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THE ECCENTUATOR - A NEW CONCEPT IN ACTUATION

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ABSTRACT

A new concept in actuation for aerospace mechanisms is presented. This actuator, called an Eccentuator, features unique output characteristics, installation and envelope efficiencies, and relative simplicity. The actuator can be powered by either hydraulic or mechanical inputs. Potential applications of the Eccentuator and development efforts are discussed.

INTRODUCTION

The new actuation concept was discovered during research for methods of actuating variable camber surfaces on supercritical wings. The supercritical contour in the region of the control surface hinge line is thinner than a conventional airfoil contour. The aft air loading of the supercritical contour causes large hinge moments on the control surfaces. The thin contours and large hinge moments coupled with an aerodynamic requirement to keep both the upper and lower surfaces free from protuberances created a need for an actuation system which did not resolve the hinge moments with a force couple parallel to the surface reference plane. The actuation system also had to be mechanically and hydraulically stiff and have good positioning resolution.

An attractive possibility for such a system was a bent beam situated along the surface reference plane and supported in the wing by a two bearing socket. Rotation of the bent beam in the wing socket would cause angular motion of the free end of the beam about a hinge line. However, attachment of the flap to the free end of the beam with an acceptable bearing structure presented a problem. The free end of the beam not only moved in an arcuate motion about the hinge line, but it also had lateral motion along the hinge line. Previous use of bent beam actuators employed rollers on the beam in spanwise tracks in the flap to accommodate this lateral motion.

The Eccentuator is a bent beam actuator that compensates for the lateral motion of the beam with an eccentric bearing on the support end of the beam. A unique indexing gear arrangement employed with the bent beam and the eccentric bearing keeps the free end of the beam in a plane normal to the hinge line. Not only does the bent beam/eccentric bearing combination result in planar arcuate motion, but the arcuate motion is twice that achievable by rotating the bent beam alone.

The Eccentuator actuator exhibits load carrying capabilities much like a straight, fixed-end beam between the flap and the wing. For small surface deflection angles around 24°, the Eccentuator carries 90%-100% of the hinge moment through its structural elements depending on the deflection position.

The remaining 10%-0% hinge moment is reacted by an actuation system. This reduced actuation force requirement is a result of the inherent variable mechanical advantage in the Eccentuator. The Eccentuator system can therefore be thought of as variable geometry structure.

The large inherent mechanical advantage which results in a low actuation requirement allows several unique actuation concepts. A small integral rotary hydraulic actuator can be placed between the two moving parts and apply actuator forces directly to the load carrying members. For mechanical drive systems, the inherent mechanical advantage reduces the load on the drive gears allowing these to be smaller. The inherent mechanical advantage reduces the effect of gear freeplay on the surface.

The drawing in figure 1 illustrates a basic single-ended hydraulic Eccentuator. The eccentric support bearing is called a carrier. The indexing ring gear is mounted onto the grounded support housing while the indexing gear is fixed to the beam.

CONCEPT SIMPLICITY

The drawing in figure 2 illustrates the basic motion of the Eccentuator. In figure 2(a) the motion generated by rotating the beam alone is shown. The angular output is twice the beam bend angle. The lateral motion along the hinge line is also noted. Figure 2(b) shows the carrier location with respect to the hinge line and illustrates with a theoretical straight beam the effect on a beam of rotating the carrier in the opposite direction from that of the beam. The free end of the theoretical beam moves in the same angular direction as the bent beam, but the lateral movement is in the opposite direction. The eccentric bearing centerline is displaced from the carrier centerline such that it generates an angle with respect to the carrier centerline equal to the bend angle of the beam. Figure 2(c) illustrates that when the bent beam is combined with the carrier, and the two are rotated in opposite directions, the carrier's lateral displacement compensates for the beam's lateral displacement and the angular motion adds to produce a total-planar angular displacement of four times the beam bend angle.

ENVELOPE EFFICIENCY

One of the principal features of the Eccentuator concept is its installed envelope efficiency. Figure 3 compares the Eccentuator installation with that of a linear actuator in a flap configuration. The Eccentuator is centered about the hingeline and lies along the centerline of the contour. Hinge moments are resolved normal to the wing reference plane rather than parallel to the reference plane as in most actuation concepts. The moment arm is therefore not a function of the wing thickness.

