ENERGY

CONSERVATION

Dr. 1403

SAN/1213-1

**NEAR-TERM
ELECTRIC VEHICLE PROGRAM**

Phase I, Final Report

August 1977

Work Performed Under Contract No. E(04-3)-1213

**AiResearch Manufacturing Company of California
A Division of the Garrett Corporation**

**ENERGY RESEARCH AND
DEVELOPMENT ADMINISTRATION**

Division of Transportation Energy Conservation

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

This report has been reproduced directly from the best available copy.

Available from the National Technical Information Service, U. S. Department of Commerce, Springfield, Virginia 22161

Price: Paper Copy ~~\$9.00~~^{10.75} (domestic)
 ~~\$11.50~~ (foreign) ^{21.50}
Microfiche \$3.00 (domestic)
 \$4.50 (foreign)

NEAR-TERM ELECTRIC VEHICLE PROGRAM

Phase 1, Final Report

NOTICE
This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

**Prepared by
B.H. Rowlett
Program Manager
and
R. Murry
Development Engineer
AiResearch Manufacturing Company of California**

**ERDA Contract No. E(04-3)-1213
AiResearch Report No. 76-13465**

August 1977

**Prepared for
Energy Research and Development Administration
Washington, D.C.**

**AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA**

MASTER

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

FOREWORD

This final report is submitted by AIRsearch Manufacturing Company of California, a division of The Garrett Corporation, in compliance with the requirements of the Phase I study program. The electric-powered passenger vehicle design study program, Contract E(04-3)-1213, is an Energy Research and Development Administration effort aimed at a preliminary design of an energy-efficient electric commuter car. This program was conducted under the direction of Robert Kirk, ERDA/TEC program manager.

Distribution of this report is provided in the interest of information exchange. The significant contribution of W. Sainio in its preparation is gratefully acknowledged. In addition, K. Wuertz was responsible for the motor/generator design, M. Chang for system performance analyses, D. Pasley for the controller design, and J. Rengert for reliability/safety analyses. The body was designed by The Brubaker Group, with aid from All American Racers, Inc., in suspension design. Also, Rath and Strong, Inc., contributed to the economic studies.

CONTENTS

<u>Section</u>	<u>Page</u>
1 SUMMARY	1-1
Program Objectives	1-1
Vehicle Design	1-3
Power System Design	1-4
Performance Characteristics	1-5
Vehicle Economic Feasibility	1-5
Reliability and Safety	1-6
Design Options and Recommendations	1-6
2 INTRODUCTION	2-1
3 REFERENCE VEHICLE PRELIMINARY DESIGN	3-1
Design Features	3-1
Vehicle Definition	3-3
Design Characteristics	3-7
Body Design	3-9
Exterior Features	3-9
Structural Design	3-14
Interior Design and Configuration	3-16
Product Acceptability Considerations	3-19
4 BASELINE POWER SYSTEM	4-1
Description and Operation	4-1
Flywheel Design	4-5
Transmission/Gearbox	4-8
Dc Motor/Generator	4-11
Field Assembly	4-14
Armature Assembly	4-18
Commutation	4-19
Final Assembly	4-19
Controller	4-20
Battery Supply Controller	4-22
Microprocessor	4-24
Control Interfaces	4-24
Sensors and Instrumentation	4-27
Power Supplies, Grounding, Noise	4-27
Battery Pack	4-28
5 SYSTEM PERFORMANCE	5-1
Introduction	5-1
Simulation Models and Program Description	5-1
Simulation Model Inputs	5-2
Loss Models	5-3

SectionPage

	Aerodynamic Drag	5-3
	Rolling Resistance	5-3
	Motor and Generator Losses	5-4
	Controller	5-5
	Cooling Fans	5-5
	Motor and Generator Gears	5-6
	Planetary Differential	5-6
	Final Drive Differential	5-6
	Flywheel Losses	5-7
	System Performance	5-7
	Flywheel Performance	5-8
	Hill-Climb Range	5-10
	Battery Performance	5-10
	Suburban Driving Cycle	5-14
	Vehicle Range	5-16
	Battery Pack Life	5-10
6	ECONOMIC ANALYSIS	6-1
	Introduction	6-1
	Acquisition Cost	6-2
	Cost Based on Conventional Steel Production Vehicle	6-4
	Cost Based on Current Plastic Production	6-6
	Cost Based on Plastic in Mass Production	6-9
	Cost Based on Mass Production of Plastics and Electric Motors	6-10
	Acquisition Cost Study Conclusions	6-12
	RDT&E Cost	6-13
	Ownership Costs	6-15
	References	6-22
7	RELIABILITY AND SAFETY	7-1
	Reliability	7-1
	System Safety	7-2
	Impact and Roll-Over	7-4
	Energy Source	7-4
	Testing and Analysis	7-6
8	ALTERNATE DESIGNS AND OPTIONS	8-1
	Introduction	8-1
	Alternate Transmission Concepts	8-1
	Four-Wheel Regenerative Braking	8-4
	Battery Locations	8-5
9	DEVELOPMENT PLAN	9-1
	Master Plan	9-1
	Program Phases	9-1
	Phase 11A: Power System Demonstrator	9-1
	Phase 11B: Experimental Prototype Vehicles	9-3
	Phase III: Development Prototypes	9-3
	Phase IV: Preproduction Prototypes	9-5

<u>Section</u>		<u>Page</u>
10	CONCLUSIONS AND RECOMMENDATIONS	10-1
	Conclusions	10-1
	Recommendations	10-1
	 Tradeoff Studies	
1	INTRODUCTION AND SUMMARY	11-1-1
	Introduction	11-1-1
	Summary	11-1-1
	Conclusions	11-1-1
2	POWER SYSTEM BASELINE DESCRIPTION AND PERFORMANCE	11-2-1
3	COMPUTER SIMULATION STUDIES	11-3-1
	Introduction	11-3-1
	Simulation Models and Program Description	11-3-1
	Simulation Model Inputs	11-3-2
	Loss Models	11-3-3
	Aerodynamic Drag	11-3-3
	Rolling Resistance	11-3-3
	Motor and Generator Losses	11-3-5
	Motor and Generator Gears	11-3-6
	Planetary Differential	11-3-6
	Final Drive Differential	11-3-7
	Flywheel Losses	11-3-7
	Simulation Model Outputs	11-3-8
	Battery Weight	11-3-8
	Overall System Efficiency	11-3-8
	Energy Recovery Efficiency	11-3-8
	System Performance Studies	11-3-10
	Tradeoffs on Propulsion System	11-3-10
	Loss Sensitivity Studies	11-3-12
4	COMPONENT AND POWER SYSTEM TRADEOFF STUDIES	11-4-1
	Flywheel Design Tradeoffs	11-4-1
	Motor and Generator Design Tradeoffs	11-4-6
	Battery Design Tradeoffs	11-4-6
	Gearbox	11-4-16
	Alternate Transmission Concepts	11-4-16
	Alternate Electrical Transmission	11-4-19
	Controller Design Tradeoffs	11-4-19

<u>Section</u>		<u>Page</u>
5	VEHICLE DESIGN TRADEOFFS	11-5-1
	Introduction	11-5-1
	Vehicle Gross Weight, Empty Weight, and Range	11-5-1
	Vehicle Layout Studies of the Power System and Battery Locations	11-5-4
	Operational Tradeoffs and Characteristics	11-5-14
	Energy Loss during Parking	11-5-14
	Startup after Extended Parking	11-5-14
	Braking Characteristics	11-5-15
6	ECONOMIC EVALUATION/ANALYSIS MODEL	11-6-1
	Introduction	11-6-1
	Objectives	11-6-1
	Model Structure	11-6-1
	Cost Estimating Relationships (CER's)	11-6-2
	RDT&E Cost	11-6-2
	Acquisition Cost	11-6-3
	Ownership Cost	11-6-4
	Total Ownership Cost	11-6-8
	Total Electric Vehicle System (Lifetime) Cost	11-6-8
	References	11-6-9
7	RELIABILITY AND SAFETY	11-7-1
	Introduction	11-7-1
	Reliability Analysis	11-7-1
	Reliability Prediction and Reporting	11-7-4
<u>Appendix</u>	Component Reliability Prediction and Reporting Index and Forms	11-A-1

SECTION 1

SUMMARY

PROGRAM OBJECTIVES

An electric-powered passenger vehicle using a regenerative power system was designed to meet the near-term ERDA electric automobile goals, as summarized in Table 1-1. The program objectives were to (1) study the parameters that affect vehicle performance, range, and cost; (2) design an entirely new electric vehicle that meets performance and economic requirements; and (3) define a program to develop this vehicle design for production in the early 1980's. The design and performance features of the preliminary (baseline) electric-powered passenger vehicle design are summarized in Table 1-1. All aspects of the baseline design have been defined in sufficient detail to verify performance expectations and system feasibility.

TABLE 1-1

AIRESEARCH VEHICLE DESIGN MEETS PROGRAM OBJECTIVES

Parameter	Near-Term ERDA Goals	AiResearch Design
Minimum passenger capacity	4 adults	4 adults
Maximum curb weight, lb	Open	2566
Minimum urban range (J227D), mi	75	85
Maximum initial cost, projected, 1975\$	5000	5000
Minimum life, mi	100,000	100,000
Maximum life-cycle cost, projected, 1975\$/mile	0.15	0.15
Maximum electric recharge energy in urban driving, kw-hr/mi	0.8	0.391
Maximum recharge time, hr	6	6
Minimum top speed, mph	55	70
Minimum accessories	Heater/defroster	Heater defroster, onboard charger
Safety features	Suitable for urban driving	All FMVSS requirements
Minimum unserviced park duration, day	7	>7
Maximum years till production ready	5	4
Maximum critical materials required	Few	None
Minimum acceleration (0 to 30 mph), sec	10	6
Minimum merging time (25 to 55 mph), sec	25	10
Sustained speed on 5 percent grade, mph	50	50
Maximum maintenance	Open	1.4¢/mi
Battery type	Open	Lead-acid
Minimum ambient temperature range, °F	-20 to +125	-20 to +125

FRP CONSTRUCTION—LIGHTWEIGHT BODY

FEATURES

- FRONT AND REAR BATTERY PACKS TO OPTIMIZE ENERGY MANAGEMENT
- REAR POWER SYSTEM TO IMPROVE FRONT END CRASHWORTHINESS
- FRONT HOOD AND FENDER DESIGNED AS SINGLE UNIT TO TRANSMIT CRASH LOADS INTO MAIN STRUCTURE
- AMPLE LEG ROOM, HEAD ROOM AND STORAGE SPACE
- DESIGNED TO SATISFY FMVSS REQUIREMENTS

PERFORMANCE SUMMARY

DRIVE TRAIN CONCEPT

NOTE: HOUSINGS NOT SHOWN

FEATURES

- REGENERATIVE POWER SYSTEM WITH A WARD-LEONARD TYPE CONTROLLER
- DIFFERENTIAL PLANETARY GEAR SET WITH AN INFINITELY VARIABLE SPLIT PATH TRANSMISSION
- EXTENDED BATTERY CAPACITY AND LIFE DUE TO LOAD LEVELING
- DRIVE CONTROL WITH THROTTLE LIKE COMMON IC ENGINE
- SUBSTANTIAL HILL CLIMBING ABILITY

Figure 1-1. Vehicle Design Features and Performance Summary

VEHICLE DESIGN

The electric-powered passenger vehicle design (described in Section 2) emphasizes reduced weight to improve performance, fewer components to maximize reliability and maintainability, good aerodynamic features to minimize drag forces at higher speeds, crashworthiness to increase passenger safety, and appealing exterior features to improve product acceptability. The layout depicted in Figure 1-2 below is unique compared with most conventional car designs because the internal combustion engine and its accessories (such as clutch and transmission, exhaust-air quality protection components, carburetor and fuel system, etc.) have been eliminated. In addition, locating the flywheel energy storage system and some of the batteries to the rear of the vehicle results in a significant reduction in front end mass; improved driver visibility; more front seat interior leg and head room (comparable to a Cadillac); and smoother airflow due to rounded corners, general narrowing in plan view, and elimination of the usual front grille required for power plant cooling. The front batteries are located low so that the space above can be used for storage and instrumentation modules.

The body styling (Figure 1-1) reflects a lightweight vehicle appearance, with wraparound glass, thin roof section, and narrowing at the nose and tail, thus saving material required for manufacture. Weight has been minimized through the use of fiber reinforced plastic (FRP) body panels and innovative arrangement, such as the use of hood structures to perform energy management tasks. Lightweight, deformable bumper fascia panels are incorporated at the vehicle nose and tail to improve crashworthiness. Structural design emphasizes transmission of loads from the fenders and cowl sections into the roof/pillar assembly and floor; the nose structures comprise battery supporting modules designed to isolate damage and permit repair through easy module replacement; and the hood structure is joined mechanically to the inner/outer fender assembly to deform as a unit on collision while preventing displacement into the windshield area.

Figure 1-2. Vehicle Design Provides Comfortable Seating, Improved Crashworthiness, and Reduced Weight

POWER SYSTEM DESIGN

The regenerative power system comprises a flywheel, motor/generator, gearbox/transmission, controller, and battery pack. The layout depicted in Figure 1-3 indicates the general arrangement of the drive train components, which are based on proven technology and can be fabricated with a minimum of new development effort and cost. The battery pack includes 16 standard-size, lead-acid golf cart batteries located to the front and rear of the vehicle. Battery power is supplemented by the regenerative power system (flywheel/generator) that stores energy during braking and delivers all of the power peaks during high-power transient operation. Thus, available capacity of the battery is greater than without a regenerative system. The flywheel is constructed of a lightweight, high-tensile-strength, S-glass/epoxy composite mounted on a spoked aluminum hub; this proven design provides higher strength and increased energy storage capability while reducing burst containment shroud requirements (smaller, lighter housing).

Figure 1-3. Power System Components Require Minimum New Development Effort and Cost

PERFORMANCE CHARACTERISTICS

Performance characteristics were determined using established performance data and a simulation model that equates flywheel-generator-motor kinetic energy to changes in vehicle kinetic energy and losses due to road loads and mechanical friction. The computer programs established that the combination of regenerative braking and battery load leveling in the baseline vehicle design results in a range of 85 miles and an overall energy consumption of 391 watt-hr/mile (see Figure 1-1). Use of flywheel stored energy during high-power demands provides acceleration characteristics that match today's standard gasoline-powered cars, thus improving consumer acceptability of the electric-powered vehicle.

In the computer studies, performance data for potential designs were generated and compared with predicted performance of the baseline design. The suburban driving cycle, SAE J227A(D), was used for most of the studies. The actual power supplied by the battery and from the flywheel (baseline systems) during the five segments of this cycle are summarized in Table 1-2. The computer simulation model and performance predictions are discussed in detail in Section 5.

TABLE 1-2
SUBURBAN CYCLE POWER SUMMARY,
REGENERATIVE FLYWHEEL DECREASES BATTERY POWER DRAIN

Driving Mode	Time Segment, sec	Flywheel		Battery		
		Δ Energy, watt-hr	Ave. Power, kw	Energy, watt-hr	Average power, kw	Current, amp
Acceleration	0 to 28	-85.65	-11.01	-46.83	-6.02	55.74
Cruise	28 to 78	-22.39	- 1.61	-98.74	-7.11	65.83
Coast	78 to 88	43.14	15.53	-19.45	-7.00	64.81
Braking	88 to 97	33.69	13.48	-13.78	-5.51	51.02
Idle	97 to 122	33.28	4.79	-45.28	-6.52	60.37

VEHICLE ECONOMIC FEASIBILITY

The economic analysis of electric vehicle costs provided some of the most important conclusions of the study. The principal parts of the study included determination of the acquisition cost and the ownership costs over a 10-year period. The study results depend heavily on recent published work sponsored by both ERDA and DOT. The study (discussed in Section 6) shows how the vehicle design can achieve an ultimate production cost of \$5000 per vehicle and an overall cost of ownership of no more than 15¢/mile. The economic analysis is summarized in Table 1-3.

TABLE 1-3

**ECONOMIC ANALYSIS SHOWS ULTIMATE
VEHICLE PRODUCTION COST OF \$5000 AND
OWNERSHIP COST OF 15¢/MI MAXIMUM**

Vehicle Component	Present Electric Vehicles per Reference 6-2		Based on Conventional Steel Production (see Figure 6-1)		Based on Current Plastic Production (see Figure 6-2)		Based on Plastic Mass Production (see Figure 6-3)		Based on Mass Production of Plastics and Motors (see Figure 6-4)	
	Weight, lb	Cost, \$	Weight, lb	Cost, \$	Weight, lb	Cost, \$	Weight, lb	Cost, \$	Weight, lb	Cost, \$
Chassis	1625	1490	1756	1530	1193	1453	1193	1199	1193	1199
Motor	315	1260	293	1048	293	1048	293	1048	293	488
Controller	85	1190	10	200	10	200	10	200	10	200
Battery	1500	630	1040	437	1040	437	1040	437	1040	437
Total	3525	4570	3210	3315	2587	3239	2587	2985	2587	2404
Sticker price	(9500) \$12,530		\$7875		\$7623		\$6805		\$5072	
Total cost/mile	\$0.28		\$0.19		\$0.18		\$0.17		\$0.15	

RELIABILITY AND SAFETY

The reliability and safety of the preliminary electric vehicle design are discussed in Section 7. As expected, the simplicity of the electric vehicle will result in much less maintenance and a higher overall reliability than with the increasingly complex automotive designs of today. However, the safety aspects are yet to be demonstrated. The design approach emphasizes achievement of safety requirements through extensive testing and modification efforts until all Federal motor vehicle safety standards are met.

DESIGN OPTIONS AND RECOMMENDATIONS

During the tradeoff analyses conducted in the Phase 1 program, several design options were identified for traction motor configurations and battery arrangement. The most promising of these options are described in Section 8, and are recommended for further study in future program phases.

In another major task of the study, a detailed program for the final design and development of the vehicle was established. This development plan (summarized in Section 9) includes a comprehensive, 3-phase, 54-month development program planned to fully demonstrate the technology, performance, safety, and cost of the prototype vehicles.

Important conclusions of the study program (summarized in Section 10) include the following:

- (a) A regenerative power system is required to meet the near-term goals of the electric vehicle program.
- (b) A lightweight vehicle structure and body is required to meet the cost and energy consumption goals of the program.
- (c) The baseline regenerative power system will substantially increase the acceleration performance of the vehicle, leading directly to greater consumer acceptance.
- (d) Increased cycle life of the batteries is required to ensure attainment of the predicted life-cycle cost of vehicle operation.
- (e) Improved batteries with higher energy density will help to reduce the weight and cost of the vehicle, provided the battery cost penalty is minimized.

SECTION 2

INTRODUCTION

The task of designing a practical, acceptable, and marketable electric passenger car requires the solution of several technical problems that have not been successfully solved in previous design efforts. Historically, battery limitations have been considered the major obstacle in overcoming design deficiencies that prevent mass acceptance. These past attempts also have suffered because of the economic necessity of using available parts and standard materials and methods. Although some minor improvements have been offered, the cost of major technological developments have been beyond the resources of the electric car industry. Thus, the Energy Research and Development Administration sponsored an electric vehicle design program under the guidelines that the electric car must be a new design specifically optimized for its intended use, and with an improved power system that will maximize the power and capacity available from an existing or near-term battery.

The Phase I study of this program, conducted by Garrett/AiResearch under Contract E(04-3)-1213, is complete; this report presents the results of the studies conducted. Phase I was performed in four major tasks as outlined below:

Task 1: Design Tradeoff Studies--The main objective of Task 1 was selection of a vehicle configuration that would best achieve the performance goals of the contract. Tradeoffs were conducted to:

- (a) Show the validity of the unique power system and its practicality of installation in a commuter-type vehicle.
- (b) Evaluate options in component sizes and assembly arrangements.
- (c) Determine system operational characteristics and overall vehicle performance.
- (d) Determine energy requirements and the optimum battery selection for the near-term vehicle design.
- (e) Determine the sensitivity of the system to component variables, technology developments, and costs.

Results of the tradeoff studies are presented in AiResearch report 76-13133.

Task 2: Conceptual Design--In this task, a conceptual design was prepared for the baseline vehicle, showing the overall vehicle configuration and alternate configurations. The location of all major components were defined and illustrated. The external configuration was selected to minimize aerodynamic drag and maintain reasonable internal space for passenger comfort, and was approved by ERDA in a conceptual design review.

Task 3: Preliminary Design--A preliminary design (reviewed in AIResearch report 76-13464) was defined for the baseline vehicle concept approved by ERDA; all power train components were defined, component and system performance characteristics predicted, vehicle weight detailed, external and internal dimensions established, and materials selected for the vehicle body and chassis.

Task 4: Development Plan--A program plan was prepared for a 54-month total vehicle system development effort (discussed in detail in AIResearch report 76-13311). The technical approach, tasks to be performed, schedules, materials, and required facilities were established. Key milestones and major decision points were identified, as well as management techniques, organizational structure, areas of responsibility and authority, and major subcontractors.

The study work conducted in the tasks listed above resulted in a preliminary design that meets all ERDA near-term electric automobile goals. This report describes the vehicle and power system preliminary design and some of the design options that may prove advantageous. Included are the cost, reliability, and safety studies. In addition, conclusions that have been made as a result of the studies are presented, and the development plan that will establish (by test demonstrations) the validity of the vehicle design to achieve the intended goals is summarized.

SECTION 3

REFERENCE VEHICLE PRELIMINARY DESIGN

DESIGN FEATURES

Using design tradeoff studies and conceptual design review discussions with ERDA as a basis, Garrett/AiResearch has completed the preliminary design of a reference (baseline) electric-powered passenger vehicle system with the potential of operating efficiently in all standard driving modes using lead-acid batteries available in the near future. The power system also incorporates a regenerative energy storage flywheel that stores energy during braking and delivers this energy to the vehicle power train during high-power transient operating modes (hill climbing, passing, etc.). Through the efforts of the Brubaker Group, under subcontract to Garrett/AiResearch, interior and exterior features and styling have emphasized simplicity, good aerodynamic performance, high reliability, ease of maintenance, and sufficient crashworthiness to ensure public acceptance and appeal. An artist's rendering of the baseline vehicle is depicted in Figure 3-1 below. Salient features of the vehicle are summarized in Figure 3-2.

Figure 3-1. Body Styling Emphasizes Good Aerodynamic Performance, Appeal, and Safety

CHASSIS AND BODY

LAYOUT	REAR DRIVE AND POWER SYSTEM, BATTERY PACKS AT FRONT AND REAR
BODY/FRAME	FRP CONSTRUCTION
SUSPENSION	FRONT MCPHERSON STRUTS, REAR CHAPMAN
STEERING	RACK AND PINION
PRIMARY BRAKES	RECUPERATIVE INTO POWER SYSTEM
SECONDARY (FRICTION) BRAKES	DISC FRONT, DRUMS REAR
PARKING BRAKES	MECHANICAL, OPERATING ON REAR DRUMS
TIRES	STEEL-BELTED RADIALS
POST STRUCTURES AND WINDOW TRIMS	SATIN BLACK FINISHES, WINDOW TRIM FLUSH WITH GLASS AND POST SURFACES
HEAD LAMPS	RECTANGULAR (200 BY 1 1/2 MM)
WINDSHIELD	STEEPLY RAKED, SLIGHTLY "V" PLAN VIEW
GREENHOUSE SURFACES	TUMBLEHOME AND ROUNDED CORNERS
DOORS	FULL WINDOW RETRACTION (OUTSIDE DOOR STRUCTURE)
WHEELWELLS	CONFIGURED CLOSELY TO TIRES
WIPERS	SINGLE, SIMILAR TO SC ROCCO

INTERIOR ACCOMMODATIONS

SEATING	FOUR PASSENGERS
FRONT SEAT LEG AND HEAD ROOM	COMPARABLE TO A CADILLAC
REAR SEAT	EQUIVALENT TO GENERAL MOTORS INTERMEDIATES
VEHICLE WIDTH AT ELBOW LEVEL	7 IN. MORE THAN VW RABBIT
INTERIOR DOOR AND QUARTER PANELS	SCULPTURED OUT
STORAGE	REAR SEAT FOLDS FORWARD, AND DEEP FORWARD STORAGE COMPARTMENT IN INSTRUMENT PANEL (10 BY 20 BY 30 IN.)
VENTILATION	COMPLETE FLOW-THROUGH

ACCESSORIES AND OPTIONS

HEATER/DEFROSTER	ACCESSORY
RADIO	OPTION
REAR WINDOW	OPTION AS HATCHBACK
BATTERY CHARGER	ACCESSORY

READOUTS

DIGITAL SPEEDOMETER	ELECTRONIC DIGITAL SPEEDOMETER LOCATED IN THE REAR VIEW MIRROR (ALSO DISPLAYS REMAINING RANGE)
INDICATORS	BRAKES, SEAT BELTS, ETC.

Figure 3-2. Salient Features Summary

VEHICLE DEFINITION

The preliminary design of the electric-powered passenger vehicle is shown in Drawing BOD30031A. This layout, reduced from a one-fifth scale drawing, depicts the power system arrangement, divided battery packs in the front and rear compartments, and location of the energy storage power system in the rear of the vehicle to improve front-end crashworthiness. Major vehicle assemblies are divided into body components and drive train components, and include the following subsystems:

- Body Subsystems

- Primary structural group/body
- Main body components
- Door group
- Hood group
- Rear deck/hatch group
- Seating group
- Trim group
- Lighting group
- Environmental control systems
- Controls and instrumentation

- Drive Train Subsystems

- Suspension system
- Power system/drive line

A detailed breakdown of the components comprising each of the categories listed above is presented in Table 3-1.

In the arrangement of the body components, ample storage space is provided in the rear trunk area above the battery pack; access to the battery pack is provided through the rear hatchback. The front battery pack is located low, and the space above can be used for storage. The front hood is designed as a structural load carrying member to help transmit a forward crash load into the main structure beginning at the A posts, thus increasing stiffness of the entire front compartment and increasing absorption of frontal crash energy. This is also one of the many weight-effective features of the design. For vehicle control, low-ratio rack and pinion steering is used to ease driver turning effort, and the steering column is collapsible to meet the applicable requirements of Federal Motor Vehicles Safety Standard (FMVSS) No. 203.

