THE DAMPING, STIFFNESS AND FATIGUE PROPERTIES OF JOINTS AND CONFIGURATIONS REPRESENTATIVE OF AIRCRAFT STRUCTURES

by

D. J. Mead

University of Southampton
England

Introduction

The damping of a structure vibrating under resonant or 'random-resonant' conditions is undoubtedly one of the most importantproperties of the structure. When an aeroplane structure is excited by acoustic pressures, it is subject to two principal damping actions, viz. the acoustic radiation damping and the internal structural damping. The source of the latter was discussed many years ago by P.B. Walker (Ref.1), when it was attributed mainly to the numerous riveted joints in the structure, but also to the internal hysteresis of the structural material. A further source was recognized in the structural discontinuities and distortions which caused minor shocks in localised areas. The latter source may have significance when very large amplitudes of panel vibration are excited acoustically, but the structural damping of most of the acoustically excited modes will derive principally from the damping action at the riveted joint.

The fundamental mechanism of the damping of a riveted joint is generally recognized to be one of slipping and dynamic friction. Pian (Ref. 2) has made a theoretical analysis of an ideal type of continuously riveted joint between a spar and a separate spar cap, and the energy loss predicted by the theory agreed well with an experimental investigation. The experiment, however, was not on a scale representative of the type of aircraft structural joint troubled by acoustic fatigue. It is obvious that the normal pressure between two plates that slip relative to one another is a very important parameter, and in the normal riveted joint no control can be exercised over its magnitude. For this reason, damping investigations into really typical riveted joints have been conducted in the United Kingdom, with a view to determining the reproducibility, or otherwise, of the joint damping and stiffness properties. This work has yet to be completed, as there are obviously numerous types and sizes of joint and rivet that could be investigated, in addition to there being different ways of loading the joint to simulate acoustically excited conditions. The present state of knowledge concerning the damping of the riveted joints so far tested is described in this paper.

The fatigue properties of joints under the normal type of longitudinal loading have received considerable attention in the past, but experience with structures in severe acoustic environments indicates that the modes of failure are different from those of the joints loaded in the orthodox manner. This is simply due to the difference in the nature and direction of the loading, but it immediately opens up the necessity for basic fatigue testing to be conducted on simple joints with the appropriate type of loading applied, So far most of the fatigue testing that has been carried out representing acoustically excited conditions has been on components containing perhaps many joints, the nature of the loads on which have been a property of the particular structure in which they were incorporated. It is therefore difficult (if not impossible) to relate a joint fatigue life in one such test to the life of an identical joint in another test in which the structure is different in some way. In general the precise nature of the joint loading is very difficult to both measure and control in this type of test. In view of the scarcity of work performed on fatigue properties of joints under these conditions, the section on fatigue properties in this paper is restricted mainly to a discussion of the modes of failure experienced in service, with some reference to the relevance of the more orthodox joint fatigue investigations to the acoustic fatigue problem.

2. The mechanism of the damping of a riveted joint.

In order to understand the mechanism of the damping at a riveted joint, it is necessary first of all to understand the nature of the contact surface between the two joint plates. This is likely to vary according to the method used to form the rivet. Suppose the rivet to be formed by a hammer or rivetting gun. In an ideal joint the forming blows will help to press the plates together as the rivet is being expanded in the hole. As the plates are pressed together, a circle of contact may be formed around the rivets, of diameter up to two or three times that of the rivet. Furthermore, the expansion of the rivet in the hole will give rise to radial compressive stresses in the plates, which in turn cause a Poisson expansion in the direction perpendicular to the plate surface. This will increase the normal pressure between the plates in an annular region very close to the rivet, and in the event of the two plates being of different hardness, a small ring may form around the rivet on one plate, whereas a corresponding indentation will be formed in the other. Examination of a broken-down joint indicates that both of these areas of contact exist together (see Fig.1).

If the rivet is formed by a squeezing device, the applied pressure will not press the plates together, and the larger circle of contact is unlikely to exist. The small ring of contact immediately around the rivet is still likely to exist, together with areas of contact at other points where the plates were actually touching at the time of the forming of the rivet.

Examination of one broken-down joint indicated that most (if not all) of the damping action took place in the larger contact circle outside the annular ring around the rivet. Over this area, the actual contact between the plates is by no means uniform owing to the roughness, on a macroscopic scale, of even the smoothest plates. This roughness takes the form of many jagged asperities, the peaks of which are embedded in or in contact with the other surface.

When a small oscillating tangential load is applied to the joint, the load will be transferred from one plate to the other by means of shear through the rivet and through the static friction at the contact area. Elastic displacements will occur in the plates, in the rivet and also in the contact asperities. Any energy loss in a complete load cycle at small loads will be largely due to 'elastic' hysteresis of the joint material, but it is important at this stage to bear in mind the nature of the shear stress distribution over the contact area. Considering the larger ring of contact around the rivet, classical elastic theory indicates that the radial variation of the shear stress is of an exponential form, tending to infinity at the circumference. An infinite shear stress would be relieved by plastic deformation of the contact asperities at the circumference, with an accompanying energy loss and consequent damping mechanism. This should occur at even the smallest load amplitudes, but the energy lost at the circumference would then be only a very small proportion of the total.

As the tangential load increases, the extent over which plastic deformation takes place also increases, and the energy lost per cycle should increase at a greater rate with respect to the load than the simple hysteresis mechanism suggests

With further increase of tangential load, a point is reached at which actual slipping begins at the boundary of the contact area, and energy is then dissipated by virtue of the dynamic frictional forces which have to be overcome. As the tangential load increases further still, slip occurs over an increasing area, the inner boundary of which approaches the rivet itself.

It is evident that the most important parameter upon which the energy dissipation depends is the normal pressure between the plates. This determines both the area of contact and the load at which slip begins. It is equally evident that it is one of the most difficult quantities either to measure or to control, and this therefore raises the problem of whether nominally identical joints can be expected to have similar damping properties. For this reason, a programme of experimental work has been carried out at the University of Southampton to investigate the reproducibility of the damping from one joint to another, and also to determine the amplitude dependence of the damping.

3. The measured damping of typical joints

3.1 Low Frequency Experiments

The early work in the United Kingdom, beginning eight or nine years ago, was directed towards an understanding of the damping of the normal modes of vibration of complete aeroplanes, and as such was carried out in the relatively low frequency range extending up to about 40 c.p.s. In all cases, a simple rivetted joint was inserted in the flange of a thin walled beam which was then made to vibrate in its fundamental free-free mode of bending vibration (see Figure 2). The joint was made by cutting the flange and covering the cut with a single butt-strap which was riveted to the flange. The rivets were therefore subjected to single shear type of loading. In all the work to date, the rivets have been formed manually by means of hammer blows. There are obviously numerous combinations of rivet size and type, plate thickness and material, and other factors upon which any quantitative results must depend. As only a limited number of these combinations have so far been investigated, it is the qualitative results which have most significance.

The first joints tested consisted of 24 S.W.G. Alclad plates and $^{1}/_{8}$in. diameter mushroom-headed duralumin rivets. The damping properties of the joint were deduced from recordings of the freely decaying oscillations of the beam fundamental bending mode of vibration. The results of these experiments indicated clearly that for a given amplitude of load on the joint, the energy dissipated per cycle did not change as the frequency of the beam was made to change (by adding mass to the beam). The frequency range covered, however, was small, extending from only 20 c.p.s. to 40 c.p.s. These early tests indicated that the logarithmic decrement of the decaying oscillation was constant as the amplitude decreased, implying that the damping followed the simple linear complex stiffness law, but the amplitude range covered was rather small. In subsequent work on some joints a marked change was observed in the logarithmic decrement as the amplitude decreased. At a certain critical amplitude, the log. dec. appeared to change from one constant value to another smaller constant value. At the time, this was attributed to the cessation of slipping at the joint, but our subsequent understanding of the mechanism of the damping does not support this view. The phenomenon of a sudden change of this nature has not occurred in the most recent work undertaken.

Of greatest interest in this early work is the comparison of the energy dissipation at joints having different sizes and types of rivet. Snap headed rivets of three different sizes were tested, and the energy dissipated per cycle of a given load amplitude varied almost exactly in inverse proportion to the rivet shear area. This is to be expected, as the larger rivets, having the greater stiffness permit less load to be transferred through the friction path. Mushroom-headed rivets of the same diameter gave rise to energy dissipation which was consistently less than that of the snap-headed rivets by up to 30%. This could possibly be due to a higher degree of fixation at the head end of the mushroom-headed rivet than on the snap-headed rivet and therefore an increased stiffness. Unpinned tubular rivets (Chobert type) yielded considerably higher values of energy dissipation, of up to four times that of snap-headed rivets. These rivets also exhibited the phenomenon of a sudden <u>increase</u> in damping as the amplitude dropped, but the work was not continued to investigate this further. The lower shear stiffness of this rivet and the smaller fixation from the head would contribute to the larger damping that was observed. One further point of interest in this work was the relatively close agreement between the damping of identical joints on two separate beams, made and tested on two separate occasions. The difference between the measured energy dissipation was no more than about 20%, which, in view of the mechanism of the dissipation and its dependence on the normal pressure, must be considered to be quite small.