ACTUATION LOADS

The basic structural advantage of the Eccentuator is that the hinge moment of the moving surface is transferred to the supporting structure primarily through structure. The positioning actuator therefore must react only a small percent of the load. The flap hinge moment (M_H) is resolved as shown in figure 4, into a couple load on the flap end of the beam and transferred in bending through the beam to a reaction couple on the support side. The couple reaction on the beam by the carrier in the mid-travel position shown is not on the centerline of the Eccentuator assembly by the amount of eccentricity in the carrier. This eccentric loading causes a rotational torque on the beam in one direction and an equal and opposite torque on the carrier. The actuation torque (M_A) on either the beam or carrier is

$$M_A = M_H \tan \theta \cos \phi \qquad (1)$$

Where: θ = beam bend angle

ϕ = rotation angle from mid-travel position

When the beam is in the mid-travel position, the actuation torque requirements are greatest. At the actuator travel extremes where ϕ is 90°, the actuation torque requirements diminish to zero. Small beam bend angles, such as 6° which produce actuator outputs of 24°, require a maximum actuation torque on both the beam and the carrier of only approximately one tenth of the hinge moment. As the Eccentuator is positioned away from the mid-travel setting, the actuation torque requirement decreases for a given hinge moment.

INHERENT MECHANICAL ADVANTAGE

The reduced actuation torque requirements are a result of an inherent mechanical advantage. This mechanical advantage varies from a maximum value of infinity at the actuator travel extremes to a minimum value at the mid-travel position. The inherent mechanical advantage is calculated by the following equation:

$$\text{Mechanical Advantage} = \frac{1}{\tan \theta \cos \phi} \qquad (2)$$

From this relation it can be seen that the mechanical advantage doubles, from its minimum value at mid-travel, when the beam and carrier have been rotated 60° from the mid-travel position. Full actuator travel and an infinite mechanical advantage occur at 90° rotation from mid-travel. Figure 5 is a typical mechanical advantage curve for a 6 degree beam bend angle.

This inherent mechanical advantage can be applied to reduce the input torque requirement at the travel extremes when encountering a constant hinge moment load, or produce an ever increasing output torque, with a constant input torque, as the actuator is positioned away from center.

OUTPUT MOTION CHARACTERISTICS

The output deflection angle (β) as a function of the rotation of either the beam or the carrier is given by the following equation of motion:

$$\beta = 2\theta\ (1 - \sin \phi) \qquad (3)$$

Large angular motion of the beam and carrier at the ends of travel produce little output motion. Therefore, avoidance of these areas of very large mechanical advantage will not produce significant output motion loss.

DRIVE SYSTEMS

The Eccentuator can be actuated by an integral hydraulic actuator or by external power sources through a unique gear drive system. Figure 6 illustrates these two drive systems. Both of these systems operate on the action-reaction principle where the force that produces the torque to turn the beam also produces the torque to turn the carrier.

Hydraulic Eccentuator

The hydraulic Eccentuator is produced by incorporating a rotary hydraulic actuator between the beam and the carrier as shown in figure 6(a). The beam becomes the shaft of the actuator and has one of the vanes attached. The carrier extension forms the actuator housing with a vane.

Section A-A of figure 6(a) shows that introduction of fluid under pressure into the actuator housing between the vanes produces a force on the beam vane and an equal and opposite force on the carrier. The hydraulic Eccentuator produces a surface installation which is very stiff hydraulically. The surface hydraulic stiffness is a function of the square of the mechanical advantage between the surface and the hydraulic chamber. Thus, a 24 degree output Eccentuator with its 10:1 inherent mechanical advantage will appear to be 100 times as stiff as it would be if the actuator were operating directly on the hinge line. Since the rotary actuator is one tenth the size required at the hinge line, the stiffness increase offered by the Eccentuator over a rotary actuator is approximately tenfold. This factor increases as the mechanical advantage increases for position away from the mid-travel setting.

Mechanical Eccentuator

The preferred mechanical system utilizes a gear on the beam and a pinion gear mounted on the carrier as shown in figure 6(b). The gear set operates as a planetary system with the pinion gear being the sun gear. The system gear ratio is the product of the inherent mechanical advantage and that of the planetary gear set as follows:

$$\text{Gear Ratio} = \text{Inherent Mech Adv} \times \left(\frac{N_g}{N_p} + 1\right) \qquad (4)$$

Where: N_g = Number of gear teeth

N_p = Number of pinion teeth

The inherent mechanical advantage amplifies the system gear ratio and minimizes the effect of any gear freeplay on the driven surface.

