THE DESIGN AND CALIBRATION OF A HIGH-SPEED DYNAMOMETER FOR SMALL GAS TURBINE APPLICATIONS

by

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THE DESIGN AND CALIBRATION OF A HIGH-SPEED DYNAMOMETER FOR SMALL GAS TURBINE APPLICATIONS

INTRODUCTION

The fact that an engine will run is only of minor importance. The important consideration is--will it do work? And if so, how much and at what efficiency? Although the gas turbine in the Oregon State College laboratory would run, it was not known how much work it could do. It was the purpose of this thesis to design, construct, and calibrate a dynamometer that would provide the answer to this question.

A dynamometer is an instrument for measuring force or power. More specifically, it is an instrument that measures the power output of an engine or motor. There are two basic types of dynamometers, the absorption type and the transmitting type. Because of the conditions of installation of a dynamometer on the gas turbine in the Oregon State College laboratory, the absorption-type dynamometer was preferable.

SPECIAL CONDITIONS OF THIS DESIGN

Some of the conditions imposed by the nature of the gas turbine made the design of this dynamometer a unique problem. It was expedient that the dynamometer be designed to run at a maximum speed of 20,000 rpm in order for it to be directly connected to the gas turbine. This direct connection was made to eliminate any error that would be made by the presence of a gear train.

A dynamometer size of 100 hp at 20,000 rpm was chosen. The turbine on which this dynamometer was mounted has run at 15,000 rpm and a similar one in the Oregon State College Forest Products Laboratory has run at 20,000 rpm. Consequently, a design speed of 20,000 rpm was the maximum considered.

Since the power developed was not to be utilized, the dynamometer should absorb it. It was necessary to have some means of disposing of this waste energy. The cooling system was required to handle 100 hp or 254,500 Btu per hour or 4240 Btu per minute. This cooling system was a major factor in the design.

The load imposed by the dynamometer had to be infinitely variable from zero to a maximum. Actually, all of the dynamometers investigated would exert some load on an engine even with the control in the "no load" position. As long as the dynamometer was turning, the bearings, oil seals, and windage would produce some load.

For the dynamometer to be of any value for experimental purposes, it was compulsory that the possible error be kept to below 1 per cent. Any larger error would make the efficiencies figured from the measured power output worthless.

For the sake of safety, it was desirable to have the dynamometer remotely controlled and remotely indicating. Any failure in the machine at such high speeds could be fatal for personnel close to it. Eventually the turbine and dynamometer should be placed in a test pit and then the remotely-controlled and indicating features would be a necessity.

To avoid the hot air blast of the turbine, the dynamometer had to be driven from the compressor end of the turbine. The best position for mounting it was on the cover of the air duct. It had to be mounted in such a way that it did not affect the measurement of the air into the compressor. This automatically placed a minimum length of approximately 10 inches on the quill shaft. An investigation of the shaft also showed that it could not be much longer than 10 inches without operating in the critical speed range.

Because of the mounting problem, the weight of the dynamometer had to be kept at a minimum. If the weight were too great, an additional mounting stand fastened to the gas turbine stand would be required. This would increase the difficulty of maintaining accurate alignment of the quill shaft.

Oil fog lubrication was imperative for operation of ball bearings at such high speed. All low speed bearings would be packed with grease but some method of sealing would have to be provided.

Some dynamometers can be used as motors to start engines or turbines. This is very desirable for some installations but the turbine in the Oregon State College laboratory was equipped with a nozzle for starting with compressed air so the starter type would not be of any advantage.

The cost was a very decisive factor in the design of this dynamometer. To keep the cost at a minimum, it was decided that it should be built in the machine shop in the Engineering Laboratory. It had to be designed so that all of the machining could be done with the existing facilities in a minimum of time.

BEST TYPE OF DYNAMOMETER

Because the speed, size, and accuracy are important items in the choice of a dynamometer, the following table is presented.

TABLE I

POWER-MEASURING DEVICES

Type	Approx Spd Limit, rpm	Usual Power, hp	Prob Error, %
Prony brakes:			
Block	2,000	10	1-5
Band	1,000	5	1
Wooden cleats on bands	1,000	200	1/2-1
Rope (3/4-in.) with wooden cleats	1,000	50	1
Fluid friction dynamometers:			
Froude (ordinary "water" brake)	10,000	25,000	1/5-1/2
Westinghouse (turbine)	4,000	5,000	1/10-1/5
Alden	1,000	5,000	1/2-1
Fan brake	2,000	200	1-5
Electric dynamometers:			
Electric eddy-current brake	6,000	300	1/2-1/5
Electric generator	750-4,000	30,000	1/10-1/5
Transmission dynamometers:			
Torsion	1,000-3,000	50,000	1-5
Kenerson	1,500	100	2

Speed limits given above are approximately the highest allowable. Limits of hp refer to the largest sizes made commercially. Marks' Handbook, Fifth Edition, page 2082.

