

Development of CFRP Rotor for Flywheel Batteries

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ABSTRACT

Carbon fiber reinforced plastic (CFRP) filament winding was selected as the forming method for the flywheel battery rotor due to its high specific energy. In order to prevent the delamination problem that often occurs with this forming process, a press fit multi-ring construction was employed. This multi-ring construction consists of a CFRP ring with a high elastic modulus, a CFRP ring with medium elastic modulus and an aluminum ring. A safety factor of 2.0 was set and each rotor ring was designed and manufactured with the optimal material and dimensions. Rotational tests were performed through 70 000 rpm whereby it was verified that the 2.0 safety factor is maintained. Further, by taking stress measurements up to 50 000 rpm, we were able to demonstrate the validity of the design.

1. Introduction

The effects of global warming on the environment is an important concern, and one of the main causes of global warming is thought to be greenhouse effects due to the emission of large amounts of CO₂. Auto manufacturers are conducting R&D on a wide range of systems such as reducing fuel consumption rates, alternate fuel vehicles, electric vehicles (EV) and hybrid electric vehicles (HEV), and a number of these efforts have already reached the practical application stage.

Based on the results of much of this research, current development trends are focusing on the development of gasoline/diesel HEV or alternative fuel HEV for the mid-term, and fuel cell vehicles (FCEV) for the long-term.

This paper reports the results of research on a flywheel battery rotor which is part of a concept for exploring the possibility of vast reductions in CO₂ emissions by combining an HEV system utilizing a natural gas engine which has lower CO₂ emissions than a gasoline engine of the same output ratio with the new technologies of a flywheel battery and an adsorbed natural gas (ANG) fuel tank. (Fig.1)

In order to increase the specific energy of the flywheel battery, fiber reinforced plastic (FRP) is often selected as the rotor material. FRP is strong in the tensile direction of the fibers, but extremely weak in the direction 90° to that. Various forming techniques have been devised to overcome this problem, but the performance anticipated from material reinforcement has not been obtained.⁽¹⁾⁻⁽⁴⁾ In addition, while strain and stress during rotation have been theoretically calculated, these results have not yet been verified through actual measurement. This study achieved the target performance by adopting a press fit multi-ring construction and designing and manufacturing a rotor with the optimal material and dimensions for each ring. In addition, the validity of the design was verified by measuring the strain and comparing the calculated and actual measured results.

This study is a part of the fiscal 1998 research for the “High Efficiency Clean Energy Vehicle R&D” project commissioned by the New Energy and Industrial Technology Development Organization (NEDO). This research aims to develop a vehicle with a running energy consumption and CO₂ emissions of no more than half that of existing vehicles through a combination of petroleum alternative clean energy sources and HEV systems.

2. Setting Performance Targets

Flywheel batteries are a type of mechanical battery that stores kinetic energy in a rotating disk. This energy is input and output in the form of electricity via a motor/generator (Mot/Gen) which acts as a motor during charge and a generator during discharge. Fig.2 shows a concept drawing of a flywheel battery.

Table 1 shows the performance targets for a flywheel battery which satisfy both the fuel and motive power performance targets of an ANG-HEV vehicle.

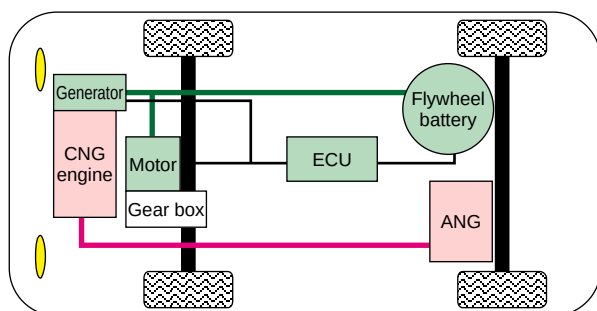


Fig.1 Outline of ANG-HEV system

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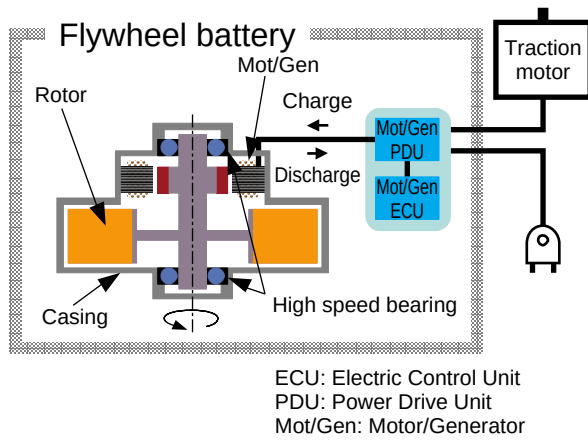


