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**COMPOSITE FLYWHEEL DEVELOPMENT
COMPLETION REPORT
(May 1 - September 30, 1976)**

R. L. Huddleston
J. J. Kelly
C. E. Knight

May 1977

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**OAK RIDGE Y-12 PLANT
OAK RIDGE, TENNESSEE**

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Fabrication Systems Department
Y-12 Development Division

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COMPOSITE FLYWHEEL DEVELOPMENT

INTRODUCTION

The purpose of this report is to summarize Union Carbide Corporation-Nuclear Division's (UCC-ND) composite flywheel development program objectives and accomplishments for the period FY 1976-1976T. This work was conducted for and funded by the Advanced Physical Methods Branch (previously, Mechanical Energy Storage Branch), Division of Energy Storage Systems, Office of Conservation, ERDA, Washington, DC.

The project was initiated at the Oak Ridge Y-12 Plant^(a) on May 1, 1976, with the primary objective being to utilize the technology base at UCC-ND to design, fabricate, and performance test a prototype, vehicular-sized, composite flywheel in the five-month period available. The project was funded at \$68,000. The overall program scope encompasses development of both the flywheel and its containment; however, the FY 1976-1976T objective was directed toward development of the flywheel and testing it in existing facilities. The development effort was a successful one, leading to successful testing of a flywheel design which demonstrated an energy density performance of 10.1 Wh/lb during spin testing.

The initial application selected for development of the composite flywheel was the heat engine/flywheel hybrid propulsion system for a vehicle. This application was selected by the ERDA Advanced Physical Methods Branch staff because of its high potential for conservation of petroleum fuel in both the near and far-term time frames. Other applications, such as utility load leveling, represent potential areas for significant energy savings but require more extensive development programs and funding resources. Successful development of a high-performance, composite, vehicular flywheel represents one step along the development path leading toward larger, higher-energy storage flywheel applications.

The balance of this report summarizes the basic development plan followed: flywheel design and fabrication, spin testing, and results of these efforts.

FLYWHEEL DEVELOPMENT PLAN

A development plan consistent with completion of the program objectives by September 30, 1976 was prepared and followed. The plan, along with the completion dates achieved, is summarized in Figure 1. Funding was received approximately one month later than anticipated by the plan; hence, most of the completion dates were one month behind schedule.

(a) Operated by the Union Carbide Corporation's Nuclear Division for the US Energy Research and Development Administration.

The plan consisted of two flywheel design, fabricate, and test iterations, as outlined in Figure 1. However, late receipt of funding necessitated rescheduling the second iteration, denoted as the "ERDA Flywheel", into the FY 1977 program plan.

Figure 1. DEVELOPMENT PLAN, LEADING TO THE FABRICATION AND TEST OF A KEVLAR-49/EPOXY FLYWHEEL. (Completion by September 30, 1976)

FLYWHEEL DESIGN, ANALYSIS, AND FABRICATION

The necessity and urgency of national energy conservation is a well-recognized fact. Mechanical energy storage by means of rotating flywheels is one of the few methods for energy storage, and flywheels probably have the highest potential effectiveness for energy storage of any method now available.

This program is directed toward development of high-performance fiber-composite flywheels. The development entails design, analysis, fabrication, and testing of flywheels and their associated containment housings. This portion of the progress report is concerned with the design, analysis, and fabrication of the flywheel itself.

The design phase included both conceptual designs and a detailed design of selected concepts. Structural and dynamic analyses of the detailed designs were utilized to evaluate

their potential performance. A design was selected for fabrication and testing to verify its potential performance rating.

Conceptual Study

Flywheel Specifications - The objective of this program was to develop a high-efficiency, fiber-composite-material flywheel and containment housing for energy storage. Based on discussions with the ERDA Mechanical Energy Storage Branch staff, the most suitable starting point for development was a unit sized to the requirements of a heat engine/flywheel hybrid propulsion system for automobiles. This device requires a storage capacity of 0.5 - 0.7 kWh. The storage capacity and automotive requirements, along with information from the flywheel literature, helped formulate the flywheel's specifications and design objectives.

A literature survey revealed that if a total kinetic energy control theory or a similar control theory were used, a flywheel package with 0.5 kWh of available energy would be sufficient for an average-size automobile. If the flywheel operates in a three-to-one speed range for energy input/output, a maximum stored energy of 0.56 kWh will suffice. The three-to-one speed range utilizes 90% of the maximum stored energy and is a reasonable range for compatibility with the other automotive components.

There have been no known composite flywheels built that are able to demonstrate this energy level. The state-of-the-art metal flywheels reach an operating storage efficiency of 15 Wh/lb. (A goal of 20 Wh/lb was set as the design operating efficiency for the first flywheel developed in this program.) Achievement of this goal will be a first step in demonstrating the potential of fiber composite flywheels. The ultimate strength-to-density ratio of current Kevlar-49/epoxy^(b) composites project an ultimate efficiency of 80 Wh/lb in a thin-rim flywheel configuration. A 20-Wh/lb efficiency requires a flywheel weighing a total of 28 pounds.

The overall size of the flywheel for an automobile application should be limited to minimize packaging problems. If the diameter is too small, the flywheel speed will be very high and the weight must be attained with a long, cylindrical unit. (Approximate maximum diameter to be allowed is 20 inches, which is suitable for an automotive system and is compatible with existing composite winding equipment and facilities, and existing spin-test facilities.)

The design objective for dynamic characteristics is the elimination of as many critical frequencies as possible in the operating speed range. Rigid body modes should be below the lower speed bound and elastic body modes above the upper speed bound, and preferably above the ultimate failure speed. The flywheel specifications and design objectives that have been established are summarized in Table 1.

(b) Kevlar-49 is the trade name for DuPont's aramid fiber. The epoxy system utilized was Dow Chemical DER 332 resin with Jefferson Chemical T403 hardener.

Configurations - There are three basic flywheel configurations suitable for use with fiber composite materials: rim, rod, and disc. These designs are illustrated in Figure 2. Practical designs with these configurations must address some specific problem areas associated with each configuration:

1. The rim configuration must have a stiff connection to the axis, or some method of maintaining the spinning rim concentric with the rotational axis. The stiffness of the connecting system must be sufficient to maintain stability of the rotating rim.
2. The rod configuration requires a method of attachment to the drive shaft and a multiple-rod arrangement to improve the volumetric energy storage efficiency.
3. The disc configuration must employ some type of radial fiber reinforcement, thickness variation, or property variation to limit the composite stresses transverse to the fiber, within acceptable bounds.

Conceptual Designs - In the conceptual study, 18 conceptual designs and optional variations on some designs were cataloged. These conceptual designs are illustrated in Figures 3 - 5 and are briefly described in the paragraphs to follow.

Designs A through E (Figure 3) employ the rim configuration and devise various methods for attaching the rim and drive shaft. The rim in each case is a circumferentially wound cylinder. The radial thickness is designed to be thin enough to prevent delamination prior to hoop-fiber failure for the particular fiber composite of choice. The rim-axial thickness may be selected to provide the desired level of energy stored.

Design A uses a curved-spoke arbor. The arbor would be machined from a suitable metal, and the curvatures are designed to allow the spoke tips to match the radial displacement of the rim. The metal is designed to remain in the elastic stress/strain region. The spoke tips could be extended into leaves to provide for more uniform contact stress and an equivalent internal-pressure-ballast-type loading.

Design B incorporates wound-fiber-composite loops into a metal hub. The hoops are designed to have a minimum potential-energy shape in the rotational potential field between the attachments to the rim and hub. The plane of the profile is perpendicular to the rotation axis. The loops are bonded to the rim and must be mechanically restrained at the hub.