Other mechanical systems are indicated by the fact that rotation of either the beam or the carrier will transmit operational torque to the other member though the indexing gears. Driving through only one part requires inputting twice as much torque as is required by the preferred system described above since the torque of the beam and the carrier are additive.

INSTALLATION FLEXIBILITY

The Eccentuator can be installed in a number of different configurations. The three shown in figure 7 differ with each other only at the hinge line bearing on the fixed structure side. The arrangement in figure 7(a) shows the structure side hingeline bearing to be a part of the carrier. Another bearing between the carrier and the beam, which is skewed to the outer bearing, allows the beam to rotate as well as pivot within the structure.

The configuration in figure 7(b) allows the beam to rotate and pivot by use of a spherical bearing.

The configuration of figure 7(c) provides the same functions as the two above, and allows for the elimination of one bearing on the beam. The hinge structure on either side of the beam provides the loadpath from the support structure reaction to the flap beam reaction bearing. The elimination of the bearing on the structure side allows the beam to be larger in the high bending load area. The actuation friction associated with turning the beam is reduced.

The concepts in figure 7 (a) and (b) are structural hinges and a hinge is not required for the surface.

CONFIGURATION FLEXIBILITY

The Eccentuator concept discussed has been that of a single-ended basic Eccentuator. Two of these basic units can be joined at the pivot point to form a double-ended Eccentuator. These two concepts are depicted in figure 8. The double-ended unit will allow twice as much deflection as a single-ended unit with the same design beam angle. This double-ended arrangement allows the hydraulic Eccentuator to have dual hydraulic chambers, each of which can drive the unit full-travel with half of the load capability.

Shown in figure 8 are multiple units. The tandem arrangement has two beams and a common carrier. This system has two hinge lines. Multiple hinge lines can be achieved with multiple carriers and beams.

CAPABILITIES

The double-ended hydraulic Eccentuator with a design angle of 6 degrees (48 degrees total output motion) can operate against a 4868.5 newton-meter

(100000 lb-in) ultimate moment load and be housed within an estimated cylindrical envelope 10.4 cm (4.1 in) in diameter and 43 cm (17 in) long. The envelope will increase in size as the design deflection angle increases because of the necessary growth of the rotary actuator. The double-ended Eccentuator has a practical angular motion capability of around 90 degrees.

A single-ended mechanical Eccentuator with an output angle of 64 degrees has been prototyped for commercial use. However, to take advantage of the inherent mechanical advantage of the Eccentuator to reduce the size of the drive system and thus the installed envelope in an aircraft, the design output angle should be limited to around 30 degrees for a single-ended unit and 60 degrees for the double-ended Eccentuator. The Eccentuator concept offers numerous trade-off parameters which allow tailoring the design to various requirements.

INSTALLATION ADVANTAGES

Typical installation of a mechanical Eccentuator in a leading edge flap application is illustrated in figure 9. In the flap up position the Eccentuator is in an overcenter position and all of the cruise flight loads are transmitted through the structural elements of the Eccentuator leaving the gear drive systems unloaded. This condition also occurs in the flap down position.

Figure 10 illustrates a typical double-ended hydraulic Eccentuator in a trailing edge surface application. The Eccentuator is in the mid-travel position in the control surface neutral position. As deflection of the surface occurs on either side of neutral, the Eccentuator's variable inherent mechanical advantage increases the integral rotary hydraulic actuator output as the airloads increase. This characteristic allows the Eccentuator to be designed to lower actuating loads than the normal maximum deflection airloads.

CONCLUDING REMARKS

Studies have found the Eccentuator to be applicable to many other actuation problems on aircraft in addition to the thin wing surfaces. Other aerospace mechanisms on missiles, helicopters and space vehicles also present application opportunities. The joints of space and industrial robots have been identified as potential uses for the Eccentuator. Many agricultural and industrial applications have been forecast.

