

A REVIEW OF THE SPECIFICATIONS
AND DESIGN OF THE MARK II OIL
LUBRICATED THRUST AND CENTERING
BEARINGS OF THE CANBERRA
HOMOPOLAR GENERATOR

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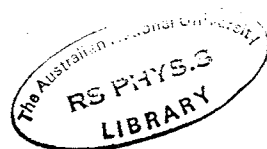
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A REVIEW OF THE SPECIFICATIONS AND DESIGN
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by

E. K. INALL

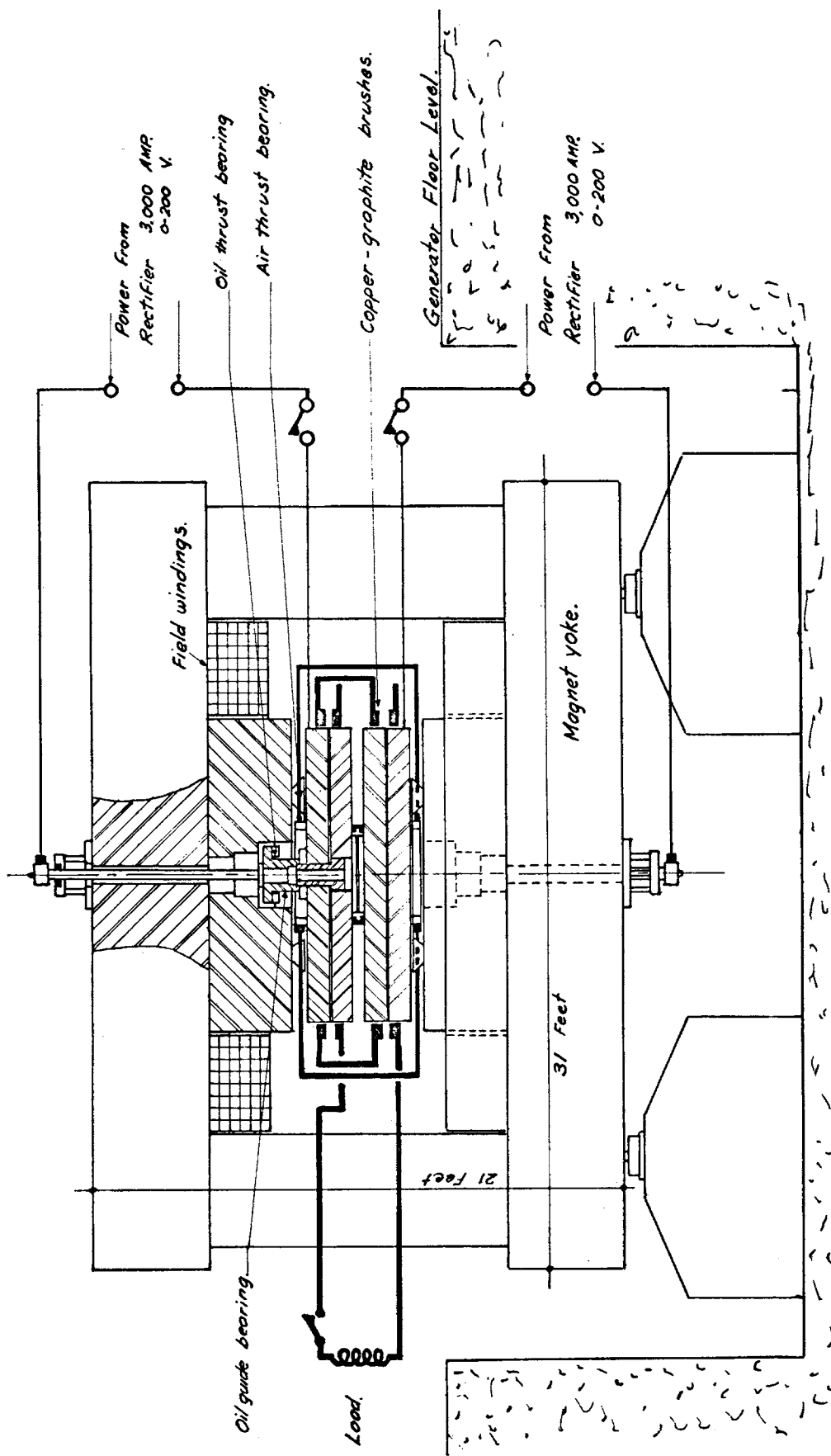
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Canberra, A.C.T. Australia



The Maximum Voltage = 900 Volts.
 " Current = 1.6×10^6 Amps.
 " Power = 1000 Mega Watts.

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.

FIGURE 1.

Introduction

This report on the bearings of the homopolar generator^{1, 2} was written after the Mark II lower bearings had been tested at 920 r.p.m. in 1963. The "Review of the Homopolar Generator Project at the Australian National University"³ and a report on the design of the rotors⁴ had been discussed and a request was made for details of the most highly stressed parts of the bearings. This report was therefore prepared. A simplified sectional drawing of the generator is shown in Figure 1.

