AD 653695

ADJUSTABLE MIRROR MOUNT DESIGN USING KINEMATIC PRINCIPLES

by

D. A. Bateman, Dip.Tech.

D D C

JUN 28 1967

ARCHIVE COPY

ROYAL AIRCRAFT ESTABLISHMENT

ADJUSTABLE MIRROR MOUNT DESIGN USING
KINEMATIC PRINCIPLES

by

D. A. Bateman, Dip. Tech.

SUMMARY

Various designs of mirror mounts are discussed in relation to their performance, construction, choice of materials, etc., together with the principles of kinematic design. Examples are given of systems incorporating many of the optimum design features described which have a precision of several seconds of arc and which are suitable for general optical and laser work.

CONTENTS

1   INTRODUCTION

Current work on solid state and gas lasers requires the use of precision mirror mounts for the adjustment and alignment of various optical components. Although many of the design principles described here are not new (e.g. Ref.1 - 1379), they are not widely known or practised and the purpose of this report is to discuss suitable designs of mirror mounts capable of adjusting to several seconds of arc with economy of materials and workshop effort.

High sensitivity of a system, in terms of angular change per adjustment increment, is not difficult to achieve, but precision, in terms of reproducibility to several seconds of arc, requires more careful design. If less precision is required, or if the mount setting is always checked or reset before use, simplifications and economies can be made. Precision to better than several seconds of arc is much harder to obtain since effects such as thermal expansion and dimensional stability become significant.

Before describing the various mirror mounts in detail it is appropriate to consider the basic principles of kinematic design, as these are applied wherever possible in this report.

2   KINEMATIC DESIGN

2.1   Basic principles

Kinematic designs in general have been described by many authors[1-6], but the more salient points are given below.

According to the principles of kinematic design[5], a body must have at least (6 - n) points in contact with a reference body if it is to have only n degrees of freedom relative to the reference body. Fig.1(a) shows a spherical ball held in a trihedral cavity by the force of gravity. Relative to the plate, the centre of the ball is uniquely defined by the three contacts with the plate. There remain three degrees of freedom of rotation for the ball about three mutually perpendicular axes. These rotational degrees of freedom are shown restrained by three more points of contact in Fig.1(b), by a ball touching two sides of a slot and a ball resting on a plane.

Features to note about this hole, slot, and plane mount are:

(a) For a given orientation the relative positions of the platform and baseplate are uniquely defined.

(b) The platform can be removed and replaced precisely in its original position, limits to this precision being the base line length between the supports, and wear or dirt on the contact points.

(c) The relative positions of the supports to each other are not critical.

(d) The direction of the slot should ideally intersect the hole for optimum restraint, but in practice this is not important. Also the angle of the slot should be approximately 90°.

(e) Point contacts are used which give high local loads and therefore hard surfaces should be used; this is one of the few disadvantages common to kinematic design.

(f) A constraining force is necessary, in this case provided by gravity.

(g) The manufacture of the mount does not require accurate machining of components, but relies on good geometric forms, such as a sphere and a plane, which can be made to near perfection.

The hole, slot, and plane method is commonly used in instruments such as a spectrometer where sets of prisms are required to be precisely and readily interchangeable.

An alternative type of mount is shown in Fig.1(c) which also satisfies the principles of kinematic design, complete restraint being given by six points of contact. This triple slot method is frequently used for locating surveying instruments, e.g. theodolites and levels, for which the triple slot has the significant advantage that instruments of different sizes can be mounted on one base.

2.2 Manufacture of holes

Various methods of manufacturing holes for kinematic designs are illustrated in Fig.2.

The simplest and most satisfactory method of producing a trihedral hole is to use a pyramidal punch as shown in Fig.2(a). Normally this method is confined to the use of a hardened steel punch and relatively soft alloys such as brass and duralumin. For harder metals, milling methods can be used but the machining operations tend to be complicated. Spark machining and ultrasonic drilling could also be used.

An extremely good alternative is shown in Fig.2(b). Three steel balls of diameter d are pressed into a cylindrical hole of diameter slightly less than D, where $D = d(1 + \sec 30°)$. A fourth ball will have a three point contact and the hard steel surfaces make this arrangement ideal for a heavily loaded or much mount. One disadvantage is that there is poor lateral restraint i.e. a small sideways force will cause the fourth ball to slip out between two of the fixed

balls, but this may be avoided by using a fourth ball which is much smaller than the other three.

Another approach[7] is given in Fig.2(c) which shows how a three sided hollow with plane faces can be machined in the surface of a hard material by making three cuts at 120° to one another with a 45° angular milling cutter or grinding wheel.

Compromise methods are illustrated in Fig.3. In the first a small pilot hole is drilled, and then suitably larger holes are also drilled at the corners of an equilateral triangle so that the holes almost touch. A 90° countersunk drill, centred on the pilot hole, is finally used to produce three narrow sloping lands which are part of the surface of a cone. A ball resting in this hole has three line contacts which form part of a common circle. Although this is not true kinematic design the line contacts can be short, and this arrangement is easy to manufacture. This technique may be used in reverse i.e. with a conical hole and a spherical ended foot having three flats machined on the surface to reduce the extent of the line contact.

A straightforward conical hole made with a countersunk drill is the simplest method of all but unfortunately does not satisfy kinematic principles. If the hole were perfectly circular and the ball perfectly spherical then a circular line contact would be obtained, but a hair or piece of grit on this line would transform the situation to one analogous to a four legged stool on an uneven surface. Furthermore, it is nearly impossible to produce circular conical holes on conventional workshop machines. A drill or countersink drill can be expected to produce errors of 5×10^{-3} inches in roundness, and by boring using a lathe errors of 5×10^{-4} inches can be expected[6]. Conical holes are often acceptable in simple design, but are shown to be unsatisfactory for the precision mounts in Section 5.1.

2.3 Manufacture of slots and planes

Milling machines offer the simplest means of producing slots. Alternatively a slot can be filed by hand into the head of a bolt, since it is generally the surface finish and hardness of the slot which is more important rather than overall form. For the plane surface it is again more important to have a smooth hard surface than attempt perfect flatness or freedom from tilt. As the quality of kinematic design is improved by using hard, unyielding contact points, it is usual to employ hard steel or glass inserts for slot or plane surfaces.

3   DEVELOPMENT OF THE MIRROR MOUNT

A device for making angular adjustments, as in a mirror mount, can be developed using kinematic principles by, for example, introducing one or more screws into the system as in Fig.4(a). The sensitivity of this arrangement is controlled by the length of the base line and the pitch of the screw, and the precision is determined by the quality of the screw, hinge, rigidity, and constructional materials.

The main screw errors are looseness and lack of straightness. A loose screw in the reference plate can give variable angular errors due to variation of the base line which may occur under vibration or when the screw is touched. If the screw is bent or if the contact point does not lie on the axis of the screw then periodic angular errors can result. Apart from the special type of screw which is manufactured for use in diffraction grating ruling engines, a micrometer head which conforms to British Standard 1734 : 1951 probably represents the most accurate form of thread readily available[8]. The accuracy of flatness of the measuring face, squareness of the spindle, and pitch errors are within 3×10^{-4} inches (Table 1). An added advantage to this precision is the graduated scale which enables a mirror mount to be conveniently adjusted through a known angle or reset to a specific reading. Fig.4(b) shows two methods of using micrometer heads. Unless extremely hard materials are used, the first is often the best way of forming a point contact with a plane in mirror mount design. The second is used if the micrometer needs to act into a hole or slot. In this case, unless the micrometer spindle has a hemispherical tip specially made (which can be done by the manufacturers at small extra cost), a ball must be restrained about the spindle axis, and the accuracy of this assembly is governed by the precision of the collar.

The most common form of hinge in kinematic design consists of point contacts sliding over a spherical surface as in Fig.4(a). Clearly the quality of the hinge is controlled by the surface finish of the contact surfaces and the uniformity of the spherical component. Steel balls as used in rolling bearings provide the best possible form of hinge: they are spherical to at least 4×10^{-5} inches[9], hard, highly polished, and inexpensive.

