Flexural Pivots for Space Applications

F. A. Seelig
The Bendix Corporation Fluid Power Division
Utica, New York

Design considerations and problems encountered in fabricating flexural pivots for use as structural members are discussed. Applications described include engine mounts, camera telescope mounts, missile support systems, gimbal assemblies, and trunnion pivots.

I. Introduction

Flexural pivots have been successfully applied in space programs by many NASA contractors and other agencies. It is intended here to discuss some of the design considerations and problems encountered in fabrication of pivots for those applications where flexural pivots are utilized as structural support members (as opposed to those used in instrument sensing devices). Typical applications are engine mounts (*Lunar Orbiter*), camera telescope mounts (ATS[1] satellites), support system (Transtage Engine, *Titan III C* Missile), support system (ERNO Third Stage Engine, ELDO Missile), gimbal assembly (Directional Control Test Missile Test Stand), and the trunnion pivots (X-ray Telescope *Apollo* Applications Program, AAP).

These applications will be discussed in the sequence mentioned above to show the interrelationship of one design to another and how improvements of one design were subsequently incorporated in another.

II. Basic Components of Flexural Pivots

All flexural pivots consist of three basic elements: flexures, core or inner housing, and mounting means or outer housing. The heart of any flexural pivot is the flexure; it must have first priority of design consideration. The configuration and the orientation within the assembly with respect to the major force vectors applied must be optimized for a particular specification. The core, or inner housing, is the member to which flexures are directly attached. It fixes the orientation of flexures while providing reinforcement of the flexures. It is fastened to the outer housing, which is the mounting member for the complete, self-contained pivot.

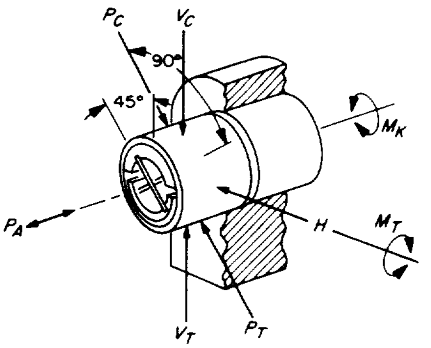
---

[1] *Applications Technology Satellite.* See also "Mechanical Design of the Spin-Scan Cloud Camera," by D. T. Upton, in these *Proceedings.*

III. Flexure Designs

Normally one thinks of flexures in terms of flat rectangular pieces of spring steel cut from lengths of rolled sheet stock. This is the simplest form that can be fabricated and stress-analyzed. The means by which to use these simple members most efficiently is the challenge to the designer. What behavior patterns do they follow when combined and used in pairs or sets of various dimensions? How are their load-carrying capabilities affected by force vector combinations? What conditions of load are design limitations?

In order to consider flexure design, one must first decide what is required of the assembly with respect to its structural capacity. As a place to start, let's consider that there are six fundamental force vectors and two basic moments shown on Fig. 1. All force vectors here are assumed to be of equal magnitude. The flexure proportions for various loading conditions may be related to each other approximately as shown in Fig. 2 with respect to length, width, and thickness. Orientation is related to the force vectors. For example: if only a P_C is considered (Fig. 2c), it can easily be seen that one relatively thick flexure will suffice. If the direction of the load relative to the flexures is as shown in Fig. 2e, the tension flexure may be relatively thin and the compression member somewhat thicker (to prevent buckling). Similarly, the flexure thickness of Fig. 2d is greater than that of Fig. 2b. (The flexure configurations for individual load conditions shown here are not realistic, since it would indeed be unusual to find such simple load systems for a given application.)



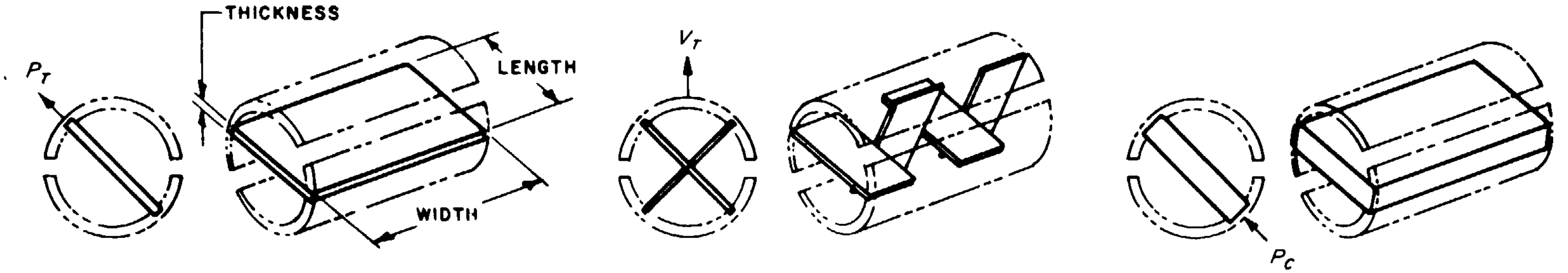
V_C VERTICAL FORCE SHARED BY ALL FLEXURES IN COMPRESSION
P_C ALL FORCE CARRIED BY ONE FLEXURE (OR PAIR IN SAME PLANE) IN COMPRESSION
H HORIZONTAL FORCE PUTS ONE FLEXURE (OR PAIR IN SAME PLANE) IN COMPRESSION
P_A AXIAL FORCE PUTTING ALL FLEXURES IN SHEAR
V_T VERTICAL FORCE SHARED BY ALL FLEXURES IN TENSION
P_T ALL FORCE CARRIED BY ONE FLEXURE (OR PAIR IN SAME PLANE) IN TENSION
M_T MOMENT APPLIED TO BEND PIVOT AXIS (NOT INCLUDING OVERHUNG MOMENT FROM TRANSVERSE FORCES ABOVE)
M_K MOMENT APPLIED TO ROTATE PIVOT AND CAUSE BENDING OF FLEXURES

