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# G2 Flywheel Module Design

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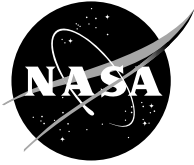
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Kirsten Duffy performed the rotordynamics analysis and Peter Kascak worked on the motor specification. Dan Clark performed the rotor windage loss calculations. Ronald Storozuk, Analex Corporation, did the solid models and the detailed drawings. The NASA team includes Kevin Konno, Jeffery Trudell, Walter Santiago, Ramon Lebron, and Barbara Kenny. Kevin Konno led the rotor design effort as well as the module fabrication and assembly team. Jeff Trudell did thermal and stress analysis. Walter Santiago and Ramon Lebron specified the avionics, sensors, and connectors. Barbara Kenny also worked on the motor specification. Aleksandr Nagorny working as a GRC research associate estimated the motor losses. The VCEL team lead by Alan Palazzolo at Texas A&M designed the magnetic bearing. Andrew Kenny performed the magnetic analysis. Jason Preuss did the solid modeling and detailed drawings of the magnetic bearings.

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Design of a flywheel module, designated the G2 module, is described. The G2 flywheel is a 60,000 rpm, 525 W-hr, 1 kW system designed for a laboratory environment; it will be used for component testing and system demonstrations, with the goal of applying flywheels to aerospace energy storage and integrated power and attitude control (IPACS) applications. G2 has a modular design, which allows for new motors, magnetic bearings, touchdown bearings, and rotors to be installed without a complete redesign of the system. This design process involves several engineering disciplines, and requirements are developed for the speed, energy storage, power level, and operating environment. The G2 rotor system consists of a multilayer carbon fiber rim with a titanium hub on which the other components mount, and rotordynamics analysis is conducted to ensure rigid and flexible rotor modes are controllable or outside of the operating speed range. Magnetic bearings are sized using 1-D magnetic circuit analysis and refined using 3-D finite element analysis. The G2 magnetic bearing system was designed by Texas A&M and has redundancy which allows derated operation after the loss of some components, and an existing liquid cooled two pole permanent magnet motor/generator is used. The touchdown bearing system is designed with a squeeze film damper system allowing spin down from full operating speed in case of a magnetic bearing failure. The G2 flywheel will enable module level demonstrations of component technology, and will be a key building block in system level attitude control and IPACS demonstrations.

## Nomenclature

<i>CMB</i>	Combination Magnetic Bearing
<i>RMB</i>	Radial Magnetic Bearing
<i>TD</i>	Touchdown Bearing
<i>M/G</i>	Motor/Generator
<i>GRC</i>	Glenn Research Center
<i>ISS</i>	International Space Station

## I. Introduction

NASA Glenn Research Center (GRC) has an ongoing effort in flywheel technology development and deployment for spacecraft applications (ref. 1). Flywheel systems can be used to replace batteries for energy storage applications. Flywheel modules can also be deployed in an array which provides both energy storage and momentum control; this type of system is called an Integrated Power and Attitude Control System (IPACS) system. A flywheel system consists of a number of flywheel modules and an electronics package which operates the flywheel motor/generators, magnetic bearings, and telemetry. The benefits of flywheel systems for energy storage applications are high energy density, high power density, long life, deep depth of discharge, and broad operating temperature ranges. In an IPACS configuration an additional mass savings can be achieved through the combination of the energy storage and the attitude control functions.

Flywheel modules for space use are designed to maximize energy density and minimize losses. Typically the energy storage component of the module is a rim composed of high strength carbon fiber. Energy is transferred to and from the wheel using a motor/generator. The flywheel module typically has some or all of these ancillary components: magnetic bearings, touchdown bearings, housing structure, sensors, connectors, and wiring harnesses. The rotating components are placed along a hub with the rim in the center of the hub. The motor, magnetic bearings, and touchdown bearings populate each end of the hub, and the stationary parts of these components are located within the housing.

This paper describes the design and analysis of the G2 flywheel module at GRC. A schematic of the G2 is shown in figure 1. The module subsystems considered in this paper include the rotor, magnetic bearings, motor/generator, touchdown bearings, and the housing.

The rotor consists of three main parts: the rim, the motor rotor, and the hub. The rim is the main energy storage component; it is made of a multi-ring carbon fiber. The motor rotor is a two pole permanent magnet synchronous motor, which was purchased from a commercial vender. The rim and motor rotor are mounted on the hub, which is made of titanium.

The magnetic bearings are used for low-loss suspension of the rotor.

The motor/generator is used to store and retrieve energy from the rotor.

Touchdown bearings are designed to capture the rotor in the event of a magnetic bearing failure, and also to support the rotor when the active magnetic bearings are turned off.

The module housing supports the static portions of the flywheel system, such as the motor/generator and magnetic bearing stators, the position sensors, the touchdown bearings etc. Additionally, the housing is sealed, allowing vacuum testing of the module in the laboratory.

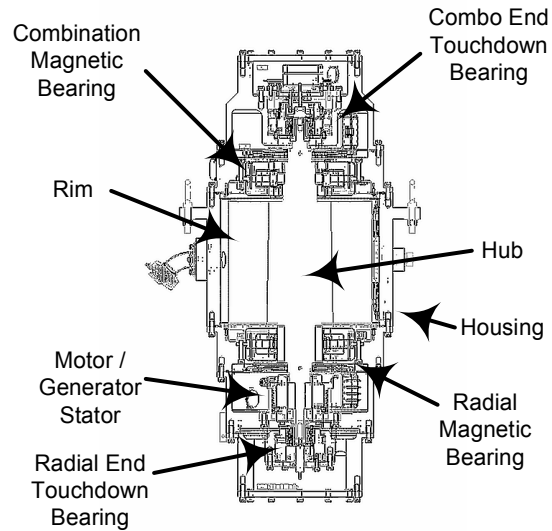


Figure 1.—G2 module drawing.

## II. G2 Requirements

The G2 Flywheel Module requirements were derived from the milestones of the NASA Aerospace Flywheel Technology Program (AFTP). The G2 flywheel module will be one of two modules used in a single axis integrated power and attitude control (IPACS) demonstration at NASA Glenn Research Center. The AFTP Program also develops component technologies that need to be tested at the flywheel module level. G2 is modularly designed to allow new components to be introduced without a complete redesign.