Lack of rigidity is unlikely to be a source of error as other demands, such as machinability, requirements for tapped holes, etc., ensure adequate thickness of components. However, the rigidity must be sufficient so that the constraining spring does not produce any appreciable bending of components over

the working angular range. The choice of materials and effects such as dimensional changes with time (ageing) are discussed in Section 5.5.

4   SIMPLE MIRROR MOUNT DESIGNS

Two simple general purpose mounts are shown in Fig.5. Both designs use the hole, slot and plane method with the common, or corner, hinge point acting in a trihedral hole, and one of the adjusting screws acting in a slot. Note that the adjusting screws are positioned so that the hinge lines intersect one another at right angles enabling orthogonal adjustments to be made - a practical and psychological advantage.

In Fig.5(a) the hinge lines intersect at the centre of the mount: this may be desirable in a focussed system where an angular adjustment would not necessarily lead to an accompanying linear displacement, as would always occur in the design in Fig.5(b).

Generally, however, it is more useful to move the contact points to give as large a base line as convenient (Fig.5(b)) thus obtaining improved sensitivity and also permitting transmission through the centre of the mount. The mount in Fig.5(b), which rests on a kinematically designed base, has been designed by colleagues and has found many applications in general optics and spectroscopy in this laboratory. The sensitivity of this system with a $1\frac{1}{2}$ inch base line and a 40 T.P.I. screw is approximately 1 degree per revolution of the screw.

Generally, the designs discussed in this report have the hole, slot, and plane lying in one plane, but Fig.6 shows a special variant of this layout. This mount was constructed for use in a laser Q-switch and illustrates how kinematic principles can be applied to a more unusual situation.

5   PRECISION MIRROR MOUNT DESIGNS

5.1   Double hinge systems

As pointed out in Section 3, the precision and sensitivity of a mirror mount depends mainly on the design of the adjusting screw and hinge points, and on the length of the base line. For these reasons all of the following mounts are larger than those discussed earlier and use micrometer screws where possible.

Fig.7 shows an interesting design due to W. T. Nichols, Weapons Department, where two kinematic systems are placed one on top of the o er enabling orthogonal adjustments to be made in two completely independent planes. Cast iron is used to give good dimensional stability, but owing to the hardness

of the material trihedral holes cannot be readily used and the steel hinge balls rest in conical holes. The sensitivity of this design with a 3.375 inch base line and a 60 T.P.I. screw is about 17' per turn of the screw and is claimed to be very stable.

A development of this design by the author is shown in Fig.8 which has several small but important differences. The shape has been simplified to reduce machining operations, an aluminium alloy is used which again simplifies machining and allows trihedral holes to be pressed into the surfac and micrometer heads are used for the adjustment screws. The micrometer graduations allow the mount to be adjusted and returned to a previous setting, and in addition, the length of the base line of the hinge can be made so that the micrometer graduations correspond conveniently to minutes or multiples of minutes of arc movement. Table 2 lists some useful sets of these multiples and their related base lines.

Using the design shown in Fig.8 some experiments were carried out with an autocollimator to check the precision of the mount with different types of holes for the hinge balls. Two mounts, both using conical holes, were found to be unsatisfactory i.e. the reflected image described a hysteresis loop when the micrometers were rotated to and fro, giving errors of about one minute of arc. The multi-ball hinge (Fig.2(b)) worked well but owing to the poor lateral constraint of this system (Section 2.2), the hinge ball could become unseated if a heavy load was placed on the front plate. The trihedral and "clover leaf" holes (Fig.2(a) and Fig.3) were satisfactory and the simplest to manufacture. Using trihedral holes and a base line giving 20' per turn (metric micrometer head) the mount setting was reproducible to within 6 seconds of arc regardless of the directions of rotation of the micrometer heads: this corresponds to the accuracy with which the micrometer graduations could be conveniently read.

Another version of the double hinge mount is shown in Fig.9 (G. Newall, University of Southampton). At the expense of sensitivity the hinge lines have been moved to intersect at the centre of the mount, and as mentioned earlier (Section 4) this is an advantage if linear motions in a focussed system must be eliminated. But of particular interest is the method of constraint. The main constraining forces for each plane are supplied by two springs acting on the hinge line, where in this position the tension remains nearly constant regardless of the angular settings, and constraint against the micrometer is by a light compression spring enabling the micrometer to have a light "feel".

5.2   Corner hinge systems

Fig.10 shows a precision corner hinge design using micrometers. One micrometer acts into a slot with a ball attached to the spindle as in Fig.4(b), with the other acting onto a recessed ball.

Errors in this mount could arise from the ball-ended micrometer if the ball did not rotate accurately about the spindle axis. Any such eccentricity could produce errors in two planes - the plane controlled by the ball-ended micrometer, and more seriously, in the plane controlled by the other micrometer. The former errors appear as periodic angular errors due to periodic changes in base line length along the axis of the slot, and the latter as oscillations due to the recessed ball sliding across the measuring face of the second micrometer. The errors will be minimum when the reference plate and adjustable plate are parallel, and maximum at an extreme setting of the ball-ended micrometer. However, calculations show that with the ball mounted to within $\pm 2 \times 10^{-3}$ inch of the spindle axis, these errors are negligible.

Because of the simplicity of this design, it is generally to be preferred to the double hinge systems, which offer no special advantages in comparison, with the possible exception of the Newall design (Fig.9) which permits axial transmission of an optical beam and where the hinge lines intersect at the centre of the mount. The double hinge system would be theoretically better for higher precision work because of the complete independence of the adjustable planes.

5.3   Locking devices

If a mount is to be used in a vibrating environment the constraining forces must be sufficient to prevent the contacts between the reference and adjustable plates from becoming separated. Any intermittent separation would accelerate wear, permit the introduction of dirt, and the adjustment screws may rotate. The screws are in fact the only items in kinematic design which can be locked; any constraint other than a strong spring between the reference and adjustable plates will violate the principles of kinematic design.

If it is required to lock screws this should be done with a threaded collet or similar clamp as in Fig.7, which is more satisfactory than lock-nuts or grub screws. Micrometer heads cannot be locked but the micrometer main nut can be tightened to stiffen the rotation of the spindle. However, this introduces the disadvantage that the spindle tends to rotate jerkily and wear is increased.

In general, a well designed mount with adequate constraining forces should not require any form of locking.

5.4   Temperature effects

The linear expansion coefficients of steel, stainless steel, cast iron, brass, and aluminium alloys lie in the range 10 - 20 x 10^{-6} per $^{\circ}$C, steel and cast iron representing the lower values and brass, stainless steel and aluminium alloys representing the higher values. The errors that could be produced in the light alloy double-hinge design (Fig.8) were computed assuming temperature differentials of 10°C between various components of the mount and in no case were the errors more than 3 seconds of arc. Temperature effects therefore, for the mounts described here, in normal room temperature conditions, may be taken as unimportant.

5.5   Choice of materials

The main factors influencing choice of constructional materials for mirror mounts are ease of machining, cost, thermal properties (although generally unimportant - Section 5.4), and dimensional stability. For ease of machining, brass and aluminium alloys are obvious choices and they are also sufficiently malleable to permit the pressing of trihedral holes without the use of a powerful press. Cost of materials generally, with the possible exception of stainless steels, is not usually a significant factor in relation to the total cost of a scientific instrument, but it is worth noting that mild steel and cast iron are cheaper than brass and aluminium alloys.

