HOMOPOLAR MOTOR-GENERATOR DESIGNS FOR CHEAP INERTIAL ENERGY STORAGE

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INTRODUCTION

One of the limitations to the research in fusion power is the need for a cheap method of storing large amounts of energy. This energy is needed for the main confinement magnetic field in the next Tokamak and must be capable of transfer from the store to the coil in a short time (1 to 10 sec.). Conventional power supplies are unnecessarily expensive since it is quite possible that the existence of the field for 1 or 2 seconds would be sufficient.

An inertial energy storage system utilizing a homopolar generator was studied for feasibility and found to be satisfactory resulting in the design, construction and testing of a bench model along with the design of an intermediate 50 M.J. machine to power the present Texas Tokamak.

ELECTRICAL AND MECHANICAL CONCEPTS

The homopolar generator is an electric machine which converts stored, rotational kinetic energy to electric energy using the Faraday effect. It is a low voltage-high current machine which can be operated in a pulsed mode and as such is a good current source for high magnetic field solenoids. The basic machine consists of a cylindrical rotor placed in a magnetic field such that the axis of the rotor is parallel to the direction of the field. Utilizing an external power source the machine is operated as a motor whereby the rotor is accelerated to a frequency ω_o at which time it has a stored kinetic energy of $1/2\ I_o\ \omega_o^2$ where I_o is the moment of inertia of the rotor. If a load is now connected to electrodes placed at the outer and inner radii of the wheel, the Faraday voltage, produced when the magnetic field is on, drives a current pulse through the load. Concurrently, there is a $\underline{J} \times \underline{B}$ force which generates a deceleration torque slowing down the wheel. The net result is a conversion of the inertial energy of the rotating wheel into electric energy in the load.

It is noted that the amount of stored energy in a given rotor is a function of ω^2 which leads to the necessity of high speeds to store appreciable amounts of energy. One of the best measures of design efficiency is the amount of energy stored per pound of rotor. The upper limit to the rotational speed, which corresponds to the maximum amount of energy stored, is reached when the rotor is operating at its limiting design stress.

The rotor must run on bearings designed for low friction and high stiffness which in general are not compatible since bearing stiffness is normally obtained at the expense of added mechanical friction. In this design trade-off, one strong constraint is that the critical speed of the rotor should be well above (at least 20%) the operating speed of the rotor.

Making electrical contact on a high speed surface for high current density is difficult. Liquid metal jets have been used to obtain less friction than conventional brushes but considerable design complication is encountered with the use of liquid metal. Recent tests have shown that copper-graphite brushes can be made to operate satisfactorily under design conditions similar to our requirements.

Counterrotating disks were investigated with the advantage of reduced torque on the frame since the angular momentum changes would approximately cancel each other. This scheme was abandoned for several reasons such as the necessity to provide for emergency deceleration of one rotor, unbalanced axial magnetic field forces, poor access to center bearings and brushes and the added possibility of vibrational instability.

2000 M.J. FEASIBILITY STUDY

Certain advantages are obtained in a large machine over a small one such as reduced bearing and windage losses relative to the amount of energy stored. Other factors such as brush design become more critical since the ratio of surface area to total rotor weight reduces with increased size. Maximum stress levels of 30,000 psi were used to determine the size of the rotor. The resulting designs required two rotors of 70 tons each at 15 feet in diameter.

Maximum voltage is 1000 volts at maximum speed. The current would vary with the load but could go as high as 2,000,000 amperes. Cycle time was set at 100 seconds thereby eliminating most of the heat problem in the rotor. The design of this machine was not completed after sufficient work had been done to verify the feasibility.

Cost estimates based upon the preliminary designs were slightly over $2,000,000. Since this preliminary study was made, a number of refinements could be applied but none have proven to be radically different from the original estimates. Perhaps the most costly addition would be required to reduce the windage losses. Several ideas on boundary layer control will be experimentally investigated on the bench model.