RADIAL FORCES SHOWN TO BE CONSIDERED IN PLANE OF M_T AXIS
ILLUSTRATION SHOWS ORIENTATION WITH FLEXURE ANGLES ONLY

Fig. 1. Primary force vectors, flexural pivot design

More realistically, most applications require an investigation of complex combinations of these force vectors. For example, let's assume that flexures must be designed to support a small variable-thrust engine in the third stage of a missile. The engine, supported on a gimbal ring which is not completely rigid, is held in a fixed position by locked-up actuators during the launch phase, with its center of gravity off the centerline of its thrust vector. Also assume that the engine must be fired and gimbaled through maximum operating angles at full thrust. It is obvious that the force vectors in this application require careful study to determine which force combinations are applied simultaneously and cause the most stress on the flexures. These combined loads set the design limits.

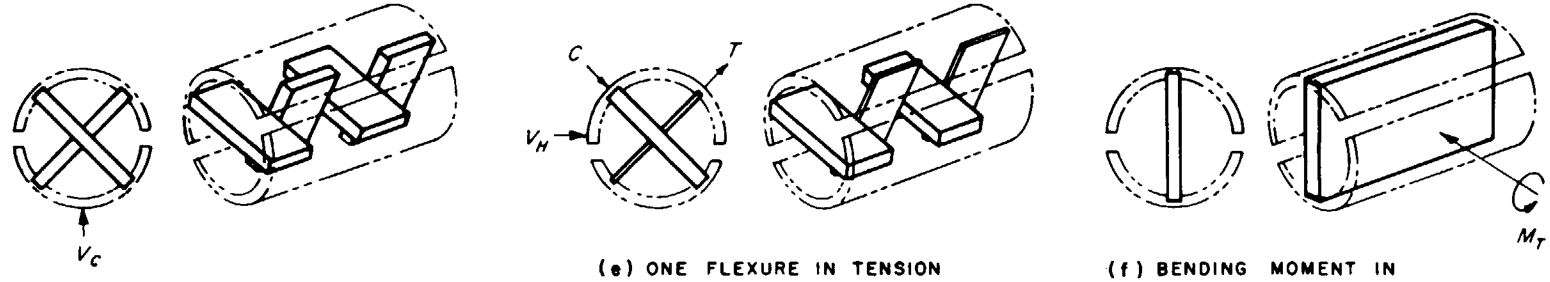
In order to consolidate the infinite number of configuration possibilities, three basic concepts will be considered, each permitting variations of flexure length, width, and thickness for design adjustment. These basic arrangements are shown in Figs. 2g, h, and i, respectively: the 90-deg symmetrical universal system, the 60-deg symmetrical nonuniversal system, and the "plus design" or unsymmetrical system. The latter uses the wide flexure as the main load-carrying member, with the narrow flexures as stabilizing elements. It has been our experience that most pivot applications for space structures can be satisfied with these three concepts. It is not intended to imply that other configurations of flexures and pivot concepts are not to be considered, but only that these form a simple, realistic design approach. We normally use the formulas developed in papers by Fred Eastman (Refs. 1–3), W. H. Wittrick (Ref. 4), Henry Troeger (Ref. 5) and others which deal with simple flat flexures. There are other possibilities, such as triflexures, cruciforms, contoured cross sections, and wire. All of these can be designed for specific requirements, but they also have limitations which may or may not be desirable. It is not intended to discuss these at this time.