A possible source of error in a mirror mount, especially if it is to remain aligned for a period of weeks or months, is the dimensional instability of the components. Dimensional changes are due to (a) decomposition of an unstable or metastable phase in a metal or alloy, and (b) the relaxation of residual stresses present in the material as the result of prior machining. In steel, phase changes in the crystal structure can occur over periods of months or years at room temperature and can affect dimensions by 0.001 inch per inch in extreme cases, but normally the changes are of the order of tens of micro-inches per inch[10,11,12]. To eliminate or minimise these defects careful heat treatment must be applied. In aluminium alloys, which are precipitation hardened, the precipitation processes can continue at room temperature, causing dimensional and hardness changes. Very little comparative data is available, but experience at the Royal Aircraft Establishment among metallurgists and gyroscope specialists indicates that dimensional stability increases in the following order: aluminium alloys, brass, carefully aged cast iron, and

correctly heat treated and stress relieved mild steel and tool steel. For
general purposes dimensional instability is unlikely to be significant;
however, for a mount which is required to be precise to better than several
seconds of arc and which must remain at a fixed setting for say, several months,
dimensional qualities should be borne in mind. Micrometer heads, which can be
a vital part of a mirror mount, are not likely to be a source of error. The
type manufactured to B.S. 1734 : 1951 is made from a 1% carbon steel which is
carefully aged to a stable condition.

5.6   Overall design

When using a precision mirror mount, attention must also be given to the
method of installing it into the optical system being used, and into the
apparatus generally. A convincing demonstration of the type of problem that
might be encountered was shown by the relative ease with which a standard 2
metre commercial optical bench could be bent. With the bench supported at its
$\frac{1}{4}$ and $\frac{3}{4}$ length points the weight of one 60 cm × 60 cm saddle stand (weight ~ 1 lb)
at either the centre or the ends of the bench was sufficient to cause a bending
of several seconds of arc.

The method of attaching mirror mounts on a single stem as shown in
Figs.6-9 is not ideal. For example, to touch one of the micrometers produces
bending of the stem, reliance being placed on good elasticity to return the
mount to its original position. Although the single stem is satisfactory for
the degree of precision considered here (up to several seconds of arc), any
improvement would necessitate placing the mirror mount, preferably using
kinematic principles, on a sturdy base, similar to the method used in Fig.5(b).

6   NON-KINEMATIC DESIGNS

In all the designs reviewed the hinge consists of point contacts sliding
over a spherical surface. An alternative to this is to use a flexible metal
pillar, plate, or spring strip attached rigidly to both reference and adjustable
platform. This is a departure from kinematic practice but the main criticism
of such a hinge is that forces in the hinge tend to be high i.e. a significant
fraction of the yield stress of the metal, and permanent strains and creep
could effect the overall stability. A flexible hinge will sag depending on
the load, and for this reason a mirror mount with such hinges could have a
variable performance for heavy or light mirrors.

Good mirror mounts however, can be made using such hinges, as described
by Collins and Smith[13]. They used the double-hinge system with micrometer
heads and the two hinge lines each consisted of two flexible hinges. Each

hinge was made from a solid beryllium copper rod, diameter 8 mm, the centre
portion being cut away on each side with a smooth radius to give a centre
thickness of 0.4 mm, most of the bending taking place across this central
portion. Springs were also used to provide constraint against the micrometer
measuring faces.

7   CONCLUSIONS

A critical discussion of several designs of mirror mount has been given,
accompanied by details of construction and performance While it is clear
that one specific design of mirror mount will not be ideal for every application,
this report does lay the foundation for the good design of systems in general,
and for the extension to even more precise forms of mirror mount which are not
discussed here. For mounts capable of precision up to several seconds of arc
the following recommendations are made:

(i)   For simplicity and ease of construction kinematic principles
should be strictly applied, preferably using a trihedral hole, a 90° slot, and
a plane - all with hard polished surfaces.

(ii)   Micrometer heads should be used for the screw adjustment.

(iii)   Steel balls of the type used in rolling bearings, which are hard,
polished, and nearly perfectly spherical, offer the simplest form of hinge.

(iv)   The corner hinge system is the simplest to use (Fig.10); the
double hinge method offers few advantages.

(v)   For ease of machining, light alloys or brass may be employed in
the main structure.

For a mount which is required to remain aligned for long periods,
e.g. months, or to have a precision of better than several seconds of arc, the
following recommendations, in addition to (i), (ii), and (iii) above, should
be noted.

(vi)   Correctly heat treated and stress relieved mild steel or tool
steel for the construction of the mount should be used to give minimum
dimensional instability and temperature effects.

(vii)   Longer base lines, and/or Type 2 or Type 3 micrometer heads with
larger diameter thimbles, should be used to give the necessary improvement in
sensitivity.

(viii) Although the single stem method of mounting used generally in this report is convenient and satisfactory, the stem should be made short or eliminated for very precise work, and the mount carefully integrated with any other apparatus, preferably using kinematic techniques.

Table 1

Maximum permissible errors for micrometer heads

From B.S. 1734 : 1951

For nomenclature see Fig.4(b)

Diameter of thimble	Flatness of measuring face	Squareness of measuring face to axis of spindle (measured over diameter of face)	Squareness of measuring face to ouside diameter of shank (measured over diameter of face)	Maximum error of traverse of measuring face at any position
Type 1 1 in	0.000 05 in (0.001 mm)	0.000 05 in (0.001 mm)	0.000 3 in (0.008 mm)	0.000 1 in (0.003 mm)
Type 2 1 in-2 in	0.000 03 in (0.0008 mm)	0.000 03 in (0.0008 mm)	0.000 1 in (0.003 mm)	0.000 1 in* (0.003 mm)*
Type 3 > 2 in	0.000 02 in (0.0005 mm)	0.000 03 in (0.0008 mm)	0.000 03 in (0.0008 mm)	0.000 1 in* (0.003 mm)*

*The periodic error shall not exceed ±0.000 05 in (±0.001 mm) for Type 2 micrometer heads and ±0.000 02 in (±0.0005 mm) for Type 3 micrometer heads.

Table 2

Base line lengths for convenient angular equivalents to micrometer head graduations

Metric micrometer heads (screw pitch 0.5 mm)

Minutes per turn	Division per* minute	Base line	
		cm	inches
50	i.e. 1 div/min	3.438	1.353
25	2 div/min	6.875	2.707
20	5 div/2 min	8.594	3.383
12.5	4 div/min	13.750	5.414
10	5 div/min	17.188	6.767

Inch micrometer heads (screw pitch 0.025 in)

50	i.e. ½ div/min	4.366	1.719
25	1 div/min	8.732	3.438
12.5	2 div/min	17.463	6.875
10	5 div/2 min	21.829	8.594

*This column applies to Type 1 micrometer heads with small thimble diameters.

REFERENCES

No.	Author	Title, etc.
1	P.G. Tait W. Thomson	Treatise on natural philosophy, Vol.I, p.152, Cambridge University Press 1879
2	J.C. Maxwell	Scientific papers, Vol. II, p.507, Cambridge University Press 1890
3	A.F.C. Pollard	The kinematical design of couplings in instrument mechanisms. London: Adam Hilger 1929
4	T.N. Whitehead	The design and use of instruments and accurate mechanisms. New York: Macmillan 1934
5	J. Strong	Modern physical laboratory practice. Blackie and Son, London 1940
6	A. Elliot J. Home Dickson	Laboratory instruments. Their design and application. Chapman and Hall, London 1951
7	J.F. Darby	A note on the hole, slot and plane method. J. Sci. Inst., 31, 224, 1954
8	British Standards Institution	Micrometer heads. B.S. 1734 : 1951
9	R.K. Allan	Rolling bearings. Pitman 1946
10	E.B. Mikus T.J. Hughel J.M. Gerty A.C. Knudsen	The dimensional stability of a precision ball bearing material. Trans. Am. Soc. Met. 52, 307-320 1960
11	F.C. Victory	Dimensional instability effected design of precision equipment. American Machinist, 1948 February 26 p.108-111 March 11 p.144-147 March 25 p.112-115 April 8 p.112-115

REFERENCES (Contd)

