Preliminary Design of a 1 Gigajoule Homopolar Generator

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Abstract— A high-energy, high-voltage homopolar generator has been designed. The HPG will have composite flywheels to maximize energy storage density and a multi-pass armature to achieve high output voltage. The homopolar generator is designed to discharge a constant 895 kA into a 460 V load for several seconds and recharge in less than a minute. The designed energy density is in excess of 15 J/g. Output current control will be achieved by increasing the field coil current proportionally to the decrease in rotational speed.

SYSTEM DESCRIPTION

Fig. 1 shows a homopolar generator (HPG) that will store 1 GJ of energy and deliver 895 kA at a maximum of 460 V. The operating speed of the generator is 5400 rpm. During discharge, the field density is increased to provide voltage matching to a load. The speed at the end of a discharge is 4000 rpm.

The composite flywheels are made of an inner titanium support web with an S-2 fiberglass-epoxy ring over the titanium. A graphite fiber-epoxy ring is fitted over the S-2 glass ring. A containment vessel surrounds the flywheels at each end. The iron core rotor, supported by two stiff hydrostatic bearings, has multiple passes of copper rings in order to attain the output voltage. A thrust bearing is positioned at one end of the HPG to maintain the axial position of the rotor. The iron stator is laminated to reduce the formation of eddy currents and thus increase the flux permeation rate. Four field coils mounted in the stator provide the magnetic flux through the rotor.

Brushes are shown contacting the copper rings underneath the field coils. The rotor is 0.6 m in diameter and 1.8 m long. A steel ring at the both ends of the rotor provides a return path through iron for the 1.8 T field. Two sets of four

Fig. 1. Cross-section of 1 GJ homopolar generator.

armatures are shrunk onto the rotor. A compression force produced by shrinking the armature conductors over the rotor transmits the discharge torque through the rings to the rotor. The insulation between the rings is provided by a tough ceramic film.

The stator surrounds the rotor, field coils, brushes, and the compensating turns. It provides a return path for the magnetic flux through iron. The compensating turns are copper rings which pass the current back to the next armature ring through the brushes. The compensating turns also reduce the magnetic field created by the current passing through the armatures.

The brushes are arranged under the field coils. The brushes make the electrical connection between the armature and the compensating turns. The brush slip speed on the outer armature is 200 m/s before discharge and 150 m/s after discharge. High brush slip speeds have been demonstrated, however, voltage drop across the brush/armature interface increases significantly above 200 m/s. Stability of the brushes on the armature is achieved using trailing arm brushes.

Small holes in the brushes eject cooling water to the brushes and armature immediately after discharge. This process also flushes away the brush-wear debris. The water is scavenged away through the bottom of the rotor cavity.

The brushes are arranged in separately actuated sectors to reduce the possibility of machine failure during discharge. Full current can be drawn by the load in the event of an isolated actuator failure. The sector arrangement also facilitates brush maintenance.

The HPG mass must be as low as possible without compromising the reliability of individual components. Table 1 is a list of the major HPG components and their associated weights. The overall energy density of the HPG is expected to be in excess of 15 J/g. While the volume was also constrained, the composite fly wheels contain most of the energy in a small space. As a result, volume requirements did not present a design restriction. The overall length of the HPG is 4.5 m and overall height is 3m.

ELECTROMECHANICAL ANALYSIS

A magnetic analysis of the HPG was performed using electromagnetic finite element analysis techniques. For the conceptual design, only the circumferential field coil current and eddy currents were analyzed. As a result, the analysis was treated as a two-dimensional axi-symmetric problem. While the radial and axial armature currents are not relevant to the transient magnetic response of the generator, they are relevant to the determination of stress levels within the machine. Therefore, these currents will have to be examined during a detailed design.