Performance requirements were set at the module level and propagated to the component levels. The total parts cost per module was minimized, and the design schedule was limited to 10 months.

Several power and energy requirements were imposed. The usable stored energy shall be greater than 300 W-hr with a charge-discharge rating greater than 1 kW, and the power bus voltage was set at 120 V. The module shall be capable of simulating Low Earth Orbit (LEO) cycles with a 90 minute nominal charge/discharge time at 90 percent depth of discharge (DOD).

G2 size and environmental requirements were set for a laboratory environment. Weight is not a significant factor but was limited to 250 pounds, and the size was limited to 20 inches in diameter by 36 inches long. A goal was set to minimize both of these quantities within the budget constraints. The operating temperature will be between 60 and 120 °F, and the G2 is required to operate in any orientation so that it can be applied to future multi-axis IPACS experiments.

A summary of the G2 requirements is included in table 1. The G2 design meets or exceeds all of the requirements.

TABLE 1.—G2 REQUIREMENTS SUMMARY

Item	Requirement	Design
Usable energy	>350 W-hr	525 W-hr
Charge/discharge power rating	>1 kW	1 kW
Electrical power bus voltage	<120V	120V
Depth of discharge	90%	90%
Weight	<250 lbs	218 lbs.
Volume	20 in. Ø x 36 in.	12 in. Ø x 30 in.
Capable of operation in any orientation	Yes	Yes

### III. G2 Rotor Design

The function of the rotor is to store energy. The G2 rotor consists of a carbon fiber rim for energy storage and a central titanium hub, on which the bearing and motor parts mount. The carbon fiber rim is a multilayer press fit design, and it is located at the axial center of the hub. The rotors of the magnetic bearings located on each side of the rim, and the radial position sensor arrays are located outboard of each magnetic bearing rotor. The motor/generator is located further down the hub, outboard of the radial magnetic bearing, and the touchdown bearings are located at each end of the shaft. An axial position sensor and angle resolver are located at the combination touchdown bearing end of the rotor.

#### A. Rotordynamics Requirements

Important rotor requirements include the total and usable kinetic energy, the 1st bending mode frequency, the polar to transverse moment of inertia ratio ( $I_p/I_t$ ), and the rotor weight.

The G2 rotor has been designed to store a total of 591 W-hr of energy, with 525 W-hr usable energy; this exceeds the requirement of 350 W-hr usable energy.

TABLE 2.—G2 ROTOR REQUIREMENTS

Item	Requirement	Design
Total kinetic energy		591 W-hr
Usable kinetic energy	350 W-hr	525 W-hr
1st Bending Mode	>72,000 rpm	71,900 rpm
Polar to transverse moment of inertia ( $I_p/I_t$ )	<0.8 or >1.2	0.5
Rotor weight	<75 lbs	50 lbs

The bending mode frequency for this type of rotor should exceed the maximum operating speed by 20 percent; the estimated bending frequency is 71,900 rpm is equal to the requirement of 72,000 rpm given the margin of error.

The  $I_p/I_t$  ratio must not be near unity; the G2 rotor  $I_p/I_t$  ratio is 0.5, providing greater margin than any of the previous GRC flywheels, and the rotor weight will be 50 lbs which is under the maximum requirement of 75 lbs.

A summary of the G2 rotor requirements is included in table 2. The G2 rotor design meets all of the requirements.

#### B. Rotordynamics Analysis

Rotordynamics analysis is central to the flywheel module design process; the rotor is designed with control in mind. The modes of the rotor can be divided into two categories: rigid body modes, and bending modes. Both sets of modes are considered in the design process.

Rigid body modes describe rotordynamic behavior of the rotor independent of rotor bending. The important radial rigid body modes for the G2 design are the *bounce* and *tilt* modes, and the relevant bending modes are the first and second rotor bending modes (fig. 2). The bounce mode describes translation of the center of mass of the rotor (A), while rotation about the center of mass is called the tilt mode (B). Although flexure of the rotor is decomposed into many bending modes, typically only the 1st (C) and 2nd (D) bending modes are low enough in frequency to be excited during flywheel operation. The bending mode frequencies are a key driver for the rotor design, because magnetic bearing control becomes significantly more difficult if the bending mode frequencies fall within the rotor operating speed.

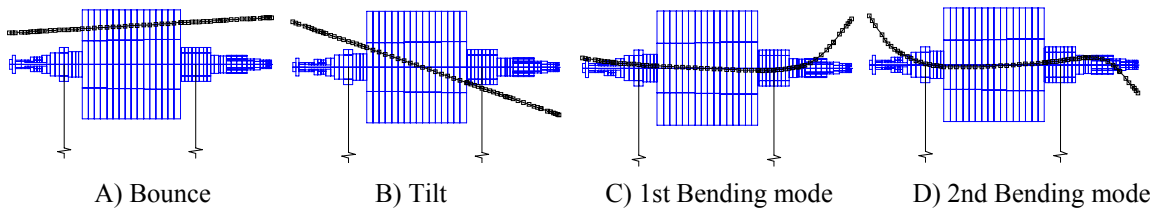
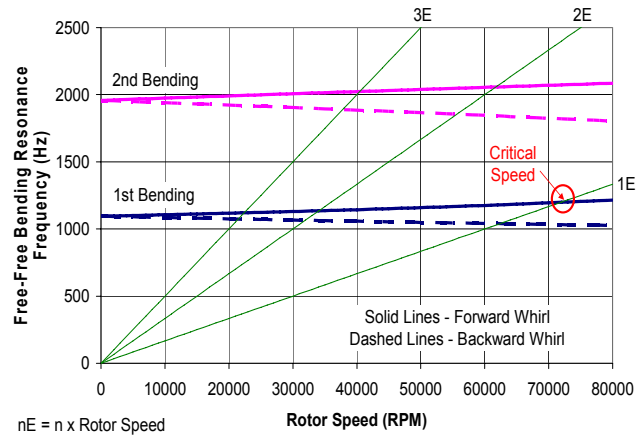


Figure 2.—Rotor modes.