Fig.2 Outline of flywheel battery system

Table 1 Specifications of flywheel battery

Max power	30kW
Usable energy stored	500Wh
Total weight	100kg

3. Design

3.1. Investigation of Materials

Like a chemical battery, in addition to the total amount of energy that can be stored, high specific energy, high specific output and high efficiency are also required of the flywheel when applied as a battery. Of these, the requirement for the rotor, which is the main unit for storing energy, is "high specific energy". Therefore, the author verified the type of rotor material that would meet the requirement for "obtaining high specific energy".

The energy that can be stored was considered assuming the use of a rotating disk such as that shown in Fig.3. The disk was considered to be a solid disk made of isotropic materials in order to simplify the model.

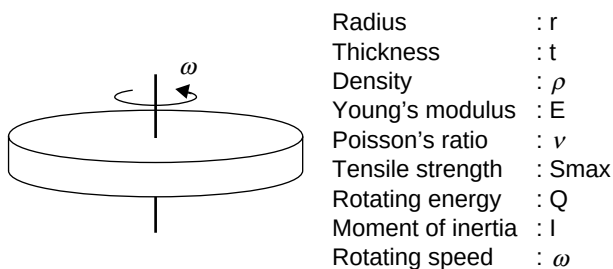


Fig.3 Rotating disk

The amount of kinetic energy that can be stored in a rotating disk is expressed by the following equation.

$$Q = \frac{1}{2} I \omega^2 = \frac{1}{4} \pi \rho r^4 t \omega^2$$

Next, in consideration of the rotational stress produced

on this disk, the maximum attainable revolutions are the revolutions at which the maximum stress produced equals the tensile strength of the materials.

$$S_{max} = \frac{1}{8} \rho r^2 (3 + \nu) \omega_{max}^2$$

Therefore, the maximum specific energy that can be achieved with a rotating disk made from a certain material can be defined by the following equation.

$$(Q/Wt)_{max} = \frac{1}{4} (r \omega_{max})^2 = \frac{2S_{max}}{\rho (3 + \nu)}$$

This equation shows that the maximum achievable specific energy of a rotating disk is determined by the specific strength, which is the value obtained by dividing the tensile strength of the material by the density.

Based on the above, the results of comparing the maximum specific energy of various materials are shown in Table 2.

Table 2 Comparison of available specific energy

	Density ρ (kg/m ³)	Poisson's ratio ν	Tensile strength S_{max} (MPa)	Specific energy Q/Wt (Wh/kg)
Maraging steel (MAS2)	8000	0.31	2942	62
Carbon steel (SCM435)	7830	0.28	1373	30
Titanium (Ti-6Al-4V)	4430	0.19	1177	46
Aluminum (A7075-T6)	2800	0.29	569	34
Thermoplastic (Nilon66)	1140	0.3	78	12
Synthetic fiber (Nilon66)	1140	0.3	665	98
Aramido fiber (Kevlar129)	1368	0.3	2334	287
Glass fiber (S-Glass)	2103	0.3	3158	253
Carbon fiber (T1000G)	1620	0.3	4459	463

Table 2 shows that carbon fiber reinforced plastic (CFRP) rotors have the highest specific energy, so CFRP was selected as the rotor material.

3.2. Investigation of Forming Methods

Carbon fibers are an anisotropic material that exhibit high tensile strength only in the tensile direction of the filaments. Therefore, it was necessary to design a construction which makes the fullest use of this high tensile strength, that is to say a construction which receives the high centrifugal stress produced by rotation in the tensile direction of the filaments.

There are two possible filament arrangements which receive the centrifugal stress in the tensile direction of the filament as follows.

- (I) Arranging the filaments so that they radiate axially outward in a radial direction from the center.
- (II) Arranging the filaments in ring shapes in the hoop direction.