A working model of a double-ended hydraulic Eccentuator has been modeled in both a unit surface and a rudder application. It is demonstrated by using pressurized Freon. An electric motor powered Eccentuator missile fin control model demonstrates electro-mechanical operation. Vehicle steering with a manual Eccentuator system has been modeled and prototyped on a late model automobile. Other applications are being either modeled or prototyped in the process of developing the Eccentuator concept into a mature actuation system that will meet the demanding requirements of the aerospace industry.

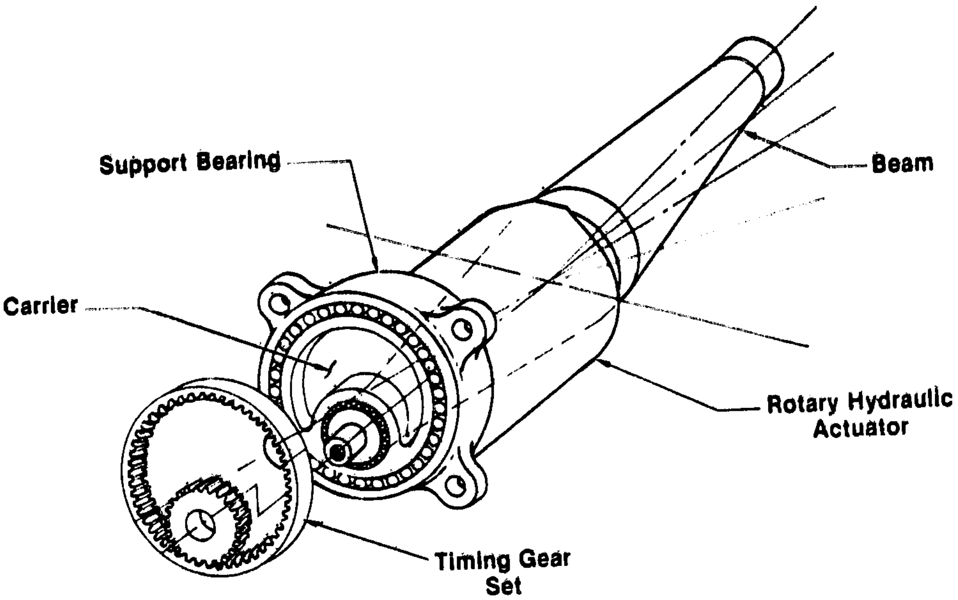

Figure 1.- Single-ended Eccentuator.

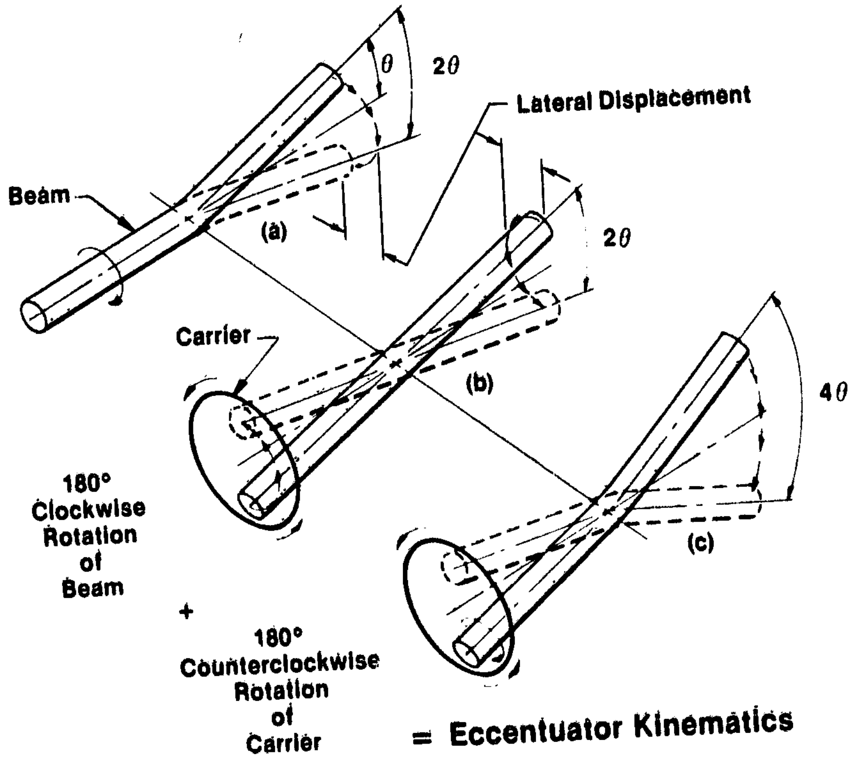

Figure 2.- Eccentuator kinematics.

