EP-RR 5

THE MARK II COUPLING AND ROTOR CENTERING REGISTERS FOR THE CANBERRA HOMOPOLAR GENERATOR

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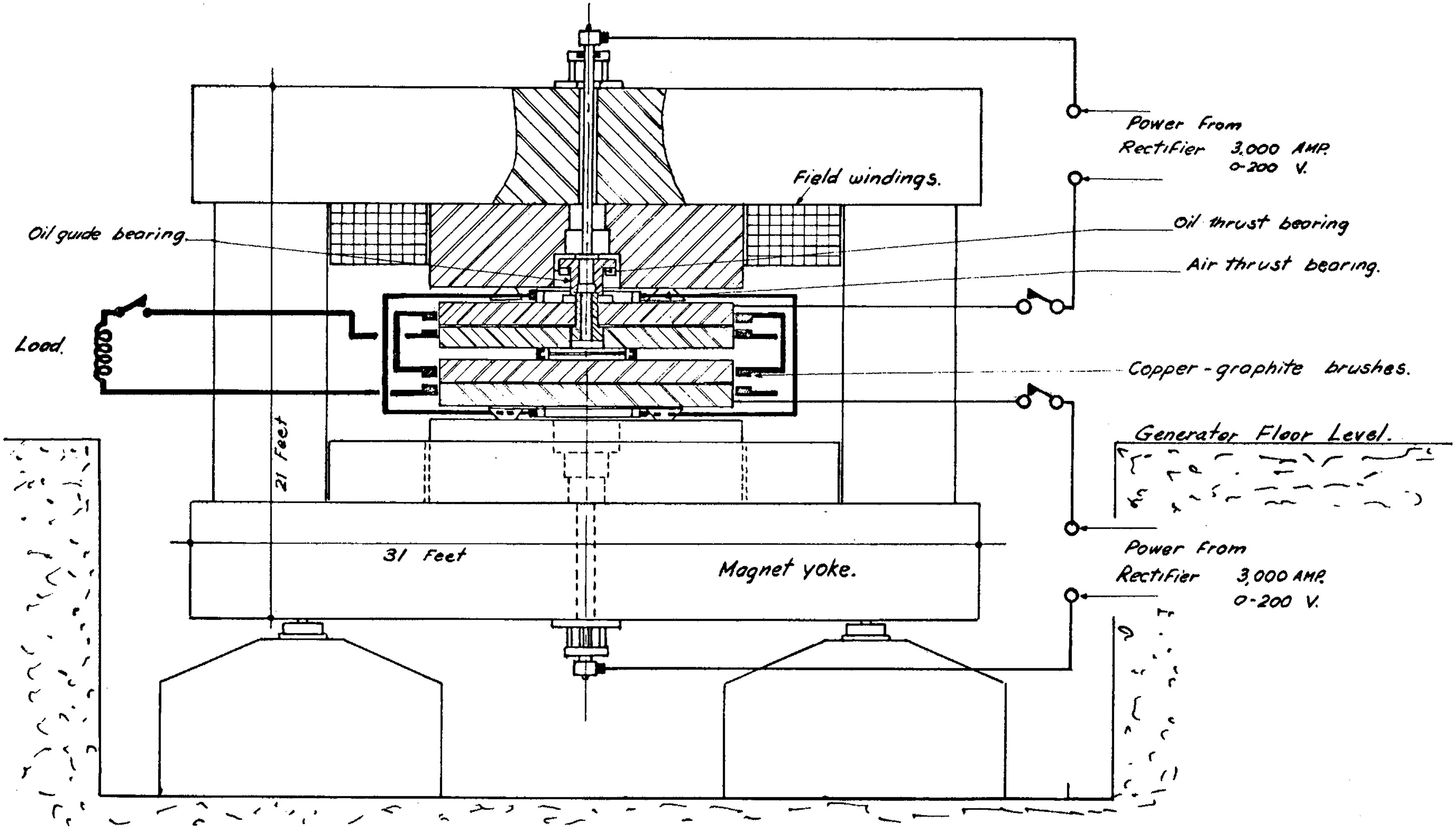
SUMMARY

In 1963, rotors of the homopolar generator were fitted to the Mark II shaft and bearings without any changes in the shape of the central bore. In 1965 it was possible to remachine the rotors to eliminate some of the stress concentrations produced by the original shape. The design proposed in this report reduced the stress at the worst concentration from 3.82 x 20,000 p.s.i. to 2.7 x 8,100p.s.i., and introduced a method of locking the screw in the shaft without the need for keyways, and the stress concentration they produce, in the rotors.



