DESIGN OF A HOMOPOLAR GENERATOR FOR 400 m/s SLIP RING VELOCITY BRUSH TESTING

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Abstract - Continued development of homopolar generator (HPG) technology by the Center for Electromechanics at the University of Texas at Austin (CEM-UT) will be aided by the installation of a 400 m/s slip ring velocity, controlled atmosphere, high current HPG for brush testing purposes. This HPG will be rated at 4.6 MJ, 36 V, 1 MA and have the capability of testing actively cooled brushes in varying environments. Inductive and resistive loads will be used for output current pulse shaping with an extensive battery of instrumentation and control equipment available for data collection and evaluation.

Operating characteristics, brush testing envelope dimensions, and HPG design parameters are presented. Additional information on component designs and status of fabrication of the HPG is included.

BACKGROUND

During the past five years, the CEM-UT has participated in a national program for the development of technology applicable to electromagnetic (EM) propulsion. The program is sponsored as a joint effort by the U. S. Army Research and Development Command (ARRADCOM) and the Defense Advanced Research Projects Agency (DARPA) of the Department of Defense (DoD). Development of electromagnetic launcher (EML) systems of interest to the DoD includes actively pursuing definitions of EMLs, power supplies, power conditioning equipment, and peripheral hardware required for specific military applications[1]. As a participant in the ARRADCOM/DARPA-sponsored program, CEM-UT is conducting several programs allied to EM propulsion technology, with specific interest placed on the continued development of pulse-power supplies and components to support EMLs in the field.

CEM-UT is in the second phase of a program to develop advanced HPGs and to demonstrate improvements in the state of the art for the machines. During the first phase of the program, an HPG based upon an "all-iron-rotating" (AIR) concept was successfully built and tested. This machine, the compact homopolar generator (CHPG), produced a peak current of 1.02 MA and stored inertial energy of 6.2 MJ (4.6 Mft-lbf). At 4 kJ/kg (1.3 kft-lbf/lbm), the machine demonstrated a significant increase in energy density (ratio of stored energy to machine mass) over previous HPGs[2].

As a parallel effort, a systems tester HPG was designed and built to evaluate next-generation HPG components including copper-finger, current-collecting brushes, stationary-shaft hydrostatic bearings, and a counter-rotating version of the AIR rotor geometry[3]. Presently, the systems tester HPG has a demonstrated energy density of 9 kJ/kg (2.9 kft-lbf/lbm) and is being fitted with a rotor suitable for testing passively and actively cooled current-collecting rim-brushes at sliding velocities up to 300 m/s (984 ft/s).

Energy densities demonstrated by these HPGs were achieved through use of high-strength materials, high-performance brushgear, robust bearings and compact field coils. Efficient utilization of available space was necessary to maximize the amount of rotating mass within the volume of the machine. However, these machines have reached a threshold in the state of the art for the materials and components used in their construction. Because HPG energy densities are most directly improved by increasing the maximum rotor angular velocity, development of new components, and materials capable of withstanding the severe loading conditions at higher rotational velocities are required (terminal voltage is increased proportionally, and stored energy is increased by the square of the rotor angular velocity). Unfortunately, present designs of high current-density, current-collecting brushgear limit advanced HPG performance at the correspondingly increased brush sliding velocities.

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For improved HPG performance, new methods for cooling and aiding current transfer in the brush/rotor interface must be explored. To study the effect of increased sliding velocities on current collecting brushgear, a high speed brush tester (HSBT) HPG has been designed to test new brushgear concepts at slip ring velocities up to 400 m/s (1,310 ft/s). The HSBT was designed with as much flexibility as possible. Brush access, the ability to easily remove, evaluate, and replace test brush sets, was of primary importance in its development. In addition, variability of brush electrical, mechanical, and environmental testing conditions, as well as component reliability, serviceability, and safety were addressed. Finally, a "facility" approach in addition to the HPG design was included to insure that all aspects of a brush testing program would be considered.