In Design C, the rim is coupled to the shaft by a tapered thickness disc of "constant stress" or similar profile. The disc has curved lips at the disc/rim interface to help alleviate strain incompatibilities.

Design D also uses fiber-composite loops or bands. In this design, the minimum-potential-energy shape in the rotational potential field is derived, with the axis of rotation lying in the plane of the profile. The loops are bonded to the rim and hub. A modification of this design would place the loops around the outside of the rim assuming that the rim outer surface was smoothly contoured at the corners. This design would improve both lateral and axial stiffness and reduce dependence on the loop-to-rim bond strength.

Design E uses a radial spoke arbor (minimum of three). In this design, strain differences (at speed) between the arbor and rim must be compensated. Either the rim must be initially deformed to fit an oversized arbor, or some type of built-in spring at the spokes' tips must accommodate the interference fit.

Designs F through I (Figure 4) employ the rod configuration in various design schemes. The rod elements of each design are of a continuous, unidirectional, composite material. The elements of each design would be individually fabricated and assembled in the flywheel form.

Design F is made up of two or more continuous unidirectional loops which carry some form of additional dead-weight loading. The dead-weight loading lowers the stored energy/unit weight efficiency, but raises the stored energy/unit volume ratio. The low stored energy/unit volume ratio is one of the primary disadvantages of the rod-configuration design.

The rod or bar of Design G is a relatively massive component. It is tapered to provide a more uniform stress distribution in the composite. The cross section should be a relatively thin rectangle for dynamic stability. The shaft-adaptor design is very critical to the potential success of this configuration.

The "brush" designs of Robenhorst, et al,¹ are identified as Design H. These designs attempt to improve the stored energy/unit volume ratio of the rod configuration.

A set of nested, continuous, unidirectional, composite loops make up Design I. Each loop has a minimum-potential-energy shape to avoid bending stresses. The loops are bonded in slots through the hub.

Designs J through R (Figure 5) are considered as disc configurations, although some designs are similar to the rim configuration. These designs attempt to either limit radial stresses in circumferentially wound composites or provide fiber orientation to accommodate the radial stresses.

Design J limits radial stresses by nesting individual rings. The thickness of each ring is designed to reach the limiting radial and circumferential stresses simultaneously. The rings are connected by polygonal hoops. These hoops could be wound progressively with the rings by using removable inserts.

Table 1 DESIGN SPECIFICATIONS FOR AN AUTOMOTIVE SYSTEM FLYWHEEL	
Available Stored Energy	- 0.50 kWh
Design Stored Energy	- 1.12 kWh (m.n)
Maximum Design Speed	- 40,000 rpm
Flywheel Weight	- 25 lb
Flywheel Size	- 20" D (approx)
Torque Load Capacity	- 100 ft lb
Dynamic Characteristic	- Minimize the Number of Critical Frequencies in the Operating Speed Range

(a) Rim Configuration.

(b) Rod Configuration.

(c) Disc Configuration.

Figure 2. BASIC FLYWHEEL CONFIGURATIONS.

(a) Design A - Curved-Spoke Arbor.

(b) Design B - Hoop-Plane Composite-Loop Arbor.

(c) Design C - Constant-Stress Disc.

(d) Design D - Axial-Plane Composite-Loop Arbor.

(e) Design E - Multi-Spoke Arbor.

Figure 3. CONCEPTUAL DESIGNS FOR THE FLYWHEEL. (Rim Configuration)

(f) Design F - Axial-Plane Loop Rod.

(g) Design G - Tapered Bar.

(h) Design H - Brush Designs. (From Robenhorst (1))

(j) Design J - Nested-Hoop Plane Loops.

Figure 4. CONCEPTUAL DESIGNS FOR THE FLYWHEEL. (Rod Configuration)

(j) Design J - Polygonal-Hoop-Connected Nested Rings.

(k) Design K - Graded-Density-Loaded Nested Rings.

(l) Design L - Multiple Geodesic Wrap.

(m) Design M - Multi-Angle Laminated Disc.

(n) Design N - Radial and Circumferential Winding.

(o) Design O - Prestressed and/or Variable-Thickness Disc.

(p) Design P - Nested-Cylinder Interleaved-Spoke Arbor.

(q) Design Q - Deltawrap Overwrapped Rim.

(r) Design R - Rockwell-Tape-Design Disc.

Figure 5. CONCEPTUAL DESIGNS FOR THE FLYWHEEL. (Disc Configuration)

Design K alters the radial-stress distribution by incorporating discrete layers of dead-weight loading in the proper proportions to limit the radial-stress buildup. The overall weight efficiency is lowered by the dead-weight layers.

A multiple geodesic pattern forms Design L. Parallel filament bands or tapes are wound in several geodesic patterns to produce multiple-fiber orientation at most points in the structure. Geodesics winding near the axis of rotation would be primarily radial in orientation, while geodesics winding near the outer diameter would be primarily circumferential in orientation. The number of different geodesics and the resulting thickness profile are tailored to produce a near-uniform fiber stress.

Design M is a laminated layup of unidirectional composite layers. The layers are oriented to produce a pseudoisotropic disc with a shaft molded in place.

Design N is an arrangement (not woven) of circumferential and radial fibers (also axial fibers for longer cylinders). The proportions are established by the desired geometry, material properties, and stress analysis. This design is described by Garber.²

Design O has a simple circumferentially wound composite, with either a variable-thickness or built-in prestress to limit the radial stress. Variable-thickness profiles were presented by Gerstle.³

Design P is similar to Design J except that the nested rings are supported by interleaved spokes. Detailed design on this concept must evaluate the capability for the metal spoke to displace sufficiently to follow each ring without applying concentrated loads to the rings.

Design Q applies an overwrap of radially oriented fibers to a circumferentially wound ring. The ring must be smoothly contoured on the outer diameter and corners. The overwrap is applied by using parallel-fiber-band-generation techniques developed for winding spherical pressure vessels by the "delta-axisymmetric" pattern.

Design R is reported by Notti, et al,⁴ as the Rockwell Composite Flywheel Tape-Wound design. The composite-material-tape-wound disc on each side of the mandrel is cut off, and the two discs are bonded together at their mandrel contact surfaces.

Design Evaluation and Selections - A method has been established for rating the cataloged conceptual designs and selecting the design(s) for further analysis and hardware development. Each concept was studied and its strengths and weaknesses listed. Based on these listings, a set of rating criteria were established as follows (numbers in parentheses represent the rating-scale value and the range for each criterion):

Rating Criterion Number	Criterion
1	Energy Stored/Unit Weight (0 - 10)
2	Energy Stored/Unit Volume (0 - 5)
3	Potential for Meeting the Objectives of this Program (0 - 10)
4	Ease of Fabrication (0 - 5)
5	Likelihood for Near-Term Successful Test (0 - 10)
6	Development Time Required (0 - 10)
7	Design and Analysis Complexity (0 - 10)
8	Economical End Product (0 - 5)
9	Potential of Significant Improvement with Further Development (0 - 5)
10	Likelihood of Minimal Dynamic Problems (0 - 10)
11	Torsional Strength and Stiffness Capability (0 - 5)

A decision table was formed and a subjective rating for each conceptual design estimated for all the criteria. Table 2 is a reproduction of the decision table. Some of