(a) ONE FLEXURE IN TENSION

(b) TWO FLEXURE PAIRS IN TENSION

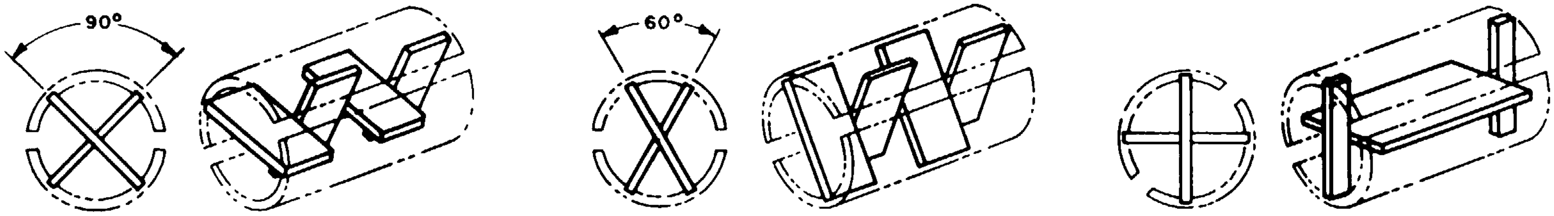
(c) ONE FLEXURE IN COMPRESSION



(d) TWO FLEXURES IN COMPRESSION

(e) ONE FLEXURE IN TENSION
ONE FLEXURE IN COMPRESSION

(f) BENDING MOMENT IN
MAJOR DIMENSION



(g) 90° SYMMETRICAL UNIVERSAL

(h) 60° SYMMETRICAL NONUNIVERSAL

(i) PLUS DESIGN UNSYMMETRICAL

Fig. 2. Flexure configuration and orientation with respect to load

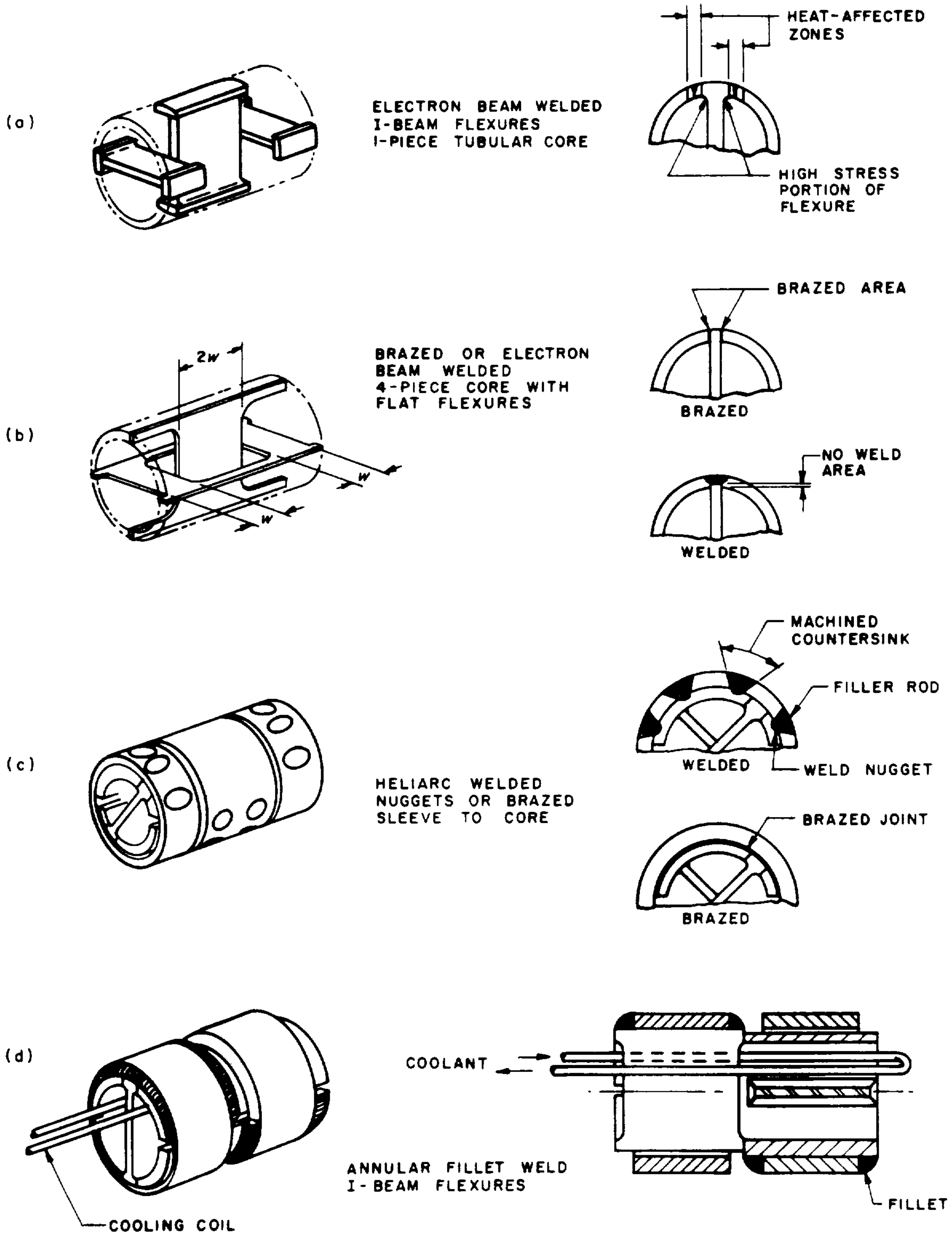


Fig. 3. Fabrication of brazed and welded assemblies

IV. Core or Inner Housing

The inner pivot housing or core can be as important for structural integrity as the flexure. Flexural pivots can be designed so that the forces are applied with the housing acting either as a cantilever beam or as a double-end-supported beam. The whole concept of mounting must be considered, particularly if weight or radial stiffness limitations are involved. The thickness of the housing depends on the method of attachment of the flexures and must be reasonably proportional to the flexure configuration. As a rule of thumb, we have used a thickness at least equal to flexure thickness for welded construction and at least 1½ to 2 times the flexure thickness for brazed construction (Fig. 3). This assumes the braze joints to be extended beyond the individual flexure widths, as shown in Fig. 3b. The core may be rectangular, square, octagonal, or round to suit the mounting conditions. The round tubular shape seems to have the greatest advantage for utilization of material and lends itself to simple machine operations such as turning, boring, and centerless grinding. It also permits the use of tubing, which reduces material cost and machine time. This type of housing may also be made from die-formed cylindrical segments or quadrants, brazed or welded together with the flexures. The housing configuration may be adjusted to suit the fabrication techniques intended.

V. Mounting Member, Outer Housing

This member of the flexural pivot is primarily an adapter to fit the flexures included in the core assembly to the other hardware. In some cases, a heavy core can incorporate tapped holes, studs, or partial flanges, eliminating the need for the outer member. The cylinder outer sleeve provides a more readily adaptable member for mounting without complicating the fabrication of the core assembly. It can be easily modified to include flanges or mounting lugs. The round shape has the same machining advantages as the core.

VI. Fabrication