In order to understand rotor bending modes, rotor design iterations are conducted using free-free bending mode analysis. Critical speeds are analyzed for operation on the magnetic bearings and the mechanical touchdown bearings, and critical speed maps, plotting mode frequencies versus bearing stiffnesses, are made for each mode. Since the maximum operating speed for the G2 module is 60,000 rpm (1000 Hz), the bending modes are designed to be above 1000 Hz throughout the operating range. Figure 3 shows the modes of the final design, showing the 1st and

2nd bending modes, and the synchronous rotor speed (1E) and its harmonics (2E and 3E). The 1st and 2nd bending mode frequencies are at 1097 and 1961 Hz with the rotor at rest, and the bending modes split with speed into forward and backward components. The 1st bending mode forward and backward frequencies are 1180 and 1042 Hz at 60,000 rpm, and the 2nd bending mode forward and backward frequencies are 2057 and 1850 Hz at 60,000 rpm. Note that our goal has been achieved; the bending modes are above 1000 Hz over the entire operating range. Note that although the 1st backward bending mode comes close to 1000 Hz at full speed, it should not pose a problem - the forward components are of greater concern because they can be excited by the rotor unbalance, while the backward components are much more difficult to excite.



**Figure 3.—Predicted G2 bending modes versus rotor speed.**

The rigid body mode frequencies are controlled by the rotor bearing stiffnesses; since we operate on active magnetic bearings, the bearing stiffnesses are set by the controller gains. Typically both bounce and tilt modes are set below 100 Hz at rest. The bounce mode frequency remains fixed throughout the speed range, while the tilt mode splits into a forward whirl and backward whirl mode. The backward whirl mode frequency decreases with speed, while the forward whirl mode frequency ideally asymptotes to the ratio of the polar and transverse moment of inertia of the rotor. Actual placement of the rigid body modes is chosen by trading off several design criteria, such as desired 1E/forward whirl crossover, bounce mode stiffness, rigid body mode separation.

### C. Thermal Analysis

Thermal analysis must be performed on the rotor system to verify that the flywheel module will be capable of continuous operation, without overheating. Because the rotor is levitated and operated under vacuum, radiation is the exclusive heat transfer path for the rotor.

In order to complete this thermal analysis, rotor losses are predicted for the magnetic bearings, the motor/generator, and the effects of windage.

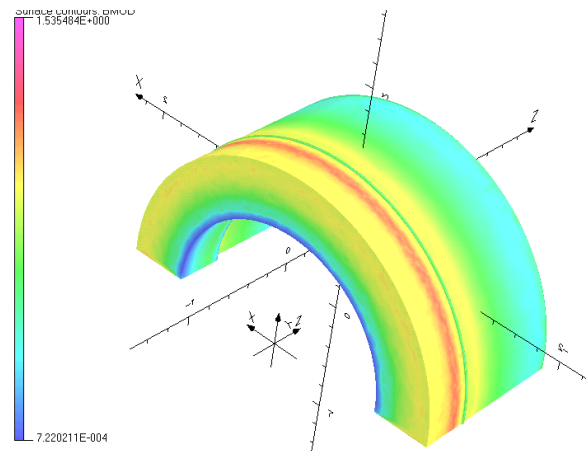
#### 1. Magnetic Bearing Spin Losses

Losses due to the magnetic bearing operation are called magnetic bearing spin losses. These losses include control flux, permanent magnet (PM) bias flux, and amplifier switching frequency losses.

Although the magnetic bearings are capable of providing significant forces to levitate the rotor, under typical operating conditions they are operated near a zero force level. Because of this, bearing control flux is neglected in the loss analysis.

Pulse width modulated (PWM) amplifiers are used to generate the currents for the magnetic bearings. The currents generated at the switch frequencies (typically 20 to 30 kHz) constitute another possible loss source. However, this current component is greatly reduced via power filters before reaching the magnetic bearing actuators, so the switching related power losses in the bearings can also be neglected.

Since the control flux and switching frequency losses can be neglected, magnetic bearing spin loss estimate is calculated with only the PM bias field



**Figure 4.—Radial MB rotor lamination PM spin losses.**



applied. Magnetic bearing eddy current and hysteresis loss estimates for the rotor components due to the PMs are made using 3D FEA (see fig. 4); the estimated bias losses are less than 1W at full speed.

### 2. Motor/Generator Spin Losses

The motor/generator was purchased from an outside vendor. The proprietary nature of the design made it difficult to predict the losses. The full power efficiency was predicted at 94 to 95 percent. The rotor losses were not predicted.

### 3. Windage Loss Analysis

Windage loss estimates were made using the method presented by Liu (ref. 2). The free-molecule wall shear stress equation with slip velocity correction is the basis for the calculations; for our analysis, it is converted into a windage power equation for a cylindrical surface (eq. 1). This equation estimates the windage power loss on the outer surface of the carbon fiber rim section; losses on the faces of the rim and the hub surfaces were neglected in this analysis.

$$P_{W,C} = P(\sqrt{2\pi/RT})L\omega^2a^3((2\lambda/L)/(1+2\lambda/L)) \quad (1)$$

Power loss for decade isobars between 1e-5 and 10 mTorr are shown in figure 5. A turbo molecular vacuum pump will be attached to the flywheel module on the central housing section which surrounds the rim, and the vacuum level is anticipated to be better than  $10^{-3}$  mTorr. Based on the calculations in figure 5, the windage power loss at this vacuum level is expected to be less than 1 W at full operating speed.

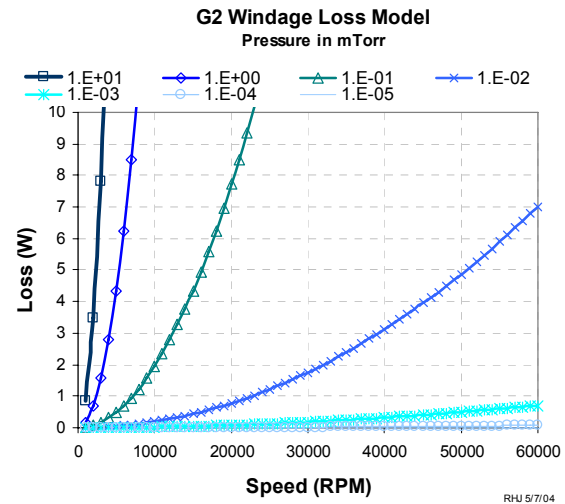


Figure 5.—G2 windage loss estimate.