HSBT DESIGN OVERVIEW

As stated above, the HSBT is a 4.6 MJ, 1.0 MA, 36 V HPG designed to test actively cooled current collecting brushes at slip ring velocities up to 400 m/s. It has a vertical axis of rotation, integral hydraulic drive system, and radial brush access capabilities so that brushgear may be tested and evaluated with relative ease. The design includes the capability for testing brushes in environments other than standard atmosphere, including low pressure and inert gas atmospheres. A large workspace is provided around the test brushes to install instrumentation and cooling support devices as well as to allow video recording of brush performance during high current transfer conditions. Eight brush-access ports with eight corresponding inner and outer brush sets allow several types of observations to be recorded simultaneously. In addition, the performance of brush sets adjacent to removed brush segments may be observed. Tables 1 and 2 list a number of interesting machine characteristics.

COMPONENT DESIGNS

For the purpose of discussion, a detailed description of the HSBT will be given in sections covering specific major component assemblies within the machine. Major components discussed are;

- Rotor
- Machine Structure
- Output Conductors and Insulation System
- Brush Mechanisms
- Bearings, Seals and Dynamics
- Field Coils
- Drive System.

Underlying design considerations for each component are addressed along with descriptions of their final designs. Please refer to Fig. 1 to view the interrelationship of the components.

Rotor

Many rotor configurations were considered in the selection of the present machine geometry. Various drum, disk, and hybrid rotor shapes were evaluated whereby the hybrid rotor shape shown in Fig. 1 was chosen. It consists of a monolithic design in that the shaft, ferromagnetic rotor, and current carrying slip rings are machined from a single 4340 vacuum-arc-remelted alloy-steel forging. The stepped and tapered shape of the rotor allows higher rotational speeds than a simple disk-shaped rotor of the same diameter and the monolithic design avoids "hollow rotor" stress concentrations. The maximum stress occurs in the rotor at the radius where the inner slip rings join the tapered rotor, and reaches 54 kpsi at 15,000 r/min. Finally, by choosing a rotor with the inner slip rings at one half of the outer slip ring diameter, one may test the outer brush sets while the inner sets act as work-horses at a slip-ring velocity reduced by a factor of two with respect to the outer slip ring.

Three aspects of the selected rotor geometry, in so far as performance of HPGs in general are concerned, should be noted. First, since the rotor is monolithic, the machine output voltage will droop from armature reaction when discharge currents diffuse into the skin of the rotor. Second, there will be a voltage gradient of about four volts across

Table 1. HSBT operational parameters.

<i>Slip Ring :</i>	<i>Outer *</i>	<i>Inner *</i>	<i>Units</i>
Surface Speed.....	400 (1,310)	200 (656)	m/s (ft/s)
Diameter.....	50.8 (20.0)	25.4 (10.0)	cm (in)
Width (each).....	5.1 (2.0)	7.6 (3.0)	cm (in)
Collector Area.....	811 (126)	608 (94.2)	cm ² (in ²)
<i>Testing Envelope Dimensions</i>			
Width.....	8.57 (3.38)	8.89 (3.50)	cm (in)
Inside Diameter.....	50.8 (20.0)	25.4 (10.0)	cm (in)
Outside Diameter.....	72.4 (28.5)	33.0 (13.0)	cm (in)
<i>Rotor</i>			
	<i>Value</i>		
Maximum Speed.....	15,038 (1,575)		r/min (rad/s)
Mass.....	198 (425)		kg (lbm)
Maximum Stored Energy.....	4.6 (3.4)		M J (M ft•lbf)
Total Machine Mass (appx.)	2,954 (6,500)		kg (lbm)

* Values given are for each of one outer and two inner slip rings.

Table 2. HSBT electrical characteristics.