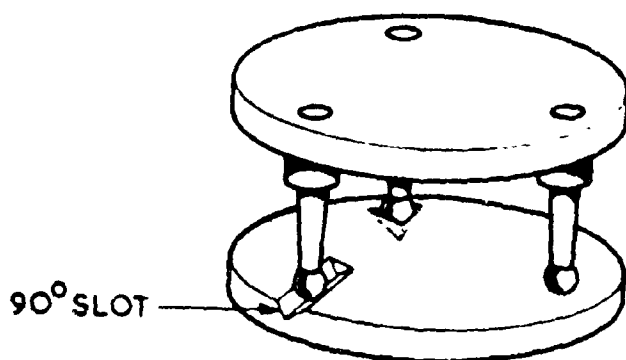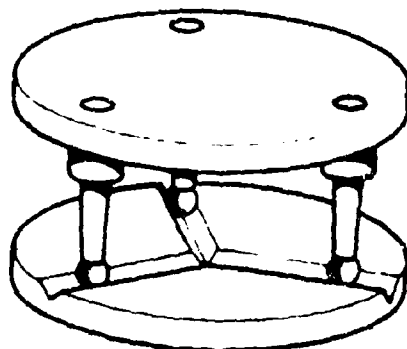| No.	Author	Title, etc.
12	National Bureau of Standards	Extremely stable gauge blocks. NBS Technical News Bulletin, 44, 82, May 1960
13	L.J. Collins J. Smith	Mirror mounts for experimental optical masers. J. Sci. Instr., 42, 499-500, 1965

008-900617

Fig.1
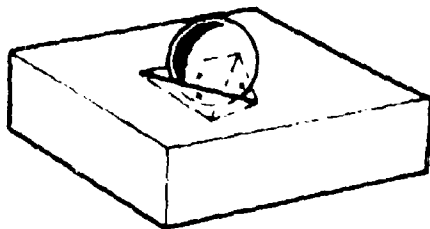
(a)   BALL IN TRIHEDRAL HOLE

90° SLOT

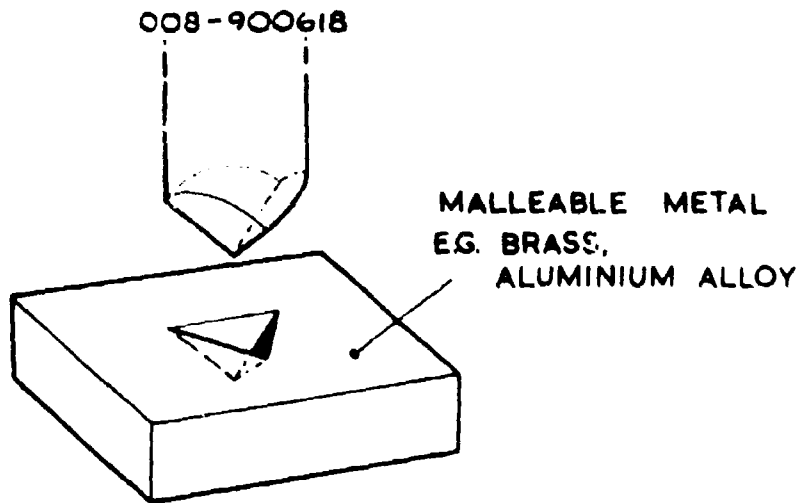(b)   HOLE, SLOT, AND PLANE MOUNT
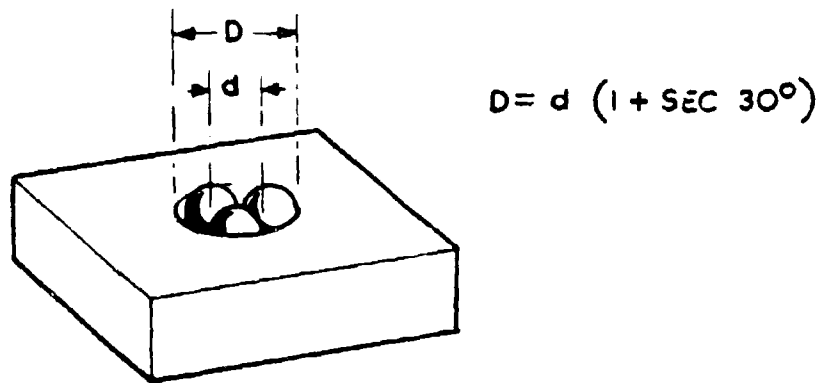SIX POINTS OF CONTACT, ZERO DEGREES OF FREEDOM

(c)   TRIPLE SLOT MOUNT

FIG.1 MOUNTS USING KINEMATIC DESIGN

Fig.2

008-900618

MALLEABLE METAL
E.G. BRASS,
ALUMINIUM ALLOY

(a)   STEEL PUNCH FOR TRIHEDRAL HOLES

$D = d \left(1 + \sec 30°\right)$

(b)   STEEL BALLS TIGHTLY FITTING IN COMMON HOLE

CUT 2

MACHINED WITH
45° MILLING CUTTER

CUT 3

CUT 1
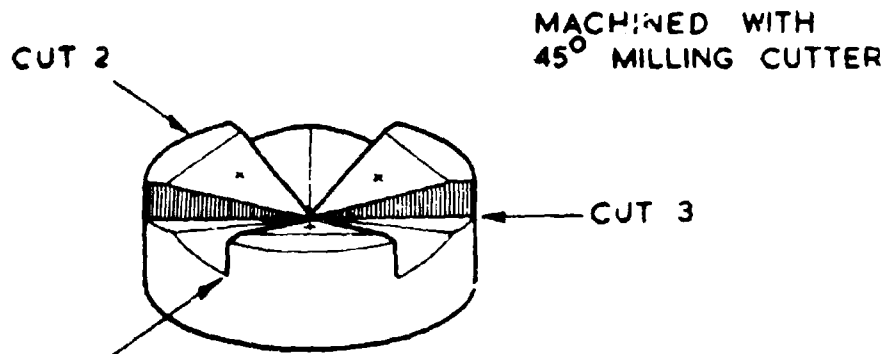
(c)   MACHINED TRIHEDRAL HOLLOW

FIG.2 METHODS OF MANUFACTURING HOLES
FOR KINEMATIC DESIGN
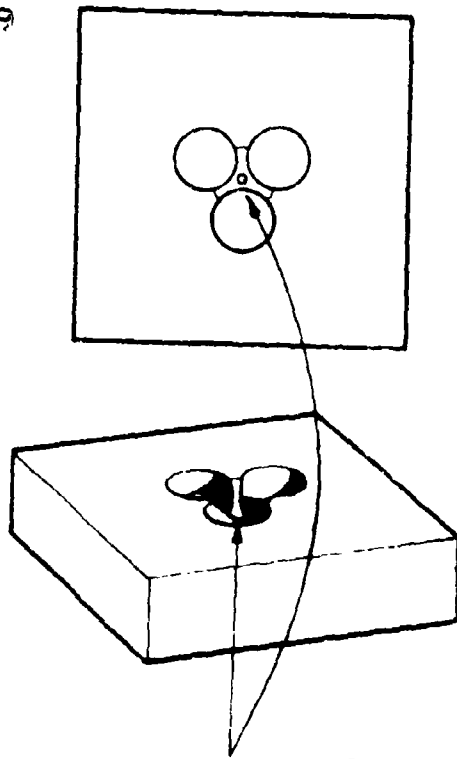
Fig.3