As an indication of the actual magnitude of the energy loss at one of the snap-headed rivets of $^1/8$in. diameter, a value of $1{\cdot}75 \times 10^{-6}$ lb. in. x Joint Load $(\text{lb.})^2$ was derived from the experimental results.

It should be emphasized here that none of these joints were subjected to long periods of continuous excitation, as a result of which no time effects on the damping would have been observed. Furthermore, the restricted amplitude range that was covered would not have been sufficient to show up clearly the amplitude effect on the damping that has since been found.

3.2 Higher Frequency Experiments

Work at the higher frequency of about 150 c.p.s. has recently been conducted at Southampton, this time on $^1/8$in. diameter countersunk rivets in an 18 S.W.G. butt strap which covered a cut in the crown of an open channel section stringer (see Figure 3). The stringer was again made to vibrate in its fundamental mode of bending, and damping measurements were made under conditions of steady harmonic loading of the joint. In order to eliminate the effect of acoustically radiated energy, the experiments were initially conducted in an evacuated cylinder. To determine the effects of all the extraneous damping sources, preliminary tests were conducted on an unjointed, but otherwise identical beam. It so happened that the extraneous damping was a very small proportion of the total damping when the joint was included.

The damping measured on one of these beams is shown in Figure 4, where the damping ratio of the beam is plotted against the amplitude of the load on the rivet. (Damping ratio = Actual damping coefficient of the beam ÷ the critical damping coefficient). If the energy dissipated at the joint was proportional to the square of the joint load, (i.e. linear damping) the damping ratio would remain constant as the load amplitude varied. It is seen therefore that the joint behaves in a sensibly linear manner up to a joint load amplitude of about one pound, but thereafter the damping increases rapidly with amplitude. This increase is presumably associated with the onset of slip at the joint. Difficulty was found in the early experiments in obtaining a smooth curve over the steeply rising portion, and this was soon traced to the fact that the damping was decreasing quite rapidly with time. This time-variation of the damping is shown in Figure 5, each curve of which shows the change of damping while the joint load amplitude was maintained at a constant value. Over the first $6^1/2$ hours, the joint load was 6·7 lb. and after the first three hours the damping was becoming steady. Testing then stopped for three days after which it was found that the damping had increased during the rest interval, some form of recovery having occurred. Upon further loading the damping began to fall once again during a three hour test, but recovery took place once more during another day's rest. The recovery this time was smaller than before. The loading was then increased to 19 lb. on the joint. The damping increased initially but fell very rapidly as the test proceeded. After six hours it was evident that an asymptotic value was being approached. The recovery effect was again observed, as shown by the Figure, which shows that a greater degree of recovery occurred during the three day rest at $13^1/2$ hours than during the one day rest at $9^1/2$ hours. Further increase of load (up to 29 lb. and then to 37 lb.) was each time accompanied by a large initial rise followed by a similar rapid decrease with time, of the damping, the rate of decrease of the damping always dropping with time. Finally, the joint load amplitude was fixed at 6·7 lb. again for two hours, during which there was little change of damping but the magnitude of it was well below that of the earlier measurements at the same load. At this stage the damping was again measured over the whole amplitude range previously covered, and the curve II of Figure 6 was obtained. Comparison with curve I, which is that of the first test, indicates that the effect of continuous loading was really to shift the whole curve to the right, i.e. to delay the load at which the slipping process appears to begin. Prolonged loading at low amplitude (below about 2 lb. joint load) caused no change in the damping, whereas prolonged loading at a high load did result in a reduction of the damping at the low load.

In view of this marked time dependence, it is interesting to observe the damping curves of two identical jointed beams. These are shown in Figure 7, curves I and II being the same as those of the first beam, and curve III and IV being those of a second beam, before and after prolonged excitation respectively. The damping over the load range before slip occurs is seen to be very nearly the same for each beam. Over this range the damping ratios of the jointed beams were about five times those of the unjointed beams. The fact that the curves for the different beams begin to rise at different load amplitudes can be attributed to the different durations of loading to which the beams had been subjected. It is probably not without significance, however, that curve IV of the second beam (which was measured after prolonged loading) when shifted to the left, superimposes exactly in magnitude and form upon curve II of the first beam. Reproducibility of the damping of such riveted joints therefore appears to be possible over the lower load range, but variation occurs in the load at which the sharp damping increase occurs. Thereafter, the rate at which the damping increases again appears to be reproducible.

The phenomenon of a drop in damping (or friction) under prolonged loading has been observed by investigators into fretting corrosion. In particular, Fenner, Wright and Mann (Refs. 6) found a drop in dynamic friction when two steel surfaces were rubbed one against the other with a constant displacement amplitude. It was evidently due to the accumulating of oxide deposit at the interface. The oxidation occurs due to the high temperatures generated at the contact asperities in the early stages of the slipping process, temperatures which are sufficient to cause local welding of mating asperities. These welds are torn apart as the displacement increases and in the presence of oxygen an oxide film readily grows on the surfaces. This, however, may be continually removed by the subsequent ploughing action of the asperities, exposing a fresh surface to the oxidizing process. The oxide forms into a layer between the surfaces and after some time reaches a constant thickness. Thereafter the coefficient of friction between the surfaces corresponds to oxide sliding on oxide.

After the tests on the countersunk rivets described above, an accumulation of aluminium oxide was found over the contact areas of both joint plate and the adjacent beam surface. These are seen in Figure 8. The fact that the contact is not in a circle around the rivet will be discussed later. The significant feature for the present argument is that when the oxide deposits were lightly rubbed with very fine emery paper, the 'tails' of the elongated marks remained, leaving a polished circle or ellipse in the middle of the mark. It would seem reasonable to assume, therefore, that slip and oxidation began at the extremities of the contact area (where the shear stress would be greatest) at the load at which the damping curve began to rise. The progressive oxidation associated with prolonged running at this load would cause the damping to drop with time. When the load was increased, the area of slip would increase towards the centre of the contact area where no oxidation had previously occurred. The damping would be expected to rise rapidly as the load was increased, but to drop again with increase of time as oxidation progressed in the new area of slip.

That the contact area was not a circle around the rivets was due to a very slight concavity in the surface of the stringer to which the butt strap was attached. Such surface irregularities are quite likely to occur at actual aeroplane joints, so the results of the work described are by no means unrepresentative of those pertaining to a real structure.

In view of the unexpected nature of these contact surfaces, further similar experiments were conducted after having ensured that the contact area was more orthodox. The surprising feature of the results here was that the general magnitude and variation of the damping was almost identical with that of the former joints, despite the great difference in the contact areas (see Figure 9). At the low, pre-slip amplitudes of joint load the energy dissipation was the same to within the accuracy of measurement. Slip began in the same range of load amplitude. The critical slip load again increased with duration of continuous testing. The rate of increase of the energy dissipation with respect to the load, after the onset of slip, corresponded closely with the former result. When the joint was opened up after the experiment the contact area was found to be a circle of about $^{3}/16$in. diameter around the rivet ($^{1}/8$in. diameter). Oxide deposits were clearly visible all round the edge of this circle. The explanation suggested for the time variation of the damping of the former joints applies equally well to this joint with its circular contact area.

It is interesting to compare the energy dissipated per cycle by this countersunk riveted joint with that dissipated by the snap-headed riveted joints of the much earlier work. At the high amplitudes of joint load, the energy loss at the countersunk rivet was in the region of $0{\cdot}75 \times 10^{-6} \times (\text{Joint Load})^2$ (in.lb.) per cycle but this, of course, was increasing rapidly with the load amplitude. At the snap-headed rivet the loss was at the rate of $1{\cdot}75 \times 10^{6} \times (\text{Joint Load})^2$ per cycle, at an unspecified joint load. The snap-headed rivets were in plates of considerably lower thickness than those of the countersunk rivets, in which case the joint stiffness could be expected to be considerably lower. This, in turn, can be shown to imply that the energy loss would be greater, other things being equal. In plates of the same thickness as those of the countersunk rivets, it can therefore be expected that the snap-headed rivets would give an energy loss at least of the same order as the countersunk.

4. The damping of multi-jointed structures

In order to calculate the damping of a simple vibrating structure which contains joints whose properties are known, it is necessary in general to specify both the stiffness and the 'equivalent hysteretic' damping coefficients of the joints. (It will be the 'equivalent hysteretic' damping coefficient that will be derived from the results of the harmonic tests on the joints that have already been described). However, when the joint flexibility makes no appreciable change to the mode of vibration and the associated stress distribution within the structure (i.e. as compared with that of an identical, but continuous structure without joints), neither the damping nor the stiffness coefficients are required explicitly. It is only necessary to know the energy dissipated at the joint per cycle of load as a function of the load amplitude on the joint (see Ref. 7). When the mode of vibration of the structure is known, the load on each joint can be calculated for any amplitude of the mode, and the total energy dissipated by all the joints may be evaluated. The damping coefficient, or the damping ratio corresponding to the mode is then easily found.