### 4. 1-D Thermal Model

In order to understand the maximum allowable heat load due to losses, a one dimensional (1-D) thermal model was made to estimate the temperature rise as a function of total losses on the rotor. This model allows input of a total power loss value and an effective emissivity of the system; in the model, the total losses are distributed between the loss mechanisms described above. The model was run for multiple total loss values, and over multiple effective emissivities, and the results are presented in figure 6.

The epoxy resin used in the G2 rim has a glass transition temperature of 180 °F; this is the thermally limiting component in the flywheel, and this upper limit is marked in red. Since the estimated effective emissivity of the G2 is estimated to be about 0.7, the maximum allowable power losses to limit rotor rim temperature to 180 °F are estimated to be about 45 W.

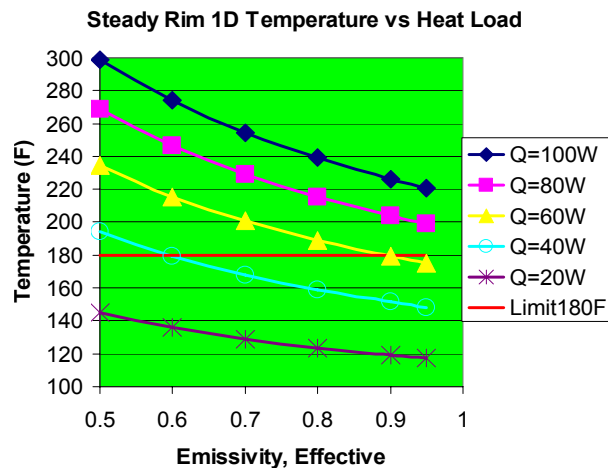


Figure 6.—G2 1-D thermal model.

#### IV. Magnetic Bearing System

The G2 flywheel utilizes an active magnetic bearing (MB) system, consisting of two magnetic bearings which provide five-axis levitation. A radial magnetic bearing (RMB) at the motor end of the shaft controls two radial degrees of freedom (X1 and Y1), while the combination magnetic bearing (CMB, located at the end opposite the motor) controls two radial degrees of freedom as well as the axial degree of freedom (X2, Y2, and Z). A simplified diagram of the feedback control system is shown in. The MB controller issues current commands to a pulse width modulated (PWM) amplifier; the PWM drives current into the magnetic bearing actuator, which in turn produces forces which levitate the rotor. The rotor position is sensed using non-contact eddy current sensors; this sensor signal is conditioned and fed back into the controller.

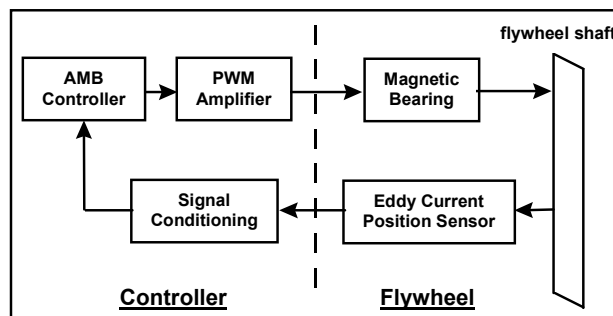


Figure 7.—G2 MB controller schematic.

##### A. Magnetic Bearing Requirements

Requirements were developed for the load, stiffness, electrical and dimensional attributes of the MB actuators.

The load specifications require that both MBs to be able to support 30 lbs radially at 1500 Hz. Both MBs beat this specification, with 40 lbs. (RMB) and 44 lbs. (CMB) capacity. Additionally, the bearings are required to support three times the rotor weight at 0 rpm; since the rotor weighs 50 lbs, this means that at 0 rpm, both the RMB and the CMB must provide 75 lbs. of force, each, radially, and the CMB must also provide 150 lbs. of force axially. The CMB bearing meets the DC load requirements, but the RMB DC load capacity is slightly below the specification (70 vs. 75 lbs), although this is not expected to be a problem.

Current and position stiffness are actuator parameters that have a significant impact on the magnetic bearing control. The minimum current stiffnesses are 7.5 lbs/A radially and 15 lbs/A axially, and the designed current stiffnesses exceed these requirements, with 8.8 lbs/A and 9.3 lbs/A radially for the two MBs, and 18.8 lbs/A axially. The design objective was to minimize position stiffness; the resultant position stiffnesses are 6160 lbs/in and 6890 lbs/in radially for the RMB and CMB, and 7910 lbs/in axially.

The electrical specifications are a result of trades between power electronics and magnetic bearing actuator issues. The bus voltage was limited to 110V and the current to 10 A, and the designed hardware operates within these limits. A minimum bandwidth requirement of 1500 Hz was established to minimize phase lag in the control system, and the RMB bandwidth is 2200 Hz and the CMB bandwidth is 2000 Hz.

The dimensional requirements reflect rotordynamics issues and the need to fit with the other components of the flywheel. The lamination ID is maximized and the overall bearing length is minimized to increase the rotor bending mode frequencies, while the lamination OD is limited by the strength of the magnetic materials.

