shows that σ_r depends on σ_θ and the radius of curvature when the clamping pressure is zero.

Consequently, nothing can be done to reduce σ_r with respect to σ_θ, since they are interdependent.

Therefore, for the material to deform plastically, the clamping pressure p should be kept as small as possible. Ideally, the best condition is achieved when a tension force is applied, but since it is extremely difficult to achieve such a condition practically, it is considered, that $p = 0$ is the best condition.

When the material is allowed to deform plastically, the tendency to buckle will be greatly reduced.

APPENDIX III

Residual Stresses in Ship Frames due to Forming by Cold Bending

The residual stresses due to cold bending are, in effect, the resultant of two stress patterns. The first pattern is due to bending into the plastic range of the material and is shown in Fig. 3(*a*). The other stress pattern results from the process of elastic recovery or spring back and is shown in Fig. 15.

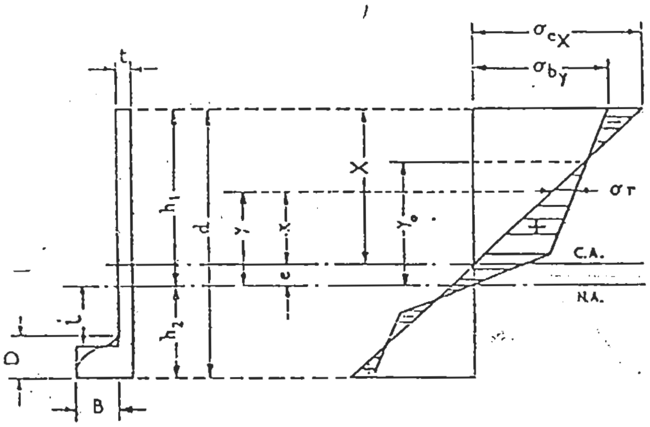

FIG. 15.—RESIDUAL STRESS DISTRIBUTION

Timoshenko (8) gave a method whereby the residual stresses can be calculated and measured on the assumption that the material is of the ideal plastic type and that a symmetrical section is used. In the following analysis, the calculation of the residual stresses is based on the assumptions:—

1. The stress strain diagram is as shown in Fig. 3, i.e. having a linear strain hardening range.
2. Stress strain diagram is identical for both tension and compression.
3. Plane sections before bending remain plane and normal to the neutral axis.

   i.e. $\epsilon = y/\rho$ . . . . . . (1)
3. Symmetrical sections are dealt with, and the plane of loading passes through the centroid of the section.
4. The material is homogeneous and isotropic.

In order to calculate the residual stresses, the bending moment acting on the section as well as the position of the neutral axis should be known.

(a) Bending Moment

The bending moment is calculated as follows:—

$$M = \int_0^A \sigma \, d A$$

where $\sigma = \sigma_y + (\epsilon - \epsilon_y) \tan \alpha$

It can be shown that

$$M = M_0 + (I_n \tan \alpha)/\rho \quad . \ . \ . \ . \ . \ . \ . \ . \quad (2)$$

where $M_0 = Z_p (\sigma_y - \epsilon_y \tan \alpha) = M_p - Z_p \epsilon_y \tan \alpha$ . . (3)

I_n = 2nd moment of area about the neutral axis

$= Z_n . h_1$

Z_n = first moment of area about the neutral axis

h_1 = distance of extreme fibres, on the web side, from the neutral axis.

From (2) and (3), we get:

$$M = M_p + Z_n \tan \alpha \{h_1/\rho - (Z_p/Z_n) . \epsilon_y\} \quad . \ . \quad (4)$$

(b) Neutral Axis Position

The neutral axis position is determined from the conditions of equilibrium: $\Sigma F = 0$ and $\Sigma M = 0$.

Using the force condition.

$$\int_0^{A_T} \sigma . d A = 0 \quad . \ . \ . \ . \ . \ . \quad (5)$$

we have $\int_0^{A_e} \sigma . d A + \int_0^{A_p} \sigma . d A = 0$

where A_T = total area of the section

A_e = area of elastic core

A_p = area of plastic zone

Neglecting the area of the elastic core, we get:

$$\int_0^{A_{p1}} [(\sigma_y - \epsilon_y \tan \alpha) + (y/\rho) \tan \alpha] \, d A = \int_0^{A_{p2}} [(\sigma_y - \epsilon_y \tan \alpha) + (y/\rho) \tan \alpha] \, d A.$$

where A_{p1} = area of plastic zone on the tension side

A_{p2} = area of plastic zone on the compression side

Therefore $(\sigma_y - \epsilon_y \tan \alpha) A_{p1} + (\tan \alpha/\rho) . A_{p1} . \bar{y}_1 = (\sigma_y - \epsilon_y \tan \alpha) A_{p2} + (\tan \alpha/\rho) . A_{p2} . \bar{y}_2.$

Putting $A_{p2} = A_T - A_{p1}$

$$A_{p1} = \frac{A_T [(\sigma_y - \epsilon_y \tan \alpha) + (\tan \alpha/\rho) . \bar{y}_2]}{2 (\sigma_y - \epsilon_y \tan \alpha) + (\tan \alpha/\rho) . (\bar{y}_1 + \bar{y}_2)} \quad (6)$$

where $\bar{y}_1$ = distance of centroid of A_{p1} from the neutral axis

$\bar{y}_2$ = distance of centroid of A_{p2} from the neutral axis.