The drive train system comprises suspension system and power plant/drive line. The layout of the four-wheel independent suspension system utilizes McPherson struts in the front and Chapman struts in the rear. Battery power is supplemented by a regenerative power system that stores energy during braking and delivers all of the power peaks during high-power transient operation; thus, the available capacity of the battery is greater than without a regenerative system. In this way, both vehicle acceleration and range in city driving is increased substantially. Furthermore, it is expected that battery cycle life will be increased due to the less severe duty cycle. The power system/drive line components are discussed in detail in Section 4.

TABLE 3-1

ELECTRIC-POWERED PASSENGER VEHICLE COMPONENT
BREAKDOWN (PRELIMINARY)

SUBSYSTEM	COMPONENTS	
BODY		
Primary Structural Group/Body	<ul style="list-style-type: none"> • Floor pan (incl. seating, sills and reinforcing members) • Firewall/cowl assembly • Upper/lower A-post assembly, right • Upper/lower A-post assembly, left • B-post assembly, right • B-post assembly, left • C-post assembly, right • C-post assembly, left • Rear quarter panel, right 	<ul style="list-style-type: none"> • Rear quarter panel, left • Inner rear wheelwell strut, right • Inner rear wheelwell strut, left • Roof skin • Front subframe assembly • Rear subframe assembly • Front fender assembly • Rear fender assembly • Front bumper assembly • Rear bumper assembly • Sandwich materials
Main Body Components	<ul style="list-style-type: none"> • Windshield • Backlight • Side glazing, right (not incl. door) • Side glazing, left • Door hinge anchorages • Door latch anchorages • Hatch hinge anchorages • Hatch latch anchorages • Seat belt anchorages, front and rear • Shoulder belt anchorages • Pedal assembly support structures • Steering column support structures 	<ul style="list-style-type: none"> • Seat support structures, front and rear • Subframe mounting points • Battery mounting structures • Hood latch attach structures • Window seating/sealing • Instrument panel structures • Sound insulation materials • Grommets/boots, body shell penetrations • Windshield/washer/wiper system • Wiring harness/fittings • Parking brake assembly • Body subframe elastomeric mounts
Door Group (Right and Left Facia)	<ul style="list-style-type: none"> • Inner skin • Outer skin • Sandwich layer • Outer handle/latch assembly • Inner handle/latch assembly 	<ul style="list-style-type: none"> • Weather stripping • Glazing • Glazing retraction mechanism • Glazing weather seal • Door hinge assembly
Hood Group	<ul style="list-style-type: none"> • Outer skin • Inner reinforcement • Hinge attach structure • Hinges 	<ul style="list-style-type: none"> • Hood latch attach structure • Latch • Weather stripping • Front bumper facia panel
Rear Deck/Hatch Group	<ul style="list-style-type: none"> • Outer skin • Inner reinforcement • Hinge assembly 	<ul style="list-style-type: none"> • Latch/lock assembly • Weather stripping • Rear bumper facia panel
Seating Group	<ul style="list-style-type: none"> • Front seat assemblies • Rear seat assemblies • Front and rear seat belts 	<ul style="list-style-type: none"> • Front shoulder belts • Seat adjust hardware, front • Head restraints

TABLE 3-1 (Continued)

SUBSYSTEM	COMPONENTS	
Trim Group	<ul style="list-style-type: none"> ● Paint/primer ● Exterior rub strips ● Exterior ID/nameplates ● Carpeting, fwd/rear components ● Instrument panel trim/padding ● Visors ● Rear view mirrors, inside/outside 	<ul style="list-style-type: none"> ● Exterior trim/brightwork ● Headliner, header, and post trim ● Door side panels, right and left ● Package shelf ● Ash tray(s)/lighter(s) ● Kick panels
Lighting Group (Including Mounting Hardware)	<ul style="list-style-type: none"> ● Headlamp assembly, right/left ● Tail lamp/brake light assembly, right/left ● Back-up light assembly, right/left 	<ul style="list-style-type: none"> ● License plate light ● Front side markers, left/right ● Rear side markers, left/right ● Interior lighting ● Luggage compartment lighting
Environmental Control Systems	<ul style="list-style-type: none"> ● Heater, ducting, and controls, complete 	<ul style="list-style-type: none"> ● Ventilation controls and ducting, complete
Controls and Instrumentation	<ul style="list-style-type: none"> ● Steering column and wheel assembly ● Pedal assembly 	<ul style="list-style-type: none"> ● Misc. hand controls ● Instruments/gages ● Electronics
DRIVE TRAIN		
Suspension System	<ul style="list-style-type: none"> ● Front tires, wheels, nuts ● Rear tires, wheels, nuts ● Front suspension, incl. brakes (McPherson struts) 	<ul style="list-style-type: none"> ● Rear suspension, incl. brakes (Chapman struts) ● Steering rack and arms (rack and pinion)
Power System/Drive Line	<ul style="list-style-type: none"> ● Flywheel assembly ● Generator ● Traction motor ● Differential, half shaft (front/rear) 	<ul style="list-style-type: none"> ● Controller (choppers, microprocessor, control interfaces) ● Power supplies, grounding ● Sensors, instrumentation ● Battery pack onboard charger

DESIGN CHARACTERISTICS

Major vehicle design characteristics are presented in Table 3-2. The parameters listed include data drawn from other sections of this report. The performance data are considered moderate design goals, and summarize the characteristics discussed in detail in Section 5. The drive train and power system information is drawn from the design details presented in Section 4. Design characteristics for the vehicle body are discussed subsequently in this section. The baseline system vehicle dimensions are 158 in. long by 70 in. wide by 57 in. high. The 95-in. wheel base is optimum for the vehicle battery distribution shown in drawing BOD30031A. The vehicle curb weight of 2566 lb is based on existing data for the components listed in Table 3-1 and a battery weight of 1040 lb. The low vehicle weight--a highly important goal in electric vehicle design--has been achieved through stringent design review of existing automobile manufacturing materials and use of new materials technology where feasible and practicable. In addition to the use of lightweight FRP body structures, weight is further minimized by eliminating much of the "extra" equipment normally provided on conventional automobiles. Examples include use of a two-headlight system instead of four, reduced instrumentation, and use of a single-arm wiper rather than a double arm/linkage system. Most of the instrument panel area is major transverse structure, with only minor trim and detailing added.

Assuming favorable marketing analysis results and consumer acceptance, additional weight savings will be achieved through implementation of a number of innovative design concepts proposed for the baseline electric-powered passenger vehicle. For the vehicle interior, customer extras such as padded glove box doors, fully upholstered door and quarter panels, extra-thick carpeting, deluxe trim, etc. have been eliminated. The conventional window drop mechanism has been replaced with simpler, lighter hardware, and the window is lowered outside the door structure rather than through the door structure. This makes for a stronger door, the inner structure incorporating textured finishes that do not need to be covered with secondary panels or additional materials. The window retracting bay is simply covered by a soft, vacuum-formed component, placing soft upholstery only where it is really needed. Lightweight, 0.187-in.-thick glass is utilized for vehicle side glazing in both door and rear quarter windows. Acrylics or polycarbonates (Lexan) glazing materials have been evaluated on a continuing basis by The Brubaker Group for automotive glazing, and although they have the potential for reducing weight, they have drawbacks in durability (resistance to scratching, even with hard coatings), stiffness, and cost. Further development may overcome these problems.

The most meaningful sources of body weight reduction will be in the intelligent selection of materials and optimization through innovative body panel arrangements, such as utilization of hood structures to perform energy management tasks. Another example would be the use of completely closed box sections on all roof pillar, header, and transverse structure rather than the open "C" sections used in conventional vehicles. Throughout the vehicle, intelligent decisions have been made with respect to hardware and functional components. If a given item can be deleted entirely, this provides the best results in terms of weight reduction. If it cannot be deleted or combined with another element, the lightest, simplest system has been selected.

TABLE 3-2
MAJOR VEHICLE CHARACTERISTICS

Parameter	Value
General	
Overall length, in.	158
Width, in.	70
Height, in.	57
Wheelbase, in.	95
Curb weight, lb	2566
Gross weight with 4 passengers, lb	3166
Vehicle inertial weight, lb	3324
Vehicle effective drag area, $C_D A$, sq ft	7
Rolling resistance coefficient	0.010
Power system type	Regenerative
Controller type	Ward-Leonard
Acquisition cost, dollars	5000
Ownership cost per mile, \$/mi	0.15
Drive Train/Power System	
Battery weight, lb	1040
Battery capacity, kwhr	18.7
Battery voltage, vdc	108
Battery recharge time, hr	6
Flywheel transmission type	Electromechanical
Flywheel capacity, kwhr	1.0
Flywheel maximum speed, rpm	25,000
Power system weight, lb	303
Final drive gear ratio	2.88 to 1
Performance	
Speed	
Cruise speed, mph	50/55
Maximum speed, mph	70
Speed up 5% grade, mph	50
Acceleration	
0 to 30 mph, sec	6.4
0 to 55 mph, sec	15.1
25 to 55 mph, sec	10.0
Range	
City driving, mi	85
50-mph cruise, mi	82
55-mph cruise, mi	70
	} Based on 18 wh/lb at C3 rate
Energy consumption for SAE J227D cycle, kwh/mi*	0.391

* Energy input to battery charger

BODY DESIGN

While a great deal of discussion with respect to vehicle styling predicts "boxy" shapes for future vehicles, this formula cannot be uniformly applied to all vehicle types or to all marketing situations. Domestic cars have been patterned after the Mercedes package, and in so doing have improved occupant accommodations and reduced vehicle weight. However, a "boxy" shape is not vital to excellent interior accommodations, and a squared-off vehicle shape does not provide the best occupant protection, aerodynamic characteristics, or manufacturing costs on a material basis. For example, the Jaguar is an extremely trim vehicle for its interior size, is not boxy, and receives very high ratings on its general appearance. It is along these lines that the proposed four passenger EV styling has been developed. Body design has been kept simple and clean to optimize product acceptance, provide good aerodynamic performance, and ensure ease of manufacture.

The body styling reflects a lightweight vehicle appearance (see Figure 3-1). The wrap-around glass, thin roof section, and character lines high on the body sides and ends enhance this effect. The tucked-under rocker panels contribute to remove "visual weight" from this area, low between the wheels, while small flares have been incorporated to prevent mud and stones from damaging the body finish. Light, quick, and trim in appearance, the greenhouse* incorporates tumblehome and rounded corners for good airflow and drag reduction, but without compromising occupant head room. The lower body incorporates substantially more plan view contouring than conventional vehicles, narrowing at the nose and tail, and thus saves material required for manufacture (and weight).

The individual panels and panel separations have been minimized for quality control and ease of repair or replacement in the field. The lower body (below the greenhouse) consists of only seven primary OML panels--soft nose, soft tail, hood (1 piece), left door, right door, and left and right rear quarter panels. (Most two-door automobiles have 9 to 11 OML panels). An exploded view of the complete body structure is shown in Drawing BOD30032A.

Exterior Features

The vehicle exterior has been designed with large, glazed areas placed flush to each other for wind-noise reduction and overall drag reduction. The entire greenhouse area could be standardized for production with finishes that are the same for each vehicle manufactured. Pillar finishes in black and roof exteriors in black or off-black satins or metallics could be matched with any customer selected lower body color desired, and would result in substantial

* The greenhouse area is considered as everything above the bottom edge of the windows.

3-10

THE BRUBAKER GROUP 10315 W. PICO BOULEVARD LOS ANGELES, CALIFORNIA 90064 (213) 277-6251	
DRAWN BY 2/25/11 DESIGNED BY 11/1/11	CHECKED BY 11/1/11 DATE 11/1/11
TITLE ELECTRIC-POWERED PASSENGER VEHICLE DESIGN STUDY PROBLEM PARTS BREAKDOWN	REV. R 1/4 REV 10030032 A
MATERIAL 6061-T6 ALUMINUM	PART CONDITION NEW
PART ASSEMBLY DO NOT SCALE DRAWING	SHEET 1 OF 2

cost savings on the line. Although the A, B, and C posts are large in section (as required), they do not detract from vehicle appearance because the pillar structures are located substantially beneath the glazed areas. All post structures and window trims of development test vehicles to be delivered to ERDA will be finished in non-glare, satin black finishes. All post cross-sections are designed to present the least possible obstruction to the driver's field of vision. In some instances, asymmetrical configurations are used to attain this objective. Body hardware such as windshield wipers, handles, locks, latches, graphics, and other projections have been deleted or incorporated in such a manner as to minimize aerodynamic drag and eliminate unnecessary hazard to pedestrians or cyclists in the event of accidental impact.

Lightweight, deformable bumper fascia panels have been incorporated at the vehicle nose and tail (See Figure 3-3). These units are configured to sustain continued abuse at low speeds with little or no cosmetic or functional damage, and to provide substantial additional protection for pedestrians in the event of an impact. The section of soft fascia has been doubled at each corner of the vehicle to allow greater deformation without damage in this vital area.

Figure 3-3. Vehicle Structural Features

Lighting systems, both front and rear, have been set back from the bumper impact zone to avoid damage during bumper stroke and to protect pedestrians or cyclists from contact with hard surfaces in the event of collision. The newly approved 200 by 142 mm rectangular headlamps (one on each side) have been designed into the vehicle front end. The headlight module includes the side marker and turn signal units. The low headlamp profile and location well aft of the bumper permits a steeply raked hood and low nose for improved aerodynamics and vision. Bumper corners are well rounded to improve aerodynamic performance and maximize pedestrian safety. The tail light and backup/license plate light system also are positioned well forward of the impact area and away from corners of the vehicle.

The entire rear window is configured to become a hatchback at a slight addition in weight due to hinges, latching hardware, and weather seal. The proposed design does not require an additional frame on the backlight (as shown in Figure 3-4), and the transverse structure just forward of the rear bumper fascia provides a high degree of body structural integrity. A rear view of the vehicle is shown in Figure 3-5. A full hatch or trunk lid may be incorporated if desired. The rear quarter windows are fixed units, providing additional strength in the B and C pillar areas, but may be hinged out from the B post at customer option. Complete flow-through ventilation utilizing the B and C posts has been incorporated.

S-11547

Figure 3-4. Frameless, Flush Backlight

Figure 3-5. Rear View of Electric-Powered Passenger Vehicle

Side glazing in the door area allows for full window retraction, although other concepts may be used to improve safety, increase crashworthiness, and strengthen the structure. The window is lowered outside the door structure rather than through the structure, thus allowing a stronger, lighter door design (see Figure 3-6 for door details).

The vehicle hood and outer front fenders are manufactured as a single unit. Because of the low hood profile and unique components of the electric vehicle design, the hood is not designed to function as a frequently removable panel as it is in conventional vehicles. This approach has led to some innovative structural concepts for crashworthiness and allows the use of more efficient interior instrument panel configurations.

Structural Design

All structures of the electric-powered passenger vehicle have been designed not only for load carrying capability, but to maximize crashworthiness--to protect the driver and passengers in case of high-speed collisions and to minimize damage upon minor, low velocity collisions.

The nose structures comprise battery supporting modules designed to isolate damage to the greatest extent possible and to permit repair through easy module replacement. Inner front fenders and cowl structures are located to conduct crash loads as directly as possible into the roof/pillar assembly and floor. The angle of the A and C pillars transmits the loads from the lower body through to the roof.

Figure 3-6. Door Structure

S-11548

The motor, generator, and flywheel components placed to the rear of the vehicle improve front end crashworthiness because all forward components will collapse without danger of penetrating the passenger compartment. Because there is no powerplant in the vehicle nose that must be installed or removed periodically, new approaches have been taken with respect to structure. Road loads from the front suspension normally enter the inner/outer fender assembly, which has no transverse structure except further aft at the cowl. Because no engine is mounted in this area, lightweight structural sections such as beams or gussets are included as part of the permanent vehicle structure. Although the hood functions as a load carrying element during impacts, it is not required to take normal road or jacking loads, and the vehicle may be safely driven with the hood off. Thus, a substantial reduction in thickness of material and reinforcements has been achieved in the preliminary design compared to conventional structures, and a net reduction in weight realized.

One of the more innovative solutions for improved front end structure with decreased weight results from the use of the inner and outer hood for energy management. The hood on conventional automobiles is a "hang-on" component, often only buckling away from the front fender on impact. Due to the limited need for access to the nose area, the proposed electrical vehicle design mechanically joins the hood structure to the inner/outer fender assembly to deform as a unit on collision. The attachment method prevents displacement into the windshield area, but allows easy access to the nose for routine inspection, service, or replacement of parts.

Although further testing will be required to substantiate the concept, it appears that battery packages could function as an excellent energy absorber. Battery construction is not unlike the early experimental water bumpers and present freeway barriers, and while containment of acid and water must be considered, controlled energy absorption is available. In a minor collision, the bumper structure and fascia panel (depicted in Figure 3-3) absorbs energy and restores itself without permanent damage. On high-speed impacts, the batteries and their containing structures are designed to deform at a controlled rate. These structures will be engineered to optimize energy management without incorporating additional systems, weight, or cost. Battery compartments have been located at the front and rear of the vehicle for this purpose.

The plastic body panels used in the electric vehicle are similar in configuration to conventional, steel-stamped panels, except that greater surface area and more complex shapes can be formed in a single operation. An additional advanced design feature is that thickness may be varied in critical areas so that materials may be more efficiently used. Although initially more expensive to produce and to assemble, FRP construction will lend itself to automated production, and eventually to the manufacture of more efficient automotive bodies. The composite body must incorporate more precise joining and bonding techniques, and will therefore result in a quieter, stronger, corrosion-free automobile. Methods of determining the location and extent of structural damage are readily available in FRP parts. Techniques that will be used to facilitate replacement of panels, such as color-coded bond surfaces and modular panel assemblies, permit reasonable and relatively low-cost repairs of FRP automobiles. Due to their high cost and weight, the use of foams has not been included; these may be considered for insulating as required. Certain portions of the electric vehicle

cowl structure may contain urethane foams for this purpose. Substantial additional testing of FRP in body structures will be required to determine the exact engineering and economic factors in support of full-scale production with these materials.

Interior Design and Configuration

Compared to most small-car designs, the interior accommodations of the electric-powered vehicle are superb. Front seat leg and head room dimensions are comparable to a Cadillac, and rear seat accommodations are equivalent to General Motors intermediate cars. Efforts have been made to produce a visually "open" and uncluttered environment, as shown in Figure 3-7. The vehicle is very roomy at elbow level, being about 7 in. wider than a VW Rabbit. Interior door and quarter panels are sculptured out to avoid the boxy, flat-sided effect of most contemporary small cars.

The rear seat folds forward to become a luggage compartment extension, which is a basic feature in several late-model cars. Somewhat more unique, however, is a deep, forward storage compartment in the instrument panel to the right of the driver. Without compromising structural integrity, a compartment averaging 10 by 20 by 30 in. can be located immediately in front of the front seat passenger and recessed forward into the nose section. The compartment may be concealed by a soft, hinged door immediately below the upper instrument panel structure. The storage area provides space for items substantially larger than the normal glove box, such as bulky clothing, tennis rackets, food, or other travel items. With this alternative to in-car storage, the forward exterior hood need not be opened except for repair or periodic service.

1. Controls

The design approach for location of vehicle controls is unique in that the traditional electrical vehicle location in the dash is not used because it is potentially injurious in a second collision, and use of an inertia reel shoulder harness restraint system can make them inaccessible in an emergency (if the driver moves too quickly, the inertia reel can lock and prevent his reaching the control). As in conventional internal combustion vehicles, the accelerator and brake for the electric vehicle are located for right-foot operation. In addition the steering column is positioned so that use of both feet on both pedals simultaneously is difficult and uncomfortable. This feature has been incorporated in the design to overcome poor habits developed in conventional vehicles (use of brake and accelerator simultaneously) that would substantially increase battery power draw and reduce range. An additional feature to reduce this possibility is a footrest for the left foot at pedal height.

2. Instrumentation

A typical instrument panel has 93 components packed into 9200 sq in. of space. Packaging and servicing these components is a major problem. The creation of new "gadgets" such as cruise controls is further complicating the problem, and manufacturers are turning to electronic solutions in an effort to save space. Low-cost integrated circuits are being utilized, and digital electronic readouts can be expected as OEM very soon. (They already exist as aftermarket speedometers and miles/gallon monitors).

Figure 3-7. Vehicle Interior View

In the proposed electric vehicle, the two most important readouts--speed and range remaining--are provided in a display that is part of the rear view mirror, as shown in Figure 3-8. A readout of time is provided as an option. Speed is correlated electronically with the remaining charge, and a readout in remaining miles (less a suitable safety factor) is provided. Electronic devices are used because they are smaller and lighter (3 to 5 lb is saved for the speedometer alone) and they can share power supplies and circuitry with the functional equipment of the car, as well as being located in a more convenient place than under the dash. Also, their power consumption is minimal (a sophisticated calculator operates for 15 hr on two conventional penlight batteries) and reliability is high. Additional indicators for brakes, seatbelts, etc. are provided in the dash. Some indicators relating to the power system, such as flywheel speed and battery charge, are provided as options.

Figure 3-8. Digital Speedometer/Readout Mirror

PRODUCT ACCEPTABILITY CONSIDERATIONS

To effectively implement ERDA near-term and long-range goals, the electric-powered vehicle design must not only provide the performance capabilities required to compete with existing passenger vehicles, but must also promote user acceptance and instill mass appeal. To achieve these goals, the preliminary vehicle design has been directed toward providing desirable interior and exterior features and performance characteristics. In addition to weighing potential safety, production, performance, and other criteria on a cost/benefit basis, design concepts were evaluated with respect to market acceptance trends. By locating the flywheel in the rear of the vehicle, constraints for the nose design were limited to providing reasonable headlight and bumper height requirements. As a result, significant reduction in front end mass; improved vision; and smoother airflow due to rounded corners, a general narrowing in plan view, and the lack of the usual grille openings for powerplant cooling were achieved.

To improve passenger appeal and aerodynamic performance, the windshield is steeply raked and has a slight "V" plan view, with window trim generally flush with the glass and post surfaces. An interesting A-post treatment is used to reduce drag and wind noise, and to generally improve the flow of air and rainwater in this area. A blade that doubles as the outboard windshield trim is incorporated into the A pillar structure. Air and water are channeled into the groove created by the blade, and forced up and into the drip rail over the door. Thus, with the side glass brought flush with the A-post structure, considerably less flow separation occurs, reducing side window turbulence and preventing dirt and water buildup in foul weather. The balance of the greenhouse surfaces are flush, softly contoured, and generally rounded. The backlight incorporates a deep "V" contour in plan view and an approximately 30-deg slope rearward from horizontal. With the clean front end configuration, airflow over the greenhouse area will remain attached, and converge smoothly above the vehicle tail.

The lower body is widest at the front door-cut and narrows in plan view toward the rear of the vehicle, which results in a drag reduction of 5 to 12 percent as compared to a straight section. The aft lower body is cut sharply at the rear bumper for clean airflow separation and minimum drag. Wheelwells are configured closely to the tires for improved airflow and to minimize impact problems in the event of collision with pedestrians, cyclists, or other vehicles. The overall body is generally devoid of projecting hardware, which contributes to drag at higher speeds. A spoiler is incorporated as part of the bumper fascia panel to reduce drag and front end lift at high speed.

Although the proposed vehicle design has the potential for meeting all ERDA goals, it is imperative that a design freeze not be imposed on configuration, solution, or concept until all of the factors that must be considered have been optimized through further vehicle development. For example, battery location within the vehicle was a major problem with respect to maintaining desirable interior and exterior features and performance characteristics. During the Phase I program, some interesting locations for the battery package were examined, including installation in a central tunnel or in either rocker sill. However,

these arrangements were eliminated in favor of others that would not compromise the basic marketing requirements of future vehicles. Present and future demands for increased interior space and ease of entry and exit (flat floor and low rocker sills) will greatly influence vehicle sales, and the design and manufacturing concepts incorporated in the four-passenger electric-powered vehicle must remain flexible.

Another area that may require modification during future development efforts is in the headlamp design. The headlamps are set rearward in recessed wells, and airflow through this area could be improved with the addition of glass covers over the opening. This would require special development and certification to ensure minimal losses in light transmission and thorough sealing, and although the concept would increase costs and manufacturing complexity, some manufacturers may want to consider the possibility. Also, several innovative door glazing concepts have been proposed that reduce weight, lower production and assembly costs, improve door strength, and enhance interior crashworthiness. Despite these advantages, the more conventional roll-down glazing may prove to have greater consumer acceptance. What appears to be an advantage in the electric vehicle design may become a source for great concern and continued engineering refinement. Finally an electric vehicle is quiet. So quiet, in fact, that once the ambient noise of the normal conventional internal combustion engine is removed, the consumer becomes aware of an entirely new range of sounds; although in a substantially lower db range, these sounds may be just as annoying to the operator. Road noise conducted through the suspension system and body will be the most noticeable, followed by body squeaks, rattles, and noise from drive-line, powerplant controls, and interior accommodations. The basic body design, anticipated assembly process, and component installation in the preliminary vehicle design is intended to minimize this new source of noise, but development testing may indicate that further design refinements are required.

SECTION 4

BASELINE POWER SYSTEM

DESCRIPTION AND OPERATION

The baseline power system for the electric powered passenger vehicle combines the flywheel and the traction motor to provide a new, unique arrangement that results in an infinitely variable clutchless, stepless, transmission operating from zero to full forward speed. The arrangement is shown schematically in Figure 4-1 and a preliminary layout is shown in Drawing L208684. The system comprises the flywheel assembly, planetary gearbox, two identical motors, the electronic controller, and the battery pack. One motor is geared to the planetary ring gear and is called the generator and the other is geared to the planetary carrier shaft and is called the motor. Both motors are used alternately in the generator mode and in the motor mode. The combined electrical rating of the motors is approximately the same as the single traction motor in the conventional drive system.

Figure 4-1. Power System Schematic

INFORMATION CONTAINED HEREIN IS UNCLASSIFIED DATE 01-20-2001 BY 60322 UCBAW/SJS/STP		AUTHORITY: 50 USC 3605 (a)(2) DATE 01-20-2001 BY 60322 UCBAW/SJS/STP	
1. TITLE AND SUBTITLE FLYWHEEL & TRANSMISSI PROTOTYPE ELECTRIC CAR		2. AUTHOR(s) 70210 L 308534	

The armatures of the two motors are directly connected. The magnitude and direction of the current are determined by controlling the field excitation of each motor. The battery also is connected to the armature circuit and the current is determined by the armature voltage.