<i>Characteristic</i>	<i>Value</i>	<i>Units</i>
Average Air-Gap Magnetic-Field Strength.....	0.94	Wb/m ²
Open Circuit Voltage.....	36	V
Capacitance.....	7,099	F
Resistance (without brushes).....	2.7×10^{-6}	Ω
Peak Output Current.....	1.0	MA

the inner slip ring (at full field and speed) due to its ferromagnetic properties and proximity to the main flux-conducting path of the generator. Finally, the rotor shape was obviously chosen for brush access, not energy density. It should be stressed that these aspects are a result of the nature of the HSBT and, in the interest of testing brushes, were quite acceptable.

Machine Structure

Stator. Once the rotor shape was chosen, it was necessary to provide a suitable flux path to complete the homopolar geometry of the machine. Placement of other machine components, such as the brush mechanisms, output conductors, bearings, field coils, and instrumentation systems, had to be integrated into the structure of the HPG. Given the desire to have unencumbered brush access, the chosen stator geometry provided many desirable advantages over others considered. The chosen configuration is based on an outside return flux path. It is constructed from a cylindrical steel forging and serves as a framework upon which all component assemblies are supported. Eight flats are machined on the outer circumference of the stator with a rectangular penetration (brush-access port) machined in the center of each. During routine testing of brushes, these penetrations provide access to both inner and outer brush gear without disassembly or disconnection of any other component in the machine.

Port Cover Plates. Environmental isolation and mechanical stiffness is obtained by bolting rectangular plates (port-cover plates) fitted with o-ring face seals over the brush-access ports. Connection of electrical loads to the output terminals inside the stator is made through sealed penetrations in the port-cover plates.

End Plates. Another component of the machine structure is the end plate. One end plate bolts into each end of the stator and provides structural support for the hydrostatic bearings, field coils, and pole pieces. Each end plate is an active part of the flux path and is therefore ferromagnetic. Environmental sealing is maintained with o-rings in the face of the mounting surfaces. Critical alignments of the hydrostatic bearings and pole pieces are assured by accurate cylindrical and flat grinding of the mating surfaces between the stator, end plates, and bearing housings.

Fig. 1. Cutaway view of the HSBT.

Pole Pieces. Last in the support structure are the pole pieces. There are 16 in all, eight mounted to each end plate. Each of the pole pieces are identical 45° segments cut from a steel ring. At their base are male, 22.5°, radially-tapered (with respect to the machine axis) dove tails which are 12.7 mm (0.5 in.) thick. Each pole piece is rigidly mounted and positively aligned by a matching female dove tail machined into the mounting surface of the end plate. This mounting arrangement allows radial removal of the brush gear through the brush-access ports in the stator. Because the radially-tapered dove tails are not self locking, stainless steel retaining clamps are required to secure the pole pieces in their proper radial location. By removing its port-cover plate and pole piece retaining clamp, any set of brush gear may be removed without disturbing the bearings, field coils, drive system, or auxiliary systems.

Output Conductors and Insulation System

Inner Brush Ring. Mounted to each pole piece is an inner brush ring and compensating turn output conductor (IBR/CTC). This component performs several functions. First, it acts as a mounting surface for the inner brush mechanisms and provides structural support as well as electrical contact area to conduct the output current of the machine. The IBR/CTC is machined from a copper forging as a continuous ring to the cross section shown, and cut into 45° segments, each to be mounted to a pole piece. Because the IBR/CTC is segmented, it is rigidly supported from the pole piece with the inner and outer stainless steel torque rings. The IBR/CTC is fastened to the torque rings, through the electrical insulation, using tapered dowels and insulated screws. During the discharge, this IBR/CTC fastening and support method reacts brush drag and Lorenz ($J \times B$) forces developed in the conductors by compression of the insulating substrate.

Outer Brush Ring. A cylindrical surface on the IBR/CTC, at the point where it bolts to the outer torque ring, radially aligns the outer brush ring with respect to the axis of the machine. At this same point, the IBR/CTC makes a transition from an axisymmetric geometry to a discrete, rectangularly cross sectioned output conductor, 5.1-cm² (2.0-in.²) in area. Also, this output conductor has a mounting surface to which the outer brush ring is bolted for support.