Of these, evenly arranging the filaments from the smaller center areas to the larger peripheral areas is not possible with arrangement (I), making it difficult to actually form a

high strength rotor with this arrangement. Therefore, when forming a flywheel rotor using composite fiber materials, filament winding (II) is selected. (Fig.4)

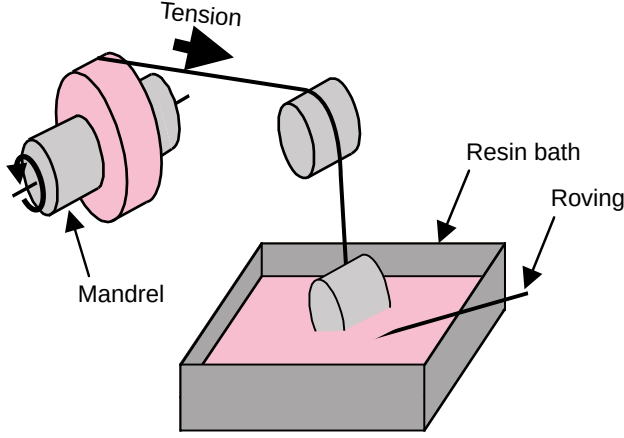


Fig.4 Filament winding

3.3. Calculating Rotational Stress

Filament winding formed rotors exhibit strong anisotropy with an extremely different material constant and strength in the hoop and radial directions. Therefore, stress calculation using a two-dimensional polar coordinate system was selected.⁽⁵⁾ The equations for calculating the rotational stress are shown below.

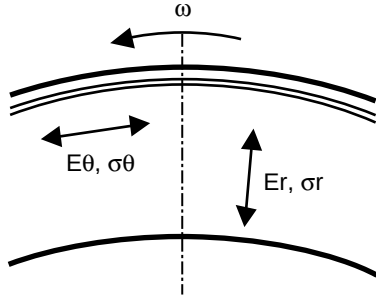


Fig.5 Stress and strain

The symbols used for calculation are defined as follows.

- r: Radius
- ϵ_θ : Hoop strain
- σ_θ : Hoop stress
- E_θ : Hoop Young's modulus
- ν_θ : Hoop Poisson's ratio
- η_θ : Hoop coefficient of thermal expansion
- ΔT : Temperature difference
- E_f : Fiber Young's modulus
- ν_f : Fiber Poisson's ratio
- ρ : Density
- u: Radial displacement
- ϵ_r : Radial strain
- σ_r : Radial stress
- E_r : Radial Young's modulus
- ν_r : Radial Poisson's ratio

- η_r : Radial coefficient of thermal expansion
- V_f : Fiber volume content
- E_m : Resin Young's modulus
- ν_m : Resin Poisson's ratio
- ω : Rotational angular velocity

The relations between the stress and strain are as follows.

$$\epsilon_r = \sigma_r / E_r - \nu_{\theta r} \cdot \sigma_\theta / E_\theta$$

$$\epsilon_\theta = \sigma_\theta / E_\theta - \nu_{r\theta} \cdot \sigma_r / E_r$$

Which can be simplified as follows.

$$F = E_r \cdot \nu_{\theta r} = E_\theta \cdot \nu_{r\theta}$$

$$E_r' = E_r / (1 - \nu_{\theta r} \cdot \nu_{r\theta})$$

$$E_\theta' = E_\theta / (1 - \nu_{\theta r} \cdot \nu_{r\theta})$$

$$F' = F / (1 - \nu_{\theta r} \cdot \nu_{r\theta})$$

The balance between the stress and centrifugal force in the radial direction is expressed by the following equation.

$$d(r \cdot \sigma_r) / dr - \sigma_\theta = -\rho \omega^2 r^2$$

The relations between displacement and strain are as follows.

$$\epsilon_r = du / dr$$

$$\epsilon_\theta = u / r$$

Thus, assuming E_r' , E_θ' and F' are constants the following equation is obtained.

$$E_r' \cdot r^2 \cdot d^2u / dr^2 + E_r' \cdot r \cdot du / dr - E_\theta' \cdot u = -\rho \omega^2 r^3$$

By solving this equation and taking into account temperature dependence, the displacement, strain and stress are expressed by the following equations.