The Requirements to be Met by the Bearings

The load on the top thrust bearing and shaft is higher than on the bottom and the design was therefore based on the top loads. The working load on the top thrust (item 4 of Figure 2 and Dwg. No. OB1070) is 80 tons, made up of 40 tons weight of the rotor and 40 tons down thrust from the top air pad. The components however have been designed to carry 400 tons made up of the above 80 tons plus 320 tons due to a possible short circuit current of 5×10^6 amps in the top disc of the top rotor. However if the stress in some small region of concentration exceeded the yield stress during such a short circuit, it would not be bad and it would be unrealistic to design all regions to prevent this. This load could not exist for more than 0.2 seconds.

The following stress regions are examined:

- a. The head of the screw (item 15). Tightening load = 200 tons, maximum load = 400 tons.
- b. The thread of the screw with loads as for (a)
- c. The shaft (item 17) at the top of the keyways for the locking cone and at the thread.
- d. The shaft thread in the thrust nut (item 4) for the load of 400 tons.
- e. The 24 screws which hold the bearing body (item 6) into the magnet pole.

Preload on the Screw

The screw is preloaded to 200 tons by tightening the screw manually, heating the section between the thread and head by an amount required to give a known thermal expansion, and then retightening manually. When the screw is cool, the stress is checked. The stressed length of the screw is 18 in., the area of steel is 56.5 sq. in. (see Figure 3).

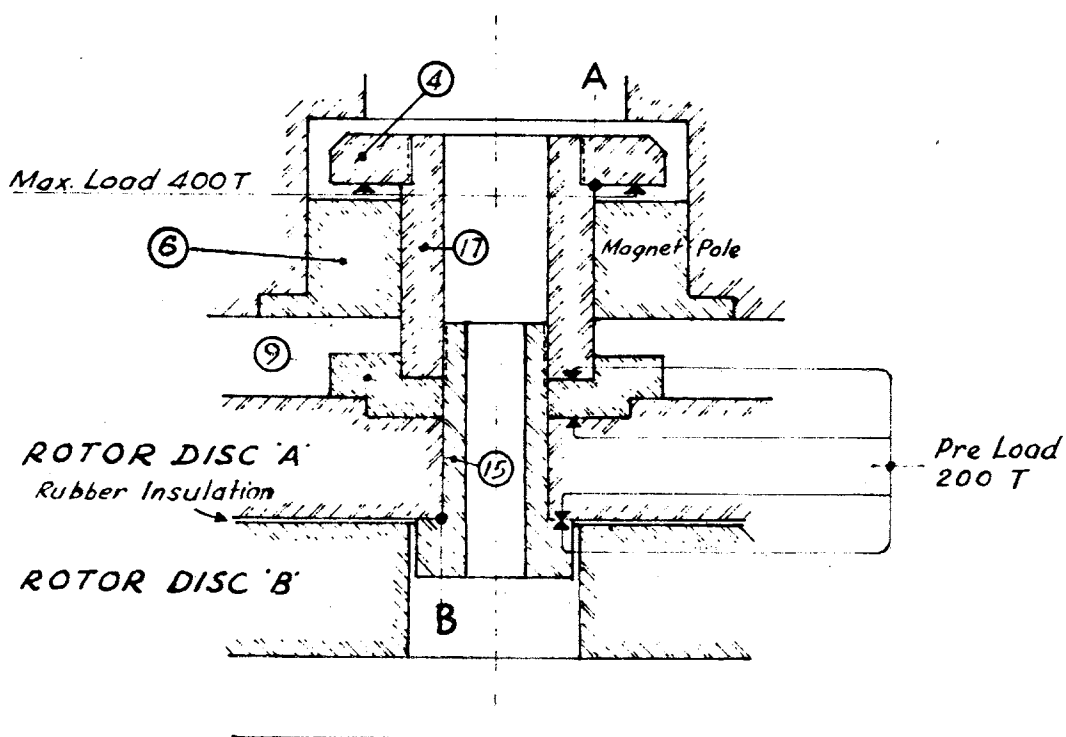


Figure 2. Simplified Drawing of the Mark II Shaft Fixed to the Top Rotor, Showing Maximum impulse load.

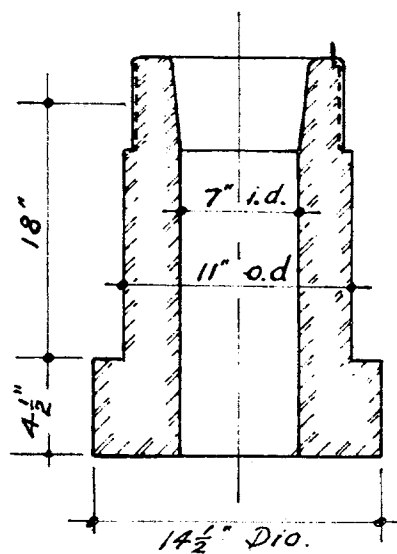


Figure 3. Details of the Mark II Screw Used to Fix the Rotor to the Shaft.