A variety of fabrication methods are utilized in making a pivot assembly. Joining processes demand careful consideration in the design configuration of the basic components. For example, if temperature and load conditions are moderate, pivots of brazed construction with flat flexures and quadrant-type housing are adequate (Fig. 3b). However, if the design is likely to carry high transverse loads creating high flexure stresses, I-beam flexures with a one-piece tubular core are a more practical and more reliable design (Fig. 3a). This permits welding and keeps the heat-affected zone of the weld away from the highly stressed portion of the flexures.

There are several possibilities for attaching the outer housing or sleeve to the core. One method is by countersinking through holes from the outside of the sleeve, welding the inner edge of the countersunk holes to the outer surface of the core, and then filling the countersunk holes with filler material (Fig. 3c). This involves good welding technique to avoid overheating the flexures. Another method is to braze the sleeve and core together (Fig. 3c). A third method, utilizing annular fillet welding at each edge of the outer housing, has proved to be the most reliable (Fig. 3d). The illustration shows four separate welds on a cantilever pivot. The double-end type requires two additional welds on the third section of the outer sleeve. This method uses a heat sink or cooling method to avoid overheating the flexures.

VII. Applications

The foregoing paragraphs have covered structural considerations and fabrication methods employed to attain the structural integrity of flexural pivots. These are not the only problems involved in pivot design, as will be shown by discussion of applications in the following paragraphs. Pivots for these are depicted in Fig. 4.

The first two applications involve standard off-the-shelf-type pivots which, incidentally, are a slightly modified version of the 90-deg symmetrical universal system flexures in Fig. 2g. Instead of having two pairs of equal flexures, this design has one pair of two equal flexures on the outboard ends of the housing and one double-width flexure on a 90-deg intersecting plane in the center. The flexures are flared out as they enter the brazed joint in a quadrant-type housing, as shown in Fig. 3b. This gives higher axial capacity, higher transverse stiffness, and a slightly higher torsional stiffness than the simpler configuration.

The *Lunar Orbiter*[2] utilized a cantilever pivot ⅝ in. in diameter by 1 in. long for the gimbal support system of a 100-lb thrust engine. Reports from the Boeing Company and NASA confirm excellent performance of the flexural pivots in all five *Lunar Orbiter* flights. These

---

[2] Work on this program was conducted for the NASA Langley Research Center under NASA Contract NAS 1-3800.

