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COMPOSITE FLYWHEEL DEVELOPMENT
(May 1-June 30, 1976)

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MASTER

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OAK RIDGE, TENNESSEE

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COMPOSITE FLYWHEEL DEVELOPMENT

INTRODUCTION

The purpose of the report is to summarize Union Carbide Corporation-Nuclear Division's (UCC-ND's) composite flywheel program objectives and accomplishments for the period from May 1 through June 30, 1976. This work was conducted for, and funded by, the Mechanical Energy Storage Branch, Division of Energy Storage Systems, Office of Conservation, ERDA, Washington.

The necessity and urgency of national energy conservation is a well-recognized fact. Mechanical-energy storage, using rotating flywheels, is one of the few known methods for energy storage, and flywheels probably have the highest potential effectiveness for energy storage of any method now available. As a result of past program activity within UCC-ND, considerable experience has been gained at the Oak Ridge Y-12 Plant^(a) in the area of design, fabrication, and testing of high-performance composite structural components. Similarly, a great depth of experience has been developed in the design, test, and containment of high-speed rotating equipment.

Because of this experience base, UCC-ND can contribute a significant amount of timely development of high-performance composite flywheels and their containment and, thus, assist ERDA in meeting national energy conservation goals.

Initial application selected for the development of the composite flywheel is the heat engine/flywheel hybrid propulsion system for a vehicle. This application was selected by the staff of the Mechanical Energy Storage Branch because of its high potential for the conservation of petroleum fuel in both the near and long-range time frames.

Discussions with the Mechanical Energy Storage Branch staff have focused UCC-ND's efforts into three key areas consistent with its experience base: state-of-the-art flywheel development, spin testing, and containment development.

The objective of the UCC-ND flywheel development effort for the budgetary period from May 1 through September 30, 1976, is to design, fabricate, and performance test an experimental 0.5-kWh composite flywheel, the associated bearings, conservatively heavy containment, and a suitable means for driving the flywheel to the design speed, as depicted in Figure 1. This work is funded at a level of \$68,000 for the five-month period.

An operating performance goal of 20 watt-hr/lb (20 Wh/lb) energy density at an energy level of 0.56 kWh has been set by UCC-ND. The 20 Wh/lb goal encompasses both the composite flywheel and the hub that connects it to the shaft. It does not include the shaft. The goal exceeds the present performance of isotropic flywheels, and is also at the upper limit of

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

Figure 1. AN EXISTING SPIN TEST STAND FOR THE INITIAL FLYWHEEL DEVELOPMENT.

current laboratory technology reported in the literature for composite flywheels^(b) as depicted in Figure 2.

At this point in the program, the thick rim with radial overwrap bands has been selected as the initial design concept. Kevlar-49/epoxy has been selected as the construction material, and the end of the design phase is near. A process for fabricating full-scale Kevlar-49/epoxy thick rims has been developed, and full-scale rims have been successfully wound.

The balance of this report provides more detailed information on the development plan for this budgetary period and discusses the present accomplishments with respect to: (1) flywheel design, analysis, and fabrication and (2) spin testing.

FLYWHEEL DEVELOPMENT PLAN

Tasks directed toward fabricating and spin testing a composite flywheel by September 30, 1976, consistent with the energy needs of the heat engine/flywheel hybrid vehicle, are summarized in Figures 3 and 4. Successful completion of these tasks will result in a 20-

(b) *Economic and Technical Feasibility Study for Energy Storage Flywheels*, Final Report to ERDA, Division of Conservation Research and Technology, prepared by Rockwell International Space Division under Contract (04-3)-1066, pp 2 - 4; December 1975.

(1) Composite and Hub.
(2) Performance ratings per ERDA-sponsored Rockwell International Study.

Figure 2. UCC-ND FISCAL YEAR 1976-T ENERGY DENSITY PERFORMANCE GOAL

Figure 3. A DEVELOPMENT PLAN, LEADING TO THE FABRICATION AND TESTING OF A KEVLAR/EPOXY FLYWHEEL. (Completion by September 1, 1976)

Figure 4. FLYWHEEL SPIN TESTING SCHEDULE IN AN EXISTING FACILITY.

