21. VIKING LANDER

ANTENNA DEPLOYMENT MECHANISM

By Kenneth H. Hopper and Dean S. Monitor
Martin Marietta Aerospace Corporation

SUMMARY

A Mars lander is currently being designed and built by Martin Marietta Aerospace as part of the Viking Project, under contract to the Viking Project Office of NASA Langley Research Center.

One of the mechanisms that is included in the configuration of the Viking Mars Lander is an antenna dish deployment mechanism which utilizes a mechanical escapement device for controlling deployment. The criteria require that the mechanism be capable of deploying the antenna under a wide range of aiding and retarding winds after landing on Mars but must limit the acceleration during deployment and latching of the antenna. Adding to the complexity of the desi · are the requirements for very accurate alignment after erection and heat sterilization prior to launch. This mechanism has been designed, fabricated and successfully tested to qualification test levels for usage on the Viking mission.

INTRODUCTION

One of the requirements of the Viking Project is that the Mars Lander have the capability of direct lander-to-Earth communication as well as relayed communication through the Mars Orbiters. This requirement resulted in the selection of a 30-inch-diameter parabolic dish antenna with the capability of being pointed at the Earth. The size of this antenna and the

PRECEDING PAGE BLANK NOT FILMED

need to locate it high enough above the lander to avoid interference, coupled with the spacecraft volume limitations, established the need for a deployment mechanism. See Figure 1 for a definition of the spacecraft envelope restrictions and the deployed relationship of the antenna to the lander.

DESIGN SELECTION

The integrated arrangement of all systems on the lander dictated the storage envelope and operational envelope following deployment. This then indicated that the deployment mechanism design would be a pivoted arm to swing the antenna through approximately 145° rotation from the stowed position to fully deployed position. A summary of the design criteria is presented in Table 1.

The main deployment loads result from the Mars surface winds. The results of wind tunnel tests upon a 45% scale model of the parabolic dish combined with gravity, cable bending and friction effects established the total maximum deployment load of 400 in-lb torque with the head wind. An aiding wind produces 200 in-lb of torque. A summary of the torque requirements is presented in Figure 2.

Trade-off studies of alternate means of powering the deployment mechanism included pyrotechnic gas generation, electric motor, clock spring drive, torsion spring, compression spring, constant-force spring, and stored gas systems. Due to the resulting minimum lander weight and overall simplicity, dual constant-force springs of laminated construction, similar to those for the Apollo lunar excursion module landing legs, were selected to power the deployment mechanism.

Studies of several means of controlling the rate of deployment and shock limitation included electrodynamic braking, hysteresis braking, centrifugal braking, deformable member shock attenuation, viscous damping, magnetic particle braking, mechanical escapements, and shock mounted limit stops. A mechanical escapement was selected as the design approach affording the best chance of success.

Since there was no need for precise control of the rate of deployment, the anchor-recoil type of escapement was selected over the balance-wheel type. It also has inherent self-starting performance, fewer mechanical elements, and the capability to operate over a wide range of torque loads. See Figure 3 for a schematic of the escapement device.

DESIGN DESCRIPTION

The deployment mechanism consists of a welded aluminum fixed support, a pivot-mounted support elbow, the power spring subassembly, a spring-loaded uplock latch subassembly, and the governor subassembly. The mechanism is shown in Figure 4 and its weight is 8.3 pounds.

The fixed support is a three-piece weldment consisting of a base plate, support tube, and hub and is made of 2219-T87 aluminum alloy. After rough machining, this assembly is stress relieved and then final machined. The base plate is pinned to the lander body for accurate angular alignment.

The support elbow, made of 2024-T851 aluminum alloy, is mounted on angular-contact, torque-tube type ball bearings within the hub of the fixed support. Fixed stop pins in the elbow contact integral limit stops in the hub to accurately position the support elbow at the end of its rotational travel. The support elbow terminates in a mounting flange which interfaces with the antenna mounting base. This mounting flange is required to be positioned within 16 arc minutes cone angular error but is typically within 1 arc minute upon completion of the elbow support travel.

The power spring subassembly consists of dual constant-force springs laminated of Elgiloy, a cobalt-chromium-nickel alloy, which together produce 400 in-lb of torque. One end is anchored to the hub and the other rolls over a spool on the elbow support.