Output Terminals. For connection of loads to complete the electric circuit of the machine, the IBR/CTC has a high contact area termination having an effective contact-area ratio of 10:1 with respect to the output conductor cross sectional area. This high contact area terminal is fabricated by silver-brazing seven parallel copper blades (1.6-mm (0.063-in.) thick, dispersion strengthened copper sheet), each spaced one blade thickness from the next, at the end of the output conductor. After brazing, the termination is etched in an acid bath to remove any scale or buildup. Any dimensional interference with a standard for measuring blade spacing is eliminated by additional acid etching steps to open up the spacing. These blades, when interleaved with blades of a similarly terminated conductor, have 13 parallel contact surfaces to conduct output current.

With this termination method, the contact resistance of an output terminal is negligible while connection of resistive and/or inductive loads to the terminal is simple and accomplished in a small volume. For instance, a resistive load may consist of parallel stainless steel sheets interleaved with and electrically shorting the blades of each positive and negative output terminal pair of the machine. Load resistance may then be easily varied by changing the number, width, or length of the stainless steel sheets used. Proper contact pressure in the interfaces between the interleaved blades is then applied by three jack-screws in the terminal clamp rings.

Electrical Insulation System. Electrical insulation from ground, of the IBR/CTC, is provided by a layer of epoxy-impregnated-fiberglass cloth between the IBR/CTC and the pole piece. This insulation system is fabricated by shaping dry fiberglass cloth to the contour of the pole piece surface and then saturating the cloth with epoxy. The premachined IBR/CTC is placed over the wet epoxy/fiberglass substrate and pressed until all excess epoxy is displaced and the brush ring is properly located. The assembly is then held in position until the epoxy cures.

Once curing has completed, the exposed surfaces of the brush ring (primarily the air gap, and inner and outer brushgear mounting surfaces) are machined to their final dimensions while still attached to the pole pieces and end plate to insure concentricity of the brushgear with the rotational axis of the machine. At this time, the individual pole piece/brush

ring assemblies may be separated from one another and the brush rings removed from the pole pieces for final assembly of the brush gear.

Fabrication of the insulation system in this manner insures an interface between the pole piece and brush ring that is structurally strong, electrically isolated, and in intimate contact to support the electromagnetic loads on the conductors.

Brush Mechanisms

Although derived from the CHPG outer brush-mechanism design[2] (Fig. 2a), the HSBT brush mechanism (Fig. 2b) has no circumferential interdependence of adjacent brush gear. Brushgear interdependence was avoided by fabrication of the brush and compensating straps as a single unit and supporting the unit with a variation of the CHPG actuator support cap. As a result, an independent and simplified assembly was obtained. Circumferential independence is the most significant advantage of the new design because no brush mechanisms were lost by segmenting the IBR/CTC.

In this independent design, the pivot-point for the trailing arm brush strap geometry was transferred from a position parallel to the top of the actuator cap to a position parallel to the bottom of the cap. Doing so enhanced the axial and lateral stability of the brush/brush-strap mechanical system by aligning the strap such that brush strap axial frictional forces and lateral $J \times B$ forces were in the plane of the strap rather than at a transverse angle to it. However, repositioning the pivot point required that the new brush strap geometry be supported by an additional member, the brush strap retainer, behind the pivot point. The brush-strap retainer was required to control bending when axial and lateral forces occur during the discharge.

An added benefit of this method of supporting the brush strap was that one area of the new cap could be drilled and serve as a manifold to supply cooling fluid to the individual brush pads. This manifold could then be connected with tubing to each brush pad and/or may be fitted with orifices that directly spray cooling fluid onto the rotor surface.

