

Research on Flywheel Batteries

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ABSTRACT

A 100 Wh flywheel battery was fabricated and demonstrated. Research was conducted on the element technologies of the battery, a carbon fiber reinforced plastic (CFRP) rotor and magnetic bearing control, with the following outcomes. Burst tests conducted on the rotor showed that its achieved strength was more than 80% of the theoretical strength. In addition, 25 000 – 50 000 rpm (operating speed range) cyclic stress tests carried out at ordinary temperatures demonstrated that the rotor is durable to 30 000 cycles without fiber breaking or delamination. The digital processing technique of a speed tracking type band-stop filter for automatic balancing control of magnetic bearing was proposed. It was clarified that the stability of conical forward mode decreases by a gyroscopic effect as speed increases. The control method of magnetic bearing providing low-vibration and stable suspension up to the maximum operating speed of 50 000 rpm was established.

The results described above confirm the feasibility of high specific energy and low loss flywheel battery systems.

1. Introduction

Flywheel batteries are also called mechanical (electro mechanical) batteries, and have superior life, environmental temperature characteristics and other features compared to chemical batteries. The principle of storing energy in flywheel batteries is kinetic energy of the spinning rotor, and the stored energy is proportional to the product of the moment of inertia of the flywheel rotor and the square of the rotor speed. This means that given the same mass and volume, more energy can be stored at higher rotor speeds. Increasing the rotor speed is an effective means of increasing the specific energy; therefore, high speed technology is the key to realizing high specific energy flywheel batteries. Most technical themes for flywheel batteries concern increasing speeds such as high strength rotors that can withstand high speeds, and bearings that can suspend high speed rotors with high stability and low friction loss. Except for low speed systems that use steel rotors, high speed flywheel batteries are currently in the development stage. Establishing practical

technology, understanding performance, and ensuring durability and reliability are currently the central themes. This paper describes the results of research on high speed rotor and bearing technologies that are vital for realizing a high speed flywheel battery system, and reports the results of system tests using a 100 Wh flywheel battery model fabricated with the aim of realizing a flywheel battery system.

2. Outline of the Flywheel Battery

2.1. System Outline

Figure 1 shows a concept drawing of the flywheel battery system.

The flywheel battery is a mechanical battery that stores kinetic energy in the form of a spinning flywheel rotor, and inputs/outputs electrical energy via an internal motor/generator.

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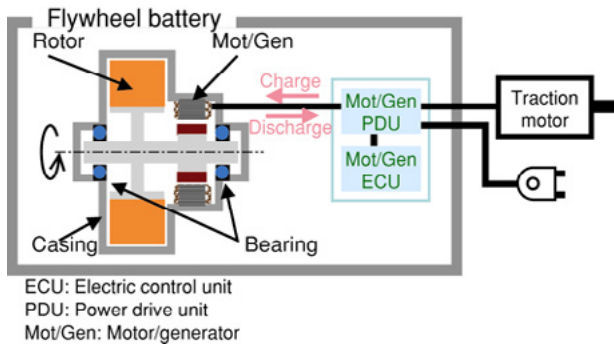


Fig. 1 Outline of flywheel battery system

2.2. Performance Targets

The flywheel battery performance targets were set with the basic concept of the battery for series hybrid vehicles. Table 1 shows the performance targets of the flywheel battery.

Table 1 Performance target of flywheel battery

Max. power (kW)	15 (@25 000 ~ 50 000 rpm)
Max. stored energy (Wh)	333
Max. operating speed (rpm)	50 000
Usable energy (Wh)	250
Operating speed range (rpm)	25 000 ~ 50 000
Total weight (kg)	50

This research aimed to achieve a flywheel battery system that satisfied the performance targets noted above. The primary goal was to realize a flywheel battery system with a usable energy of 100 Wh by shortening the rotor thickness in the axial direction while maintaining the same rotor diameter and rotor speed.

3. Element Technologies of the Flywheel Battery

As shown in Fig. 1, the flywheel battery is composed of the following four main technical elements.

- Rotor which performs the energy storage function
- Motor/generator that performs the charging and discharging (mutual conversion of electrical and kinetic energy) function
- Vacuum housing for reducing windage loss
- Bearings which suspend the high speed flywheel rotor within the vacuum

Research on a high strength rotor and magnetic bearings which are particularly important for realizing a compact and lightweight flywheel battery system is described below.

3.1. Rotor

The specific energy of the rotor is determined by the specific strength which is the value obtained by dividing the tensile strength of the material by the density. Various high specific strength materials were compared, and carbon fiber reinforced plastic (CFRP) was selected as the rotor material. Filament winding was selected as the rotor forming method, and in order to prevent delamination, a press fit multi-ring construction shown in Fig. 2 was adopted. This “high elastic modulus CFRP ring + medium elastic modulus CFRP ring + aluminum ring” construction has different ring densities and elastic moduli in the radial direction⁽¹⁾. Considering ease of fabrication and testability, a CFRP rotor with a usable energy

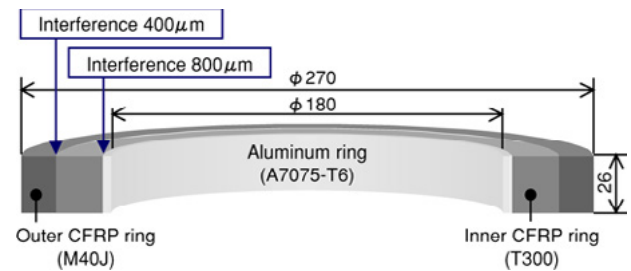


Fig. 2 Press fit multi-ring rotor

of 250 Wh was comprised of five CFRP rings. The 100 Wh flywheel battery model used two CFRP rings.