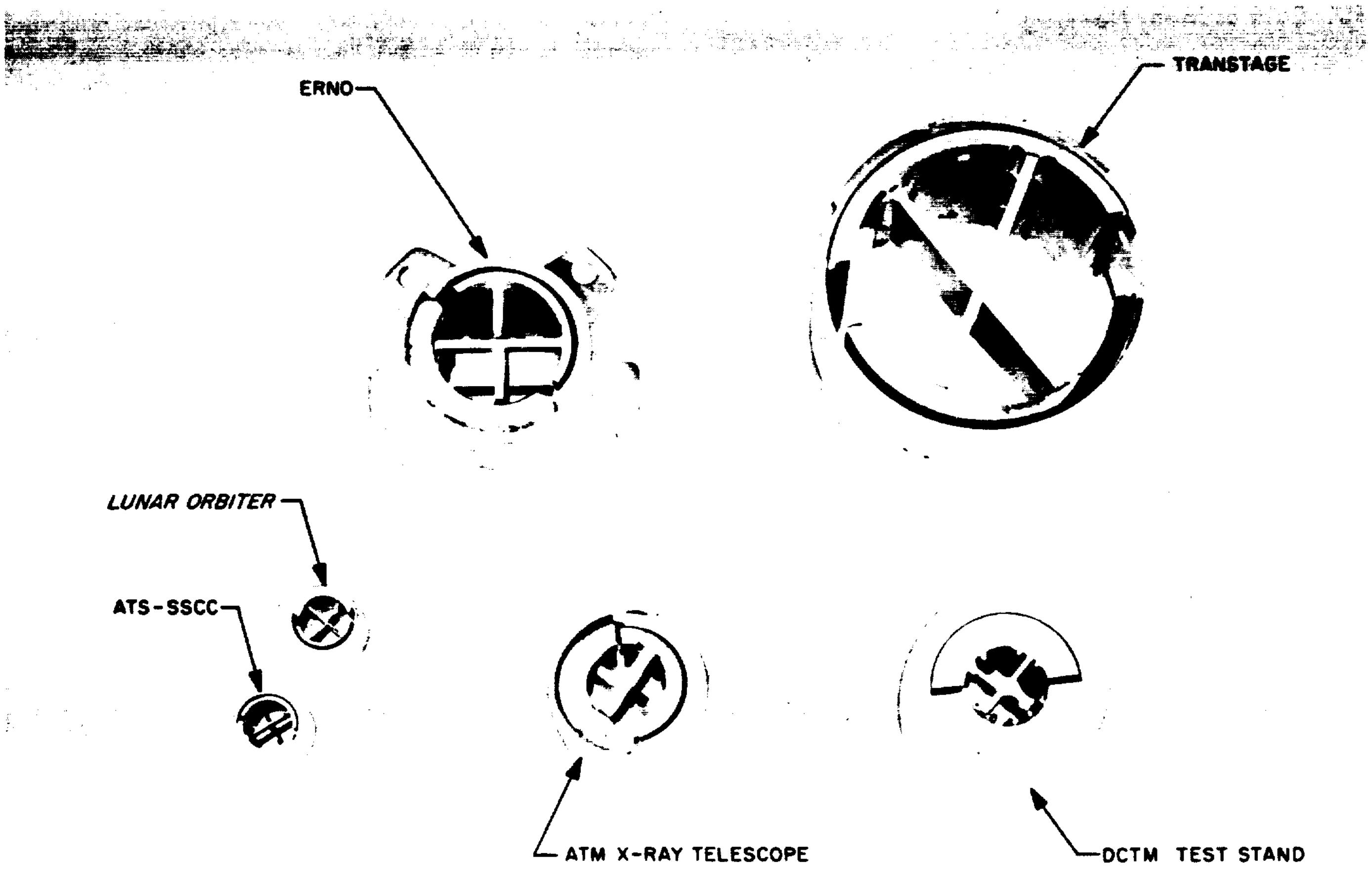


Fig. 4. Flexural pivots

pivots were purchased from stock with a 100% reliability load test prior to shipment.

Specifications for these pivots were as follows:

(1) Rate of motion: 0.05 to 10 deg/ s.

(2) Temperature: 400° F maximum to +35° F minimum.

(3) Nominal angular motion: ±4 deg, maximum ±15 deg at no load.

(4) Life: 5,000 nominal angular cycles at maximum temperature and nominal radial load; 200 nominal angular cycles at maximum temperature, maximum radial load and maximum axial load.

(5) Torsional spring rate through maximum angular motion: 13 lb in./rad ±10%.

(6) Radial load: 200 lb maximum, 110 lb nominal.

(7) Axial load: 35 lb maximum.

(8) Pressure environment: 1×10^{-14} to 760 torr maximum.

The reasons for selection of these pivots were: (1) no lubrication requirements, (2) elimination of the effect of varying operating temperatures on bearing clearances, and (3) small required angular motion (see Fig. 5).

The ATS Spin-Scan Cloud Cameras utilize a double-end-type pivot ⅝ in. in diameter and 1 in. long. These pivots are used in both the black-and-white and the color versions of the camera. They provide trunnion mounts for the camera telescopes, which oscillate ±7.5 deg and ±9 deg respectively at a rate of 1 cycle every 24 min. These applications demand extremely accurate position repeatability as the telescope moves through its angular travel. Figure 6 is a sectional view of the color camera assembly showing the pivot locations. Transverse forces applied to the pivots in both of these applications were relatively small, permitting the use of the flat flexure brazed fabrication.