The two principal advantages of the system are regenerative braking and load leveling of the battery current; these greatly improve the available energy. The time averaged power requirements for a given driving cycle is provided by the battery pack. The flywheel absorbs the transient, above average power demands enabling the battery to discharge at a near-optimum rate. For power requirements below the average, such as moderate speed cruising or short term idling, the excess battery power is used to recharge the flywheel. During this type of recharging, the generator at the ring gear operates as a motor energized by the output from the battery. The major characteristics of the power system are summarized in Table 4-1.

TABLE 4-1
BASELINE POWER SYSTEM MAJOR CHARACTERISTICS

Parameter	Value
Flywheel	
Total energy, w-hr	1000
Available energy, w-hr	750
Maximum speed, rpm	25,000
Loss at maximum speed, w	380
Total assembly weight, lb	61
Traction motor	
Rated power, hp	13.7
Rated speed, rpm	12,000
Nominal efficiency, percent	88
Weight, lb	106
Generator	
Rated power, hp	13.7
Rated speed, rpm	12,000
Nominal efficiency, percent	88
Weight, lb	106
Gearbox weight, lb	20
Controller weight, lb	10
Total power system weight, lb	303
Battery pack weight, lb	1040

STARTUP OPERATION

Startup is accomplished by accelerating the flywheel to operational speed using the generator as a motor powered by the battery or by the onboard charger. The motor connected to the output wheels is held stationary. When the system is ready for use, the motor can be switched to either forward or reverse to start the vehicle. The battery voltage is reduced through the chopper at the initial low speeds.

ACCELERATION MODE

During acceleration, the generator supplies most of the armature current and the battery supplies a low current dependent on the speed of the flywheel. (A description of the battery controller is presented in Section 5 under Battery Performance). The flywheel decelerates reacting to the torque applied by the generator at the ring gear. The generator torque is transmitted as mechanical power through the planetary output shaft and as electrical power through the generator/motor armature circuit. The overall efficiency of the combined power transmission exceeds the product of the motor and generator efficiencies because of the higher efficiency of the mechanically transmitted portion of the power. Maximum acceleration is achieved (when the throttle is fully depressed) by automatically switching the battery current off. This increases the generator current to its maximum value, which increases the energy extraction from the flywheel.

DECELERATION AND BRAKING

Electrical braking is used to decelerate the vehicle and recover a substantial portion of the vehicle kinetic energy. The deceleration case is the reverse of the acceleration mode. The armature current is reversed by the controller action, which adjusts the motor and generator fields. The motor, acting as the generator, supplies current to the generator; the generator, acting as a motor, accelerates the flywheel.

A small amount of deceleration is initiated when the throttle pressure is removed thus simulating the normal compression braking of piston powered vehicles. Brake pressure then increases the electrical braking up to a maximum beyond which the friction brakes will be blended in. The electrical braking will be sufficient for all normal driving at speeds as low as about 8 mph, such as during the Federal City Driving Cycle. Below this speed, the friction brakes will be blended in to stop the vehicle.

HILL CLIMBING

The kinetic energy storage capability of the flywheel is sufficient to provide for a limited amount of hill climbing without the high battery current demands normally required. The uphill case is similar in operation to the acceleration case. The generator is supplying a major portion of the current to the motor and the flywheel is supplying a major portion of the energy. As the flywheel energy is depleted, the battery current gradually increases assuming more of the total load. Downhill operation is similar to the deceleration case such that the potential energy from the wheels is delivered back into the flywheel. If the downhill grade is of sufficient length to cause the flywheel to reach its maximum speed, the friction brakes will be automatically blended in by the controller.

CONSTANT SPEED CRUISE OPERATION

Constant speed cruise operation is similar to the conventional traction motor powered system; all the energy must be supplied by the battery except for the initial acceleration requirement. However, for constant speeds up to 55 mph without the acceleration requirement the battery efficiency is very good; therefore, acceptable vehicle ranges are achieved even though the battery also is used to supply a small amount of makeup energy into the flywheel.

FLYWHEEL DESIGN

The primary variables that must be considered in the design of a flywheel are material choice, shape, peripheral tip speed, and rotational speed. Other design considerations include the vacuum environment, containment housing design, and operational environment (such as the cycle life and the generation of gyroscopic loads induced by external angular velocities of the vehicle). Steel disc flywheels are currently in use in railcar propulsion systems and in some experimental cars. The success of these applications has stimulated the development of higher energy density wheels using composite materials.

The baseline flywheel design comprises a rim made of several annular cylinders of S-glass/epoxy composite mounted upon a spoked aluminum hub (Figure 4-2). This design allows concentration of a high-tensile-strength S-glass/epoxy laminate at the rim of the flywheel, where it is most effective. The rim is mounted on a hub containing three contoured aluminum spokes that combine low weight with high strength under the centrifugal loads of rotation. This design is based on a new concept that has been proven in full-scale laboratory testing, including 1000 full-speed deep-discharge cycles. The laboratory testing was performed with a flywheel made from E-glass and epoxy material that was cycled to 1920 ft/sec. The flywheel weighed 11.08 lb, stored 0.242 kwh of energy, was made by conventional winding techniques, and exhibited no special problems or failures during the entire test program.

Figure 4-2. S-Glass/Epoxy Composite Flywheel for Electric Commuter Vehicle

The flywheel characteristics are summarized in Table 4-2. The higher strength S-glass (as compared to E-glass) has been under development at AiResearch for the past 2 years. The use of S-glass requires the same winding and processing techniques as used with the successful E-glass flywheel, and the material properties are well known. However, the higher strength of S-glass permits it to be designed for operation at 2500 ft/sec, for a 63 percent increase in energy density over the E-glass, and with the same long fatigue life. Use of this composite flywheel design in the electric vehicle results in a much lighter system as compared to the steel disc flywheels developed for onboard energy storage in rail systems. Also since they are of higher strength and of a lighter material, much weight is saved in fabrication of the burst containment shroud.

The baseline flywheel is mounted in an aluminum housing that supports the bearings and seals, and provides an evacuated enclosure around the flywheel. The housing has thin walls for low weight, but is adequately ribbed to provide for stiffness against collapse from evacuation. The flywheel is supported by two straddle-mounted bearings that are located within the partially evacuated housing. The bearings are oil lubricated, with the lubricant being pumped out through the vacuum pump system.

TABLE 4-2
SUMMARY OF FLYWHEEL CHARACTERISTICS

Parameter	Value
Flywheel 100 percent speed, rpm	25,000
Energy storage at 100 percent speed, watt-hr	1000
Available energy (down to 50 percent speed), watt-hr	750
Flywheel size	
Rim OD, in.	23
Rim ID, in.	18.4
Rim length, in.	2.8
Polar moment of inertia, ft-lb-sec ²	0.7750
S-glass weight, lb	33
Hub weight, lb	12
Housing weight, lb	16

A seal is located on the high-speed flywheel shaft to separate the flywheel cavity from the transmission cavity. This seal is designed for the smallest practical diameter to reduce sealing area, surface speed, and power loss. This small shaft diameter is achieved by mounting the flywheel rotor bearings within the sealed flywheel housing and having only a torque-carrying shaft protrude through the seal. Several types of seals are available, with the most-likely candidate being a ferro-fluidic type in which an annulus of oil is maintained in position between the rotating shaft and the stationary housing by means of a mass of magnetic particles in a strong magnetic field. This seal would be completely gas-tight except for air dissolved in any oil that might be carried through the seal.

A second candidate seal is a carbon-faced seal against a hardened seal runner on the flywheel shaft. In this design, the seal is well lubricated, sealed, and cooled by oil on both sides of the seal. Additional seals are located on the motor, generator, and axle shafts. These seals are for lower speeds and are not as critical in regard to leakage rates. Seal requirements are summarized in Table 4-3.

TABLE 4-3
SUMMARY OF SHAFT SEALS, ELECTRIC CAR

Location	Nominal Shaft Diameter, in.	Seal Ring Diameter, in.	Maximum Shaft Speed, rpm	Maximum Seal Surface Speed, ft/min	Seal ΔP , psi	Allowable Leakage Rate, scfm
Flywheel rotor shaft	0.4	0.65	25,000	4254	2	0.001
Generator shaft	0.75	1.00	12,000	3142	13	0.05
Motor shaft	0.75	1.00	12,000	3142	13	0.05
Axle shaft	1.00	1.30	840	286	13	0.02 (each)

The reduced air pressure is maintained in the flywheel housing by means of a two-stage vacuum pumping arrangement -- a molecular pump first stage, which maintains the housing pressure at less than 50 microns, and a second-stage internal gear pump, which maintains a low pressure at the discharge of the molecular pump. The molecular pump is essentially a disk on each end of the flywheel rotor in close proximity with the housing. Spiral grooves in the rotor and housing, under the speed of rotation, permits the air molecules to migrate inward toward the flywheel shaft centerline. To accomplish this, the pressure must be low enough to allow a sufficiently long molecular free path in which the molecules bounce directly back and forth across the gap between the disk on the flywheel and the wall of the housing. One of these pump disks is located at each end of the flywheel rotor.

In designing the size (or capacity) of the flywheel, the ideal goal of eliminating all peak loads from the battery was emphasized. The peak loads occur during acceleration, maximum speed passing maneuvers, and hill climbing. The hill climbing requirement is the most severe. If only acceleration to 55 mph (or 70 mph for passing) were required, the flywheel could be sized for an available capacity of about 250 w-hr. However, if the flywheel is to furnish the power to maintain 50 mph up a 5 percent grade for a distance of one mile, the required available capacity is about 650 w-hr, for a total capacity of about 900 w-hr. Figure 4-3 shows the total capacity as a function of distance up a 5 percent grade. The baseline system selected includes a total capacity of 1-kwh, which is sufficient to accelerate the car to 50 mph and then go up a 5 percent grade for a total distance of 1.2 miles before having to rely on the battery only. Figure 4-4 shows the distance and speed capability for a 5 percent grade.

The rotational speed of the flywheel was determined by the maximum diameter that could be installed and by the recommended gear speeds in the planetary differential. The volume requirements for a flywheel of 1-kwh total capacity are shown in Figure 4-5 in terms of diameter and axial length. The rotational speeds for the same range of diameters are shown in Figure 4-6. A 23-in.-dia flywheel operating at a maximum speed of 25,000 rpm was selected as a compromise between installation volume and reasonable planetary gear speeds. Smaller-diameter flywheels would have less windage losses, but the higher rotational speeds would have higher bearing seals and planetary gear losses.

TRANSMISSION/GEARBOX

Various alternate versions of the planetary differential gearbox have been previously analyzed. Analysis has shown that the primary variable in the transmission is the speed ratios of the three shafts. As long as the gear arrangement will accommodate the most desirable speed ratios of the flywheel, generator, and motor, then no further improvement can be made. Although additional speed changing devices would enhance the overall operational capability of the system (i.e., would extend the speed ranges of the two motors, which have electrical commutation limitations that restrict their speed ratios), these are not considered for the electric vehicle design because of the added weight, cost, and complexity of the gear changing mechanisms.

S-8915

Figure 4-3. Total Flywheel Capacity for a 5-Percent Grade

S-11584

Figure 4-4. Distance and Speed Capability for a 5-Percent Grade

Figure 4-5. Volume Requirements for 1-kwh Flywheel

S-8913

Figure 4-6. Rotational Speed for Different Flywheel Diameter

The function of the transmission is to mechanically connect the various drive elements (axle, motor, generator, and flywheel) and provide the speed ratios required of the drive. The principal speed ratio problem (and the maximum power transmission) occurs between the axle and the flywheel. These two elements are connected by means of a split power train differential planetary gear set, a concept that provides an infinitely variable gear ratio between the axle from standstill to maximum speed.

The sun gear of the planetary set is mounted on the flywheel shaft and rotates at flywheel speed. The planet gear carrier (planetary) shaft is coupled to the drive axle(s) through a fixed ratio drive mechanism (gear, chain, or belt). The electric motor also is coupled to the planetary shaft through a fixed ratio drive. The ring gear is attached to the generator through a fixed ratio drive, but the independently controlled generator speed provides for an infinitely variable ratio within the design limits of the ring gear speed, and consequently of the differential planetary drive ratio. The speed and torque relationships of the various elements in the planetary gearset are described in Tables 4-4 and 4-5, respectively. At present, the gear ratio (term a of Tables 4-4 and 4-5) is 4.25, and will be finalized in the detail design of future phases.

The entire transmission assembly is enclosed in an aluminum housing that provides the interconnecting and mounting structure for the various power train elements. The motor, generator, and flywheel are attached by bolted flanges to the transmission case. Other minor elements, such as the lube/vacuum pump assembly, oil filter, etc., also are mounted on the transmission housing. The fixed ratio drive connections between the various elements can be accomplished in a number of ways: gears, multiple strand roller chain, silent chain, or toothed belts. The high speeds involved with the motor and generator drives suggest either gears or toothed belts. The drive between the planetary shaft and the axle (differential ring gear) is a much higher-torque, lower-speed drive; thus, a chain drive is most suitable.

All rotating parts are mounted on antifriction rolling element bearings to reduce the losses. The bearings and gears are lubricated by means of jets from a pressurized oil system. The oil system contains the oil pump, filter, cooler, and necessary valves. The gearbox is of the dry-sump type to reduce oil churning and associated losses. The oil will be pumped to an oil tank (which may be integral with the transmission housing) that is vented to atmospheric pressure.

DC MOTOR/GENERATOR

The preliminary design of the motor/generator is shown in cross-section in Drawing 519890-1. Both motor/generators are considered to have effectively the same requirements and to operate over the same speed range. Therefore, only one design is discussed. Subsequent evaluation may indicate sufficient system benefit to use two different-size machines to best accomplish the specific functions of each. While both units operate in the motor and generator mode at times during the duty cycle, for simplicity the machine is referred to as the motor throughout the following discussion.

TABLE 4-4

PLANET GEAR SPEED RELATIONSHIP

 N_s = Sun gear rpm N_c = Planetary shaft rpm N_r = Ring gear rpm $a = \frac{\text{planet gear diameter}}{\text{sun gear diameter}} = \frac{\text{planet teeth}}{\text{sun teeth}}$ N_p = Planet gear rpm (relative to its axis) N_p' = Planet gear rpm (absolute)

$$(1) \quad N_r = \left(\frac{2a+2}{2a+1} \right) N_c - \frac{N_s}{2a+1}$$

$$(2) \quad N_c = \left(\frac{2a+1}{2a+2} \right) N_r + \frac{N_s}{2a+2}$$

$$(3) \quad N_s = (2a+2) N_c = (2a+1) N_r$$

$$(4) \quad N_p = \frac{N_r (1+2a) - N_s - 2aN_c}{2a}$$

$$(5) \quad N_p' = \left(\frac{a+1}{a} \right) N_c - \frac{N_s}{a}$$

TABLE 4-5

PLANET GEAR TORQUE RELATIONSHIPS

 T_s = Sun gear torque T_c = Cage torque T_r = Ring gear torque $a = \frac{\text{planet PD}}{\text{sun PD}} = \frac{\text{planet teeth}}{\text{sun teeth}}$

$$(1) \quad T_c = 2T_r \left(\frac{a+1}{2a+1} \right) = T_r \left(\frac{2a+2}{2a+1} \right)$$

$$(2) \quad T_c = T_s (2a+2)$$

$$(3) \quad T_s = \frac{T_r}{2a+1}$$

$$(4) \quad T_s = \frac{T_c}{2a+2}$$

$$(5) \quad T_r = \frac{T_c}{2} \left(\frac{2a+1}{a+1} \right) = T_c \left(\frac{2a+1}{2a+2} \right)$$

$$(6) \quad T_r = (2a+1)T_s$$

The preliminary motor design described herein is intended for use in the first generation development units to ensure high efficiency and low weight, with less emphasis on cost and producibility. As operating data and experience are gained during the development program, it is intended that design improvements will be incorporated that will result in a more optimum design from the standpoint of both weight and cost. These changes will include materials changes (such as the armature laminations or brush material) and overall size and winding changes (resulting from pole face/interpole tradeoffs). The preliminary design provides a starting point for a high output-to-weight motor/generator with high efficiency.

The motor design is based on the typical considerations for a traction motor application. There are, however, some variations from conventional motors of this rating. These variations were made within the guideline of minimizing weight without lowering motor efficiency. There are three main differences in approach, as listed below:

- (a) Aluminum castings are used instead of steel for end bells to directly save weight.
- (b) Vanadium-permendum laminations with their higher magnetic saturation level are used in the armature. This provides for superior specific performance (kw/lb) without requiring excessive field excitation, with its resultant high losses and reduced efficiency.
- (c) Pole-face windings are used to improve control of armature reaction magnetomotive force (mmf), thus permitting operation at smaller main field air gaps. This results in a net improvement in efficiency by reducing the main field losses.

The significant dimensions, weights, and basic design parameters are summarized in Table 4-6. To predict motor capability, a performance prediction program was run on a digital computer using the maximum control field excitation (1800 amp-turns per pole) as input. The speed, armature current, power output, and efficiency are plotted as a function of shaft torque in Figure 4-7. The motor design and assembly are described below under field assembly, armature assembly, commutation, and final assembly.

Field Assembly

The field, or stator, assembly is built within the main magnetic frame, a cylinder of AISI 1010 steel. Being a four-pole machine, the motor has four main (or control) field poles located at 90-deg spacing within this ring. These field poles comprise laminated silicon steel riveted into a rigid body. The control field winding is wound around the neck of each field pole, and the pole face winding is placed in two slots in the pole close to the air gap. The actual laminated pole comprises two sections: the main section (described above) and a section that is placed between the two pole-face winding coils after they are inserted. This second section has a slot in its lamination. A bar with three tapped holes is inserted in this slot and the ends swedged to effectively rivet the stack of laminations together. Screws passing through clearance holes in the stator frame and main pole thread into the holes in the

TABLE 4-6
DC MOTOR/GENERATOR CHARACTERISTICS

Rating

20.0 kw, 106 vdc, 200 amp
3200 to 10,000 rpm at maximum power, 12,000 rpm maximum
4 poles

Dimensions, in.

Housing O.D.	8.90
Housing I.D.	7.86
Field pole I.D.	5.12
Stack length	5.25
Field coil end extension	1.75
Armature O.D.	5.00
Armature slot I.D.	3.71
Slot width	0.28
Armature coil end extension	1.91
Commutator O.D.	4.25
Commutator length	2.52
Brush width	0.29
Brush length	2.00

Electromagnetic Weight Summary, lb

Field assembly

Shunt field iron	10.93
Shunt field copper	6.96
Pole face winding copper	4.03
Interpole iron	6.18
Interpole copper	4.08
Housing	21.78

Armature assembly

Armature iron	12.89
Armature copper	7.45
Commutator	11.79
Brushes	0.20

Total EM weight 86.29

Total Weight, lb

Support hardware

End bells	6.89
Brush rigging	3.28
Shaft	5.03
Bearings	1.04
Miscellaneous hardware	3.00

Total unit weight 105.53

TABLE 4-6 (Continued).

Circuit Resistance at 20°C, ohm

Armature winding	0.0238
Shunt (control) winding	0.5767
Interpole winding	0.0071
Pole face winding	0.0093

Circuit Inductances, mh

Armature winding	0.1244
Shunt (control) winding	14.2 to 71.1
Interpole winding	0.1909
Pole face winding	0.0469

Winding Descriptions

Armature: 75 bar commutator, 25 slot, 6 conductors/slot, wave winding, rectangular wire of 0.0138 sq in.

Shunt field: 60 turns No. 13 AWG square wire

Interpole: 6 turns rectangular wire, 0.0350 sq in.

Pole face: 5 turns rectangular wire, 0.0304 sq in.

Materials

Housing	AISI 1010 steel
Wire	Annealed copper, Kapton wrapped
Shunt poles	Silicon steel
Armature laminations	Permendur
Interpoles	Silicon steel
Commutator	Silver bearing copper
Brushes	Electrographitic
End bells	Aluminum

Electrical Loading, amp

Maximum armature current	200.0
Maximum shunt field current	30.0
Conductor/in., maximum	954.9
RMS armature current	90.3
RMS shunt field current	15.8

Current Density, amp/sq in. (at RMS loads)

Armature winding	3272
Shunt (control) field winding	3123
Interpole winding	2580
Pole face winding	2970
Brushes	78

TABLE 4-6 (Continued)

Flux Density (at Maximum Field Excitation), kgauss

Armature teeth	23.15
Armature core	9.93
Main pole	19.08
Housing	14.06

Electrical Losses (at rms Loads and 68°F), watt

Armature iron	112.9
Armature copper	180.6
Interpole copper	57.9
Pole face copper	75.7
Shunt field copper	144.5
Brush electrical drop	45.3
Brush friction	190.6

Commutation

Commutating zone/neutral zone	0.650
Commutating coil voltage	4.283
Brush width/bar pitch	1.625

Figure 4-7. Motor Performance at Maximum Field Excitation (106 vdc)

bar, thus clamping the entire pole assembly snug against the inner diameter of the frame. The windings are wound on fixtures and formed to fit the allotted space. The magnet wire is annealed copper with Kapton tape insulation. Additional insulation is provided for the full coils by external wrapping with glass tape and lapped Kapton tape. In addition, the pole face slots are insulated with a nomex slot liner. Interpoles that aid the motor commutation by minimizing the magnetic flux in the commutating zone are located half-way between the main field poles. As in the main poles, the interfaces are riveted assemblies of silicon steel laminations. The interpole winding uses the same basic insulation system as described for the main pole windings. The design provides for complete fabrication of the interpole and winding outside the stator, and then inserting the poles and securing them in place with screws. With its ease of pole attachment and removal, this technique facilitates shimming of the interpole for proper adjustment of the armature reaction correction.

The main (or control) field winding is separately excited, with the actual number of ampere-turns dependent on the input from the controller. The four control field windings (one on each pole) are connected in series, and all interpole and pole face windings are connected in series; these are subsequently connected in series with the armature circuit.

Armature Assembly

The armature assembly comprises a stack of Vanadium permendur laminations mounted on the shaft. The laminations have 25 rectangular slots for insertion of the armature winding. The armature is wound with a retrogressive wave winding,

with six conductors per slot, each conductor being one-half of an armature coil. The coil is formed in the "hairpin" configuration. A digital computer program calculated the precise dimensions of the coil after considering acceptable bend radii for the specific wire geometry. Each armature coil is formed to shape individually, and is then assembled into groups to complete a slot group. For insulation, the annealed copper magnet wire is wrapped with Kapton tape. A nomex separator is located between the top three and bottom three conductors, and the total assembly is wrapped with glass tape and Kapton tape. The armature slot has a nomex slot liner, and after the coil group is inserted in the slot, a fiberglass slot wedge is driven in place to secure the windings. The winding end turns, wire extensions beyond the stack itself, are then taped with fiberglass to provide for full support under the centrifugal loads encountered in operation. The open ends of the hairpin windings are welded to appropriate commutator bar risers, and the entire assembly is vacuum-varnish impregnated. The final step is to undercut and diamond-turn the commutator to clean and tune up the brush running surface. The armature laminations are keyed to the shaft, as is the commutator. In addition to providing a positive drive, these keys index the armature slot to commutator bar position.

Commutation

As part of the armature, the commutator is a 75-bar configuration that uses silver bearing copper and bonded mica insulation. Mica extensions are used to maximize creep distances, which are paths between bars or from the commutator bars (which are electrically energized) to the shaft or other potential ground surfaces. Electrographitic material is used for the brushes to ensure long life. The brushes are sized so that the current density at maximum load is within brush design limits. The brush rigging (or support hardware) includes brush holders, springs, spring retainers, and brush holder base, which allows proper interconnection of the brushes and insulation from the outer housing. The brushes are placed in the brush holders through openings in the rear motor bell. The brush shunts are connected to the proper terminals, and the springs and spring retainers mounted. A sheet metal closure then covers the end bell openings. The entire brush rigging assembly has an angular adjustment to permit location of the brushes in a true neutral commutation position. The springs are designed to provide 4-psi brush pressure on the commutator. The brushes are split to maintain better brush-commutator contact.

Final Assembly

The final stage of motor fabrication consists of pressing the front (or drive end) bearing onto the shaft and clamping and locking the back (or commutator) end bearing on the shaft. The armature is then placed in the field assembly, the end bells registered on the field frame, and the entire structure bolted together. The outer race of the back bearing is then clamped. This secures the commutator-brush position. The front bearing is designed to float, or take up any difference in axial thermal expansion, between the outer structure (field assembly and end bells) and the armature. The brushes are loaded and the shroud put in place. The motor is then operated at low speed and current until the brushes are seated. Any final trimming to achieve the true brush neutral is done after brush seating, and the motor is ready for test.

In service, the motor is air cooled by air ducted from a separate blower. The cooling air enters at the commutator end, proceeds through the air gap and the spaces between the field windings, and then exits through ventilation holes in the forward end bell. A flow of 50 cfm is provided.

CONTROLLER

The control system is an adaptation of the solid-state field control system developed for use in the DOT advanced concept train (ACT-1) program. The basic schematic is shown in Figure 4-8. In this system, separately excited motors are used, and the two armatures are connected directly. The controller adjusts the two field excitations to determine the armature current and direction in response to the command signal from the throttle or brake.

S-11500

Figure 4-8. Control System Schematic

All scheduling of control variables is based on feedback of system state or output variables independent of road profile or vehicle initial conditions. The control system comprises the following:

- (a) Field supply controllers (2)
- (b) Battery supply controller
- (c) Microprocessor
- (d) Control interfaces
- (e) Sensors and instrumentation
- (f) Power supplies

These components are described in detail below.

The two field supply controllers function to determine the output torque of the power system. The battery supply controller is only used when the motor(s) are below base speed, and is normally bypassed. The microprocessor is the central processor for the data from the command signals and the sensors, and determines the proper armature voltage and the battery input current into the armature circuit. The control interfaces are links between the processor and all of the control or command variables. Speed and position sensors are used to supply power system data to the processor and data display for the driver (or recorder). Power supplies are provided to convert the line voltage to that required for the various components of the control system (such as the microprocessor, the analog-to-digital converters, and the sensors).