Therefore the elongation under a 200 tonload is 0.0047 in.

(Material of screw, Com Steel K 1030)

When the screw was stressed for the first time after reassembly a thermal expansion of 0.010 in. was produced between the two tightenings of the screw. The elastic component of the loading was then checked by allowing the temperature to equalise and then reheating the screw to determine the expansion required to free it. After the first tightening only 0.002 in. of elastic deformation existed. The tightening was repeated a number of times. After the second, the elastic component was 0.002 in.; after the third 0.004 in.; and after the fourth 0.0045 in. By that stage, 0.016 in. of non-elastic closure had occurred. Since some elastic deformation would occur to the screw head, the rotor edge, the spigot plate and the shaft, the final load was somewhat less than 200 tons.

At 800 r.p.m. the tangential stress around the bore of the rotor would be 20,000 p.s.i.

Since Poisson's Ratio equals 0.26 and Young's Modulus 30×10^6 p.s.i. the 11 in. of rotor thickness will shrink by 0.002 in. Therefore the load will relax by about 80 tons at full speed.

Calculations of Stresses on the Rotor Support

1. The stress in the screw under the tightening load of 200 tons is 3.55 tons per sq. in.

The stress concentration at the head is given in Petersen's book⁵ on page 121 for which

$$d = 4 \text{ in.}, \quad D = 7\frac{1}{2} \text{ in.}, \quad m = 4\frac{1}{2} \text{ in.}, \quad m/d = 1.1,$$

$$\sigma = 3.55 \text{ tons/sq. in.}, \text{ from the curve}$$

$$\frac{\sigma_{\text{max.}}}{\sigma} = 5.2$$

Thus $\sigma_{\text{max.}} = 18.4 \text{ ton/sq. in.}$ so that some yielding in the corner will occur under the tightening load. There will then be a fluctuation of 7 tons amplitude with each pulse. This will not cause fatigue.

If yielding resulted in the 400 ton load being taken uniformly around the head of the screw in shear, the shear stress would be 2.56 tons/sq. in. Such yielding and distribution of the load occurs in all fully loaded bolts.

The screw was shown to be free of cracks or flaws by magnetic testing before it was accepted. It was normalised. The specification for this steel states that the Nil-Ductility Transition (N. D. T.) temperature is below 10°C , so there is no danger of brittle fracture of the screw.

2. The stress concentration for the thread can be estimated on the basis of the discussion in Petersen⁵ on page 112. The thread on the screw has about the same depth in relation to twice the wall thickness as a Whitworth thread has in relation to its diameter. The stress concentration can therefore be taken as about 4. This would give a peak stress of 14 tons/sq. in. under the tightening load.

The tightening load is transmitted to the top surface of the rotor via the end of the shaft and the spigot plate (Figure 2, item 9). If the 400 ton pulse load occurred, the preload would be relieved and the stress would then be in the same direction in the threads on the screw and in the shaft. This case is shown in Figure 97(c) of Petersen⁵ and the stress concentration is about half that of the normal nut and bolt. Thus the increase in load would be compensated by the reduction in concentration if the thread engagement were 4 in., that is twice the thickness, being equivalent to an engagement of one diameter of a Whitworth screw. The actual engagement is only 3 in. so that some yielding may occur.

If it is assumed that only half the root area of the thread is effective in taking the load over the full length of engagement, the shear stress due to the 400 tons on this area would be 7.7 tons/sq. in. which would be considered safe for this service.

(The manufacturers quoted that three samples of K 1030 gave 70 ft. lbs at 20°C on Izod tests and 36 tons/sq. in. tensile on 1 in. diameter specimens. Our forgings may give a little less because of size (about 10%)).

3. In order to relieve the stress concentration at the end of the key-ways provided for the locking cone described earlier,⁴ and to enable the key-ways to be cut, a groove of $\frac{1}{2}$ in. radius and $\frac{1}{2}$ in. deep will be turned on the inside of the shaft. The basic stress at this point due to a load of 400 tons is 3.8 tons/sq. in.

The stress concentration is obtained from Figure 16 of Petersen.⁵ For this case

$$D = 5\frac{1}{2} \text{ in.}, \quad d = 4\frac{1}{2} \text{ in.}, \quad r = \frac{1}{2} \text{ in.},$$

$$D/d = 1.22, \quad r/d = 0.111$$

$$\text{So that } K_t = 2.37.$$

The peak stress then becomes 9 tons/sq. in.