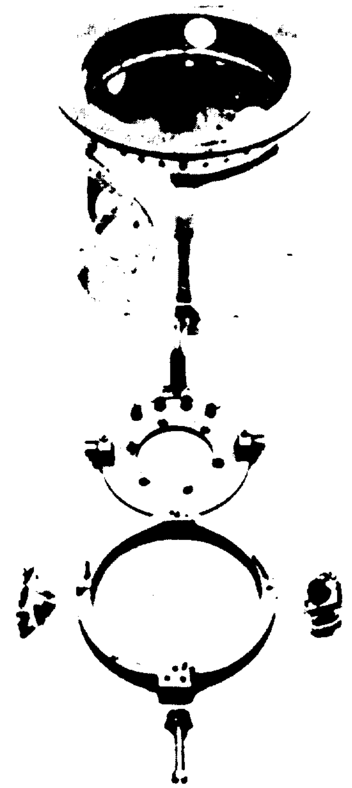


Fig. 5. *Lunar Orbiter* gimbal and bearing assembly, thrust vector control, photograph courtesy of The Boeing Company

The Engine Gimbal System for the Transtage Engine (*Titan III C* Missile), Fig. 7a, involved a breakthrough in the development of the weld fabrication techniques discussed in earlier parts of this paper. Several earlier unsuccessful fabrication attempts provided the experience and pointed to the need for new design and fabrication approaches. This program started with some basic investigations of the joining process. Weight was important, as were high strength, low torsional stiffness, and low radial deflections. After preliminary investigations, the 60-deg symmetrical nonuniversal system of flexures was selected with a double-end pivot configuration. The previous unsuccessful attempts at joining both by brazing and welding of flat flexures into a cylindrical housing led to the development of the I-beam type flexure and electron beam welding.

The housing joining method shown in Fig. 3c (heli-arc with countersunk holes) was used. A weld failure on the first prototype made it necessary to increase weld size. For weight reduction, it was also mandatory to remove a considerable amount of the core thickness and outer sleeve material, which further limited the available weld area. In spite of these requirements, the welding technique solved the problem.

The ERNO Engine (ELDO Third Stage) gimbal pivots (Fig. 7b) required a flexure configuration change due to

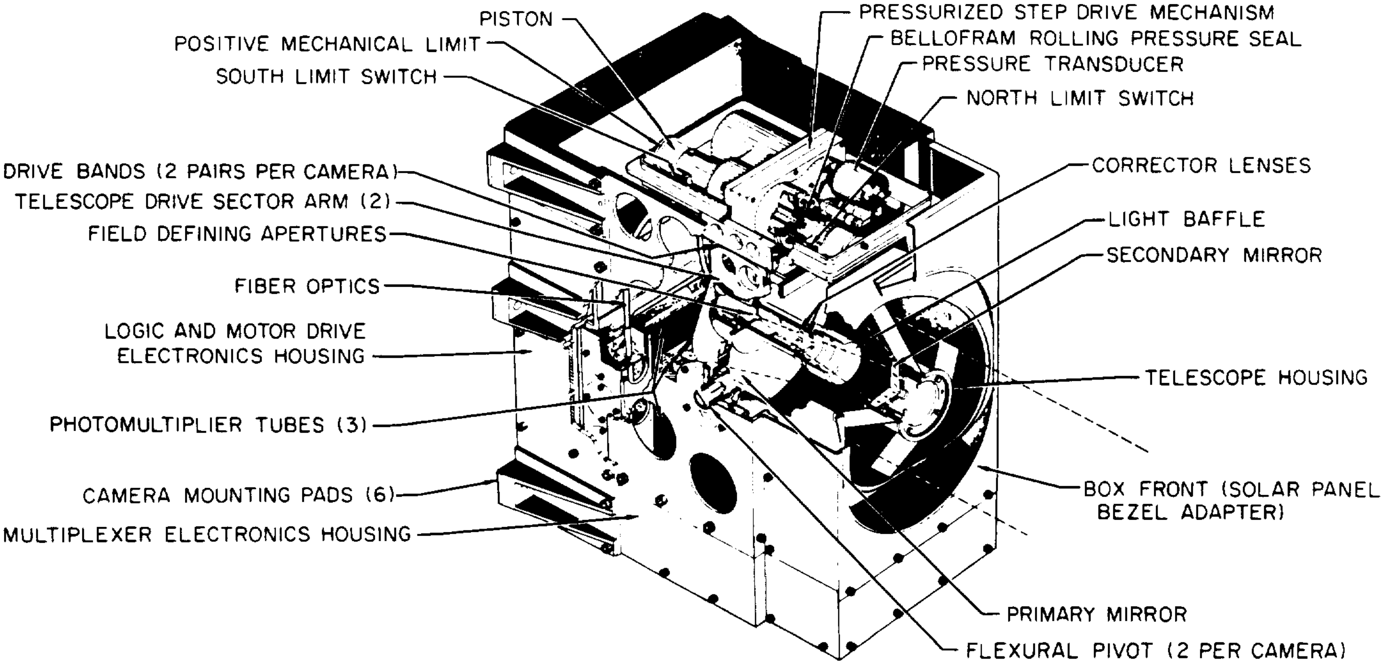


Fig. 6. ATS Spin-Scan Cloud Camera, photograph courtesy of the University of Wisconsin

