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A COMPOSITE-RIM FLYWHEEL DESIGN*

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Abstract

This paper describes the design of a flywheel which incorporates a circumferentially wound rim. The design required the selection of materials and configuration for rim and hub, and a means of attaching the rim to the hub. The method used to fabricate prototype flywheels of this design is discussed.

1. INTRODUCTION

A hybrid heat engine/flywheel propulsion system requires a rotor which can both store energy efficiently and also fail in an easily contained manner. A high performance, filamentary composite rotor appears as a likely candidate to meet these design requirements.[1] An extensive listing of conceptual composite flywheel designs is given in Ref. 2. The three designs which utilize circumferentially wound composites are the (1) solid disk, (2) multi-ring disk, and (3) rim attached to a hub by bands or spokes.

Flat circumferentially wound disks have been fabricated from glass/epoxy and Kevlar/epoxy.[3,4] These rotors developed circumferential cracks during spin testing at rather low rotational speeds and hoop stress levels. Substantial radial tensile stresses are induced by the rim's rotation and under certain conditions by fabrication and thermal variations.[5,6] The rotors failed because the radial stresses developed in these spinning disks exceeded the low transverse strength of the circumferentially wound composites. By properly tailoring the axial thickness of a disk, the radial stress distribution can be depressed;[7,8] however, it appears that the radial stresses cannot be

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reduced sufficiently to permit a solid disk design.

The multi-ring flywheel design nests a set of hoop wound rings. The radial thickness of each ring is chosen to avoid a radial tensile failure. There is a radial displacement incompatibility between rings. The multi-ring concept still awaits the identification of an adequate method to accommodate this displacement incompatibility between rings without degrading the dynamic stability of the rotor.[9]

A flywheel incorporating a Kevlar 49 composite rim was designed, built, and tested by the Oak Ridge Y-12 Plant.[2,10] Although this flywheel did achieve a higher energy density per pound than the solid disks, it too failed in radial tension before the full potential of the hoop strength was utilized. This failure was due, in part, to a non-optimal choice of the rim's inner-to-outer radius ratio. The peak radial stress in a rotating rim with a fixed outer radius is decreased by decreasing the radial thickness of the rim.[11] Therefore, a proper choice of the inner-to-outer rim ratio will suppress a premature radial failure.

In this paper a flywheel incorporating a circumferentially wound rim is described. The flywheel was required to satisfy weight and size limitations which seem appropriate for utilization in a hybrid vehicle. The rim material and design were chosen to optimize the rotor's energy storage capacity. A method of attaching the rim to a hub is discussed as well as a method of fabricating prototype flywheels.

II. ANALYSIS OF A CONSTANT AXIAL THICKNESS RIM

A design which optimizes the energy storage of a flat, circumferentially wound rim was sought. This choice could then be used as a basis for designing a thick rim flywheel. The rim designs considered were (1) a rim fabricated from a single material, (2) a rim with lead ballast uniformly distributed along its interior edge, and (3) a rim composed of two concentrically wound materials (Fig. 1). The uniform lead ballast was modeled by requiring that the inner material of a two-material rim have a radial modulus E_r equal to that of lead, a vanishingly small circumferential modulus E_θ, and a zero in-plane Poisson's ratio $\nu_{r\theta}$. Therefore, designs 1 and 2 are special cases of the two-material rim. The rims were required to have an outside radius $r_3 = 10$ in., an axial height $H \le 3$ in., a weight

$W \leq 20$ lb, and a rotational speed $\omega \leq 40{,}000$ rpm. These constraints appeared appropriate for a rotor utilized in a hybrid vehicle which requires 0.5 kWh of mechanical energy storage. The inside radius r_1 and the interface radius r_2 were chosen to maximize the rim's energy storage without exceeding an anisotropic maximum stress failure criterion. Circumferentially wound rims reinforced with graphite, Kevlar 49, S-glass, and E-glass were considered. The nominal static properties of these composites were taken from Refs. 8, 12, 13 and are shown in Table 1. The details of the analysis are presented in Ref. 14.

III. RESULTS

The optimal choice of the rim's inside radius r_1 was identified for one-material rims. The optimal configuration as well as the associated energy storage at failure are given in Table 2 for rims reinforced with either graphite, Kevlar 49, S-glass or E-glass. The optimal graphite composite rim has $r_1/r_3 = 0.775$ and stores more energy (1.13 kWh) than any other rim reinforced with only one material.