Table 2
DECISION TABLE FOR CONCEPTUAL DESIGN RANKING

Conceptual Design	Rating Criterion Number											Total Rating (85,max)	Rank
	1	2	3	4	5	6	7	8	9	10	11		
A	5	3	10	3	7	6	3	1	2	6	4	50	8
B	8	3	10	3	6	6	4	2	2	3	1	48	9
C	6	4	10	3	7	9	3	2	2	8	5	59	4
D	9	3	10	4	8	10	9	3	3	7	2	68	1
E	5	3	10	3	5	4	2	1	2	7	3	45	11
F	2	1	0	3	1	3	3	2	0	0	1	16	17
G	3	2	1	2	2	2	5	2	2	1	2	24	14
H	3	1	1	1	1	1	3	1	2	3	2	19	15
I	3	1	2	1	1	1	2	1	2	2	2	18	16
J	5	4	10	2	3	2	2	3	2	3	2	38	13
K	6	5	10	3	5	3	10	4	2	6	4	58	5
L	6	4	8	1	4	2	1	3	4	6	4	43	12
M	6	4	10	4	5	5	5	5	2	10	5	61	3
N	5	4	10	0	7	0	5	2	3	10	5	51	7
O	5	4	10	3	3	1	3	5	3	8	3	48	10
P	5	4	10	3	3	5	5	2	3	5	4	49	8
Q	10	3	10	4	7	6	3	4	5	9	4	65	2
R	6	4	8	3	6	5	3	3	3	10	5	56	6

the criteria and the rating-scale values were selected to meet the objectives of the FY 1976-1976T program. Thus, the final rating and selection of designs for development may be reevaluated at a later time. The rating values were summed for each design to determine an overall ranking for each conceptual design and, from these values, the designs to be developed were selected.

The five highest ranking designs were selected from the decision table. These designs were: D, Q, M, C, and K, respectively. The ratings for Design D were based on the plan of having the composite loops pass around the rim exterior. This form of Design D looks like a natural forerunner of Design Q. Design Q has the highest rating for potential improvement with further development.

Design D was selected for initial development in the FY 1976-1976T program, possibly leading to development of Design Q in following programs. Designs M, C, and K will be carried at a low activity level as backup designs.

Analysis and Design

Material Selection and Characterization - All the designs selected for development use the rim configuration as one component of the design. Since most of the energy is stored in the rim, it must have sufficient mass at the design speed to store the desired level of energy. The rim, therefore, cannot be an ideal "thin" rim. Also, in order to maintain a disc-type geometry to minimize dynamic problems, the axial rim thickness must be limited.

The material chosen for fabricating the rim must have a high strength-to-weight ratio to give a high-energy-stored/unit-weight efficiency ($E/W = \sigma/2\rho$ for a "thin" rim) where E represents the energy stored, W the weight, σ the strength, and ρ the specific weight. The material must also have the properties and characteristics needed to form a relatively thick rim. The candidate material systems for use in this program have been limited to those systems with which the Y-12 Composite Structures group has had fabrication experience.

Systems applicable to the flywheel are: graphite/epoxy, S-Glass/epoxy, E-Glass/epoxy and Kevlar-49/epoxy. Of these, the aramid fiber (Kevlar-49) in an epoxy matrix has the highest strength-to-weight ratio. The graphite/epoxy systems were eliminated because the strength-to-weight ratios are too low to give an adequate margin of capability above the program objectives.

One consideration for a thick rim is the radial and hoop-stress distributions in a rotating orthotropic ring. The inner-to-outer radius ratio, material density, material elastic properties, and angular velocity control these distributions. The equations for the stresses are developed in the book by Lekhnitskii.⁵ These equations were programmed, and the radius ratio which produced a simultaneous occurrence of ultimate radial and hoop stresses was determined for each material system. The minimum-radius ratios were 0.75 for both the glass/epoxy and the Kevlar-49/epoxy composites.

The rims can be wound and used at this radius ratio, provided the residual stresses are not significantly unfavorable. A very unfavorable distribution was expected but it was not realized, as will be shown in the subsequent section on **Residual Stress Analysis** (Page 26).

Since the minimum radius ratios were the same for both the glass/epoxy and Kevlar-49/epoxy composites, the Kevlar-49/epoxy system was selected for use because of its higher strength-to-weight ratio. Elastic properties for Kevlar-49/epoxy have previously been characterized. Hoop strength of wound rings has also been well established.

During this phase of the program, three full-size rims were fabricated. The first and third rims were wound with 1420-denier Kevlar-49 roving; the second rim was wound with 4560-denier Kevlar-49 roving. Approximately four spools of roving were used on each rim.

The strength of the roving used in each rim was determined by winding test rings from each spool. The test rings were nominally 6.000 inches in ID, 0.250 inch wide, and 0.125 inch thick. A ring was wound from each of the spools before the material was utilized in the rim. A remnant was left on the spool for a second test ring after most of the roving was applied to the rim. The test rings were mounted in a hydro burst fixture and the burst pressure/circumferential elongation curve recorded during the test. Both the hoop strength and elastic modulus were calculated from the recorded data. The pressure-elongation curves were linear to failure. Test results are summarized in Table 3.

The transverse tensile strength is normally significantly dependent on the epoxy resin system used. A special test was set up to measure the radial tensile strength of a thick ring. A segment of a ring was mounted in an apparatus illustrated schematically in Figure 6. A pure bending moment was applied to each end of the segment. This action produced a flexural hoop stress and a radial tensile stress if the moments were applied in the direction

shown. The radial-stress distribution is similar to that in a rotating ring and should, therefore, be indicative of the expected ultimate radial strength in the flywheel rim. Small-scale thick rings wound, for residual stress evaluation, provided specimens for this radial-strength measurement.

The stresses generated by the flexure of a curved beam are derived in a number of texts, one of which is authored by Seely and Smith.⁶ A curved beam of rectangular cross section is illustrated in Figure 7. The bending moment generates both circumferential and radial stresses in the curved beam, but the radial stresses are of most concern here. The circumferential stresses may be approximated by the standard beam-flexure equation. The radial stress at any radial position through the section located by Coordinate y is given by.

Table 3
STRENGTH AND MODULUS OF THE
KEVLAR 49 ROVING USED IN THE
RIM FABRICATION

Rim Number	Tensile Strength [ksi (CV)(1)]	Modulus [Msi (CV)]
1	263.5 (7.1%)	12.5 (3.1%)
2	225.6 (6.2%)	12.1 (1.7%)
3	262.7 (7.2%)	12.6 (3.5%)

(1) CV - coefficient of variation.

$$\sigma_r = \frac{Ma'}{Rta(R+y)} \left(1 - \frac{z''}{z}\right) \quad (1)$$

For rectangular cross sections:

$$z = -\frac{1}{a} \int_{-c}^{+c} \frac{y}{R+y} da = -1 + \frac{R}{2c} \left[\ln \left(\frac{R+c}{R-c} \right) \right], \text{ and} \quad (2)$$

$$z' = -\frac{1}{a'} \int_{-c}^y \frac{y}{R+y} da = -\frac{1}{c+y} \left[c+y - R \ln \left(\frac{R+y}{R-c} \right) \right] \quad (3)$$

where:

σ_r represents the radial stress, M the applied moment, a' the cross-hatched area (Figure 7), R the radius of curvature to the section centroid, t the beam thickness, a the total section area, and y the coordinate referenced to the centroid.

The maximum radial stress will occur at the location of the neutral axis. The y coordinate of the neutral axis is given by:

$$y_0 = -zR/(z+1).$$

Figure 6. SPECIAL-RING-SEGMENT TRANSVERSE TENSILE TEST.