An approach of this nature is justifiable when the joints are transmitting bending stresses in the flanges or surfaces of a beam. Elementary beam theory may be used to determine the bending stresses and joint loads.

When the joint forms a shear connection, as for example, between a stringer and the skin to which the stringer is attached, the exact approach to the damping calculation is to allow for the shearing deformation of the joint in the differential equation for the mode of bending deflection of the configuration. However, the stiffness of a normal line of rivets is so high that their shearing deformations do not cause any significant change in the stress distribution predicted by the elementary beam theory, so that the above method of calculating the damping may still be used.

It is interesting to notice with respect to configurations containing such a shear joint that the maximum possible structural damping that can be obtained from the joint occurs when the hysteretic damping coefficient of the joint is equal to the stiffness coefficient. This fact is equivalent to saying that the optimum 'loss factor' of the joint is unity. The result may be shown by using either the exact theory including the rivet deflection effects, or the elementary beam theory approach. (These two approaches converge as the joint stiffness increases and as the distance between the adjacent surfaces of the two parts of the cross-section becomes very small).

Calculations of the total damping of two simple multi-jointed specimens have been carried out on the above basis. One of the specimens had the joints in the bending stress carrying flanges of a beam identical in all other respects to that in which the basic joint properties had initially been measured (Ref. 4). Good agreement between measured and calculated damping was found. The other specimen (Ref. 5) was in fact a typical aircraft stringer riveted along its length to a strip of thin plate representing, say, a strip of fuselage skin. The estimated value of the contribution of the joints to the total damping coefficient was only 0·0005% of the critical damping of the beam! The beam was considered to be vibrating in its fundamental mode of free-free vibration, with none of the rivets being loaded sufficiently highly to bring them into the slipping condition. For slip to occur, very high vibration amplitudes of the beam were required which could not be accommodated on the beam that was subsequently to be tested. The amplitudes would also be much greater than anything that could be expected in a structure excited by noise. The small value of damping predicted would be swamped by all the extraneous damping present in any test to verify the calculation. However, a test was carried out, and the damping coefficient of the stringer skin combination was found to be about 0·013% of critical, whereas the damping of a stringer on its own was found to be about 0·01% of critical. It is not legitimate to subtract these damping ratios, since different beam stiffnesses are involved in each case, but this preliminary experimental evidence does point to the smallness of the contribution of the stringer-skin joints to the total damping.

An important conclusion emerges from these results. It is known, of course, that acoustic pressures can excite the modes of bending vibration of stringers and frames in a fuselage. These modes are necessarily coupled to skin panel modes, and with one another, but the stringer-skin or frame-skin combination may be regarded as a beam and the attachment rivets to the skin will be damping the motion in the same way as above. The order of the damping ratio contribution from these rivets should not differ greatly from that from the rivets in the particular beam just discussed, i.e., 0·0005%. This would be greater for a beam of shorter length, increasing in inverse proportion to the square of the length, provided no slip occurred. It is almost certain that this contribution to the damping will be far exceeded by the contributions from the joints which transfer bending loads and which necessarily occur in the frame and stringer flanges, and also from the other types of joint, such as the connection between a stringer and frame, where there may be a considerable transfer of shear load from one member to the other. If any effort is to be made to increase the damping of the structure by additives at the joints, it is at these latter joints and not at the stringer-skin joint that the greatest effect will be obtained. Unfortunately, it is at such joints that the worst effect on fatigue life is likely to be realised, as a result of such treatment.

5. The stiffness of riveted joints

The definition of a stiffness is, of course, the rate of change of the elastic restoring force within a system with respect to the displacement of that system. In visualizing the stiffness of a riveted joint it is more convenient to use the concept of flexibility, i.e., the reciprocal of the stiffness, which is the rate of change of displacement with respect to the elastic restoring force. When a simple, singly-jointed specimen is loaded longitudinally, its overall extension exceeds that of a similar, but unjointed specimen by virtue of the following displacements:

(a) Actual bending and shearing of the rivet or connection

(b) Bearing deformation of the rivet

(c) Elastic (and possibly plastic) deformations in both plates due to the stress concentrations around the rivet. (This includes the bearing deformation adjacent to the rivet and the elastic displacements of contact asperities).

Each of these displacements contributes to the overall extension of the jointed specimen and may be regarded as the 'components' of the joint flexibility. The flexibility of the joint may then be defined as the difference between the flexibility of a single-jointed specimen and an unjointed, but otherwise identical specimen.

The measurement of the flexibility of a joint is beset with the difficulty of having to measure the very small difference of two large quantities. In the experiments on the countersunk riveted joints, the addition of a joint to the centre of the beam caused a drop in the natural frequency of the beam. This resulted from three factors:

(a) the additional mass of the joint plate

(b) the joint flexibility

(c) the change in the mode of the vibrating beam.

It was found that a first order approximation to extract a value of the joint flexibility from the measured natural frequency drop was not sufficiently accurate, and an accurate account had to be taken of the mode change. The joint stiffness so calculated is shown in Figure 10, where it is plotted against the amplitude of the harmonic load on the joint. For a comparison, the energy dissipated per cycle by the joint is also plotted. The figure shows clearly that the characteristic rise in damping is accompanied by a marked reduction in the 'equivalent' stiffness of the joint. Such a reduction is to be expected as the slipping mechanism progresses.

Knowing the stiffness of the joint and the energy dissipated per cycle, it is possible to calculate an 'equivalent loss factor' for the joint, by defining the loss factor, η, as $\frac{1}{2\pi}$ x The energy dissipated per cycle of load $\div$ the maximum strain energy stored in the course of the cycle. This is also plotted on Figure 10. Comparing this with the values of loss factors for typical damping materials (which are usually in the range 0·5 to 1·5) they are seen to be very low. The combination of a low loss factor and a high stiffness results in the very low damping of the stringer-skin configuration already discussed, in which the rivet forms the shear connection between the two components.

The effect of prolonged loading upon the stiffness of the joint is also of interest, for as the damping decreased with time (initially very rapidly), the beam natural frequency, and therefore the joint stiffness, increased. Now increasing the stiffness of a linear joint without changing the damping coefficient means that there will be a reduction of the energy dissipated when a cycle of constant load amplitude is applied. There need be no change whatsoever in the damping mechanism itself. This provides an alternative, (though probably complementary) explanation for the drop in damping to that discussed before. Nevertheless, it still remains to account for the increase in the stiffness. This could conceivably be due to work hardening of some of the contact asperities, and perhaps to an improvement of the fixity of the rivet in the plate holes by a welding action taking place. The latter would not normally be expected to occur if oxygen had free access to the surfaces, but it is questionable whether the atmosphere could penetrate into the intimate contact areas inside the rivet hole.

It should be emphasised that the joint stiffness measurements have been evaluated from the tests on one beam only, and despite the care taken in the associated theory, certain approximations had still to be made. The experiment had not been initially designed to measure the joint stiffness, and for this purpose a modified arrangement is required in future work.

An understanding of the stiffness of a riveted joint is of greatest value (in the context of acoustic fatigue) in connection with the non-linear vibrations of panels, which are attached around the edges to relatively stiff members. In such configurations, the non-linearity of the joint stiffness may play an important part in the resonant response of the panel at high amplitudes. It is important to recognise here that the rivets would be subjected not only to the normal type of shear load, but also to an oscillating load perpendicular to the direction in which the joint was designed to be loaded. This oscillating load might well modify completely the effect of time of loading on both the damping and stiffness of the joint.

6. The damping of a bonded honeycomb configuration

The damping of a honeycomb sandwich type of structure is derived from the energy loss at the bond which is subjected to shear, from the internal hysteresis losses in the sandwich skins, and from the joints at the edges of the panels. The panel edge joints could only be tested adequately for damping when incorporated in a much larger configuration than has been used for the rivet damping tests, but the losses at the bond can be investigated on simple beam type specimens. Preliminary tests have been conducted on beams 36 in. long x 2 in. wide x $^1/2$ in. honeycomb thickness vibrating in the fundamental free-free mode. Four different beams having different skin thicknesses were used. The damping ratios obtained are shown in Figure 11 and are compared with that for the riveted stringer-skin combination of Paragraph 4. The decreasing order of the honeycomb damping as the skin thickness increases is in opposition to the simple theory relating to the damping of the bond, which predicts an increase of damping. No explanation for this phenomena has yet been found.

A comparison between the stringer-skin damping curve and those of the honeycomb beams is not strictly justifiable, as a structure made from the honeycomb of thinnest skin thickness would be considerably stiffer than the stringer-skin type. However, the most significant feature of the results is the very low values of damping in all cases; all of which are only slightly greater than the damping ratio of an unjointed aluminium alloy beam. It has already been asserted that joints carrying bending moments and bending stresses will contribute much more to the total damping of a structure than those which transmit shear stresses only (as in the riveted string-skin beam). Consequently, the maximum value from any future work relating to the intrinsic damping of a structure would appear to be obtained by giving greater attention to the joints carrying the bending loads.