CENTRE REMOVED WITH 90° COUNTERSUNK
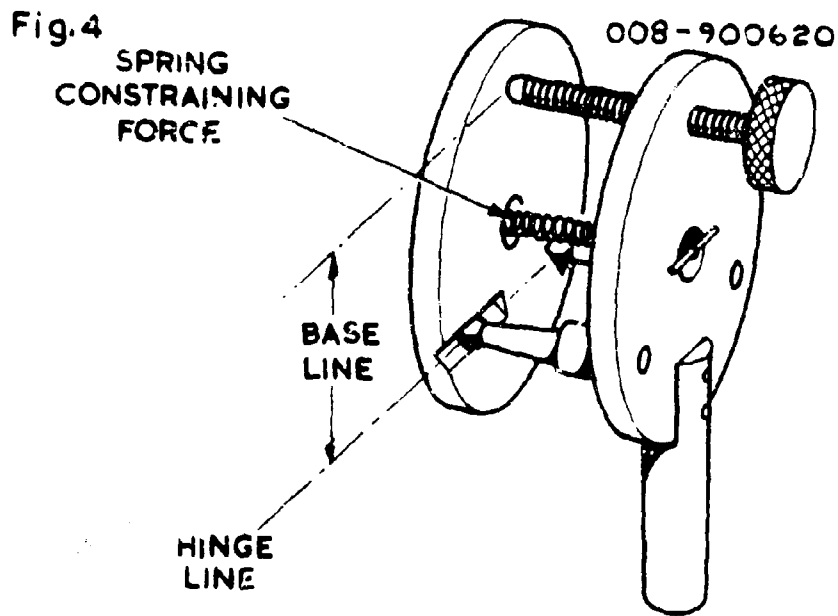DRILL TO LEAVE THREE NARROW SLOPING LANDS
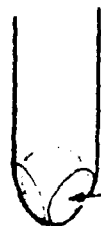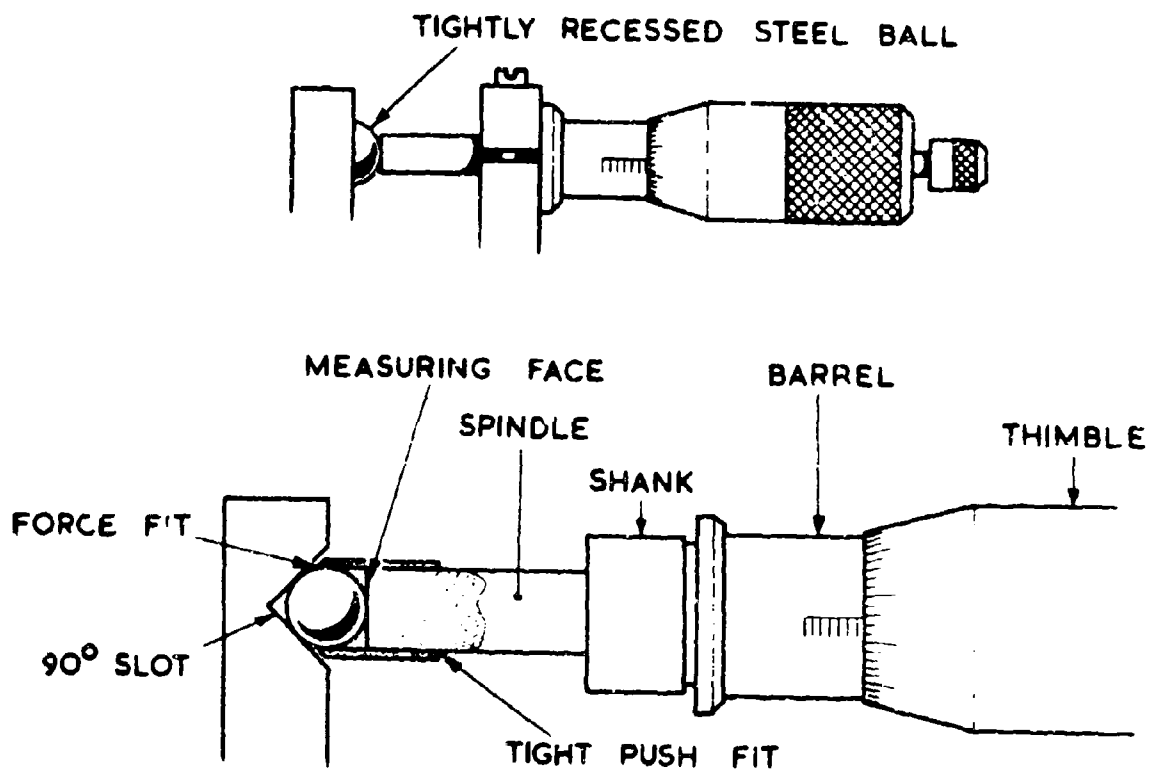
THREE FACES CUT TO
LEAVE NARROW CONTACTING
RIDGES
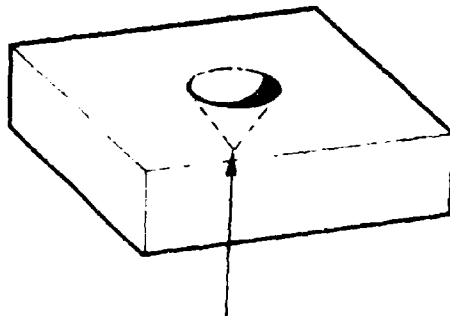
CONICAL HOLE

FIG. 3 METHODS OF MANUFACTURING
HOLES, COMPROMISE FORMS

Fig. 4

008-900620

SPRING
CONSTRAINING
FORCE

BASE
LINE

HINGE
LINE

(a)   SIMPLE ADJUSTABLE MOUNT

TIGHTLY RECESSED STEEL BALL

MEASURING FACE   BARREL

SPINDLE

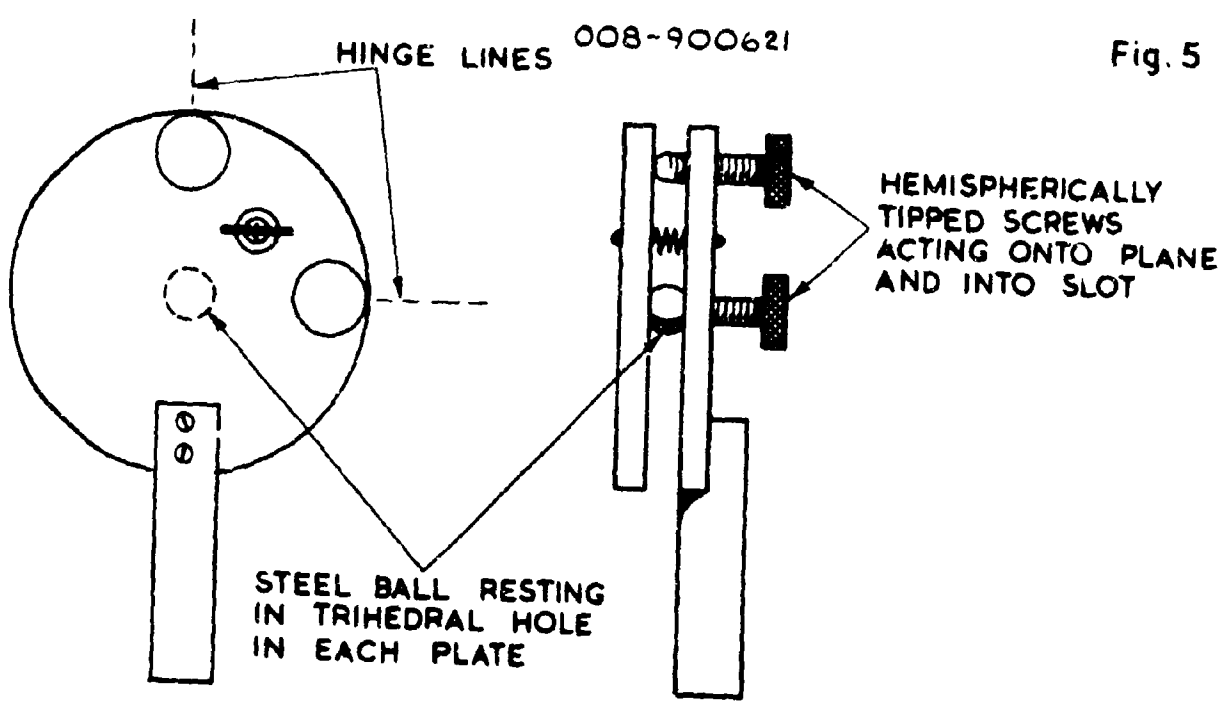SHANK   THIMBLE

FORCE FIT

90° SLOT

TIGHT PUSH FIT

(b)   METHODS OF USING MICROMETER HEADS

FIG. 4 DEVELOPMENT OF ADJUSTABLE MOUNT
FROM HOLE, SLOT, AND PLANE METHOD

HINGE LINES   008-900621   Fig. 5

HEMISPHERICALLY
TIPPED SCREWS
ACTING ONTO PLANE
AND INTO SLOT

STEEL BALL RESTING
IN TRIHEDRAL HOLE
IN EACH PLATE

(a)