THE A.N.U. HOMOPOLAR GENERATOR is an energy storage device to store 560×10^6 Joules, taken from the Electricity Supply over a period of 15 minutes. The unique feature of the machine is that all this energy can be delivered to the load in one second.

The Maximum Voltage = 900 Volts.
" Current = $1{\cdot}6 \times 10^6$ Amps.
" Power = 1000 Mega Watts.

FIGURE 1.

INTRODUCTION

The main contents of this report were written in October, 1964 after the lower Mark II bearings had been tested to 920 r.p.m. with the rotors as they were originally machined for the Mark I bearings.[1, 2] The "Review of the Homopolar Generator Project"[3] dealt at some length with the questions raised by our own design studies concerned with the stress in the rotors and how the stress was affected by the shape of the rotor location registers. This report was written as part of the discussion of alternative action to that proposed in the Review.

The homopolar generator is shown in Figure 1, which is a simplified sectional view showing the rotor discs in the magnet which produces a uniform magnetic field of 17,000 gauss across the surface of the rotors. The discs are unstable in this field being subject to a tilting torque caused by the field concentrating at the part of a tilted rotor nearest to the pole. In order to decide on the modification needed, the following information was presented.

GENERAL BEARING DESIGN LOADS AND CONDITIONS

1. The tilting torque on a rotor if each deflects equally is about 3×10^9 lb. in. per radian of tilt.

2. A short circuit of 5×10^6 amp in one disc would load the thrust bearings with about 400 tons. The machine would be dismantled and checked after such a discharge. Some plastic deformation of supports could be allowed under such loads.

However, all components should run under an impulse load of 200 tons and not require the machine to be dismantled.

This load of 200 tons is specified as the preload on the screw which fixes the rotor to the shaft (item 15, Dwg. No. OB1070 and Figure 2).

3. The proposed design allows for a maximum normal speed of 800 r.p.m. if the rotors are not subject to brittle fracture at 20°C. This will stress the bore of the discs B and C to 20,300 p.s.i.

4. A total peak to peak alternating stress of 40,000 p.s.i. would be permissible at a stress concentration, on the basis of fatigue considerations. The design chosen and checked in the following calculations calls for only a range of 27,000 p.s.i. This is not likely to be enough to cause yield and so will not go to compression when the rotor is at rest.

5. It is recognised that the preload on the screw (item 15) introduces sufficient friction to prevent the shaft or screw turning with respect to the rotor, during a pulse. However this is not considered positive enough and radial keys have been provided to contend with the possibility of the preload relaxing or the decelerating forces being larger than expected.