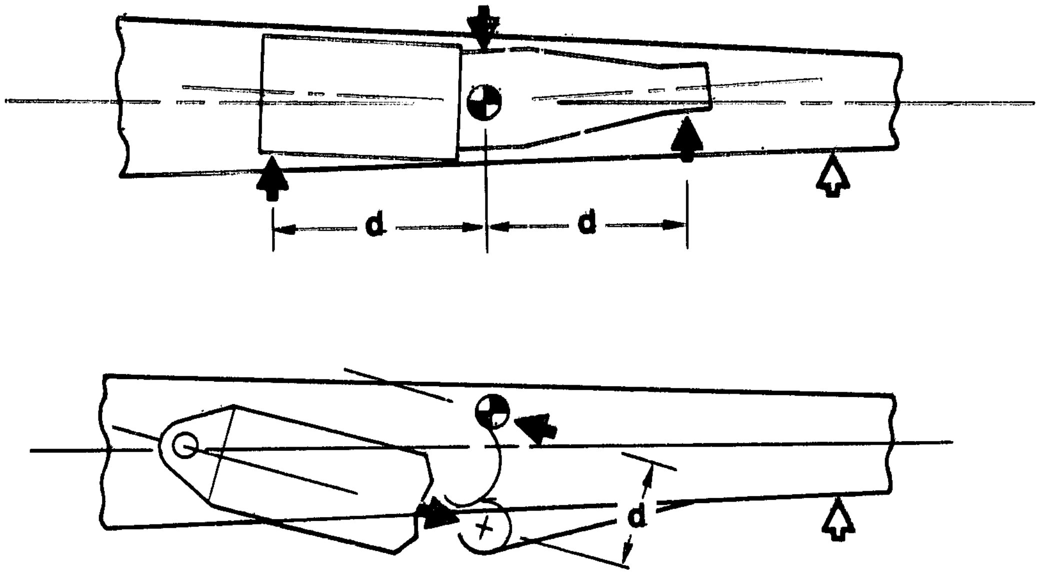

Figure 3.- Installation envelope efficiency.