3.1.1. CFRP rotor stress limit and durability

An important theme for rotor research is ensuring strength and durability. This research set the targets for ensuring performance of the flywheel battery system as follows.

- Designing a CFRP rotor with an initial guaranteed rotor speed of 70 000 rpm (a stress safety factor of 2.0 with respect to the system maximum operating speed of 50 000 rpm)
- Designing the system to prevent overspeed of 50 000 rpm or more

Thus, the rotor research goals were to guarantee the rotor initial performance and durability. In consideration of the fiber strength dispersion, the stress limit of the CFRP rings used for rotor design was assumed to be 70% of the theoretical strength calculated from the fiber tensile strength and the fiber volume fraction (V_f). The rotor was designed so that the hoop stress of the rotor at 70 000 rpm did not exceed this stress limit. In research thus far, rotational tests using the designed and fabricated rotor have verified this speed of 70 000 rpm, and confirmed the following points:

- Stress measurements match the design values.
- The stress limit exceeds 66% of the theoretical strength.

This means that the remaining theme for ensuring performance of the CFRP rotor is to confirm durability. In addition, confirming the stress limit is also important for improving the accuracy of future rotor design. This paper describes the stress limit and durability of the designed and fabricated CFRP rotor.

(1) CFRP rotor stress limit and evaluation method

The theoretical strength of the CFRP rotor ($V_f = 60\%$) is 2.64 GPa for the outer ring which uses the high elastic modulus fiber M40J, and 2.12 GPa for the inner ring which uses the general-purpose fiber T300. Figure 3 shows the relationship between the hoop stress of each CFRP rotor and the rotor speed. The stress reaches the theoretical strength at the rotor speed of 88 600 rpm for the outer ring (M40J fiber) and 92 900 rpm for the inner ring (T300 fiber). The stress limit of an actual CFRP rotor is thought to be lower than the

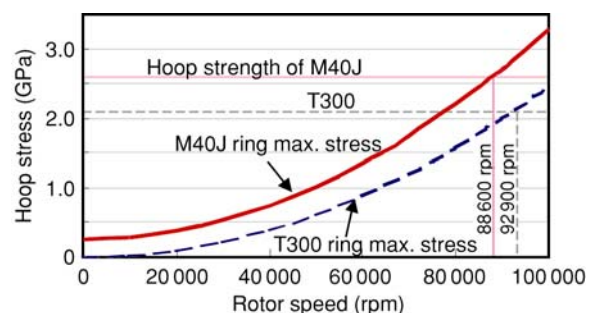


Fig. 3 Hoop stress of CFRP rotor

theoretical strength due to factors such as fiber breaking caused by applying tension during shaping and the contact with the guide rollers, and fiber cutting due to machining.

While there are cases of burst tests having been performed by creating deviations in portions of rotors in order to understand the burst phenomenon of composite material rotors, there are no reports on the evaluation of composite material rotor strength. Strength evaluation of CFRP rotors is important for achieving the optimum rotor design, and understanding the burst phenomenon is also important for ensuring performance with regards to burst. This research entailed conducting CFRP rotor burst tests with the aim of confirming the CFRP rotor stress limit and understanding the burst phenomenon. The stress limit of the CFRP rotor is evaluated from the results of analyzing the burst phenomenon, and the burst speed.

Table 2 Mechanisms and inspections of fatigue

Mechanism	Phenomenon	Nondestructive inspection	Destructive inspection
Breaking of fiber	Deterioration of · Young's modulus · Strength	· Dimension check · Evaluation of diameter during rotation	Burst test
Degradation of resin	Deterioration of strength	· Ultrasonic inspection · Color check · X-ray analysis	
Delamination of layer	Deterioration of strength	—————	

(2) CFRP rotor endurance deterioration and inspection method

Like general deterioration, the deterioration of composite materials can be broadly divided into the following two causes.

- Chemical action (corrosion, chemical changes produced by ultraviolet rays or other factors)
- Physical action (fatigue, creep)

The CFRP rotor exists in a vacuum environment, so deterioration due to chemical action can be ignored. Carbon fibers are said to experience almost no fatigue and creep, and numerous test data supporting this have been published⁽²⁾. However, there are no cases which clearly describe the fatigue mechanisms of the CFRP rotor, and also no reports of research on deterioration due to the centrifugal force of the rotor. Therefore, this research focused on changes in the physical properties of each constituent element of the CFRP rotor, and investigated deterioration phenomena and methods for inspecting deterioration. Table 2 shows the deterioration phenomena and inspection methods. This research employed non-destructive inspection for measurements during the endurance tests, and performed evaluation by measuring the dimensions before and after the endurance test and by ultrasonic inspection after the test.

3.1.2. Evaluation of CFRP rotor stress limit

(1) Rotor for burst test

Figure 4 shows an external view of the rotor used for the burst test.