$$u = X \cdot r^\alpha + Y \cdot r^{-\alpha} + A \cdot r^3 + F \cdot r$$

$$\epsilon_r = \alpha \cdot X \cdot r^{\alpha-1} - \alpha \cdot Y \cdot r^{-(\alpha+1)} + 3A \cdot r^2 + F$$

$$\epsilon_\theta = X \cdot r^{\alpha-1} + Y \cdot r^{-(\alpha+1)} + A \cdot r^2 + F$$

$$\sigma_r = B \cdot X \cdot r^{\alpha-1} - C \cdot Y \cdot r^{-(\alpha+1)} + D \cdot r^2 + G$$

$$\sigma_\theta = B \cdot \alpha \cdot X \cdot r^{\alpha-1} + C \cdot \alpha \cdot Y \cdot r^{-(\alpha+1)} + E \cdot r^2 + G$$

$$A = \rho \cdot \omega^2 \cdot (\alpha^2 - \nu_{\theta r}^2) / (E_\theta \cdot (\alpha^2 - 9))$$

$$B = E_\theta / (\alpha - \nu_{\theta r})$$

$$C = E_\theta / (\alpha + \nu_{\theta r})$$

$$D = \rho \cdot \omega^2 \cdot (3 + \nu_{\theta r}) / (\alpha^2 - 9)$$

$$E = \rho \cdot \omega^2 \cdot (\alpha^2 + 3\nu_{\theta r}) / (\alpha^2 - 9)$$

$$F = (\eta_\theta \cdot (\alpha^2 - \nu_{\theta r}) + \eta_r \cdot (\nu_{\theta r} - 1)) \cdot \Delta T / (\alpha^2 - 1)$$

$$G = E_\theta \cdot (\eta_\theta - \eta_r) \cdot \Delta T / (\alpha^2 - 1)$$

3.4. Determining Rotor Specifications and Investigating Strength

Flywheels are characteristic in that energy is stored as rotational kinetic energy, so more energy can be stored at higher revolutions. However, the design requirements were determined as shown in Table 3 according to the following required performances and limitations.

- Specific energy requirements as a system
- Canceling out the gyro-moment using two rotors rotating in opposite directions
- Mot/Gen control capability limit
- Achieving a safety factor of 2.0 based on results for compressed natural gas (CNG) tanks, etc.

CFRP was selected as the rotor material. In order to

Table 3 Rotor design requirements

Usable energy stored	250Wh*1
Total energy stored	333Wh
Available speed range	0 to 5000rpm
Operating speed range	25000 to 50000rpm
Burst speed	over 70000rpm
Rotor radius	$\approx$ Rotor Thickness
Inside diameter/outside diameter	$\approx$ 0.7

*1 Use two rotors for canceling out the gyro-moment

satisfy the performance targets noted in Table 3, the outside diameter of the rotor was changed and the amount of energy that can be stored in those rotors is indicated in Fig.6. This graph shows that a rotor outside diameter of $\phi 270$ is needed to store an effective energy of 250 Wh.

Therefore, the rotor dimensions were determined as follows.

Outside diameter: $\phi 270$ (mm)

Inside diameter: $\phi 190$ (mm)

Thickness: 130 (mm)

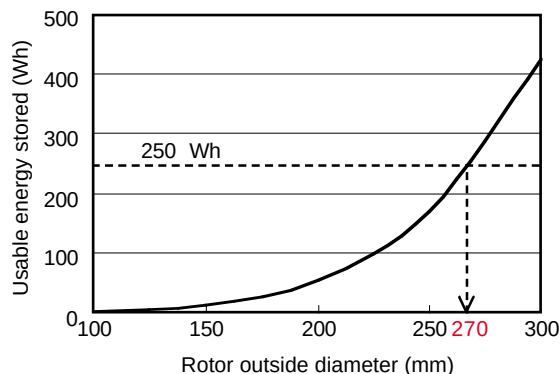


Fig.6 Relation between rotor outside diameter and stored energy

Fig.7 shows the stress calculation results at 70000 rpm for a filament wound rotor having an outside diameter of 270 mm and an inside diameter of 190 mm formed with Toray T1000G, which has the highest tensile strength of currently available carbon fibers. Table 4 shows the Toray T1000G ($V_f = 60\%$ CFRP) allowable stress limit and material constants used for the calculation.