7. The fatigue failure of joints under acoustic loading

7.1 The Relationship between the Damping and Fatigue Strength of a Joint

The investigations into joint damping described in Paragraph 3.2 have not yet been extended to include the failure of the joints under repeated loading, and hence to examine the relationship between joint damping and fatigue life. There is little doubt that there will be some relationship between the fatigue life and the damping, under given load conditions. The damping has been shown to be associated with a fretting action, and this is well known as a process precipitating fatigue failure. The damping is greatest when the joint stiffness is lowest and the relative displacements of the adjacent plates are greatest. This implies a more severe fretting action with the likelihood of an earlier fatigue failure, always provided that the failure is in the joint plates and not in the rivet itself. It is known that the damping capacity of the structural material increases as the limit of the fatigue life is approached, but this will probably have a negligible effect upon the damping of the joint as a whole, since the energy dissipated by internal friction within the joint material is a negligible proportion of the energy dissipated by the slipping mechanism.

Fenner, Wright and Mann (Ref. 6) and others have shown in tests on aluminium alloy specimens that even a small normal 'fretting pressure' of 20 lb. per in.^{2} has a very pronounced effect upon the fatigue life. The pressure between two joint plates has never been measured, but examination of some joint surfaces after being opened after a damping test indicates that considerably greater pressures than this must have been present. In these fretting fatigue tests, the minimum applied fatigue stress was about 9,000 lb. per in.^{2}, which is considerably greater than the average stress encountered in acoustic fatigue. It is perhaps platitudinous to remark in connection with this that in the domain of acoustic fatigue, it is the effect of a very large number of reversals of small stresses that account for the extensive damage, and it is this fact which limits the usefulness and relevance of much of the previous work.

It is possible to increase the damping capacity of a riveted joint by the insertion of a visco-elastic layer at the joint interface, (Ref.10). This immediately removes the previous damping mechanism with the associated danger of fretting fatigue at the plate surfaces. The extent to which the static strength is reduced by the elimination of the friction path, and the extent to which bending of the rivet may precipitate rivet fatigue failure must be carefully investigated before the method can be used.

7.2 The Nature of Joint Fatigue Loading Actions

Joint fatigue failures under acoustic loading seem to fall into one or other of the following categories:

(1) Failure of cleats connecting fuselage frames and stringers

(2) Failure of frame flanges attaching frames to skin and stringers

(3) Failure of stringers at stringer joint plates

(4) Cracking of panels along or close to rivet lines

(5) Failure of countersunk head rivet by the pulling off of the head

(6) Failure of the interface bond of honeycomb structures

In order to understand the nature of the loading causing these failures, it is useful also to consider certain failures which have occurred in areas away from joints in stringers, frames and tail structure ribs. These failures have often taken the form of cracks running in the direction of the <u>length</u> of the member and usually originating in the bend of the cross-section. Such cracks could only arise if high bending stresses existed due to vibratory distortion of the cross-section. Manual inspection of a fuselage under jet-noise excitation has indicated clearly that such modes of vibration were in fact being excited on both frames and stringers. Furthermore, it appeared that the wide web of a frame was vibrating in a 'panel mode', with a greater amplitude near the centre than at the edges. In addition, both the frame and the adjacent stringers were vibrating in bending modes in the plane of their webs.

The complexity of the loading on a stringer-to-frame cleat can now be envisaged (see Figure 12). The vibration of the frame in the plane of its web will impose a load F_y on the cleat, to be transferred through to the stringer. Cross-section distortion modes of the frame will impose a load F_x, and the 'panel mode' of the web a couple M_y. Cross-section distortion of the stringer imposes a load F_z. The problems of testing and defining the fatigue properties of such a component are obvious.

The failure of stringers at stringer joint plates appears to be caused simply by bending vibration of the stringer in the plane of its web or centre line. There is no difficulty in simulating the failure under normal harmonic testing procedure. Of interest here is the experience in the United Kingdom where a joint plate that had failed due to acoustic excitation was replaced by one of greater stiffness. The latter failed after a shorter time than the former. Further increases of stiffness had the same result, but when a joint plate of lower stiffness than the original was substituted, a greater fatigue life was obtained. This, of course, was not due to the fatigue properties of the joint but to the effect the different stiffnesses of the joints had on the load and moment distribution within the structure.

Cracking of panels along, or close to rivet lines can be attributed either to the bending moment at the edge of the panel due to the edge being partially fixed when the panel vibrates in bending, or to membrane tension stresses being developed under non-linear conditions at high amplitudes. The former is the more likely cause. The fatigue cracks often skirt around the edge of the countersunk rivets, though still passing very close to it. At dimpled joints, the crack then coincides with the bent edge, where the process of forming the dimple would have produced the greatest work-hardening effect, and where fretting due to the edge of the rivet head might be expected to occur. A considerable volume of literature exists on the fatigue strength of riveted joints under conventional longitudinal loading of the joint, and comparisons have been made of the merits of different types of rivet, countersinking and dimpling processes (Ref. 8). This work mostly relates to much higher stress levels than are encountered in acoustic fatigue, but at the lowest stress levels for which results are given there is usually a very wide scatter in the fatigue life. In some flexural fatigue tests on simple specimens with dimpled holes (i.e., no rivet present), at the lowest stress level of testing, 16,000 lb. per in.^{2}, the scatter in fatigue life was 50:1 (Ref. 9). These specimens were made with average manufacturing care, and not with the close control usually applied in fatigue specimen manufacture. It is most likely that an even wider scatter range would be encountered in the presence of a rivet and the possibility of fretting at the rivet head.

The failure of rivet heads has been encountered under conventional loading, and is probably due to shearing fatigue of the rivet across the neck. Under acoustic loading, the cause is more likely to be an oscillating tension load on the rivet arising from the inertia forces associated with panel modes. If panel membrane stresses are present at the same time then a shearing action may also take place on the rivet. It is conceivable that fretting at the rivet neck may play an important part in this type of failure.

Honeycomb structures are susceptible to failure of the interface bond when in very intense noise fields. The direct stress exerted upon the bond will be the vector sum of the incident sound pressure, the reflected and re-radiated sound pressure and a stress due to the inertia force on the vibrating skin surface. Furthermore, shear stresses will exist on account of the deformation of the configuration. There should be no difficulty in simulating on a small scale the fatigue failure under a pure stress normal to the surface, nor under combined direct and shear stresses.

7.3 Means of Improving the Fatigue Life of Joints

From the foregoing remarks, it will be seen that one of the major causes of acoustic fatigue failure at joints is that the joints are being loaded in a different manner or direction from that which was intended when the joint was designed. It is generally recognised that careful attention to detail design is the most effective means of prolonging the life of a structure, when the orthodox loadings are concerned. The same must be said for structures under acoustic loading. Where the loads in the unorthodox directions cannot be prevented, consideration must be given in the early design stages to the structure being well able to carry them. As an example, cracking along rivet lines can be prevented by suitably reinforcing the skin, either by bonding on a reinforcing strip, or by chemically machining the skin and leaving landings along all rivet lines. The possibility then exists of using rivets (if they cannot be dispensed with) especially designed to carry small tension loads.

The alleviation of fretting by the introduction of lubricants or anti-fretting agents at the joints will obviously be accompanied by a loss of structural damping, which in turn means an increase of vibrationamplitude and stress. The durability of these agents over a long period of time would have to be proved. The introduction of a visco-elastic interface (as remarked already) may inhibit fretting and increase the damping, and so decrease the stresses. The effect on the rivet fatigue life under shear must be investigated.

8. Conclusions

The tests on the damping of riveted joints indicate that at low rivet loads the magnitude of the energy dissipated at a joint is reproducible from one joint to another. The joints carrying bending loads are deemed to contribute most to the damping of a whole structure. It would be of very great value to measure the damping of some typical modes on a full-scale structure, but the difficulty must be recognized of exciting pure modes of a system with many close natural frequencies, and unless pure modes are excited no damping measurements can be taken. If such measurements are ever attempted, it is necessary to remember that the damping of highly loaded riveted joints will vary with the duration of loading. This variation will also be important when the structure is subjected to prolonged periods of acoustic excitation. The magnitude of the acoustic radiation damping of the modes compared with that of the structural damping is the most important factor in assessing the value of the study of the intrinsic structural damping.

Acoustic failure of joints is considered to be due to their being loaded in directions different from that for which they were initially designed. Joints which contribute most to the damping may be most susceptible to fretting fatigue. Improvement of the fatigue life should be sought by careful detail design, together with the possibility of using visco-elastic layers at the joint interfaces which will both increase the damping and reduce the susceptibility of the joint plate to fretting fatigue.