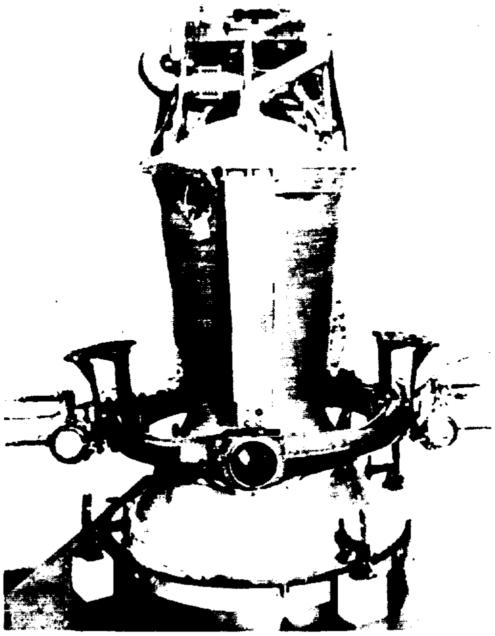


Fig. 7a. Transtage engine and gimbal, photograph courtesy of Aerojet General Corporation

Fig. 7b. ERNO engine and gimbal, photograph courtesy of ERNO Gmbh, Bremen, Germany (designer and producer of the main rocket motor and the structure of the third-stage *Europe I* rocket)

the limited envelope allowed for the pivot. The square gimbal ring tends to reduce the M_T type of moment (see Fig. 1), allowing a cantilever design in the space available. Normally a double-end-type pivot would be recommended for this high load application to better distribute the major forces on the flexures. The "plus design" configuration solved this problem. The outer housing-to-core welds had passed acceptance load tests but failed in a static load test when installed in the customer's hardware. Available area for welding was limited by the flexures and four large mounting lugs on each half of the outer housing. After some development, we adopted the annualar weld shown in Fig. 3d. This approach was successful – both static and vibration tests were performed satisfactorily and the system was approved for flight status.

The Directional Control Test Missile (DCTM) Wiggle Test Stand (Fig. 8), involved another gimbal system with much higher radial stiffness and smaller deflection angle than the two previous gimbals. In spite of high radial stiffness requirements, it appeared that the flexures would be sufficiently stiff with the 90-deg symmetrical universal system. Weight was no limitation, and a double-end pivot with a square gimbal ring was designed. From this job we learned that the influence of the housing thickness on the radial stiffness of the pivot was of prime importance. The diameter had to be increased from an original diameter of 1⅛ in. to 1¾ in. to provide satisfactory results for the application.

The most recent pivot application for structural space hardware is the AAP telescope mount. These pivots support the X-ray telescope and are used to minimize stresses in the telescope tube, particularly during the launch phase of the program. Figure 9 shows the arrangement of the pivots on the X-ray telescope. The specifications required small deflection angles, moderate loads, and high radial stiffness. The experience gained from investigating the core stiffness on the DCTM test stand application and from the annular weld method used on the ERNO gimbal pivots was utilized in the design of the X-ray telescope mount, resulting in only minimal development testing. In fact, this application was satisfied without the benefit of a prototype unit.

VIII. Summary

It has been attempted here to demonstrate some of the primary considerations in the design and fabrication of flexural pivots as structural members. The purpose was