Wh/lb, 0.5-kWh high-performance flywheel. The initial flywheel will not be suitable for direct installation into an automobile, however. Additional problems, such as failure speed and statistics, acceleration/deceleration stability, stability due to vehicle motions, fatigue, and speed life, must be solved. The plan, summarized in Figures 3 and 4, includes two experimental iterations in flywheel development with a period of approximately six months after receipt of funding needed to achieve the objectives. As noted in Figure 3, funding was received on May 1, 1976, allowing five months of development effort prior to the end of Fiscal Year 1976T (September 30, 1976). One fabrication, test, and evaluation cycle will, however, be completed by September 30, 1976. The second iteration, planned to eliminate any design defects and identified as the "ERDA flywheel" in Figure 3, will be meshed into the Fiscal Year 1977 program.

FLYWHEEL DESIGN, ANALYSIS, AND FABRICATION DEVELOPMENT

This program is designed to develop high-performance fiber composite flywheels. Development entails design, analysis, fabrication, and testing of flywheels and their associated containment housings. This portion of the report is concerned with the design, analysis, and fabrication of the actual flywheel.

The design phase includes both the conceptual designs and detailed design of selected concepts. Structural and dynamic analyses of the detailed design will be used to evaluate their potential performance. A design has been selected for fabrication and testing to verify its potential performance rating.

Conceptual Study

Flywheel Specifications - The objective of this program is to develop a high-efficiency, fiber-composite-material flywheel for energy storage. Based on discussions with the Mechanical Energy Storage Branch staff, the most suitable starting point for development is a unit that is sized to the requirements of a heat engine/flywheel hybrid propulsion system for automobiles. This requires a storage capacity of 0.5 - 0.7 kWh. This storage capacity and automotive requirements, along with information from the open 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 is used, then a flywheel package with a 0.5-kWh available energy is 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.

A search of the literature and personal communications have not revealed any composite flywheels that demonstrate this energy level. Small state-of-the-art metal flywheels reach

an operating storage efficiency of 15 Wh/lb. A goal of 20 Wh/lb has been set as the design efficiency for the first flywheel. This goal will demonstrate the potential of fiber composite flywheels (if it is not overly optimistic), since the ultimate strength-to-density ratio of the current Kevlar-49/epoxy composite would yield an ultimate efficiency of 80 Wh/lb in a thin-rim flywheel configuration. A 20 Wh/lb efficiency requires a flywheel weighing 28 lbs, total.

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. A reasonable maximum diameter is 20 inches. This dimension is suitable for an automotive system and is compatible with existing composite winding equipment and other 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 specification and design objectives are summarized in Table 1.

Table 1
DESIGN SPECIFICATIONS FOR AN AUTOMOTIVE
SYSTEM FLYWHEEL

Configurations - There are three basic flywheel configurations suitable for use with fiber composite materials. These basic designs are illustrated in Figure 5. Practical designs with these configuration must address themselves to specific problems associated with each configuration. For instance, the rim configuration must have

Available Stored Energy	- 0.50 kWh
Design Stored Energy	- 1.12 kWh (min)
Maximum Design Speed	- 40,000 rpm
Flywheel Weight	- 28 lb
Flywheel Size	- 20" D (approx)
Torque Load Capacity	- 100 ft lb
Dynamic Characteristics	- Minimize the Number of Critical Frequencies in the Operating Speed Range

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

Conceptual Designs - In the conceptual study, 18 conceptual designs, and optional variations on some designs, were cataloged. These designs are illustrated in Figures 6 through 8 and are briefly described in the paragraphs to follow.

All of the designs given in Figure 6 employ the rim configuration and show 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 small enough to prevent delamination prior to hoop-fiber failure for a particular fiber composite. The rim-axial thickness may be selected to provide the desired level of energy to be stored.