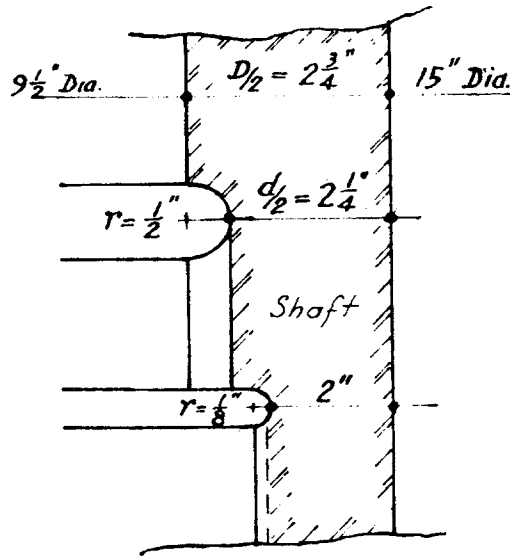


Figure 4. Details of the Stress Concentration Points in the Shaft.

4. At the end of the thread the basic stress is 4.9 tons/sq. in.

Here $D = 5\frac{1}{2}$ in., $d = 4$ in., $r = 1/8$ in.,

Then from Petersen's Figure 57,⁵

$$r/d = 0.031, \quad D/d = 1.37, \quad \text{and} \quad K_t = 3.1$$

The peak stress then becomes 15 tons/sq. in. Because of this stress the Nil-Ductility Transition temperature of the shaft material will be sought. (It has since been reported to be less than 10°C).

5. The thrust nut (Figure 2, item 4) is screwed onto the shaft with epoxy resin in the thread and two round pins are cemented into the thread parallel to the axis of the shaft. The stress concentration is determined by considering the nut and shaft as a parallel cylinder with a groove as deep as the relief groove at the end of the thread. This assumes that the araldite is not rigid enough in tension to introduce a radial tension in the first threads.

In this case $D = 5\frac{1}{2}$ in., $d = 3\frac{1}{4}$ in., $r = 1/8$ in.,

Then in Petersen's Figure 17⁵ $d/D = 0.59$, $r/D = 0.0227$,

so that $K_t = 4.2$.

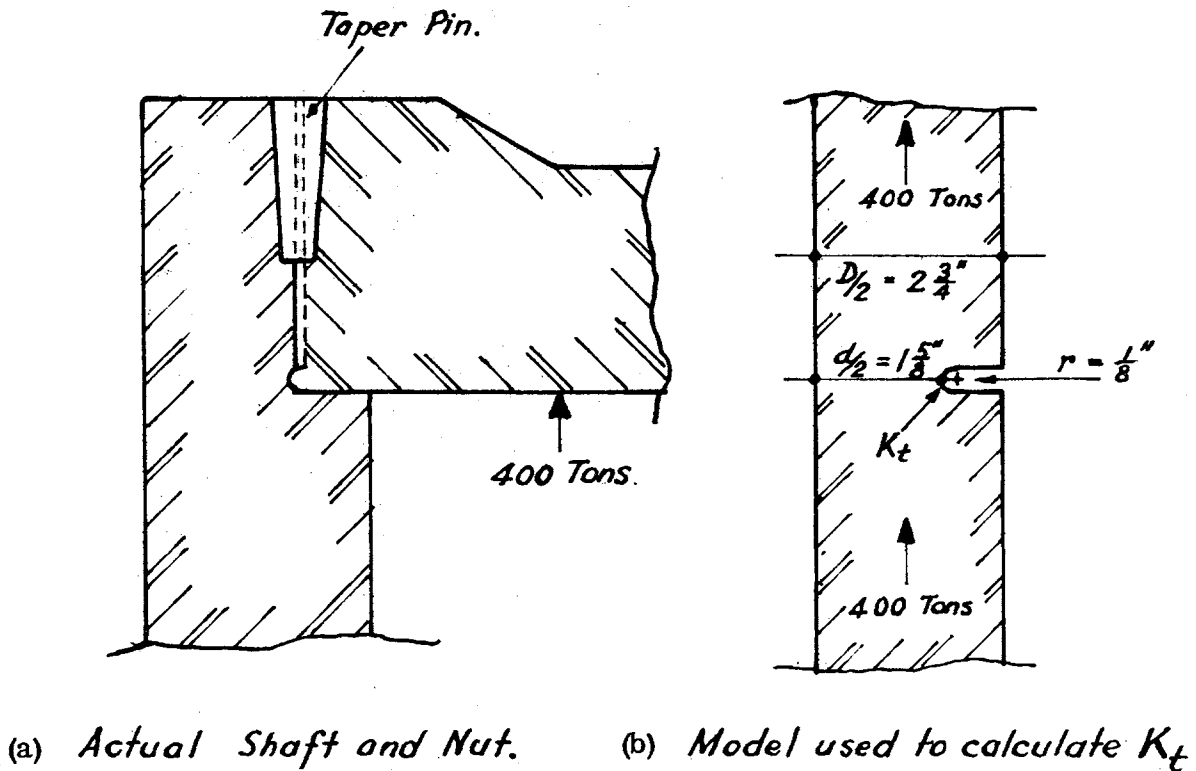


Figure 5. Details of the Stress Concentration due to the Attachment of the Thrust Shoulder to the Shaft.