Rotation produces large hoop stress in the tensile direction of the fibers. Further the elongation at the inner and outer hoops differs. Therefore stress is also produced in the radial direction. Filament winding shaped rotors exhibit

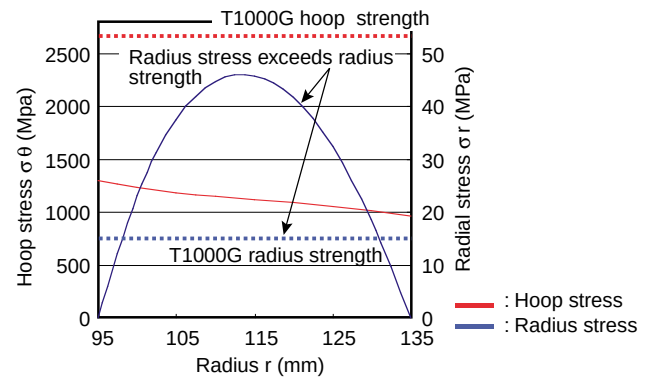


Fig.7 Stress of T1000G rotor at 70000rpm

extreme anisotropy and have a construction where in fibers distributed in layers in the radial direction are fixed by resin. Therefore, the tensile strength is particularly low in the radial direction. Fig.7 shows that the radial stress produced by rotation greatly exceeds the radial tensile strength for a CFRP single layer simple ring winding rotor, leading to delamination in the layers between the fibers. Therefore, suppressing radial stress due to rotation and improving the rotor construction to prevent delamination are important issues.

3.5. Investigation of Construction

The following approaches can be considered to prevent inter-layer delamination of the rotor:

- (I) Adopting a construction which produces compressive force in the radial direction due to rotation.
- (II) Applying an initial compressive force to the rotor which exceeds the produced radial tensile stress.
- (III) Improving the matrix resin strength.
- (IV) Adding reinforcing materials in the radial direction.

Possible forming methods for each of these are as follows:

- (I): Providing a density slope according to radius so that inner rings have a higher density, or providing an elastic modulus slope according to radius so that outer rings have a higher elastic modulus.
- (II): Adopting an integrated construction where in multiple rings are press fit with tightening margins.
- (III): Using a thermoplastic resin matrix.
- (IV): Adopting a three-dimensional fiber construction, or a structure with ring winding disks and radial arrangement disks adhered together.

Of these methods, forming technologies which satisfy the required reliability and strength do not exist for (III) thermoplastic resin matrixes or (IV) three-dimensional fiber constructions. The adhered construction (IV) is commonly used for FRP, but is excluded here because there have been

Table 4 T1000G material specification

Hoop strength σ_{θ} max (MPa)	Radial strength σ_r max (MPa)	Density ρ (kg/m ³)	Hoop modulus E_{θ} (GPa)	Radial modulus E_r (GPa)	Poisson's ratio ν
2675	15	1568	167.6	10	0.33

reports that cracking occurs along the reinforcing fibers rendering this method unusable in forming high strength disks.⁽³⁾ As a result of investigating (I) and (II), a multi-ring press fit construction (multi-ring construction) with an internal metal ring was selected. The densities and elastic moduli for each ring were varied by changing the fibers in each ring to create a construction that prevents delamination. The fibers used were Toray M40J high elasticity carbon fibers for the outer ring, Toray T300 general-purpose carbon fibers for the middle ring, and A7075-T6 for the inner metal ring.

Some filaments are cut when forming and machining the rings. Fig.8 shows the stress calculation results assuming an elastic modulus of 95%.

The stress calculation judgment criteria are that the stress in each part does not exceed the strength limit, and that the stress in the press fit portions does not become tensile. In other words, the press fit portions must not separate.

Fig.8 shows that the stress of the innermost hoop where the hoop stress is the highest for each ring does not exceed hoop strength, radial stress does not exceed radial strength, and the stress on the press fit portions is compressive. Therefore separation does not occur.

Based on the above, the rotor specifications were determined to be as follows: outside diameter, $\phi 270$ mm, maximum operating revolution, 50000 rpm, and limit revolution, 70000 rpm.

Table 5 shows these rotor specifications.

Table 5 Specification of flywheel rotor

Specification of flywheel rotor				
Item	Specification		Remarks	
Outside diameter	$\phi 270$ (mm)			
Inside diameter	$\phi 180$ (mm)			
Thickness	$26 \times 5 = 130$ (mm)			
Weight	6.87 (kg)			
Maximum energy stored	336 (Wh)		50000rpm	
	Outer ring	Middle ring	Inner ring	
Material	M40J	T300	A7075-T6	
Outside dia. (mm)	$\phi 270$	$\phi 235$	$\phi 190$	
Inside dia. (mm)	$\phi 235$	$\phi 190$	$\phi 180$	
Interference	400 (μm)		800 (μm)	
Vf of CFRP	60 (%)		60 (%)	

4. Rotor Fabrication

Fig.9 shows a single ring (thickness 26 mm) of the CFRP portion fabricated based on the rotor specifications shown in Table 5. The inner portion with a lighter color is the T300 ring, and the outer portion which appears blacker is the M40J ring. The M40J and T300 rings are mechanically press fit.