Table 1
Estimated Mass of HPG components

HPG Component	Mass (kg)
Steel in rotor	4,273
Cu in Rotor	1,016
Steel in stator	6,800
Cu in stator	503
Brushes	1,063
Field coil	1,205
Ti Flywheels	3,253
Glass Flywheels	4,917
Graphite Flywheels	1,793
Steel Shaft	1,415
End Plates Aluminium	10,807
Containment End plates	6,863
Containment ring	9,209
Bus	1,220
End plate Stiffeners	2,264
Ground support	1,500
Thrust Bearing Brace	300
Total HPG mass	58,401

The aim of the analysis was to determine the controllability of the machine. Accomplishing this aim required the model to include the rotor iron, stator iron, armature sleeves, compensating turns, and field coil windings. Since the problem is axi-symmetric, only one-half of the machine was modelled. The steady-state response is shown in Fig. 2, and the transient response is shown by Fig. 3 [1].

Examination of the steady-state plot reveals that the flux has completely penetrated the iron rotor and stator. As a result, the maximum flux density of 1.8 T has been reached in the air gap. However, the transient response shows that some time is required to reach full flux density.

Fig. 2. Steady state magnetic response

Fig. 3. Transient magnetic response

Eddy currents have been induced in the surrounding conductive materials. According to Lenz's law, these currents are mirrors of the driving current, and as such, they tend to cause magnetic field in opposition to the air-field. The field caused by the eddy currents reduce the air-gap field until a steady-state condition is reached.

Fig. 4 shows how the field in the air gap rises in time. Curve number 1 shows the total flux in the air gap after two seconds. If the formation of eddy currents could be prevented, then the field coil current could rise more quickly. Eddy currents can be reduced by laminating surrounding conductive materials in the direction of eddy-current flow. Curve number two shows the effect of laminating the stator. The rise of useful flux is slightly better, but after two seconds, the total air gap flux is the same as the solid-stator case. When the stator and compensating turns are laminated, the rise is significantly increased, as shown in curve number three. The magnetic analysis indicated that the HPG could be successfully con-

Fig. 4. Transient response of air gap flux

trolled. Since peak air-gap flux can be achieved within the time required for discharge, then the field-coil current can be manipulated to provide peak voltage at the end of the discharge.

The next step in the analysis was to determine the electrical parameters of the machine so that a lumped equivalent circuit could be developed and used in a load simulation. A detailed electromechanical circuit model of the field coil circuit was developed to account for the eddy currents induced in the rotor iron, the armature conductors, the stator, and the end plates. Careful examination of the model and the FEM analysis showed that the end plate losses could be ignored without significant error. The circuit was therefore simplified through traditional transformer analysis so that the stator and rotor circuits were combined into a single eddy-current loop. This simplified model is shown in Fig 5. The circuit equations that describe this circuit are

$$\frac{di_{FC}}{dt} = \frac{V_{FC} - \frac{di_e}{dt} M - I_{FC} R_{FC}}{L_{FC}}$$

$$\frac{di_e}{dt} = \frac{-I_e R_e - \frac{di_{FC}}{dt} M}{L_e}$$

Where

- I_{FC} = current in the field coil
- V_{FC} = driving voltage on the field coil
- L_{FC} = self-inductance of the field coil = $5.95 \mu H$
- R_{FC} = resistance of the field coil = $5.09 \mu \Omega$
- I_e = induced eddy current
- R_e = resistance of the eddy current path = $1.84 \mu \Omega$
- L_e = Inductance of the eddy current path = $3.94 \mu H$
- M = Mutual inductance between the field coil and surrounding conductive material = $4.55 \mu H$

The equations that relate the field coil response to the HPG terminal voltage are

$$\Phi = k_{FC} I_{FC} - k_e I_e$$

$$V_T = 4\omega_a \Phi - I_{HPG} R_{HPG}$$

- Where Φ = Air gap flux
- k_{FC} = Coupling constant = 2.89×10^{-6}
- k_e = Coupling constant = 2.52×10^{-6}
- V_T = HPG terminal voltage
- ω_a = Rotor speed
- I_{HPG} = HPG output current
- R_{HPG} = Resistance of HPG

Fig. 5. Simplified electrical model

The circuit constants were determined from the FEM analysis using standard energy techniques. The response of the HPG calculated with the simplified circuit model is compared to the response calculated with the FEM model in Fig. 6. The figure shows excellent agreement between the two methods, particularly in the case where the initial field coil voltage is set to 1.21 V/turn.