TABLE 3.—MAGNETIC BEARING REQUIREMENTS

Item	Requirement	Design
<i>Load</i>		
RMB – DC Radial Load Capacity	75 lbs	70 lbs
RMB – 1500 Hz Load Capacity	30 lbs	40 lbs
CMB – DC Radial Load Capacity	75 lbs	74 lbs
DC Axial Load Capacity	150 lbs	150 lbs
CMB – 1500 Hz Radial Load Capacity	30 lbs	44 lbs
<i>Stiffness</i>		
Current Stiffness – RMB – Radial	7.5 lbs/A	8.8 lbs/A
CMB – Radial	7.5 lbs/A	9.3 lbs/A
CMB – Axial	15 lbs/A	18.8 lbs/A
Position Stiffness – RMB – Radial	minimize	6160 lbs/in
CMB – Radial	minimize	6890 lbs/in
CMB – Axial	minimize	7910 lbs/in
<i>Electrical</i>		
Voltage	110V max	102V
Current	10A max	10/8 A
Bandwidth – RMB – Radial	1500 Hz	2200 Hz
CMB – Radial	1500 Hz	2000 Hz
<i>Dimensional</i>		
Rotor lamination ID – RMB	2 in.	2.0 in.
CMB	2 in.	2.0 in.
Rotor lamination OD – RMB	3 in.	2.9 in.
CMB	3 in.	2.9 in.
Bearing Length – RMB	2.5 in.	2.07 in
CMB	2.5 in.	2.28 in

The G2 MB requirements are summarized in table 3. Both magnetic bearing actuators exceed the specifications in almost all aspects; the small load capacity exception in the RMB is not expected to have any performance impact.

### B. Magnetic Bearing Actuator

Both the combination and the radial magnetic bearings were designed for the G2 module; a drawing of the combo MB is presented in figure 8. Two key requirements for the MB design were low power consumption and load capacity.

In order to minimize losses, a PM bias, homopolar design was selected for the radial and combo. PM bias was selected because this approach eliminates the need for bias current in the MB, preventing the associated  $I^2 R$  losses. The homopolar bearing design was selected because, in this design configuration, the magnetic field variation in the rotor laminations due to the bias flux is minimized; this is desirable, as these field variations cause hysteresis and eddy current losses. Finite element methods were used to optimize the stator poles to further reduce the flux variation.

The design target for the MB static load capacity was three times the rotor weight of 50 lbs, or 150 lbs total. The radial portion of both final design bearings each have a capacity of 75 lbs, meeting the total 150 lb requirement, while the axial portion of the combo bearing also provides 150 lbs capacity.

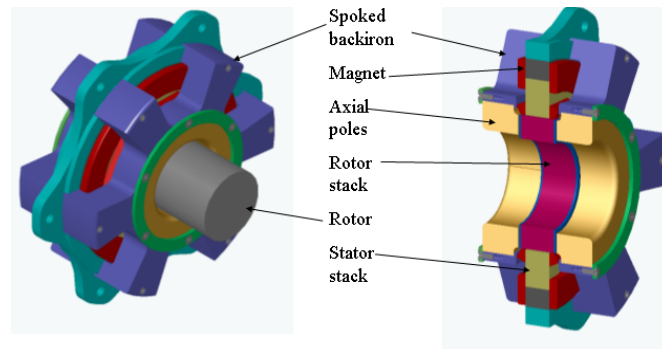


Figure 8.—G2 MB design.

### C. Magnetic Bearing Power and Control System

A detailed schematic of the magnetic bearing power and control system is shown in figure 9. The major components of the MB control system include the position sensors, sensor signal conditioning, MB controller, and PWM/power filters.

Non-contact eddy current sensors are used to locate the rotor in the G2 unit. Ten sensors are used in all: two sets of four radial sensors are located above and below the rim, near the magnetic bearings, and two additional sensors, which measure axial position and angular reference, are located at the top of the unit. Each set of radial sensors is run off of common oscillators, to avoid beat frequency crosstalk noise which can arise due to differences in sensor oscillator drive frequencies (ref. 3).

One key portion of the sensor signal conditioning is the anti-alias filters, which are first order 8 kHz low pass filters. These filters prevent aliasing in the digital controller. The filter corners were selected to efficiently eliminate potential noise while minimizing phase lag.

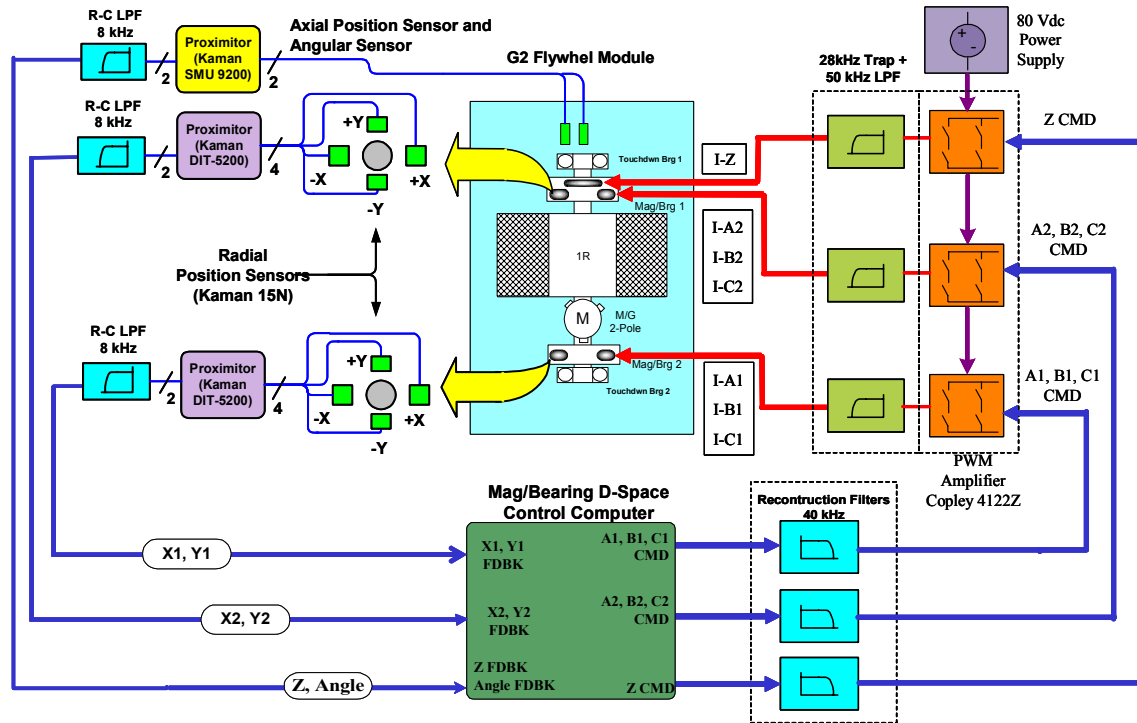


Figure 9.—G2 MB power and control system.

