velocity range of the Charpy machine (up to approximately 20 fps), the energy values agreed.[7]

The integration of the stress-strain curve furnished the same energy values as the pendulum method, and this method should furnish the correct energy value regardless of the rigidity of the pendulum, of the tup, or of the flywheel.

The agreement of the energy values obtained by the pendulum method with the energy values of the Charpy machine, and with the energy values obtained by integration seems to indicate the adequacy of the pendulum method within the checked velocity ranges.

The discussion contained the suggestion to measure the stress in a section near the pendulum, and near the tup. This has been done, and the curves of Fig. 2(*a*) and (*b*) of the paper give the values obtained from these measurements.

The discussion contained the further suggestion to locate the strain gages within the test section (Section II of Fig. 1 of the discussion).

This has indeed been tried, but it was found rather impractical since plastic flow occurred within this section. Strain gages attached within this section became inoperative before fracture occurred. It was then tried to measure the strain in a direction perpendicular to the axis of the specimen. This, however, made the translation of the strain records into stress values rather complex and even doubtful, hence the method described in this paper seemed to be preferable.

The Effects of Web Deformation on the Torsion of I-Beams[1]

B. J. Aleck.[2] This significant paper introduces the concept of relative deflection between points of the web and flanges of a thin-webbed I-beam, loaded torsionally. As already mentioned by the authors, an important extension of the work concerns the correction which it introduces to the theory for lateral buckling.

It is to be noted that an I-beam is an inefficient member torsionally, and therefore would seldom be used in aircraft structures where this loading is present to any great extent.

A similar correction, however, arises in the case of the closed box beam. The box beam forms the basis for wing design and for the design of many torsionally resistant structures. In a two-spar wing, for example, a box is formed by the upper and lower wing surfaces and by the front and rear spars. The practical problem is somewhat complicated by the presence of vertical stiffeners and regularly spaced cutouts in the spar webs. An indication of the importance of the web deflection on the torsional stresses could be obtained, however, even with these factors neglected.

The importance of the general problem of the distortion of cross sections under load may be illustrated by the results of tests designed to measure the torsional stiffness of a complete wing structure. The displacements of at least four chordwise points per cross section are usually measured. To find the angle of twist of a cross section which does not deform, it is unimportant which two points of the cross section are used; any two will give the same angle. The test results, however, usually indicate different angles of twist, depending upon which set of points is considered as representative of the cross section. This is evidence that the cross section of actual structures do deform under torsional loading.

R. L. Templin.[3] The theoretical analysis presented by the authors regarding the behavior of I-beams under torsional forces would be expected to apply more closely to the actual behavior of such members when the material of the members is stressed within the elastic range. In some structures, however, it may be desirable to know more about the behavior of such members in the plastic range.

In certain structural research work being carried out in the Aluminum Research Laboratories, investigating the behavior of aluminum-alloy I-beams, H-sections, channels, and zees under torsional loads up to point of rupture, some rather interesting behaviors in the plastic range have been observed. In Fig. 1 of this discussion are shown some of the specimens after testing to failure. Reading from the top down, the first four specimens are I-beam or H-beam sections (flange width 2 in., depth $2^1/_2$ in., webs and flanges $^1/_8$ in. thick), while the bottom specimen is a Z-section made by removing diagonally opposite flanges from the H-section. In these specimens, the ratio of outstanding flange width to thickness is $7^1/_2:1$ and the ratio of the clear depth of the web to its thickness is 18:1.

Material from these experimental aluminum-alloy sections had a tensile strength of 75,900 psi, a yield strength (0.2 per cent offset) of 69,500 psi, and an elongation in 2 in. of 17.9 per cent.

In testing these specimens, initial failures tend to occur at the ends of the grips in the form of short buckles in the outer edges of the flanges. Subsequently rupture occurs in these buckles. The triangular deformation pattern in the webs persists throughout the specimens of the proportions indicated. It may be noted also that a secondary type of failure occurs by longitudinal fracture in the center of the flange, after the flanges have bent inward

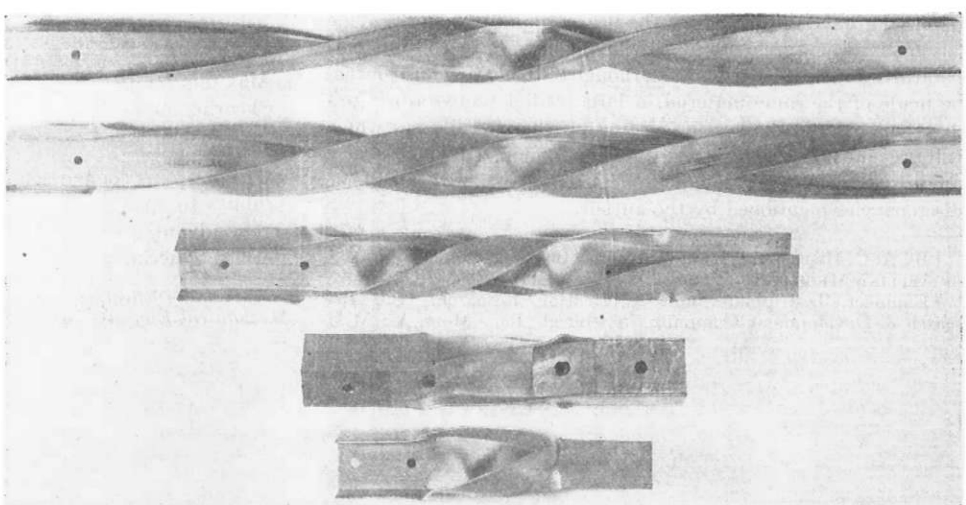

Fig. 1 Aluminum-Alloy Structural Shapes Tested in Torsion

[7] "The Effect of Strain Rate Upon the Tensile Impact Strength of Some Metals," by E. R. Parker and C. Ferguson, Trans. American Society for Metals, vol. 30, 1942, pp. 68–85.