Design A uses a curved-spoke arbor. The arbor would be machined from a suitable metal and the curvatures 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 a more uniform contact stress and an equivalent internal-pressure-ballast-type loading.

Design B incorporates wound-fiber-composite loops as spokes and 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 rotational 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 displacement 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 could be 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, with minimum of three spokes. In this design, displacement 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 kind of built-in spring at the spoke tips must accommodate the interference fit.

The designs depicted in Figure 7 employ the rod configuration in various design schemes. The rod elements of each design are of a continuous, unidirectional, composite material. The components of each design would be individually fabricated and assembled in the flywheel form.

(a) Rim Configuration.

(b) Rod Configuration.

(c) Disc Configuration.

Figure 5. BASIC FLYWHEEL CONFIGURATIONS.

Figure 6. CONCEPTUAL DESIGNS FOR THE FLYWHEEL. (Rim Configuration)

Figure 7. CONCEPTUAL DESIGNS FOR THE FLYWHEEL. (Rod Configuration)

Figure 8. CONCEPTUAL DESIGNS FOR THE FLYWHEEL. (Disc Configuration)

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 designs.

The rod or bar of Design G is a relatively massive component. It is tapered to provide a more uniform stress distribution in the assembly. 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 Rabenhorst, et al, ⁽¹⁾ are cataloged 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 8) are considered as disc configurations although some designs are similar to the rim configuration. These designs either attempt to limit radial stresses in circumferentially wound composites or provide fiber orientation to accommodate the radial stresses.

Design J limits the radial stresses by nesting the 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.

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. Geodesic-patterns winding near the axis of rotation would be primarily radial in orientation while geodesic-patterns winding near the outer diameter would be primarily circumferential in orientation. The number of different geodesics and the resulting thickness profile is 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 was described by Garber.⁽²⁾

Design O is a simple circumferential composite winding, 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 was 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.

Evaluation and Selections - A method has been established for evaluating the cataloged conceptual designs and selecting the designs for development. Each concept was studied, and its pros and cons were listed. Based on these lists, the following set of rating criteria were established, with the numbers in parentheses representing the numerical rating scale for the criterion listed:

Rating Criterion Number	<u>Criterion</u>
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 a 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 for 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 was estimated for all the criteria. The decision table is presented as Table 2. Some of the criteria and the rating-scale values were selected to meet the objectives of the FY 76, FY 76T program. Thus, the final rating and selection of designs for development may be reevaluated at a later time. The rating values are summed for each design to determine the overall ranking of conceptual designs and, thereby, to suggest the designs to be developed. From the rating table, the first five flywheel designs are, in order: D, Q, M, C, and K. The ratings

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

for Design D were based on the composite loops passing 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.

For these reasons, Design D has been selected for initial development in the FY 76 program, leading to the development of Design Q in following programs. Designs M, C, and K will be carried along at a low activity level as backup designs.

Analysis and Design

Material Selection and Characterization - It should be noted that 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 the disc-type geometry necessary 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-per-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 weight density). 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 that the Y-12 Composite Structures group has had experience in fabricating.

Systems that are 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 are eliminated at this time 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 are able to 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 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 providing the residual stresses are not significantly unfavorable. A very unfavorable distribution was expected but not realized, as will be shown in the subsequent discussion on residual stress analysis.

Since the minimum radius ratios were the same for all the composites, the Kevlar-49/epoxy system has been selected for use because of its higher strength-to-weight ratio. Elastic properties for Kevlar-49/epoxy have previously been characterized. The theoretical hoop strength of wound rings is well established; however, the actual material used in fabricating the rims will be certified by winding ring-tensile specimens from the material spools. The transverse tensile strength is variable and significantly dependent on the epoxy resin system used. Transverse tensile strength in the ring form has not been measured. A special test was set up to measure the radial tensile strength of a thick ring.

In the transverse tensile test, a segment of a ring is mounted in an apparatus illustrated schematically in Figure 9. A pure bending moment is applied to each end of the segment. This action produces a flexural hoop stress and a radial tensile stress if the moments are 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, will provide specimens for this radial-strength measurement.