The addition of a lead ballast to the composites listed in Table 1 did not improve the maximum energy storage capacity of rims reinforced with graphite, S-glass, or E-glass, but it did increase the maximum energy that can be stored in a Kevlar 49 composite rim by 21% (Table 3). However, this optimum ballasted Kevlar 49/epoxy rim still stores less energy than the optimum unballasted graphite/epoxy rim.

Two-material rims composed of the six possible combinations of composites listed in Table 1 were analyzed. It was found that the only synergistic combinations have Kevlar 49 composite as the outer rim material and either an S-glass or E-glass composite as the inner material (Table 3). The energy storage capacities of the optimal E-glass/Kevlar 49 and S-glass/Kevlar 49 composite rims were respectively 22% and 25% higher than the optimal Kevlar 49 composite rim. This synergistic effect occurred only when the inner composite had a hoop modulus lower than a Kevlar 49 composite. The effect was stronger as the density of the inner composite decreased. This suggests the use of Kevlar 29/epoxy, which has a hoop modulus of 7.25×10^6 psi[15] and a density equal to that of a Kevlar 49/epoxy, for the inner composite. Except for hoop modulus, the mechanical properties of Kevlar 29 and Kevlar 49 composites were assumed identical. The optimal Kevlar 29/ Kevlar 49 reinforced rim has an energy storage capacity 31% higher than the optimal Kevlar 49 composite rim (Table 3). However, this rim's energy storage capacity is 94% of that of the optimal graphite composite rim. Therefore, the optimal graphite/epoxy rim is the first choice for incorporation into a thick rim design.

IV. HUB DESIGN

The two conceptual methods for attaching a rim to a hub are (1) running spokes from the hub to the inside edge of the rim, and (2) wrapping bands over the rim and hub. The chief obstacle confronting the spoke design is the difficulty of attaching a spoke to the inside of a hoop wound composite rim without degrading the rim's strength. Filament wound band designs avoid this attachment problem and also appear easier to fabricate. For these reasons an overwrapped band design was chosen.

The results of the analysis of a flat rim, presented above, indicate that a graphite/epoxy rim with an inner-to-outer radius ratio of 0.775 should be chosen to optimize energy storage capacity. In this analysis the rim's cross section was assumed rectangular. A band wrapped design, however, requires that the rim shape be chosen in a manner which minimizes band bending. Accordingly, a semi-elliptical rim cross section with the flat edge facing the hub was selected. An axisymmetric finite element analysis was utilized to determine the three dimensional stress state induced in a rotating semi-elliptical torus. The hoop and radial stress distributions in this contoured rim were slightly perturbed from those given by a plane stress analysis of a rim with the same inner-to-outer radius ratio. The finite element analysis indicated that to increase energy storage the inside radius of the semi-elliptical torus should be 7.625 in. as compared with a radius of 7.75 in. for the constant axial height disk.

A rim with an inside radius of 7.625 in., a semi-major axis (aligned radially) of 2.375 in., and a semi-minor axis of 1.5 in. rotating at 40,000 rpm will have (1) a maximum hoop stress of 206 ksi and (2) a maximum radial stress of 4.7 ksi. These stress levels are just below the strength of the graphite/epoxy composite (Table 1). This rim weighs 16.4 lb and has a moment of inertia of 3.212 in.-lb-sec^2.

A rotor which stores 0.56 kWh of energy can deliver 0.5 kWh with a 3 to 1 speed reduction. This free-spinning, semi-elliptical torus will store the required 0.56 kWh at 31,800 rpm.

The method chosen for band attachment is shown in Fig. 2. This design, which was called the Wagon Wheel Flywheel, incorporates a tubular hub made from 7075-T6 aluminum and six circumferentially wound bands. Each band is directed radially and passes over the geometric center of the rim. The bands are axially segregated by hub castellations of varying heights (Fig. 3). The bands are not tightly bound to the hub. They can move radially through the hub, resisted only be frictional forces. The design therefore limits the radial forces applied to the hub. The bands are constrained against tangential motion relative to the hub by the walls of the slots. This constraint permits torque to be applied to the rim.

The bands are required (1) to be in tension along their entire length and (2) not to fail when the flywheel is rotating at 40,000 rpm. The bands are subject to inertial loads induced by the flywheel's rotation. Additional band loads may result from any radial displacement incompatibility between the rim and the overwrapped bands. The band loads are dependent upon the choice of the band material.