The basic stress is as for 3(a), equal to 3.8 tons/sq. in.

Therefore the load of 400 tons produces a peak load of 16 tons/sq. in.

6. There are 24 one-in. diameter B. S. F. screws holding the bearings into the poles. The maximum load per bolt is therefore 16.7 tons. Ajax mild steel bolts are rated at 17 tons maximum load. High strength alloy steel bolts have been used in this position.

The peak stresses show that fatigue will not be important. However, since some exceed the yield point under normal loading, brittle fracture could cause a failure if the material is below its Nil-Ductility Transition temperature. Because of this, the stress concentration factors quoted have not been reduced by the notch sensitivity allowance.

The head of the screw could crack off due to the tightening if the material were brittle at any temperature reached on a cold day. This would be much worse under short-circuit conditions. Similarly the shaft could crack under short-circuit

conditions, if it were brittle. The alloy from which these parts were made is not brittle at room temperature.

Table of Calculated Stresses in Shaft and Screw

(Tons per square inch)

Location	Working Load		5 x 10 ⁶ amp Fault Load	
	Basic Stress	Stress at Concentration	Basic Stress	Stress at Concentration
Head of Screw	3.55	18.4*	7.1	Beyond Yield*
Screw Thread	3.55	14.0	7.1	15.0
Relief Groove in Shaft	0.6	1.4	3.8	9.0
Screw Thread in Shaft	2.45	7.5	4.9	15.0
Thrust Thread on Shaft	0.6	2.5	3.8	16.0

*Fluctuation after yielding at 15 tons per square inch = 15 to 8 tons per square inch

All the basic stresses for the working load and the short-circuit load are low so that the components would suffer only minor distortion if the 5 x 10⁶ amp short-circuit occurred.

Oil Supply to the Thrust Bearing

The thrust bearing is supplied with oil to four recessed areas in the face of the thrust piston. The total flow is 15 gallons per minute (g.p.m.) from pumps which deliver a substantially constant flow at pressures up to 5000 p.s.i.

The total thrust area is 346 sq.in., so that the mean pressure required to carry the working load of 80 tons is 518 p.s.i. The corresponding supply pressure is about 600 p.s.i. and would reach 3,300 p.s.i. under the short-circuit load.

The compressibility of the oil in the supply lines is such that the oil to the bearing would stop momentarily until the pressure built up to the value required to carry a short circuit load. This may allow the steel to bear onto the Ferobestos

face of the bearing for a short time. The anti-rotation keys in the thrust piston were designed to withstand a tangential force of 40 tons assuming a coefficient of friction of 0.1 for steel on oiled Ferobestos.

At the normal running temperature (55°C) the viscosity of the oil is 25 centi stokes, and the nominal clearance of the bearing is 0.003 in. under a load of 90 tons.

Details of the Centering Bearing

The shaft of the centering bearing is 15 in. in diameter and is covered with a layer of Ferobestos $1/16$ in. thick. This runs in the steel body of the bearing (above section) with a radial clearance of 0.005 in. An oil supply of 10 g. p. m. is split by three throttle tubes and enters the bearing at three equally spaced grooves. The bearing is $10\frac{3}{4}$ in. long, made up of two lands $4\text{--}3/8$ in. wide, separated by the oil recess 2 in. wide. The oil is supplied through flow control valves from the same pumps as the thrust bearings. At the normal running temperature the supply pressure into the oil supply throttles is 1,100 p. s. i. There is a pressure drop of 700 p. s. i. in the throttles, so that the pressure in the oil recess is 400 p. s. i.

The radial restoring force in the bearing has been measured while the shaft was at rest. The value obtained was 2.5 tons per 0.001 in. off centre. At 800 r. p. m. the hydrodynamic centering force has been calculated to be only about 3 tons per 0.001 in. off centre.

At 900 r. p. m. the centrifugal load on the bearing is about one ton for every 0.001 in. which the centre of gravity of the rotor is off the axis of rotation. During past tests, the rotor has been 0.0035 in. off the axis while at rest and this has increased to 0.0055 in. at 800 r. p. m., and then decreased to 0.0035 in. at 900 r. p. m. It is assumed that the return to the rest value at 900 r. p. m. was due to the second centering land on the rotor coming into play as intended. When the bottom rotor was mounted last, it was 0.0013 in. off centre and it is expected that in future the rotors can be turned and belt ground after mounting. This should result in the eccentricity at rest being as low as 0.001 in. (These accuracies have been achieved during subsequent operation.)

Calculations of Stresses in the Rotors and Bearings

1. Stress Due to the Working Load of 80 tons on the Top Rotor

This rotor droops down on the outer edge, giving a hoop tension on the top surface.