Figure 7. CURVED-BEAM SEGMENT WITH THE RECTANGULAR CROSS SECTION LOADED IN PURE BENDING.

The negative sign indicates that the neutral axis is nearer to the inside radius. This value of y is used in Equation 3 to find z' , then in Equation 1 to find $\sigma_r \max$.

As a check on the applicability of these equations for orthotropic composite materials, a finite-element model was generated and analyzed. The model, given in Figure 8, is a

Figure 8. MESH FOR A FINITE-ELEMENT ANALYSIS OF THE COMPOSITE-RADIAL-STRENGTH, FOUR-POINT FLEXURE TEST.

segment from a ring 6.000 inches in ID, 1.09 inches in radial thickness, and 0.250 inch wide. The radial-stress distributions from Equation 1 and the finite-element model are compared in Figure 9. The agreement is good and lends confidence to the use of Equations 1 through 4 for calculating the radial tensile strength.

Segments from some of the thick rings wound for residual-stress analysis were tested. In addition, some rings were wound with different resin systems in an attempt to find a higher radial-strength composite. The radial strength was believed to be ~ 4 ksi (based on literature data), but the results summarized in Table 4 indicate that the radial strength is ~ 1 ksi for the composite systems used here. This low radial strength severely limits the rim-radius ratio that would not delaminate before circumferential stress failure. As discussed in subsequent parts in this report, the first flywheel tested did delaminate prior to attaining the expected design point.

Structural Analysis - Detailed stress analyses for Designs D and C were performed by using the finite-element method. The computer code that was used solves two-dimensional axisymmetric or plane-stress problems with orthotropic, temperature-dependent, material properties. Design D required both axisymmetric and plane-stress modeling. A code modification was inserted to apply the rotational-body-force loading to plane-stress cases. Minor modifications were made to allow the code to calculate the weight and moment of inertia of the input model.

Figure 9. RADIAL-STRESS DISTRIBUTIONS FROM FINITE-ELEMENT ANALYSIS AND EQUATION 1.

Table 4
TRANSVERSE TENSILE STRENGTH OF
KEVLAR 49 COMPOSITES

Epoxy Resin ⁽¹⁾	Composite Transverse Strength [psi (CV) ⁽²⁾]
DER332/T403	1100 (29.2%)
XD7818/XD7575.02/	
ERL4206/TONOX 60-40	260 (22.1%)
XD7818/D230/A298	580 (32.1%)

(1) DER332, XD7818, and XD7575.02 are products of the Dow Chemical Company; T403, D230, and A298 are products of Jefferson Chemical; ERL4206 is a product of Union Carbide Corporation, and TONOX 60-40 is a product of the Uniroyal Company.

(2) CV - coefficient of variation.

Figure 10. FINITE-ELEMENT MODEL OF THE DESIGN D FLYWHEEL, SYMMETRIC ABOUT THE R AXIS.

A finite-element model of Design D was prepared through the use of a mesh-generation program. One such model is reproduced in Figure 10. The structure is symmetric about the R axis; therefore, only half the structure was modeled. The Z axis is the center line of rotation. Notice that the rim is essentially rectangular in cross section, but has radiused outer corners. The mesh was input along with the material properties and angular velocity. The model was run first in the axisymmetric mode. Stresses and displacements in the rim were computed. The rim-to-loop interface displacements were used as input data, and the case was rerun for the plane-stress mode. This run determined the stresses and displacements in the loop. An accurate model of Design D would require a 3-D analysis, but the procedure described here should be a valid approximation.

Several design and analysis iterations led to the design represented by the model in Figure 10. The rim has a 7.500-inch inside radius and a 10.000-inch outside radius. The total axial rim height is 3.400 inches, and the outer surface corners have a 1.000-inch radius. The band has a 0.100-inch uniform thickness passing around the rim outer contour. A thin interface layer was placed between the rim and band in order to determine the required bond strength.

Results from the last analysis of Design D, modeled in Figure 10, are presented in Figures 11 through 14. As mentioned, the model was run first in the axisymmetric mode. This run determines the stresses in the rim due to centrifugal loading and its corresponding displacements. In this step of the analysis, the material density of the band and interface materials was zeroed. The elastic properties were also factored down to allow for a free expansion of the rim. This procedure is probably a closer approximation to the real case than the full-rim-overwrap covering provided by the axisymmetric model. Computed stresses in the rim at an angular velocity of 2827 rad/sec (27,000 rpm) are given in the graph of Figure 11. At this angular velocity, the flywheel stores 0.56 kWh of energy and has a theoretical weight efficiency of 22.8 Wh/lb. Figure 11, Graph a, is a plot of the hoop-stress distribution; Graph b is a plot of the radial-stress distribution.

Rim displacements in the axisymmetric computer code run were recorded and used as specified displacements on the rim cross section in the plane-stress analysis. The band and interface-material densities and

Figure 11. STRESS DISTRIBUTIONS ALONG A RADIAL LINE IN THE PLANE OF SYMMETRY THROUGH THE RIM THICKNESS. (Obtained by Finite-Element Analysis of the Model in Figure 9)

Figure 12. SHEAR AND NORMAL STRESSES ALONG THE RIM/BAND INTERFACE WITH AN AVERAGE EPOXY INTERFACE MATERIAL IN A PLANE-STRAIN ANALYSIS.

Figure 13. SHEAR AND NORMAL STRESSES ALONG THE RIM/BAND INTERFACE WITH A LOW SHEAR-STIFFNESS INTERFACE MATERIAL BETWEEN THE RIM INSIDE RADIUS AND THE RIM-CORNER RADIUS, AND AN AVERAGE EPOXY MATERIAL ALONG THE REMAINDER OF THE INTERFACE IN A PLANE-STRAIN ANALYSIS.

Figure 14. LONGITUDINAL BAND STRESSES IN THE CASES OF THE INTERFACES OF FIGURES 12 AND 13.

properties were restored and the plane-stress problem solved for the same angular velocity. The interface material may be represented in many different ways. It could be modeled as a bandline with average epoxy properties, low modulus epoxy adhesive properties, or near-incompressible rubber-like properties; or, portions of the interface could be free and other portions bonded. All of these may be plane-stress or plane-strain analyses with property transformations.

Results from two of the modeled interfaced schemes are presented here. The interface shear and normal stresses are plotted in Figures 12 and 13. Figure 12 presents the shear-stress distribution and the normal-stress distribution when the interface is modeled by an average epoxy ($E = 500,000$ psi, $\nu = 0.35$) in plane strain; Figure 13 presents the shear-stress distribution and the normal-stress distribution when the interface is not bonded along the rim side out to the beginning of the corner radius, with the remainder at the interface modeled by an average epoxy in plane strain.

Stresses in the band are plotted in Figure 14. The average longitudinal stress in the fully bonded band is plotted in Graph a, the average longitudinal stress in the band that is not bonded on the rim side is plotted in Graph b.

These results from the analysis of Design D looked very promising. Based on preliminary property data, the design appeared to be more than adequate to reach the design operating goal. The radial tensile strength test data reported previously were not available until after the first flywheel of this design had been fabricated. These data indicated that the flywheel would likely fail by delamination, premature to the design level.

Design M was not analyzed, but can be evaluated by using composite laminate theory. This approach is a relatively straightforward, calculator-type analysis and will not require a large amount of analysis time if the design is utilized in the program.