An aluminum hub (A7075-T6) with spokes was prepared for the burst test, and the CFRP rings were colored white in order to film the burst phenomena. The hub spokes had a tapered shape which became thicker toward the center to alleviate the stress concentration at the spoke joint on the shaft center side. The stress on the spoke joint reached the stress limit for A7075-T6 at a rotor speed of approximately 82 000 rpm, which means that the CFRP rotor strength that

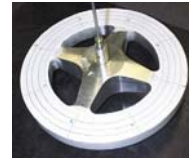


Fig. 4 Rotor for burst test

can be verified at this rotor speed is 85% of the theoretical strength. The burst test was conducted with the test target set as the rotor speed at which hub destruction occurs.

(2) Burst test

Figure 5 shows the transition in test rotor speeds. The test was performed twice, and instantaneous burst occurred at 79 766 rpm (81.1% of CFRP rotor theoretical strength) and 78 507 rpm (78.5% of same). Figure 6 shows a fragment of the burst CFRP rotor. The CFRP rotor shattered into fiber bundles with a maximum length of 80 mm and a maximum width of 3 mm. Based on filmed images of the burst instant and analysis of the destroyed hub, the conclusion was reached that in both cases, hub spoke destruction (originating in the balancing portion of the spoke joint) occurred first, and resulted in bursting of the CFRP rotor.

The fact that burst originated from hub destruction confirmed that the stress limit of the CFRP rings is 80% or more of the theoretical strength. In addition, the burst phenomenon is an instantaneous release of energy, which confirmed the importance of preventing burst.

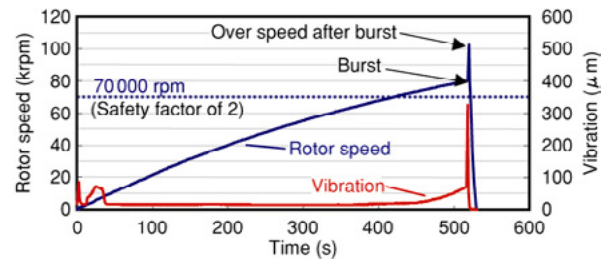


Fig. 5 Burst test of CFRP rotor



Fig. 6 Broken CFRP

3.1.3. Evaluation of CFRP rotor durability

(1) Setting the endurance test conditions

As shown in Fig. 3, the centrifugal force at 50 000 rpm generates stress of approximately 1 GPa on the CFRP rotor. Endurance fatigue was evaluated by repeating this stress. The test pattern was set at a repeated cycle between 25 000 and 50 000 rpm (1 cycle: 25 000 → 50 000 → 25 000 rpm) which produces the maximum stress amplitude in the operation speed range. Cyclic fatigue was assumed to be dependent on the stress amplitude and the number of cycles, and the number of test cycles was calculated based on the charge/discharge pattern of the flywheel battery during cruising modes. The cruising modes used were the 10-15 mode (Japan) and the city mode (US). The number of cycles

when cruising for 100 000 km was approximately 21 500 cycles in 10-15 mode, and approximately 20 500 cycles in city mode. This research set the number of test cycles at 30 000 cycles, and conducted the endurance test at ordinary temperature as the first step in evaluating durability. The test conditions are shown in Table 3.

Table 3 Conditions of cyclic stress test

Speed range (rpm)	25 000 ~ 50 000
Number of cycles	30 000
Spin up/down time (sec/cycle)	60
Vacuum level (Pa)	< 10

(2) Endurance test

Figure 7 shows an external view of the rotor used for the endurance test. The endurance test was conducted for 30 000 cycles under the conditions given in Table 3. The rotor dimensions were measured and ultrasonic inspection was performed every 10 000 cycles, and expansion of the diameter during rotation was also measured before and after the endurance test. Figure 8 shows the dimension measurement results, and Fig. 9 shows the measurement results for expansion of the diameter during rotation. These results can be summarized as follows.

- Measurement of the dimensions of each part showed that the diameter changed by approximately 10 μm after 10 000 cycles but did not change thereafter, and that the change in thickness was 10 μm or less.



Fig. 7 Rotor for cyclic stress test

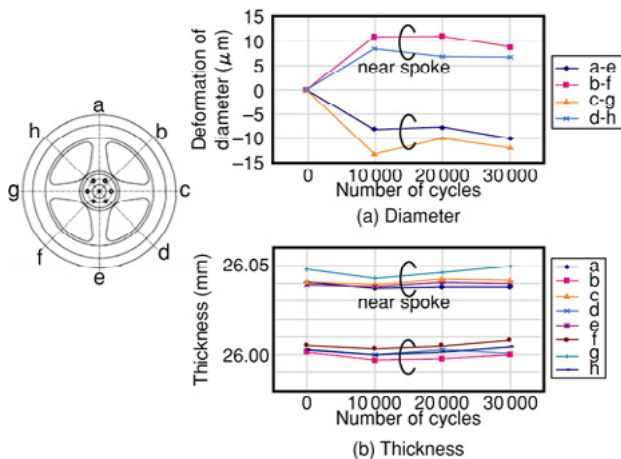


Fig. 8 Inspection of dimensions

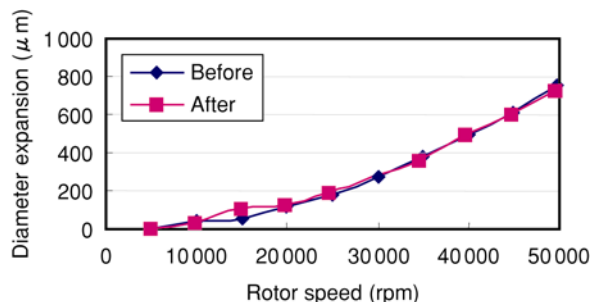


Fig. 9 Evaluation of diameter during rotation

- Ultrasonic inspection did not confirm any delamination around the entire circumference of the rotor.
- Expansion of the diameter during rotation was the same both before and after the endurance test.