HEMISPHERICALLY TIPPED
SCREW AS CORNER HINGE
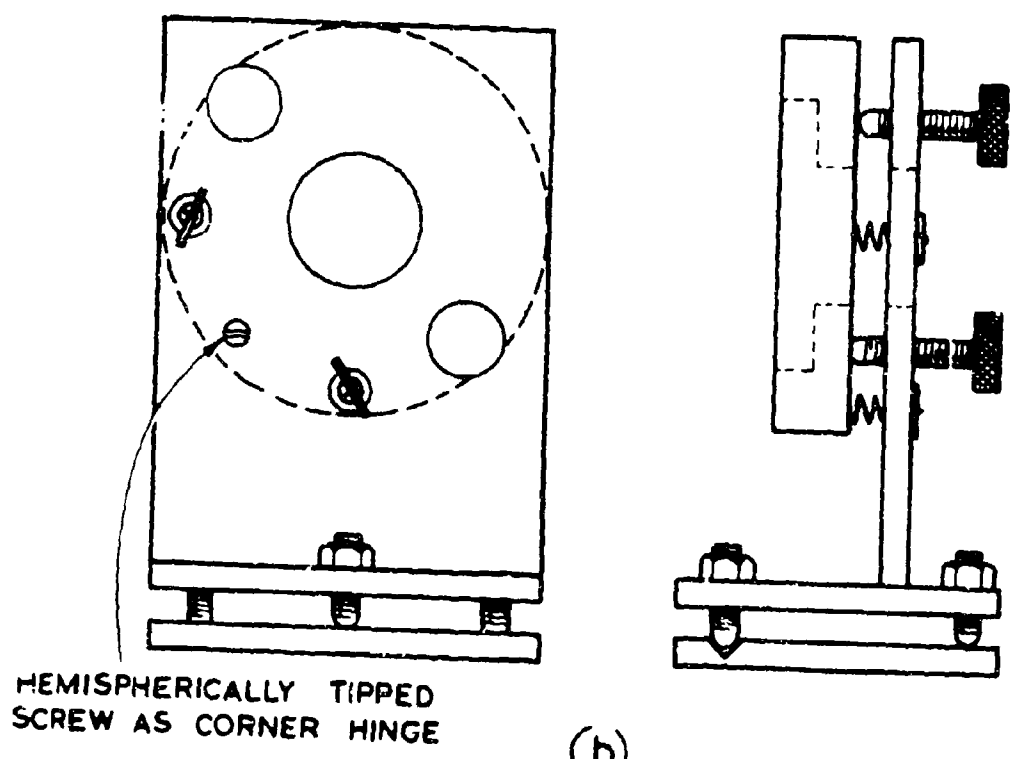
(b)

FIG. 5   SIMPLE MIRROR MOUNT DESIGNS
WITH CORNER HINGES

Fig.6   008-900622

PAIR OF
CONSTRAINING
SPRINGS

BALL IN HOLE

HEMISPHERICALLY TIPPED
SCREW ACTING ONTO PLANE

HEMISPHERICALLY TIPPED
SCREW IN SLOT
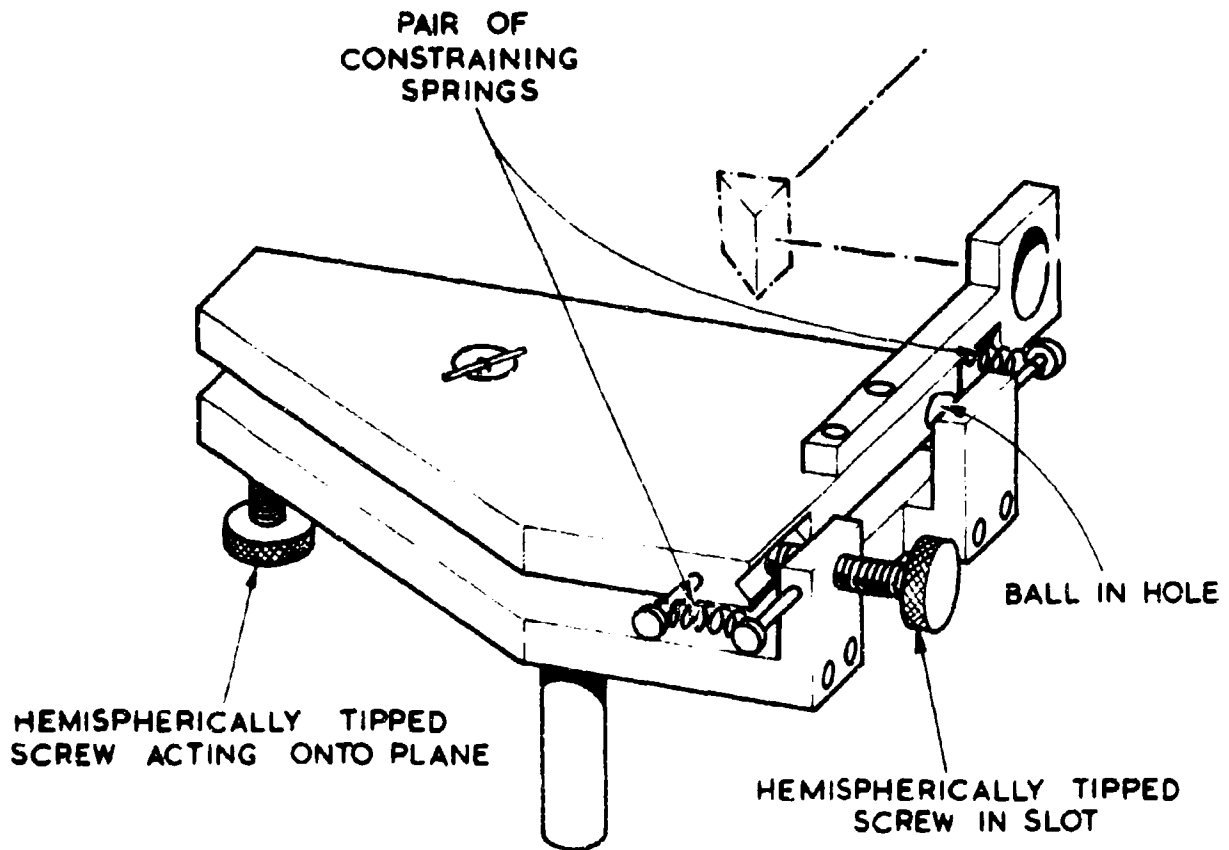
FIG. 6 SPECIAL PURPOSE MOUNT USING
KINEMATIC PRINCIPLES

$3\frac{1}{8}$ IN.