Finally, with the independent design, tooling for fabrication of the brush straps may be used for both inner and outer brush mechanisms and consequently reduced tooling costs. As a result, both inner and outer mechanisms are identical except for the number of brushes in the set, and is easily adaptable to many slip ring widths and diameters.

Although the HSBT designed includes a full set of inner and outer actively cooled brush mechanisms, the outer set of brush gear may be replaced by any other type which will fit into the available brush testing envelope (see Table 2). Adequate space was allocated around the outer diameter of the outer brush ring so that air cylinders could be used if desired.

Bearings, Seals and Dynamics

Bearing Selection. The duty cycle of a pulsed HPG is largely comprised of accelerating (or decelerating) from one speed to another speed in its operating range. Since the rotor must routinely operate at virtually any rotational speed, it is undesirable to have a rotor-to-stator critical frequency within its operating range. In other words, the machine must operate subcritically. Consequently, subcritical operation plays a major role in the design and selection of the bearings for pulsed HPGs.

Three classifications of bearings were considered for use on the high speed brush tester, the first being rolling-element bearings. Rolling-element bearings were ruled out early on for several reasons. HSBT operational dynamics required shaft diameters sized for stiffness which at 15,000 rpm, resulted in rotational speeds well beyond the theoretical limit of correspondingly sized rolling-element bearings. Also, preloads necessary for subcritical operation would significantly decrease the statistical life of the bearings and increase the possibility of sudden catastrophic failure during operation. Finally, rolling-element bearings are quite susceptible to arc-pitting caused by high magnetic fields and would require magnetic shielding.

Other bearings considered fall under the classification of fluid film bearings. Fluid-film bearings are generally more resistant to damage from arc-pitting and, if designed properly, can have a theoretically infinite life. Two basic types of fluid film bearings, tilting-pad hydrodynamic and

Fig. 2a. CHPG Brush Mechanism.

Fig. 2b. HSBT Brush Mechanism.

externally-pressurized hydrostatic, were studied. The more popular hydrodynamic bearing utilizes a naturally converging fluid film to separate the bearing surfaces and provide load carrying capacity. Hydrostatic bearings, on the other hand, use an external high-pressure lubricant supply to provide the majority of their load carrying capacity and stiffness. Consequently, the load capacities and stiffnesses of hydrostatic bearings are not greatly affected by rotational speed while hydrodynamic bearings require moderate rotational speeds to begin to develop load capacity and stiffness. Because hydrodynamic bearings analyzed did not provide adequate stiffness at low speeds, hydrostatic bearings were chosen for this application.

Radial Bearings. Conventional orifice-compensated radial hydrostatic bearings were designed for rotor support. Table 3 lists the flow and dynamic characteristics of the HSBT hydrostatic bearings.

The calculated performance of the radial bearings looks quite good. Each bearing has a no-load stiffness providing a 56% critical frequency avoidance margin. Their peak stiffness and load capacity provide additional protection against most possible bearing loading conditions. Combined, a load of over 124.8 kN (29,000 lbf) is necessary to

Table 3. Hydrostatic bearing performance characteristics.

Characteristic	Radial*	Thrust*	Units
No-load Stiffness.....	1.1 (6.0)	4.61 (25.6)	G N/m (M lbf/in)
Peak Stiffness.....	1.49 (8.28)	4.61 (25.6)	G N/m (M lbf/in)
Peak Load Capacity.....	64.5 (14.5)	114 (25.7)	k N (k lbf)
Lubricant Temperature Rise.....	18 (32)	21 (37)	°C (°F)
Lubricant Flow Rate.....	61 (16)	140 (37)	l/min (gpm)
Fluid Friction Drag.....	7.5 (10)	38 (51)	k W (hp)
Lubricant Supply Pressure.....	207 (3,000)	207 (3,000)	bar (psi)

* Values given are for each of one thrust and two radial bearings.

make the radial bearings rub. There is a low lubricant temperature rise through the bearings and fluid-friction drag was minimized at 15,000 rpm.