It is immediately apparent that the prony brake type dynamometer would be unsuited for this application because of themeans of applying the load and the method of control.

An investigation of the power absorbed by the Froude water brake will show why it was not satisfactory. The formula (7, p168) is

$$hp = \frac{3.18}{10^{10}} n^2 d^5$$

where "n" is the speed in rpm and "d" is the rotor diameter in inches. Solving for "d" when "n" is 20,000 rpm and the bhp is 100 hp,

$$d^5 = \frac{100 \times 10^{10}}{3.18 \times 20{,}000^2} = 785$$

$$d = 3.74 \text{ inches}$$

Accurate, easy control of the load that this type of dynamometer imposed on the gas turbine would be very difficult. This difficulty of obtaining accurate control at high speeds is characteristic of all fluid brakes. For instance, each disk of the Webb viscous brake absorbed power according to the formula (7, p170)

$$hp = \frac{n^3 \ (r_2{}^5 - r_1{}^5)}{130{,}000{,}000}$$

where "n" is the speed in rpm, "r_2" is the radius of the plate in feet, and "r_1" is the radius of the annular ring of water in the casing. If the casing were full of water, the speed, 20,000 rpm, and the power, 100 hp, then the disk radius, "r_2", would be 0.277 feet or 6.7 inches in diameter.

For this disk to absorb 1 hp at 20,000 rpm, the annulus of water required would be 0.275 feet radius or 0.024 inches of water in contact with the disk. Accurate control, especially at the light loads, would be almost impossible at the high speeds.

The fan brake would not be accurate enough to control due to changes in atmospheric pressure, density of the air, and air currents.

The electric generator is probably the most widely used type of dynamometer. The Boeing 501 gas turbine was tested with a d-c generator, but a gear train was used to reduce the speed. The high speeds of 20,000 rpm would cause commutation difficulty. The maximum speed for commutation with copper wire gauze brushes and oil lubrication is 13,000 fpm (5, p16). At 20,000 rpm the maximum size of commutator would be 2.44 inches in diameter.

A two-inch commutator would be satisfactory for this application but the building of such precision commutators requires machines and skill that were not available to the author.

With their specialized machinery and technical skill, General Electric Company has built a dynamometer of this type with a product of horsepower and rpm of 2.5×10^6 (5, p17). The dynamometer needed here would require a product of 2.0×10^6 and the skill and special machines were not available for its construction.

When a shaft transmits a torque there is some twisting of the shaft. The amount of twist in a particular shaft will depend upon the amount of torque transmitted. A circuit, usually electronic or optical, may be set up to measure the amount of twist. The torsional shaft and the indicating circuit constitute a torsional dynamometer.

A torsional type dynamometer would operate very well at the high speed encountered. A model of this type that operated at 30,000 rpm has been made and marketed. It was priced in the $3,000 range. This cost was a major factor in deciding against it. This type of dynamometer would only transmit the power so some method of absorbing the power output would still have to be provided. This alone would be as large a problem as the design and construction of a dynamometer of the absorbing type. The indicating circuit of a torsional dynamometer is of complex nature. The complexity naturally increases the cost.

Because it most nearly satisfied the conditions imposed by the problem, the eddy-current type dynamometer was chosen. The rotor was symmetrical about the axis of rotation so it could be balanced easily to withstand the high speed. The accuracy of this type could be well within the 1 per cent allowable error. This type would lend itself very well to remote control and with additional work could be remote indicating. It was a comparatively light machine and could be mounted on the inlet air duct and be directly connected to the compressor. A dynamometer of this type could be built with the skill and facilities available.

There was some question about the eddy current dynamometer operating satisfactorily at speeds above 6,000 rpm (see Table I). An investigation of published material available revealed no reason why 6,000 rpm was the limit so this type of dynamometer was chosen partly to determine whether or not it would function properly at the higher speeds.

THEORY OF EDDY CURRENT BRAKES

Any time a conductor is moved through an electrical field, a current is induced in it. If the conductor should extend beyond the edge of the electrical field, the induced current will flow in a circular path within the conductor. From the circular shape of the path comes the name eddy current.

An eddy current brake consists of an electrical field and a conductor. Either one may be connected to a prime mover and the other be stationary. In the particular design dealt with here, the eddy currents are induced in the rotor by the stationary field mounted in the stator. To obtain a more definite change in flux along the periphery of the rotor, four poles were used to concentrate the flux at 90° intervals. The more definite change of flux yielded induced currents larger than would have been obtained without the poles. From an investigation of the phase relation of motion and electrical fields, the induced current was found to flow radially outward in the region of the electrical field. The right hand rule (10, p284) was used to determine the direction of the induced current.