The CFRP rotor did not burst during the endurance test. While minute changes occurred in the shape, ultrasonic inspection of the CFRP rotor and measurement of expansion of the diameter during rotation after the endurance test showed that there was no delamination which might indicate deterioration of resin, and no drop in Young's modulus which might indicate fiber cutting. This confirmed that the CFRP rotor has a durability of 30 000 cycles at ordinary temperature.

3.2. Bearings

Requirements for flywheel battery bearings include high speed, low loss, use in a vacuum, and so on. Magnetic bearings that can satisfy these requirements were selected.

3.2.1. Magnetic bearings

Magnetic bearings are non-contact bearings that suspend the rotor by magnetic force. Objects (rigid bodies) have three translational and three rotational degrees of freedom (DOF) for a total of six DOF. The five DOF excluding the rotor spinning motion should be suspended by magnetic bearings. In this system, five DOF active magnetic bearings were adopted. Figure 10 shows a schematic drawing of a general magnetic bearing. Radial magnetic bearings are located in two places in the axial direction, and thrust magnetic bearings are located at the end of the shaft. In addition, non-contact displacement sensors for sensing the rotor motion are located near each electro magnet.

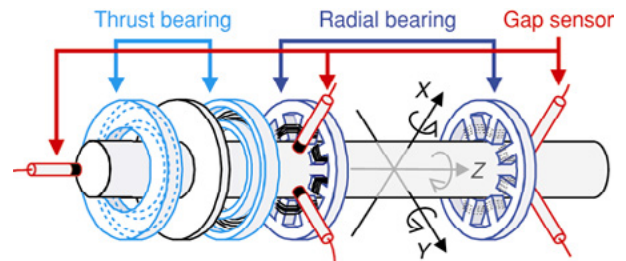


Fig. 10 Schematic of 5 DOF active magnetic bearing

3.2.2. Modeling

The coordinates are as defined in Fig. 11.

This system is a multi-input multi-output (MIMO) (multivariable) system. Here, the state equation for radial magnetic bearings is derived assuming the application of modern control theory which can easily handle a MIMO system and allows systematic design of the control system.

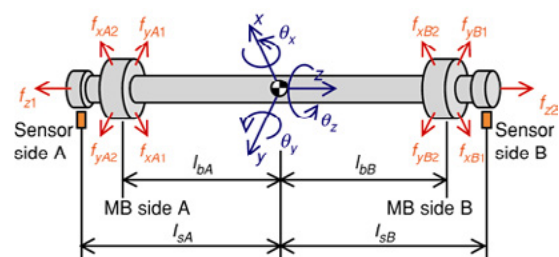


Fig. 11 Definition of Axes

The equation of motion for flywheel rotor in the radial direction is expressed as follows.

$$\begin{cases} m\ddot{x} = (f_{xA2} - f_{xA1}) + (f_{xB2} - f_{xB1}) \\ m\ddot{y} = (f_{yA2} - f_{yA1}) + (f_{yB2} - f_{yB1}) \\ J_d\ddot{\theta}_x = -J_p\omega\dot{\theta}_y + (f_{yA2} - f_{yA1})l_{bA} - (f_{yB2} - f_{yB1})l_{bB} \\ J_d\ddot{\theta}_y = J_p\omega\dot{\theta}_x - (f_{xA2} - f_{xA1})l_{bA} + (f_{xB2} - f_{xB1})l_{bB} \end{cases} \quad (1)$$

Where m is the rotor weight, and J_p , J_d are the moments of inertia around the z axis and the x (y) axis, respectively. Equation (1) is converted to state space form as follows.

$$\begin{aligned} \frac{d}{dt} \begin{bmatrix} \mathbf{x}_g \\ \dot{\mathbf{x}}_g \end{bmatrix} &= \begin{bmatrix} \mathbf{O}_{4 \times 4} & \mathbf{I}_{4 \times 4} \\ \mathbf{O}_{4 \times 4} & \mathbf{A}_{r1} \end{bmatrix} \begin{bmatrix} \mathbf{x}_g \\ \dot{\mathbf{x}}_g \end{bmatrix} + \begin{bmatrix} \mathbf{O}_{4 \times 4} \\ \mathbf{B}_{r1} \end{bmatrix} \mathbf{f}_r \\ \mathbf{x}_g &= [x \ y \ \theta_x \ \theta_y]^T \\ \mathbf{f}_r &= [f_{xA2} - f_{xA1} \ f_{yA2} - f_{yA1} \ f_{xB2} - f_{xB1} \ f_{yB2} - f_{yB1}]^T \\ \mathbf{A}_{r1} &= \begin{bmatrix} \mathbf{O}_{2 \times 2} & \mathbf{O}_{2 \times 2} \\ \mathbf{O}_{2 \times 2} & \mathbf{A}_{r2} \end{bmatrix}, \mathbf{A}_{r2} = \frac{J_p}{J_d} \omega \begin{bmatrix} 0 & -1 \\ 1 & 0 \end{bmatrix} \\ \mathbf{B}_{r1} &= \begin{bmatrix} \frac{1}{m} & 0 & \frac{1}{m} & 0 \\ 0 & \frac{1}{m} & 0 & \frac{1}{m} \\ 0 & \frac{l_{bA}}{J_d} & 0 & -\frac{l_{bB}}{J_d} \\ -\frac{l_{bA}}{J_d} & 0 & \frac{l_{bB}}{J_d} & 0 \end{bmatrix} \end{aligned} \quad (2)$$