The stress in the rotor bore at A, Figure 6 due to it bowing under the weight of both discs can be estimated as follows by using a formula given in Roark's book, "Formulas for Stress and Strain." ⁶

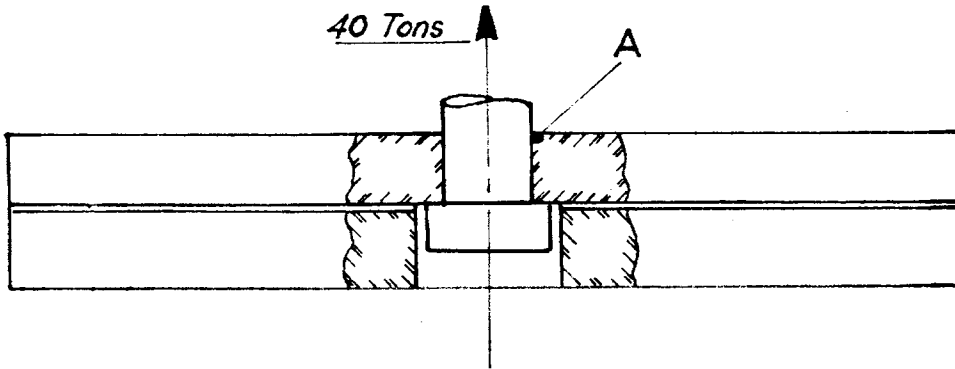


Figure 6. Showing the Region A of Maximum Stress Due to the Rotor Bowing Down at the Outer Edge.

Roark,⁶ page 199, example No. 16 applies with:

$$m = 1/0.26 = 3.8, \quad w = 6.02 \text{ lb/sq. in.},$$

$$b = 5.5, \quad a = 69.5, \quad t = 11 \text{ in.}$$

$$S_t = \frac{3 \times 6.02}{121 \times 4 \times 3.8 \times (69.5^2 - 5.5^2)} \left[4 \times 69.5^4 \times 4.8 \times \log \frac{69.5}{5.5} + \right. \\ \left. 4 \times 69.5^2 \times 5.5^2 + 5.5^4 \times 2.8 - 69.5^4 \times 6.8 \right] \\ = 2,000 \text{ p. s. i.}$$

This is an over estimate since it does not allow for the stiffness of the lower disc. If the lower disc were all suspended at the centre it would take up the same shape as the top and the stress in each would be:

$$S_t = 1,000 \text{ p. s. i.}$$

However some load will be taken all the way out to the circumference, the top disc will bow more and the lower one less. The stress will therefore be between 1000 and 2000 p. s. i.

The 40 tons from the air bearing will bow the pair as if there were 20 tons per disc. Therefore we can calculate the extra stress in the top disc using case No. 15,

page 198 of Roark⁶ with

$$d = b = 5.5, \quad a = 69.5, \quad W = 44,800 \text{ lb},$$

$$t = 11 \text{ in.}, \quad c = 37 \text{ in.}$$

Then

$$S_t = \frac{3 \times 44.8 \times 10^3}{2 \bar{A} \times 3.8 \times 121} \left[\frac{2 \times 69.5^2 \times 4.8}{75 \times 64} \log \frac{37}{5.5} + 2.8 \times \frac{42.5 \times 31.5}{75 \times 64} \right]$$

$$= 900 \text{ p.s.i.}$$

Thus on the top disc of the top rotor there will be a hoop stress of about 2500 p.s.i. total (tension on top face) due to the weight of both discs plus the preload on the air bearing.

2. Stress Due to 400 Tons Down on Top Disc

The rubber bond between the discs is rigid enough to hold the center of the lower disc with a force of 200 tons if the upper disc bowed downwards around its rim. However, there is little resistance to shear deflection between the discs. Thus the 400 ton load due to a short circuit on the upper disc would stress them both like two separate discs carrying 200 tons each. Assume this is centered at a slightly larger radius than the air pad. This would result in an extra 10-12000 p.s.i. at the bore which, added to the stress due to centrifugal forces, would certainly produce some yield at the corner. This could lead to trouble if the rotor were brittle.

3. The Effect of a Radial Load on the Centering Bearings

Assume a side load of 50 tons could occur during a fault in the generator. The significant dimensions are shown in Figure 7. Whenever the preloading is more than 80 tons, the deflection would be as for a cylindrical cantilever.

$$\text{Moment of Inertia} \quad I = \frac{\bar{A}}{4} (R^4 - r^4)$$

$$= \frac{\bar{A}}{4} (7.5^4 - 5.5^4) = 1,790 \text{ in.}^4$$

$$\text{Roark,}^6 \text{ page 100 No. 4 with } a = 4 \text{ in.}, \quad b = 1 = 14 \text{ in.}$$

Then the angle θ of the shaft at the top of the bearing will be

$$\theta = \frac{50 \times 2.24 \times 10^3}{6 \times 30 \times 10^6 \times 1.79 \times 10^3} (16 + 56 + 196)$$

$$= 0.06 \times 10^{-3} \text{ rad}$$

or 0.0006 in. in 10 in.