A finite-element mesh of a Design C flywheel is given in Figure 15. Again, the structure is symmetric about the R axis, and the Z axis is the axis of rotation. The mesh shown

Figure 15. FINITE-ELEMENT MODEL OF THE DESIGN C FLYWHEEL, SYMMETRIC ABOUT THE R AXIS.

represents the best form of a detailed Design C obtained thus far. Hyperbolic, constant-stress, and linear-taper-thickness profiles for the disc arbor were evaluated. The constant-stress profile is best in the disc portion of the arbor if the disc is solid. A bore hole through the disc is required to be compatible with existing spin-test equipment. A bore hole with added thickness reinforcement becomes one of the arbor weak points, regardless of the thickness profile used. However, the lip region has the highest stresses and is, therefore, the critical design point.

Several design and analysis iterations led to the design modeled in Figure 15. As mentioned, the model is symmetric about the R axis; and, therefore, all nodal points in the model which lie on the R axis are constrained so that only R direction displacements are allowed. The rim has a rectangular cross section, a 7.500-inch inside radius, a 10.000-inch outside radius, and a total height of 3.2 inches (a 1.6-inch height was modeled). The split-lip disc arbor has a linearly tapered thickness from 1.000 inch at the hub to 0.050 inch at a radius from the Z axis of 6.75 inches. The lips are 0.025 inch thick and have a 0.750-inch radius with a 0.250-inch-long cylindrical extension in contact with the rim. The design speed required to store 0.56 kWh of energy is 2780 rad/sec (26,500 rpm).

Stresses in the rim at 2780 rad/sec are very close to those shown in Figure 9 for the Design D flywheel. Therefore, the primary concern in Design C is the split-lip disc arbor. In the analysis, the rim properties are those of Kevlar-49/epoxy, and the arbor is represented by properties for aluminum.

The stresses through most of the disc and hub do not exceed 40 ksi at the design speed. The highest stresses occur in the lip region. An enlarged view of this region is illustrated in Figure 16. Some of the highest effective stress values which result from the combined stress state in this elastic analysis are shown. When the effective stress exceeds the yield stress, the material plastically strains. Some small areas in this model thus indicate that the yield stress will be exceeded, regardless of the aluminum alloy chosen for use.

Based on these results, this arbor design could reach the design operating speed, assuming the rim did not fail. An interference fit at assembly between the arbor and rim probably would keep the arbor material in the elastic-strain range. The potential of this design to reach much higher speeds, however, is limited by the onset of significant plastically strained areas. There are no plans at this time to fabricate a flywheel with this design.

Dynamic Analysis - Work is underway to analyze a three-dimensional model of Design D and determine its natural vibration frequencies and mode shapes. The 3-D code, NONSAP,⁷ performs two or three-dimensional nonlinear static and dynamic structural analyses. The current version of the program is being modified to account for several factors of importance to the flywheel analysis. The three-dimensional element in the program only considers the material nonlinearity. Orthotropic material capability will be added; and, for the present, the axes of orthotropy will be aligned with the element's local coordinates. Addition of "geometric" stiffening will be made if a preliminary analysis indicates that this nonlinear term could be significant. Rotational body-force loading must also be added to the program.

Figure 16. LOCATION OF HIGH, EFFECTIVE STRESS VALUES IN THE LIP REGION OF A DISC-ARBOR FLYWHEEL DESIGN BY FINITE-ELEMENT ANALYSIS.

While code modifications were being programmed and inserted, a model of Design D was prepared. The model was submitted and run in the unaltered code version. This model was for analysis of the body's critical frequencies in the operating speed range. The model is a 180-degree portion of the rim and band sets, which should provide for all the possible vibration modes. Difficulties were encountered in running the model, hence no reliable results are available at this time.

A different analysis method was approached to obtain the expected natural frequencies. The model is formulated by transfer matrix theory, as presented in the book by Pestel

and Leckie.⁶ Results from this model indicated three natural frequencies in the design speed range. An axial vibration of the rim relative to the hub occurs at ~ 800 rpm; a torsional vibration of the rim relative to the hub occurs at $\sim 10,500$ rpm, and a wobble vibration of the rim relative to the hub occurs at $\sim 13,400$ rpm. These values appear to be in reasonable agreement with experimental data on the first flywheel test. Details on these results and formulation of the dynamic analyses are presented in a report by Dodge and Fanning.⁸

Design for Fabrication - The first flywheel design to be fabricated is illustrated in Figure 17. The rim was constructed of the Kevlar-49/epoxy composite. The wound composite was room-temperature cured to minimize residual stresses. With an outside diameter of 20 inches, the allowable radius ratio fixes the inside diameter at 15 inches, or a 2.5-inch-thick-wall rim.

Figure 17. DESIGN OF THE FIRST FLYWHEEL TO BE FABRICATED.

The design goal of 20 Wh/lb established the total computed flywheel design weight at 28 pounds. The bands have a cross section that is 0.750 inch wide by 0.100 inch thick. The eight bands in the set weigh approximately 1.6 pounds. If 2.0 pounds were allowed for the hub, a 23.4-pound rim would leave a 1.0-pound margin for design changes in the bands or hub should further analyses show any deficiencies. With these constraints, the rim was 3.400 inches wide.

The first hub design is illustrated in an isometric view in Figure 18. The material is of Type 6061-T6 aluminum. A finite-element model of the hub by itself showed a more-than-adequate speed capability. In addition, the design provided a self-locking ability to the shaft that fits through the bore. When the hub is spinning, the stresses in the end plates try to expand the bore, and this action applies a bending moment to the ends of the connecting cylinder, causing it to close on the shaft. The lugs on the end plates of the hub help maintain the bands in their position during spinning.

Figure 18. HUB DESIGN FOR THE BANDWRAP FLYWHEEL.

Fabrication

Process Development - The conceptual designs chosen for further study and development employ a thick-rim element for the primary energy storage capacity. Therefore, initial development was directed toward fabrication of thick rims.

A six-inch-diameter cylindrical mandrel was available on which to wind test rings. A conventional wet-winding process line was set up with a circumferential winding machine, seen in Figure 19. The Kevlar spool was mounted on a rotating pretensioning device. The strand passed to the resin impregnation drum, then to the final fiber tension controller. The fiber tension controller was closed-loop controlled from the input from the tension monitor. The strand passed to the winding machine guide pulley and was wound on the mandrel.

A room-temperature-curing epoxy resin system (Dow Chemical DER 332/Jefferson Chemical Jeffamine T403) was selected for test winding. Prior experience with highly orthotropic fibers such as graphite or Kevlar-49 has indicated that a room-temperature-curing resin minimizes problems with microcracking, delaminations, and residual stresses.

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Figure 19. A WET-WINDING PROCESS LINE WITH A CIRCUMFERENTIAL WINDING MACHINE.

The first cylinder was wound to a width of two inches and a thickness of one inch. This thickness produced a 0.75 radius ratio. A circumferential load rate of 0.024 in/rev gave a side-by-side strand laydown for the 380-denier Kevlar strand. At wind completion, the part was allowed to rotate until resin gelation had taken place, then the part was allowed to cure for three days before removing the mandrel. The winding tension was held constant at 500 grams. The resulting composite rim was 74.2% by volume fiber, 23.7% by volume resin, and a 2.1% void content. The part was easily removed from the mandrel, and visual inspection found no delaminations or localized fiber buckling. X-ray inspection of the part also showed no delaminations. These results were very encouraging, but the ability to scale to larger rims was uncertain.