Figure 9. SPECIAL-RING-SEGMENT TRANSVERSE TENSILE TEST.

Structural Analysis - A detailed stress analysis of Designs C and D has been performed, using the finite-element method. The computer code 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.

A finite-element model of Design D was prepared through the use of a mesh-generation program. One such model is indicated in Figure 10. The structure is symmetric about the R

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

axis; therefore, only half the structure is modeled, and the Z axis is the centerline of rotation. The rim is essentially rectangular in cross section, but has radiused outer corners. The mesh is input along with the material properties and angular velocity. First, the model is run in the axisymmetric mode. Stresses and displacements in the rim are found. Rim-loop interface displacements are used as input, and the case is rerun in the plane-stress mode. This run determines the stresses and displacements of the loop. An accurate model of Design D would require a 3-D analysis, but the procedure just described should be a valid approximation. A few meshes with different outer-corner radii were prepared and analyzed, and a radius of 1.0 inch was selected for short-term use.

Based on the results thus far, the Design D flywheel should provide a comfortable margin of capability above the program objectives, if the flywheel performance is controlled by stresses rather than dynamic stability. Additional parameters will be studied to evaluate the bond integrity needed between the loop and rim, static stiffnesses in coordinate directions, and other design factors.

Design M can be primarily evaluated by using the composite-laminate theory. This analysis is a relatively straight-forward, calculator-type study and will not require a large amount of analysis time.

A finite-element mesh of a Design C flywheel is depicted in Figure 11. Again, the structure is symmetric about the R axis, and the Z axis is the axis of rotation. The mesh shown represents the best form obtained to date of a detailed Design C. Hyperbolic, constant-stress and linear-taper-thickness profiles for the disc arbor have all been 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 provide compatibility with the 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. Design and analysis thus far

Figure 11. FINITE-ELEMENT MODEL OF THE DESIGN-C FLYWHEEL, SYMMETRIC ABOUT THE R AXIS.

indicate that this type of design could reach the energy storage goal of 20 Wh/lb, but the metal in the lip region would be near its yield point at this level and, thus, not possess an adequate safety factor.

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, titled NONSAP,⁽⁶⁾ performs two or three-dimensional nonlinear static and dynamic structural analyses. The current version of the program is being modified to account for several important factors in the analysis. The three-dimensional element in the program only allows isotropic material properties, and only considers material nonlinearity. Orthotropic material capability will be added; and, for the present, the axis of orthotropy will be aligned with the element's local coordinates. The 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.

While code modifications are being programmed and inserted, a model of Design D is in preparation. The model will be submitted and run first in the unaltered-code version. This model is for analysis of the body's critical frequencies in the operating speed range. The model is a 180-degree portion of the rim and radial bands, therefore it should provide for all the possible vibration modes.

A different model is required for the stress analysis due to body-force loading. More symmetries may be used to reduce the portion of the circumference that is necessary to model and, thereby, allow a more refined element mesh for detailed evaluation of the stress distribution. The stress-analysis model will most likely be run after most of the code modifications have been completed.

Design for Fabrication - An isometric drawing of the first flywheel to be fabricated is presented in Figure 12. The rim will be constructed of Kevlar-49/epoxy composite.^(c) The

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

Figure 12. DESIGN OF THE FIRST FLYWHEEL TO BE FABRICATED.

wound composite is 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, producing a 2.5-inch-thick rim.

The design goal of 20 Wh/lb establishes the total flywheel weight at 28 lb. 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 lb. If 2.0 lb are allowed for the hub, a 23.4-lb rim would leave a 1.0-lb margin for design changes in the bands or hub if further analysis shows any deficiencies. With the constraints just listed, the rim is then 3.400 inches wide.

The first hub design to be fabricated and tested is illustrated in an isometric view in Figure 13. The material will be of Type 6061-T6 aluminum. Finite-element analysis of the hub shows a more-than-adequate speed capability. In addition, the design provides 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.

Fabrication

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

Figure 13. HUB DESIGN FOR THE BANDWRAP FLY-WHEEL.