The uplock latch is attached to the elbow support with a hinge pin. A torsion spring causes the opposite end to engage the outer diameter of the hub, drop into a pocket, slide up a ramp machined in the hub, and then lock the elbow support upon completion of the deployment rotation.

The governor subassembly, shown in Figure 3, includes a two-piece gear box, a three-pass gear train of a 300:1 ratio, the escapement, torsion pendulum, two R-4 ball bearings, six R-3 ball bearings, and a cover. The first gear mesh is driven from the end of the elbow support. The gear box is fabricated from 2024-T851 aluminum alloy, the gears are made from A-286 stainless steel, and the escapement from 17-4PH stainless steel. The escapement is the anchor-recoil type which produces an oscillation frequency:

$$f = K\sqrt{\frac{T}{I}}$$

Where - T is input torque and
I is pallet axis inertia

The escapement wheel also includes a ratchet clutch to provide for retracting the mechanism to the stowed position without a rate governing function. The active pallet faces and escapement wheel teeth are machined to 32 microinch finishes.

Dry film lubrication of bonded molybdenum disulphide types are incorporated in the ten ball bearings, the gears, the escapement, and within the laminations of the drive springs. A Teflon filled hard anodize coating serves to lubricate the spool upon which the power springs roll and the surfaces of the uplock latch.

Since the surfaces of the deployment mechanism reflect sunlight which may affect the lander cameras, a low reflectance coating is applied over Iridite surfaces on the mechanism.

DEVELOPMENT PROBLEMS

The design of the mechanism, with the exception of the escapement device, utilized technology that is well within the current state of the art and presented no particular problems during development testing. This application of an escapement device is rather unique, however, and required several iterations during test before the current design was finalized.

The major problems encountered were related to material selection, surface finish and lubrication, the detailed geometry of the pallet and wheel, and the stiffness of the pallet and its shaft. The initial configuration of the pallet and wheel is shown in Figure 5(a). After a few operating cycles the mechanism would hang up because of brinelling of the aluminum pallet. A change to A-286 stainless steel pallet and wheel tended to correct this problem; however, hang-ups still occurred short of the desired operating life.

Under the assumption that a reduction in the impact load, obtained by reducing the stiffness of the pallet, would increase the operating life, the pallet and wheel geometry shown in Figure 5b was tested. This configuration eliminated the hang-ups but proved to be too flexible, resulting in a run-away failure mode when subjected to the simulated maximum aiding wind. It was concluded that the stiffness of the pallet could probably be fine-tuned to solve the run-away problem and still avoid the hang-up mode; however, in the interest of economy and schedule, a stiff pallet with a revised contact surface geometry was tried and, since it worked well enough to meet the life cycle goal of 100 cycles, no further development of the pallet stiffness was considered to be warranted. The final geometry is illustrated in Figure 5c.

The materials used in the final configuration are 17-4PH H900 hardness for the pallet and 17-4PH H950 hardness for the wheel. This selection was made to provide a harder material on the pallet than on the wheel since the pallet is impacted 24 times for each impact on a given wheel tooth. Also, a chrome plating was applied to the pallet; nickel plating, to the wheel; and as Lubco 905 dry lube, on both surfaces.

The development test article was retrofitted with the updated escapement hardware and the qualification level test program was reconducted. (See table 2.) After completing the vibration, pyro shock and landing shock tests, with approximately 36 operating cycles, the escapement mechanism again experienced a hang-up. A failure analysis concluded that the chrome plating on the pallet was breaking down and chipping off, resulting in sharp edges that cut into the softer wheel and causing the hang-up. Consequently, the chrome plating was removed from the pallet design. However, the Lubco 905 was retained. Retesting of this configuration has verified its adequacy for its intended purpose.

CONCLUSIONS

The constant force spring system proved to be lightest of the candidate drives examined. The escapement type of rate governor provides adequate rate control throughout the large temperature range combined with the large torque variation in a simple, reliable manner. Although only one deployment on the Mars surface is required, the design goal of 100 deployments under worst case loading has been successfully demonstrated with this mechanism.