[1] By J. N. Goodier and M. V. Barton, published in the March, 1944, issue of the Journal of Applied Mechanics, Trans. A.S.M.E., vol. 66, p. A-35.

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over the web. Such fracture may be seen in the middle specimen of the group shown in Fig. 1.

The triangular pattern, obtained in the plastic deformation of the webs of these specimens, reveals the effect of compressive stresses in the web and suggests that in some cases it may be necessary to consider the possibility of web buckling within the elastic range.

Authors' Closure

The authors are indebted to Messrs. Aleck and Templin for their valuable discussions. An analysis similar to that made in the paper could be made for thin-webbed box beams and would undoubtedly account for some distortion of section not only near the root but wherever the twisting moment exerted by the loads varies along the axis. If open section conditions should exist, as at a cutout in a wing made, for instance, to accommodate landing gear, they will frequently be confined to a short length, and then the effect of the cross-section deformation will probably be significant, as Figs. 14, 15, 16, and 18 of the paper indicate.

Mr. Templin's account of the twisting to failure of various sections demonstrates very effectively the existence of compression in the flanges, and in addition the curious triangular buckling pattern of the web. In the severely twisted beam, if the central axis is unstretched, the fibers along top and bottom of the web are considerably extended on account of their helical form. In fact of course the middle part will be put in compression and the stretch of the outer parts reduced so as to have zero longitudinal resultant force—provided the grips permit axial movement freely. The compressive stress in the middle may be responsible for the buckling. It is not at all unlikely that this buckling is a large deflection phenomenon as the hexagonal buckling of the compressed cylindrical shell is known to be.

Heat Effects in Lubricating Films[1]

P. G. Exline.[2] The author has made an important contribution to the thermodynamics of lubricating films in this paper. Although the conditions of operation are rarely encountered in engineering applictions, the results are undoubtedly of use for lightly loaded, high-speed bearings. Certainly the concentric bearing lends itself most readily to analysis and experimental confirmation.

It is suggested that, with an expression for the variation of viscosity across the film available, the next step would be to derive an expression for the flow of oil from the central circumferential groove to the ends of the bearing. This would provide a sensitive measurement of h which will vary with the temperatures and heat losses. Even though both bearing and journal be made of the same material, a large radial temperature gradient will exist in the bearing setting up thermal stresses which will prevent its expanding as rapidly as the journal which is at a uniform temperature throughout. This may explain some of the discrepancies mentioned by the author.

From time to time the writer has been confronted by the question of the existence of turbulence in the oil film of a bearing. The agreement of the experimental work with the analysis, based on viscous behavior, would indicate that turbulence would not be encountered in range of surface velocities and clearances covered by these tests.

Author's Closure

It should be pointed out that internal heating of the lubricating film of a bearing is approximately the same whether the bearing is loaded or unloaded. While the formulas for shearing stress are primarily applicable to concentric or unloaded bearings, the results should emphasize that the maximum shearing stress, and hence the maximum load, of a bearing is diminished by film heating.

Mr. Exline suggests that differential thermal expansion of the journal and bearing would alter the film thicknesses, and this is of course true; however, the temperature of the journal and the temperature distribution in the bearing were known and the necessary corrections were made. It appears that the basic measurements of bearing and journal diameters are the major source of error in the accepted film thicknesses.

In the oral discussion at the meeting Mr. Hersey asked for a comparison of Mr. Kingsbury's graphical method[3] and the present formulas. Considering an example which Mr. Kingsbury used to illustrate his method we have the following data in the dimensions of inches, pounds, seconds, and deg F.

Surface speed of the journal, $U = 2000$
Temperature of the journal surface, $\theta_2 = 140$
Temperature of the bearing surface, $\theta_1 = 100$
Conductivity of the oil (avg), $K = 0.0163$
Viscosity of the lubricant at 140 deg, $\mu_2 = 1.50 \times 10^{-6}$
Viscosity of the lubricant at 100 deg, $\mu_1 = 3.29 \times 10^{-6}$

Using the viscosity data at temperatures θ_1 and θ_2 to evaluate the constants in the logarithmic viscosity formula we can readily calculate the results of column 1 in the table. On the basis of the maximum film temperature θ_0 we might make a more appropriate choice of temperatures for evaluating the constants in the viscosity formula, say, 115 F and 150 F, and we get the more accurate results of column 2. Column 3 of the table gives Mr. Kingsbury's corresponding results. It can be noted that the percentage difference between the results of the formulas and Kingsbury's graphical method are about 1 per cent or less.

Table

Quantity	Formulas		Kingsbury
	(1)	(2)	(3)
Max film temp, θ_0	165.6	166.5	166.6
Shearing stress, S, ($h = 0.0010$ inch)	2.48	2.54	2.568

The foregoing comparison illustrates that the advantages of the formulas in speed and convenience of calculation, and the inherent advantages of the analytical method are not gained at the sacrifice of accuracy.

[1] By A. C. Hagg, published in the June, 1944, issue of the Journal of Applied Mechanics, Trans. A.S.M.E., vol. 66, p. A-72.

[2] Engineer, Instruments and Lubrication Research, Gulf Research & Development Company, Pittsburgh, Pa. Mem. A.S.M.E.

[3] "Heating Effects in Lubricating Films," by Albert Kingsbury, *Mechanical Engineering*, vol. 55, 1933, pp. 685–688.