The second 6-inch ring was wound to 1.5 inches in thickness (a 0.667 radius ratio) to ascertain if the 0.75-radius-ratio, 6-inch ring was on the threshold of delamination. A larger strand (4560-denier Kevlar roving) was used because it is more suitable for winding flywheel rims. The winding tension was a constant 4500 grams. It could not be held proportional to the tension at the previous winding for the higher-denier strand because the machine torque capacity was too low. The wound part was cured for three days, then removed from the mandrel. The resulting rim was 69.0% by volume fiber, 26.4% by volume resin, and a 4.6% void. The higher void content was most likely due to the greater difficulty in impregnating the larger strand. Visual and X-ray inspection again indicated a high-quality composite, with no delaminations or fiber buckling.

Full-scale flywheel rims must be wound in a larger circumferential winding machine. A lathe modified with an automatic reversing lead mechanism provided both the diameter and torque capacity required.

A scaled-up ring was wound on another available mandrel which was 13 inches in diameter and 1.9 inches wide. The ring was wound with 4560-denier Kevlar roving at a 4500-gram tension to 18 inches in diameter (0.722 radius ratio). The composite ring was 70.7% by volume fiber, 24.4% by volume resin, and had a 4.9% void content. Visual and X-ray inspection again indicated a high-quality part, with no delaminations or fiber buckling. These results provided additional confidence in the capability to fabricate a 15-inch-ID by 20-inch-OD composite rim.

Before any residual-stress measurements were made on the rings fabricated thus far, a variable-winding-tension profile was tried on another 6-inch-diameter ring. The profile was linearly increasing from 2000 to 9500 grams through a one-inch wall thickness. This profile was selected to roughly approximate the reverse of the expected residual-stress distribution due to resin shrinkage. After curing and removal from the mandrel, a serious region of localized fiber buckling was observed. (The part is shown in Figure 20.)

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Figure 20. LOCALIZED FIBER BUCKLING IN A RING, DUE TO AN IMPROPER PROGRAMMED WINDING TENSION.

Obviously, this condition is not acceptable in a flywheel. Based on this result and the results from the residual stress analysis reported in the next section, the tension profile used for this ring is far from the correct profile.

Another six-inch-diameter ring was wound. The fiber was 1420-denier Kevlar yarn tensioned at 1800 grams (proportioned to the 500-gram tension on 380-denier yarn). The ring was wound to a one-inch wall thickness on the lathe with the automatic reversing lead mechanism. The composite ring was 71.8% by volume fiber, 24.4% by volume resin, and had a 3.8% void content.

Residual Stress Analysis - The residual stresses in a composite structure may be of critical importance, especially in the direction transverse to the fiber axis. The described process studies verified the capability to fabricate thick rims from the Kevlar-49/epoxy composite. The residual stresses present in the fabricated rims were determined to evaluate the potential flywheel performance.

The procedure for evaluating residual stresses is: A test ring is parted from the fabricated rim; the ring is instrumented with strain gages, and strain changes, recorded as thin layers of material, are machined off the diameter. The strain changes are used to calculate the residual stresses that were present initially.

The stresses were calculated incrementally, corresponding to the strain change produced when an increment of material was removed. The ring is illustrated schematically in Figure 21 after removal of the i^{th} layer. The

radial stress, σ_{r_i} , was calculated from the equation for the tangential stresses in a thick-wall, orthotropic ring subjected to an external pressure and the total strain change from the initial reading. All the rings examined thus far have been strain gaged on the inside radius. The tangential stress change on the inside radius produced by the strain change was determined by:

$$\Delta\sigma_\theta = \Delta\epsilon_\theta E_\theta = \frac{2KaK - 1_rK + 1}{r2K - a2K} \sigma_{r_i}$$

Rearranging to yield the radial stress gives:

Figure 21. TEST-RING RESIDUAL STRESS ANALYSIS MODEL.

$$\sigma_{r_i} = \frac{r^2 K - a^2 K}{2KaK - 1rK + 1} \Delta\epsilon_\theta E_\theta$$

where

σ_{r_i} represents the radial "initial" stress existing inside the i^{th} layer,

r the radius to Layer i ,

a the ring inside radius,

K a constant equal to $\sqrt{E_\theta/E_r}$ (where E_θ is the tangential modulus of elasticity and E_r the radial modulus of elasticity), and

$\Delta\epsilon_\theta$ the tangential strain change from its initial zero reading.

Thus, the radial stresses were determined at all incremental points through the wall. The tangential stress in any layer, i , was found by calculating the membrane stress in the cylindrical layer due to the differential pressure ($\sigma_{r_{i-1}} - \sigma_{r_i}$), thus:

$$\sigma_{\theta_i} = (\sigma_{r_{i-1}} - \sigma_{r_i}) \frac{r}{\Delta r}$$

where:

σ_{θ_i} represents the tangential "initial" stress existing in the i^{th} layer, and Δr the layer thickness. The radial and tangential "initial" or residual stresses are plotted versus radial position to display their distribution.

The residual-stress distributions determined for the first three test rings are represented by the graphs of Figures 22 through 24. The shape of the distributions is similar for each ring, the distributions differing only in their magnitudes. Overall, these are very satisfactory distributions. Radial stresses were mostly compressive and very low. The tangential stresses were high near both boundaries, but most serious near the outside radius. No adequate explanation or theory exists yet for the formation of these distributions. The most obvious solution for reducing the outer surface residual stress was to wind an excess of material and then machine it away after curing. The type of distributions found were not as expected, based on prior experience. The expected distributions would be similar to the curved-beam bending stresses. The radial stress would be zero at both boundaries and gradually build to a peak near the center of the wall. The tangential stress would be an almost linear bending distribution, with compression on one boundary and tension on the other. The expected residual produced by resin shrinkage and/or thermal contractions from an elevated-temperature cure has tensile radial stresses through the wall and tensile tangential stresses on the inner surface.

Figure 22. RESIDUAL STRESSES IN A SIX-INCH-INSIDE-DIAMETER BY ONE-INCH-THICK CIRC-WOUND KEVLAR-49/EPOXY RING.

Figure 23. RESIDUAL STRESSES IN A SIX-INCH-INSIDE-DIAMETER BY 1.5-INCH-THICK CIRC-WOUND KEVLAR-49/EPOXY RING.

Figure 24. RESIDUAL STRESSES IN A THIRTEEN-INCH-INSIDE-DIAMETER BY 2.5-INCH-THICK CIRC-WOUND KEVLAR-49/EPOXY RING.

Figure 25. RESIDUAL STRESSES IN THE FIRST FULL-SIZE RIM. (A 15-Inch-ID by 20-Inch-OD Circ-Wound Kevlar-49/Epoxy Composite)

Residual stresses were measured in two full-size rims wound for flywheel manufacture and test. A ring was parted from the rim and instrumented similar to that described previously. However, in this case, the strain gages were mounted on the side of the ring so that the residual stresses on both the inner and outer surfaces could be determined with better accuracy. Machining of layers was conducted first from the outer surface incrementally toward the strain-gage location, then from the inner surface toward the strain-gage location. The data were reduced in a manner similar to the method described previously. However, the strain increment that results when a layer is removed in this case depends on both the radial and tangential stress components at the strain-gage location.

Results from the two rims were very close and similar to the results on test rings shown in Figures 22 - 24. Stress distributions from the first full-size rim are indicated in Figure 25. When the rim was used in a flywheel, the outer surface was machined to a contour, and approximately 0.125 inch of stock removed. This removal also removes the high residual stress layers on the outside. On the tangential stress curve in Figure 25, this removal occurs at approximately the 8-ksi stress level. Based on these results, the finished rim used in the flywheel did not have any significant residual stress problem.