An analysis was performed to determine an appropriate choice for the band material. The hoop wound bands were modeled by a rod spinning about its center and with a half length L = 10 in. (Fig. 4). The radial displacement U of the rod at its tip can be easily found from one dimensional elasticity analysis and is given by

$$U = \frac{\rho\omega^2 L^3}{3E} \qquad (1)$$

where ρ = material density
ω = angular speed
E = longitudinal Young's modulus

This relation can be inverted to express specific stiffness

$$\frac{E}{\rho} = \frac{\omega^2 L^3}{3U} . \qquad (2)$$

At ω = 40,000 rpm, a finite element calculation performed for the chosen graphite/epoxy rim gave U = 0.088 in. The specific stiffness E/ρ required for the bar to achieve this displacement is 172×10^6 in. The specific hoop stiffness of the four filamentary composites under consideration is given in Table 1. The Kevlar 49/epoxy composite with a specific stiffness of 220. x 10^6 in. is the material of choice. Its specific stiffness exceeds the value desired by the least amount. A stiffer composite than that which was calculated to match the rim displacement will insure the rod will be in tension.

The maximum stress developed in a rotating rod with its tip displacement U prescribed occurs at its origin and can be shown to be

$$\sigma_r = \frac{EU}{L} + \frac{\rho(\omega L)^2}{6} . \qquad (3)$$

A Kevlar 49 epoxy rod rotating at 40,000 rpm with U = 0.088 in. will induce a maximum stress of 135,000 psi. This is well below the composite's strength (Table 1).

V. FABRICATION

The Wagon Wheel Flywheel design was fabricated by Brunswick Corporation of Lincoln, NB. A sand mandrel containing the flywheel hub was prepared. The outside diameter of the sand mandrel was machined to the desired inside diameter of the rim. The rim was filament wound using a four end roving of 3617 denier Thornel 300 graphite yarn and a DER 332-Jeffamine T403 (100:36 by weight) resin system cured at room temperature. A constant winding tension of 4 pounds was employed. The dams were removed and the rim was machined to the required semi-elliptical cross section. Slots were also machined into the face of the mandrel to contain the bands during filament winding. A sealer coat was applied to the rim. The bands were wound with a 4560 denier Kevlar 49 yarn and a DER 332-Jeffamine T403 resin system toughened with ATBN and with a DMP30 accelerator (100:36:6:1 by weight). After a room temperature cure, the sand mandrel was dissolved with water.

VI. SUMMARY

An analysis of constant thickness rims identified a graphite/epoxy rim with an inner-to-outer radius ratio of 0.775 as having the highest energy storage capacity of all design and material choices considered. This rim was used as the basis of a thick rim flywheel design. The Wagon Wheel design (Fig. 2) incorporates a tubular aluminum hub and over-wrapped Kevlar 49/epoxy bands. If this flywheel reaches a rotational speed of 32,000 rpm it could deliver 0.5 kWh of energy with a three to one speed reduction.

VII. ACKNOWLEDGEMENTS

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IX. BIOGRAPHY

E. David Reedy, Jr. obtained a B. S. in Mechanical Engineering from the University of Pennsylvania in 1972. He received a M. S. and Ph.D. in engineering from Harvard University. Since his graduation in 1976, he has been employed by Sandia Laboratories, Albuquerque, New Mexico. His responsibilities are in composite material research and development. His current interests include composite flywheel design and fracture of unidirectional composites.

TABLE 1. PROPERTIES OF CIRCUMFERENTIALLY WOUND COMPOSITES

Property	Graphite/ Epoxy	Kevlar 49/ Epoxy	S-Glass/ Epoxy	E-Glass/ Epoxy
Hoop tensile modulus 10^6 psi (10^4 MPa)	18. (12.4)	11. (7.6)	7.8 (5.4)	6. (4.1)
Radial tensile modulus 10^6 psi (10^4 MPa)	1.3 (0.88)	0.7 (0.45)	3. (2.1)	2. (1.4)
Poisson's ratio	0.27	0.34	0.25	0.25
Hoop tensile strength ksi (MPa)	220. (1520.)	200. (1380.)	226. (1560.)	150. (1040.)
Radial tensile strength ksi (MPa)	4.8 (33.1)	2.3 (15.9)	5.8 (40.)	4. (27.6)
Density lb/in.^{3} (10^3 Kg/m^3)	0.054 (1.50)	0.05 (1.38)	0.072 (1.99)	0.075 (2.08)
Specific hoop stiffness 10^6 in. (10^6 cm.)	333. 846.	220. 559.	108. 274.	80. 203.

TABLE 2. OPTIMAL DESIGNS FOR RIMS MADE OF A SINGLE MATERIAL.

$r_3 = 10$ in.