TABLE 1

DEPLOYMENT MECHANISM DESIGN CRITERIA

Deployment with 40 m/sec Restraining or 70 m/sec Aiding Wind

Up-Latching During Deployment That Prevents Back Travel

No Erection Velocity Required to Engage Uplock

Governor Life 20 Cycles With 40 m/sec Aiding Wind

Governor Life Goal of 100 Cycles

Erection Time - <5 Minutes With No Wind

Acceleration of High Gain Antenna Center of Gravity Less Than 2 G

Angular Error Mast/High Gain Antenna Interface - <16 Arc Minutes Root Sum Square

TABLE 2

DEPLOYMENT MECHANISM DEVELOPMENT TESTS

<u>Environmental Effects</u>	
Heat Compatibility	Five Cycles of 40 Hours at + 267°F Two Deployments
Vibration	Resonance Survey Sinusoidal Vibration Sine Vibration Requirement 5 to 19 Hz at 0.4 Double Amplitude (DA) 15 to 50 Hz at 7.5 G's Peak Rolloff at 20 dB/Octave from 50 to 100 Hz 100 to 250 Hz at 0.75 G's Peak Sweep Rate - 1 Octave/Min Random Vibration Normalized Spectrum 20 to 250 Hz: + 3 dB/Octave 250 to 100 Hz: at 0.75 G^2/Hz 1000 to 2000 Hz: -6 dB/Octave Overall Level: 10g rms 5 Minutes per Axis Six Deployments
Landing Shock	30 G 1/2 Sine Pulse Three Times Each Direction in Each Axis Six Deployments
Surface Thermal	Two-Hour Exposures at + 145°F Two-Hour Exposures at - 195°F Two Deployments at Each Temperature
Sand and Dust	Six-Hour Exposure Two Deployments
<u>Operating Pyro</u>	Six Deployments with Flight-Type Pyrotechnic Pin Puller Release System
<u>Deployment Life</u>	3/8 G Attitude With 70 Meter/Sec Aiding Wind 74 Deployments Total deployments under all environments is 100.