The electrical circuit is completed by the current flowing radially outward in the region of the electrical field, along the surface of the rotor, and radially inward at a point between the poles.

The current moving in the region of the electrical field experiences a side thrust in a direction opposite to the motion of the conductor (10, pp227, 285). Thus the motion of the conductor is opposed and a braking torque is applied to the prime mover connected to the conductor. The opposite reaction of the braking torque occurs in the field coil and tends to rotate the stator with the rotor. If the stator is mounted on anti-friction bearings, the torque required to keep it from rotating may be measured. Since the measured torque is an equal and opposite reaction to the braking torque felt by the rotor, it is the load applied to the prime mover.

ELECTRIC DESIGN

The first step in the design of the electrical circuit was to compute the flux required to develop 100 hp. For this computation the resistance of the path of the induced current in SAE 1020 steel was assumed to be 20 micro-ohm.

Because of the low cost and the availability, SAE 1020 steel was used. Computations (see Appendix I) with the data from Fig. 1 showed that it would be satisfactory.

Knowing the resistance and the power absorbed per path, the induced voltage "E" was computed by use of Ohm's law. Then the total flux ϕ in maxwells was computed from the equation (10, p287, eq 12-4)

$$\phi = E\, t \times 10^8$$

$E = \frac{\phi}{t}$

where "E" is the induced voltage and "t" is the time in seconds required for a point on the rotor to pass from one pole to the next.

The flux per unit area required was found to be 46 kilomaxwells per square inch (see Appendix I). However, enough ampere turns to saturate the magnetic path were considered advantageous so a value of 60 kilomaxwells per square inch was used.

Assuming dimensions for the magnetic path, the ampere turns necessary to maintain the flux were computed