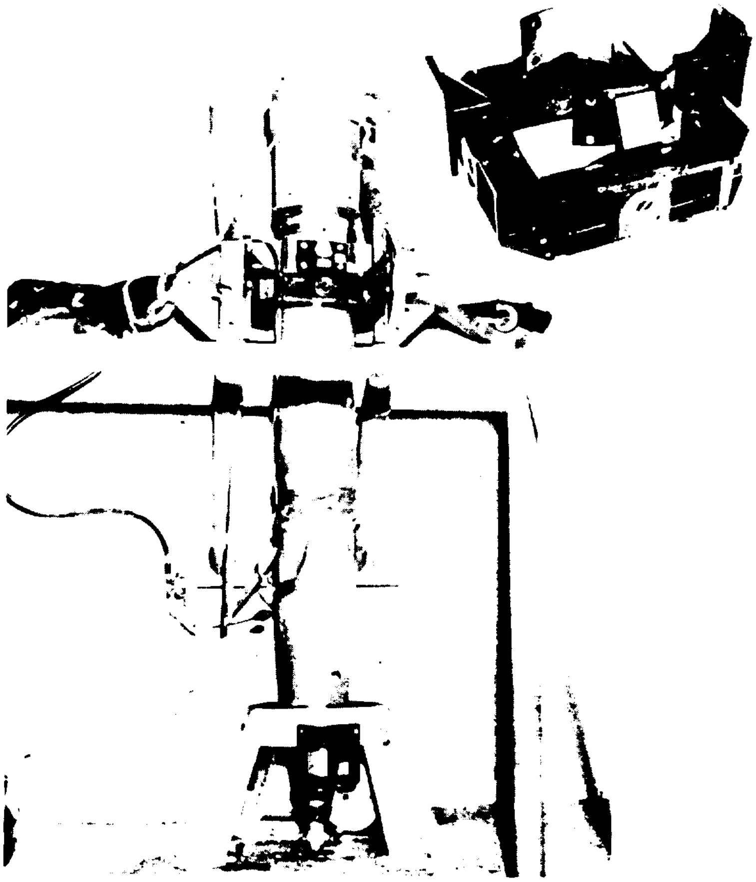


Fig. 8. DCTM Wiggle Test Stand, photograph courtesy of Grumman Aircraft Engineering Corporation

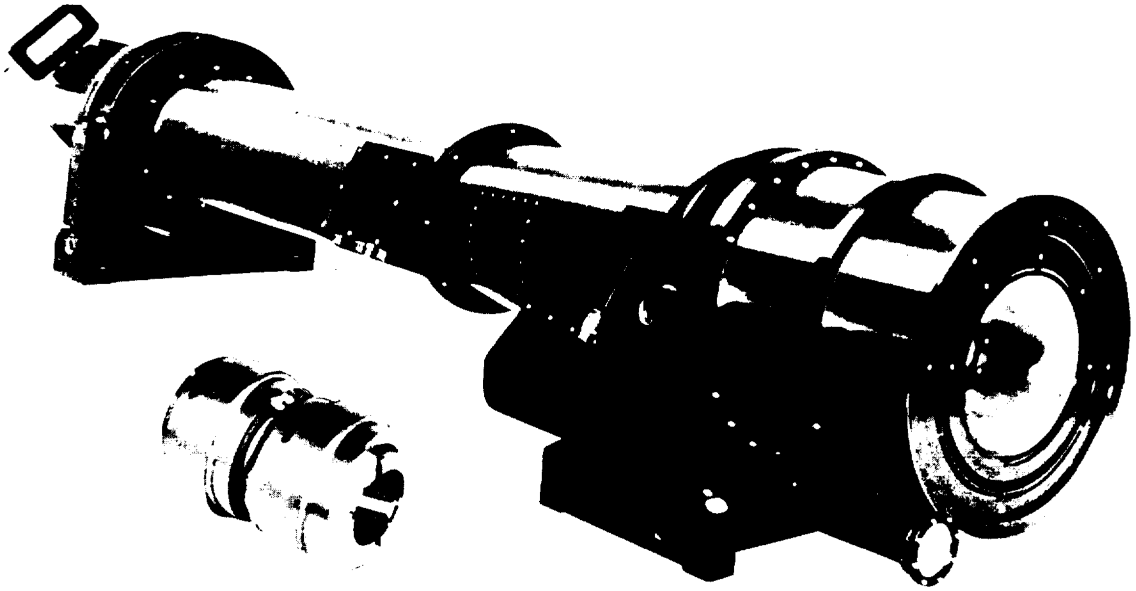


Fig. 9. ATM X-ray telescope, photograph courtesy of American Science and Engineering Company

to demonstrate the necessity to thoroughly investigate all conditions of load and environment before deciding on a pivot size and configuration. The fabrication methods cannot be separated from design requirements, since they often dictate the feasibility of the use of flexural pivots – both from design and cost considerations.

Acknowledgments

The author acknowledges the courtesy of the companies that have given permission for publication of photographs in this paper, as listed in the figure captions. Acknowledgment for publication of the Spin-Scan Cloud Camera photograph is made to the NASA Goddard Space Flight Center and to V. E. Suomi and R. J. Parent of the University of Wisconsin, prime investigator and co-investigator, respectively, for the camera.

References

1. Eastman, F. S., *Flexural Pivots to Replace Knife Edges in Ball Bearings*, Eng. Exp. Sta. Bull. No. 86. University of Washington, Seattle, 1935.
2. Eastman, F. S., "Design of Flexural Pivots," *J. Aeronaut. Sci.*, Vol. 5, pp. 16–21, Nov. 1937.
3. Eastman, F. S., "The Unique Properties of Flexural Pivots," *The Trend in Engineering*, Vol. 12, No. 1, University of Washington, Seattle, 1960.
4. Wittrick, W. H., *The Theory of Symmetrical Crossed Flexure Pivots*, Department of Aeronautical Engineering, University of Sydney, Australia, 1948.
5. Troeger, H., "Considerations in the Application of Flexural Pivots," *Automatic Control*, Vol. 17, No. 4, pp. 41–46, Nov. 1962.