Reinforcement	r_1 in.	H in.	W lb	AT FAILURE ω_f rpm	kWh	$\frac{\text{w-hr}}{\text{lb}}$	$\frac{\text{w-hr}}{\text{in.}^3}$
Graphite	7.75	2.95	20.	39,850	1.13	56.7	1.22
Kevlar 49	8.35	3.00	14.3	38,680	0.81	56.6	0.86
S-Glass	8.40	3.00	20.	34,230	0.90	44.5	0.94
E-Glass	8.45	2.97	20.	27,300	0.57	28.5	0.61

TABLE 3. OPTIMAL DESIGNS FOR SYNERGISTIC COMBINATIONS WITH KEVLAR 49/EPOXY

$r_3 = 10$ in.

Inner Material	r_1 in.	r_2 in.	H in.	W lb	AT FAILURE ω_f rpm	kWh	$\frac{\text{w-hr}}{\text{lb}}$	$\frac{\text{w-hr}}{\text{in.}^3}$
Lead Ballast	7.8	7.83	2.99	20.	37,280	0.98	48.9	1.04
S-Glass/Epoxy	7.7	8.1	2.93	20.	37,940	1.01	50.4	1.10
E-Glass/Epoxy	7.7	8.	2.96	20.	37,570	0.99	49.5	1.07
Kevlar 29/Epoxy	7.5	8.5	2.91	20.	38,960	1.06	52.9	1.16

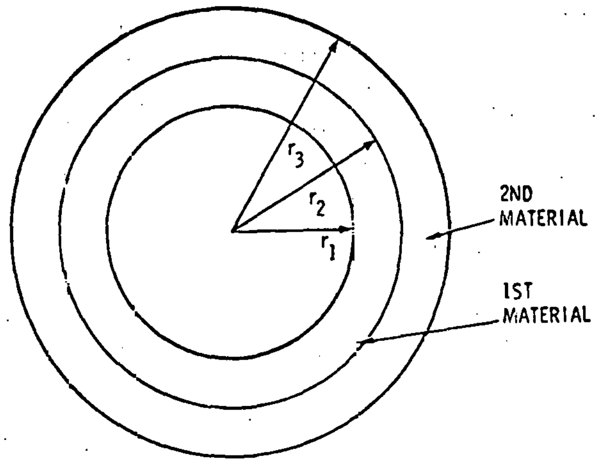

Fig. 1. Rim composed of two concentrically wound materials.

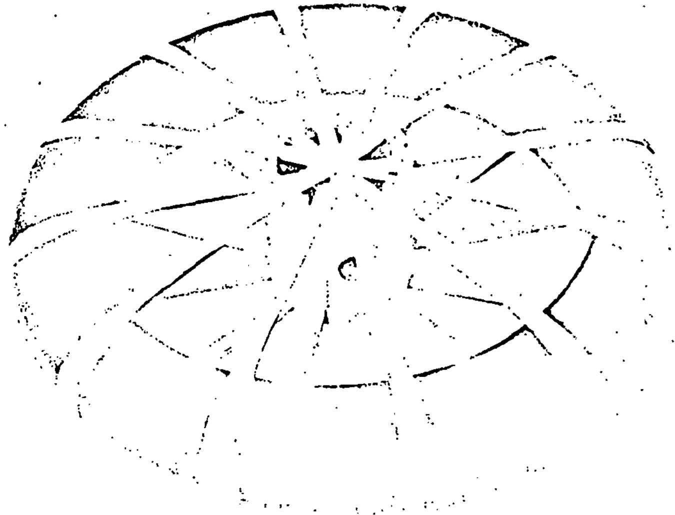

Fig. 2. Wagon Wheel Flywheel.

Fig. 3. Wagon Wheel Flywheel Hub.

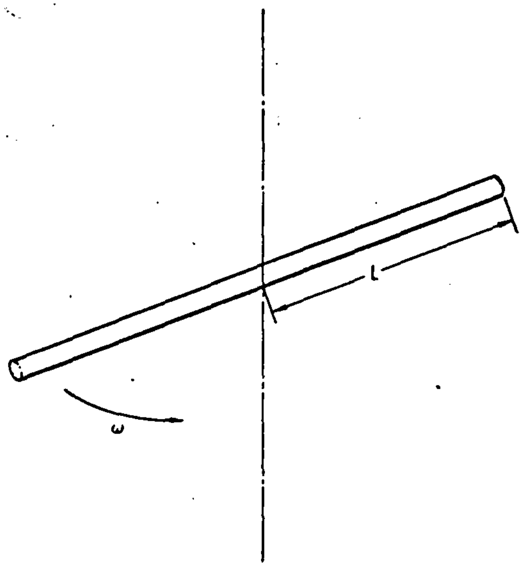

Fig. 4. Spinning rod model of band.