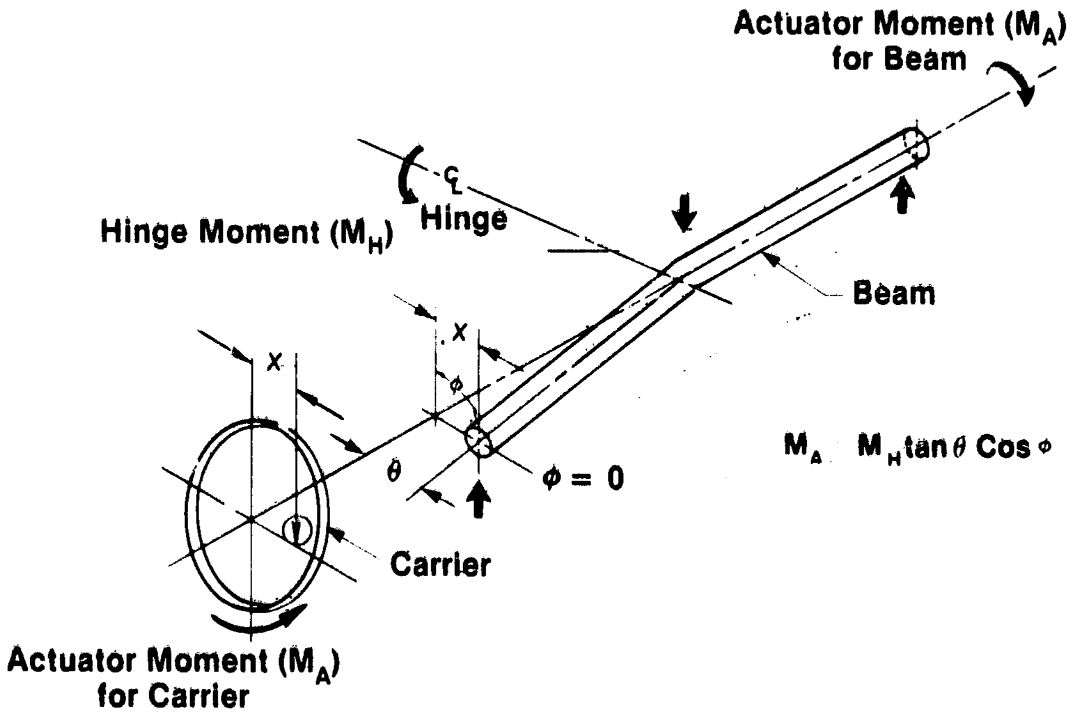

Figure 4.- Actuation loads analysis.

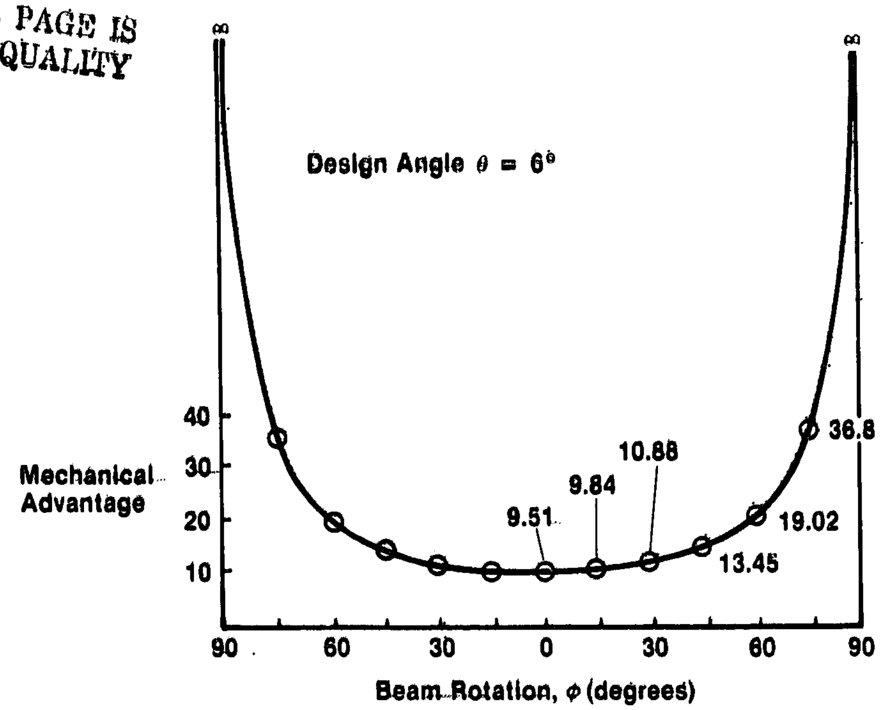

Figure 5.- Inherent mechanical advantage.