The flywheel generator and traction motor armature field controls are implemented as pulse-width-modulated (PWM) current regulators to provide high efficiency. With this technique, a pulse train with a period much smaller than the electrical time constant of the field winding is applied to the motors. The pulse amplitude is constant and approximately equal to the battery voltage. The pulse width is increased or decreased as more or less current is commanded. With this type of controller, all power transistors are switched fully off (dissipating little power), accounting for the high efficiency.

A simplified schematic of the electronics for the field control is shown in Figure 4-9, with $V_{C1} - V_{C4}$ on/off PWM control signals and V_f analog feedback signal proportional to instantaneous field current. The feedback signal is compared to an input command signal to provide closed-loop control of the field current by adjustment of the $V_{C1} - V_{C4}$ switching on times. The bridge is biased such that transistors Q3 and Q4 are on and Q1 and Q2 are off for zero input. An input pulse from V_{C1} and V_{C3} will invert Q1 and Q3, allowing current to flow from the battery through Q1, the field winding, Q4, and R SENSE (+). When V_{C1} turns off, Q1 switches off and energy stored in the field inductance forces the current to continue flowing through Q4 and CR3. This "free-wheeling" path results in a continuous, smooth motor current. To reverse the current direction shown, control is through V_{C2} .

Figure 4-9. Field Supply Controller

The amount of power dissipation in the semiconductor components ranges from approximately 10 to 35 watts, necessitating an adequate heat sink and/or heat removal techniques. Typical size and weight characteristics for a production field controller (including all PWM controls) are 30 cu in. and 2.0 lb (if adequate heat removal capability is supplied).

Battery Supply Controller

The battery supply controller is shown in Figure 4-10. Current regulation occurs with voltage chopping by the chopper switch under pulsewidth control. The input LC filter allows current smoothing of the battery. The chopper switch must handle currents up to 100 amp at a voltage of 108 v. The choice of switching elements is related to cost, size, efficiency of the switches, and switch impact on the control and filtering circuitry. A load sharing configuration using lower current, high voltage transistors is shown in Figure 4-11. Peak current through each of the N transistors is approximately $1/N$, with sharing guaranteed by overdriving the transistors through the base resistors rather than by less efficient emitter ballast resistances. Three or four inexpensive Darlington transistors designed for 25 amp at 400 v can therefore be used, which result in a low drive requirement and high reliability at a reasonable cost.

The power dissipation in the switches at 100 amp is estimated as 208 watts. At this power level, an approximate package size is 200 cu in. and a 4-lb weight.

S-11454

Figure 4-10. Battery Supply Controller

S-11472

Figure 4-11. Transistor Driver

Microprocessor

A microprocessor is employed for control because the control function can be expanded by additional program memory and the system can be adapted to other control functions by reprogramming. For control system use, a 12-bit microprocessor is generally required to meet accuracy and control requirements. The 12-bit tri-state output and input buses provide the interfaces between the input, feedback, and output control variables and the processor. The input and feedback variables are sampled, and appropriate commands are issued as control variables. A functional diagram of this process is shown in Figure 4-12, which includes the processor, signal conditioners, and control interfaces.

The functions implemented in the processor will be linear and nonlinear function generation, schedule generation, discrete control generation (on/off functions), and some filtering functions. Aside from the filter functions, the remaining generators are implemented as table look-up procedures.

For prototype units, an AiResearch RF- μ 16 processor will be used for the controller functions. The RF- μ 16 is a third-generation central processing unit design for general-purpose, dedicated-processor applications. Although the processor may have more capability than is required for this application, the additional processing capacity would be an asset in simplifying the development of the control algorithms. In production units, the processor requirements will be well-defined and the choice of microprocessor will be reevaluated.

Evaluation of the mode controls and various flywheel control schemes will be simulated by a computer model to include vehicle controls and various load profiles. The computer model (shown in Figure 4-13) is an example of the analytical aid that will be used in detail design and evaluation of the control algorithms.

Control Interfaces

The function of the processor is to examine the instantaneous state of the vehicle and issue appropriate commands to the control interfaces in response to operator input commands. The interfaces form the buffers between the processor and the control variables. Figure 4-12 shows a functional block diagram of the proposed control interfaces to I_{FG} (generator field current), I_{FM} (motor field current), I_{C110P} (battery/current), and the friction brake control interface. The three current controls are shown as external minor-loop current regulators commanded through D/A converters from the processor. This allows any dynamic loop compensation $G(S)$ to be implemented with analog circuits and frees the processor from this function. The PWM control includes the modulating capability as a function of the feedback error signal and the functional controls for forward/reverse current discrimination. Implementation with either discrete (on amp) or monolithic regulating circuits is feasible for the PWM control elements.

The friction brake interface control blends the friction brake system operation with the electrical regenerative braking system. Techniques of blending the operation of the two systems to allow maximum regeneration of energy and maintain safe operating conditions will be considered during development of the drive control functions.

S-11455

Figure 4-12. Control Interface and Sensor Conditioners

S-11471

Figure 4-13. Vehicle Computer Model

Sensors and Instrumentation

The control functions require feedback sensing of battery current, traction motor armature current, flywheel generator armature current, traction motor speed, and flywheel speed. Current sensing requires conditioning of the voltage drop across an in-line sense resistor (typically four-terminals for low inductance and low resistance at high currents). The armature currents must be converted to digital signals for use by the processor; hence, an A/D converter (12 bits) is required for this conditioning. A single A/D converter can be employed for both lines, with the appropriate current sense signal multiplexed to the A/D as required by the control mode.

Traction motor speed and flywheel speed will be sensed by variable reluctance monopoies, providing a frequency output proportional to speed. This frequency is converted to a digital signal by counting external clock pulses between cycles of the monopole output. The effect of the threshold around zero speed and the requirement for sensing bidirectional rotation will be required as part of the control function analysis.

Additional input sensors include the throttle sensor, brake sensor, ignition (start) sensor, and shift (park, forward, reverse) sensor. The accelerator (throttle) and brake sensors can be either linear potentiometers connected to the accelerator or brake pedal with an A/D converter providing the conditioning for the processor, or digitally encoded LVDT's that allow direct interface to the processor. The choice of sensor type will depend on availability of the digital type (the analog sensor has been widely used in automotive programs) and on reliability and cost tradeoffs. The direct interface to the processor is attractive because no A/D converter is required. The ignition (start) and shift sensors are simple discrete switches that will interface directly with the processor.

Instrumentation for operator use also requires some sensor and conditioning hardware. Typical dashboard displays for prototype units will include vehicle speed, flywheel capacity (percent maximum rpm), battery current, battery voltage, field currents, external charging indicator, and recharging (regeneration) indicator. No processor capability is required for these displays in the prototype unit because simple analog interfaces to meters and/or lights are sufficient. In addition, a battery charge indicator (similar in function to the fuel gage) will be included.

Power Supplies, Grounding, Noise

The power supply and power distribution array are often overlooked in the design of a system, but their consideration as part of the overall system operation is essential, particularly in a system such as this that mixes high and low power analog and digital (switching) circuitry. A proper grounding sequence, including separate analog and digital grounds, eliminates an assortment of noise and EMI problems in the digital circuitry (especially A/D conversions). Shielding of some signals will be considered for low-voltage signals such as sense lines. Power supplies for the analog and digital control components can operate efficiently from the 108 v line if a switching regulator is used. Generation of positive and negative supply voltages will be required.

BATTERY PACK

The concept of the flywheel hybrid power system is based on the premise that regenerative braking in conjunction with load leveling will result in a sufficient improvement in vehicle range and performance to achieve the near-term goals of the program. With the possible exception of cycle life and ownership cost, this is true for all of the batteries being considered, even the near-term lead-acid battery. For this reason, it is recommended that the baseline design be based on the use of the near-term, lead-acid battery so that the development and demonstration program can be initiated as soon as possible. Later improvements can only improve the weight and life-cycle cost aspects.

The vehicle battery goals established by the Argonne National Laboratory are given in Tables 4-7 through 4-10, which show both the (improved) near-term goals and the advanced versions. Figures 4-14 through 4-16 show data from one manufacturer, Eagle Picher, for both current, state-of-the-art and improvements expected by 1980.

From the design studies of the hybrid power system and the vehicle gross weight variations, it is possible to determine the required battery weights as a function of energy density of the battery. Figure 4-17 shows the required weights of the various batteries for the 75-mile range.

For the lead-acid battery, the data from Table 4-7 and Figure 4-14 have been used to establish the battery ratings for the baseline system. These ratings are:

Specific energy (C3 rate)	40 w-hr/kg
Specific power (sustained)	15 w/kg
Cycle life (80 percent discharge)	800 cycles
Cost	\$50/kwh

The cycle life of the batteries is based on the conventional traction service duty cycle. However, for the hybrid system (where the load is to be nearly constant at the C3 rate), the cycle life of a given battery should be significantly better. The battery manufacturers generally agree, but do not have any data to quantify this premise.

The voltage of the battery pack will be determined by the number of 2-v cells. At the present time, most of the high-energy traction batteries available are 6-v golf cart batteries of a standard size that weigh about 65 lb. If the design is to be based on these standard-size batteries, then the total battery weight will determine the number of batteries, and consequently the voltage. For the baseline vehicle system, the requirement for 983 lb of batteries would dictate the use of 15 or 16 standard batteries, for a maximum voltage of either 90 or 96 v. Generally, the dc traction motors will be more efficient at higher voltages because of the brush voltage drop. In the hybrid system, the armature circuit includes two sets of brushes that are in series during transient

TABLE 4-7

PRELIMINARY GOALS FOR LEAD-ACID ELECTRIC VEHICLE BATTERY
(Argonne National Laboratory)

	Improved State-of-the-Art	Advanced
Fiscal year	1978	1980
Duty cycle	2 to 4 hr discharge 4 to 8 hr charge	
Energy efficiency (round trip)	> 60 percent	
Cycle life*	800 cycles	1000 cycles
Installed cost	\$50/kwh	\$40/kwh
Specific energy	40 w-hr/kg (C3 rate)	60 w-hr/kg (C3 rate)
Specific power		
Sustaining	20 w/kg	30 w/kg
Peak	100 w/kg	150 w/kg
Typical installation size	20 to 30 kwh	
Environmental impact	Minimal	

*Corresponding to 70 to 80 percent depth of discharge

TABLE 4-8

PRELIMINARY GOALS FOR NICKEL-IRON OXIDE ELECTRIC VEHICLE BATTERY
(Argonne National Laboratory)

	Improved State-of-the-Art	Advanced
Fiscal year	1977	1979
Duty cycle	2 to 4 hr discharge 4 to 8 hr charge	
Energy efficiency (round trip)	> 60 percent	
Cycle life*	1500 cycles	1500 cycles
Installed cost	\$70/kwh	\$50/kwh
Specific energy	50 w-hr/kg (C3 rate)	60 w-hr/kg (C3 rate)
Specific power		
Sustaining	40 w/kg	50 w/kg
Peak	150 w/kg	250 w/kg
Typical installation size	20 to 30 kwh	
Environmental impact	Minimal	

*Corresponding to 70 to 80 percent depth of discharge

TABLE 4-9

PRELIMINARY GOALS FOR ZINC/NICKEL OXIDE ELECTRIC VEHICLE BATTERY
(Argonne National Laboratory)

Duty cycle	2 to 4 hr discharge 4 to 8 hr charge
Energy efficiency (round trip)	>60 percent
Cycle life	1000 cycles
Installed cost	50/kwh
Specific energy	110 w-hr/kg (C5 rate)
Sustaining	50 w/kg
Peak	200 w/kg
Typical installation size	20 to 30 kwh
Environmental impact	Minimal

TABLE 4-10

PROGRAM GOALS FOR THE LITHIUM/IRON SULFIDE ELECTRIC VEHICLE BATTERY
(Argonne National Laboratory)

	Mark I 1977	Mark II 1979	Mark III 1981	Potential for Future
Specific energy, w-hr/kg				
Cell	100	125	160	180 to 200
Battery	75	100	130	145 to 160
Peak power, w/kg				
Cell	100	150	200	240
Battery	75	120	160	200
Lifetime				
Deep discharges	400	700	1,000	>1,000
Automobile cycles	1,200	2,100	3,000	>3,000
Equivalent miles	40,000	85,000	150,000	>250,000

Figure 4-14. Lead-Acid Electric Vehicle Batteries, Power Density Versus Energy Density

Figure 4-15. Nickel-Iron Electric Vehicle Batteries, Power Density Versus Energy Density

Figure 4-16. Nickel-Zinc Electric Vehicle Batteries, Power Density Versus Energy Density

Figure 4-17. Battery Weight as a Function of Specific Energy for a Range of 75 Miles in City Driving

performance and parallel during cruise. It would therefore be desirable to have smaller, 6-v batteries so that the voltage could be at least 108 v, and possibly higher.

This problem has been discussed with the battery manufacturers, who feel that as long as the battery proportions do not change much from the present models, there is no reason for not having the more ideal size, except for the tooling that would be required to make them. Therefore, for the purposes of this design study, 16 standard-size golf cart batteries have been chosen as the preliminary model for use in the vehicle layouts, assuming that smaller batteries could be made available later. The electrical system will therefore be designed for 108 v for the higher efficiency.

The major problem with the present battery technology is the effect on total vehicle costs. The initial battery costs are a penalty, and the heavy weight that is required adds to the cost of the vehicle chassis, either in additional size and weight to carry the batteries or in additional sophistication in the attempt to minimize the penalty of additional weight. Furthermore, the cycle life of the batteries is such that frequent replacement is required. Some of the advanced batteries have promise for increasing the cycle life, but only at the expense of increased cost. A certain amount of increased cost may be justified, but only if the performance or cycle life is sufficiently improved. The replacement cost of batteries is shown in Figure 4-18 in terms of cost per mile as a function of battery cost, performance, and cycle life.

Figure 4-18. Battery Replacement Cost as a Function of Battery Cost, Performance, and Cycle Life

SECTION 5

SYSTEM PERFORMANCE

INTRODUCTION

This section presents the performance capabilities predicted for the baseline electric-powered passenger vehicle. Performance predictions are based on a system simulation model and computer program developed to determine how the system operates and how component performance affects overall performance. The detailed analytical models developed for each component and the digital program used to predict system performance variations are discussed below. Performance predictions are presented at the end of this section for the flywheel, battery, and representative driving cycles and modes.

SIMULATION MODELS AND PROGRAM DESCRIPTION

The flywheel propulsion system simulation equates transient changes in flywheel-generator-motor kinetic energy to changes in vehicle kinetic energy and losses due to road loads and mechanical friction. The flywheel, motor, and generator shafts are connected by a differential planetary gear set. The rate of change of flywheel kinetic energy is controlled by the torque applied at the ring gear by the generator. The system losses are computed on a component basis as a function of speed, torque, current, or other appropriate parameters so that realistic system performance can be predicted.

Three system models were developed to provide the required flexibility in simulating system operation over a given driving condition: (1) ACCEL, (2) SUBURB, and (3) UPHILL. ACCEL simulates maximum system performance during full throttle acceleration to a given maximum speed followed by full power regenerative braking. For this mode, the armature current is fixed at the design maximum, the power output of the generator increases to respond to the increasing speed (and power) of the motor, and the motor and generator field excitations are adjusted by the controller to maintain a constant armature voltage. Upon reaching the prescribed maximum vehicle speed, electrical braking is initiated. The controller reduces the generator field and increases the motor field sufficiently to cause a complete reversal of the armature current. This maximum braking operation is continued until the motor (and vehicle) speed is reduced to an armature voltage value that drops below a minimum for significant braking effect, which occurs at about 5 to 6 mph. Complete system and component performance is calculated for each time interval (usually 1 sec) throughout the entire operation, and includes all component speeds and accelerations, torque forces, and system losses. In addition, the computer provides plots of some of the major variables during acceleration and deceleration to facilitate comparison and evaluation of predicted performance.

Simulation program ACCEL is used to study maximum system power capability as a function of the component sizes and efficiencies. The output shows the effect of the gear ratios used; the relation between the selected generator, flywheel, and motor speeds; and the resulting vehicle acceleration and deceleration performance. Also shown are the overall efficiency and the split between mechanically and electrically transmitted power.

The second program, SUBURB, simulates vehicle operation over any arbitrary driving cycle comprising any number of specified segments. It is presently set up to simulate the five segments of the SAE J227A(d) suburban driving cycle. The power requirement is determined by the vehicle speed and acceleration prescribed in the driving cycle. The battery is integrated into the power system armature circuit by means of a current schedule that is keyed to the flywheel speed. The program logic solves for the generator armature current, which results from applying the correct generator torque at the planetary. The generator torque, transferred mechanically through the planetary, plus the motor torque resulting from the sum of the battery and generator currents, must then equal the required torque. During idle, the motor at the ring gear is driven by the power out of the battery while the carrier shaft motor unit is turned off.

The cumulative battery energy delivered to the power system and both the vehicle and flywheel kinetic energy are summarized at the end of each segment. Energy consumption is itemized for each component in the system over each of the successive segments of the cycle. The total of all segments is the energy requirement that must be supplied by the battery for the entire cycle. Simulation program SUBURB is used to (1) compute the range of the vehicle from the total summation of the energy requirements for the cycle, (2) examine the individual losses in the system, and (3) show the sensitivity of the resultant range to each of the system component losses.

The third program simulation, UPHILL, simulates the operation of climbing up a grade at a given speed. This program logic is the same as SUBURB. In use, three segments are simulated: (1) level ground acceleration to a given speed, (2) a short transition to the specified grade, and (3) vehicle steady-state speed up the grade powered by the energy from both the flywheel and the generator, and is terminated when the generator no longer maintains a sufficient voltage to overcome the system losses. This occurs at about 50 percent flywheel speed, signifying that about 75 percent of the total energy stored in the flywheel has been extracted. The peak power out of the battery is less than 10 kw, supplying less than 25 percent of the energy required for the hill climb.

SIMULATION MODEL INPUTS

The model input array permits the user to describe the vehicle weight and road resistance, and to specify the characteristics of the propulsion system components. The inertial properties of the vehicle are described by gross weight, frontal area, drag coefficient, and tire-wheel bearing performance. The basic propulsion system configuration is described by the energy capacity of the flywheel and the power of the motor-generator. The flywheel rotational moment of inertia and maximum speed fix the energy capacity. A detailed description of the size and electromagnetic properties of both the motor and the generator is used in the program to calculate performance as a function of armature and field current, armature speed, and torque.

The power developed by the motor or generator is the product of the output torque times the armature speed, minus the losses due to friction and resistance heating. The output torque, in turn, is the product of the armature current and the internal voltage constant. Thus, the input description must include the saturation curve, giving the internal voltage as a function of field strength, and all motor design details that affect motor losses.

Battery efficiency is calculated as a function of the instantaneous power out of the battery and integrated over the complete cycle to determine the total energy withdrawn from the battery. The computation requires that an array describing battery specific energy as a function of specific power be supplied as input for the type of battery being considered.

The drive train is specified by the gear ratios of the final drive and the planetary differentials. The losses at the final drive and the planetary are given as a function of the speed and output torque of the rear axle and the flywheel, respectively. The losses at the motor and generator gears are also given as functions of the respective speed and torque.

Loss Models

The power dissipation for each component in the system is calculated for the most part on the basis of well-documented empirical equations. All losses are expressed in terms of watts of power dissipated.

Aerodynamic Drag

The aerodynamic drag is calculated using Newton's resistance law for form drag as follows:

$$F_{aero} = C_D A \rho V^2$$

where C_D is the drag coefficient available from test data for various compact cars and

A is the frontal area

ρ is the density of air

V is the velocity of the vehicle

Rolling Resistance

The rolling resistance calculation utilizes the empirical expression for tire and wheel bearing losses approved for use in federally funded automotive performance studies.* The expression is:

$$F_{RL} = \mu W_V [1 + (1.4 \times 10^{-3})V + (1.2 \times 10^{-5})V^2]$$

* See Prototype Vehicle Specification, Compact Passenger Car, Alternative Automotive Power Systems Division, Environmental Protection Agency, Feb. 1974.

where W_v is the gross weight of the vehicle

V is the velocity and

μ is the tire rolling resistance coefficient

The force due to aerodynamic drag and rolling resistance is applied as a torque at the rear axle. This power is transmitted through the final drive differential to the carrier shaft.

Motor and Generator Losses

The motor and generator losses comprise the difference between the power into the generator and the power out of the motor. Thus, the power into the generator minus the power out of the motor equals power loss due to:

Motor armature

Generator armature

Motor field

Generator field

The power loss analysis utilized the computer package developed by AiResearch for the design of specialized motors for the aerospace and transportation industries.

1. Motor and Generator Armature Power Losses

Power lost in the motor and generator armatures includes copper, brush, windage, and stray load losses; also included are iron losses, which comprise hysteresis and eddy losses. The following equations are used to calculate these losses.

Copper-- $P_{cu} = I^2 R$, where I is the armature current and R is the resistance of the armature, adjusted for the predicted operating temperature.

Brush Voltage Drop-- $P_{BDP} = I \times (0.25 + 0.0025 I/A_B)$, where A_B is the area of the brushes and I is the armature current.

Brush Friction-- $P_{BF} = N_B \times A_B \times P_{sp} \times C_F \times D_{COM} \times N_A / 168.9$, where N_B is the number of brushes, P_{sp} is the spring pressure, C_F is the coefficient of friction of brushes, D_{COM} is the diameter of the commutator, and N_A is the rotation speed of the armature.

Windage-- $P_w = (4.83 \times 10^{-5})(\pi D_A N_A)^2$, where D_A is the diameter of the armature, and N_A is the rotation speed of the armature.

Stray Load-- $P_{SL} = 0.01 \cdot P_s$, where P_s is the shaft power.

Hysteresis Loss-- $P_H = \left(\frac{W}{10^7 D} \right) f \eta B^X$, where W is the weight of the core, f is

the frequency, d is the thickness of core lamination, B is the flux density, D is the density of core material, X is the hysteresis loss exponent for B, and η is the hysteresis loss coefficient.

Eddy Loss-- $P_E = \left(\frac{1.645 d^2 W}{16 D} \right) f^2 B^2 \lambda$, where the terms are as defined under hysteresis loss above and λ is the eddy loss coefficient.

Thus, power loss in each of the armatures is the summation of P_{cu} , P_{BDP} , P_{BF} , P_W , P_{SL} , P_H , and P_E .

2. Motor and Generator Field Losses

The field losses are limited to copper losses as follows:

$$P_{GF} = I_{GF}^2 R_{GF}$$

$$P_{MF} = I_{MF}^2 R_{MF}$$

where I_{GF} and I_{MF} are the field currents to the generator and motor, respectively, and R_{GF} and R_{MF} are resistances of the field coils, adjusted for the predicted operating temperature.

Controller

The power loss in the controller consists of the power necessary to drive the microprocessor and the power dissipated across the choppers at the generator and motor fields.

$$P_{MP} = 58 \text{ watts}$$

$$P_{GFC} = I_{GF} V_{GC}$$

$$P_{MFC} = I_{MF} V_{MC}$$

where I_{GF} and I_{MF} are the field currents to the generator and motor, respectively, and V_{GC} and V_{MC} are the voltage drops across the field choppers.

Cooling Fans

On the basis of a preliminary thermal analysis, it is assumed that the steady-state power required to drive the two cooling fans is 42 watts.

Motor and Generator Gears

The expressions for power losses at the motor and generator gears are adapted from approximate available test data for power and windage dependent losses. Equations are

$$P_{GL} = [(3 \times 10^{-6})T_G N_G + N_G^2 \times 10^{-10}], \text{ hp}$$

$$P_{ML} = [(2 \times 10^{-6})T_M N_M + N_M^2 \times 10^{-10}], \text{ hp}$$

where T_G and T_M are the torque of the generator and motor, respectively, and N_G and N_M are the respective armature speeds

This model implies an approximate 2 percent power loss over the usable range of speed and torque.

Planetary Differential

The loss expression for the planetary differential gear set is similar to the power and windage dependent equation used for the generator and motor gears. Test data on a dry sump planetary gear set with a rotating ring gear suitable for predicting losses on the proposed set is not available. The current expression is based on an adjusted wet sump efficiency curve as follows:

$$P_{PL} = [(4 \times 10^{-6})T_F N_F + N_F^2 \times 10^{-9}], \text{ hp}$$

where

T_F is the torque of the flywheel, and

N_F is the flywheel speed

This equation implies gear loss at idling of about 0.62 hp, increasing to over 2 hp at the high-power acceleration conditions.

Final Drive Differential

The power loss model for the final drive differential is consistent with industry standards for straight bevel geared differentials of modest gear ratios. The equation is

$$P_{FD} = [(4.0 \times 10^{-6})T_A N_A + (2.7 \times 10^{-9})N_A^2], \text{ hp}$$

where

T_A is the torque at the axle, and

N_A is the speed of the axle

This equation implies an approximate 4 percent power loss over the operating spectrum.

Flywheel Losses

The full speed flywheel power losses are estimated from the test data of past AiResearch flywheel programs. The run-down time results indicate that the power losses decrease as the square of the ratio of flywheel speed to maximum flywheel speed. Thus:

$$P_{FL} = 380 \left(\frac{N_F}{N_{FMAX}} \right)^2$$

where 380 is the power loss in watts at maximum flywheel speed (see Table 5-1)

N_F is flywheel speed, and

N_{FMAX} is maximum flywheel speed

TABLE 5-1

FLYWHEEL ASSEMBLY FRICTION LOSSES (AT 100 PERCENT RPM)

Item	Loss, watts
Windage	100
Rotor bearings (2)	120
Rotor seal	80
Lube pump and 2-stage vacuum pump	80
Total	380

SYSTEM PERFORMANCE

Critical performance data for the baseline vehicle design have been determined through the use of the simulation models. Maximum power output and vehicle range for the suburban, cruising, and hill-climb driving cycles were examined in the performance study, which was separated into battery-limited and flywheel-limited operations. The typical suburban cycle utilizes only 15 percent of the available flywheel energy for load leveling. In the sustained highway cruise, the power is delivered entirely from the battery. In contrast, during full throttle acceleration to maximum speed, the flywheel delivers all of the power, and the battery is switched off. This consumes 35 percent of the flywheel total energy. Similarly, during hill climbing the battery supplies only 25 percent of the required power; thus, the range is flywheel limited.