Before completing the design of this machine it was decided to build a bench model which had been partially designed by a senior Mechanical Engineering class and to proceed with the detail design and specifications of a 50 M.J. prototype unit to use with our present Tokamak as a replacement for the batteries.

50 M.J. PROTOTYPE UNIT

Designs for a 50 M.J. prototype unit are nearly complete. A sectioned drawing of this unit is shown in Figure 1 with the basic components identified. The split rotor is equivalent to twin rotors bolted together--but they are electrically separate with each rotor having a set of inner and outer brushes. Basically this unit uses the design concepts developed for the full scale 2000 M.J. unit along with the information obtained from the bench model tests.

In this machine the inner brushes on each side are not electrically connected together just as the outer brushes are not connected since the current will flow in the same direction in each. The machine would produce the same voltage with the twin rotors as a single rotor if the brushes were not separated.

The following sections explain the design analysis and show how many of the design decisions were made.

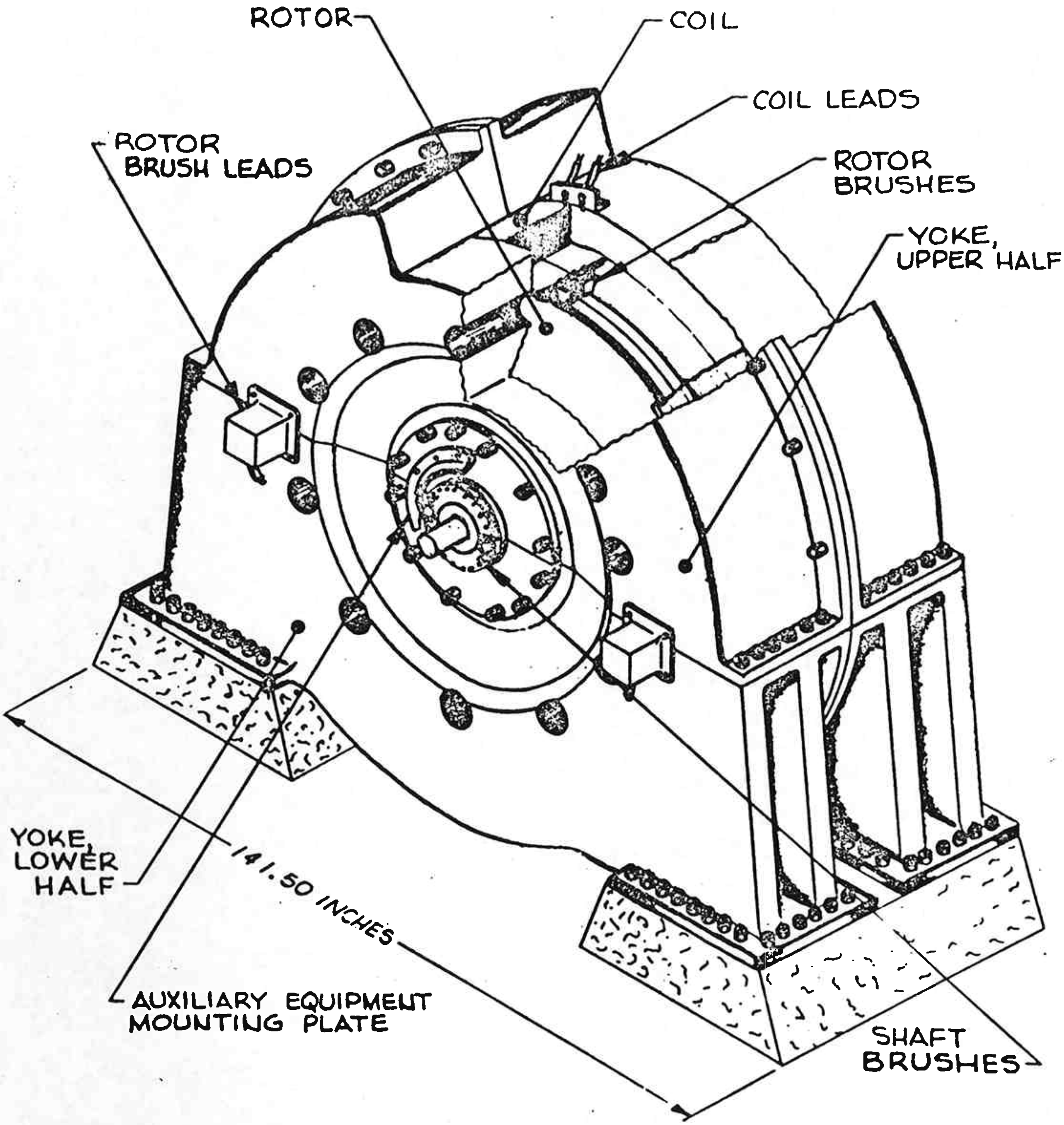


Figure 1 50 M.J. Prototype Unit

Rotor Design

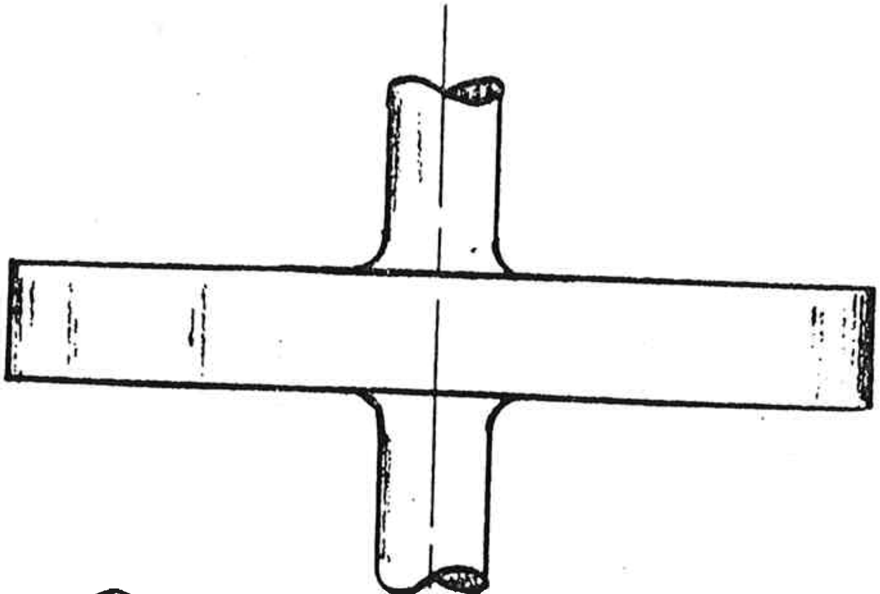
Preliminary Considerations

The size, shape, and material of the rotor were selected primarily on the basis of stresses produced at given energy storage levels. In the final selection, however, constraints imposed by magnetic and electrical consideration played a significant role.

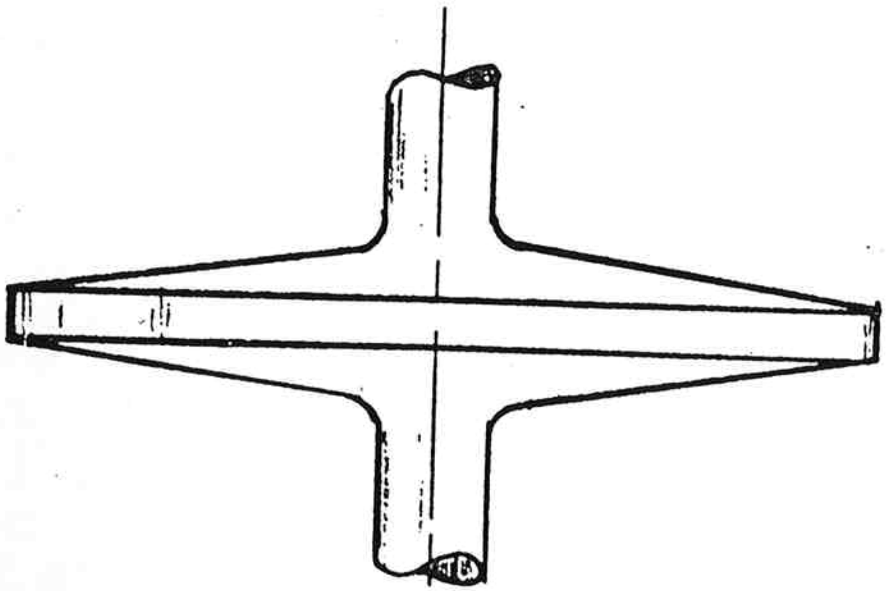
The major stress producing factor is the radial body force imposed by the angular velocity. Secondary loadings are torque produced in stopping the rotor and non-uniform temperatures induced by brush losses, by current flux, and by air friction. The stresses produced by secondary loadings will be similar in all configurations, and play no part in the initial selection of shape.