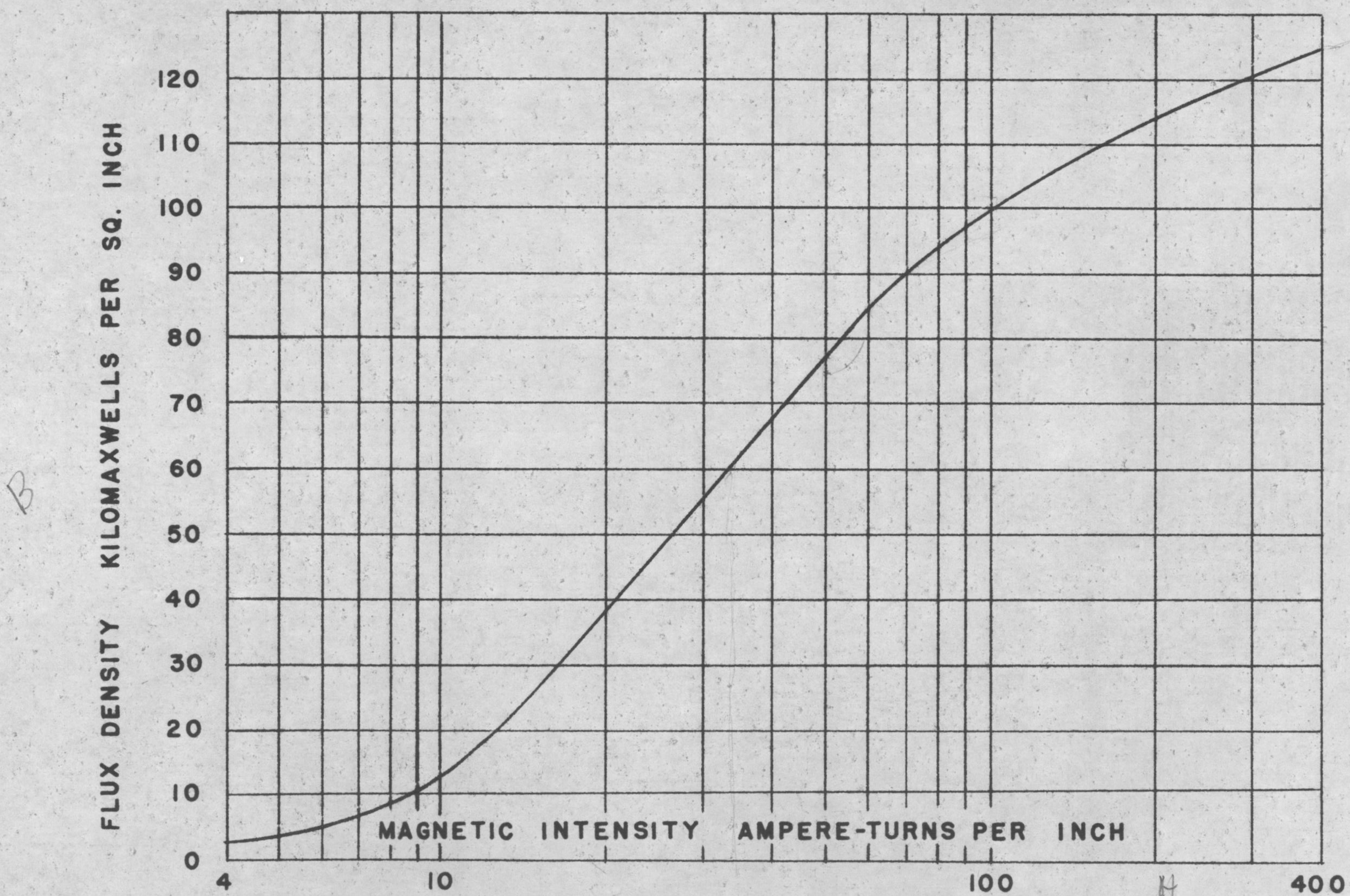

FIG. I MAGNETIZATION CURVE FOR SAE 1020 STEEL (9, p.48) (see Appendix I). Allowing one ampere per 2,000 circular mils of wire area, the necessary number of turns of a given size of wire was computed. The cross sectional area of the coil was checked against the space available determined by the assumed dimensions. This was repeated for different sizes of wire until a size was found that would fit in the available area. The total resistance of the coil was checked to make certain it would not limit the current to a value less than the desired current when 120 volts were used across the coil.

The heat generated in the coil by the field current had to be disposed of. This amounted to approximately 24 watts per second. If the entire amount were given off by radiation, about 0.32 watts would be radiated by each square inch of outside coil surface per second. With the open type of caging employed to house the dynamometer, this amount of radiation presented no difficulty.

A rheostat was used to control the load by controlling the strength of the field. Choosing a rheostat with low change in resistance per turn of control knob made it much easier to get accurate control of the field strength.

The sum of the rheostat and coil resistances was enough to limit the field current to the 0.2 ampere desirable (see Appendix I) under full load conditions.

MECHANICAL DESIGN

The dynamometer was assembled from the following parts: The rotor, stator, and the housing (see Fig. 2). The rotor was mounted on high speed ball bearings in the stator. The stator also held the field coil and poles. It was mounted in anti-friction bearings in a stationary housing fastened to the gas turbine.

The bearings used to mount the high speed rotor in the stator cradle were precision type bearings No. 6213 YC/3 manufactured by the SKF Company. The product of inside diameter in millimeters and the speed in rpm, called the "dn" value, were listed as 700,000 for these bearings. The "dn" value at which these bearings were to operate (1,300,000) was almost twice that recommended. Consideration of the exceptionally light loads and the choice of oil fog lubrication showed that the bearings should give satisfactory operation. The knowledge that they would not be run at maximum speed for any extended length of time was a factor in favor of the decision.

Lubrication of the high speed bearings presented a problem. An oil fog was necessary so it was obtained by allowing the oil to drop on the rotor so the impact of the oil and the rotor would atomize the oil.

<u>The Rotor</u>. The rotor (see Fig. 3) was machined from a solid billet of SAE 1020 steel. It was symmetrical

about the axis of rotation, making it easy to balance. Absolute balance of the rotor was of utmost importance to reduce vibration to a minimum.

The rotor was bored out to a depth of $8\frac{1}{2}$ to 9 inches from the end away from the turbine. Tap water was to be used to cool it internally, although an air blast was also considered.

The water was released at the full depth of the cavity by a pipe through the bored-out end of the rotor. It then flowed the full length of the cavity to be discharged by centrifugal force from the end through which it had entered. A ring collected the discharged water so it could be disposed of to a floor drain.

The Quill Shaft. A shaft was necessary for the direct connection between the rotor and the gas turbine. Because the dynamometer was mounted on the inlet air duct cover, a shaft of 10 or 12 inches in length was required. The type of shaft best suited for this connection was a quill shaft. This is a hollow shaft of small cross sectional area. The main advantage of this type is that it can stand slight misalignment without vibrating.

To enable the shaft to withstand even more misalignment, a semiflexible coupling was used to connect it to the gas turbine. This coupling (see Fig. 4) employed rubber rings under compression to transmit the torque.

The critical speed of the quill shaft was investigated to make certain it would not be operating in the critical range. Computations showed that a quill shaft of three-fourths inch outside diameter and one-half inch inside diameter, and twenty inches in length would pass through critical range at approximately 60,000 rpm. However, the quill shaft, the dynamometer rotor, and the turbine rotor comprised a two-dimensional torsional system with a critical speed of 20,300 rpm. By shortening the shaft to 15 inches, the natural frequency of the system was increased to 23,400 rpm. Since this was still too close to the operating speed, it was shortened to 12 inches to obtain a natural frequency of 26,200 rpm.

This quill shaft would transmit 30 foot-pounds of torque satisfactorily.