The electro magnetic force can be expressed by the following equation.

$$f = \frac{\mu_0 N^2 S}{4} \left(\frac{i}{g} \right)^2 = k \left(\frac{i}{g} \right)^2 \quad (3)$$

Where g is the magnetic bearing clearance, i is the electro magnet coil current, N is the number of electro magnet coil turns, S is the sectional area of the magnet core, and μ_0 is the permeability of free space ($= 4\pi \times 10^{-7}$).

Based on equation (2), the linearized force vector $\mathbf{f}_r$ can be expressed as follows.

$$\begin{aligned} \mathbf{f}_r &= \mathbf{C}_{r1} \mathbf{x}_{rb} + \mathbf{C}_{r2} \mathbf{i}_r \\ \mathbf{i}_r &= [i_{xA} \ i_{yA} \ i_{xB} \ i_{yB}]^T \\ \mathbf{C}_{r1} &= \text{diag} \left(\left[4k_{rA} \frac{I_{rA}^2}{G_{rA}^3} \ 4k_{rA} \frac{I_{rA}^2}{G_{rA}^3} \ 4k_{rB} \frac{I_{rB}^2}{G_{rB}^3} \ 4k_{rB} \frac{I_{rB}^2}{G_{rB}^3} \right] \right) \\ \mathbf{C}_{r2} &= -\text{diag} \left(\left[4k_{rA} \frac{I_{rA}}{G_{rA}^2} \ 4k_{rA} \frac{I_{rA}}{G_{rA}^2} \ 4k_{rB} \frac{I_{rB}}{G_{rB}^2} \ 4k_{rB} \frac{I_{rB}}{G_{rB}^2} \right] \right) \end{aligned} \quad (4)$$

Where $\mathbf{x}_{rb}$ represents the rotor displacement vector, $\mathbf{i}_r$ represents the control current vector, G_{rA} and G_{rB} are the normal clearance of each magnetic bearing, and I_{rA} and I_{rB} are the bias current of each electro magnet. Using the following relational equation of the control voltage and current,

$$\begin{aligned} \mathbf{e}_r &= \mathbf{L}_r \frac{d\mathbf{i}_r}{dt} + \mathbf{R}_r \mathbf{i}_r \\ \mathbf{e}_r &= [e_{xA} \ e_{yA} \ e_{xB} \ e_{yB}]^T \\ \mathbf{L}_r &= \text{diag}(L_{rA} \ L_{rA} \ L_{rB} \ L_{rB}) \\ \mathbf{R}_r &= \text{diag}(R_{rA} \ R_{rA} \ R_{rB} \ R_{rB}) \end{aligned} \quad (5)$$

and the following relational equations,

$$\begin{aligned} \mathbf{x}_{rb} &= \begin{bmatrix} x_{bA} \\ y_{bA} \\ x_{bB} \\ y_{bB} \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & -l_{bA} \\ 0 & 1 & l_{bA} & 0 \\ 1 & 0 & 0 & l_{bB} \\ 0 & 1 & -l_{bB} & 0 \end{bmatrix} \mathbf{x}_g = \mathbf{T}_{gb} \mathbf{x}_g \\ \mathbf{x}_{rs} &= \begin{bmatrix} x_{sA} \\ y_{sA} \\ x_{sB} \\ y_{sB} \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & -l_{sA} \\ 0 & 1 & l_{sA} & 0 \\ 1 & 0 & 0 & l_{sB} \\ 0 & 1 & -l_{sB} & 0 \end{bmatrix} \mathbf{x}_g = \mathbf{T}_{gs} \mathbf{x}_g \end{aligned} \quad (6)$$

where vector $\mathbf{x}_{rs}$ represents rotor displacement of each sensor portion, combining equations (2) and (4) results in the following state equation.

$$\begin{aligned} \frac{d}{dt} \begin{bmatrix} \mathbf{x}_{rs} \\ \dot{\mathbf{x}}_{rs} \\ \mathbf{i}_r \end{bmatrix} &= \begin{bmatrix} \mathbf{O}_{4 \times 4} & \mathbf{I}_{4 \times 4} & \mathbf{O}_{4 \times 4} \\ \mathbf{T}_{gs} \mathbf{B}_{r1} \mathbf{C}_{r1} \mathbf{T}_{gb} \mathbf{T}_{gs}^{-1} & \mathbf{T}_{gs} \mathbf{A}_{r1} \mathbf{T}_{gs}^{-1} & \mathbf{T}_{gs} \mathbf{B}_{r1} \mathbf{C}_{r2} \\ \mathbf{O}_{4 \times 4} & \mathbf{O}_{4 \times 4} & -\mathbf{L}_r^{-1} \mathbf{R}_r \end{bmatrix} \begin{bmatrix} \mathbf{x}_{rs} \\ \dot{\mathbf{x}}_{rs} \\ \mathbf{i}_r \end{bmatrix} \\ &= \begin{bmatrix} \mathbf{O}_{4 \times 4} \\ \mathbf{O}_{4 \times 4} \\ \mathbf{L}_r^{-1} \end{bmatrix} \mathbf{e}_r \end{aligned} \quad (7)$$

3.2.3. Magnetic bearing control

Rotational vibration is an issue for suspending a high speed rotor, and the causes of this vibration can be broadly divided into the following two factors:

- Rotor unbalance
- Resonance

Countermeasures for these factors are described below.