A six-inch-diameter cylindrical mandrel was available on which to wind test rims. A conventional wet-winding process line was used in conjunction with the circumferential winding machine seen in Figure 14. The spool of Kevlar-49 is mounted on a rotating pretensioning device. The strand passes first to the resin-impregnation drum, then to the final fiber-tension controller. The fiber-tension controller is closed-loop controlled, using the input of the tension monitor. Finally, the strand passes to the winding machine guide pulley and is wound on the mandrel.

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

The first cylinder was wound to 2 inches wide and 1 inch thick. This thickness produced a 0.75 radius ratio. A circumferential lead rate of 0.024 inch/revolution gave a side-by-side strand laydown for the 380-denier Kevlar strand. At the wind completion, the part was allowed to rotate until resin gelation occurred, then it 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% fiber by volume, 23.7% resin by volume, and 2.1% void. The part was easily removed from the mandrel, and visual inspection did not reveal delaminations or localized fiber buckling. X-ray inspection of the part also showed no delaminations. These results were very encouraging, but they may not be directly scalable to larger rims.

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Figure 14. A WET-WINDING PROCESS LINE WITH A CIRCUMFERENTIAL WINDING MACHINE.

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

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

Before any residual stress measurements had been made on the fabricated rims, a variable-winding-tension profile was tried on another 6-inch-diameter rim. The profile

linearly increased from 2000 to 9500 grams through a 1-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 region of severe, localized fiber buckling was observed. A view of the part is provided in Figure 15. Obviously, this

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Figure 15. LOCALIZED FIBER BUCKLING IN A RING, DUE TO AN IMPROPER PROGRAMMED WINDING TENSION

condition cannot be allowed in a spinning flywheel. Based on this result, and the results from the residual stress analysis reported in the section to follow, the variable-tension profile used for this rim is far from the correct one.

Another 6-inch-diameter rim was wound. The fiber was 1420-denier Kevlar yarn tensioned at 1800 grams (proportional to a 500-gram tension on 380-denier yarn). The rim was wound to a 1-inch wall thickness on the lathe with an automatic lead-reversing mechanism. The composite rim was 71.8% fiber by volume, 24.4% resin by volume, and 3.8% void.

Residual Stress Analysis - Residual stresses in a composite structure may be of critical importance, especially in the direction transverse to the fiber axis. Process studies have shown the capability to fabricate thick rims from the Kevlar-49/epoxy composite.

Residual stresses present in the fabricated rims must be determined to evaluate the potential flywheel performance.

The procedure used to evaluate the performance of the potential flywheel is as follows: A test ring, parted from the fabricated rim, is instrumented with strain gages. The strain changes, recorded as thin layers of material, are machined from the diameter. The strain changes are used to calculate the residual stresses that were initially present.

The stresses are calculated in increments that correspond to the strain change produced when an increment of material is removed. (The ring is illustrated schematically in Figure 16 after removal of the i^{th} layer.) The radial stress, σ_{r_i} , is 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 is given by:

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

Rearranging to yield the radial stress gives:

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

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 is the radial modulus of elasticity), and
- $\Delta\epsilon_{\theta}$ the tangential strain change from its initial zero reading.

Figure 16. RESIDUAL-STRESS-ANALYSIS MODEL OF A TEST RING.

Radial stresses are determined at all incremental points through the wall. The tangential stress in any layer, i , may be 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 is the layer thickness. The radial and tangential "initial" or residual stresses are plotted versus radial position to display their distribution.

The residual stress distributions found in the first three test rims are indicated by the graphs of Figures 17 through 19. The shape of the distributions is similar for each rim, differing only in their magnitudes. Overall, these are very satisfactory distributions. Radial stresses are mostly compressive and very low. The tangential stresses are high near both boundaries, but of most concern near the outside radius. The most obvious solution for reducing the outer-surface residual stress is to wind an excess of material and then machine away the unneeded material after curing.