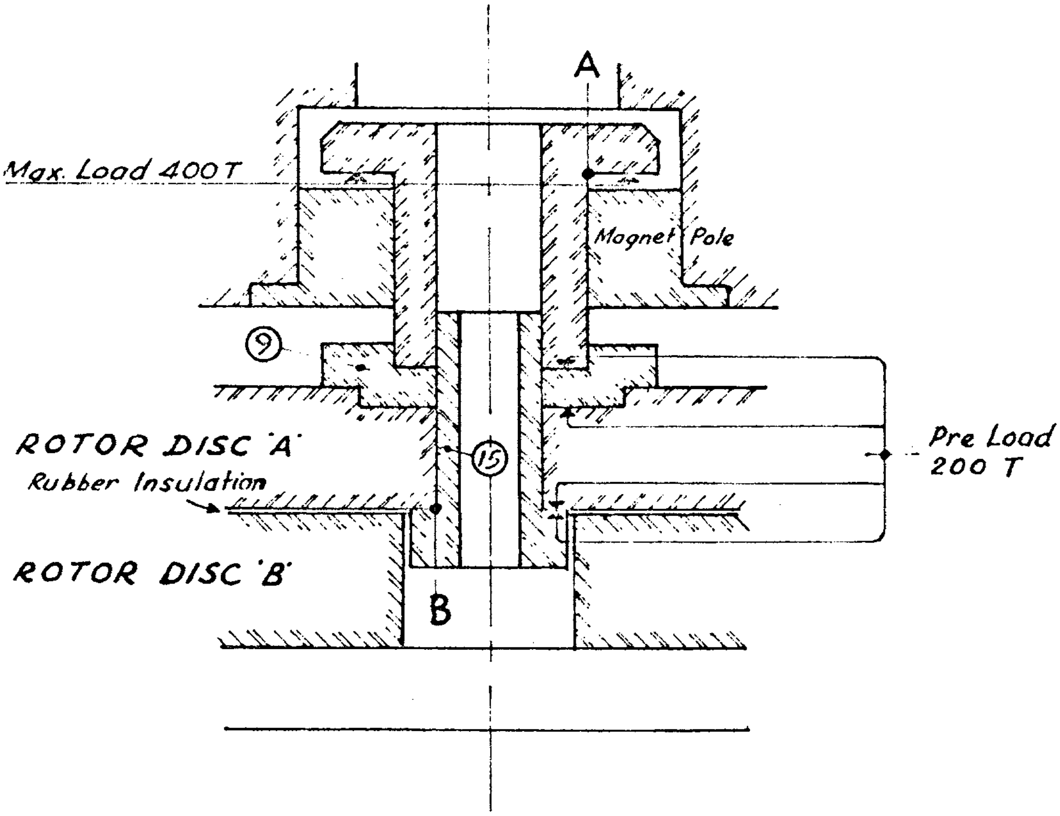


Figure 2. Showing the method of attaching the Mark II shaft to the rotor.

The following modifications can be made to the rotor and the centering ring (item 9) to keep the stresses due to centrifugal forces within the limits set out above. The calculations of the stresses and machining procedures are set out later.

RECOMMENDED CHANGES TO ROTOR SHAPE

1. Machine off the centering rib and keyway on discs A and D and round out the outer corner of the existing groove to a radius of $\frac{1}{2}$ in. as shown in Figure 3.

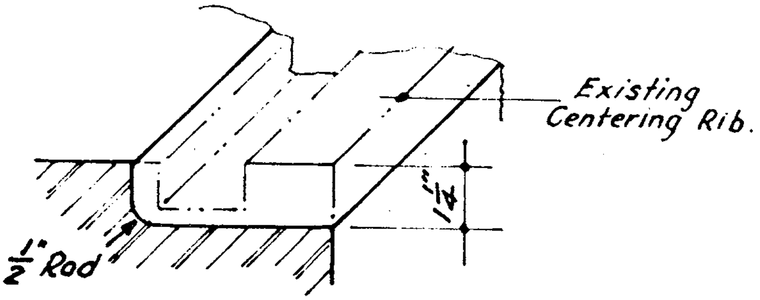


Figure 3. Showing section of disc A at the registers used to provide centering.

2. Machine a groove $1\frac{1}{2}$ in. wide by one in. deep on a radius of $10\frac{3}{4}$ in. to $12\frac{1}{4}$ in. on discs A and D (see Figure 4). Across the top of the $\frac{3}{4}$ in. wide ridge so formed,36 half holes can be drilled one in. in diameter spaced at $1\frac{3}{4}$ in. centres.

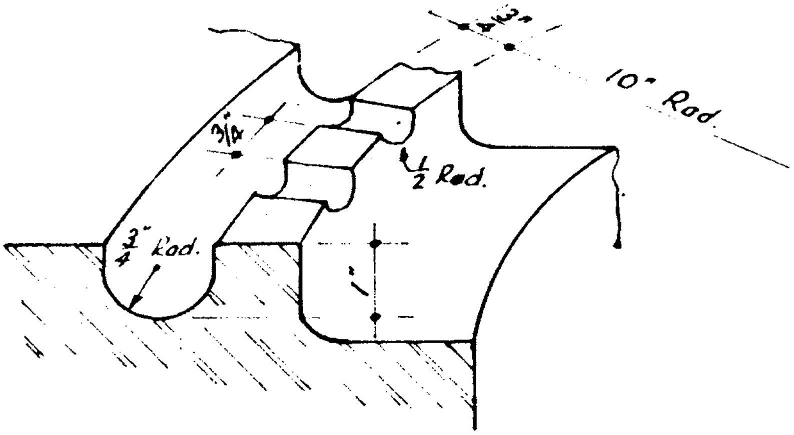


Figure 4. A section of disc A showing the proposed centering register on a radius of ten in. and the keyways to prevent the shaft rotating on the rotor.