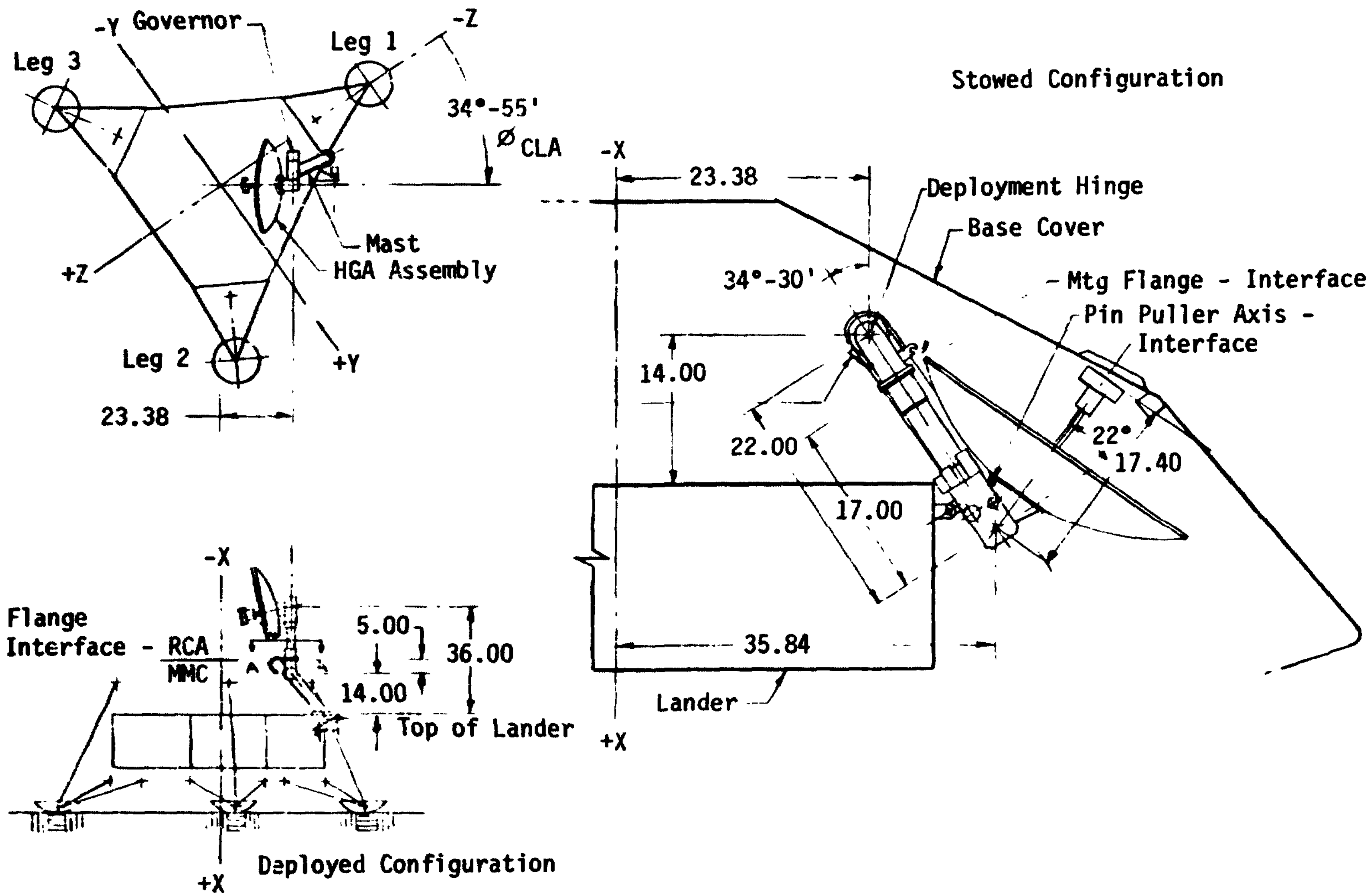

Figure 1.- Antenna deployment mechanism arrangement on Viking Lander. All linear dimensions are in inches.

NOTES: Curve A - Retarding 70 m/sec wind

Curve B - Aiding 70 m/sec wind

Curve C Is a Composite, Assuming Wind Direction Reversal and A 10% Factor for Friction, Bending of Cables, Sand in Dish, etc & 40 m/sec retarding wind