The MB controller is written in a rapid development software environment, then compiled and run on a digital controller which includes A/D and digital inputs, a CPU, and D/A outputs. The user interface for the controller is implemented on a PC (ref. 4). The MB control algorithm issues seven current commands based on the position inputs. Each magnetic bearing actuator has three radial control axes (A,B,C) instead of the traditional two axes (X,Y) for redundancy, and the combination magnetic bearing has an axial (Z) control axis. First order 40 kHz low pass filters are used for reconstruction of the D/A signal; this corner frequency was also selected to reduce noise while eliminating phase lag.

Seven PWM amplifiers produce current proportional to the commands. The seven amplifiers have synchronous switching frequencies to reduce amplifier to amplifier beating noise picked up by the sensors<sup>5</sup>, and they operate on a 120 V Bus. The current output from the PWMs is filtered using a 50 kHz low pass filter and a trap filter at the PWM switching frequency (ref. 6). At the actuator the currents produce magnetic fields resulting in forces on the levitated rotor.

## V. Motor/Generator System

A permanent magnet motor/generator (M/G) system is used to transfer energy to and from the flywheel rotor. Flywheels designed at NASA Glenn Research Center are intended for use in Low Earth Orbit (LEO), thus, they will typically be either charging or discharging, and will have only short periods where there will be no power transfer. For this type of load profile, permanent magnet motors provide the highest efficiency and the highest power density. Other types of motors would be more appropriate for applications which required short high-power transfer periods and long hold periods.

### A. Motor/Generator Requirements

The M/G requirements stem from the need to charge and discharge the flywheel over a 90 minute LEO orbit. The flywheel is intended to operate between 20,000 and 60,000 rpm; the motor power requirement is 1 kW at 20,000 rpm and 3 kW at 60,000 rpm. The number of motor poles is minimized to reduce the frequency requirements of the power electronics and controls; the selected motor has two poles, which is the minimum possible number. The motor voltage is selected to work with the ISS bus and future satellites, and the design value of 78 V meets the requirement of  $72 \text{ V} \pm 10$  percent. A total harmonic distortion (THD) <5 percent is required to minimize torque ripple, and the selected motor's THD is less than 3 percent. The physical dimensions of the motor are limited by the other parts of the system; the motor has a rotor outer diameter (OD) or 1.5 in., and the stator OD is 3.0 in. and it is 3.1 in. long. The G2 motor requirements are summarized in table 4. The motor meets all of the specifications.

TABLE 4.—G2 MOTOR/GENERATOR REQUIREMENTS

Item	Requirement	Design
Power @ 20,000 rpm	1 kW	1 kW
Number of poles	$\leq 4$	2
Back EMF Voltage	$72 \text{ V} \pm 10\%$	78
Sinusoidal Back EMF, THD	< 5%	$\approx 3\%$
Dimensions Rotor OD	< 2.9 in.	1.5 in.
Stator OD	< 8 in.	3.0 in.
Stator Length	<4 in.	3.1 in.

### B. Motor/Generator Design

The M/G was designed by a subcontractor; it is a two pole permanent magnet motor rated at 3 kW at 60,000 rpm, and it can produce 1 kW at 20,000 rpm. The line to line back EMF voltage was measured on a sister unit at 78 V with less than 3 percent THD. The physical dimensions of the motor are appropriate for this application, and the motor full power efficiency was estimated at 94 to 95 percent.

### C. Motor/Generator Power and Control System

Figure 10 is a schematic of the power and control system for the motor/generator.

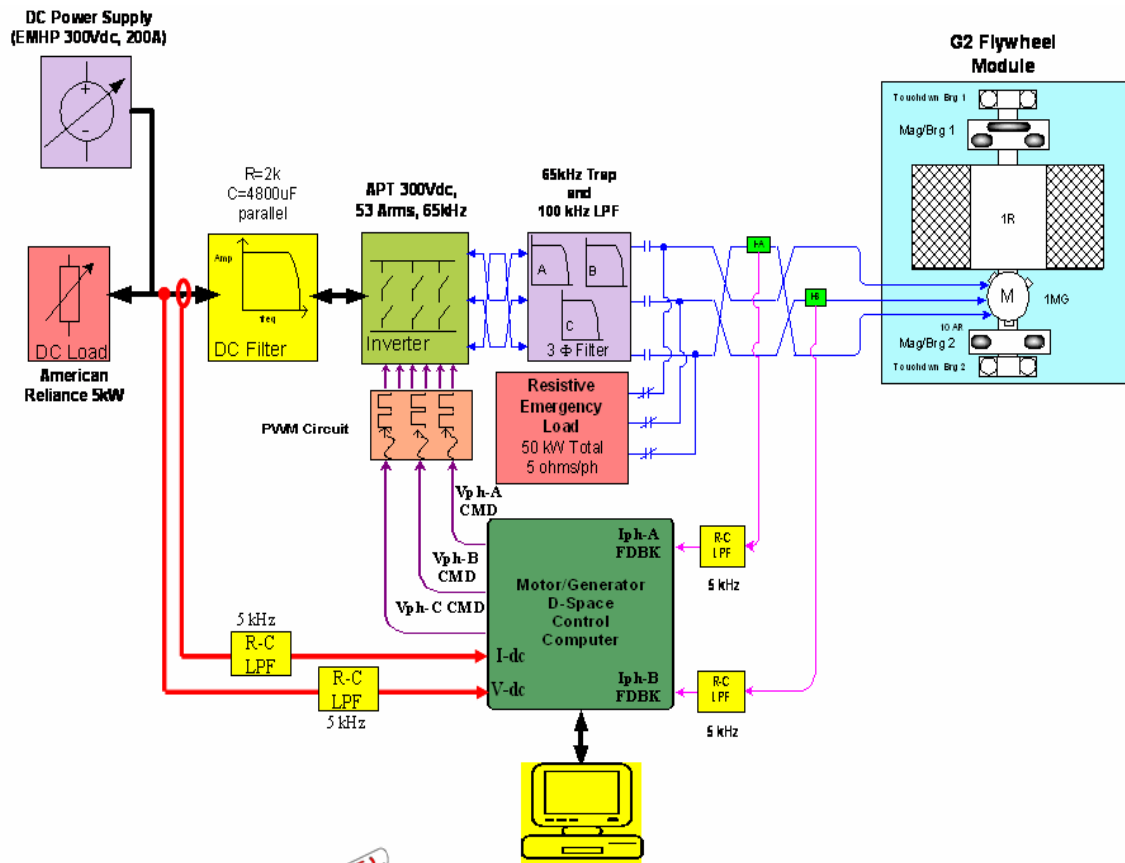


Figure 10.—G2 Motor/generator power and control system.