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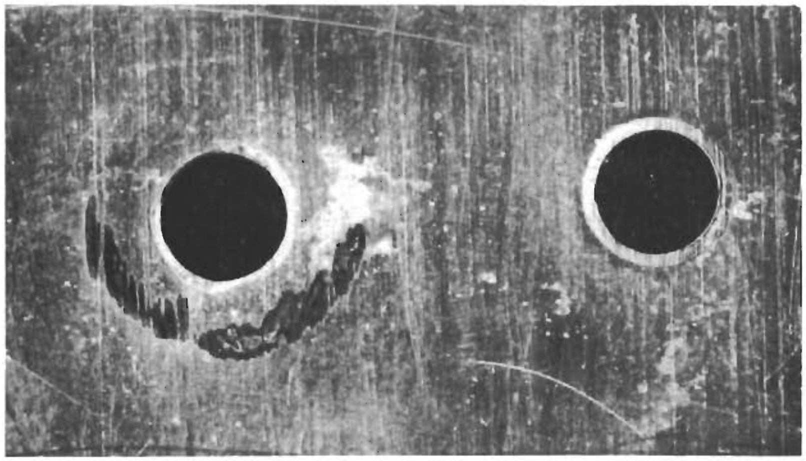

Fig. 1 The Contact Areas Around a Well-Formed Rivet

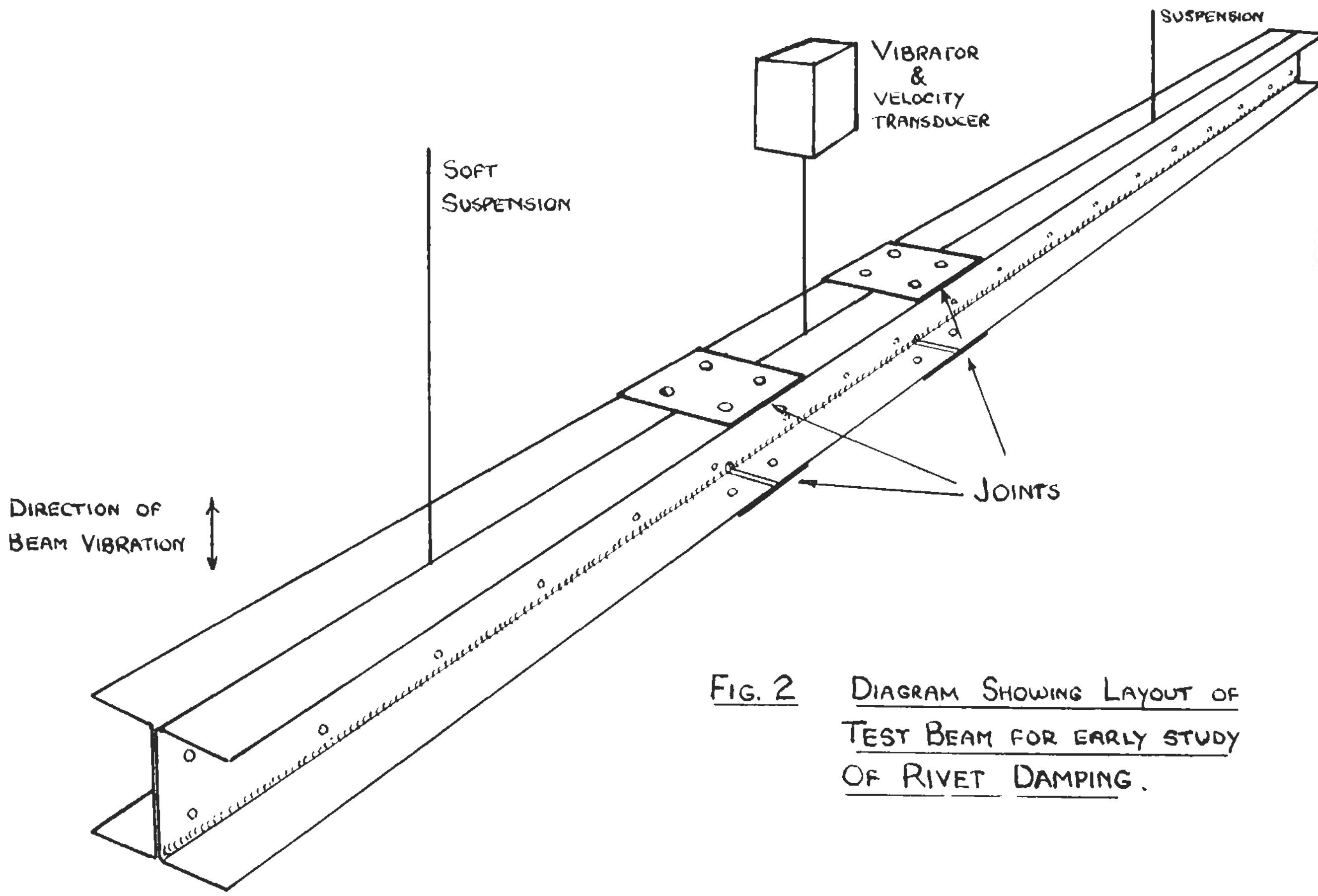

Fig. 2 Diagram Showing Layout of Test Beam for Early Study of Rivet Damping.

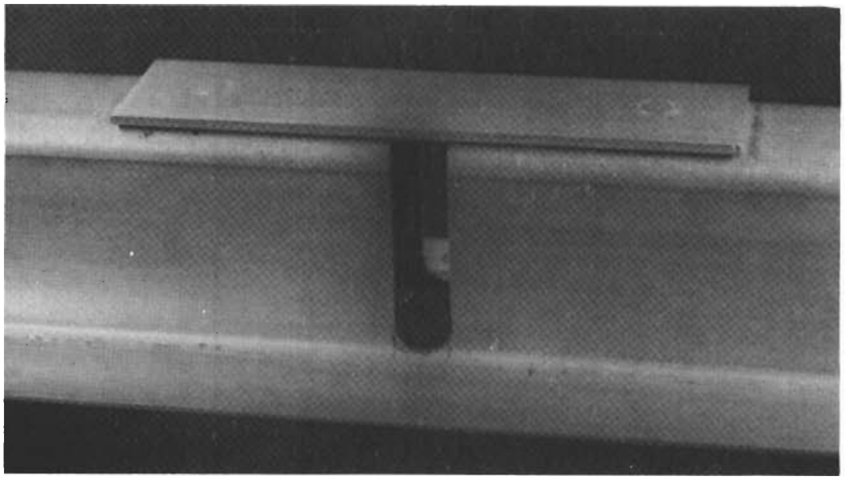

FIG. 3 THE COUNTERSUNK RIVETTED JOINT IN THE TEST BEAM

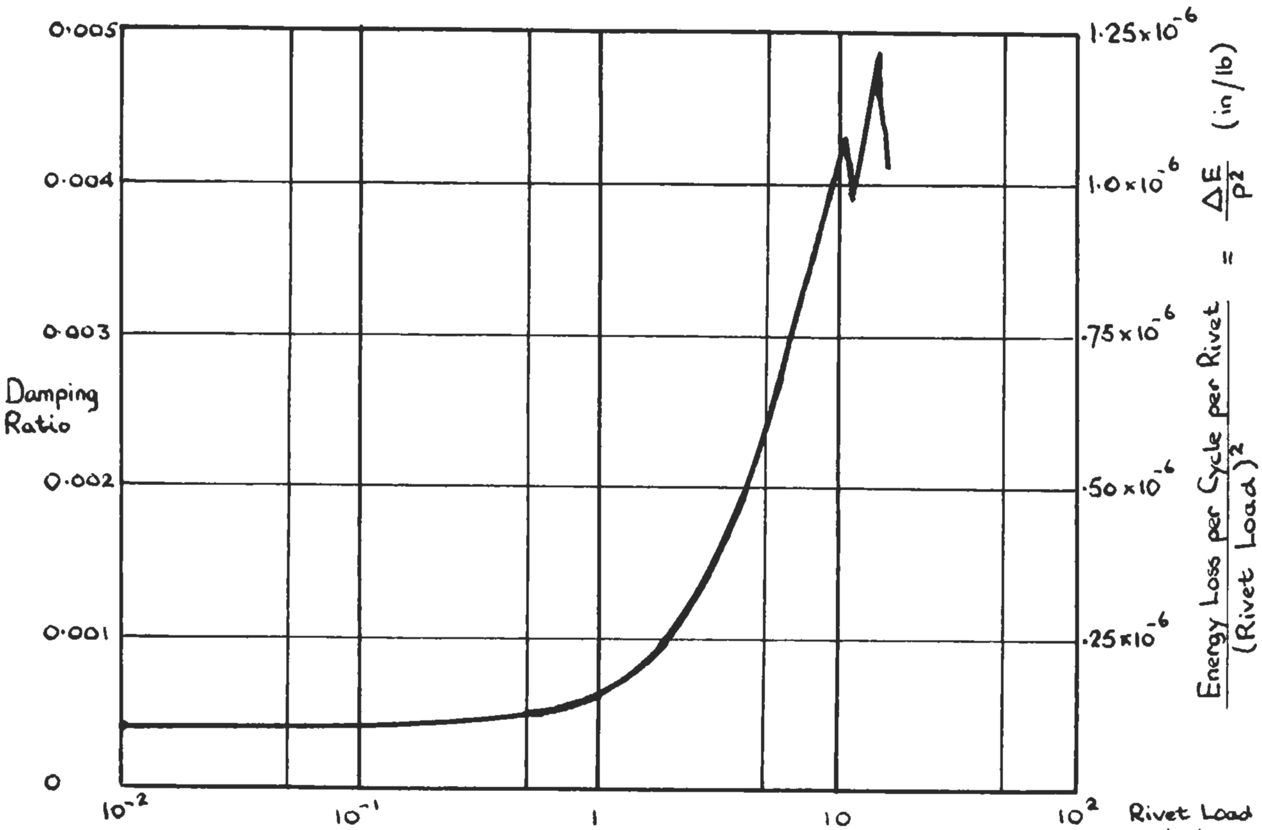

FIG. 4. THE VARIATION OF THE DAMPING OF A JOINTED BEAM WITH RIVET LOAD AMPLITUDE.