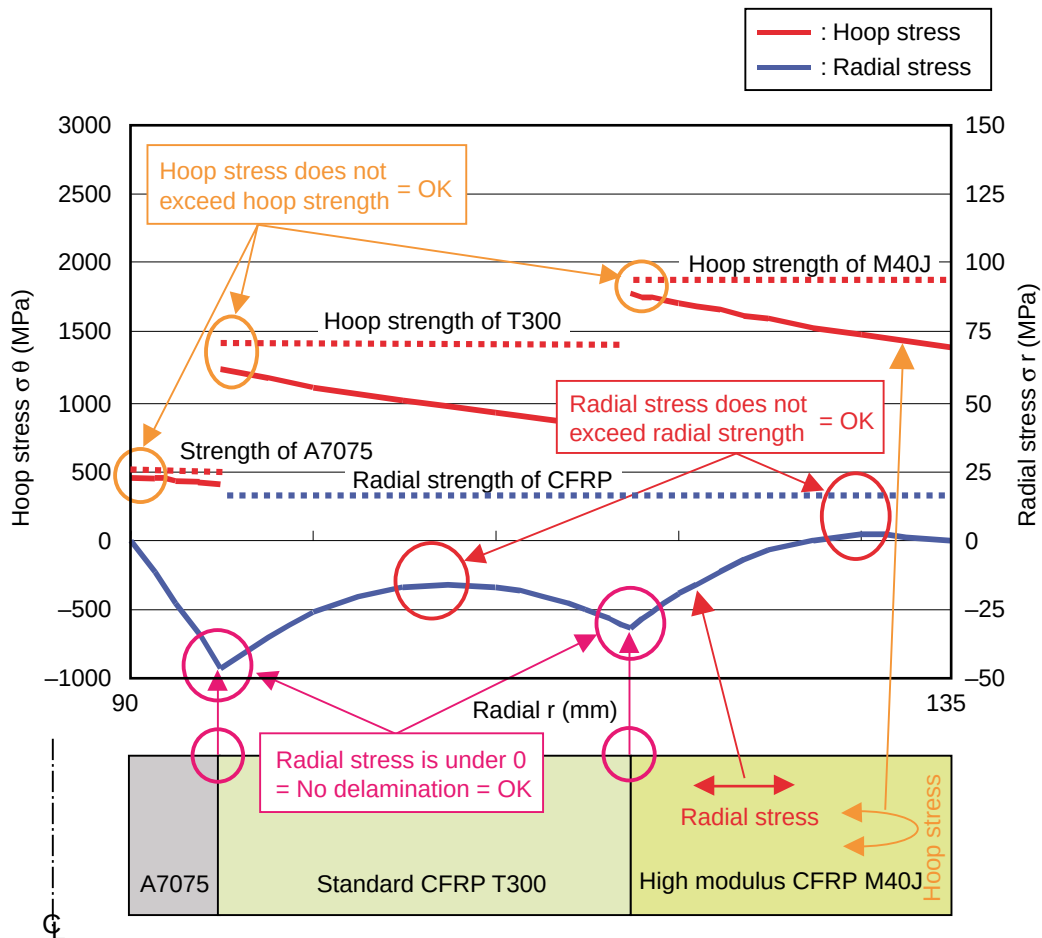


Fig.8 Stress calculation results

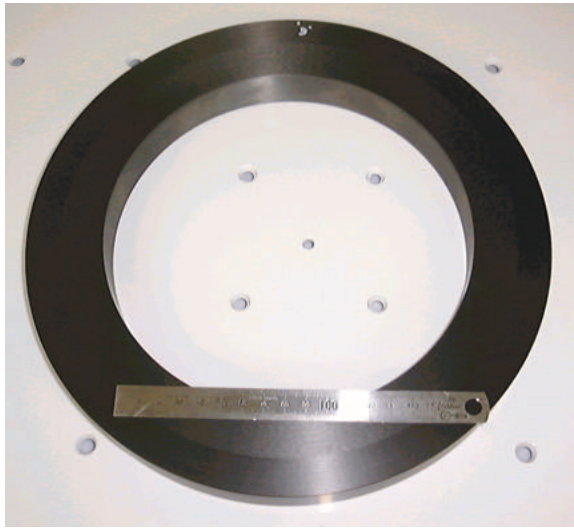


Fig.9 CFRP ring

5. Tests

5.1. Measuring the Change in Inside and Outside Diameter Due to Cooling

Carbon fibers have a negative coefficient of thermal expansion that is near zero. Thus, when carbon fibers are formed into a ring by filament winding and the ring is then cooled, only the resin contracts, and it does not contract in the direction of the fibers. Therefore, the outside diameter contracts and the inside diameter expands. The appropriateness of the dimensional change characteristics with respect to temperature was verified for the various rotor stress calculations. This was carried out by measuring the inside and outside diameters of the prototype rings at a normal temperature (27°C) and at -40°C and comparing these results with the calculated results of expansion and contraction.

Measurement was performed at four points on each of the three individual rings for a total of twelve points.

Fig.10 shows a comparison of the measured and calculated results, and shows that the measured and

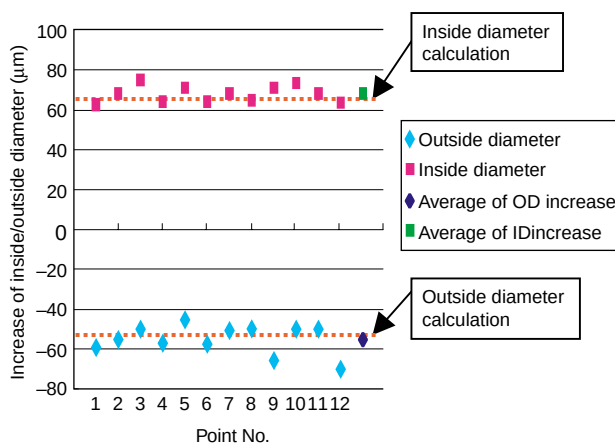


Fig.10 Increase of inside/outside diameter of CFRP ring at -40°C

calculated results match closely. Thus the calculated values for temperature change in inside and outside diameters were verified to be appropriate. Further cold fitting was judged to be possible with the press fitting margin setting determined assuming the expansion of the CFRP ring inner diameter.