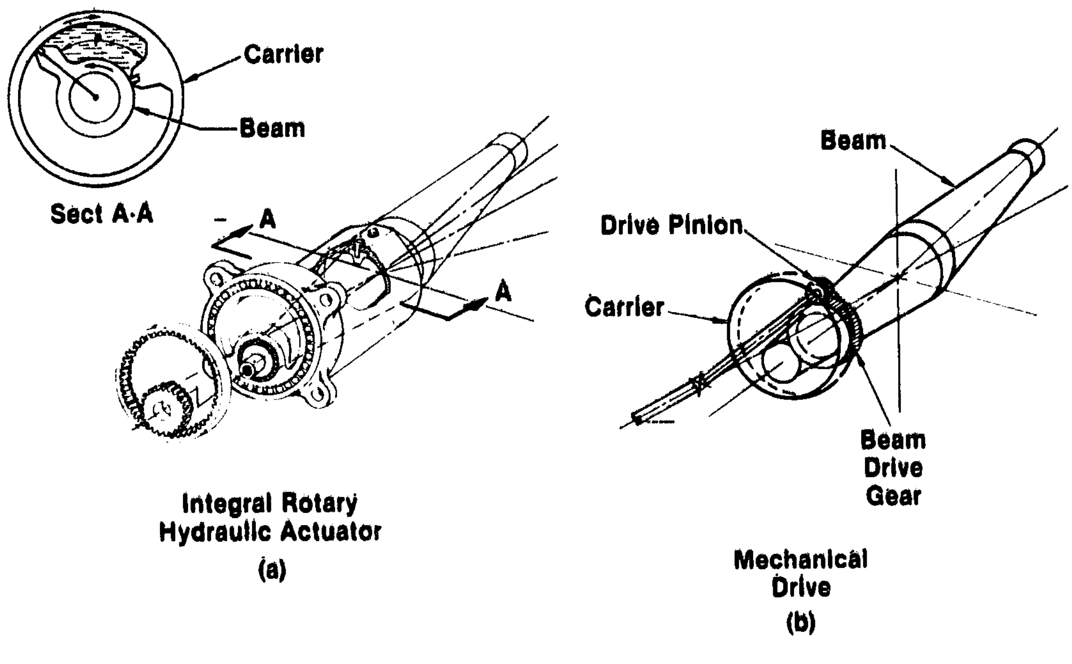

Figure 6.- Drive systems.

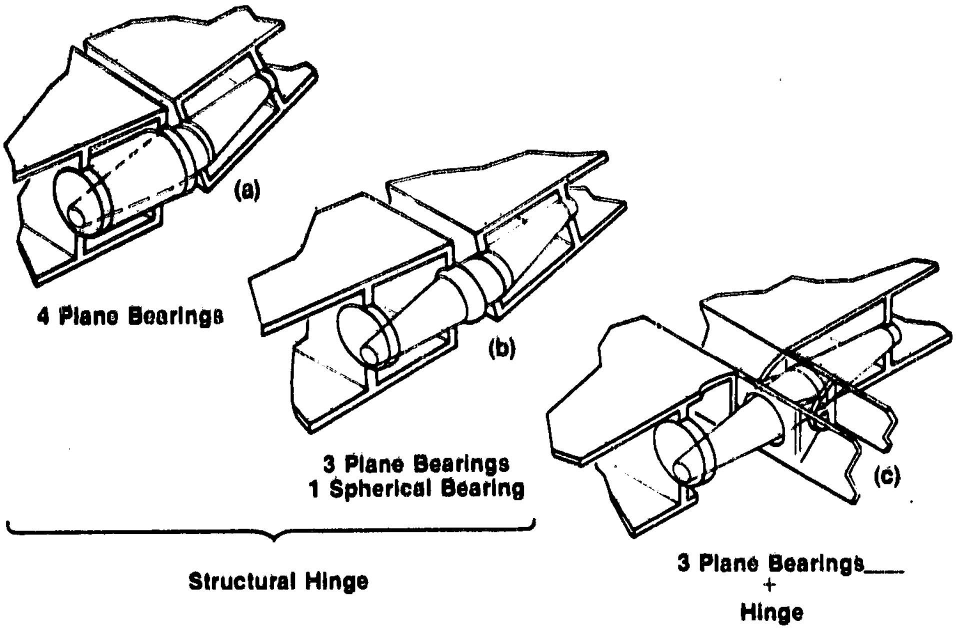

Figure 7.- Installation flexibility.