The major components of this system include the DC bus, antialias filters, the motor/generator controller, the PWM circuit and inverter, the power filter and the emergency load.

The DC bus includes a DC power supply, the DC load, and the DC filter; the bus voltage and current signals are fed back to the motor-generator controller through first order low pass anti-alias filters.

As with the MB controller, the motor-generator controller is written in a rapid development software environment, then compiled and run on a digital controller which includes A/D and digital inputs, a CPU, and D/A outputs, and the user interface for the controller is implemented on a PC. Advanced motor/generator control features include position sensorless control, and bus regulation. The position sensorless motor controller algorithm uses the phase A and B currents as inputs (ref. 7), eliminating the need for an angular position sensor. The bus regulation algorithm uses the DC Bus current and voltage as feedback (ref. 8).

The M/G control algorithm outputs duty cycle information for three phases to the PWM board, which performs a comparison to a triangle wave reference to produce switch states. A driver chip is used to run the gates on the six switch inverter.

Power filters are used to minimize high frequency noise in the system (ref. 9), and the emergency load is used at the end of a test run, to dump the stored rotor power.

## VI. Touchdown Bearing System

The touchdown bearing system on the G2 module has two functions: to support the flywheel rotor when the magnetic bearings are turned off, and to control the rotor position in the event that the magnetic bearing system has suffers a failure during operation.

When the bearings are turned off, the touchdown bearing supports the weight of the rotor, with assistance from the permanent magnets in the motor and magnetic bearings.

The touchdown bearing system is also designed for the worst case magnetic bearing failure, which requires spin down from full operating speed on the touchdown bearings. This case could be caused by loss of power to the magnetic bearings, a major malfunction in the controller hardware, or a combination of several failures in the magnetic bearing actuator.

### A. Touchdown Bearing Requirements

Touchdown bearing requirements can be grouped into five categories: bending mode critical speeds, radial displacement limits, radial force limits, temperature limits, and radial gap.

As is the case with the magnetic bearings, the T/D bearings are designed so that their bending mode critical frequencies are kept out of the operating speed range. The G2 bending critical on the touchdown bearings is expected to be 84,300 rpm, which provides a 40 percent margin above top operating speed of 60 krpm.

The displacements must be limited by the T/D bearing during the touchdown event and subsequent spin down; displacements need to be kept below 20 mils to ensure that the rotor does not contact the stationary parts of the flywheel (the magnetic bearing and motor stators have gaps of 20 and 30 mils, respectively). Per the design, the G2 touchdown displacements will be held to below 20 mils in the event of a worst case (full speed) touchdown, and will decrease to less than 2 mils during the spin down.

The maximum radial force specification is 1500 lbs; by selection of the T/D mechanical bearing, the G2 bearings loads are designed to be below 500 lbs. during rotor touchdown and spin down.

High temperature materials (>200 °F) are necessary because the rotor energy during a touchdown is dissipated through friction in the T/D bearing system. All of the materials used in the G2 touchdown bearing are rated to operate beyond 300 °F.

The required T/D bearing radial gap is >5 mils, and the gap was designed for 7 mils.

The G2 T/D bearing requirements are summarized in table 5. The T/D bearings meet all of the specifications.

TABLE 5.—TOUCHDOWN BEARING REQUIREMENTS

Item	Requirement	Design
Bending Critical Speed on TD's	>72000 rpm	84300
Radial Displacement during spin down on TD's	< 20 mils	< 2 mils
Radial force on TD's during spin down	< 1500 lbs.	<500 lbs
Radial displacement on TD's during touchdown at 60 krpm	<20 mils	<20 mils
Radial force on TD's during touchdown at 60 Krum	<1500 lbs.	<500 lbs
Materials to withstand high temperature operation	200 °F	o-rings-400 °F grease-482 °F
Touchdown bearing radial gap	>5 mils	7 mils

## B. Touchdown Bearing Design

The design of the touchdown bearing is centered about a rotordynamics analysis of the touchdown event.

One key requirement is that, in the event of a touchdown, the transient behavior cannot cause the rotor to enter a whirl mode instability; this type of instability can cause damage to the module.

Additionally, displacement amplitudes and forces are estimated throughout the operating range to ensure that they are within the limits through all the critical frequencies. The key touchdown bearing properties, stiffness, damping, and friction are modified to minimize loads and displacements of the rotor.

The G2 combination and radial touchdown bearing designs are shown in figure 11 and figure 12. Radial stiffness and damping are controlled by O-rings and a squeeze film damper, and axial stiffness of the combination touchdown bearing is controlled by wave springs. Friction is determined by the ball bearing and the touchdown surface between the bearing and the rotor, and anti-rotation pins stop the bearing housing from turning with respect to the housing. Adjustments of the outer touchdown bearing housings with respect to the adjoining parts of the flywheel housing are used to align the touchdown bearing, magnetic bearing and motor centerlines.

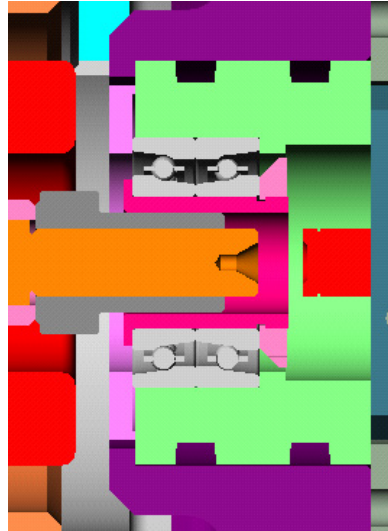


Figure 11.—Radial T/D bearing.