Thrust Bearing. A thrust bearing using a clearance-sensing flow-compensating feature, developed in the systems tester HPG program, was also chosen for this application. A detailed discussion of the thrust bearing design is warranted because of the unusual nature of its operation. Normally, the bearings are the softest "spring" in the machine dynamic system and there are performance trade-offs between stiffness, fluid friction losses, and lubricant supply requirements. Usually in thrust bearings, the performance of one attribute is sacrificed to improve the performance of another. With this in mind and the aid of a clearance-sensing restrictor developed by M. R. Vaughn[4], a new hydrostatic thrust bearing was designed which provided high stiffness, low fluid-friction losses, and low fluid flow requirements.

This bearing utilizes a set of "nested" pockets which are machined into the faces of the thrust-bearing disk and ported together such that they act as "smart" fluidic restrictors. Fig. 3 shows a cutaway view of the thrust-bearing disk with the simplified operation and equivalent hydraulic circuit shown in Fig. 4. Notice in Fig. 4 that the restrictors Rt3 and Rb3 refer to both the inner and outer circumferential lands of the top and bottom faces, respectively. Rt1 and Rb1 correspond to the "smart" restrictors and are located opposite to the pockets they supply through the cross-feed tube restrictors, Rt2 and Rb2. Any reduction in clearance on one side of the bearing reduces flow and pressure to the opposite side thus counteracting the load causing the reduction in clearance.

Lubricant is supplied through a stationary oil feed tube which has a clearance, R leak, with the thrust disk and allows a small amount of leakage into the sump. This design results in a simplified hydraulic network required to supply the bearing with lubricant and eliminates one plate usually required in conventional hydrostatic thrust bearing designs.

Though analysis of the bearing is as of yet incomplete, initial analyses indicate no load stiffnesses on the order of 1.6 times greater than an equivalent geometry, orifice-compensated bearing. Also, rotation of the pockets tends to increase bearing stiffness with increasing speed. To keep fluid-friction losses down, thin lands will be used for all restrictors. In addition, the pockets are deep to minimize the effects of fluid-turbulence losses. Finally, the stationary lands are made of aluminum-oxide ceramic disks, epoxied to the thrust bearing housing plates, to minimize fluid-flow erosion and to electrically insulate the rotor from the stator.

In summary, the overall HSBT hydrostatic bearing performance is quite satisfactory, given the operational speeds and dynamic stiffnesses required for the application. Overall drag for the bearing system will be about 53 kW (71 hp) at 15,500 rpm. The system will require about 260 l/min (69 gpm) of lubricant at 207 bar (3,000 psi) to provide subcritical operation at 15,500 rpm. The simplified hydraulic networking of the thrust-bearing disk will eliminate one bearing plate required in preceeding designs and simplifies oil supply porting in the thrust-bearing housing.

Shaft Seals. Sealing the hydrostatic bearing sump cavities from the environmental chamber where the brushes will be tested was provided for with a triply redundant seal system. This is because at 15,000 rpm, the velocity of the shaft sealing surface, 100 m/s (328 ft/s), prohibits the use of polymeric type static seals and at zero speed, high-speed dynamic shaft seals perform poorly. Consequently, three types of seals are used, two

Fig. 3. Thrust-bearing disk.

Nomenclature:

- Q = fluid flow rate
- P = pressure in the region indicated
- R = fluid flow restriction (fixed or variable)
- t = "top" Bearing surface
- b = "bottom" bearing surface

Fig. 4. Schematic flow and circuit diagrams for the clearance sensing thrust bearing.

dynamic and one static, and these are staged in series with intermediate active sumps inbetween each.

First, an air-retractable seal is used for static and low speed sealing requirements. It will seal for initial testing of the environmental chamber and will remain seated during rotor rotation up to the speed at which the dynamic seals begin to work.