3. Machine out the lifting grooves in the four discs to provide corner radii of 1/8 in.

4. The counter bored steps in discs B and C should be re-machined to give a corner radius of 1/16 in.

5. The sides of the shallow $31\frac{1}{4}$ in. diameter grooves in discs A and D can be sloped off and polished.

6. The 12 tapped holes in disc A on 37-3/8 in. pitch circle can be rounded out to a polished depression having a minimum radius of one in.

STRESS CALCULATIONS

The basic stresses have been calculated for uniform discs spun at 800 r.p.m. taking the density of the steel as 0.284 lb. in.$^{-3}$ and Parson's Ratio as $Y = 0.26$. This gives the following values.

	Tangential Stress		Radial Stress	
	Discs A & D	Discs B & C	Discs A & D	Discs B & C
Inner Radius	20,300 p.s.i.	20,300 p.s.i.	0	0
Radius 10 in.	13,100 p.s.i.		7,170 p.s.i.	
Radius $10\frac{3}{4}$ in.	12,600 p.s.i.			
Radius $12\frac{1}{4}$ in.	12,200 p.s.i.		8,100 p.s.i.	
Radius 18.5/8 in.	11,000 p.s.i.		8,730 p.s.i.	

As a result of the machining, the following stress concentrations occur.

a. The Filletted Corner Shown in Figure 4

This is at a radius of 10 in. in discs A and D where the radial stress is 7,200 p.s.i. The stress concentration can be calculated using the curves given in Petersen's book,[4] "Stress Concentration Design Factors," Figure 37 (p 66).

Here $D = 22$ in., $d = 20$ in., $D/d = 1.1$

$r = 0.5$ in., $r/d = 0.025$.

Therefore the stress concentration, $K_t = 2.45$

K_t would be reduced by the groove shown in Figure 4 and the reduction could be estimated as shown on page 63 of Petersen's book. The stress of 17,600 p.s.i. obtained using $K_t = 2.45$ is low so the reduction is of no importance.

b. Relief Groove Centred on Radius of $11\frac{1}{2}$ in. in Discs A and D

For these calculations the stress at $12\frac{1}{4}$ in. radius is taken.

Using curves and symbols shown in Figure 17 of Petersen,

$D = 22$ in., $d = 20$ in., $d/D = 0.909$,

$r = 0.75$, $r/D = 0.0341$

Therefore $K_t = 2.9$

This value of K_t will be higher than the actual value because of the relieving effect of the step near to the groove. Petersen does not give data for K_t at the corner of relief grooves but if Figure 21 is used with the values shown for Figure 5 (of these notes). there is a reduction from 3.07 to 2.78 for one notch compared to two. A similar reduction in our case would reduce K_t from 2.9 to 2.6. Thus it would be on the safe side if K_t were taken as $K_t = 2.7$. For a radial stress of 8,100 p.s.i. the concentrated value becomes 21,900 p.s.i.

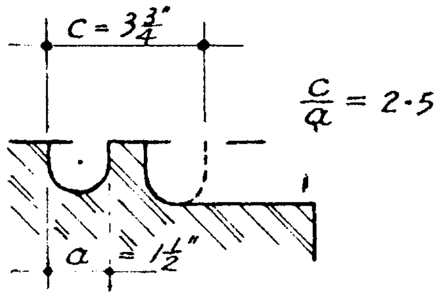


Figure 5. Showing the value ot the parameters used by Petersen to determine stress concentration factors.

c. The Half-Cylinder Keyways Shown in Figure 4

The highest tangential stress is on the 10 in. radius where it is 13,100 p.s.i. The stress concentration due to the keyways can be calculated from Figure 20 of Petersen's book. For our case

b = 1.75 in., a = 1 in., b/a = 1.75

Then from the broken curve for a row of notches $K_t = 2$.

The stress then becomes 26,200 p.s.i.

d. The 12 Rounded Out Holes at a Radius of 18.6 in., Disc A

These can be rounded out to a minimum radius of one in. to a depth of ½ in.