Flywheel Performance

1. Full-Power Acceleration/Deceleration

Vehicle performance during full-throttle acceleration demonstrates the flexibility of the proposed power system. The acceleration capability shown in Figure 5-1 is adequate to meet even the toughest demands of both urban and suburban traffic patterns. The elapsed time to 55 mph is 15.1 sec, and the maximum acceleration rate is 5.47 mph/sec (8.02 ft/sec^2). The 25 to 55 mph merging maneuver requires 9.8 sec at the average acceleration rate of 3.1 mph/sec. The maximum power developed by the system at 250 amp maximum armature current is 86.2 hp at 60 mph (see Figure 5-2). The drop in power above 60 mph can be explained by examining the torque curves plotted in Figure 5-3. The generator drops below base speed as the velocity increases above 60 mph. The generator is then operating at full flux in the field coils and maintaining a constant torque output. As generator speed continues to decrease, the output power decreases. The maximum output torque at the carrier shaft is 338 ft-lb at 15.6 mph.

S-11521

Figure 5-1. Maximum Acceleration, Flywheel Powered, Full Throttle

S-11517

Figure 5-2. Power Output, Full Throttle

S-11520

Figure 5-3. Torque Characteristics at Maximum Power

Full-power regenerative braking provides strong braking power at good system efficiency. The minimum stopping distance required for speeds up to 70 mph is shown in Figure 5-4. Friction brakes are needed below 8 mph. The full power deceleration rates over the vehicle speed range are shown in Figure 5-5. The power system efficiency over a full-power deceleration from maximum speed is 89 percent. Of the 198 watt-hr of vehicle kinetic energy at 70 mph, work in the form of overcoming tire losses and aerodynamic drag comprises 26 watt-hr, while 151 watt-hr are recovered in flywheel kinetic energy.

Hill-Climb Range

The high power required for hill climbing at highway speed is delivered by the flywheel without approaching the peak power capability of the traction motor. The power map for the climb of a 5 percent grade at 50 mph (presented in Figure 5-6) shows the power split between the system components as a function of total climbing distance. The sum of the battery and generator power equals the power into the traction motor. The balance of the required torque is transferred mechanically to the carrier shaft from the flywheel. The mechanical fraction increases as the generator drops below base speed after 0.7 miles of climbing. The vehicle range capability on various grades is shown in Figure 5-7 as a function of velocity. In addition to the obvious reduction in power requirement, climbing distance also improves with decreasing velocity because a greater fraction of the total flywheel energy can be converted into potential energy.

Battery Performance

The vehicle range over typical driving cycles is determined by the total energy stored in the battery pack and the efficiency with which this energy is consumed. For the fixed battery weight of 1040 lb, the energy storage capability is limited by the specific energy of the given battery type. The graph of specific energy as a function of specific power for the advanced lead-acid tubular plate battery used in this study is shown in Figure 5-8. In addition to the limit imposed by the inherent losses in the power system, energy consumption efficiency also is limited by battery discharge efficiency. For a given duty cycle, maximum battery efficiency occurs for the power drain that most nearly maintains a constant level, and is the ideal solution for a load leveling system.

The battery controller serves as a load leveling control by limiting battery current as a function of flywheel speed and a limited time history of the preceding driving pattern. This control function is shown schematically in Figure 5-9. The lower curve in the figure is adequate for normal driving, such as the mixed, highly variable transients encountered during the Federal City Driving Cycle (LA-4). In this case, the battery current is a simple function of flywheel speed. The high acceleration rates and the high transient cruise speeds are easily accommodated by this schedule. However, the shallow slope and average power output prescribed by this curve, which has been found to be optimum for load leveling of the normal driving transients, is not adequate for all types of driving patterns. The schedule must be adjusted to sustain high power cruise speeds for distances greater than 3 or 4 miles and

S-11518

Figure 5-4. Full Powered Regenerative Braking

S-11519

Figure 5-5. Regenerative Braking, Deceleration

S-11522

Figure 5-6. Hillclimb Power Map, 5 Percent Grade at 50 mph

S-8921

Figure 5-7. Distance and Grade Capability from Surplus Flywheel Energy with Steady-State Battery Drain of 60 amp

S-11516

Figure 5-8. Lead-Acid Tubular Plate Battery Performance

S-13972

Figure 5-9. Battery Current Schedule as a Function of Flywheel Speed

to sustain other high power conditions over a longer time than the average transients. During a limited time interval, the system will detect that the flywheel energy is being depleted and that the battery make-up current schedule must be increased. The controller will respond by increasing the base schedule to a higher value.

This is somewhat better than the concept of constant total energy (vehicle plus flywheel) since a delay function is involved. Furthermore, this system has the additional capability of accommodating the case of uphill grades (potential energy transfer) that would not be included in the simple total kinetic energy case.

Suburban Driving Cycle

The SAE J227D driving cycle has been used for most of the computer studies to optimize the system and the components. The performance over this cycle is summarized in Figure 5-10. These data show the energy requirements for each segment of the cycle, the energy level stored in the flywheel and in the vehicle at the end of each segment, and the battery energy that is required to overcome the system losses. These data also can be used to show the effectiveness of the current control schedule in load-leveling the current from the battery. The data from Figure 5-10 are converted to units of power in Table 5-2 to show the actual power supplied by the battery and from the flywheel during the cycle. The battery current drain is very nearly constant, which will result in a nearly ideal battery efficiency.

TABLE 5-2
SUBURBAN CYCLE POWER SUMMARY

Driving Mode	Time Segment, sec	Flywheel		Battery		
		Δ Energy, watt-hr	Average Power, kw	Δ Energy, watt-hr	Average Power, kw	Current, amp
Acceleration	0 to 28	-85.65	-11.01	-46.83	-6.02	55.74
Cruise	28 to 78	-22.39	-1.61	-98.74	-7.11	65.83
Coast	78 to 88	43.14	15.53	-19.45	-7.00	64.81
Braking	88 to 97	33.69	13.48	-13.78	-5.51	51.02
Idle	97 to 122	33.28	4.79	-45.28	-6.52	60.37

ENERGY STORAGE SYSTEM SUMMARY IN MI-HR

TIME	VEHICLE INTERNAL ENERGY	VEHICLE KINETIC ENERGY	BATTERY ENERGY INTO P.S.	ENERGY CONSUMPTION
.0	792.19	.00	.00	47.76
28.0	706.54	84.72	46.83	121.13
78.0	684.15	84.72	145.57	22.36
88.0	727.29	38.67	165.02	18.77
97.0	760.98	.00	178.80	12.00
122.0	794.26	.00	224.08	222.02 (TOTAL)

ENERGY INPUT TO POWER SYSTEM/MILE = 235.57 WATT-HOURS

ENERGY INPUT TO BATTERY/MILE = 337.06 WATT-HOURS

ENERGY INPUT TO BATTERY CHARGER/MILE = 396.54 WATT-HOURS (85.00 PERCENT CHARGER EFFICIENCY)

RANGE = 81.2 MILES (1040.0 LBS. OF BATTERIES)

Figure 5-10. Suburban Cycle Energy Summary

The battery reaches its highest average power level during sustained cruise (28 to 78 sec) after the flywheel has supplied the peak power needed for the acceleration requirement. The flywheel begins recharging during the coast segment (78 to 88 sec), and by the end of the cycle has regained all of the flywheel energy delivered over the cycle. In this equilibrium operating range, the vehicle is capable of repeating the cycle until the battery pack is depleted. The specific energy consumption data at the bottom of Figure 5-10 permit the battery controller to be evaluated effectively. The vehicle range at 337.06 watt-hr/mile (including 3.9 miles on residual flywheel energy) is 85.1 miles. With an average velocity over the suburban cycle of 27.81 mph, the optimum battery discharge rate is 2.92 hr, corresponding to a battery efficiency of 0.702. The battery efficiency achieved with the present control schedule is 0.699, which shows how nearly the schedule approaches the ideal constant load case. The near-constant battery discharge current over the SAE J227D cycle is shown in Figure 5-11. The overall power system approaches a steady-state cruising condition during the middle portion of the cycle. By the end of the idling segment, the flywheel-controlled discharge current has decreased to its initial value indicating that the flywheel has been fully recharged.

S-13860

Figure 5-11. Power System Load Leveling J227D Driving Cycle

Federal City Driving Cycle

Computer simulation studies also were conducted for the Federal City Driving Cycle (LA-4). In contrast to the SAE J227D cycle, this pattern reflects a variety of intra-city conditions derived from actual driving records. The velocity-time distribution shown in Figure 5-12 includes segments representative of free-way, surface street, and congested urban driving. The cycle covers a total distance of 7.5 miles at an average velocity of 19.6 mph with an average of 2.4 stops per mile.

The battery discharge current computed for the LA-4 cycle is shown in Figure 5-12. The battery output adjusts to changes in the average power requirement over the successive segments of the cycle while the power demand for acceleration and braking is absorbed by the flywheel. The maximum battery power is 5.9 kw in contrast to the peak power system output of 36.4 kw. The regenerative braking capability as shown in Figure 5-5 is adequate to meet the deceleration requirements over the entire cycle. The load leveled duty cycle permits the near optimum utilization of battery energy. The minimum battery discharge rate for the 94 mile range at an average velocity of 19.6 mph is 4.80 hours, implying a maximum battery efficiency of 0.801. The achieved efficiency for this application is 0.785.

S-13859

Figure 5-12. Power System Load Leveling LA-4 Driving Cycle

VEHICLE RANGE

The range is a function of the gross weight, system efficiency (including the battery), and available battery capacity. Further, the range varies with the type of driving mode. For the studies of the baseline system, a near-term, lead-acid battery with a nominal available capacity of 18 watts-hr/lb (at the C-3 rate) has been assumed to be available in time for the first experimental cars. The range of this baseline system is shown in Figure 5-13 for city driving (using the SAE J227D and LA-4 cycles) and for constant cruising speeds. Figure 5-14 shows the energy consumption (measured at the inlet to the recharger) for the same driving conditions.

Performance also has been studied for the use of the advanced batteries with higher energy density. For these higher energy batteries, it is important to consider the advantages of reduced battery weight. The improvement in range is shown in Figure 5-15, and the reduction in energy consumption with the use of the nickel-zinc battery as compared to lead acid is shown in Figure 5-16.

BATTERY PACK LIFE

The total life cycle miles available from a given battery pack are determined by the acceleration, speeds, and distances of the average daily driving cycle. The acceleration and speeds affect the energy cost per mile, whereas the distance per daily recharge cycle affects the depth of discharge. Since the depth of discharge (distance) determines the cycle life of the battery, the total energy (total miles) varies with the daily use pattern. Figure 5-17 shows the variation in battery cycle life as a function of depth of discharge.

The curve in Figure 5-17 shows the assumed life characteristics of lead-acid batteries based on the Argonne National Laboratory 1975 near-term development goal of 800 cycles at a depth of discharge of 80 percent. Depth of discharge is defined (for the lead-acid cell) as 100 percent when the open circuit cell voltage has dropped to 1.75 volts/cell. Analysis of this battery cycle life curve shows that the total energy storage capability over the life of the battery is maximum when the battery is regularly cycled to 50 percent depth of discharge. Analysis also shows that regular cycling to 100 percent depth of discharge results in a decrease in total stored energy of 19 percent. This effect is translated into daily distance versus battery life (miles or years) in Figure 5-18.

The upper curve in Figure 5-18 is based on the LA-4 Federal City Driving Cycle as was shown in Figure 5-12. This cycle includes 2.4 stops per mile and an average speed of 20 mph in city traffic. The results for the SAE J227D cycle are shown by the middle curve. Although the J227D cycle represents a less typical city driving cycle (1 stop per mile and an average speed of 28.6 mph), the area in between the two curves can be expected to represent the normal variations in driving habits and traffic conditions.

An important conclusion can be drawn from these data. The relatively small penalty of extending the usable range of the vehicle by use of the curve between 80 and 100 percent is small and is outweighed by the advantage of the additional 20 percent in permissible range. It should be noted that the power

Figure 5-13. Vehicle Range with Lead-Acid Batteries

Figure 5-14. Vehicle Energy Consumption

Figure 5-15. Vehicle City Driving Range as a Function of Battery Weight

Figure 5-16. Vehicle Energy Consumption as a Function of Battery Weight

Figure 5-17. Battery Cycle Life vs Depth of Discharge
ANL ISOA Pb-Acid Battery

Figure 5-18. Total Battery Pack Miles vs Nominal Miles
per Charging Cycle

system without regeneration and load leveling (as represented by the bottom curve) must cut off the permissible range at approximately 80 percent discharge because the vehicle will no longer meet established power and acceleration demands of the J227D cycle.

Obviously, with the advantage of the hybrid system, battery wet life becomes more of a design and development criteria than cycle life per se.

SECTION 6

ECONOMIC ANALYSIS

INTRODUCTION

The marketability of the electric passenger car has been severely questioned in the literature, the principal obstacles being both performance and cost. The performance data presented in Section 5 of this report demonstrate the suitability of the proposed baseline vehicle design to meet ERDA marketing objectives, which includes an existing second car market that has been estimated at 1,011,000 vehicles in the Los Angeles area alone (Reference 6-1)*. Marketability then reduces to a simple question of cost, which is addressed in this section.

A number of past electric vehicle studies (for example, References 6-1 and 6-2) have been necessarily pessimistic about vehicle cost projections due to several factors. The apriori assumption has been that electric vehicles using existing batteries will never obtain general consumer acceptance due to poor performance; thus, the economic analyses have been pessimistic. Further, the traditional concluding economic comparison normally made between the mass-produced internal combustion engine (ICE) compact and the limited-production electric vehicle (EV) does nothing to establish the fundamental competitive relationship between the two technologies. These studies merit a second look.

The current trend in automotive technology indicates that the ICE/EV cost advantage must decrease. The legislative pressure brought about by the existing air quality crisis and the undesirable dependence on imported oil has forced an increase in ICE complexity. Modifications such as the catalytic converter and the lean-burn, stoichiometric, closed-loop-controlled, or stratified charge engine technologies, each offered as a potential solution, increase ICE cost by 15 to 80 percent. The manufacturing cost of the Detroit product also is highly sensitive to the cost of steel, which comprises about one-half the vehicle weight. According to the study conducted by the Council on Wage and Price Stability in December 1976, as cited in a recent article (Reference 6-3), the cost of sheet steel has increased 20 percent since the time price controls were lifted in mid-1974.

This upward trend in the cost of automotive technology can be effectively reduced through the introduction of the proper EV technology. The 1978 automobile exhaust emission standards for HC, CO, and NO_x can readily be met through the use of electric vehicles that use existing coal-fired power plants for off-peak hour recharging (Reference 6-2). The success of EV's is closely tied to the development of cost-effective, lightweight vehicle construction technology. The work reviewed later in this study indicates that such a technology is being developed by the plastics industry. Relative to steel, plastics benefit from reduced tooling costs and the further economies of scale that will accompany entrance into the automotive market. Also, the proposed

*References presented at the end of this section.

EV will benefit from having far fewer parts than the ICE vehicle, which should result in lower assembly costs and reduced maintenance. In the economic analysis presented below, several EV designs are evaluated using cost projections from both Detroit and recent plastics industry studies. The lifetime vehicle costs for these EV designs were calculated using a special computer program, the electric vehicle economic analysis model (EVECOM). The lifetime costs are broken down into three categories: (1) acquisition; (2) research, development, testing, and evaluation (RDT&E); and (3) ownership. The critical cost element studied during this phase of the program was acquisition cost. The remaining two categories impact the total vehicle cost/mile, but are held fixed at a baseline estimate for the several designs considered.

ACQUISITION COST

The vehicle acquisition cost comprises all costs associated with the manufacturing, distribution, and marketing of the vehicle as reflected in the sticker price. In determining the total vehicle life-cycle costs, the costs of financing incurred by the owner and any salvage value remaining after the vehicle's useful life are also included in the acquisition cost. In estimating the manufacturing costs, namely labor, raw materials, and capitalization, the vehicle was broken down into components and subsystems (see Table 6-1). A cost estimating relationship (CER) based on the weight of the component was developed from the vehicle manufacturing data available in References 6-2 and 6-4. The distribution and marketing costs are accounted for in the dealer markup factor, which is computed as a function of manufacturing cost from an expression developed in Reference 6-4. The financing terms were established after a local survey of currently available auto financing plans. The salvage value was set at 7 percent, which is consistent with the analysis presented in Reference 6-2.

The study results presented in this section separately demonstrate the impact on acquisition cost of four new technology EV design alternatives, each of which feature the new electric propulsion system. These alternatives are:

- (a) Based on conventional steel production
- (b) Based on current plastic production
- (c) Based on plastic mass production
- (d) Based on mass production plastic and motors

The four design alternatives focus on potential material and production options for a few critical vehicle subsystems that total 40 percent of the vehicle weight. Referring to Table 6-1, these subsystems are frame and body structure, removable panels, basic body, motor, generator, and flywheel. The weight and cost specification of each of these subsystems are developed separately for each design alternative.

TABLE 6-1

BASED ON MASS PRODUCTION PLASTICS AND ELECTRIC MOTORS,
VEHICLE COST BREAKDOWN CATEGORIES

Item	Category	Determination Method	Units
1	Frame and body structure (unitized frame)	$C_{FB} = 0.67$ where C (in general) is in \$/lb or \$/mi and subscripts refer to appropriate individual components	\$/lb
2	Removable panels (2 doors)	$C_{RP2D} = 1.17 - 0.0000403 W_c$, where W_c = vehicle curb weight (lb)	\$/lb
3	Removable panels (4 doors)	$C_{RP4D} = 1.17 - 0.0000310 W_c$	\$/lb
4	Basic body (includes interior and exterior trim, accessories, lights, glass, heater, wiper/washer, crash and panel pads)	$C_{BB} = 1.45$	\$/lb
5	Drive system, rear drive	$C_{DSR} = 0.79 - 0.0000719 W_c$	\$/lb
6	Drive system, front drive	$C_{DSF} = 0.83 - 0.0000639 W_c$	\$/lb
7	Suspension system, coil/leaf	$C_{SUS c/l} = 0.61 - 0.0000170 W_c$	\$/lb
8	Suspension system, coil/coil	$C_{SUS c/c} = 0.67 - 0.0000179 W_c$	\$/lb
9	Brake system, drum/drum	$C_{BR DRM/DRM} = 0.536 - 0.0000213 W_c$	\$/lb
10	Brake system, disk/drum	$C_{BR DSK/DRM} = 0.60 - 0.0000379 W_c$	\$/lb
11	Steering system, front drive, ball and rack and pinion	$C_{STR,BALL,R\&P,F} = 1.32 - 0.0000379 W_c$	\$/lb
12	Steering system, rear drive, ball and rack and pinion	$C_{STR,BALL,R\&P,R} = 1.35 - 0.0000368 W_c$	\$/lb
13	Wheels, 4 or 5	$C_{WH (4),(5)} = 0.109 + 0.000109 W_c$	\$/lb
14	Tires, 5 bias ply	$C_{TIRE B1A (5)} = 0.404$	\$/lb
15	Tires, 5 radial ply	$C_{TIRE RAD (5)} = 0.572$	\$/lb
16	Tires, 4 run flat	$C_{TIRE RUN FLAT (4)} = 0.850 + 0.000115 W_c$	\$/lb
17	Restraints	$C_{RES} = 2.042$	\$/lb
18	Air Conditioning	$C_{A/C} = 4.427$	\$/lb
19	Propulsion System, Motor	$C_{MOT} = 1.35$	\$/lb
20	Generator	$C_{GEN} = 1.35$	\$/lb
21	Flywheel	$C_{FW} = 3.57$	\$/lb
22	Flywheel Housing	$C_{FWH} = 1.10$	\$/lb
23	Gearbox	$C_{GB} = 1.10$	\$/lb
24	Controller	$C_{CONTR} = 20.00$	\$/kw
25	Battery Charger	$C_{BC} = 8.00$	\$/kw
26	Battery	$C_{BAT} = 0.42$	\$/lb
27	Vehicle Assembly	$C_{ASSEM} = 0.0668$	\$/lb curb wt
28	Total Manufacturing Cost (includes variable and fixed material and labor costs; engineering tooling costs; and quality control costs)	$C_{MANF, TOT} = \sum \text{Items 1 through 27}$	\$
29	Markup Factor (includes factory profit; warranty costs; marketing costs; facilities equipment costs; corporate profit; and dealer markups)	$MF = 0.0002918 (\text{manufacturing cost}) + 1.4083$	\$
30	Sticker Price (suggested retail selling price; does not include transportation, dealer preparation, or sales taxes)	$P_s = (MF)(C_{MANF, TOT})$	\$

Of the remaining categories listed in Table 6-1, those making up the running gear and vehicle assembly (categories 5 through 18 and 27) are defined using the data in Reference 6-4. This DOT report presents weight and Detroit manufacturing cost data on six production car sizes, and specifies over 200 components per vehicle. For this analysis, the cost and weight of each running gear component was defined as a function of curb weight so that the equivalent Detroit component could be specified for each design alternative. The cost and weight of the flywheel housing, gearbox, and battery charger (categories 22, 23, and 25) are based on AiResearch estimates. The controller cost (category 24) is estimated from data furnished in Reference 6-2, and this is consistent with AiResearch cost projections. The controller comprises a microprocessor, sensors, and three choppers of 16.4-kw total peak power rating. At the current unit price for choppers of \$7.68/kw, 40 percent of the estimated \$200 controller cost can be assigned to the microprocessor and sensors. The battery unit OEM cost of \$0.42/lb is taken from Reference 6-2 for lead acid batteries.

Cost Based on Conventional Steel Production Vehicle

This economic analysis is for the conversion of an existing Detroit subcompact--the 1975 Chevrolet Vega--to battery power (see Figure 6-1). The conversion is carried out by substituting electric components for all ICE related components (as specified in Reference 6-4). The 606-lb Detroit propulsion system, which includes a 40-lb battery, a 16-lb dc starter motor, and a 47-lb catalytic converter, is replaced with the 354-lb AiResearch propulsion system. Although both systems feature flywheels, a 45-lb composite flywheel is essentially substituted for the 36-lb flywheel of the ICE. The gasoline energy storage system, a 15-lb tank and nominally 80 lb of fuel, is replaced by 1040 lb of lead acid batteries and 60 lb of installation trays. The remainder of the vehicle is considered to be as defined in Reference 6-4 for the Detroit subcompact (i.e., steel frame and body structure at \$0.40/lb, sheet steel removable panels at \$0.79/lb, and the standard subcompact interior and running gear).

The conversion to battery power increases the vehicle weight from 2567 lb with a full tank of gas to 3208 lb, a 25 percent net gain totaling 641 lb. The concurrent increase in manufacturing cost, from \$1720 to \$3315, reflects the assumed change in propulsion system production technology. The 606-lb Detroit ICE and related components are mass produced at \$0.56/lb, while the AiResearch propulsion system reflects limited production costs of \$3.82/lb. Current EV traction motor production cost (reported in Reference 6-2) is \$4.00/lb for the traction motor industry as a whole. The AiResearch cost estimate for the proposed flywheel, based on limited production, is \$4.00/lb, as compared to the Detroit production cost for a 1.5-kw dc starter motor of \$0.50/lb.

The converted Detroit subcompact is very similar in battery type, chassis design, and production costs to the reference EV presented in the DOT report of August 1976 (Reference 6-2). The latter economic study assumed a conventional EV propulsion system, and therefore offers a meaningful cost comparison between the AiResearch design and conventional designs. The cost and weight of the principal subsystems of the two vehicles are summarized in Table 6-2, which shows a 27 percent reduction in battery weight, traction motor size, and

ERDA ELECTRIC VEHICLE PROGRAM PHASE I AUGUST 1976

VEHICLE ACQUISITION COST SUMMARY

(1975 \$)

ANNUAL PRODUCTION RATE 100000 VEHICLES/YR

PAGE 2

RUN NO. 101

DATE 30 NOV 76

VEHICLE DESIGN I.D. 101

VEHICLE COMPONENT/SYSTEM	WEIGHT (LBS)	COST (\$)	COST/LB (\$/LB)	% OF STICKER PRICE
FRAME + BODY STRUCTURE (UNITIZED FRAME)	431.97	172.79	.40	2.19
REMOVABLE PANELS (2 DOORS)	443.71	382.44	.79	4.86
BASIC BODY	225.00	279.00	1.24	3.54
DRIVE SYSTEM (REAR DR)	110.00	84.61	.77	1.07
SUSPENSION SYSTEM (ROT/COIL)	138.10	84.99	.62	1.08
BRAKE SYSTEM (DRK/DRM)	140.50	88.99	.59	1.13
STEERING SYSTEM (RACK + PITON, REAR CR)	37.00	49.65	1.34	.63
WHEELS (4)	69.00	31.66	.46	.40
TIRES (4 RUN FLAT)	105.00	128.01	1.22	1.63
RESTRAINTS (4)	6.00	12.25	2.04	.16
AIR CONDITIONING	-.00	.00	.00	.00
PROPULSION SYSTEM				
MOTOR	106.00	424.00	4.00	5.38
GENERATOR	106.00	424.00	4.00	5.38
FLYWHEEL	45.50	182.00	4.00	2.31
FLYWHEEL HOUSING	16.00	17.60	1.10	.22
GEAR BOX	20.00	22.00	1.10	.28
CONTROLLER	10.00	200.00	20.00	2.54
BATTERY CHARGER	30.00	80.00	2.67	1.02
SYSTEM INSTALLATION	20.00	.00	.00	.00
BATTERY	1040.00	436.80	.42	5.55
BATTERY INSTALLATION	60.00	.00	.00	.00
TOTAL PROPULSION SYSTEM	353.50	1349.60	3.82	17.14
TOTAL BATTERY SYSTEM	1160.00	436.80	.40	5.55
VEHICLE ASSEMBLY		214.41	.07	2.72
VEHICLE CURR WEIGHT	3209.78		.00	.00
TOTAL MANUFACTURING COST		3315.21	1.03	42.10
FACTORY + DEALER MARKUPS		4560.33	1.42	57.90
SUGGESTED RETAIL STICKER PRICE		7875.54	2.45	100.00

Figure 6-1. Acquisition Cost Summary for the EV
Based on Conventional Steel Production

controller power rating. The sticker price published in the DOT report is \$9500, although the Detroit pricing formula used in the present study would result in a price of \$12,530.