Figure 2 shows the cross section of three shapes which were considered in the preliminary stages of the selection procedure. The constant thickness disk has the advantage that simple, closed form solutions of the stress problem are available. The disk with linear taper cannot be solved in closed form, but is more efficient than the uniform disk and is a simple shape to fabricate. The constant stress disk has a theoretical solution and is, in principle, of optimum shape. As later considerations show, however, it is not feasible to take full advantage of this efficient but complicated shape.

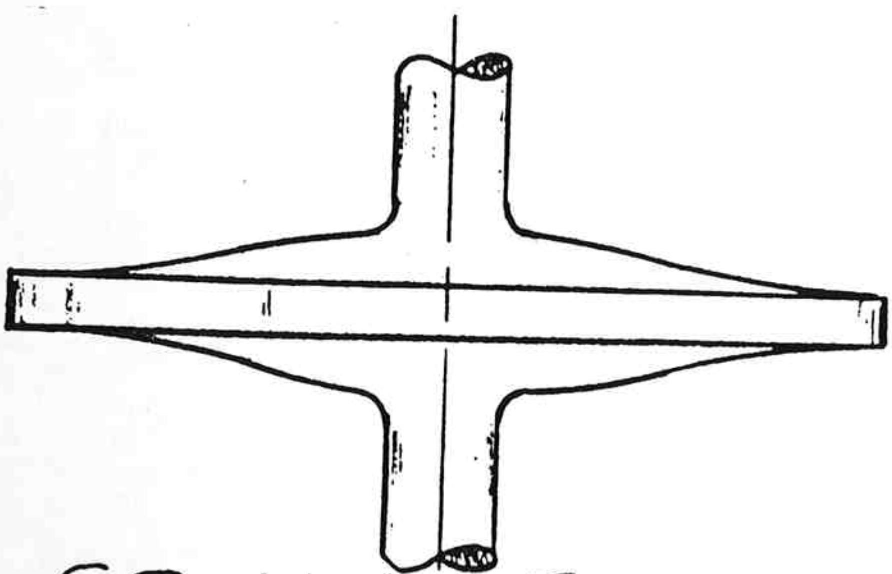
The preliminary calculations for sizing of disks of these three shapes are given in the following sections. The dimensions and symbols used in these sections are defined in the symbol table.



CONSTANT THICKNESS



LINEAR TAPER



CONSTANT STRESS

Constant Thickness Disk

Basic relations

a) Kinetic Energy "E"

The kinetic energy stored in a spinning disk is given by

$$E = \frac{1}{2} I \omega^2 = \frac{1}{2} \frac{\gamma}{g} \omega^2 \int_0^c \int_0^{2\pi} \int_0^b r^3 \, dr \, d\theta \, dz$$

$$E = \frac{\pi}{4g} \gamma \omega^2 c b^4 \text{ INCH - LBS}$$

(1)

b) Weight "W"

The weight of the disk is

$$W = \pi \gamma c b^2 \text{ LBS}$$

(2)

c) Stress

Due to the steady spinning of the disk a state of plane stress will exist in which the stresses are given by the following[1].