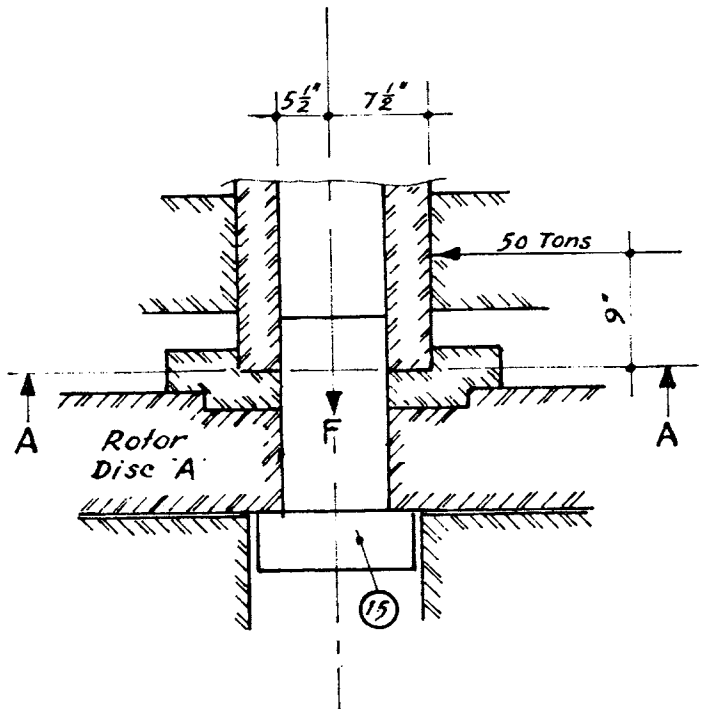


Figure 7. Diagram for Calculating Maximum Side Thrust Adequately Restrained by Preload.

If there is no preload on the screw the inclination could be as much as 0.004 in. in the length of the bearing. This would not mean that the bearing would fail, but it does emphasise the need to maintain a preload on the screw.

4. Dishing of the Thrust Flange Under a Working Load of 100 Tons

It is clear from Figure 8 that the thrust flange will roll under the influence of the tension in the shaft and the reaction from the thrust bearing.

This can be calculated using the expression in Roark⁶, page 199, No. 16.

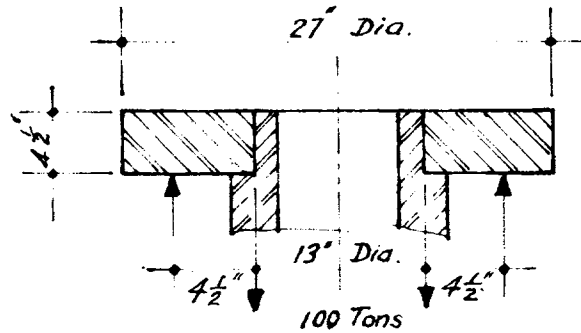


Figure 8. Showing Simplified Details of the Thrust Flange, item 4 of Figure 2

Assuming a uniform load, $w = \frac{100 \times 2.24 \times 10^3}{345}$

$$Y = \left[\frac{3 \times 100 \times 2.2 \times 10^3 \times 2.5}{345 \times 16 \times 3 \times 10^7 \times 3.5^2 \times 4.5^3} \right]$$

$$\left[13.5^4 \times 27.5 + 6.5^4 \times 18.5 - 13.5^2 \times 6.5^2 \times 46 \right.$$

$$\left. - 4 \times 13.5^2 \times 6.5^2 \times 11.5 \times \frac{4.5}{2.5} \log \frac{13.5}{6.5} \right]$$

$$+ \frac{16 \times 13.5^4 \times 6.5^2 \times 4.5^2}{(13.5^2 - 6.5^2) \times 2.5} \log^2 \frac{13.5}{6.5}$$

$$= 7.7 \times 10^{-3} \text{ in.}$$

This would be about 0.007 in. rise over the outer 5 in. of the radius. That is, the width of the bearing face.

This is an angle of roll of

$$\frac{7}{5} \times 10^{-3} \text{ rad} = 1.4 \times 10^{-3} \text{ rad} = \theta$$

The moment required to twist the piston to follow the flange is found from

$$\theta = \frac{MR^2}{EI} \quad (\text{Roark}^6, \text{ page 231})$$

$$\text{where } I_{yy} = \frac{5 \times 2.5^3}{12} = 6.55 \text{ in.}^4 \quad (\text{see Figure 9})$$

$$\therefore 1.4 \times 10^{-3} = \frac{M \times 11^2}{30 \times 10^6 \times 6.55}$$

$$M = 2.28 \times 10^3 \text{ lb.in./in. of circumference.}$$

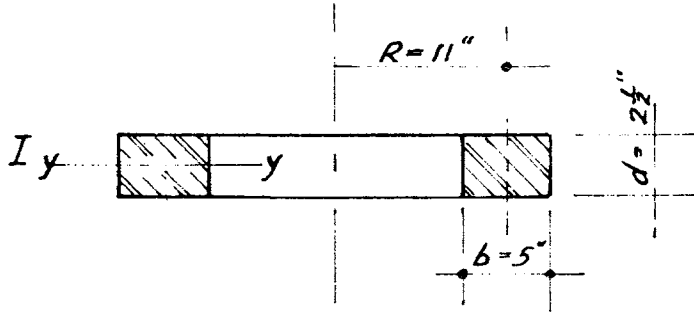


Figure 9. Details of the Thrust Bearing Piston.