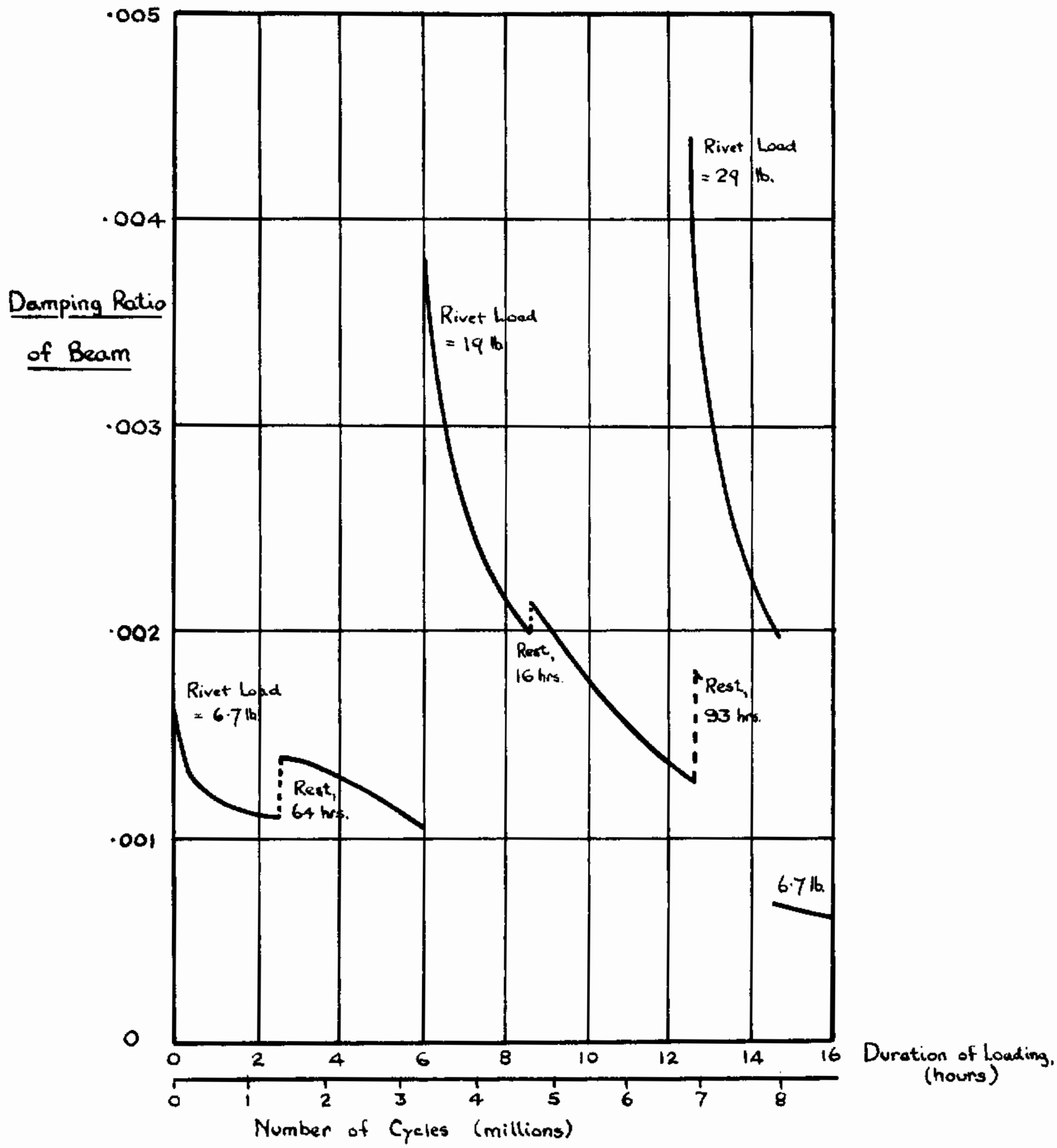

FIG.5 VARIATION WITH TIME OF THE DAMPING OF A BEAM CONTAINING A RIVETED JOINT.

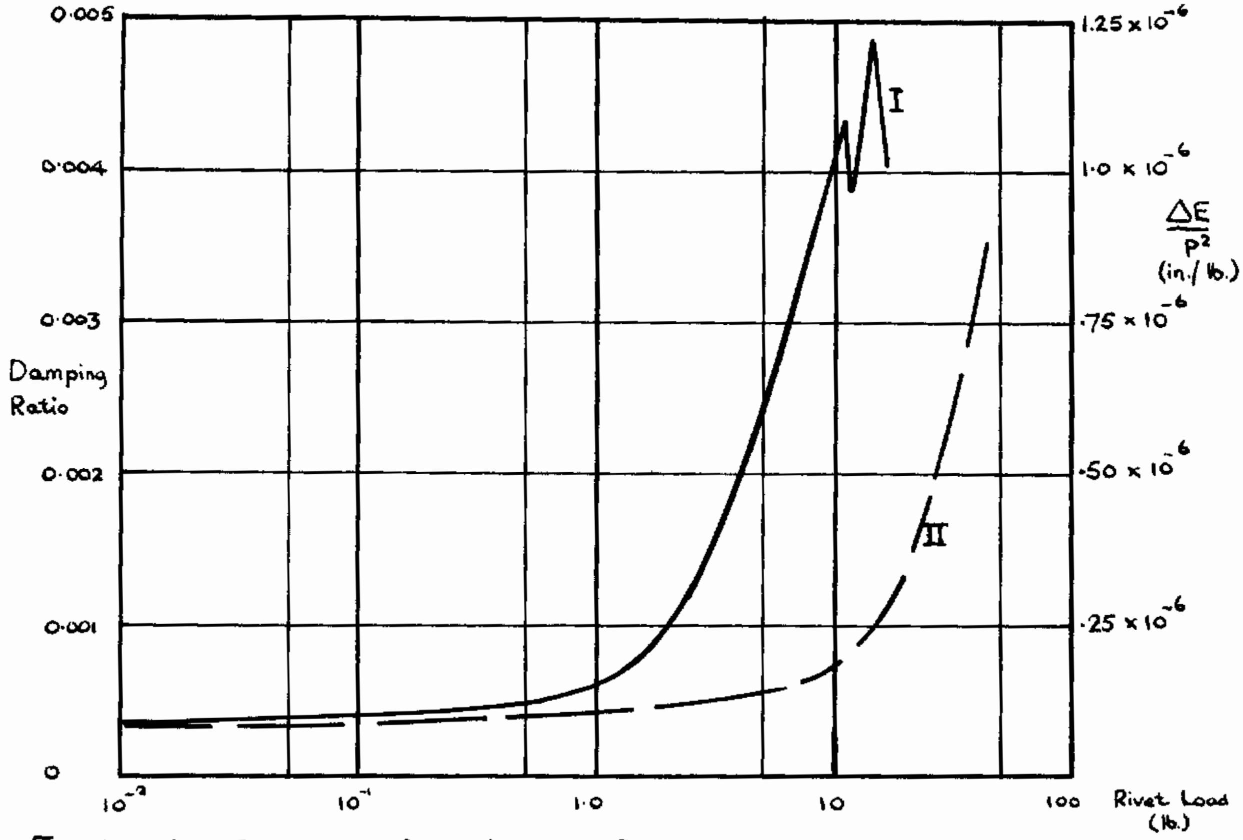

Fig. 6 The Damping of a Jointed Beam before and after prolonged loading.