TABLE 6-2
ECONOMIC ANALYSIS SUMMARY

Vehicle Component	Present Electric Vehicles per Reference 6-2		Based on Conventional Steel Production (see Figure 6-1)		Based on Current Plastic Production (see Figure 6-2)		Based on Plastic Mass Production (see Figure 6-3)		Based on Mass Production of Plastics and Motors (see Figure 6-4)	
	Weight, lb	Cost, \$	Weight, lb	Cost, \$	Weight, lb	Cost, \$	Weight, lb	Cost, \$	Weight, lb	Cost, \$
Chassis	1625	1490	1756	1530	1193	1455	1193	1199	1193	1199
Motor	315	1260	293	1048	293	1048	293	1048	293	488
Controller	85	1190	10	200	10	200	10	200	10	200
Battery	1500	630	1040	437	1040	437	1040	437	1040	437
Total	3525	4570	3210	3315	2587	3239	2587	2985	2587	2404
Sticker price	(9500) \$12,530		\$7875		\$7623		\$6805		\$5072	
Total cost/mile	\$0.28		\$0.19		\$0.18		\$0.17		\$0.15	

Cost Based on Current Plastic Production

This economic analysis is for a four-passenger electric commuter vehicle that makes vigorous use of lightweight materials at the expense of traditional Detroit steel (discussed in Section 3). The breakdown of materials in the frame and body structure and removable panels is given in Table 6-3.

TABLE 6-3
WEIGHTS OF CHASSIS MATERIALS

Component	Aluminum	FRP	Steel	Adhesive	Davisorb	Total
Frame and body structure, lb	67	234	20	20	0	341
Removable panels, lb	80	90	10	0	55	235

Fiber-reinforced plastic (FRP) replaces much of the steel in the frame and body structure. FRP, aluminum, and Davisorb are all utilized as weight saving materials in the removable panels. Davisorb is a trade name for a resilient fiber plastic used in bumpers. The weight breakdown for the entire vehicle is given in Figure 6-2. Several weight reductions relative to the subcompact defined in Figure 6-1 are achieved. The reduction of the weight of the basic body subsystem reflects more weight-efficient seat design and the elimination of all accessories except instrumentation. Some weight savings in the running gear also are gained. The drive system weight is reduced due to elimination of the drive shaft. The brake system size requirement could be reduced because of the vehicle regenerative braking capability. Finally, the weight of the steering system benefited from the reduced length of the steering column.

The production costs for all chassis components except the frame and body structure and the removable panels were generated from the cost data detailed in Reference 6-4. The costs of the latter two components were developed from the unit cost data listed in Table 6-4; these data were obtained from industry sources.

TABLE 6-4
COST DATA FOR FRAME/BODY AND REMOVABLE PANELS

Material	Cost, \$/lb
Aluminum	1.25
FRP	1.11
Steel	1.00
Adhesive	1.00
Davisorb	2.20

The production cost for aluminum includes tooling costs similar to those for steel, but reflects the increase in raw material cost from \$0.18/lb for steel to \$0.59/lb for aluminum. The cost data on FRP is based on a low volume specialty car application, and is thus more labor intensive than traditional Detroit assembly lines. The Davisorb cost is direct from the supplier. The cost/lb of these two subsystems is computed from the data listed in Tables 6-3 and 6-4. Thus, the relative cost for the frame and body and removable panels can be determined as follows:

$$\text{Frame and Body, } C_{FB} = \frac{67(1.25) + 234(1.11) + 20(1.00) + 20(1.00)}{341} = 1.18$$

$$\text{Removable Panels, } C_{FP} = \frac{80(1.25) + 90(1.11) + 10(1.00) + 55(2.20)}{235} = 1.14$$

ERDA ELECTRIC VEHICLE PROGRAM PHASE I AUGUST 1976

RUN NO. 103

VEHICLE ACQUISITION COST SUMMARY

DATE 30 NOV 76

(1975 \$)

VEHICLE DESIGN I.D. 103

ANNUAL PRODUCTION RATE 100000. VEHICLES/YR

VEHICLE COMPONENT/SYSTEM	WEIGHT (LBS)	COST (\$)	COST/LB (\$/LB)	% OF STICKER PRICE
FRAME + BODY STRUCTURE (UNITIZED FRAME)	341.05	402.44	1.18	5.28
REMOVABLE PANELS (2 DOORS)	245.00	330.35	1.41	4.33
BASIC BODY	132.20	191.69	1.45	2.51
DRIVE SYSTEM (REAR DR)	45.00	51.34	.60	.67
SUSPENSION SYSTEM (COIL/COIL)	120.00	67.92	.57	.89
BRAKE SYSTEM (DISK/DRM)	100.00	54.46	.54	.71
STEERING SYSTEM (PARK + PINTON, REAR DR)	30.00	37.64	1.25	.49
WHEELS (4)	44.00	17.20	.39	.23
TIRES (4 RUN FLAT)	100.00	114.75	1.15	1.51
RESTRAINTS (4)	6.00	12.25	2.04	.16
AIR CONDITIONING	0.00	.00	.00	.00
PROPULSION SYSTEM				
MOTOR	106.00	424.00	4.00	5.56
GENERATOR	106.00	424.00	4.00	5.56
FLYWHEEL	45.50	182.00	4.00	2.39
FLYWHEEL HOUSING	16.00	17.60	1.10	.23
GEAR BOX	20.00	22.00	1.10	.29
CONTROLLER	10.00	200.00	20.00	2.62
BATTERY CHARGER	10.00	80.00	2.67	1.05
SYSTEM INSTALLATION	20.00	.00	.00	.00
BATTERY	1000.00	436.80	.42	5.73
BATTERY INSTALLATION	0.00	.00	.00	.00
TOTAL PROPULSION SYSTEM	343.50	1349.60	3.82	17.70
TOTAL BATTERY SYSTEM	1000.00	436.80	.42	5.73
VEHICLE ASSEMBLY		172.79	.07	2.27
VEHICLE CURR WEIGHT	2546.75		.00	.00
TOTAL MANUFACTURING COST		3239.25	1.25	42.49
FACTORY + DEALER MARKUPS		4384.05	1.69	57.51
SUGGESTED RETAIL STICKER PRICE		7623.30	2.95	100.00

Figure 6-2. Acquisition Cost Summary for the EV
Based on Current Plastic Production

The comparison of the production costs in Figure 6-2 with the steel subcompact of Figure 6-1 leads to some interesting observations. The wholesale substitution of FRP for steel in the frame and body structure decreased weight by 20 percent, but increased costs by over 200 percent. In contrast, the use of aluminum and FRP in the removable panels decreased weight by 50 percent, while cost increased by just over 10 percent. The cost reductions reflected in the remaining chassis components are essentially benefits of the change in overall vehicle design.

In total cost, the limited-production, lightweight EV is slightly less than the mass-produced steel EV. The use of FRP results in fewer parts and reduced assembly costs. The lightweight design succeeded in decreasing the curb weight from 3210 to 2587 lb, which is within 1 percent of the fully fueled ICE Detroit subcompact discussed in the previous analysis. The weight reduction of 623 lb converts directly to improved vehicle efficiency.

Cost Based on Plastic in Mass Production

The analysis presented below examines the effect of plastic mass production on lightweight vehicle construction. Owens/Corning (Reference 6-5) has projected the impact of high throughput compression molding SMC technology on the production cost of vehicle components, developed in this case for a typical door design. SMC refers to sheet molding compound, a type of FRP for which an applicable cost analysis has been performed. SMC cost data is representative of mass production FRP technology and is suitable as a data base for the Phase I study. Further work on FRP technology for the automotive application is expected. The cost components for the production of a bare door, exclusive of the hinges, latches, window, and interior panel, are summarized in Table 6-5.

TABLE 6-5
CORNING DOOR ANALYSIS

Category	Material	
	Steel	SMC
Labor Content, \$/door	1.29	0.44
Raw Material, \$/door	9.93	14.02
Investment, \$/door	8.92	2.97
Total (bare door), \$/door	20.14	17.43
Weight (bare door), lb	55	37
Unit Cost, \$/lb	0.37	0.47

The SMC technology is cost-competitive on a component basis with the Detroit steel fabrication assembly line technology. The labor content of the SMC bare door is decreased by 81 percent by the substitution of high throughput compression molding for a conventional compression molding found in a custom molder's shop. The cycle time of the molding operation is reduced from 2.4

minutes to 0.4 minutes. Steel benefits from a two-to-one material cost edge (\$0.18/lb to \$0.36/lb). However, SMC requires 66 percent less capital investment for tooling, equipment, and building. The capital required to construct and tool-up a steel door stamping/assembly line far exceeds that needed for an SMC molding operation, and SMC offers the advantages of lighter weight and inherent corrosion resistance. On the basis of this analysis, the production costs for the lightweight components used in the preceding analysis can be revised if mass production technology is assumed. The revised costs are listed in Table 6-6.

TABLE 6-6

REVISED COSTS FOR FRAME/BODY AND REMOVABLE PANELS

Material	Cost, \$/lb
Aluminum	1.05
FRP (SMC)	0.47
Steel	0.37
Adhesive	1.00
Davisorb	2.20

Using these revised costs in the calculation of the component unit cost parameters, the frame and body and removable panel costs can be calculated as follows:

$$\text{Frame and Body, } C_{FR} = \frac{67.35(1.05) + 234(0.47) + 20(0.37) + 20(1.00)}{341} = \$0.67/\text{lb}$$

$$\text{Removable Panels, } C_{RP} = \frac{80(1.05) + 90(0.47) + 55(2.20) + 10(0.37)}{235} = \$1.07/\text{lb}$$

The revised cost summary for the mass-produced, lightweight vehicle is shown in Figure 6-3. Although the cost of the frame and body structure has been reduced by 43 percent by the introduction of mass production technology, it remains 33 percent greater than the comparable steel design. The cost of the removable panels is reduced by 24 percent. The sticker price is lowered by 11 percent to \$6805.

Cost Based on Mass Production of Plastics and Electric Motors

This cost analysis includes the economies of scale in the production cost of the propulsion system. The DOT EV report (Reference 6-2) projects the mass production costs (>1,000,000) for traction motors based on the ratio of materials cost to OEM cost for the electric motor industry as a whole, given to be 2.38. Under the assumption that this class of tooling can be included in the manufacturing plan for the production of 100,000 vehicles/year, this result is applied to the production cost of the current motor and generator as shown in Table 6-7.

ERDA ELECTRIC VEHICLE PROGRAM PHASE I AUGUST 1976

VEHICLE ACQUISITION COST SUMMARY

(1975 \$)

ANNUAL PRODUCTION RATE 100000, VEHICLES/YR

PAGE 2

RUN NO. 104

DATE 30 NOV 76

VEHICLE DESIGN T.O. 103

VEHICLE COMPONENT/SYSTEM	WEIGHT (LBS)	COST (\$)	COST/LB (\$/LB)	% OF STICKER PRICE
FRAME + BODY STRUCTURE (UNITIZED FRAME)	341.05	228.50	.67	3.36
REMOVABLE PANELS (2 DOORS)	235.00	250.45	1.07	3.64
BASIC BODY	132.20	191.69	1.45	2.82
DRIVE SYSTEM (REAR DR)	85.00	51.34	.60	.75
SUSPENSION SYSTEM (COTL/COIL)	120.00	67.92	.57	1.00
BRAKE SYSTEM (DSK/DRM)	100.00	54.46	.54	.80
STEERING SYSTEM (PACK + PITON, REAR DR)	30.00	37.64	1.25	.55
WHEELS (4)	44.00	17.20	.39	.25
TIRES (4 RUN FLAT)	100.00	114.75	1.15	1.69
RESTRAINTS (4)	6.00	12.25	2.04	.18
AIR CONDITIONING	0.00	0.00	0.00	0.00
PROPULSION SYSTEM				
MOTOR	106.00	424.00	4.00	6.23
GENERATOR	106.00	424.00	4.00	6.23
FLYWHEEL	45.50	182.00	4.00	2.67
FLYWHEEL HOUSING	16.00	17.60	1.10	.26
GEAR BOX	20.00	22.00	1.10	.32
CONTROLLER	10.00	200.00	20.00	2.94
BATTERY CHARGER	30.00	80.00	2.67	1.18
SYSTEM INSTALLATION	20.00	0.00	0.00	0.00
BATTERY	1040.00	436.80	.42	6.42
BATTERY INSTALLATION	0.00	0.00	0.00	0.00
TOTAL PROPULSION SYSTEM	353.50	1349.60	3.82	19.83
TOTAL BATTERY SYSTEM	1040.00	436.80	.42	6.42
VEHICLE ASSEMBLY		172.79	.07	2.54
VEHICLE CURB WEIGHT	2546.75		0.00	0.00
TOTAL MANUFACTURING COST		2985.41	1.15	43.87
FACTORY + DEALER MARKUPS		3819.40	1.48	56.13
SUGGESTED RETAIL STICKER PRICE		6804.82	2.63	100.00

Figure 6-3. Acquisition Cost Summary for the EV
Based on Plastic in Mass Production

TABLE 6-7
COST OF MATERIALS FOR MOTOR/GENERATOR

Category	Weight, lb	Unit Price, \$/lb	Cost, \$
Copper	32.5	0.88	28.62
Iron	51.8	0.10	5.17
Steel	8.5	0.14	1.22
Aluminum	6.9	0.49	3.38
Other	<u>6.3</u>	<u>1.00</u>	<u>6.34</u>
Subtotal	106		44.73
Scrappage (35 percent of purchased material) at \$0.42/lb	37.1		15.50
Total purchased material	143.1		60.31
Priced materials/lb of motor produced			\$ 0.57
OEM price/lb at 2.38 (unit cost of material), $C_{MOT} =$			\$ 1.35/lb

This type of calculation is repeated in Table 6-8 for the flywheel to establish a similar limit given a large production capability.

The introduction of mass production technology to the fabrication of the motor and generator results in a 66 percent reduction in component cost. A reduction of this order, as detailed in Reference 6-2, is predicated on a total annual production rate of 1,000,000 traction motor units/year. The current annual production rate is 200,000 units/year. Thus, this production cost must be viewed as a far-term industry capability. The impact on the sticker price of the mass-produced lightweight vehicle is dramatic. The \$582 savings in manufacturing cost indicated in Figure 6-4 reduces the suggested retail sticker price to \$5072.

Acquisition Cost Study Conclusions

The preceeding analyses, summarized in Table 6-2, show that the acquisition cost for the four-passenger electric commuter vehicle can be reduced from an initial \$7875 to \$5072, while reducing the curb weight from 3210 lb to 2587 lb. The Detroit material and production vehicle is immediately obtainable, with little possibility of any significant reduction in chassis production cost.

TABLE 6-8
COST OF MATERIALS FOR AN S-GLASS FLYWHEEL

Category	Weight, lb	Unit Price, \$/lb	Cost, \$
Aluminum	12.0	0.49	5.88
S-Glass	<u>33.5</u>	1.60	<u>53.60</u>
Subtotal	45.5		59.48
Scrappage (15 percent of purchased material) at \$1.35/lb	6.8		8.91
Total purchased material	52.3		68.39
Price of materials/lb of flywheel produced			1.50
CFW, \$/lb at 2.38 (unit cost of material)			3.57

Also, the severe weight penalty substantially reduces the performance, and therefore the marketability of the vehicle. In contrast, the limited-production lightweight vehicle exceeds the baseline performance requirements, and should be considered as a marketable goal for vehicle design in the near term.

In the last two analyses presented above, the potential economies of scale are considered. A mass production manufacturing facility utilizing high-throughput SMC compression molding should achieve the mass production chassis economies in the near-term timeframe. The final decrement down to the \$5072 sticker price must be considered the far-term goal necessary for gaining consumer acceptance in the general commuter vehicle market.

RDT&E COST

This cost category includes the research and development costs that would be incurred by an electric vehicle manufacturer prior to undertaking a large-scale production program for a specific electric vehicle model or models. This is distinct from, but not independent of, U.S. Government programs such as the current ERDA electric vehicular program and other possible future government-sponsored electric vehicular programs. It represents the manufacturer's in-house development effort prior to committing the large capital investments required for mass production and incorporates all knowledge and expertise gained from outside programs, such as the on-going ERDA program, and the manufacturer's own background and experience. In estimating the total RDT&E costs, the following cost elements have been defined:

ERMA ELECTRIC VEHICLE PROGRAM PHASE I AUGUST 1976

VEHICLE ACQUISITION COST SUMMARY

(1975 \$)

ANNUAL PRODUCTION RATE 100000 VEHICLES/YR

PAGE 2

RUN NO. 106

DATE 01 DEC 76

VEHICLE DESIGN I.D. 103

VEHICLE COMPONENT/SYSTEM	WEIGHT (LBS)	COST (\$)	COST/LB (\$/LB)	% OF STICKER PRICE
FRAME + BODY STRUCTURE (UNITIZED FRAME)	301.05	228.50	.67	4.51
REMOVABLE PANELS (2 DOORS)	235.00	250.45	1.07	4.90
BASIC BODY	132.20	191.69	1.45	3.78
DRIVE SYSTEM (REAR DR)	85.00	51.34	.60	1.01
SUSPENSION SYSTEM (FOTL/COIL)	120.00	67.92	.57	1.34
BRAKE SYSTEM (DISK/DRM)	100.00	54.46	.54	1.07
STEERING SYSTEM (RACK + PITMAN-REAR DR)	30.00	37.64	1.25	.74
WHEELS (4)	44.00	17.20	.39	.34
TIRES (4 RUN FLAT)	100.00	114.75	1.15	2.26
RESTRAINTS (4)	6.00	12.25	2.04	.24
AIR CONDITIONING	0.00	.00	.00	.00
PROPULSION SYSTEM				
MOTOR	106.00	143.10	.35	2.82
GENERATOR	106.00	143.10	.35	2.82
FLYWHEEL	45.50	162.44	3.57	3.20
FLYWHEEL HOUSING	16.00	17.60	.10	.35
GEAR BOX	20.00	22.00	.10	.43
CONTROLLER	10.00	200.00	20.00	3.94
BATTERY CHARGER	30.00	80.00	2.67	1.58
SYSTEM INSTALLATION	20.00	.00	.00	.00
BATTERY	1040.00	436.80	.42	8.61
BATTERY INSTALLATION	0.00	.00	.00	.00
TOTAL PROPULSION SYSTEM	353.50	768.23	2.17	15.15
TOTAL BATTERY SYSTEM	1040.00	436.80	.42	8.61
VEHICLE ASSEMBLY		172.79	.07	3.41
VEHICLE CURB WEIGHT	2586.75		.00	.00
TOTAL MANUFACTURING COST		2404.05	.93	47.40
FACTORY + DEALER MARKUPS		2667.86	1.03	52.60
SUGGESTED RETAIL STICKER PRICE		5071.91	1.96	100.00

Figure 6-4. Acquisition Cost Summary for the EV
Based on Mass Production of Plastics
and Electric Motors

- (a) Initial engineering
- (b) Development tooling
- (c) Development support
- (d) Manufacture of the R&D vehicles
- (e) Test facilities
- (f) Test operations
- (g) Contractor profit

The general calculational algorithm for RDT&E costs is:

$$\text{Total Cost} = (\text{element hours}) (\text{composite rate}) + (\text{man-hour related costs}) + (\text{profit}), (\$)$$

The RDT&E costs used in the present study are based on initial AIResearch estimates. The impact on vehicle life-cycle costs is minimal for the assumed five-year amortization period and 100,000-vehicle/year production rate. A comprehensive study during subsequent program phases is anticipated.

OWNERSHIP COSTS

The ownership costs cover the expense due to operating and maintaining the vehicle. The following items are included:

- (a) Maintenance and repair cost for the complete vehicle
- (b) Tire replacement cost
- (c) Accessories cost (such as expenditures for floor mats, seat covers, etc.)
- (d) Insurance, including comprehensive coverage, collision, and liability
- (e) Registration and licensing
- (f) Garaging, parking, and toll costs
- (g) Battery recharging energy cost
- (h) Battery replacement cost, including financing charges

State and federal taxes on gasoline, oil, and tires are not broken out separately because they are assumed to be either not applicable for the electric car or, as in the case of tires, are included in the appropriate cost element.

The maintenance and repair cost of the propulsion system has been analyzed for maintenance and repair requirements by breaking it down into its constituent components and estimating the wearout or replacement life of the critical parts and assemblies in each principal component. In the case of the traction motor, for example, the critical items identified were the motor brushes and the rotor bearing assembly. Based on previous history with similar components and duty cycle environments, it was then estimated that brush replacement would be required after 25,000 miles of vehicle operation; bearings will be designed for a life of at least 100,000 miles of operation. Similarly, it was assumed that a finite percentage of controller components such as switches and transistors would be susceptible to failure due to overheating, random power spikes, or corrosion, and would require servicing every 50,000 miles.

The maintenance and repair cost for the chassis and body was averaged over the life of the vehicle as a function of curb weight. The data for this CER and for the remainder of the items listed above were taken from References 6-6 and 6-7. The entire list of CER's is given in Table 6-9. The complete cost analysis, including RDT&E, acquisition, and ownership costs, for the EV based on mass-produced plastics and motors is given in Figure 6-5.

TABLE 6-9
OWNERSHIP COST BREAKDOWN

Item	Category	Determination Method	Units
1	Maintenance and Repair Cost, other than Propulsion System	$C_{MCR, \text{ other}} = \frac{65.4 + 0.0118 W_c}{\text{annual mileage}}$ <p style="text-align: center;">where W_c = vehicle curb weight, lb</p>	\$/mi
2	Maintenance and Repair Cost, Propulsion System	$\text{Total Cost} = \left(\frac{\text{total vehicle life (mi)}}{\text{Scheduled maint. Interval (mi)}} - 1 \right) \left[(\text{mean time to repair})(\text{labor rate}) + (\text{parts cost per maint. Interval}) \right]$	\$
3	Tire Replacement Cost	$C_{TIRE \text{ REPL}} = \left(\frac{\text{total vehicle life (mi)}}{\text{Tire life (mi)}} \right) \left[\frac{(\text{OEM cost of tire})(\text{retail markup factor})}{\text{total vehicle life (mi)}} \right]$	\$/mi
4	Accessories Cost	$C_{ACCESS} = \frac{27.40}{\text{annual mileage}}$	\$/mi
5	Insurance Cost	$C_{INSUR} = \frac{375.00}{\text{annual mileage}}$	\$/mi
6	Registration and License Cost	$C_{REG \ \& \ LIC} = \frac{17.00}{\text{annual mileage}}$ <p>This formulation assumes that no personal property tax is levied against the vehicle and that no taxes in lieu of the State and Federal gasoline taxes are imposed.</p>	\$/mi
7	Garaging and Parking Cost (Includes tolls)	$C_{GAR \ \& \ PARK} = \frac{213.00}{(\text{annual mileage})}$	\$/mi
8	Battery Recharging Energy Cost	$C_{ENERGY} = (\text{recharge energy requirement, kwh/mi}) (\text{energy cost, \$/kwh})$	\$/mi
9	Battery Replacement	$C_{BAT \text{ REPL}} = \left[\frac{\text{total vehicle life, yr} - 1}{\text{battery life, yr}} \right] \left[\frac{(\text{OEM cost of battery})(\text{retail markup factor})}{(\text{total vehicle life, mi})} \right]$	\$/mi

ERDA ELECTRIC VEHICLE PROGRAM PHASE 1 AUGUST 1976
 RESEARCH, DEVELOPMENT, TESTING, & EVALUATION (RDTE) COST SUMMARY
 (1975 \$)

PAGE 1

RDA AC. 106

DATE 01 DEC 76

VEHICLE DESIGN I.D. 103

NUMBER OF R & D VEHICLES 3.