For deep sections having only one flange, e.g. Tee section, bulb plate, the area A_{p1} is given by:

$$A_{p1} = h_1 . t \quad . \ . \ . \ . \ . \ . \ . \quad (7)$$

From (6) and (7), we get:

$$h_1 = \frac{A_T [(\sigma_y - \epsilon_y \tan \alpha) + (\tan \alpha/\rho) . \bar{y}_2]}{t [2 (\sigma_y - \epsilon_y \tan \alpha) + (\tan \alpha/\rho) . (\bar{y}_1 + \bar{y}_2)]} \quad . \ . \ . \ . \ . \ . \quad (8)$$

The effect of the strain hardening is not large, in this instance, since it affects both the tension and compression zones, and it is to be expected that the neutral axis determined from the assumption of an ideal stress strain diagram (i.e.) $\tan \alpha = 0$ will not be greatly in error.

Therefore, when $\tan \alpha = 0$

$$h_1 = A_T/2t \quad . \ . \ . \ . \ . \ . \ . \quad (9)$$

(c) Calculation of the Residual Stresses

The residual stresses are the resultant of the loading and unloading stresses, see Fig. 15.

The loading stresses are as given by:

$$\sigma_b = (\sigma_y - \epsilon_y \tan \alpha) + (y/\rho) \tan \alpha \quad . \ . \ . \quad (10)$$

The unloading stresses are calculated from the bending moment as follows:—

$$M = Z_e . \sigma_e \quad . \ . \ . \ . \ . \ . \quad (11)$$

where M = unloading moment = bending moment.

Z_e = elastic modulus of the section.

σ_e = elastic stress.

From equation (4), we have:

$$M = (M_p - Z_p \epsilon_y \tan\alpha) + (h_1/\rho) . Z_n \tan\alpha \quad . \quad (12)$$

from equation (11) and (12), we get:

$$\sigma_{eX} = M_p/Z_c - (Z_p \epsilon_y \tan\alpha)/Z_c + (h_1 Z_n \tan\alpha)/\rho Z_c \quad . \quad (13)$$

where σ_{eX} = elastic stress at the extreme fibres of the section.

therefore elastic stress at any depth x from the centroid is:

$$(\sigma_e)x = [M_p/Z_c - (Z_p \epsilon_y \tan\alpha)/Z_c + (h_1 Z_n \tan\alpha)/\rho Z_c]\frac{x}{X} \quad . \quad (14)$$

where X = distance of extreme fibres from the centroid of the section.

Residual stresses $= \sigma_r = (\sigma_b)_y - (\sigma_e)_x$

where

$(\sigma_b)_y$ = bending stress at a distance y from the neutral axis

$(\sigma_e)_x$ = unloading stress at a distance x from the centroidal axis.

therefore

$$\sigma_r = (\sigma_y - \epsilon_y \tan\alpha) + (y/\rho) . \tan\alpha$$
$$- [M_p/Z_c - (\epsilon_y Z_p \tan\alpha)/Z_c$$
$$+ (h_1 Z_n \tan\alpha)/\rho Z_c](y - e)/X \quad . \quad (15)$$

Let y_0 be the value of y when the residual stress becomes zero, see Fig. 15.

i.e. at $x = y_0 - e$

therefore y_0

$$= -\frac{(\sigma_y - \epsilon_y \tan\alpha) + e(M_p/I_c - (\epsilon_y Z_p \tan\alpha)/I_c + (I_n \tan\alpha)/\rho . I_c}{\tan\alpha/\rho - M_p/I_c + (\epsilon_y Z_p \tan\alpha)/I_c - (I_n \tan\alpha)/\rho I_c} \quad . \quad . \quad . \quad . \quad (16)$$

Equation (15) gives the residual stress at any depth y from the neutral axis, whereas equation (16) gives the value of y when σ_r becomes zero.

As an example, consider the case of a 12·0 in. O.B.F. section bent to an arc of a circle of radius 80·0 in. (measured on the neutral axis (i.e. ρ = 80·0 in.). It will be assumed also that bending will take place without strain hardening (i.e. $\tan\alpha = 0$, and that the yield stress σ_y = 16·0 tons/in.².

The residual stress in the extreme fibres is calculated from equation (15) by putting $X = y - e$

therefore $\sigma_r = -11{\cdot}36$ tons/in.²

and by using equation (16), the released bending moment at the extreme fibres for:

$$y_0 < y < h_1$$

could be calculated. The spring back in curvature at midspan could then be calculated and for a span of 10·0 in., it is given by

$$\frac{1}{R_r} = 0{\cdot}351 \times 10^{-3} \text{ in.}^{-1}$$

where: $\frac{1}{R_r}$ = spring back in curvature.

For a frame 30 ft. long, the corresponding spring back at the end point is 0·316 in. An error of this order of magnitude could occur at a number of points along the frame and these could accumulate to produce a serious distortion in the section on completion of welding.

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