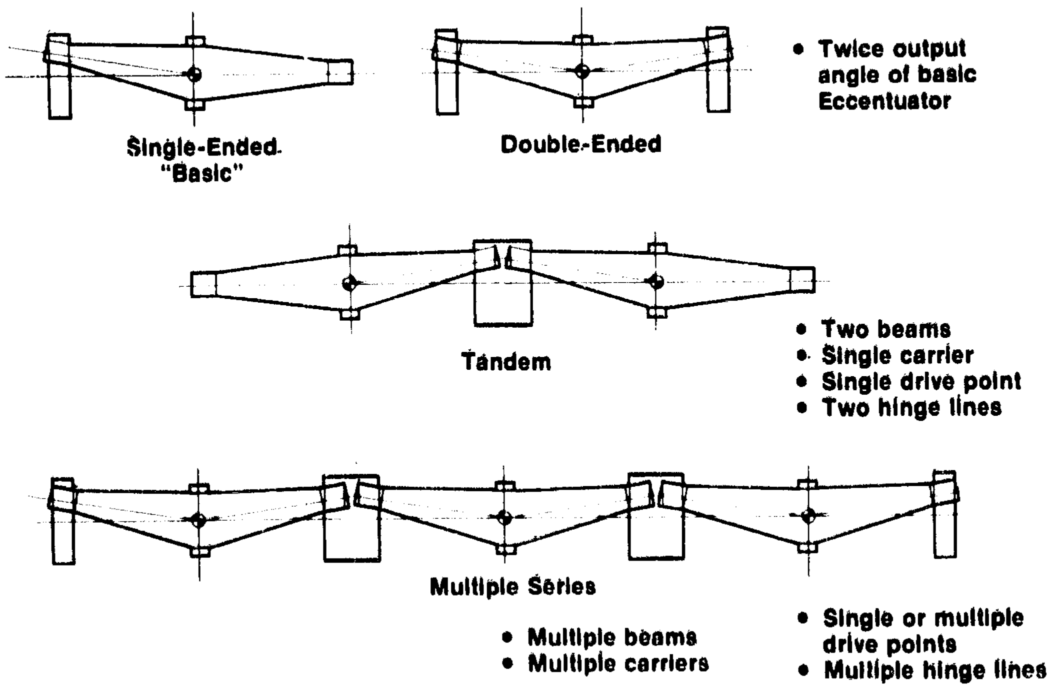

Figure 8.- Configuration flexibility.

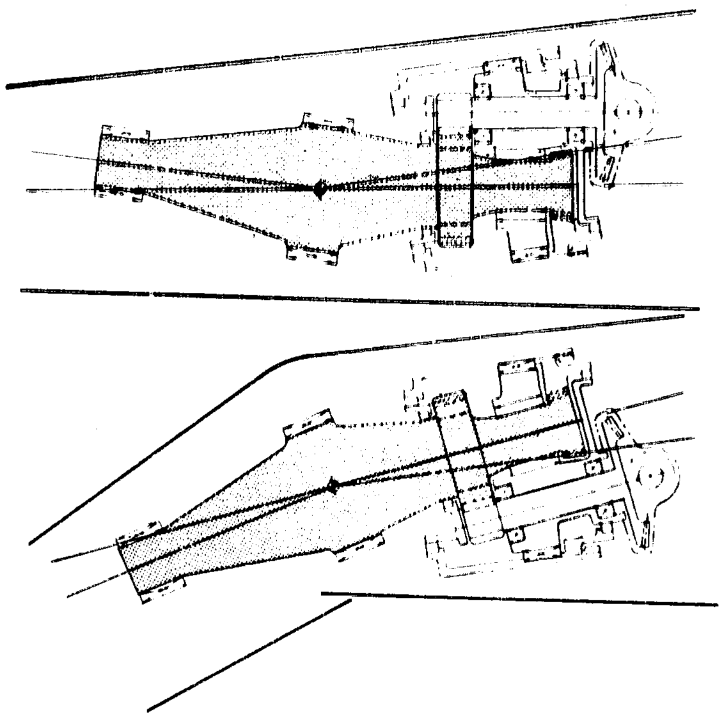

Figure 9.- Leading edge installation.

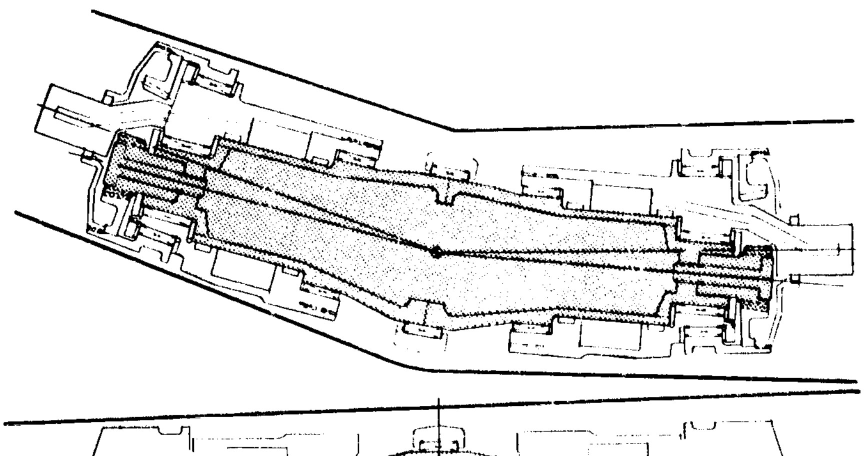

Figure 10.- Trailing edge installation.

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