Tooling Design - The tooling required to fabricate the circ (circumferentially)-wound rims was relatively simple. A steel cylinder, 15 inches in diameter by 4.100 inches long, was fitted with two side plates. The side plates had a 0.050-inch recess for the cylinder and had an outside diameter of 22 inches. A two-inch-diameter shaft passed through the side plates. The side plates were 0.750 inch in thickness to provide adequate stiffness to resist the side pressure during winding. The wound rim could then be up to 22 inches in outside diameter by 4 inches wide.

The tooling for winding bands around the rim was more complex. A form had to be provided for each band to be wound. This form controlled the fiber-laydown position and the overall dimensions of the band. A multipiece disc was fitted between the hub described earlier and the rim. This disc positioned the rim concentric with the hub and aligned the band forms in the correct position relative to the hub features. Matched multipiece discs were required for each side of the flywheel.

Rim Fabrication - The rim tooling was received and the first full-scale rim was wound. The winding setup is shown in Figure 26. The fiber was 1420-denier Kevlar-49 yarn, the resin system was DER 332/T403 cured at room temperature, and the winding tension was held constant at 1800 grams. The rim was wound to approximately 20.2 inches in outside diameter so that the high residual stress region at the outside could be machined away. The total winding time was 28 hours at a laydown rate of 28 rpm. A view of the rim is seen in Figure 27.

Two other rims were wound, one using 1420-denier Kevlar yarn and the other using 4560-denier Kevlar yarn. The wound rim used 4560-denier yarn with 6000 grams of winding tension. Winding time was seven hours at a laydown rate of 28 rpm. The third rim used 1420-denier yarn with an 1800-gram winding tension. An insert placed on the

Figure 26. WINDING SETUP FOR FABRICATING THE FULL-SCALE RIMS.

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mandrel reduced the part width to approximately 3.6 inches. The winding time was 19 hours at a laydown rate of 37 rpm. Pertinent data on the three rims are summarized in Table 5.

Flywheel Fabrication - Fabrication of a finished Design D flywheel for testing involved rim preparation, machine setup, rim and band tooling assembly, and alignment and winding of the band set. Rim preparation required machining of the sides and outer contour. A planar winding machine was set up with special alignment fixtures to mount the rim and provide the desired band orientations.

Machining of the as-wound rim presented some special problems. Composites, in general, are difficult to machine. It is particularly difficult to achieve a satisfactory surface integrity on Kevlar-49/epoxy composites. The usual surface condition after machining is a rough, hairy texture with many loose fiber ends raised on the surface. This condition is illustrated in Figure 20, which shows a ring section cut from a test cylinder. A high-speed parting-saw operation with coolant flooding was used.

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Figure 27. COMPLETED AS-WOUND RIM AFTER REMOVAL FROM THE WINDING MANDREL.

Obviously, the condition illustrated in Figure 20 is not satisfactory for a flywheel, which must spin at high speeds. The machined surface must be on the same order of smoothness as the surfaces that are wound against a smooth mandrel. Fortunately, machining technology existed in Y-12 which was able to produce the desired surface.

Table 5
FABRICATION DATA ON FLYWHEEL RIMS MADE OF KEVLAR-49/EPOXY (DER 332/T403)

Rim Number	Yarn (denier)	Winding Tension (g)	Winding Speed (rpm)	Fiber Volume (%)	Resin Volume (%)	Void Volume (%)
1	1420	1800	28	73.0	23.9	3.1
2	4560	6000	37	73.2	23.6	3.2
3	1420	1800	37	73.1	23.6	3.3

A unique machining process has been developed in the Y-12 Plant for providing a super finished ($< 2 \mu\text{in}$, peak to valley) surface on a machined part and has been used to produce mirrors for several laser research programs around the country.⁹ The process uses a special cutting tool called a "diamond knife" which is made from a single crystal diamond and is extremely sharp because of a near-perfect edge. The turning machine employs an air-bearing spindle and other very stiff and very precise components. The technique had been used exclusively on metal parts, but was adapted to machine the Kevlar-49/epoxy composite.

A machined rim mounted on the turning machine is shown in Figure 28. Machining is normally done wet with a coolant; but, since the effects of coolant on the composite are

Figure 28. DIAMOND-KNIFE-MACHINED FLYWHEEL RIM OF KEVLAR-49/EPOXY COMPOSITE.

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unknown at this point, rim machining was done dry. Results were very good, as seen in Figure 28, and were felt to be satisfactory for spin testing. The surface was not perfect, however, as shown by the closeup view in Figure 29, but the small imperfections did not seem to be serious enough to warrant further machining development.

The procedure for winding bands across the hub and around the rim called for a planar winding machine. In addition, the planar path had to have a built-in motion normal to the plane of winding in order to generate a parallel fiber band of the desired width. An existing machine had these required features.

Another requirement called for the winding plane to be parallel to the flywheel or part axis. This winding setup was not normal, and the present machine did not have this capability. The problem was solved easily by installing two riser blocks at the top of the machine's rotating yoke. A picture of the machine and flywheel setup is presented in Figure 30.

The rim and band tooling assembly was designed such that the multipiece tooling fitted between the hub and rim, holding the rim in position relative to the hub bore. A machined hub is shown in Figure 31. The multipiece tooling was manufactured by mounting all the pieces assembled in a set on a face plate, then machining the band slots and inner and outer diameters simultaneously. The set was then match marked so that it could be reassembled in the same relationship. A gap was left between the hub and tooling-set diameters and between the tooling-set and rim diameters, so that shims could be used to adjust the alignment and provide for easier disassembly after the bands were cured.

This first flywheel had the hub and band tooling assembled to the rim with a 0.002-inch TIR on the rim OD and a 0.004-inch TIR on the rim face. The band set was wound with 380-denier Kevlar-49 yarn, and the same epoxy resin was used in the rim. The first flywheel data are summarized in Table 6. The finished flywheel after removal of the tooling and readied for spin testing is shown in Figure 32.

FLYWHEEL TESTING

Spin testing of the prototype 11.16-kg (24.55-lb) composite flywheel entailed adapting the existing test stand to accommodate safe flywheel testing, low-speed balancing, and spin testing.

Test Stand

Spin testing of the flywheel was accomplished in the top-loading turbine stand seen in Figure 33. Drive and braking power is supplied by the air turbine. Speed is measured by using an eddy-current probe to monitor the passage of six equally spaced holes in the top of the turbine wheel and feeding this signal into a frequency counter.

Figure 29. CLOSEUP VIEW OF A MACHINED COMPOSITE SURFACE.

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Figure 30. PLANAR WINDING MACHINE AND FLYWHEEL SETUP FOR WINDING BANDS.

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The test-stand casing is constructed of heavy steel plate (1 1/2 inches thick) with vacuum-tight welds and O-ring seals to permit vacuum testing. The vacuum system can provide test environments down to 0.1 micrometer.

Primary radial containment and momentum decoupling is provided by a heavy steel crash ring. Containment evaluations, such as those of Recht,¹⁰ indicated that the primary containment afforded by this crash ring (2 inches) should be sufficient; however, the casing provides additional secondary capability since it is 1 1/2 inches thick. The crash ring rests on three bearing pads and is centered in the casing with roll pins to provide an easily rotatable inertia ring. An analysis of possible crash-ring motion was carried out by assuming totally plastic impact of the flywheel with the crash ring and applying the principles of the conservation of momentum and energy. These analyses indicated that such a total angular-momentum-transfer failure of the flywheel at operating speeds would result in an energy reduction from the 1.5×10^6 ft lb_f contained in the flywheel to 7.1×10^3 ft lb_f stored in the crash ring. This level of residual kinetic energy would be dissipated by friction at the base of the crash ring after 2.03 revolutions of the crash ring.