$$\sigma_r = \frac{\gamma \omega^2}{g} \left[\frac{3+\nu}{8} (b^2 - r^2) + \frac{(1+\nu)}{b(1-\nu)} (C^2 - 3z^2) \right]$$

$$\sigma_\theta = \frac{\gamma \omega^2}{g} \left[\frac{3+\nu}{8} b^2 - \frac{1+3}{8} r^2 + \frac{(1+\nu)}{6(1-\nu)} (C^2 - 3z^2) \right]$$

$$\sigma_z = \tau_{R\theta} = \tau_{rz} = \tau_{\theta z} = 0$$

The critical stress condition, i.e. maximum shear or principal stress difference, will occur at $r = z = 0$, at which point

$\sigma_r = \sigma_\theta = \sigma$. Taking Poisson's ratio as 0.3 we have

$$\sigma = \frac{\gamma \omega^2 b^2}{g} \left[0.413 + 0.092 \left(\frac{c}{b}\right)^2 \right]$$

or, assuming that the thickness to radius ratio c/b is less than a quarter, we have

$$\sigma = 0.419 \frac{\gamma}{g} \omega^2 b^2 \quad \text{LBS/INCH} \tag{3}$$

d) Design Calculations

The specific energy of the disk is given by

$$u = \frac{E}{W} = \frac{\frac{\pi}{4g} \gamma \omega^2 c b^4}{\pi \gamma c b^2} = \frac{\omega^2 b^2}{4g} \tag{4}$$

Now, if the value of σ is to be held to a certain value, the allowable angular velocity is given by

$$\omega^2 = \frac{\sigma g}{0.419 \gamma b^2}$$

so that the energy that can be stored per pound of material is

$$u = \frac{\sigma g}{0.419 \gamma b^2} \quad \frac{b^2}{4g} = 0.597 \frac{\sigma}{\gamma} \text{ INCH-LBS/LB} \tag{5}$$

Equation (5) can now be used to calculate the size rotor required to store a given amount of energy. It is of interest that the thickness to radius ratio of the disk is not a factor, since only material properties influence the amount of material required.

e) Example Designs

As points of departure, we consider sizing structural steel and aluminum rotors for energy requirements of 50 MJ. For both materials 30 ksi is a reasonable allowable stress level. We take the densities as $\gamma_{steel} = 0.283\ lb/in^3$, $\gamma_{aluminum} = 0.105\ lb/in^3$.

For the 50 MJ steel rotor the calculations proceed as follows.

$$U = \frac{0.597 \times 30 \times 10^3}{0.283} = 6.33 \times 10^4 \text{ INCH-LBS/LB}$$

$$= 6.33 \times 10^4 \times 10^3 \times \left(\frac{1}{12 \times 0.74 \times 10^6}\right)$$

U = 7.12 MJ/KIP

Since W = 50/7.12 ≑ 7.02 KIPS we have

$$\text{Volume} = \frac{7.02 \times 10^3}{0.283 \times 1728} = 14.4 \text{ CU FT}$$

If we take the thickness of the disk as approximately one fourth the radius we find that a disk 8" thick with a radius of 32" has a volume of

$$V = 0.667 \times \pi \times (2.67)^2 = 14.9 \text{ CU FT}$$

Once the dimensions of the disk are known, the allowable angular velocity can be calculated. In this case

$$\omega = \left(\frac{30 \times 10^3 \times 32 \times 12}{0.419 \times 0.283 \times (2.67)^2 \times 144}\right)^{1/2} = 480 \text{ RAD/SEC}$$

$$\omega = 480 \text{ RAD/SEC} \times \frac{60}{2\pi} = 4600 \text{ RPM}$$

Energy	51.7 M.J.	
Material	Steel	Aluminum
Weight (KIPS)	7.27	2.7
Volume (cu.ft.)	14.9	14.9
Thickness (ft.)	.67	.67
Radius (ft.)	2.67	2.67
Speed (rpm)	4600	12,400

Table 1. Typical Constant Thickness Disk Designs

Table 1 gives results for the examples cited above.