CAST IRON

60 T.P.I. THREAD

STEEL BALL IN HOLES

SCREW LOCK

BALL IN HOLE AND SLOT

ELEVATION HINGE LINE

AZIMUTH HINGE LINE
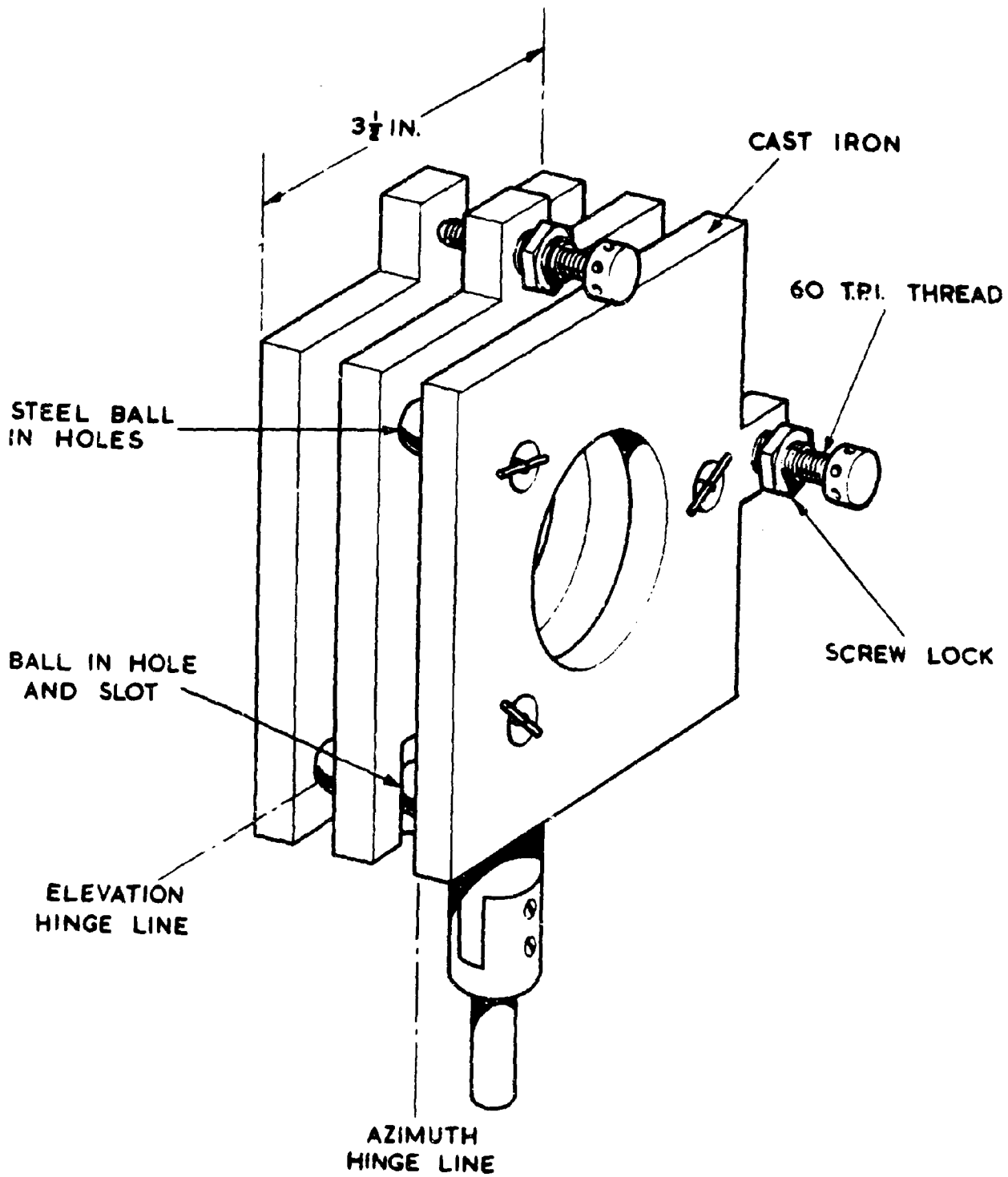
FIG.7 A VERSION OF A DOUBLE HINGE MOUNT

Fig. 8   008-900624

4 IN.

LIGHT ALLOY

RECESSED STEEL BALL

STEEL BALL IN TRIHEDRAL HOLES

BALL IN TRIHEDRAL HOLE AND SLOT
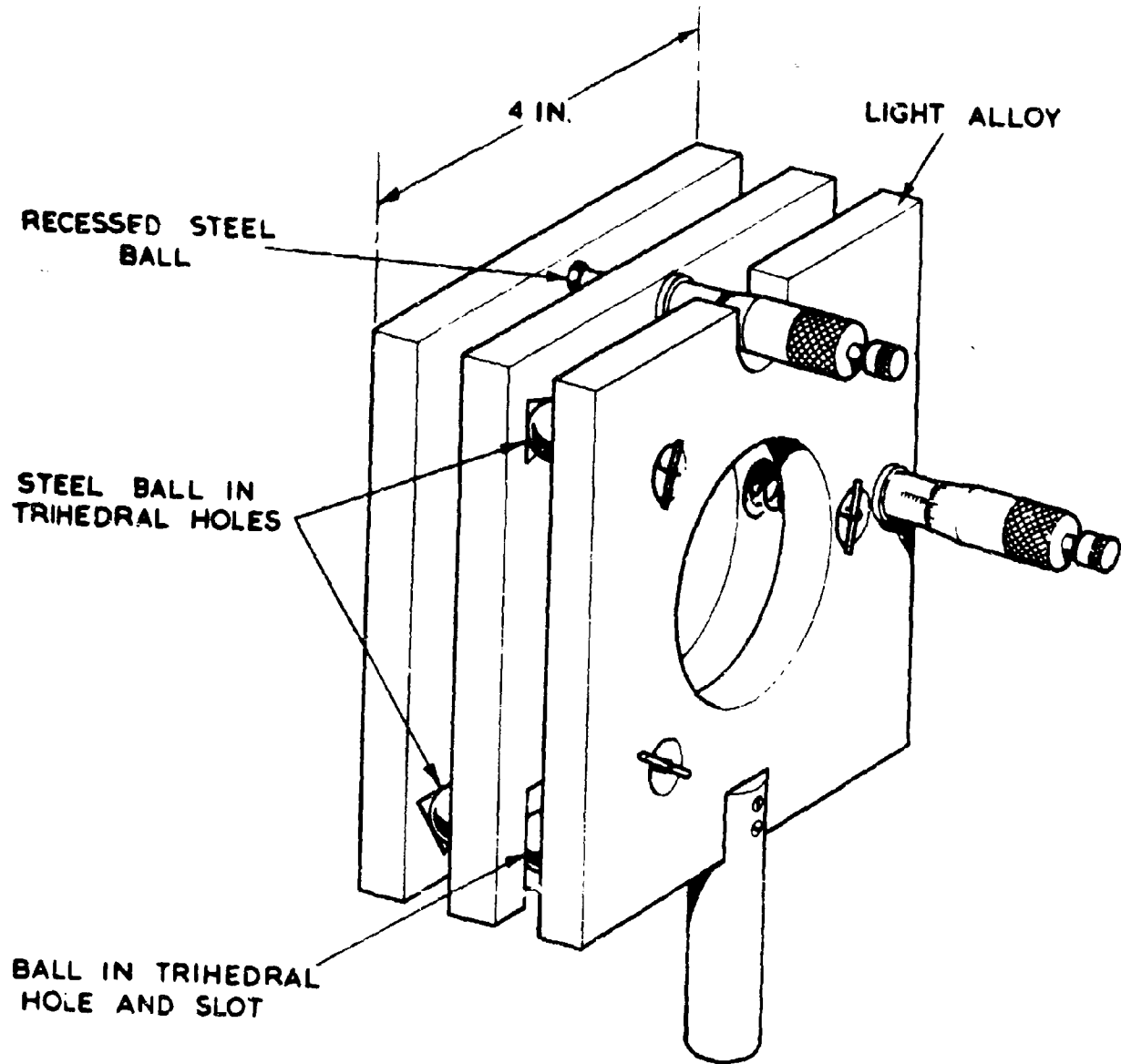
FIG. 8 A DOUBLE HINGE MOUNT USING
MICROMETERS