(1) Reduction of unbalanced vibration

A residual unbalance exists in the rotor, and this unbalance produces rotational vibration. Various methods using observers and filters have been proposed to suppress unbalanced vibration⁽³⁾, and these methods can be broadly divided into the following two types. The first is methods that suspend the rotor around the axis of rotation (geometrical axis) by control force that negates the centrifugal force produced by the unbalance, and the second is methods that suspend the rotor around the principal axis of inertia without generating control force for unbalanced vibration. The latter type is also called automatic balancing (rotation on principal axis of inertia) control.

This research selected automatic balancing control using a band stop filter. The filter eliminates the rotational frequency vibration (unbalanced vibration) from signals and the filtered signals are fed back, so the control forces (stiffness) with respect to the whirling vibration are set to zero. In particular, this research devised a method of digital filter processing so that the center frequency (band stop frequency) tracks the rotor speed. Figure 12 shows the block diagram for this processing. The filter characteristics are varied based on the rotor speed signal. Figure 13 shows the rotational vibration reduction effects due to automatic balancing control using this filter.

(2) Avoiding resonance

Resonance of a rotor suspended by magnetic bearings can be broadly divided into rigid body mode and bending mode. Rigid body mode is produced by the stiffness of the magnetic bearings (spring and damper suspension), and has the two modes of parallel and conical. Bending mode is the so-called bending resonance of the rotor. The issues in rotating are the excitation and instability of these resonance modes, and these factors are the critical speed and gyroscopic effect. The

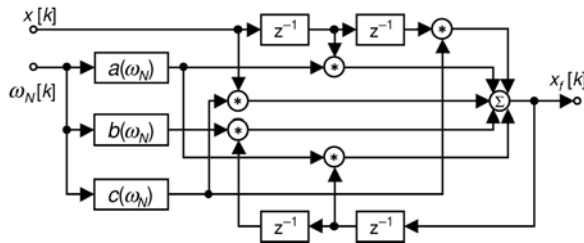


Fig. 12 Block diagram of a speed tracking band stop filter

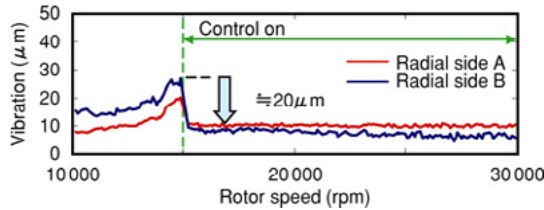


Fig. 13 Effect of automatic balancing control

critical speed indicates the rotor speed at which resonance is excited by the aforementioned rotor unbalance force, that is to say the rotor speed at which the rotational frequency matches the resonance frequency. The gyroscopic effect is a phenomenon unique to rotors. As shown in equation (1), an interference term due to the gyroscopic effect is present in the first term on the right hand side of the two rotational equations. Due to this effect, each resonance frequency when the rotor is stopped ($\omega = 0$), branches into two high and low frequencies as the rotor speed increases. Figure 14 shows a typical representation of this phenomenon. The modes that change into the high and low frequencies are called the forward and backward modes, respectively. The influence of this gyroscopic effect increases for high speed flywheels which also have a large moment of inertia J_p , resulting in a system where the dynamic characteristics vary relative to the rotor speed.

The critical speed of rigid body mode resonance can basically be traversed by applying damping characteristics through magnetic bearing control. The feedback gain was calculated for a model with a gyroscopic term of zero (non rotating) in equation (7), and the eigenvalue of a closed loop system when controlled by this feedback gain was calculated to evaluate the probability for instability as the rotor speed increases. Figure 15 shows the results, which revealed that the damping ratio for the conical forward mode decreases as the rotor speed increases. In actual control, the effects of the phase lag characteristics of the sensors and filter are also added, so this mode has probability for instability. Therefore, a means for adjusting the feedback gain according to the rotor speed was provided to improve the damping characteristics of conical forward mode (avoidance of instability).

The relationship between the bending resonance and gyroscopic effect of rotors suspended by mechanical bearings and other high stiffness bearings are described in detail in reference (4). Research to traverse the critical speed of bending mode resonance by magnetic bearing control is also being performed, but generally the rotor is stiffened so that the critical speed is not within the operating speed range.