Using Petersen's Figure 16, 3/8 in. measured depth

D = 22 in., d = 21 in., D/d = 1.054

r = 1 in., r/d = 0.05

These values would give $K_t = 2.3$ for a plane section with cylindrical groove. Since the radial and tangential stresses are about equal at this radius and since the holes will be sections of a sphere in this stressed "membrane" a value of $K_t = 2$ would be reasonable. The concentrated stress would then be 22,000 p.s.i. (tangential).

Summary of the Peak Stresses for 800 r.p.m.

	K_t	Stress in p.s.i.	
Inner Bore of each disc	1	20,300	Tangential
Corner of centering bore	2.45	17,600	Radial
Relief groove	2.7	21,900	Radial
Half Cylinder keyways	2	26,000	Tangential
Holes in disc A	2	22,000	Tangential

METHODS OF PREVENTING THE SHAFT SLIPPING ON THE ROTOR

The design would be greatly simplified and the highest stress concentrations would be removed if it was considered reasonable to rely on friction alone to stop the shaft and slip-ring assembly from spinning on the rotor during a pulse.

The following points arise when considering this possibility.

1. The tightening load of 200 tons on the screw stretches it 0.004 in. At 800 r.p.m. the rotor will reduce in thickness by 0.002 in. and move radially under the head of the screw 0.005 in. This means the clamp load is reduced to half its initial value.

The tightening load in the thread of the shaft can expand the shaft as much as 0.0025 in. in diameter. The bearing clearance is 0.010 in. in diameter. Therefore it would not be wise to increase the tightening load.

2. The torque due to the inertia of the shaft and slip ring would be about six ton feet for the full designed pulse from the generator. During the next series of tests, using one disc of each rotor it would not exceed three ton feet unless the machine were short-circuited.

Torques of this magnitude can be produced by a radial component of the current flowing through the slip-ring from the brushes to the rotor. However, since the contact band between the slip-ring and the rotor extends one in. radially, centered on the brush radius, the radial flow of the current is not well defined. Thus torques adding to the inertia or opposing it may arise.

These torques appear low in relation to the friction available, but the uncertainty due to the elastic movements under the head of the screw makes one hesitate to rely on the friction. This is particularly so when one considers that the above design for keyways comes within the criteria for safe operation below the Nil-Ductility Temperature (N.D.T.), namely, that there are no large flaws and no localised stress reaches yield point.

MACHINING AND ASSEMBLY CONSIDERATIONS

1. Most of the machining required on the rotors is turning, and this can be done most readily on the boring mill. It is possible to bore the bottom rotor without turning it over.

2. The radial keyways can be indexed to ± 0.0002 in. on the boring mill and drilled then bored to size (only the 12 to be fitted with dowels need to be bored) at the same time. Unfortunately it will not be possible to drill and bore the spigot plate (item 9, Dwg. No. OB1070) while in place on the rotor. This is because item 9 is too deep to allow the drill head to reach down to the keyways. Instead, a plate one in. thick could be clamped to the top of the rotor to provide a full hole for the drills. The spigot plate (item 9) can be indexed and bored on the jig-borer, before the centering spigot is machined on it. The location of the dowel holes in the spigot plate will be within ± 0.0002 in.

3. Because of the tolerances on the indexing, some of the 12 dowels may have an interference of 0.0004 in. on one side and a corresponding clearance on the other. This will lead to some yielding of the keyways and the dowels. The residual stress in the keyways will add to the dynamic loads and would be bad in a material subject to brittle fracture. If the dowels are made of much softer material such as copper or bakelite the deformation would take place in them, so producing less stress in the keyways.

Because of the uncertainties in allowing for the stresses due to misalignment of keys, the use of friction alone is reconsidered. The greatest danger in this case is that the repeated radial movements of the rotor under the head of the screw may work the thread loose. This is more likely if the inertia torque on the shaft can appear unchecked at the rotor surface where the radial movements occur. It is therefore important to consider methods of locking the screw thread, and if a reliable lock can be introduced it would not be necessary to incorporate any keys.

THE DESIGN OF A LOCK TO SECURE THE SCREW IN THE SHAFT

A locking device which should be suitable is shown in Figure 6. This will be tested to determine its effectiveness as soon as test pieces are made. Design calculations are given below. (Tests have since been made and the lock proved to be more effective than these calculations indicate.)

If no keys are employed the relief groove will still be provided and the spigot plate will be machined to fit as shown in Figure 7.