AZIMUTH HINGE LINE

STEEL BALL IN
TRIHEDRAL HOLE
AND SLOT

ELEVATION
HINGE LINE

STEEL BALL IN
TRIHEDRAL HOLES

COMPRESSION SPRING
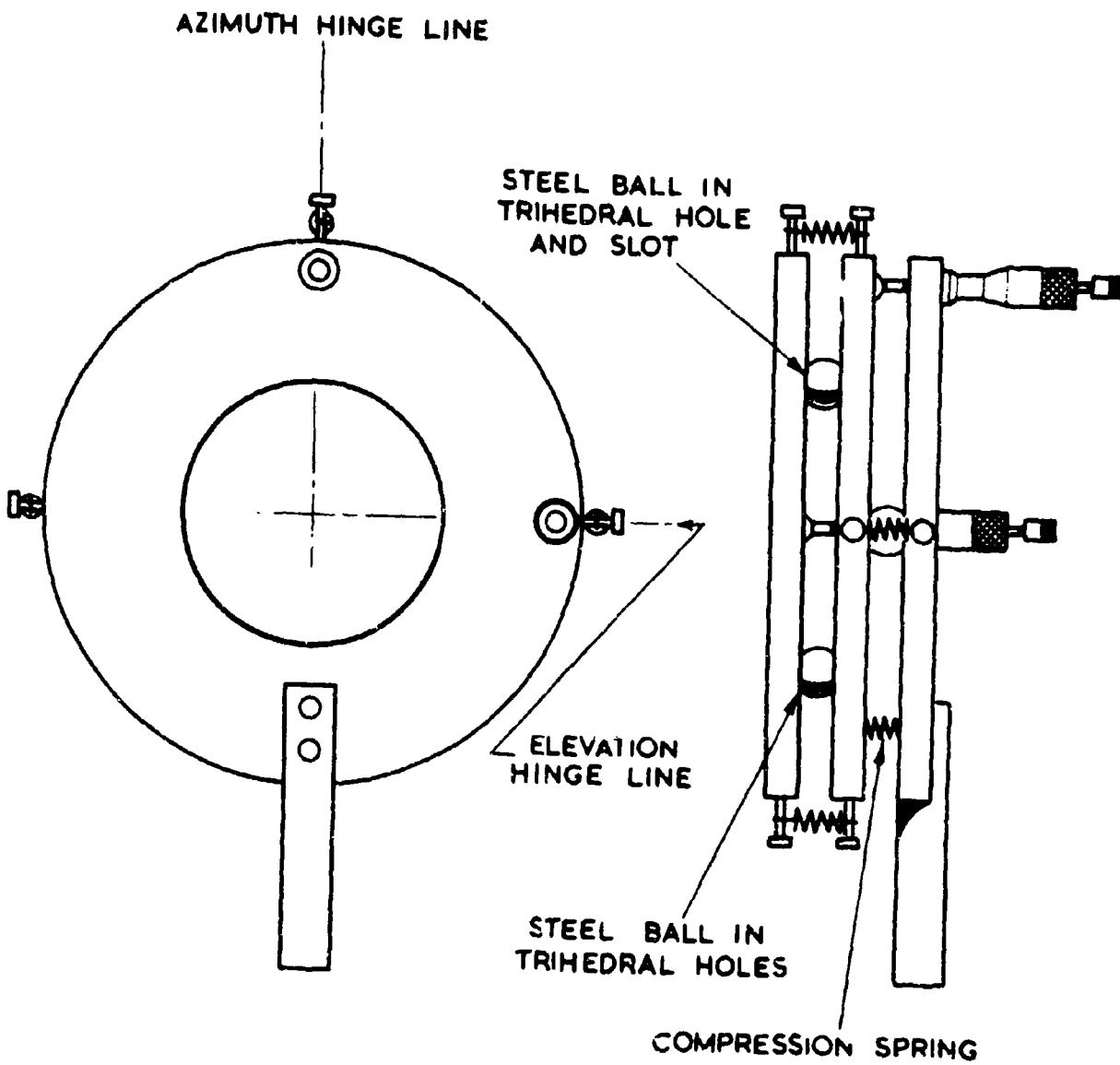
FIG.9 ANOTHER VERSION OF THE DOUBLE
HINGE MOUNT

Fig.10   008-900626

RECESSED STEEL BALL
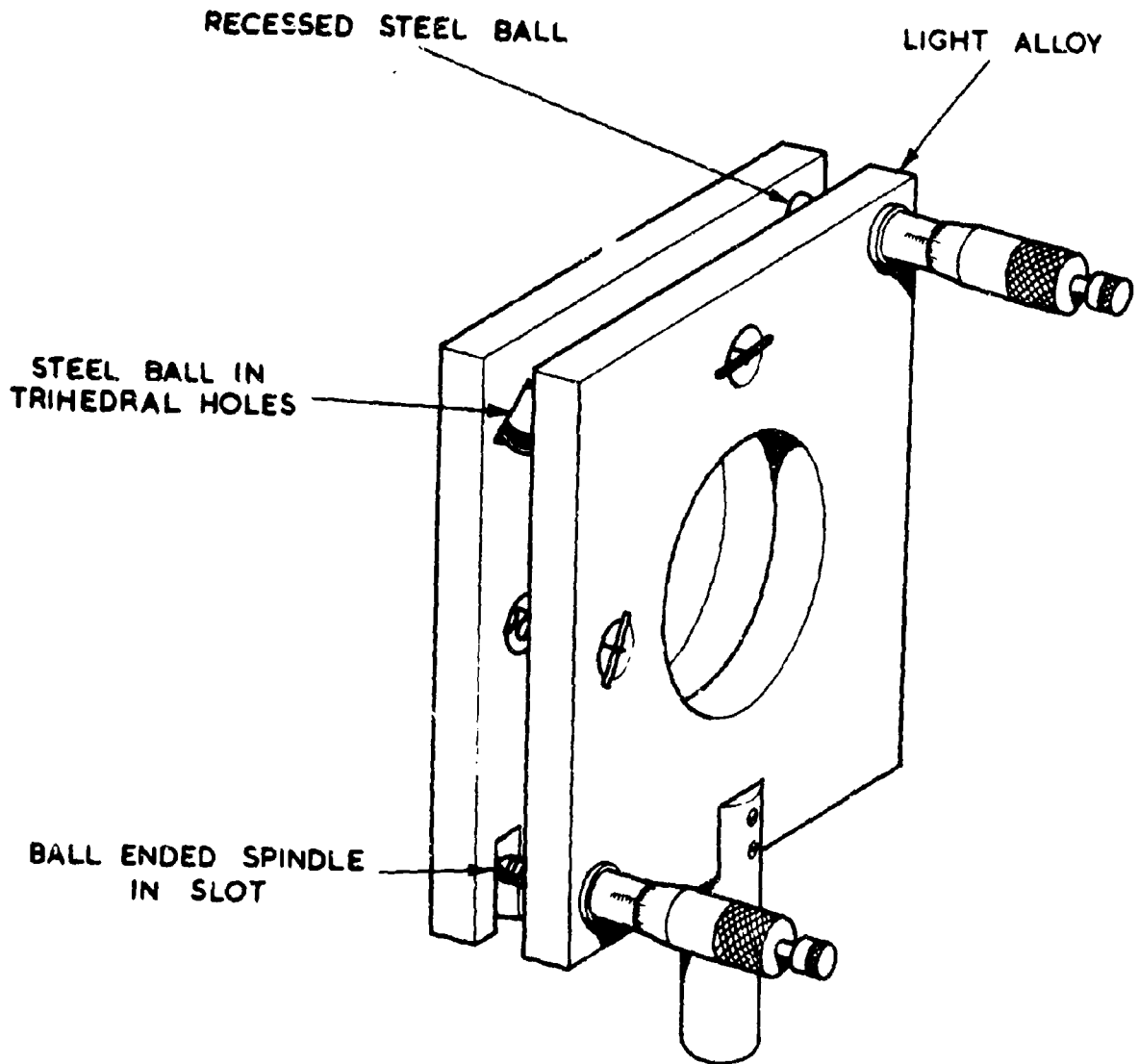
LIGHT ALLOY

STEEL BALL IN
TRIHEDRAL HOLES

BALL ENDED SPINDLE
IN SLOT

FIG. 10 PRECISION CORNER HINGE DESIGN