PRELIMINARY COMPONENT ANALYSIS AND DESIGN

Critical speed

A critical-speed study of various machine configurations was performed. Fig. 7 shows six different combinations of the flywheels, the rotor, and the bearings.

The multiple bearing cases are discounted since alignment of more than two bearings is difficult. A coupling between the flywheel and rotor overcomes the alignment problem,

Fig. 6. Comparison of TEXMAP results with simplified model

but the 375,000 Kg-m² torque on the coupling during discharge is higher than the rating of any existing practical couplings. The first critical speed of the three remaining cases was investigated. Fig. 8 shows that the HPG, with the flywheels mounted outside the bearings, has the highest first critical speed. This result is logical since flexibility of a shaft is proportional to the cube of the spanned length.

Configuration 6 was selected because it operates sub-critically and with the best margin of safety. There was careful consideration taken during selection to ensure that assembly, disassembly, and maintenance of the machine would be possible.

Energy Storage Flywheels

Composite materials such as glass-fiber/epoxy and graphite-fiber/epoxy have high strength and low density

Fig. 7. Possible HPG configurations

Fig. 8. HPG critical speeds

compared to metallic materials. Since stress in a rotating flywheel is proportional to the density and the speed squared, low density materials such as composites are ideal for high speed flywheels.

The composite flywheels will be made from a number of concentric rings which have interference fits. The interference fits cause a compression prestress in the rings when assembled. This reduces the hoop stress in the composites when rotating at full speed.

The hubs are not subjected to high stress because the tip speed is lower. A material such as Titanium, which has relatively high strength and low density compared to other metallic materials, is ideal to support the composite outer rings.

Graphite-fiber/epoxy composites have been tested on flywheels with tip speeds up to 1405 m/s without failure. Glass-fiber/epoxy composites have a lower allowable tip speed but store more rotational energy than graphite because the density of glass composite is higher. Tests on glass-fiber/epoxy have shown tip speeds in excess of 600 m/s, however, it is generally considered safe to operate at 600 m/s. The ideal flywheel design will have high density glass rings operating at a safe speed over a supporting hub, and low density graphite rings operating at safe higher speed over the glass rings.

The limited tip speed of 600 m/s for glass fiber composites leads to a maximum diameter of 2.12 m when operating at 5400 rpm. The overall diameter of the flywheels is limited by available space to 2.44 meters. Therefore the maximum tip velocity of the outer graphite composite ring at 5400 RPM is given by

$$V_{\text{maxgraphite}} = w \frac{D}{2} = 690 \text{ m/s}$$

This shows that the graphite is operating well within its demonstrated speed limitations. The graphite rings over the glass rings also help support and lower the rotational stress in the glass. Fig. 9 shows how the flywheel mass and length are significantly reduced when the graphite ring thickness is increased.

Bearing System

Hydrostatic bearings are a proven method for shaft support. Advantages of using hydrostatic bearings as opposed to hydrodynamic or ball and roller bearings for this HPG are:

(1) Low friction

Hydrostatic bearings have lower power loss due to drag than hydrodynamic bearings.

(2) High Damping

Hydrostatic bearings have higher damping coefficients than hydrodynamic bearings and ball and roller bearings.

Fig. 9. Effect of graphite ring

(3) High Stiffness

Hydrostatic bearings have higher stiffness coefficients than hydrodynamic bearings thereby allowing this HPG to maintain sub-critical operation.

(4) Reliable design

Hydrostatic bearings have been used for many years and they are more reliable than ball and roller bearings.

(5) Extremely low wear

Hydrostatic bearings do not have any metal to metal contact, even on start up. Therefore, they wear less than hydrodynamic and ball and roller bearings.

(6) Load capacity at high speeds

Hydrostatic bearings have higher load capacity than ball and roller bearings. Ball and roller bearings cannot be used for the designed HPG because the combined load and speed is beyond their capability.

These factors lead to the decision to use hydrostatic bearings

The equivalent mass supported by each bearing is 12,800 kg. The bearings were designed using a method outlined in reference [2]. The resulting bearing design is shown in Tables 2 and 3.