The lock should provide a resistance of the same order of magnitude as that due to friction under the head of the screw, and get greater or at least, not diminish if the preload on the screw diminishes. The lock shown in Figure 6 meets this requirement. The left hand thread on the inside of the screw causes the locking cone to bed more tightly if the screw starts to slacken. The eight $\frac{3}{4}$ in. diameter screws are to be tightened initially to 10,000 lb. each and this allows a margin of two for increase in the load before the screws reach their normal working load.

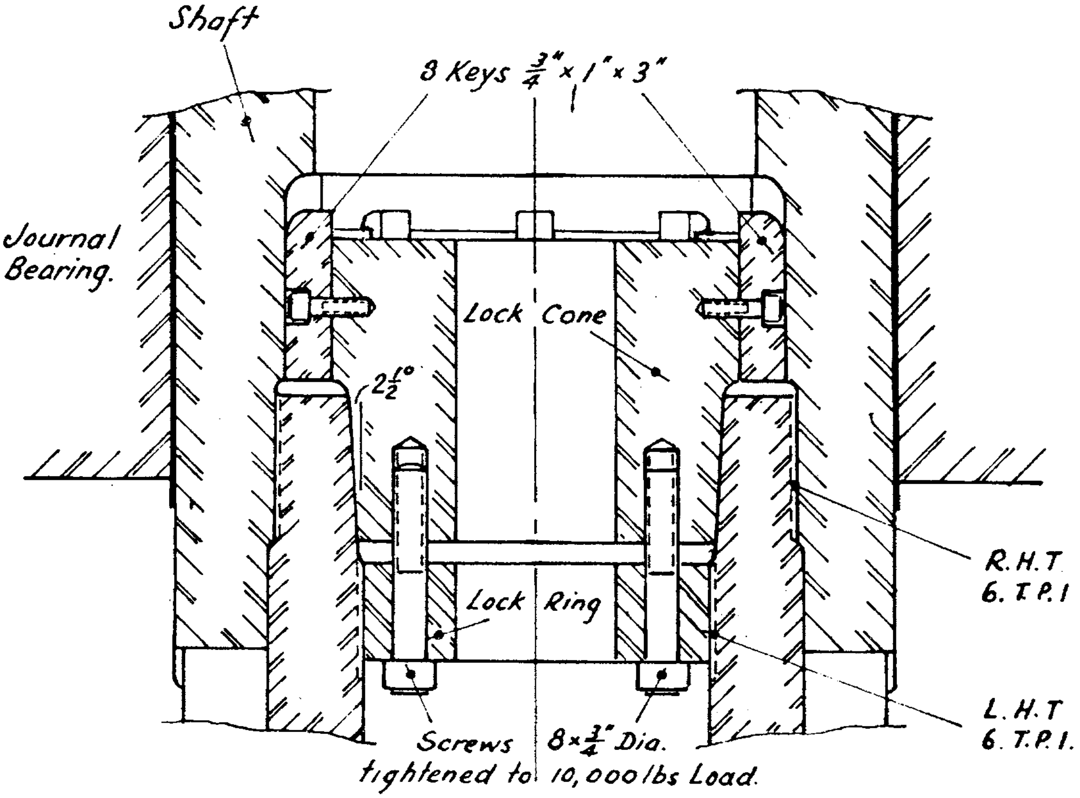


Figure 6.

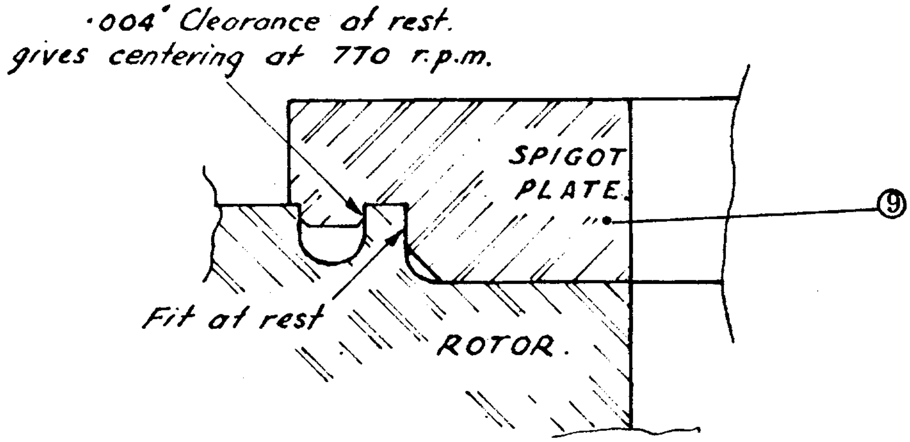
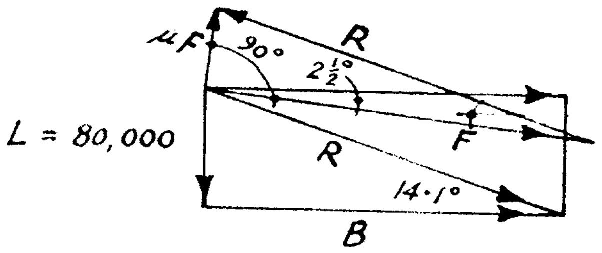