The Torque Arm. The torque or lever arm was designed to be as short as practical to obtain a maximum scale reading when determining the torque because the maximum torque expected was approximately 25 to 30 foot-pounds.

A simple method of measuring this torque would be to apply a known, infinitely variable force to the stator through a lever arm of known length. A good set of scales would supply such a force very satisfactorily. The desired torque is the product of the force and the known lever arm.

Once the load torque is known and the speed is determined, the horsepower could be computed from the equation (8, p24)

$$hp = \frac{Prn}{63025}$$

where P in pounds is the force applied at the distance r in inches from the axis of rotation and n is the speed in rpm. The distance r was chosen so that 63025/r was an integer times 10^n to simplify the equation for horsepower and make the computations much easier. The distance r was chosen to be 10.5 inches which made the brake constant, $\frac{1}{6000}$. Thus the equation for horsepower was simplified to

$$hp = \frac{Pn}{6000} .$$

<u>The Cage</u>. The cage and the cradle were of the open type to facilitate cooling of the coil and rotor by radiation and convection. This type of construction was desirable from the weight saving it effected.

CONSTRUCTION AND CALIBRATION

The end bells of the stator and housing consisted of 1/4 inch steel plate with bearing hubs welded on. Bars of 1/4 x 1-1/2 inch steel were welded between the end bells to form the cage. This type of fabrication was unsatisfactory because of warpage of the parts being welded.

Some difficulty was experienced in lining up the dynamometer rotor with the gas turbine shaft. The face of the compressor inlet duct was perpendicular to the compressor shaft so the quill shaft was set perpendicular to this surface with a dial indicator. The dynamometer rotor was then lined up with the shaft in a similar manner.

The no-load friction in the dynamometer was constant above 2,000 rpm. It was difficult to rotate the mechanism backwards at any appreciable speed to determine the tare so the scale was adjusted to a positive reading with the rotor stopped and the difference between this reading and the reading at any rpm with no current flowing in the coil was used as the no-load friction.

The curves of Fig. 5 show the scale reading in pounds versus the current in the coil. Due to the limited power output of the gas turbine and the short runs necessitated by the high temperatures, complete and conclusive data were not obtained. However, there was a tendency of the torque to increase with speed while the field strength remained constant.

RECOMMENDATIONS

1. It is recommended that a new housing be built that is true and not warped by welding.

2. Testing should be continued to include a complete investigation of torque versus coil current for all speeds from 6,000 rpm up to 20,000 rpm. Some other means of driving the dynamometer must be provided.

3. Thermocouples should be staked into the bearing hubs. These could be used in conjunction with the temperature of the cooling water in and out to measure and maintain a constant rotor temperature.

4. An investigation of the effect of rotor temperature on the characteristics of torque versus coil current should be made for all speeds.