$$\rho = 10.38 \times 10^{-5} \ \frac{\text{lbf-sec}^2}{\text{ft}^4}$$

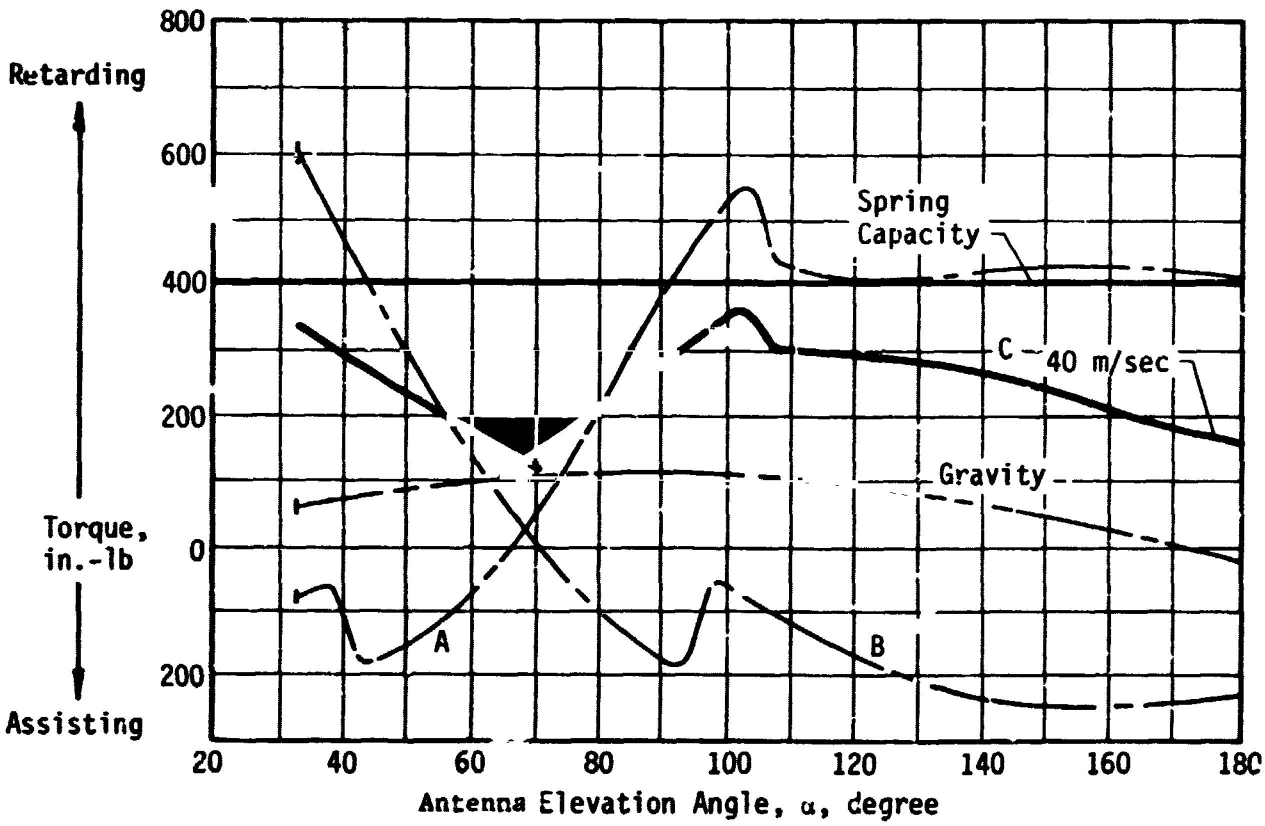

Figure 2.- Deployment loads.