Figure 12.—Combo T/D bearing.

## C. Touchdown Bearing Rotordynamics

Transient analysis is used to determine whether the rotor enters an unstable whirl phenomenon when it initially touches down. Typical transient analysis results are shown in figure 13 corresponding to: A) stable touchdown with gravity, B) stable touchdown without gravity and C) unstable touchdown which will induce whirl.

Stable touchdown with gravity (Case A) must be achieved; this covers the flywheel operation on earth in horizontal orientation. Additionally, stable touchdown without gravity (Case B) must also be achieved with touchdown; this case covers operation in space, and also operation of the flywheel in a vertical orientation.

A touchdown like the one modeled in Case C must be avoided, as an unstable touchdown which will induce whirl can result in major damage to the flywheel module.

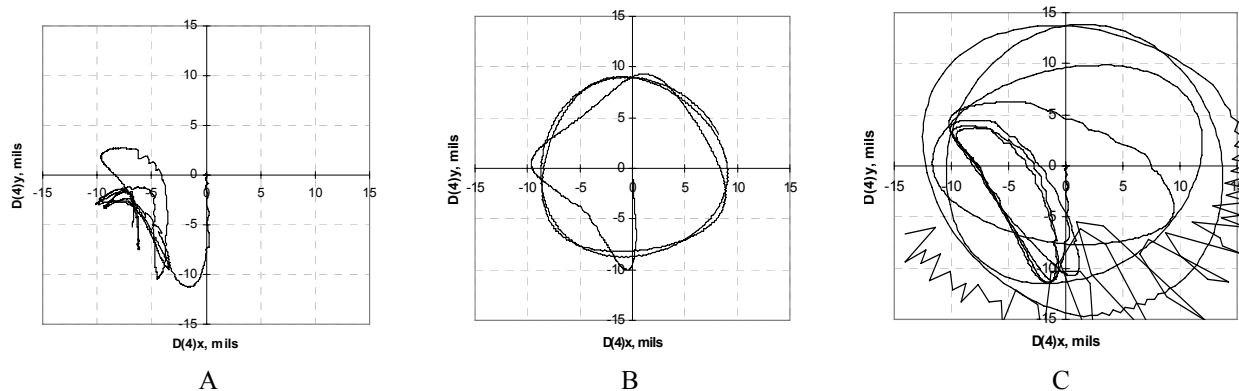


Figure 13.—G2 T/D bearing transient analysis.



The combination of touchdown bearing stiffness and damping, along with the friction coefficient of the touchdown surfaces and the bearing drag, determine operation during touchdown. In order to ensure that the unstable whirl mode is avoided, a map of the touchdown stability region as a function of damping and stiffness is made, and the touchdown bearing is designed to operate within this region, while simultaneously meeting the response and force requirements (table 5).

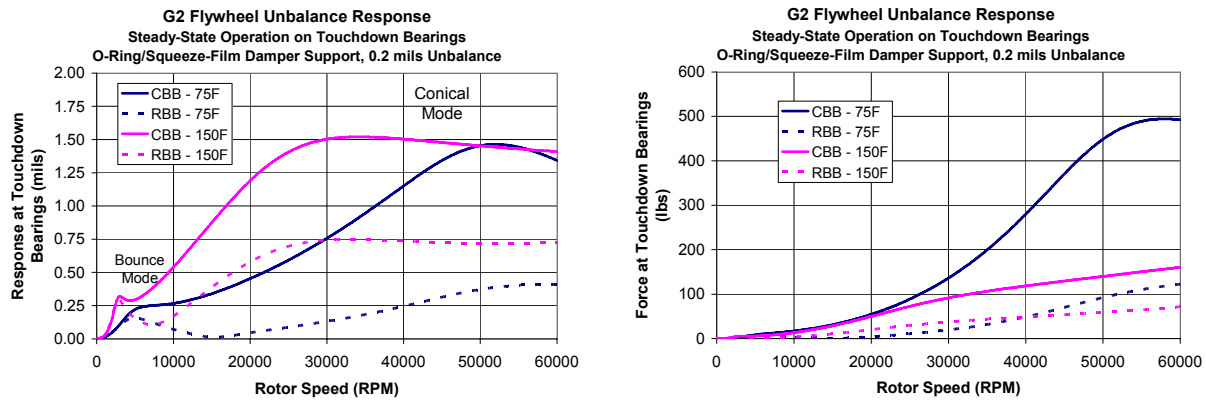


Figure 14.—G2 T/D bearing steady state analysis.

Steady state rotordynamics analysis is used to predict the shaft response and forces at the touchdown bearings (fig. 14). A combined 0.2 mil mass eccentricity unbalance force is added at the faces of the rim to simulate the worst case unbalance, and the bounce mode is stimulated with parallel unbalance vectors at each end and the conical mode is stimulated with orthogonal unbalance vectors. Note that the analyses demonstrate that the force and displacement specifications are met (<2 mils, <1500 lbs), per table 5.

## VII. Housing

The housing is layered to allow reconfiguration of the module with less hardware modifications. There are six layers: combination touchdown bearing, combination magnetic bearing, rotor, radial magnetic bearing, motor/generator and radial touchdown bearing. Each layer has all the external power, signal, thermocouple, and cooling interfaces needed to operate the component.

The housing was not weight optimized. The wall thickness is relatively high to ensure that no significant structural modes exist in the operating speed range.

## VIII. Conclusions

NASA Glenn Research Center has completed the design of the G2 flywheel module. G2 is a modular low cost flywheel which can be used for component and system level flywheel hardware research and systems demonstrations. The G2 flywheel will deliver 525 W-hr usable energy operating at 90 percent depth of discharge between 60,000 and 20,000 rpm. It will provide 1 kW of power at 20,000 rpm and 3 kW of power at 60,000 rpm on a 120V electrical bus. The module weights 218 lbs. It is approximately 12 inches in diameter and 30 inches long. G2 is intended to be operated in a laboratory environment and can be mounted in any orientation. Completion of the G2 flywheel will enable component flywheel technology development, and in combination with D1 will allow full speed, power and torque testing to demonstrate single axis IPACS at NASA Glenn Research Center.

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