The possibility of large lateral loads, which could develop from the flywheel orbiting on the inner surface of the crash ring, was obviated by leaving sufficient radial clearance between

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Figure 31. MACHINED ALUMINUM HUB WITH LOGS TO MAINTAIN BAND POSITION.

the crash ring and casing wall to permit the flywheel, crash-ring mass system to move radially unrestrained about the mass center. This arrangement reduces the shaking loads on the mounts to that provided by the radial friction forces between the crash-ring base and the tank floor during the radial motion of the crash ring.

Low-Speed Balancing

Prior to spin testing, the flywheel was low-speed balanced (at 1200 rpm) on a commercially available balancing machine. The flywheel, seen in Figure 34 is mounted on a balance arbor and installed in the machine for balancing.

Using this equipment, the flywheel was carefully force-and-couple balanced with the intent of eventually studying the sufficiency of a low-speed balance for the requisite high-speed operation. A low-speed balance should be sufficient if the following conditions are met:

1. Sufficient sensitivity of the balancing equipment. (The speed-squared dependency can greatly magnify seemingly small low-speed imbalances.)
2. Flywheel does not experience plastic or creep deformations within the operating speed range.
3. Flywheel does not experience significant, nonsymmetric elastic deformations at the higher speeds.
4. Flywheel does not experience large-amplitude flexural criticals within the operating speed range.

Table 6
PHYSICAL DATA ON THE
FIRST FLYWHEEL

	Weights (lb)	Moments of Inertia (in-lb-sec ²)
Rim	21.29	4.256
Bands	1.48	0.150
Hub	1.78	0.019
Total	24.56	4.425

Figure 32. FIRST BAND-WRAPPED FLYWHEEL, COMPLETED AND READY FOR SPIN TESTING.

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Figure 33. EXISTING SPIN TEST STAND AND ASSEMBLY.

Conditions 1, 3, and 4 could be solved by high-speed balancing, but Condition 2 must be accounted for by a flywheel design which eliminates or reduces those deformations to acceptable levels.

Also of interest is the effect of manufacturing tolerances on the imbalance of the flywheel. Of particular interest were the possible misalignments of the rim relative to the hub axis, resulting during the final assembly of the flywheel. This procedure required using a removable fixture for winding the hub attachment bands. Since the removable feature of the fixture negated using interference fits between the fixture and flywheel components, the resulting positional tolerances were on the order of mils. Noting the large mass of the flywheel rim, misalignments of this magnitude could give significant imbalances. With this in mind, the axial and radial runouts of the rim relative to the shaft axis were measured.

Figure 34. FLYWHEEL MOUNTED IN THE BALANCE MACHINE FOR LOW-SPEED BALANCING.

Force and couple-imbalance corrections were made by applying metal tape weights to the inside diameter of the flywheel rim. On this surface, the force imbalance required 1.88 grams of tape, and the couple imbalance required 5.93 grams diametrically opposed with a 2.65-inch axial separation.

Shown in Figure 35 is a plot of the axial runout of the rim relative to the shaft axis. At a diameter of 18 inches, the plane of the rim had a maximum runout of 5.1 mils. If this

Figure 35. PLOT OF AXIAL RUNOUT VERSUS ANGULAR POSITION ON THE FLYWHEEL.

measured angular tilt is used, along with Euler's equations to calculate the couple produced by this misalignment, and subsequently used to calculate the weight necessary to balance the couple, it is found that 5.6 grams are required for a 2.65-inch axial separation. Thus, it appeared that the couple imbalance could be attributed to misalignment of the rim during final assembly. Careful control of manufacturing procedures could minimize this effect.

Spin Testing

The flywheel was suspended from the vertical spindle of the turbine, as depicted in Figure 33. Runout probes were positioned to monitor radial runout of the shaft and axial runout of the flywheel rim. Both signals were displayed on an oscilloscope as run monitors and recorded on tape for future reference. After several low-speed runs (< 10,000 rpm) to check out instrumentation, the flywheel was driven up in speed while closely monitoring axial and

radial runout and vacuum pressure. At 18,000 rpm, both runout signals and the vacuum pressure experienced rapid increases and then stabilized; hence, the flywheel was stopped for inspection. Subsequent visual inspection showed that two of the bands had begun to separate from the outer circumference of the rim, and the rim had what appeared to be a circumferential crack. The circumferential crack (radial delamination) was confirmed by subsequent radiographic examination and was located at a radial position approximately midway through the rim wall. Radial and axial-runout data generated in driving the flywheel up 18,000 rpm are summarized in Figure 36. Computed natural frequencies for axial, torsional, and wobble-type vibrations are depicted along the speed axis. Data from the two probes are, of course, not adequate to determine the vibration modes and/or mode shapes.

Figure 36. FLYWHEEL SHAFT RADIAL RUNOUT AND RIM AXIAL RUNOUT AS FUNCTIONS OF SPEED.

SUMMARY OF RESULTS

A catalog of flywheel concepts was studied, preliminary evaluations were made, a design was selected for development, and the prototype composite flywheel (Figure 32) was designed, fabricated, and spin tested in the test stand setup depicted in Figure 33. The flywheel rim and spokes were fabricated of Kevlar-49/epoxy composite with a combined weight of 22.77 pounds. The hub was fabricated of Type 6061-T6 aluminum with a weight of 1.78 pounds. Therefore, the total flywheel weight was 24.55 pounds. The flywheel was radial and couple balanced and subsequently spin tested to a speed of 18,000 rpm. At this speed, the stored kinetic energy was 0.25 kWh and the energy-density performance level was 10.1 wH/lb.

The maximum speed attained did not result in catastrophic failure of the flywheel. Radial delamination was, however, sensed via increased pressure in the test chamber and some increased runout, as noted in Figure 36. The flywheel operated very stably while being driven up to the 18,000-rpm level and also while being reduced in speed after sensing the delamination.

The fact that some transverse composite fractures occurred during the test, confirmed the expected low transverse strength. When the prototype flywheel design and fabrication were started, the transverse strength was thought to be approximately 4 ksi; however, test data on the fiber composite system used in constructing this flywheel came after fabrication was essentially complete and indicated a strength of approximately 1 ksi. The test specimens were not ideal, however, and the planned flywheel test was continued. The calculated transverse stresses at the rim delamination position and at the rim-to-band bond were approximately 1 ksi at the 18,000-rpm test speed.

Future work will be directed toward improving the composite transverse strength or using design alternatives to eliminate the need for transverse tensile strength. The strength improvement will likely only come with a resin system which adequately bonds to the fiber, but it must also be a low or room-temperature curing resin for fabricating thick rims. Design alternatives to be studied include a shaped outer contour on the rim, dead-weight loading inside the rim, prestressing the rim, decreasing the rim/radius ratio, and varying the modulus of elasticity through the rim. Incorporation of one or more of these design alternatives into a bandwrap flywheel should produce significant performance improvements.

Unfortunately, a direct and realistic comparison of the energy-density performance with other flywheel test data from the literature could not be made on an objective basis due to lack of the raw data in the literature needed to make such assessments. Various methods are currently being used by the scientific community for reporting flywheel performance. These methods range from the energy-density performance for the outermost fiber, based on a one-dimensional stress situation or material test, to the average for the total rotor system. It would appear that a realistic method for assessing prototype flywheel energy-density performance is to divide the total stored kinetic energy of the flywheel and hub or shaft attachments by the total weight of the assembly. The performance level reported herein was determined in this manner.

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