COST ELEMENT	HOURS (HRS)	COMPOSITE RATE (\$/HR)	TOTAL COST(\$)	TOTAL COST INCLUDING CONTRACTOR PROFIT
INITIAL ENGINEERING	192400.00	25.00	4835000.00	5318499.90
DEVELOPMENT TOOLING	125000.00	20.00	2500000.00	2860000.00
DEVELOPMENT SUPPORT	100500.00	21.00	2110500.00	2364999.90
MANUFACTURE OF R&D VEHICLES			82894.62	91184.08
TEST FACILITIES			100000.00	110000.00
TEST OPERATIONS	2500.00	22.50	56250.00	67375.00
TOTAL, RDTE			9829144.50	10812058.40

Figure 6-5. Complete Cost Analysis for the EV Based on Mass Production of Plastics and Electric Motors

PAGE 2

ERDA ELECTRIC VEHICLE PROGRAM PHASE I AUGUST 1976

RUN NO. 106

VEHICLE ACQUISITION COST SUMMARY

DATE 01 DEC 76

(1975 \$)

VEHICLE DESIGN I.D. 103

ANNUAL PRODUCTION RATE 100000. VEHICLES/YR

VEHICLE COMPONENT/SYSTEM	WEIGHT (LBS)	COST (\$)	COST/LB (\$/LB)	% OF STICKER PRICE
FRAME + BODY STRUCTURE (UNITIZED FRAME)	341.05	228.50	.67	4.51
REMOVABLE PANELS (2 DOORS)	235.00	250.45	1.07	4.94
BASIC BODY	132.20	191.69	1.45	3.78
DRIVE SYSTEM (REAR DR)	85.00	51.38	.60	1.01
SUSPENSION SYSTEM (COIL/COIL)	120.00	87.92	.97	1.38
BRAKE SYSTEM (DSK/DRM)	100.00	54.46	.54	1.07
STEERING SYSTEM (RACK + PINION, REAR DR)	30.00	37.64	1.25	.74
WHEELS (4)	44.00	17.20	.39	.34
TIRES (4 RUN FLAT)	100.00	114.75	1.15	2.26
RESTRAINTS (4)	6.00	12.25	2.04	.24
AIR CONDITIONING	0.00	.00	.00	.00
PROPULSION SYSTEM				
MOTOR	106.00	143.10	1.35	2.82
GENERATOR	106.00	143.10	1.35	2.82
FLYWHEEL	45.50	162.44	3.57	3.20
FLYWHEEL HOUSING	16.00	17.60	1.10	.39
GEAR BOX	20.00	22.00	1.10	.43
CONTROLLER	10.00	200.00	20.00	3.98
BATTERY CHARGER	30.00	80.00	2.67	1.58
SYSTEM INSTALLATION	20.00	.00	.00	.00
BATTERY	1040.00	436.80	.42	8.61
BATTERY INSTALLATION	0.00	.00	.00	.00
TOTAL PROPULSION SYSTEM	353.50	768.23	2.17	15.15
TOTAL BATTERY SYSTEM	1040.00	436.80	.42	8.61
VEHICLE ASSEMBLY		172.79	.07	3.41
VEHICLE CURB WEIGHT	2586.75		.00	.00
TOTAL MANUFACTURING COST		2404.05	.93	47.40
FACTORY + DEALER MARKUPS		2667.86	1.03	52.60
SUGGESTED RETAIL STICKER PRICE		5071.91	1.96	100.00

Figure 6-5. Continued

PAGE 3

ERDA ELECTRIC VEHICLE PROGRAM PHASE I AUGUST 1976
PROPULSION SYSTEM MAINTENANCE COSTS
(1975 \$)

RUN NO. 10A
DATE 01 DEC 76
VEHICLE DESIGN I.D. 103

AVERAGE ANNUAL MILEAGE 10000. (MI) TOTAL VEHICLE LIFE 10.00 (YRS)

COMPONENT	SCHEDULED MAINT INTERVAL (MTS)	MEAN TIME TO REPAIR/ REPLACE/INSPECT (HRS)	LABOR RATE (\$/HR)	PARTS COST/MAINT INTERVAL (\$)	LIFETIME TOTAL COST (\$)	COST/MILE (\$/MI)
1. MOTOR						
BRUSHES	25000.0	.50	12.00	10.00	48.00	.000480
BEARINGS	100000.0	1.50	12.00	20.00	.00	.000000
2. GENERATOR						
BRUSHES	25000.0	.50	12.00	10.00	48.00	.000480
BEARINGS	100000.0	1.50	12.00	20.00	.00	.000000
3. FLYWHEEL						
SEALS	50000.0	1.00	12.00	12.00	22.00	.000220
BEARINGS	50000.0	2.00	12.00	25.00	49.00	.000490
4. GEARBOX						
BEARINGS	50000.0	2.00	12.00	25.00	49.00	.000490
5. CONTROLLER	50000.0	1.25	12.00	200.00	215.00	.002150
6. BATTERY CHARGER	100000.0	1.30	12.00	12.50	.00	.000000
7. BATTERY	800.0	4.00	12.00	.10	12.40	.000124
TOTAL MAINTENANCE COST					443.40	.004434

Figure 6-5. Continued

ERDA ELECTRIC VEHICLE PROGRAM PHASE I AUGUST 1976

TOTAL SYSTEM COST SUMMARY

(1975 \$)

PAGE 4

RUN AC, 106

DATE 01 DEC 76

VEHICLE DESIGN I.D. 103

TOTAL VEHICLE LIFE 10. (YRS) AVERAGE ANNUAL MILEAGE 10000. (MI)
ANNUAL PRODUCTION RATE 100000. (VEHICLES/YR) WOT+E AMORTIZATION PERIOD 5. (YRS)

SYSTEM COST CATEGORY	LIFETIME COST (\$)	ANNUAL COST (\$/YR)	LIFETIME COST PER VEHICLE MILE (\$/MI)	% OF TOTAL
WOT+E	21.62	2.16	.00027	.1
ACQUISITION				
PURCHASE PRICE	5071.91	507.19	.05072	33.0
FINANCE CHARGE AT INTEREST RATE 12.00% AND TERM 40.0 MOS	1376.62	137.66	.01377	9.0
SALVAGE VALUE 7.0% OF ACQ.	355.03	35.50	.00355	2.3
TOTAL ACQUISITION	6093.49	609.35	.06093	39.6
OWNERSHIP				
MAINTENANCE + REPAIR				
PROPULSION + BATTERY SYSTEMS	443.40	44.34	.00443	2.9
OTHER	959.24	95.92	.00959	6.2
TOTAL MAINT	1402.64	140.26	.01403	9.1
TIRES REPLACEMENT TIRE LIFE 30000.0 MI.	468.55	46.86	.00469	3.0
ACCESSORIES	274.00	27.40	.00274	1.8
INSURANCE	3750.00	375.00	.03750	24.4
REGISTRATION + LICENSE	170.00	17.00	.00170	1.1
GARAGING, PARKING + TOLLS	7.06	.71	.00007	.0
ENERGY RATE 1 .002 \$/KWH	1848.00	184.80	.01848	12.0
BATTERY REPLACEMENT				
BATTERY	1351.68	135.17	.01352	8.8
FINANCE CHARGE AT INT. RATE 12.00% AND TERM 12.0 MOS	162.20	16.22	.00162	1.1
TOTAL BATTERY REPLACEMENT	1513.88	151.39	.01514	9.8
TOTAL OWNERSHIP	9264.87	926.48	.09265	60.2
TOTAL SYSTEM COST	15379.98	1536.00	.15360	100.0

Figure 6-5. Continued

REFERENCES

- 6-1. Hamilton, W.F., et al. Impact of Future Use of Electric Cars in the Los Angeles Region; Volume III, Task Reports on Impact and Usage Analyses, U.S. Environmental Protection Agency Report EPA-460/3-74-020-C, Ann Arbor, Michigan, October 1974.
- 6-2. Advisability of Regulating Electric Vehicles for Energy Conservation, report to the Congress and the President from the Secretary of Transportation, August 1976.
- 6-3. Fisher, Dan, "Close Look at Steel Costs Puts Price Hike in Dispute", Los Angeles Times, December 5, 1976, pp. VII-1, 2.
- 6-4. Fitzgibbon, R. G., and L. H. Lindgren, Estimated Weights and Manufacturing Costs of Automobiles, Fuel Economy Goals Beyond 1980, report prepared for U.S. Department of Transportation, 1976.
- 6-5. Elliot, B., "Fiberglas/Plastic Door Study," Owens-Corning Fiberglas Corporation.
- 6-6. Liston, L. L., and R. W. Sherrer, Cost of Operatng an Automobile, U.S. Department of Transportation, Federal Highway Administration, Washington, D.C., April 1974.
- 6-7. Kirkwook, T. F., and A. D. Lee, A Generalized Model for Comparing Automobile Design Approaches to Improved Fuel Ecomony, Report R-1562-NSF, The Rand Corporation, Santa Monica, Calif., January 1975.

SECTION 7

RELIABILITY AND SAFETY

RELIABILITY

The greatest increase in reliability achieved by the baseline electric-powered vehicle design is the simplification of the power train system and the reduction in the number of systems and components. A flywheel, traction motor, and simple planetary differential transmission are used in place of the engine cooling system, internal combustion engine, fuel system, exhaust system, and transmission (either a manual with clutch or an automatic) in a conventional car.

The traction motor and flywheel system are air cooled; are sealed from dirt and dust with low-cost, reliable seals; and use permanently lubricated, long-life bearings (100,000 customer miles). This system eliminates the need to change or add engine oil; to change engine oil filters; to change or add engine system coolant; to repair or replace corroded radiators; to replace deteriorated or leaking radiator and heater hoses; and to replace worn, deteriorated, or broken fan and cooling belts. Thus, reliability is significantly increased.

The planetary differential transmission is simple, is infinitely variable, and exceeds the efficiency and durability of a manual transmission without the synchronizer complexity, the clutch wearout and shudder problems, and/or clutch mechanism and linkage problems. It has the convenience of the automatic transmissions without the complexity and inefficiency. Existing automatic transmissions have been made even more inefficient to help meet engine emission control requirements.

The electrical system is simplified, and the number of parameters to be monitored is reduced. The ignition and emission systems are eliminated; many of today's complex emission systems require electronic controls, solenoids, plugs, and cables. Although the electronic controller is more complex than the standard electronic ignition module, it performs braking and speed control functions. The electronic controller and batteries also replace the fuel system, and the as yet unproven variable venturi carburetor.

The fiberglass body construction contributes to reliability and low maintenance by virtually eliminating corrosion, increasing resistance to dents, and eliminating the need for painting. Vehicle crashworthiness has been taken into consideration in the body design, and is discussed in greater detail under system safety at the end of this section.

Components from other systems will not have any adverse additional impact of reliability over the conventional vehicle because they will represent current technology. These components (steering gears, lamps, restraint systems, seats, interior trim, tires, and wheels) have proven reliability and are continually tested in the respective manufacturer's laboratory.

Vehicle testing is scheduled for three phases, each corresponding to a design phase. The test schedule is set up to give design direction for the next design phase. The test results will provide data for crash, structural, handling, performance, and durability evaluation. The test schedule is shown in the master plan schedule in Section 9.

Throughout the program, established reliability methods have been used to ensure reliability in design. For subsequent program phases, the following methods will be implemented to achieve high reliability at minimum cost.

- Engineering Review Request System--Provides a means of reporting problems or deficiencies uncovered during the design and value analysis reviews.
- Trouble Reporting System--Provides a means of reporting problems, deficiencies, and failures during testing.
- Reliability Prediction and Reporting--Provides a system approach to reliability prediction.
- Reliability of Parts--Provides a means of controlling reliability through selection, application, and control. Parts are selected from preferred parts and supplier lists. The part application is analyzed for stress, environmental, and operational requirements. Part control is maintained through inspection, installation, and in-process testing. In addition, other test methods--durability (in vehicle), fatigue (in laboratory), performance, and metal-lurgical--are used to obtain and maintain part and vehicle reliability.

The system approach to reliability is continued to the component level. System reliability will meet SAE test standards and FMVSS standards. Component reliability predictions are based on the minimum sample size to estimate the population mean for the confidence level required and monitored by the use of a design verification and status form (Figure 7-1).

SYSTEM SAFETY

Many systems are safety or safety-related either by nature or law (FMVSS code). This section does not consider, in detail, safety items such as restraints, seat, lighting, steering systems, suspension system, wheels and tires, and rear drive axle because the reliability and state of the art are well known and readily available to meet the appropriate safety requirements. The unique areas of safety that are discussed below include those with no directly applicable law (FMVSS), at present, because of the type of body construction, propulsion system, energy source, and/or control system.

DESIGN VERIFICATION AND STATUS

SYSTEM <u>08.000</u>		ASSEMBLY <u>Brake Assembly</u>		PROGRAM <u>Electric Car</u>		SUBSYSTEM _____		DATE <u>8/12/76</u>	
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT	
				REQ.	TESTED	REQ.	ACTUAL		
Fluid Test	SAE J 291 J 1702 J 1703 FMVSS 116								
Master Cylinder	SAE J 66			10		P1.0 = .68			
Brake Tubes and Lines	SAE J 1401 J 1043			100		P1.0 = .96			
Brake Lining Test	SAE J 160 J 379,380 J 660,866 J 998,840			100		P1.0 = .96			
Brake Tube and Line	Performance FMVSS 106			100		P1.0 = .96			

ALL BRAKE SYSTEM COMPONENTS AT 95 PERCENT CONFIDENCE LEVEL

Figure 7-1. Sample Design Verification and Status Form

Impact and Roll-Over

Of the major safety considerations (as listed in Table 7-1), one that leads to the most difficult design problem is battery containment during vehicle impact or roll-over. The containment objective of this design is uniquely different from previous design concepts.

The design philosophy of this program is to absorb the impact loads with a lightweight structure and the batteries. Other electric car design philosophies are either to absorb the impact loads entirely with a heavy structure or to disregard the need to absorb the impact loads with any structure. The question of passenger compartment integrity also must be considered in this design.

Another very difficult design problem is acid containment during impact and/or roll-over. Compartment material, cap design and retention, battery design, and battery compartment design will be analyzed, and an extensive development, test, and analysis program will be conducted to solve the acid containment problem. In the event of vehicle roll-over, motor power shutdown will be accomplished in a manner similar to engine shutdown in the conventional powered vehicles. The conventional powered vehicle is automatically shut down during roll-over because of carburetor design. The electric vehicle will require an attitude-sensitive switch to turn off all electric power to prevent electrical hazards (shock or fire) and mechanical hazards (rotating parts). The reliability of the containment design concept, along with all the other safety roll-over and impact considerations, will require considerable effort in test, development, and analysis.

There will be a minimum of one vehicle during the three design phases that will be crash-tested. The results of these crash tests will be used in existing computer programs to predict the response of the vehicle/occupant for specific crash conditions and provide design direction.

Energy Source

Some of the safety considerations that are unique to the electric car are: (1) hydrogen gas, (2) acid, (3) high voltage, (4) overcharging, and (5) overheating. Proper ventilation of the car and in the garage is provided to prevent harmful or combustible levels of hydrogen gas from accumulating. A simple interlock system is used to eliminate potential shock hazard when the battery tray is open or the traction motor covers are removed.

Human factors also are a consideration in this design by the relative location of the positive and negative battery terminals to minimize the potential shock hazard and elimination of any high-voltage circuits in the passenger compartment. The cable routing, insulation, and gage will be designed to prevent any electrical fire. Battery overcharging with the associated problems (heat, deformation, etc.) can be prevented through the use of a voltage regulator. Battery overheating can be eliminated by proper ventilation. The flywheel temperature will be monitored and a housing (similar to a clutch housing) developed to provide the energy containment of the flywheel.

TABLE 7-1
SAFETY HAZARD CONSIDERATIONS

Condition	Problem	Solution
Roll-Over	1. Acid containment	1. Cap design 2. Cap retainers 3. Sealed compartment with one-way valves. Proper vents must be available during normal operation. 4. Material 5. Combination or all of items 1 through 4
	2. Fire hazard	1. Turn power off
	3. Shock hazard	1. Turn power off 2. Covered connections
	4. Mechanical hazard	1. Turn drive train off
	5. Retain batteries	1. Test, development, and analysis
Impact	1. Acid containment	1. Cap design 2. Cap retainers 3. Compartment (body) deformation rate 4. Battery deformation rate 5. Test, development, and analysis 6. Compartment containment 7. Compartment must have proper ventilation 8. Combination of 1 through 7
	2. Fire hazard	1. Cable routing 2. Insulation 3. Turn power off 4. Acid containment 5. Test, development, and analysis 6. Combination of 1 through 5
	3. Retain batteries	1. Mounting 2. Body absorbs part of load 3. Nonmetallic 4. Extensive test, development, and analysis

The electrical vehicle is designed with the standard emergency and service brake system and a regenerative brake system. The regenerative brake system performs two functions:

- Provides an energy recovery source while braking
- Provides an anti-skid brake system for safe, sure, and straight vehicle stops

Testing and Analysis

The design objective for the electrical vehicle is to meet all the appropriate existing FMVSS standards (Table 7-2). A 30-mph crash test (In conjunction with FMVSS 208) and a static roll-over will be performed to evaluate fire hazards from battery and spillage.

The fault tree (Figure 7-2), hazard classification, and failure mode and effects analysis, which include all components and systems having a critical or major failure effect, will be used to evaluate and identify the safety of the vehicle. All failures and their effects will be evaluated for their impact on reliability, warranty, and/or safety versus cost and value required to reduce the probability of failure.

TABLE 7-2

FEDERAL MOTOR VEHICLE SAFETY STANDARDS*

571.101	Standard No. 101; Control location, identification, and illumination.	571.124	Standard No. 124; Accelerator control systems.
571.102	Standard No. 102; Transmission shift lever sequence, starter interlock, and transmission braking effect.	571.125	Standard No. 125; Warning devices.
571.103	Standard No. 103; Windshield defrosting and defogging systems.	571.126	Standard No. 126; Truck-camper loading.
571.104	Standard No. 104; Windshield wiping and washing systems.	571.201	Standard No. 201; Occupant protection in interior impact.
571.105	Standard No. 105; Hydraulic service brake, emergency brake and parking brake systems.	571.202	Standard No. 202; Head restraints.
571.105-75	Standard No. 105-75; Hydraulic brake systems. (Effective Jan. 1, 1976).	571.203	Standard No. 203; Impact protection for the driver from the steering control system.
571.106-74	Standard No. 106-74; Brake hoses	571.204	Standard No. 204; Steering control rearward displacement.
571.107	Standard No. 107; Reflecting surfaces.	571.205	Standard No. 205; Glazing materials.
571.108	Standard No. 108; Lamps, reflective devices, and associated equipment.	571.206	Standard No. 206; Door locks and door retention components.
571.109	Standard No. 109; New pneumatic tires.	571.207	Standard No. 207; Seating systems.
571.110	Standard No. 110; Tire selection and rims.	571.208	Standard No. 208; Occupant crash protection.
571.111	Standard No. 111; Rearview mirrors.	571.209	Standard No. 209; Seat belt assemblies.
571.112	Standard No. 112; Headlamp concealment devices.	571.210	Standard No. 210; Seat belt assembly anchorages.
571.113	Standard No. 113; Hood latch system.	571.211	Standard No. 211; Wheel nuts, wheel discs, and hub caps.
571.114	Standard No. 114; Theft protection.	571.212	Standard No. 212; Windshield mounting.
571.115	Standard No. 115; Vehicle identification number.	571.213	Standard No. 213; Child seating systems.
571.116	Standard No. 116; Motor vehicle brake fluids.	571.214	Standard No. 214; Side door strength.
571.117	Standard No. 117; Retreaded pneumatic tires.	571.215	Standard No. 215; Exterior protection.
571.118	Standard No. 118; Power-operated window systems.	571.216	Standard No. 216; Roof crush resistance-passenger cars.
571.119	Standard No. 119; New pneumatic tires for vehicles other than passenger cars.	571.217	Standard No. 217; Bus window retention and release.
571.121	Standard No. 121; Air Brake Systems.	571.218	Standard No. 218; Motorcycle helmets.
571.122	Standard No. 122; Motorcycle brake systems.	571.219	Standard No. 219; Windshield zone intrusion. (Effective Sept. 1, 1976)
571.123	Standard No. 123; Motorcycle controls and displays.	571.301-75	Standard No. 301-75; Fuel system integrity. (Effective Sept. 1, 1975, with additional requirements, effective Sept. 1, 1976 and Sept. 1, 1977.)
		571.302	Standard No. 302; Flammability of interior materials.

*As of October, 1975.

a. SAFETY FAULT TREE ANALYSIS

S-8922

b. FIRE RELATED FAULT TREE ANALYSIS

Figure 7-2. Safety/Fire Fault Tree Analysis

SECTION 8

ALTERNATE DESIGNS AND OPTIONS

INTRODUCTION

During the design tradeoff analyses conducted as a part of the Phase 1 program (particularly in Task 1), several alternate design concepts were considered. These included the following:

- Alternate transmission concepts
- Four-wheel regenerative braking
- Battery location

The more promising of these are discussed below, and will be considered in more detail in future program phases.

ALTERNATE TRANSMISSION CONCEPTS

Several transmission concepts for hybrid or regenerative power systems have been treated in the literature. Of these, the design presented in Section 4 and three different alternates were analyzed. These systems are depicted in the general schematics of Figures 8-1 through 8-4. The all-electrical concept is shown in Figure 8-1. This system has been developed by AiResearch for the NYTA R-32 railcars and also for the DOT advanced concept train (ACT-1). They have demonstrated substantial energy savings by energy recovery during deceleration, as well as other operational advantages. However, the overall efficiency of the system is too low to be considered for the battery-powered car. Furthermore, since all of the power must be transmitted through both of the motors, the weight and cost become excessive.

A new and interesting variation of this system considered by AiResearch is based on the use of a high-speed, Samarium-cobalt, permanent-magnet generator. This system is shown schematically in Figure 8-2. There are several advantages to this concept. The generator can be very small and lightweight, which permits it to be driven at very high speeds (such as the flywheel speed). This eliminates the gearbox losses entirely. The flywheel and generator housing can be hermetically sealed, which eliminates the vacuum seals. Also, very high generator efficiencies (such as 96 percent) are possible. The principal disadvantage of this system is the need for complex electronics to convert ac to dc in the primary power circuit. Another disadvantage is that the hermetically sealed flywheel bearings must have a lubrication system developed that will operate in the vacuum environment. Also, the traction motor size is large because all of the power must be transmitted electrically.

The baseline system is shown in Figure 8-3. This system was selected because of its comparative simplicity and low weight. All of the components, except for the flywheel, are based on existing technology and can be demonstrated without additional development. Analytical studies have shown that

S-8925

Figure 8-1. All-Electrical Regenerative Power System with Two Dc Motors

Figure 8-3. Electromechanical Regenerative Power System with Two Dc Motors and Differential Planetary

S-8925

Figure 8-2. All-Electrical Regenerative Power System with Permanent-Magnet Motor and Ac-to-Dc Controller

S-11293

Figure 8-4. Mechanical Regenerative Power System with One Dc Motor and Continuously Variable Transmission

high-energy recovery efficiencies can be expected and that the near-term goals of range and performance can be demonstrated.

The system concept (shown in Figure 8-4) is based on an all-mechanical, continuously variable transmission. Such transmissions have been under intensive development for automotive use in recent years. Most of these transmissions suffer from the limited range of speed ratios possible, and all suffer from low efficiencies at low speeds and/or high torque ratios. One new and interesting version is shown in Figure 8-5. This transmission, based on the traction drive principal, has the capability of continuously variable-speed from slow reverse to full forward speed. This transmission has been under development at the Garrett-AiResearch division located at Phoenix, Arizona for the past four years. An existing 60-hp version has demonstrated full-load efficiencies as high as 97 percent. Preliminary studies of the performance potential of this transmission indicate that it may be competitive with the performance of the baseline system. Although it appears to be more complex, it would be lighter in overall weight.

Because of the time limitations of the present study contract, it was not possible to make a detailed comparative study of these various transmission concepts. Therefore, it is recommended that these studies be continued during the next phase of work.

Figure 8-5. Continuously Variable Drive

FOUR-WHEEL REGENERATIVE BRAKING

A possible disadvantage of the regenerative braking concept could be the use of only two wheels for braking. Although the full-power braking effect is modest, as shown in Figure 5-5, Section 5, it is sufficient for all normal driving conditions. However, the minimum coefficient of tire adhesion required to prevent a skid is approximately twice the value required for four-wheel braking. This may be objectionable during wet or icy driving conditions.

The split-power characteristics of the power system permit the design to be arranged for four-wheel braking. This would be accomplished by driving one axle with the traction motor and one axle with the planetary carrier output shaft. The final drive gear ratio would be different for each shaft to compensate for the different shaft speeds. This problem has been analyzed and layouts have been made to examine the relative benefits of the concept. Figure 8-6 is a schematic of one possible arrangement. In this case, the traction motor has been mounted on the front axle because there is more braking effect from the motor than from the carrier shaft.

The disadvantage of the two-wheel braking system may be eliminated by the four-wheel hydraulic braking system. This system includes a control system that will provide the smooth transition from electrical braking to the hydraulic braking system. It is probable that this control system can include a positive anti-skid feature with a minimum of additional complexity. This feature would override the electrical braking command in the event of an incipient skid, and thus minimize the disadvantage of the two-wheel braking concept.

5-8924

Figure 8-6. Split Transmission Schematic for Four-Wheel Drive

BATTERY LOCATION

There are essentially three concepts for the location of the large number of batteries in the design of the vehicle. They can be mounted in a center tunnel running fore and aft through the passenger area, underneath the passenger area, or installed in the front and back compartments. The center tunnel location permits a reasonable weight distribution, but the volume encroaches on the passenger space. Furthermore, the safety aspect of this design may become a problem during severe crashes. The location of the batteries underneath the passenger area requires that the batteries be redesigned for a shorter vertical dimension. This will result in a penalty to the specific energy of the batteries, and therefore cannot be considered until much-improved batteries are available.

The installation in the front and rear compartments was chosen for several reasons. First, separation of the batteries into two unequal parts permits adjustment of the weight distribution to an optimum value. Second, the access for installation and servicing appears to be the best. Third, the crash safety aspects appear to be more manageable in this approach because the structure is designed so that the load path is directed from the bumpers into the heavy weight of the batteries and the battery retention structure. The disadvantage in this design is that the increased polar moment of inertia may have an unfavorable effect on the handling qualities. Since time did not permit a determination of this effect, it should be determined early in the next phase.

SECTION 9

DEVELOPMENT PLAN

The development proposed by AiResearch has been carefully planned to develop the required technological improvements, integrate these improvements into a safe and economical commuter car design, and establish (by test demonstrations) the validity of the design to achieve the intended goals. A detailed discussion of this plan is contained in AiResearch report 76-13311, which was submitted to ERDA in compliance with the requirements of Task 4 of this study program. The comprehensive 54-month development program described in this detailed report is summarized below.

MASTER PLAN

The development program is planned in three major phases that overlap to allow completion of all objectives. As shown in Figure 9-1 (master program schedule), the phases are staggered to allow feedback of test results into subsequent design tasks. The 54-month program contains start and stop milestones for all major program activities, including design and development, program reviews, testing, and fabrication. The development program (beginning with Phase 11A) is based heavily on the Phase 1 results described in this final report, including the selected preliminary design that will be fabricated as a part of Phase 11B activities. The work to be done in each phase of the development program is summarized below.

PROGRAM PHASES

Phase 11A: Power System Demonstrator

The objective of this phase is to demonstrate the regenerative power system concept in an existing electric vehicle of known performance. Thus, an immediate and meaningful comparison of the performance advantages can be made, evaluated, and demonstrated. Two complete power systems will be fabricated--one to be used in laboratory tests and the other to be installed in an ERDA-supplied, modified Renault 12 demonstrator electric vehicle to demonstrate the hybrid concept. The laboratory unit will be subjected to shakedown, performance, and endurance tests using a laboratory flywheel facility to simulate vehicle loads. Performance testing will establish system efficiency, power losses, output torque, output power, and operational characteristics. Performance data will be obtained by simulating various driving modes such as spinup, accelerating, cruising, coasting, braking, and idling. The various driving modes tested in the laboratory will provide the necessary results for changes in power system modeling and help define computer simulation of vehicle operation.

The second power system will be installed in the Renault 12 electric vehicle on loan from ERDA. This vehicle has been extensively tested by ERDA and NASA over numerous driving cycles and modes. Therefore, direct correlation of parameters can be made, and the improved performance of the new design can be

Figure 9-1. Master Program Schedule

evaluated. The power system will be installed in the Renault and initial testing conducted to debug the system and to correlate actual vehicle operation and performance with that obtained in the laboratory test model. Following these correlations, the vehicle will be driven in the Los Angeles area, and complete system data will be recorded. The vehicle will then be delivered to ERDA for further testing and evaluations.

Phase IIB: Experimental Prototype Vehicles

This phase will be a continuation of the Phase I study program reported herein and will comprise detail design and development of experimental vehicles. These experimental prototypes will be designed to meet ERDA goals, and design reviews will be held to approve the detail design. Concurrent with finalizing power system and vehicle design details, support analyses, safety and maintainability analyses, and cost analyses will be conducted. In addition, marketing and manufacturing plans will be prepared to establish economic guidelines to be used throughout the program to ensure that production cost objectives are met.