Fig.11 shows the rotor constructed by cold fitting the innermost aluminum ring at the temperature of liquid nitrogen.



Fig.11 Complete rotor

5.2. Rotation Test up to 70000 rpm

A rotation test was conducted up to 70000 rpm to check whether the prototype rotors could achieve revolutions of 70000 rpm. Fig.12 shows the rotor in the set condition. The test was conducted in a vacuum using a spin tester with the rotor suspended and rotated by an air turbine.

Axial oscillation was monitored during the test, and the apparatus was set so that oscillation in excess of a certain value due to damage, delamination or any other reason would trigger a reverse in the air supply to the turbine thereby causing it to decelerate.

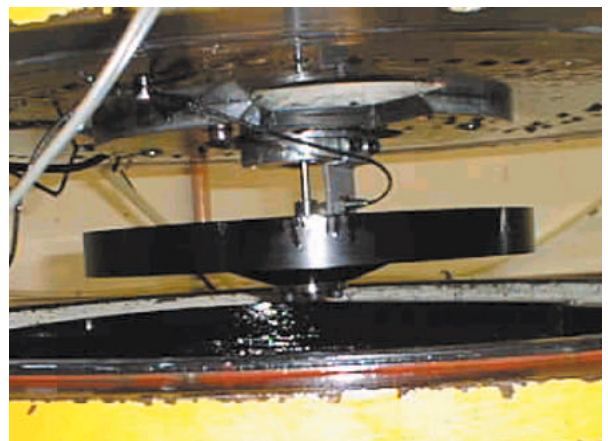


Fig.12 Rotor setting

The test conditions were set as follows, and the results of tests conducted using three rotors are shown in Fig.13.

- Degree of vacuum inside the spin tester: 133 Pa (1 Torr) or less
- Maintain 70000 rpm or more for 3 minutes

The horizontal axis shows the duration, and the vertical axis shows the revolutions. All rotors were able to reach and maintain 70000 rpm for 3 minutes without activating the trigger.

Based on these results, it was determined that a rotor could be obtained with a safety factor of 2.0 with respect to maximum operating revolutions of 50000 rpm.