The next seal used is a nitrogen-buffered labyrinth type which has seven stages, the fourth pressurized with dry nitrogen gas. This seal works from zero to 15,000 rpm and forces a positive flow of gas into both the bearing sumps and the rotor chamber. For low pressure brush tests, a vacuum pumping system will be provided which will continuously evacuate the constant flow of nitrogen into the rotor chamber.

Finally, a wind-back dynamic seal is used as the third stage and will pump any fluid flowing into the clearance between the shaft and seal back into the sump cavity.

Dynamic Analysis. Due to the transient nature of the HSBT duty cycle, as stated earlier, subcritical rotor dynamic behavior is desirable. A careful dynamic analysis was performed whereby two dynamic modes were considered, radial and axial.

In the axial mode, a three degree of freedom model was formulated and incorporated the masses of the rotor, stator, and mount along with several other springs connected in series to describe individual machine components.

The rotor/bearing system was modeled with six springs in series and included contributions from the rotor and shaft, thrust bearing disk, thrust disk bolts, hydrostatic fluid film, bearing housing plate, and the housing plate bolts. The bearing film stiffness was calculated with the aid of a computer code written to analyze the new bearing design. The minimum film stiffness required was 3.85×10^9 N/m (2.2×10^7 lbf/in). In addition to these stiffnesses, a negative magnetic stiffness from the field across the air gap was computed as 3.708×10^7 N/m (2.117×10^5 lbf/in). This negative spring rate was subtracted from the equivalent stiffness between the rotor and stator.

An eigenvalue solver was written and used for solving the natural frequencies and mode shapes. The fundamental natural frequency was found at 24,460 rpm, yielding a critical avoidance margin of 1.63 for operation at 15,000 rpm.

Because rotor imbalance may cause heavy excitation of the radial vibration modes, equal care was taken in calculation of the radial component stiffnesses. Like the axial mode model, the radial model consists of three lumped masses connected by a series of springs resulting in a three degree of freedom system. It was interesting to find that the lumped-mass dynamic model assumed for the rotor was within five percent of the value calculated by a finite-element analysis and indicated that the rotor geometry was almost "classically" rigid in behavior. With these inputs and again using an eigenvalue solver, the first critical was predicted at 23,414 rpm. This yielded a critical avoidance margin of 1.56 at 15,000 rpm.

Field Coils

Magnetic field excitation is developed by two field coil assemblies within the machine. Each field coil assembly consists of a four start, 15 turn per start, tape wound conductor configuration. Each start in a coil assembly is positioned at a 90° spacing with respect to the next start and consists of a 64-mm (2.5-in.) wide x 0.8-mm (0.0313-in.) thick copper strips. Combined, each field coil assembly has 60 turns, each having a resistance of 0.00523 Ω. With the starts of the two coil assemblies connected in series, the total resistance of the eight, 15 turn coils is 0.0784 Ω with a drop of 78.4 V, at 1,000 A per turn, resulting in 120,000 A-turns available for field excitation. Heating during a 10 s excitation period will cause a maximum temperature rise of 29°C, assuming adiabatic conditions.

Either external or self excitation of the machine, by connecting the eight individual 15 turn coils in series or parallel respectively, is possible. External excitation is to be provided via an existing 1,200 A, 100 V dc, rectified power supply connected to terminals penetrating the end plate.

Self excitation of the machine will require a set of electrically isolated brushes, in addition to the main current carrying brushes. This brush set would be independently actuated to provide excitation current initiated by residual magnetism in the HPG back iron.