Figure 7. Showing centering locations at rest and near top speed.

When the initial tightening has been applied the forces will be as shown in Figure 8.



$$\tan^{-1}\mu = \tan^{-1} 0.2 = 11.6^\circ$$

Figure 8.

If F = normal force on interface of the cone

$\vec{R} = \vec{F} + \mu\vec{F}$

$\vec{B}$ = Bursting force on end of screw.

This should not exceed 2 x the collapsing force due to the preload of 200 tons on the screw thread, i.e. it should not exceed 220 tons.

It is partly on the basis of this bursting pressure that the tightening load on the eight $\frac{3}{4}$ in. diameter screws has been chosen as 10,000 lb. per screw.

The cone face will be degreased and the coefficient of friction μ is assumed to be 0.2. A higher value may occur giving more locking friction and less bursting force than that allowed for here.

$$R = \frac{80,000}{\sin 14.1^\circ} = 33 \times 10^4 \text{ lb.}$$

$$B = 33 \times 10^4 \sin 75.9^\circ = 32 \times 10^4 \text{ lb. (143 tons)}$$

$$F = 33 \times 10^4 \sin 78.4^\circ = 32.4 \times 10^4 \text{ lb.}$$

$$\mu F = 65,000 \text{ lb.} = 30 \text{ tons on 4 in. radius}$$

As stated above the inertia forces on the shaft during a full design pulse would be six tons at one foot, that is 18 tons on 4 in. radius.

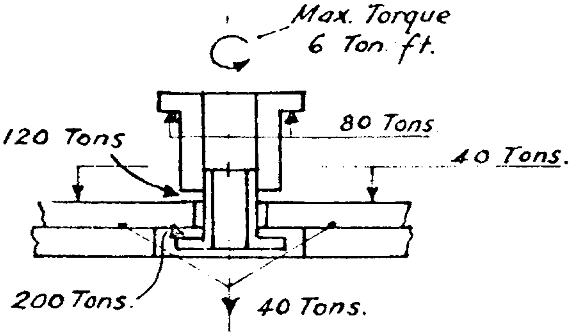


Figure 9. Showing the distribution of the loads on the screw.

The tightening load on the head of the screw will be 200 tons. The weight of the rotor plus the down load from the air bearing reduces the load between the end of the shaft and the rotor to 120 tons. Since the screw thread must be lubricated with "Molyslip" to prevent seizure, the shaft would turn on the thread rather than the head of the screw turning, if the inertia forces were great enough to cause a slip. If it were not for the locking cone this could occur if the preload were lost due to slow creeping of the screw due to radial movements as outlined earlier. The left hand thread on the locking ring (greased with Molyslip) will cause the locking cone to tighten and stop any creeping out of the screw. This would maintain the tightening load on the screw at least to a value where friction would prevent the shaft turning on the rotor without keys being fitted.

It is therefore proposed that the generator be reassembled without keys and with the locking assembly.

Test of the locking cones showed first movement at 20 ton feet followed by a rapid rise of resistance to movement, which rose to 30 ton feet. The cone could then be separated with the screws provided for this purpose. These tests were interpreted as adequate proof that this method of locking the screw was satisfactory.

The generator has been in operation for two years with the locking arrangement and no keyways, as recommended in this report.

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6. INALL, E. K.: "A Review of the Specifications and Design of the Mark II Oil Lubricated Thrust and Guide Bearings of the Canberra Homopolar Generator," The Australian National University, Department of Engineering Physics, Publication EP-RR 6, November, 1964.

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