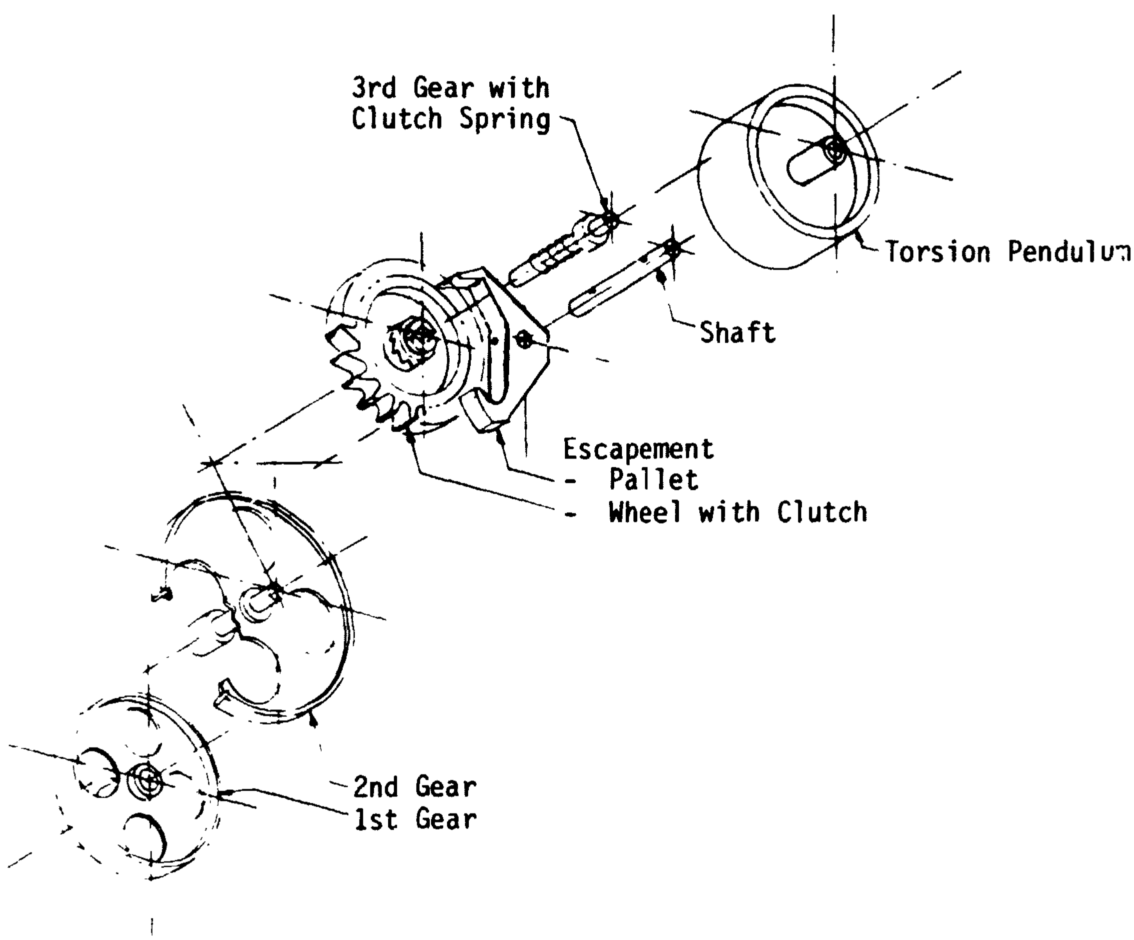

Figure 3.- Governor subassembly.