Such a large torque would not be produced by the greatest pressure gradient possible in the oil film and the load would be taken on a smaller radius since at the inner edge the oil film would be thinner and may be nearly zero. The tilt on the flange will then be less than the 0.007 in. calculated and the roll of the piston will not need to be so great. This whole effect seems to be marginal at 100 tons and would certainly cause contact with boundary lubrication under a 400 ton load. This emphasises the advantage of new longer shafts made from forgings with the head integral with the journal. In spite of the tilts indicated by these calculations the bearings have operated well at loads of 100 tons.

The Preload Required on the Air Bearing to Limit the Tilt of the Rotor

The magnetic field causes the rotors to tilt with a torque of 3×10^9 lb.in./radian of tilt. If the preload is unaffected by the tilt, that is if the oil thrust does not move in any way under the tilt load, the rotor will be just restrained by the preload being developed on one side of the air bearing (see Figure 10).

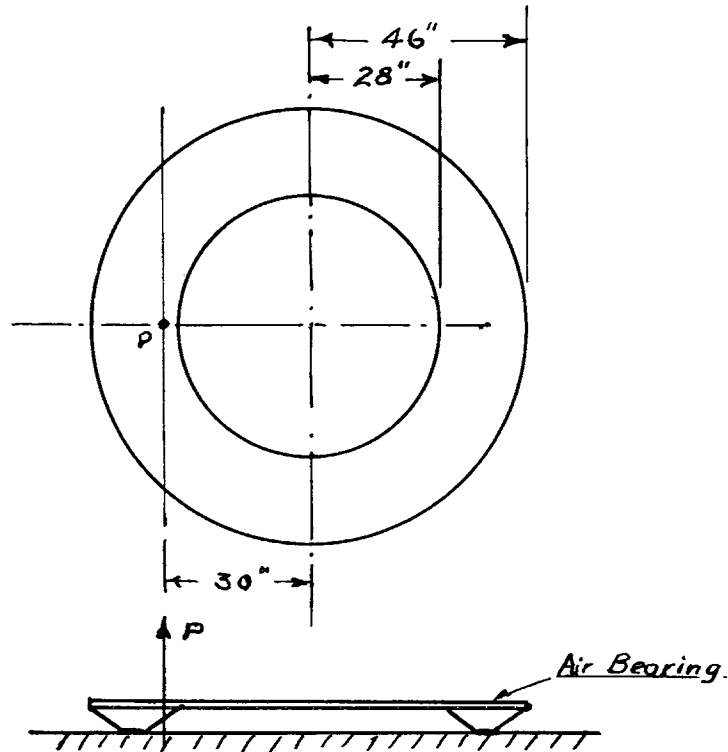


Figure 10. Showing the Dimensions of the Air Bearing and the Location of the Reaction^{which} Controls the tilt of the Rotor.

Taking a preload of P pounds and assuming that a tilt occurs causing this to be generated on an area to one side of the air bearing with an effective centre 30 in. from the axis, the restoring torque will be T where,

$$T = 30 P \text{ pounds inches}$$

If the tilt is such as to take all the air bearing clearance of 0.010 in. on the edge at a radius of 46 in., the tilt would be limited to

$$\frac{10 \times 10^3}{46} = 2.18 \times 10^{-4} \text{ rad.}$$

The magnetic tilting torque at this angle is $3 \times 10^9 \times 2.18 \times 10^{-4} \text{ lb/in.}$
If the rotor is just restrained,

$$P = \frac{6.55 \times 10^5}{30} = 2.2 \times 10^4 \text{ pounds or 10 tons}$$

A minimum preload of 20 tons was considered justified on the basis of this calculation. The results of subsequent tests are reported in an article on the bearings.⁷ In that report it is shown that the preload was increased to obtain a higher axial stiffness of the bearings so as to raise the frequency of the tilt vibration of the rotors on the bearings, to at least 22 cycles per second so as to avoid any resonance with the frequency of rotation. This sets the minimum load required at 40 tons.

Conclusion

As in the Mark I design, the stress concentration at the head of the rotor fixing screw leads to very high local stresses. The screw has been made from a high quality steel alloy and was subjected to magnetic crack test before acceptance. Under these circumstances the screw can be relied upon to withstand all normal loads. If it is ever subjected to the 400 ton fault load it would be necessary to remove the screw and subject it to thorough crack tests before it could be relied upon for further use. In such a case a new screw should be used as a matter of safety and reliability. In any modification of the designs it would be important to strengthen the screw.

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as a reference.