Drive System

Given the rotor speeds at which this machine will be operated and the frictional losses associated with the hydrostatic bearings, the design of a drive system to accelerate the rotor to full speed in a reasonable amount of time was a nontrivial exercise. A two stage, speed increasing, 4.84 to 1, parallel shaft gear box (designed by R. Cushman and Associates, Inc.) connected to a Rexroth AA6VM 250 HA 250 cc/rev, variable displacement hydraulic motor, was chosen as the drive train for this application. By designing a gear box that would directly mount to the generator bearing housing and, in turn, have the hydraulic motor directly mounted to it, a considerable reduction in the space required for the system would be realized and alignment of the devices would be simplified. The major design problem, with the drive train directly coupled to the HPG rotor, was that it must be designed to follow the rotational transients associated with the HPG discharge conditions. These transients could include rotor deceleration rates in excess of $21,000 \text{ rad/sec}^2$ at a short-circuit fault current of 3 MA and full field strength. During these transients, deceleration of the drive train component inertias must not impart torques which would exceed the torque transmitting capabilities of any part of the system.

Compared to a single stage parallel shaft gear box of the same speed ratio, a two stage gear box has an apparent output shaft inertia less than one sixteenth of the single stage box and less than the inertia of the hydraulic motor alone. The two-stage gear box will transmit 1,630 N·m (1,200 ft·lb) input torque from the 250 cm³/rev (15.2 in³/rev), 414 bar (6,000 psi) hydraulic motor and develop an output torque, at 4.84 to 1 speed ratio, of 337 N·m (248 ft·lb). Total transmissible power for the hydraulic motor, at maximum speed, displacement, and pressure, will be in excess of 448 kW (600 hp) but the system will be limited to 298 kW (400 hp) by the laboratory hydraulic utility at its full flow of 417 l/min (110 gpm) at an output pressure of 414 bar (6,000 psi).

Power transmission by the drive train will follow a piece-wise continuous curve because of the variable displacement control of both the pump and hydraulic motor. With displacement control on both units, the drive train can transmit energy to the rotor faster than a constant torque, fixed displacement hydraulic motor drive, with the same peak output power. Finally, this drive system will require approximately 35 s to accelerate the rotor from zero to 15,000 r/min.

CONCLUSIONS

In the design of the HSBT, many obstacles had to be overcome in order to reach the design goal of a 400 m/s slip ring velocity. As a result, new concepts for improving HPG performance were developed. Some of the concepts developed include;

- easy radial access for service and testing of brushgear
- a fully sealed brush testing chamber allowing tests in many environments
- quick-disconnect, high contact area output terminals
- independently mounted, enhanced stability, actively cooled brush mechanisms
- a clearance sensing, hydrostatic thrust bearing which simplifies bearing fabrication and increases bearing stiffness
- a shaft seal that works at zero and low shaft speeds yet may be retracted when the dynamic shaft seals begin to work at high shaft speeds
- a compact, direct connected, 15,000 rpm, 448 kW (600 hp), drive train.

Because of these developments, the HSBT will be a rugged, reliable, easy to use machine serving as a valuable test bed for advancing the state of the art in high current carrying brush research.

At the time of this writing, much of the raw material for construction of the machine has been purchased. In addition, the hydraulic motor has been received, and the two stage gear box has been designed and is being fabricated. Assembly of the machine should be completed and brush testing begun sometime in the fourth quarter of 1986.

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REFERENCES

- [1] H. D. Fair, Jr., "Electromagnetic propulsion: a new initiative," in IEEE Transactions on Magnetics, Vol. MAG-18, no. 1, Jan. 1982, pp. 4-6.
- [2] J. H. Gully, et al., "Design of a Compact, Lightweight Pulsed Homopolar Generator Power Supply," IEEE International Pulsed Power Conference, 3rd, Albuquerque, New Mexico, June 1-3, 1981.
- [3] J. H. Gully, et al., "Compact Homopolar Generator Development at CEM-UT," IEEE Symposium on Electromagnetic Launch Technology, 2nd, Boston Massachusetts, October 11-14, 1983.
- [4] M. R. Vaughn, "A Clearance Sensing Hydrostatic Bearing Restrictor for the Homopolar Generator Systems Tester," PhD dissertation, The University of Texas at Austin, Austin Texas, August, 1985.