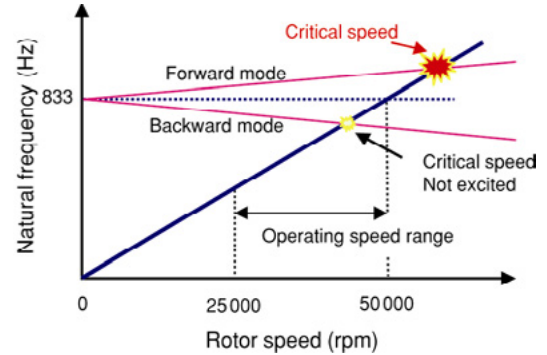


Fig. 14 Gyroscopic effect

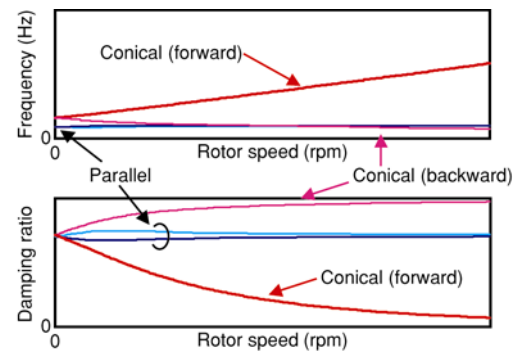


Fig. 15 Rigid mode natural frequencies and damping ratios as a function of rotor speed (closed loop simulation)

This research set the flywheel rotor bending mode frequencies of 833 Hz or more to avoid both the critical speed and instability.

3.2.4. Control system

The controller hardware employs a commercially available rapid prototyping environment based on a digital signal processor (DSP) as shown in Fig. 16, and performs digital magnetic bearing control.

Figure 17 shows the block diagram of a magnetic bearing controller. The controller performs the appropriate filter processing, state vector calculations and feedback gain processing for each sampling signal (displacement, current), and calculates the control commands. The controller based on the optimum regulator provides aforementioned means for adjusting each filter and feedback gain parameter according to the rotor speed.

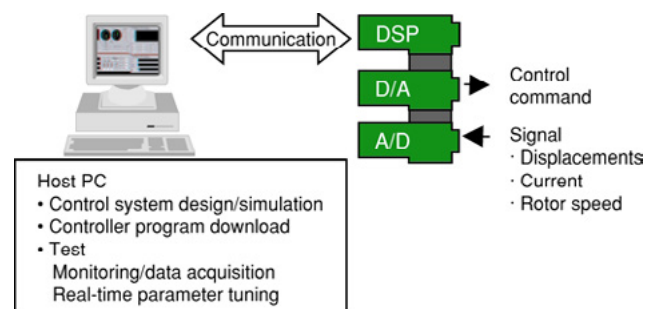


Fig. 16 Rapid prototyping system

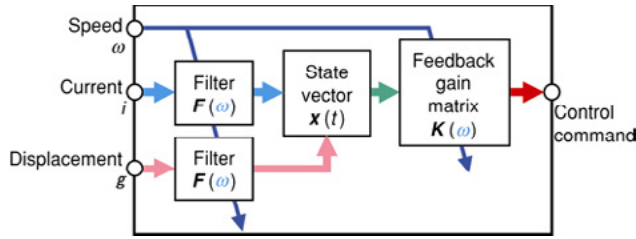


Fig. 17 Block diagram of a magnetic bearing controller

4. Research on a Flywheel Battery System

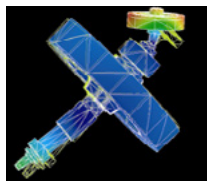
4.1. 100 Wh Model

This section describes research on a flywheel battery system that applies the aforementioned rotor and bearing technologies. A 100 Wh model was fabricated as the first step towards realizing the 250 Wh flywheel battery system. Table 4 lists the main specifications of the 100 Wh model. Compared to the final 250 Wh system that will have a five CFRP ring construction, the 100 Wh model used two CFRP rings.

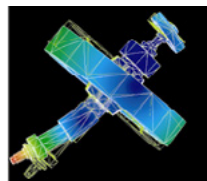
Figure 18 shows the analysis of eigenvalue problems for the designed flywheel rotor. These results show that the designed flywheel satisfied the bending mode frequencies of 833 Hz or more. Figure 19 shows the 100 Wh model rotor assembly (weight 7.6 kg) and an external view of the entire model.

Table 4 Specifications of 100 Wh model

Max. operating speed (rpm)	50 000
Operating speed range (rpm)	25 000 ~ 50 000
Max. power (kW)	15 (@25 000 ~ 50 000 rpm)
Usable energy (Wh)	100 (CFRP ring $\times$ 2)
Natural freq. (Hz)	> 833
Bearing type	5 DOF Active magnetic bearing
Mot/Gen type	Brushless DC

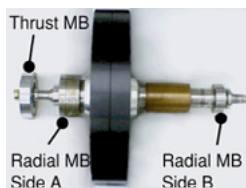


(a) 1st mode 872 Hz



(b) 2nd mode 1083 Hz

Fig. 18 Bending mode shapes and frequencies



(a) Rotor assembly



(b) External view

Fig. 19 100 Wh model

4.2. Rotational Test

Figures 20, 21 and 22 show the results of rotational tests conducted after setting and adjusting the magnetic bearing control parameters. Figure 20 shows the vibrational amplitude of each part during rotation. The vibration in both the radial and thrust directions has a low amplitude (maximum approximately 20 μm) at all speeds, and stable rotation was achieved up to the full speed of 50 000 rpm. Figure 21 shows the whirling orbit (Sides A and B) of the rotor at 50 000 rpm, and stable rotation could be confirmed here as well. Figure 22 is the FFT analysis results of the rotor vibration during rotation, and shows that there are no resonance or other abnormal vibrations other than the rotor synchronous vibration. There is no instability in the conical forward mode, and appropriate control is performed up to full speed. These rotational test results confirmed the validity of the adopted magnetic bearing control.