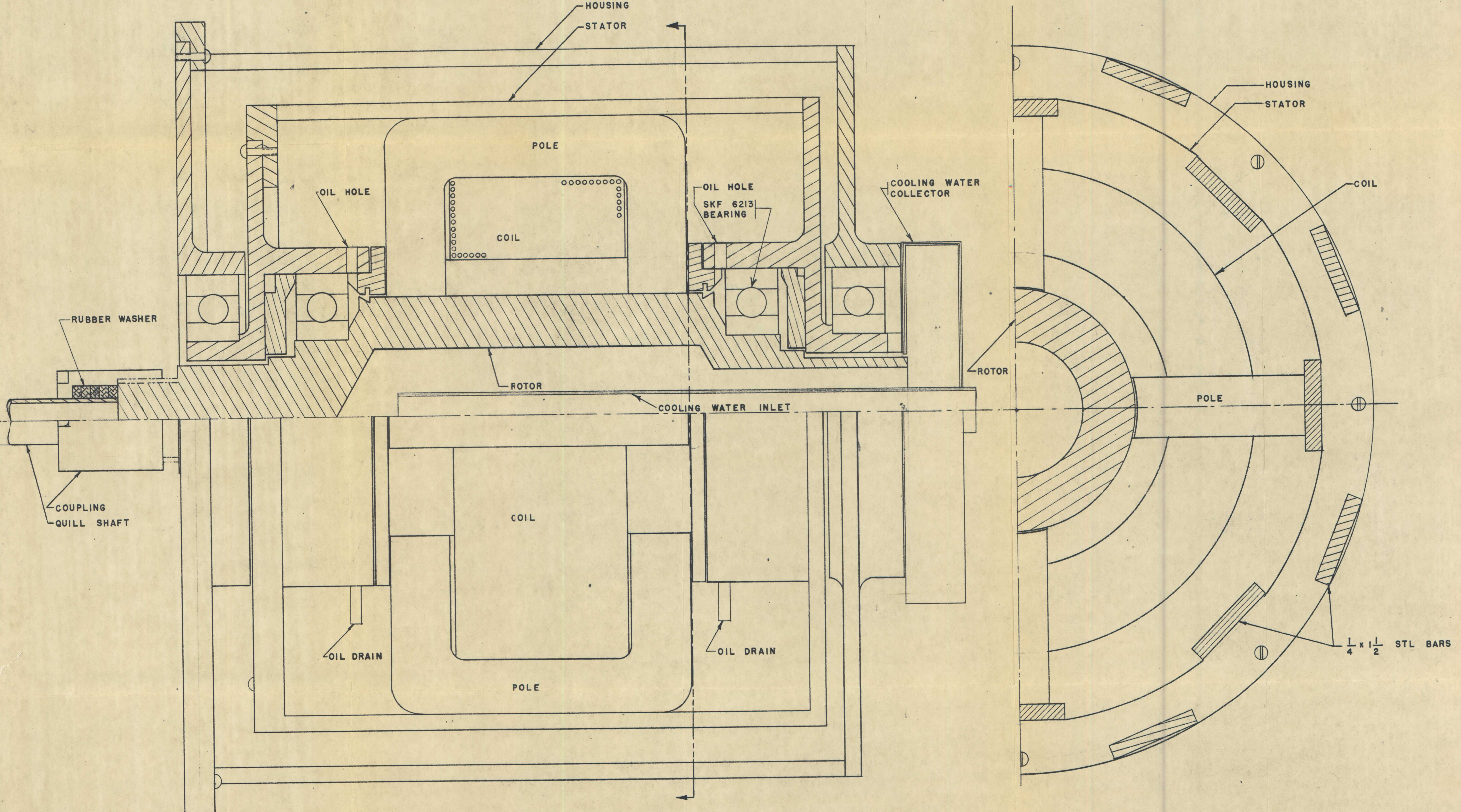

FIG. 2 ASSEMBLY —FULL SIZE

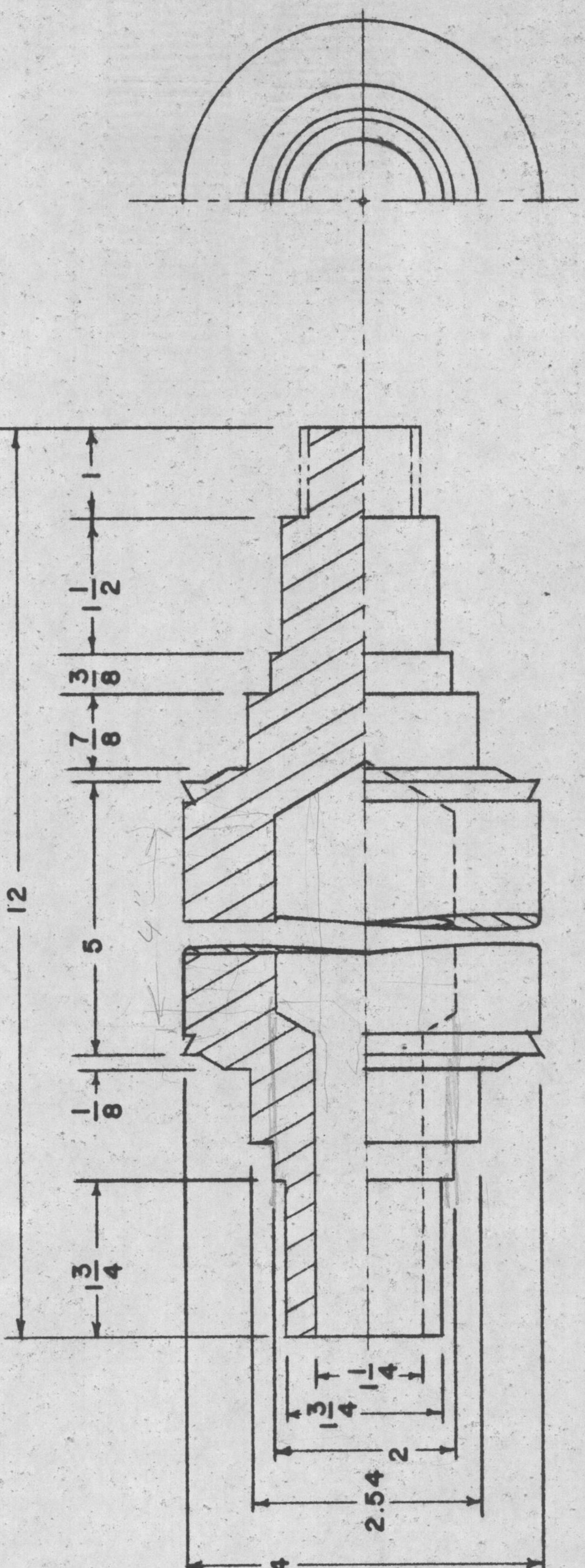

FIG. 3 ROTOR — HALF SIZE

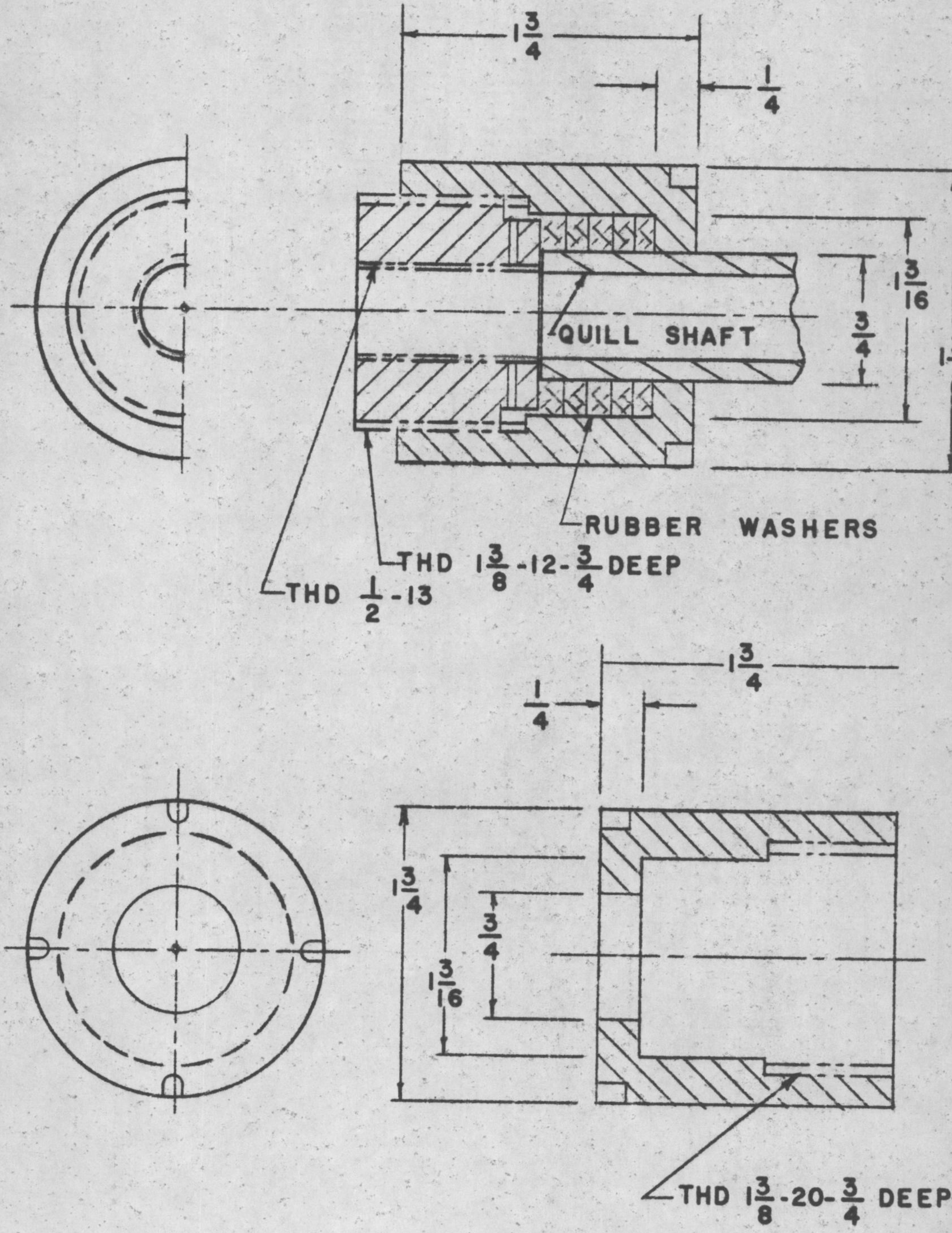

FIG. 4 COUPLINGS —FULL SIZE

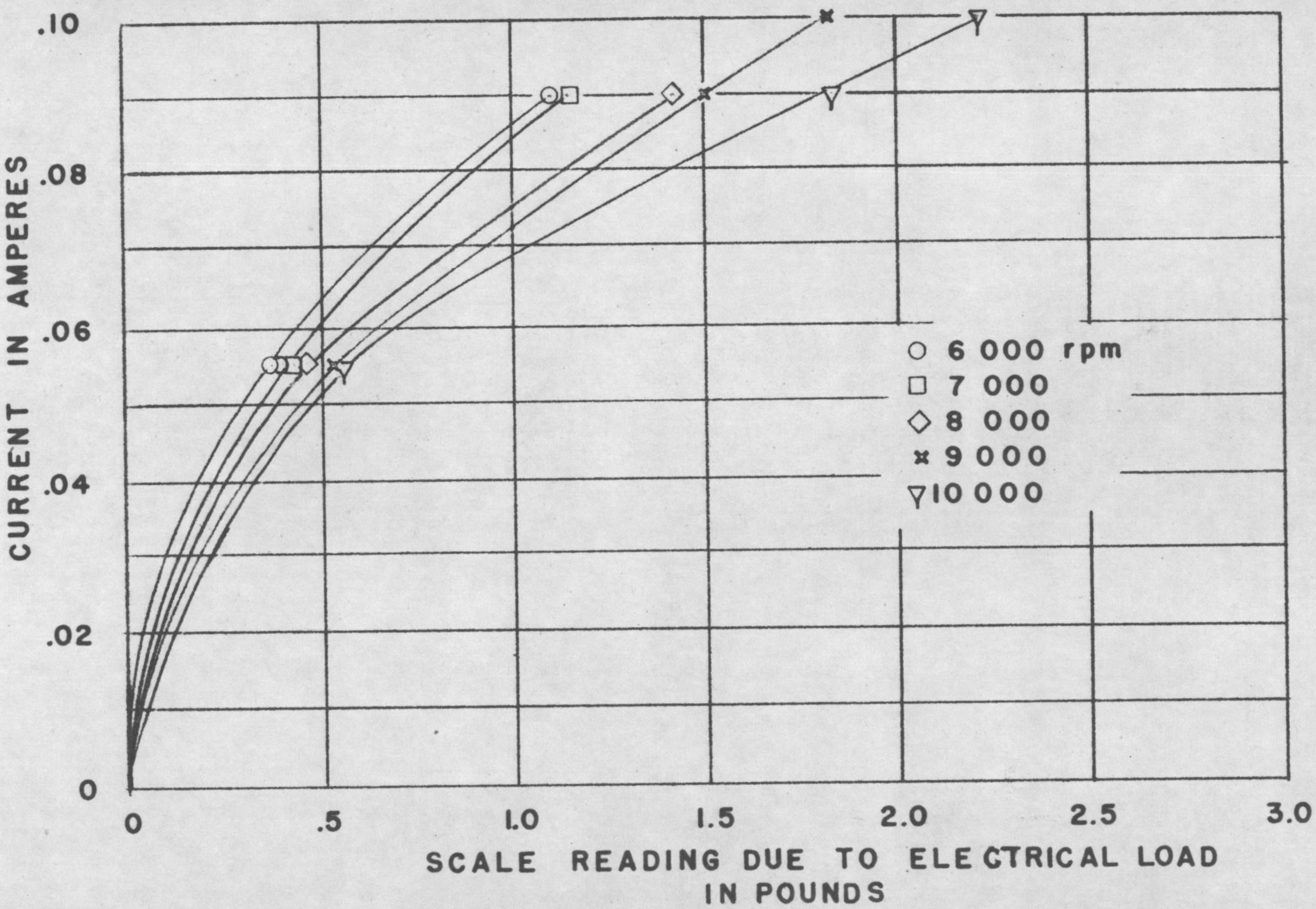

FIG. 5 ELECTRICAL CHARACTERISTICS

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APPENDIX

APPENDIX I

Assume 100 hp at 20,000 rpm.

The power absorbed per path $= \frac{1}{4} \times 100$ hp $= 25$ hp.

$= 18650$ watts.

The resistance R was assumed to be $2.02 \cdot 10^{-5}$ ohms per path.

Since $P = \frac{E}{R}$, the induced voltage $E = \sqrt{PR}$,

$$\text{or } E = \sqrt{18650 \cdot 2.02 \cdot 10^{-5}}$$

$$= 0.613 \text{ volts}$$

The equation for flux is

$$\phi = ET \cdot 10^{8} \text{ maxwells}$$

Where T is the time in seconds required for a particle on the rotor to pass from one pole to the next.

$$\text{So } \phi = 0.613 \cdot 7.5 \cdot 10^{-4} \cdot 10^{8}$$

$$= 46 \text{ kilomaxwells}$$

If 60 kilomaxwells per square inch were used to saturate the steel the magnetic intensity required was 34 ampere-turns per inch. With an average path of 14 inches of steel and two air gaps of 0.025 inches each, the total ampere-turns required was:

Steel: $14 \times 34 = 476$

Air gap: $NI = \frac{\phi L}{\mu S} = \frac{60 \times 0.050}{3.2 \times 1} = 894$

Total = 1370 ampere-turns

Used: 1400 ampere-turns

No. 25 AWG wire was used. The allowable current was 0.2 amperes making 7000 turns necessary.

The resistance of the coil was 457 ohms. The area needed for the coil was 3.18 square inches and the available area was 3.9 square inches.