Of course, other aspect ratios could be utilized to provide the same volume and hence the same energy. It is noted that the speed of the disks will be inversely proportional to the radius.

Linearly Tapered Disk

The tapered disk, with dimensions as shown in Figure 2, has the relative advantage of easier fabrication than the more complicated constant stress disk. Closed form solutions are not available for the tapered disk so the stress analysis was performed numerically using the finite element program. The particular shape of taper that was chosen as representative of this class had the dimensions:

a = 3.5 in.	c = 1.0 in.
b = 32.0 in.	d = 15.0 in.

The finite element analysis which was used yielded a complete state of stress for the body, but only the center line stress σ is of interest here. Reducing the stress to the form of equation (3) we have

$$\sigma = 0.212 \frac{\gamma}{g} \omega^2 b^2 \qquad (6)$$

The kinetic energy of the tapered disk is readily calculated for the particular disk described. If the shape is the same then the energy can be written in a form similar to equation (1) and the weight in a form similar to (2). Thus

$$E = 0.405 \frac{\gamma}{g} \omega^2 c b^4 \text{ JOULES} \qquad (7)$$

and

$$W = 2.18 \times 10^{-2}\, \gamma c b^2 \quad \text{KIPS}$$

(8)

We can use (6) - (8) to determine the specific energy which can be stored in a disk of this shape at a given stress level. From

(6)

$$\omega^2 = \frac{\sigma b}{0.212\, \gamma b^2}$$

(9)

so that the specific energy is

$$u = \frac{E}{W} = \frac{0.405 \frac{b}{g} C b^4 \frac{\sigma g}{0.212\, \gamma b^2}}{2.18 \times 10^{-2}\, \gamma c b^2} = 87.8 \frac{\sigma}{\gamma} \quad \text{JOULES/KIP}$$

Taking σ = 30,000 psi we have for steel and aluminum tapered disks,

$$U_{st} = 87.8 \times \frac{30 \times 10^3}{0.283} = 9.3 \text{ MJ/KIP}$$

and

$$U_{al} = 87.8 \times \frac{30 \times 10^3}{0.105} = 25.5 \text{ MJ/KIP}$$

It is clear that the tapered disk is more efficient than the constant thickness disk, 9.3 MJ/KIP vs 7.12 MJ/KIP for the flat disk, but the capacity, for given radius and width is less. As examples of tapered disk designs we consider a disk with a capacity of 50 MJ. The data are shown for both steel and aluminum in Table 2.

	Steel	Aluminum
Energy	50	50
Weight (kips)	5.4	2.0
Volume (cu.ft.)	11	11
Radius (ft.)	2.66	2.66
Thickness (center line)	1.2	1.2
Speed (rpm)	5350	8830

Table 2. Comparison of Steel and Aluminum Disks

Constant Stress Disk

If the disk is to be shaped then one might consider calculating the shape to reduce the stress. Wang[2] gives the expression for thickness as a function of radius so that the stress will be uniform throughout the disk and thus all material utilized to capacity. The expression is

$$h(r) = h_o \exp\left(-\frac{\gamma}{g}\,\omega^2 r^2/2\sigma\right) \tag{10}$$

From this expression it is apparent that, for fixed centerline thickness and outer radius the shape can be adjusted so that any stress level and speed can be attained simultaneously. If we fix the stress level and vary the speed we find that the capacity and efficiency of a given size disk behave as shown in Figure 3. In Figure 3 we see that there is a speed at which the maximum capacity is reached, but that the efficiency continues to increase with speed.

Table 3 shows some data for two constant stress disks. As always the stress level is held at 30 KSI.

Figure 4 shows the profile of the two constant stress disks described in Table 3.

It is clear that by shaping the disk, the stress can be better distributed and higher efficiencies obtained. This is done, however, at the expense of capacity within overall dimensions and also at the expense of increases in speed. Any true optimization must include all factors such as stability, bearing cost, frame cost, etc.