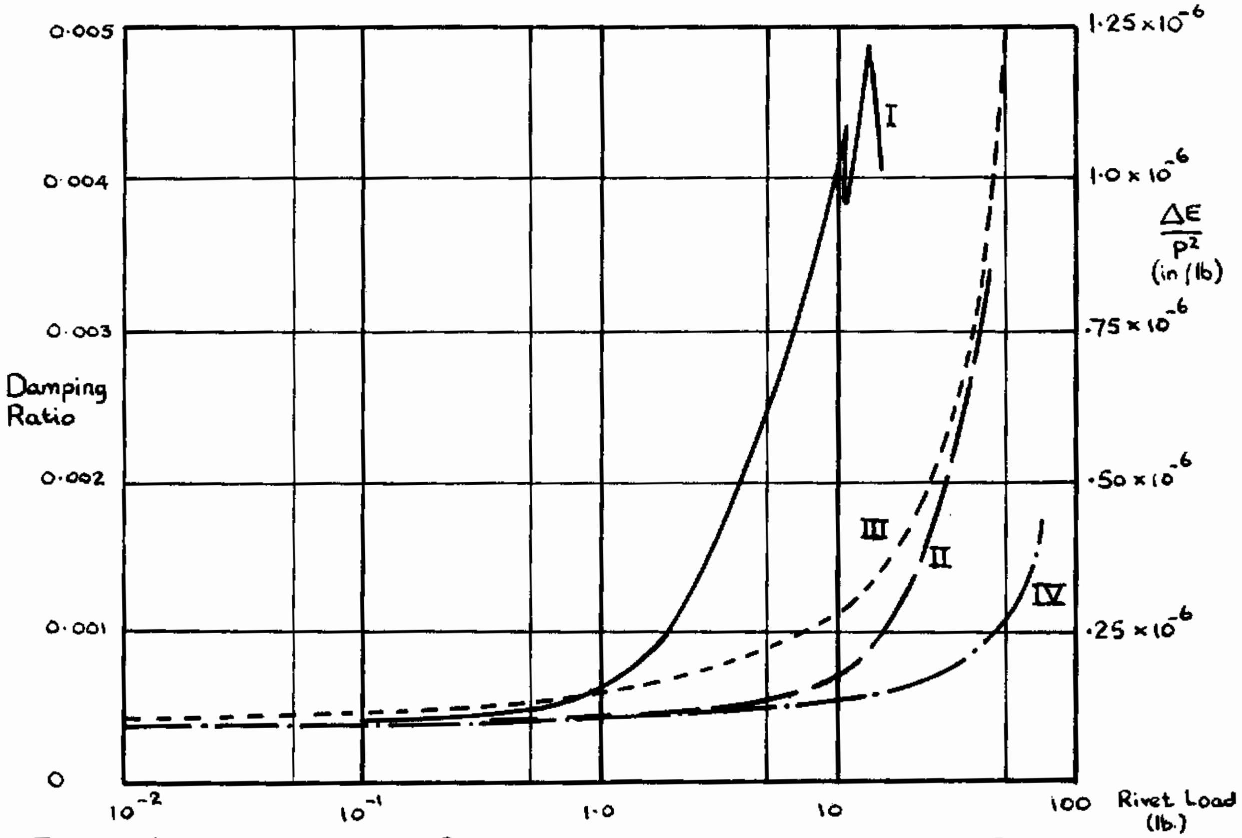

Fig. 7 Comparison of the Damping of two identically Jointed Beams.

FIG. 8 CONTACT AREAS AND FRETTING OXIDATION IN COUNTERSUNK RIVETED JOINT.

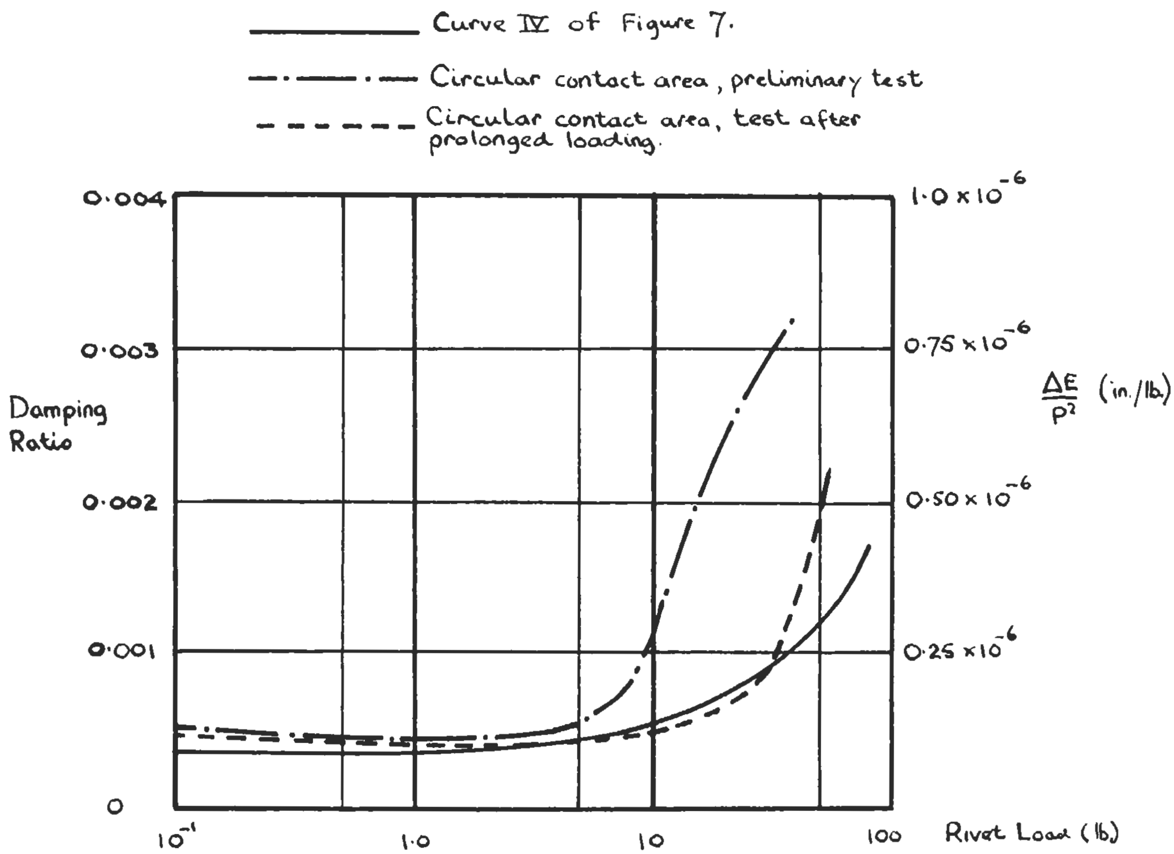

FIG. 9 COMPARISON OF THE DAMPING OF IDENTICALLY JOINTED BEAMS WITH DIFFERENT CONTACT AREAS.

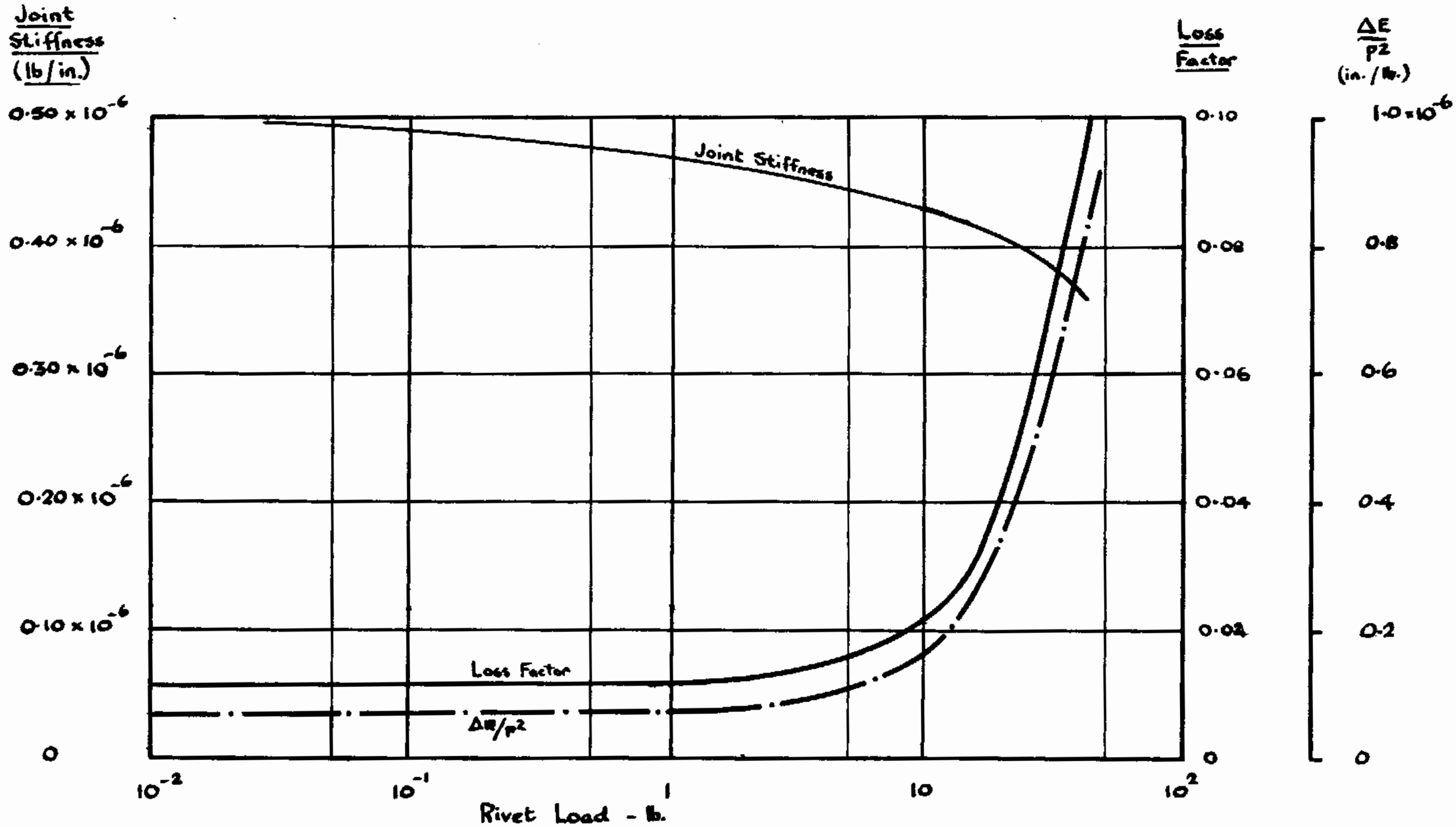

FIG. 10. THE VARIATION WITH LOAD AMPLITUDE OF THE JOINT STIFFNESS, LOSS FACTOR AND ENERGY DISSIPATED PER CYCLE.