Table 2
Radial bearing parameters

Number of pockets	14
Shaft Diameter (in)	.3048
Bearing Clearance (in)	.005
Supply Pressure (psi)	3000
Stiffness lb/in	74 x 10 ⁶
Flow Rate (gpm)	90
Pump Horse Power	100
Friction Horse Power	200
Temperature Rise (°F)	65

Table 3
Thrust bearing parameters

Number of pockets	6
Outer Diameter (in)	.3048
Bearing Clearance (in)	.003
Supply Pressure (psi)	3000
Stiffness lb/in	78×10^6
Flow Rate (gpm)	64
Pump Horse Power	112
Friction Horse Power	122
Temperature Rise (°F)	50

Heat Rejection

A major problem encountered during the machine design was the build up of heat within the generator after repeated discharge. Heat is generated by resistive losses from the field coil, the viscous drag within the bearings, brush drag, and resistive heating from the output current. This heat build-up drastically increased the thermal stresses imposed upon the components within the HPG and reduced the predicted mechanical design margins and component life. The stresses imposed on the ceramic and epoxy materials were of particular concern since these materials are quite susceptible to failure resulting from cyclical thermal fatigue. The heat generated within the bearings was of lesser concern because the bearing oil will be externally cooled and will produce no significant heat input to other components.

The greatest input of heat comes from brush drag, bulk resistance heating of the brush assembly, and heating of the interface between the brush and armature surfaces. The largest of these is caused by the voltage drop across the sliding interface between the brushes and the armatures. If the voltage drop across each brush is assumed to be 1 V and there are 16 brush set interfaces, then the total heat input is

$$E_v = V I \Delta t = 35.2 \text{ MJ}$$

The heat produced by brush drag during discharge is 5.2 MJ. Another 6 MJ are generated in the bulk electrical resistance of the brushes. As a result, the total heat input from the brushes alone is 46.4 MJ.

The other major heat producers are the losses in the field coil, the compensating turns and the armature. The heat created by these components is 14.4 MJ for a total heat production within the HPG of approximately 61 MJ.

This excess energy must be removed between discharges in order to maintain the HPG at thermal steady state. Heat removal from the brushes and armature is accomplished immediately after discharge by injecting water through the brush strap and out through holes in each brush. The water cools the brushes and the armature rings. The water is then scavenged away through the bottom of the HPG. Cooling of the field coil and compensating turns is accomplished

by forcing water through tubes running through the stator sections.

Auxiliaries

The main auxiliary systems for the HPG are:

- (1) Bearing oil supply system
- (2) Bearing oil scavenge system
- (3) Brush actuation air system
- (4) Brush cooling water system
- (5) Flywheel vacuum system
- (6) Field coil power supply system

Since the entire generator system is designed for high reliability, each auxiliary system has a redundant backup system. Brush actuation and field coil power are the most critical since their failure would result in an abort of the discharge. Bearing oil supply and scavenge, brush cooling, and flywheel vacuum failure would not affect the discharge but would prevent further discharges. The auxiliaries are powered by an independent power source (IPS) driving a 4KV, 3 phase, synchronous AC generator rated at 2.5 MW

Controls

The HPG control system is separated into three distinct subsystems. The first is an analog loop controller, the sole function of which is control of the field coil voltage and thus the HPG output voltage. The second is a PLC based system controller which controls all the auxiliaries, the IPS, and the HPG driver. A critical function controller (CFC) is the third control subsystem. It features high speed, hard wired, quad redundant logic and provides appropriate instructions for all fault conditions.

CONCLUSIONS

A reliable, high energy, high voltage HPG with fast recharge times can be built with the application of existing technology. FEM and standard modelling techniques allowed performance predictions of the HPG so that the output requirements could be met. Hydrostatic bearings are the best choice for a large rotating machine with high reliability requirements. Composite flywheels increased the energy density of the HPG so that mass and volume may be minimized. A critical-speed study was important to determine the best configuration of the HPG. Redundant auxiliaries and controls will be provided to increase the reliability of the HPG.

REFERENCES

- [1] Magnetic calculations were performed by the Center for Electromechanics, University of Texas at Austin using their proprietary TEXMAP finite element analysis code.
- [2] W. B. Rowe, Hydrostatic and Hybrid Bearing Design, Butterworths, London