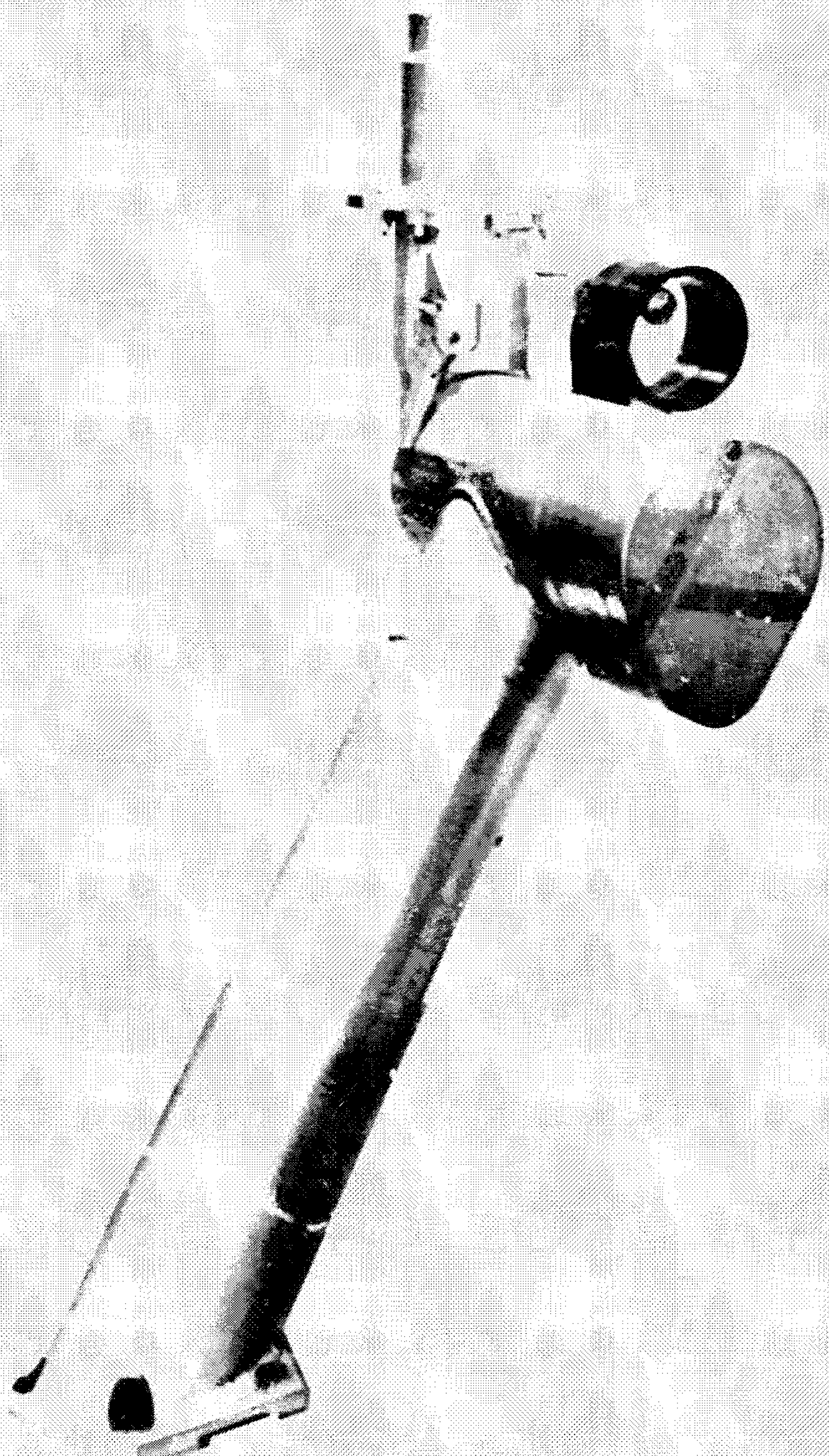

Figure 4.- Deployment mechanism.

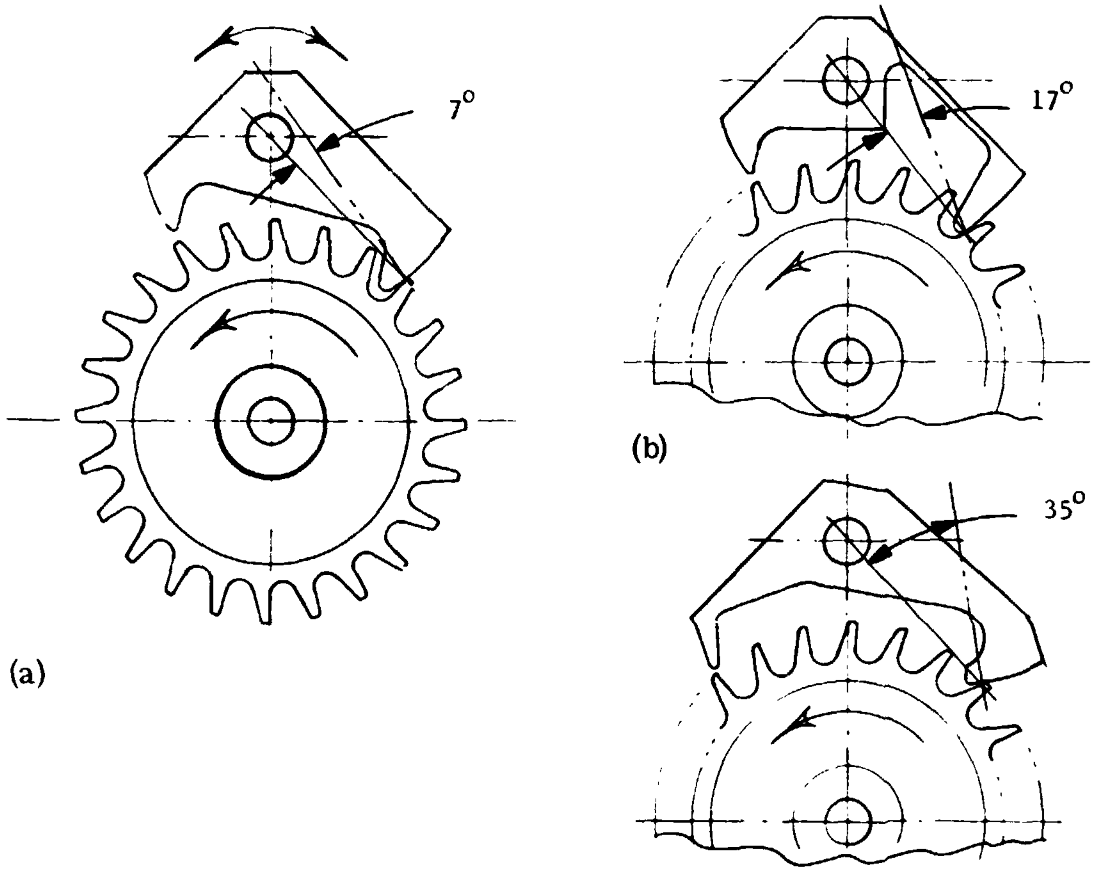

Figure 5.- Escapement geometry.