Thus, the maximum operating speed of 50 000 rpm was achieved, and storage of usable energy of 100 Wh was verified.

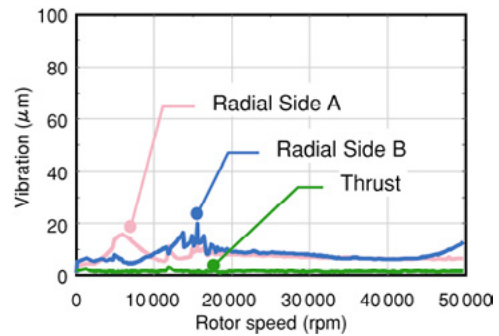
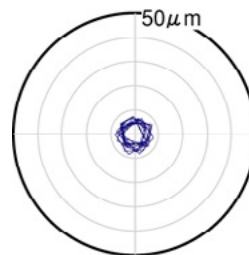
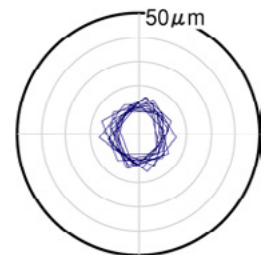


Fig. 20 Synchronous displacement up to full speed



(a) Side A



(a) Side B

Fig. 21 Whirling orbit @ 50 000 rpm

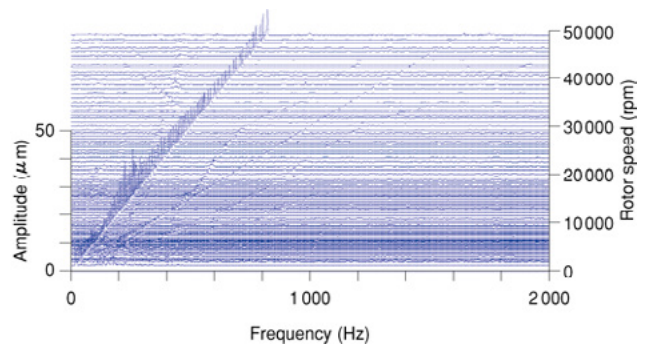


Fig. 22 Waterfall plot

4.3. Charge/Discharge Test

A test was conducted to confirm the charge/discharge performance of the flywheel battery system. Charging and discharging were performed with the drive and regenerative power command for the internal motor/generator control held constant at 7.5 kW. Figure 23 shows the results. Here, charging and discharging were performed for a total of two cycles, with one cycle consisting of charging and discharging from 25 000 rpm to 50 000 rpm. The results confirmed that the basic battery functions of charging and discharging can be performed, and that stable charging and discharging can be repeated, thus verifying compatibility as a battery. The charge/discharge efficiency for a single cycle was 86.3%.

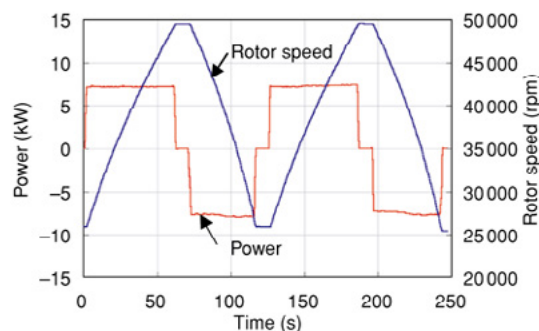


Fig. 23 Results of charge/discharge test

5. Conclusion

Research on carbon fiber composite material rotors and magnetic bearings that are the technical elements of flywheel batteries, and research on a flywheel battery system that applies these results, were carried out with the aim of realizing a 250 Wh flywheel battery system. Burst tests were conducted to evaluate the strength of a designed and fabricated CFRP rotor, and confirmed that the stress limit was 80% or more of the theoretical strength. Cyclic tests were conducted for 30 000 cycles at ordinary temperature within the operating speed range to evaluate the CFRP rotor durability, and confirmed that fiber cutting and delamination did not occur, and that the rotor had sufficient durability. For the magnetic bearings, a rotational speed tracking type digital filter was constructed for automatic balancing control. In addition, the deterioration of conical resonance stability due to the gyroscopic effect was clarified, and a corrective control method was established. A 100 Wh flywheel battery that employs the results of the rotor and magnetic bearings research was fabricated and used to achieve the maximum operating speed of 50 000 rpm, and to verify the charge/discharge functions, thus realizing a 100 Wh flywheel battery system.

The results of this research are thought to indicate the feasibility of high specific energy and low loss flywheel battery systems.

6. Afterword

In the future the authors plan to fabricate a 250 Wh flywheel battery system that applies these research results, and verify the target performance.

Note that part of this research is being conducted in conjunction with the "Research and Development of Advanced Clean Energy Vehicle (ACE)" project led by the Ministry of Economy, Trade and Industry. This project aims

to develop vehicles with CO₂ emissions of half or less than existing vehicles by combining petroleum substitute clean energy fuels with hybrid systems, and has been commissioned by the New Energy and Industrial Technology Development Organization (NEDO). A part of this research is the results of research conducted during fiscal years 1999 through 2002^{(5), (6), (7), (8)}.

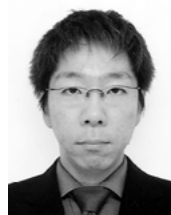
Acknowledgements

The authors would like to take this opportunity to express their deepest thanks to all related parties both in-house and at other companies for their generous guidance and cooperation with this research.

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