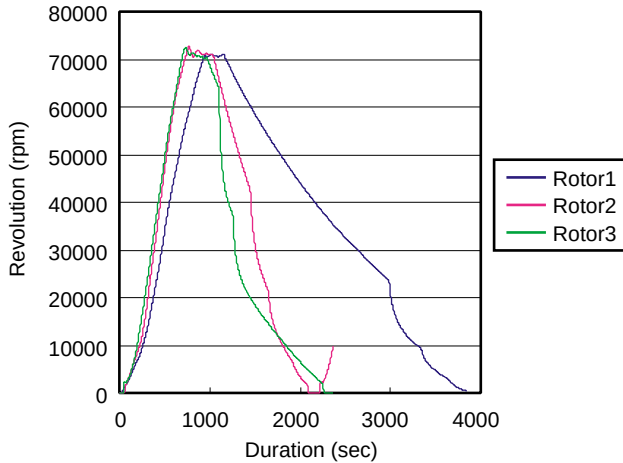


Fig.13 Results of 70000rpm testing

5.3. Rotational Strain Measurement

Aluminum rings were press fit into the individual prototype rings and strain gauges were attached to measure the rotational strain. These tests were conducted using a spin tester as in the 70000 rpm test. Fig.14 shows the rotor with the strain gauge attached, and Fig.15 shows a model drawing of the measurement points.

Hoop strain was measured at the three points of the center of the outer hoop and the center side and inner hoop of the CFRP portion. Radial strain was measured at one point at the center side of the CFRP portion.



Fig.14 Rotor with strain gauge

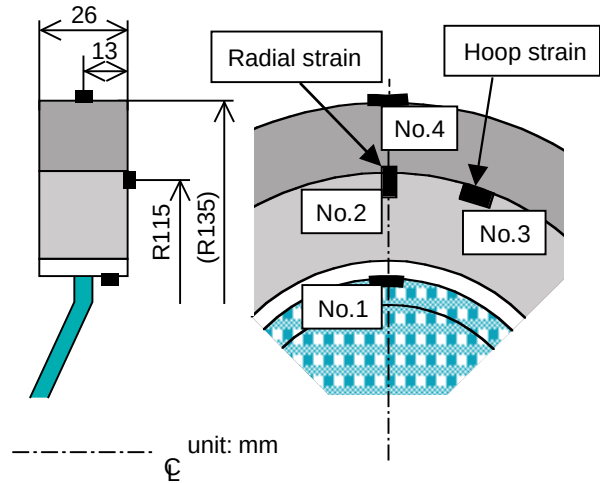


Fig.15 Outline of strain measurement points

Fig.16 shows a comparison of the strain measurement results and calculations up to 50 000 rpm. The strain was measured from two to four times at each point. This graph shows that extremely good measurement values were obtained, and that the measured strain closely matched the calculated values, thus verifying the validity of the filament winding formed anisotropic rotor design method.

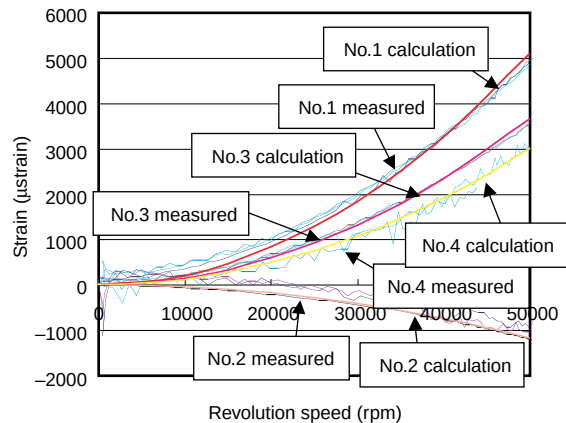


Fig.16 Comparison of calculation and measured strain

6. Conclusions

Regarding a flywheel battery rotor,

- (1) Materials and a construction that allow the specific energy to be increased were determined,
- (2) Rotor dimensions and shapes that satisfy the performance requirements were determined,
- (3) Rotation tests were conducted up to 70000 rpm, and
- (4) Strain was measured up to 50000 rpm.

These results proved that a rotor can be obtained which secures a safety factor of 2.0 over operating ranges up to 50000 rpm. Further, the calculated and measured values matched closely, thus verifying the validity of the design method.

Acknowledgment

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