The type of distributions found were not those expected, based on prior experience, which 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 stress produced by resin shrinkages and/or thermal contractions from an elevated-temperature cure would be radial tensile stresses through the wall and tangential tensile stresses on the inner surface.

Tooling Design - The tooling required to fabricate the circ-wound rims is relatively simple, namely: a steel cylinder (15" D x 4.100" L) fitted with two side plates. (The side plates have a 50-mil recess for the cylinder and have an outside diameter of 22 inches. A two-inch-diameter shaft passes through the side plates.) The side plates are 0.750 inch in thickness to provide adequate stiffness to resist the side pressure during winding. The wound rim can then be up to 22 inches in outside diameter by four inches long.

The tooling for winding bands around the rim is more complex. A form must be provided for each band to be wound. This form controls the fiber-laydown position and the overall dimensions of the band. A multipiece disc fits between the hub, described earlier, and the rim. This disc positions the rim concentric with the hub and lines up the band forms in the correct position relative to the hub features. Matched multipiece discs are required for each side of the flywheel.

Figure 17. RESIDUAL STRESSES IN A SIX-INCH-INSIDE-DIAMETER BY ONE-INCH-THICK CIRC-WOUND KEVLAR-49/EPOXY RING.

Rim Fabrication - The rim tooling was received and the first full-scale rim was wound. The fiber was 1420-denier Kevlar-49 yarn. The resin system was DER 332/T403, cured at room temperature. The winding tension was held constant at 1800 grams. The rim was wound to approximately 20.2 inches OD to permit machining away the high-residual-stress region on the outer edge. The total winding time was 28 hours. The completed rim was 73.0% fiber by volume, 23.9% resin by volume, and 3.1% void. The completed rim is shown in Figure 20.

FLYWHEEL SPIN TESTING

In order to accelerate the program and provide cost-effective development, existing spin-test equipment will be used for initially testing the flywheel. To assure safety in using the existing equipment for testing the relatively massive flywheels, probable flywheel failure and breakup modes were forecast and the structural integrity of both the containment and

Figure 18. RESIDUAL STRESSES IN A SIX-INCH-INSIDE-DIAMETER BY 1.5-INCH-THICK CIRC-WOUND KEVLAR-49/EPOXY RING.

Figure 19. RESIDUAL STRESSES IN A THIRTEEN-INCH-INSIDE-DIAMETER BY 2.5-INCH-THICK CIRC-WOUND KEVLAR-49/EPOXY RING.

Figure 20. COMPLETED AS-WOUND RIM AFTER REMOVAL FROM THE WINDING MANDREL.

mounts were reevaluated. Minor modifications will be made to accommodate initial flywheel testing; however, more extensive modifications would be required for testing at energy levels in excess of 0.5 kWh. In accordance with standard UCC-ND safety procedures, it is also necessary to prepare a test-stand safety-qualification document, which is being prepared.

FUTURE WORK

Plans for the balance of the five-month budgetary period include fabrication and testing of a Design D flywheel. The rim will be machined with radiused corners on the outside diameter. Tooling for the multiple-band set is scheduled to arrive, and will be used to apply the overwrapped bands.

Finite-element structural and dynamic analyses will be run to further check out the detailed design and assist in establishing testing procedures. Computer-code modifications on NONSAP and/or the acquisition of a new code called ADINA will give improved structural and dynamic analyses.

Additional rims will be fabricated and prepared for test iterations that may be needed to correct any design deficiencies which show up in the initial flywheel tests. Initial spin tests will include balancing, driving the flywheel to its operational speed, defining the rigid-body vibration speeds as well as any flexible body critical speeds, and obtaining shaft runout and axial displacement data. Test-stand modifications and the safety Qualification Document will be completed prior to flywheel testing. Performance testing of the first flywheel concept will be used, primarily, to evaluate critical speeds, shaft and rim runouts, and operational speed capability up to a maximum energy level of 0.56 kWh. Significant modifications and/or a new test stand will be necessary prior to failure testing of flywheels above an operating speed which is consistent with the 0.56-kWh energy level and 20 Wh/lb energy density.

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