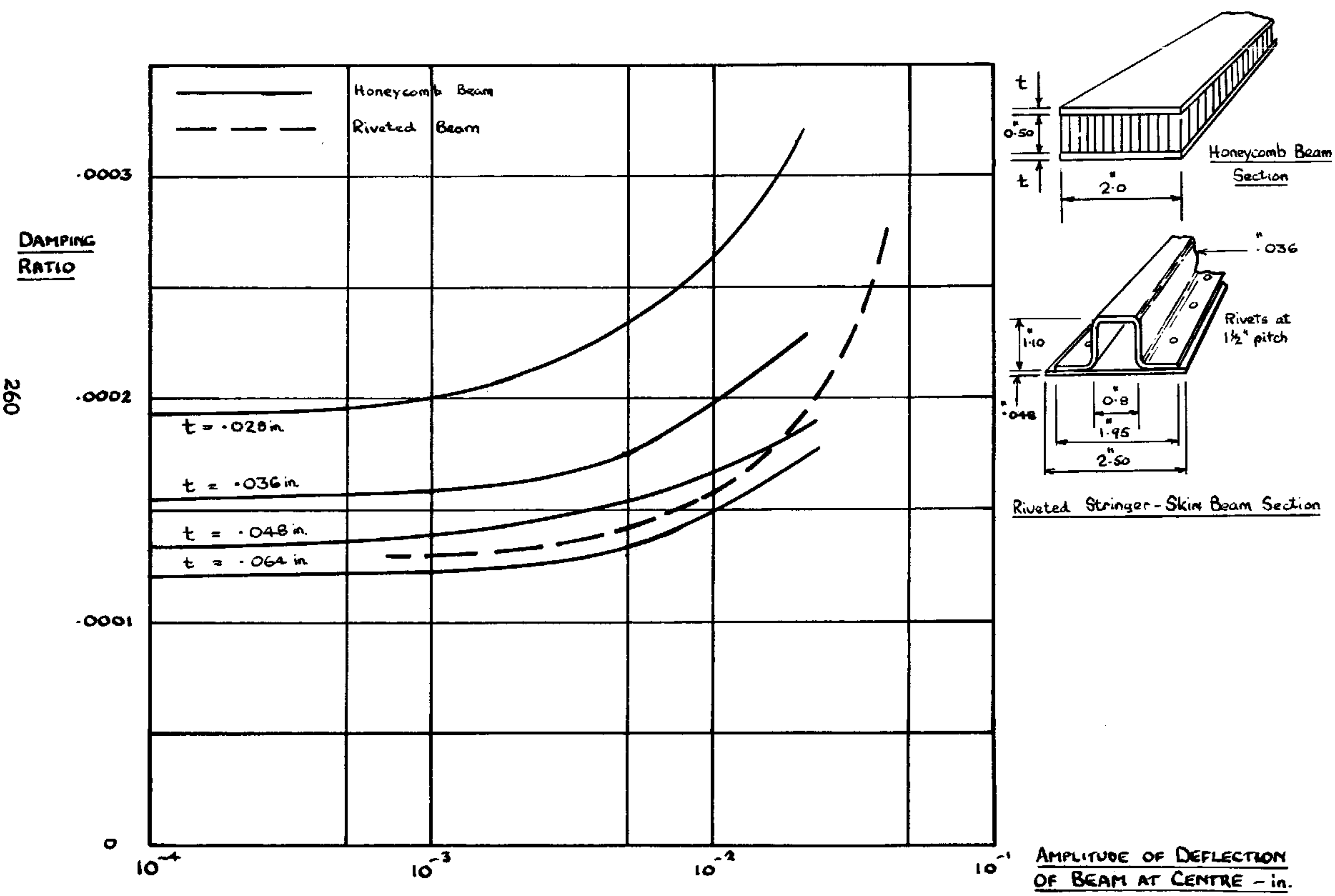

FIG. 11 DAMPING RATIOS OF HONEYCOMB SANDWICH & RIVETTED BEAMS

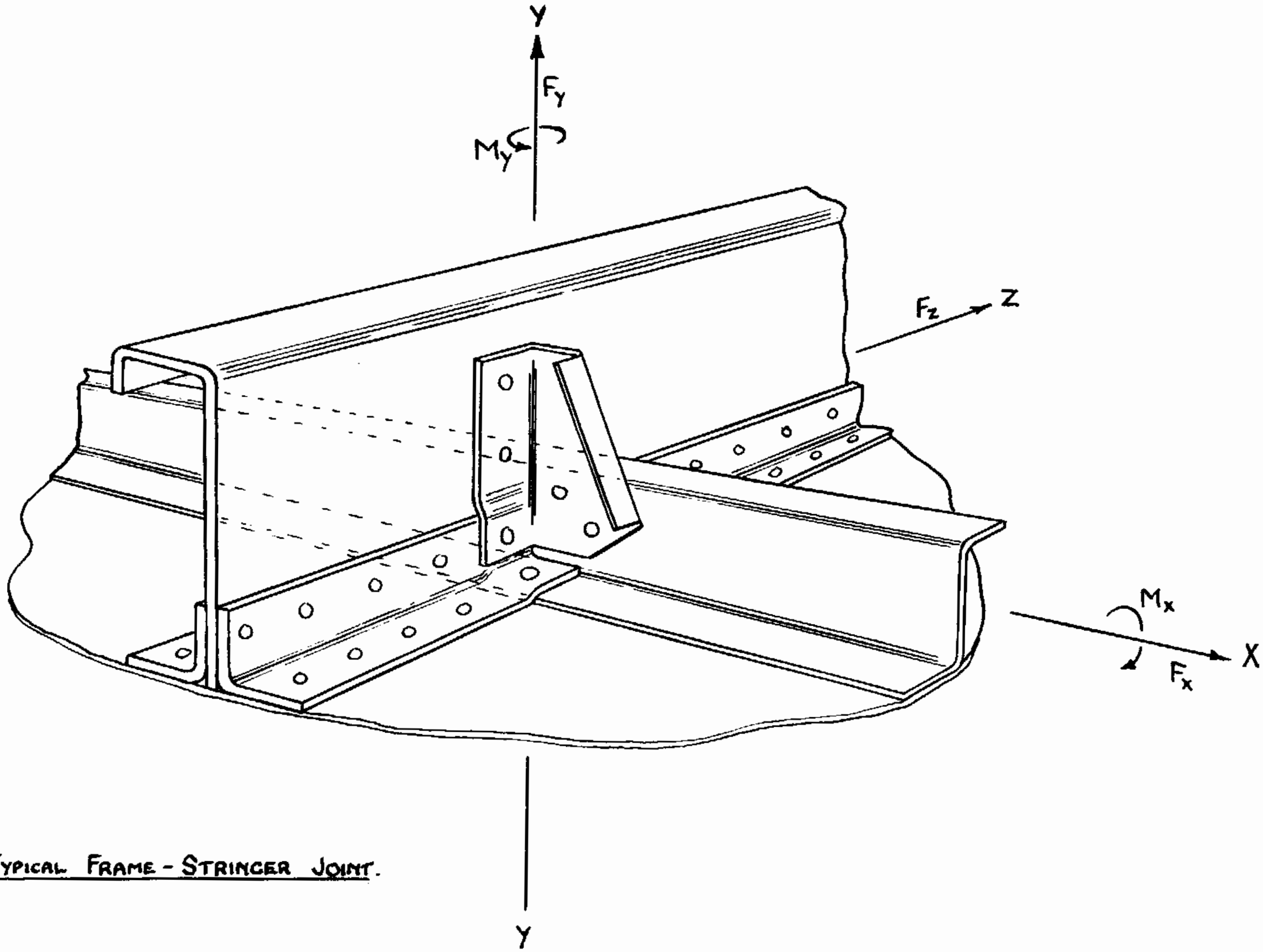

FIG. 12 TYPICAL FRAME - STRINGER JOINT.