The power system flywheel, motor/generator, transmission, controller, and ancillary equipment designs will be verified in laboratory development tests. After component interfaces and compatibility have been established by testing, the power system will be assembled and tested both in the laboratory and installed in the experimental vehicles.

Three experimental prototype vehicles will be fabricated. One will be a frame and body shell to be used for critical evaluation of the plastic body structure and crashworthiness of the battery pack/structural subsystem. A nonoperational battery pack and metal mockups of the power system components will be assembled, and the vehicle shell will be subjected to at least a front collision. The other two experimental prototypes will be functioning, driveable vehicles, and will include all items and parts necessary to determine power system and vehicle operation, performance, and handling qualities. One will be used for development testing, and the other will be performance-checked and delivered to ERDA.

The Phase IIB test program will verify (1) the preliminary design concepts (in tests such as material and static crush tests and battery crash tests); (2) body design and manufacturing techniques (in an initial crashworthiness test), and (3) vehicle operation, performance, driveability, handling, and maintainability (in vehicle development tests). These initial test results will provide important data to be used in the next design phase (Phase III). In addition, the acceptability of the basic design will be partially established prior to tests of the development prototype vehicles to be conducted in Phase III. Phase IIB testing also will serve to define specific performance tests considered necessary for the next phases of prototype vehicle testing.

Phase III: Development Prototypes

The Phase III design will be an improved, second-generation prototype vehicle design with increased structural integrity, improved performance, and lower weight. Concepts verified in Phase II will form the basis for the

design. Design improvements or additions required as a result of power system, crash, or development tests also will be incorporated. Five vehicles will be fabricated and used for the following:

- Performance testing and demonstration (one vehicle)
- Safety and vehicle performance tests (one vehicle)
- Crashworthiness tests (one vehicle)
- Delivery to ERDA (two vehicles)

A safety test program will be conducted to establish that the vehicle meets the critical structural, performance, and safety requirements of the design specification. This series of safety tests comprises the most critical of the tests that will be conducted later for final certification, and is planned at this stage to reveal any basic deficiency in the structural design or driveability. The tests are listed in Table 9-1.

TABLE 9-1

SAFETY TEST PROGRAM FOR DEVELOPMENT PROTOTYPES

Test Description	Number of Tests
<ul style="list-style-type: none"> ● Evaluation of Vehicle Design Requirements <ul style="list-style-type: none"> Initial inspection tests Consumer information tests Nonoperating safety evaluation ● Electrical Performance Evaluation <ul style="list-style-type: none"> Maximum speed Grade speed Constant speed range Driving cycle range Acceleration characteristics Energy consumption Battery recharge time Battery cycle life Gradeability characteristics ● Key Vehicle Safety Performance Tests <ul style="list-style-type: none"> Accident avoidance tests Crashworthiness tests Fire hazard tests ● Handling Tests 	<ul style="list-style-type: none"> 1 3 1 1 2 2 4 2 2 6 4 2 20 14 1 10

Evaluation of vehicle design requirements will include initial inspections to ensure that all basic design requirements such as weight parameters and dimensions are within specification and recorded. Electrical performance evaluations will include those key electrical performance parameters that significantly affect overall vehicle performance. These tests will be conducted only at room temperature, whereas the final design confirmation tests to be conducted in Phase IV will include a range of temperatures and charge/discharge rates. Key vehicle safety performance tests will include accident avoidance and fire hazard categories such as those applicable to passenger cars in current Federal Motor Vehicle Safety Standards. Crashworthiness tests will start with a frontal collision. This will be followed by a side impact test if the structure is not severely damaged in areas critical to a second test. A series of handling tests will include steering performance, handling performance, and overturning immunity.

Phase IV: Preproduction Prototypes

The prototype vehicle will be designed to its final configuration during the Phase IV design effort. The main thrust of the design effort will be producibility. The parts and equipment of this prototype vehicle will be designed in a configuration suitable for pilot production. Recommendations from the automotive industry will be solicited for incorporation into this final design phase. Concurrent with the design, the marketing and manufacturing plans updated in Phase III will be finalized. Crash, life-cycle cost, safety, and maintainability analyses also will be used in finalizing the prototype vehicle design to provide the best combination of running efficiency, appearance, passenger comfort, low cost, maintainability, and producibility. Seven vehicles will be fabricated for the following purposes:

- Vehicle performance evaluation and demonstration (one vehicle)
- Crashworthiness and FMVSS handling tests (three vehicles)
- Delivery to ERDA (three vehicles)

The safety and handling test program conducted in Phase IV will be based on Federal Motor Vehicle Safety Standards (FMVSS). Current test requirements for electric vehicles (SAE J227A) and conventional passenger vehicles will also serve as a basis for the selection of a test program. The proposed test program will be divided into four major categories:

- Evaluation of vehicle design requirements
- Electrical performance evaluation
- Passenger vehicle safety performance
- Handling tests

Other possible tests will be aggressivity and endurance tests. Testing will be conducted over a range of temperatures and charge/discharge rates.

At the completion of the preproduction phase, the vehicles will have demonstrated the performance, safety, and economic goals established by ERDA. The vehicles will have been modified to ensure ultimate producibility and will be ready for production at a rate desired by ERDA.

SECTION 10

CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

Based on the analysis and design activities conducted under the Phase 1 study effort, a preliminary (baseline) design for an electric-powered passenger vehicle has been defined that will effectively meet ERDA near-term performance, weight, cost, and safety objectives. The baseline design reflects the following study conclusions:

- (a) A regenerative power system is required to meet the near-term goals of the electric vehicle program.
- (b) A lightweight vehicle structure and body is required to meet the cost and energy consumption goals of the program.
- (c) The baseline regenerative power system will substantially increase the acceleration performance of the vehicle, leading directly to greater consumer acceptance.
- (d) Increased cycle life of the batteries is required to ensure attainment of the predicted life-cycle cost of vehicle operation.
- (e) The required costs are attainable, provided the battery cost penalty is minimized.

RECOMMENDATIONS

Recommendations for future development activities include:

- (a) A 54-month development program (as discussed in Section 9) is recommended to develop and demonstrate the new, optimized electric vehicle and power system technology.
- (b) Close coordination between the advanced battery development contractors and the vehicle designers is strongly recommended to ensure optimum utilization of the potential improvements in this critical area.

Additional areas for analysis include the design alternates discussed in Section 8 and optimizing body design for consumer acceptance (as described in Section 3).

Task 1 Tradeoff Studies

Prepared by

**B.H. Rowlett,
Program Manager**

Prepared for

**The Energy Research and Development Administration
Oakland, California**

Contract E(04-3)-1213

**AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA**

FOREWORD

This tradeoff study is submitted by the AiResearch Manufacturing Company of California, a division of The Garrett Corporation, in compliance with the Task 1 requirements of the Phase I study. The electric powered passenger vehicle design study program, Contract E(04-3)-1213, is an Energy Research and Development Administration effort aimed at developing a preliminary design of an energy efficient electric commuter car.

The methodology established herein has been applied to the development of the design presented in the Phase I Final Report. Further definition and refinement of the baseline power system subsequent to the completion of this tradeoff study has resulted in some minor deviation between the performance reported here and in the Final Report. In all cases, however, the tradeoff study methodology and conclusions have been retained.

SECTION 1

INTRODUCTION AND SUMMARY

INTRODUCTION

This report presents the results of the Task I studies conducted under Contract E(04-3)-1213, Electric Vehicle Design Study, in accordance with the contract specification and as further defined in AiResearch report 76-12948, Program Plan. The tradeoff studies discussed in this document were conducted to establish a valid design base so that more detailed design comparisons can be made during subsequent program tasks. Thus, establishment of accurate and detailed computer models to facilitate subsequent design studies was emphasized.

SUMMARY

The baseline vehicle and power system design established during Task I meets the performance goals of the program. The baseline design is tentative and will be compared with selected design variations during the next program task. Three power system computer simulation programs were prepared for the basic vehicle tradeoff studies. These programs simulate the performance of the power system and vehicle over different types of driving conditions such as maximum power acceleration, deceleration, city driving cycles, and hill climbing. The use of these simulation programs permits accurate determination of (1) the benefits of the unique hybrid power system, (2) the total energy required for the suburban city driving cycle, and (3) the extremes of the operating envelopes of the components so that component design options can be defined and studied. The baseline power system and performance are discussed in Section 2; computer simulation programs and examples of their use are presented in Section 3.

Component design tradeoff studies were conducted, including sensitivity studies to show the criticality of the various losses and unknowns in the analytical models. These are discussed in Section 4. Also, preliminary vehicle layout studies were performed to determine the best locations of the power system and the batteries. Three basic design options are identified for further study in Section 5.

Economic studies were initiated using analytical models to establish the complete vehicle weight and cost breakdowns. Expertise from various consulting firms with extensive experience in establishing automotive manufacturing and operating costs was used as a basis. Initial results are given in Section 6. Preliminary reliability and safety studies were completed, and maintainability and safety certification criteria established. These studies and the formats established for use in subsequent program tasks are described in Section 7. A sample design verification and status report is contained in the Appendix.

CONCLUSIONS

Although the objective of the Task I studies was to establish the design base for subsequent studies, some conclusions can be made based on the results

of Task 1. The detailed analysis of the power system has verified the feasibility of the system and of the performance expectations. Also, the feasibility of energy recovery from regenerative braking has been confirmed. The sensitivity analysis of the power system shows that sufficient margin for unknown design variables is provided. The preliminary economic analysis indicates that the most difficult objective of the study will be the selection of the cost and weight relationships which are required to achieve the overall cost objectives. Accordingly, this portion of the study will be emphasized.

SECTION 2

POWER SYSTEM BASELINE DESCRIPTION AND PERFORMANCE

A baseline power system that achieves all performance goals of the contract and has very low energy requirements has been selected for future work during extensive tradeoff studies. The multiple objectives of the studies conducted during the Task I effort were:

- (a) To show, by detailed analysis, the validity of the unique power system and the practicality of its installation in a commuter-type vehicle
- (b) To evaluate the options in component sizes and assembly arrangements in sufficient depth to initiate the vehicle layout studies
- (c) To determine the system operational characteristics and the overall vehicle performance
- (d) To determine the energy requirements and the optimum battery selection for the near-term vehicle design
- (e) To determine the sensitivity of the system to component variables, technology developments, and costs

To accomplish these objectives and to achieve the level of performance desired, the following tradeoff studies were conducted:

- (a) Size, capacity, and material selection for the flywheel
- (b) Gross vehicle weight vs battery weight and empty weight
- (c) Size, speed, and power ratings of the motor and generator
- (d) Planetary gear ratios and arrangements
- (e) Selection of the battery type
- (f) Power system and battery installation

Results are described in subsequent sections of this report.

The basic power system is shown schematically in Figure 2-1. Preliminary layout sketches of two versions of the power system are shown in Drawings SK71641 and SK71642. As shown, the flywheel shaft is connected to the sun gear of the differential planetary transmission. The generator is geared to the ring gear of the planetary and the carrier output is connected to the final drive differential gearbox. The traction motor is geared to the carrier shaft. The flywheel housing will be held to a low pressure of about 0.050 mm by a small motor-driven vacuum pump and a molecular pump inside the housing and the flywheel disc.

S-8927

Figure 2-1. Baseline Schematic Power System

11-2-3

SILENT CHAIN
DRIVE

FWD

GENERATOR

MOTOR

FLYWHEEL
HOUSING

PLAN VIEW

AIRESEARCH MANUFACTURING COMPANY OF CALIFORNIA A DIVISION OF THE BARRETT CORPORATION TORRANCE, CALIFORNIA		
ELECTRIC VEHICLE TRANSMISSION (REAR WHEEL DRIVE)		
SIZE	CODE IDENT NO	DWG NO
D	70210	SK-71641
SCALE 1/2		SHEET 1 OF 2

11-2-4

SIDE VIEW

11-2-5

AIRSEARCH MANUFACTURING COMPANY OF CALIFORNIA A DIVISION OF THE GARRETT CORPORATION TORRANCE, CALIFORNIA
	ELECTRIC VEHICLE TRANSMISSION (FRONT WHEEL DRIVE)

11-2-6

BODY
PANEL

TIRE

AXLE

FLOOR LEVEL OF
PASSENGER COMPARTMENT

ROAD

FRONT VIEW

VEHICLE CENTERLINE

10 INCHES

ENVELOPE OF ENCLOSURE
PROJECTING INTO
PASSENGER COMPARTMENT

GENERATOR

FLYWHEEL HOUSING O.D.

FLYWHEEL HOUSING
FLANGE O.D.

PLANETARY GEARSET
HOUSING

MOTOR

DIFFERENTIAL

The generator is used to spin up the flywheel to its maximum speed of about 25,000 rpm. During vehicle acceleration, the generator acts to decelerate the flywheel, thereby extracting energy. The portion of the energy that is supplied to the generator is transmitted electrically to produce power in the traction motor. The remaining portion of the energy from the flywheel is transmitted out through the carrier shaft and directly to the final drive gearbox.

The armature circuit of the generator is connected directly to the armature of the motor. All control is accomplished by the two field circuits. Varying the current in the two field circuits determines the amount of armature current and the direction of the current.

The major characteristics of the power system are summarized in Table 2-1. These values are the result of the Task 1 studies, and are subject to change as the study continues.

The torque and power characteristics are unique and are shown in Figures 2-2 and 2-3. The output torque is shown as the sum of two components, one from the traction motor and one from the planet carrier of the planetary gearbox. Figure 2-3 shows the total output power as a function of vehicle velocity. The data are shown for the full power transient case where all of the power is extracted from the flywheel and no battery current is supplied. The acceleration of the vehicle is shown in Figure 2-4, and the electrical braking deceleration capability is shown by Figure 2-5. As shown, the electrical braking is substantial down to a speed of about 6 mph. Hydraulic friction brakes will be used to back up the electrical braking.

The energy capacity of the flywheel provides for substantial hill climbing before needing excessive current from the battery. Figure 2-6 shows the capability of the power system to augment the normal 60-amp current to the motor during uphill grades at various speeds. For distances greater than those shown, the current limiter is overridden and the system will operate as a conventional motor-powered drive system. During downhill grades, the reverse will occur and energy is recovered into the flywheel.

The operational limitations with respect to the uphill and downhill grades are obvious. The energy can be completely drained from the flywheel if its maximum grade capability is used, and no more performance can be taken from it until it is recharged. On downhill grades, if the maximum energy capacity of the flywheel is reached, then no more recovery is possible and the friction brakes would be used. Thus, the grade condition is the critical design sizing condition for the flywheel.

The energy requirements of the entire system are summarized in Table 2-2 for the suburban driving cycle. The column labeled Required Energy represents the sum of all losses in the regenerative system where kinetic energy is conserved. The column labeled Ideal Energy represents an ideal system with no component losses, but where kinetic energy is not conserved. As shown, the ideal system has no road load penalty during the last three segments because the battery system can be turned off. The regenerative system is still on, so the losses continue. However, the regenerative system

TABLE 2-1

MAJOR CHARACTERISTICS OF THE PRELIMINARY
BASELINE SYSTEM

<u>Vehicle</u>	
Vehicle gross weight, lb	3000
Vehicle inertial weight, lb	3150
Rolling resistance coefficient	0.010
Vehicle effective drag area, $C_D A$, sq ft	7
<u>Propulsion System</u>	
Flywheel	
Total energy, w-hr	1000
Available energy, w-hr	750
Maximum speed, rpm	25,000
Loss at maximum speed, w	380
Total assembly weight, lb	65
Traction motor	
Rated power, hp	13.7
Maximum speed, rpm	7000
Nominal efficiency, percent	88
Weight, lb	80
Generator	
Rated power, hp	13.7
Maximum speed, rpm	7000
Nominal efficiency, percent	88
Weight, lb	80
Gearbox weight, lb	15
Controller weight, lb	10
Battery pack weight, lb	874
Total propulsion system weight, lb	250

Figure 2-2. Torque Characteristics of the Drive System

Figure 2-3. Power Characteristics of the Drive System

Figure 2-4. Full Power Vehicle Acceleration

Figure 2-5. Maximum Electrical Deceleration Rate as a Function of Vehicle Speed

11-2-10

S-8921

Figure 2-6. Distance and Grade Capability from Surplus Flywheel Energy with Steady-State Battery Drain of 60 amp

TABLE 2-2

ENERGY REQUIREMENTS FOR THE J227a(d)
SUBURBAN DRIVING CYCLE
GROSS WEIGHT = 3000 LB

Segment	Mode	Time, sec	Required Energy, w-hr	Ideal Energy, w-hr
1	ACCEL	28	44.42	98.28
2	CRUISE	50	111.91	87.78
3	COAST	10	19.59	0
4	BRAKING	9	19.08	0
5	IDLE	25	8.23	0
Total (0.967 miles)		122	203.25	186.06
Total per mile			209.8	192.41
<p>Overall efficiency = $\frac{192.41}{209.8} = 0.915$</p> <p>Battery discharge rate = $\frac{209.8 \times 28.53 \text{ mph}}{108 \text{ volts}} = 55.43 \text{ amp}$</p> <p>Battery capacity required = $209.8 \times 75 \text{ mi} = 15,735 \text{ w-hr}$</p> <p>Weight of lead acid batteries = $\frac{209.8 \times 75 \text{ mi}}{18 \text{ w-hr/lb}} = 874 \text{ lb}$</p>				

total losses are only slightly more (8 percent) than for the ideal, no-loss system. The calculations at the bottom of the table show how the total energy requirement is used to compute the overall efficiency and the required battery capacity and weight.

SECTION 3

COMPUTER SIMULATION STUDIES

INTRODUCTION

Particular emphasis was placed on development of the system simulation model because of its importance in determining how the system operates and how component performance affects overall performance. Detailed analytical models were developed for each of the components, and a digital computer program for total system simulation was completed. This program was used to conduct tradeoff studies for each of the many variables within the system. The component models and program used to evaluate and predict the system performance variations are discussed below. The section is organized as follows:

- Simulation models and program description

- Simulation model inputs

- Simulation model outputs

- System performance studies

SIMULATION MODELS AND PROGRAM DESCRIPTION

The flywheel propulsion system simulation equates transient changes in flywheel-generator-motor kinetic energy to changes in vehicle kinetic energy and losses due to road loads and mechanical friction. The flywheel, motor, and generator shafts are connected by a differential planetary gear set. The rate of change of flywheel kinetic energy is controlled by the torque applied at the ring gear by the generator. The system losses are computed on a component basis as a function of speed, torque, current, or other appropriate parameters so that realistic system performance can be predicted.

Three system models were developed to provide the required flexibility in simulating system operation over a given driving condition: (1) ACCEL, (2) SUBURB, and (3) UPHILL. ACCEL simulates maximum system performance during full throttle acceleration to a given maximum speed followed by full power regenerative braking. For this mode, the armature current is fixed at the design maximum, the power output of the generator increases to respond to the increasing speed (and power) of the motor, and the motor and generator field excitations are adjusted by the controller to maintain a constant armature voltage. Upon reaching the prescribed maximum vehicle speed, electrical braking is initiated. The controller reduces the generator field and increases the motor field sufficiently to cause a complete reversal of the armature current. This maximum braking operation is continued until the motor (and vehicle) speed is reduced to an armature voltage value that drops below a minimum for significant braking effect, which occurs at about 5 to 6 mph. Complete system and component performance is calculated for each time interval (usually 1 sec) throughout the entire operation, and includes all component speeds and accelerations, torque forces, and system losses. In addition, the computer provides

plots of some of the major variables during acceleration and deceleration to facilitate comparison and evaluation of predicted performance.

Simulation program ACCEL is used to study maximum system power capability as a function of the component sizes and efficiencies. The output shows the effect of the gear ratios used; the relation between the selected generator, flywheel, and motor speeds; and the resulting vehicle acceleration and deceleration performance. Also shown are the overall efficiency and the split between mechanically and electrically transmitted power.

The second program--SUBURB--simulates vehicle operation over any arbitrary driving cycle comprising any number of specified segments. It is presently set up to simulate the five segments of the SAE J227A(d) suburban driving cycle. The power requirement for the generator is determined by the vehicle acceleration prescribed in the driving cycle; the armature current is determined by the torque necessary to achieve this acceleration; and the motor and generator field currents respond to changes in the armature speeds. During the coast and idle modes, both field currents are set to zero so that only the mechanical losses (brushes, bearings, seals) are computed. Energy consumption is itemized for each component in the system over each of the successive segments of the cycle. The total of all segments is the energy requirement that must be supplied by the battery for the entire cycle. Simulation program SUBURB is used to (1) compute the range of the vehicle from the total summation of the energy requirements for the cycle, (2) examine the individual losses in the system, and (3) show the sensitivity of the resultant range to each of the system component losses.

The third program simulation--UPHILL-- simulates the operation of climbing up a grade at a given speed. This program logic is the same as SUBURB. In use, three segments are simulated: (1) level ground acceleration to a given speed, (2) a short transition to the specified grade, and (3) vehicle steady-state speed up the grade using flywheel stored energy to maintain that speed. This simulation shows the steady deceleration of the flywheel and the generator, and is terminated when the generator no longer maintains a sufficient voltage to overcome the system losses. This occurs at about 50 percent flywheel speed, signifying that about 75 percent of the total energy stored in the flywheel has been extracted.

SIMULATION MODEL INPUTS

The model input array permits the user to describe the vehicle weight and road resistance, and to specify the characteristics of the propulsion system components. The inertial properties of the vehicle are described by gross weight, frontal area, drag coefficient, and tire-wheel bearing performance. The basic propulsion system configuration is described by the energy capacity of the flywheel and the power of the motor-generator. The flywheel rotational moment of inertia and maximum speed fix the energy capacity. A detailed description of the size and electromagnetic properties of both the motor and the generator is used in the program to calculate performance as a function of armature and field current, armature speed, and torque.

The power developed by the motor or generator is the product of the output torque times the armature speed, minus the losses due to friction and resistance heating. The output torque, in turn, is the product of the armature current and the internal voltage constant. Thus, the input description must include the saturation curve, giving the internal voltage as a function of field strength, and all motor design details that affect motor losses.

The drive train is specified by the gear ratios of the final drive and the planetary differentials. The losses at the final drive and the planetary are given as a function of the speed and output torque of the rear axle and the flywheel, respectively. The losses at the motor and generator gears are also given as functions of the respective speed and torque.

Loss Models

The power dissipation for each component in the system is calculated for the most part on the basis of well-documented empirical equations. All losses are expressed in terms of watts of power dissipated.

Aerodynamic Drag

The aerodynamic drag is calculated using Newton's resistance law for form drag as follows:

$$F_{aero} = C_D A \rho V^2 / 2$$

where C_D is the drag coefficient available from test data for various compact cars (see Table 3-1) and

A is the frontal area

ρ is the density of air

V is the velocity of the vehicle

Rolling Resistance

The rolling resistance calculation utilizes the empirical expression for tire and wheel bearing losses approved for use in federally funded automotive performance studies.* The expression is:

$$F_{RL} = \mu W_V [1 + (1.4 \times 10^{-3})V + (1.2 \times 10^{-5})V^2]$$

where W_V is the gross weight of the vehicle

V is the velocity and

μ is the tire rolling resistance coefficient

* See Prototype Vehicle Specification, Compact Passenger Car, Alternative Automotive Power Systems Division, Environmental Protection Agency, Feb. 1974.

TABLE 3-1

AERODYNAMIC COMPARISON OF SEVERAL COMPACT CARS

Vehicle	Frontal Area (A), sq ft	Drag Coefficient (C _D)	C _D A, sq ft	Length, in.
Alfa Romeo Alfetta sedan	19.9	0.42	8.4	177
Chevy Vega sedan	18.5	0.52	9.6	176
Citroen Ds-21 sedan	22.7	0.33	7.5	191
Citroen CX sedan	20.0 est	0.29	5.8 est	182
Citroen SM coupe	20.9	0.27	5.6	194
Ford Capri II.	18.7	0.45	8.42	175
Ford Capri MA (experimental)	18.7	0.40	7.48	175
Ford Pinto sedan	19.3	0.64	12.4	169
VW 1500 Beetle	20.0	0.41	8.3	163
Volvo 1800 ES	18.7	0.44	8.2	177
SPS Cal car	20.2	0.45	9.1	172
Target AiResearch	19.0	0.37	7.0	162

The force due to aerodynamic drag and rolling resistance is applied as a torque at the rear axle. This power is transmitted through the final drive differential to the carrier shaft.

Motor and Generator Losses

The motor and generator losses comprise the difference between the power into the generator and the power out of the motor. Thus, the power into the generator minus the power out of the motor equals power loss due to:

Motor armature

Generator armature

Motor field

Generator field

The power loss analysis utilized the computer package developed by AiResearch for the design of specialized motors for the aerospace and transportation industries.

1. Motor and Generator Armature Power Losses

Power lost in the motor and generator armatures includes copper, brush, windage, and stray load losses; also included are iron losses, which comprise hysteresis and eddy losses. The following equations are used to calculate these losses.

Copper-- $P_{cu} = I^2 R$, where I is the armature current and R is the resistance of the armature.

Brush Voltage Drop-- $P_{BDP} = I \cdot (0.25 + 0.0025 I/A_B)$, where A_B is the area of the brushes and I is the armature current.

Brush Friction-- $P_{BF} = N_B \cdot A_B \cdot P_{sp} \cdot C_F \cdot D_{COM} \cdot N_A / 168.9$, where N_B is the number of brushes, P_{sp} is the spring pressure, C_F is the coefficient of friction of brushes, D_{COM} is the diameter of the commutator, and N_A is the rotation speed of the armature.

Windage-- $P_w = (4.83 \times 10^{-5}) (\pi D_A N_A)^2$, where D_A is the diameter of the armature, and N_A is the rotation speed of the armature.

Stray Load-- $P_{SL} = 0.01 P_S$, where P_S is the shaft power.

Hysteresis Loss-- $P_H = \left(\frac{W}{10^7 D} \right) f \eta B^X$, where W is the weight of the core, f is

the frequency, d is the thickness of core lamination, B is the flux density, D is the density of core material, X is the hysteresis loss exponent for B , and η is the hysteresis loss coefficient.

$$\text{Eddy Loss}--P_E = \left(\frac{1.645d^2w}{16D} \right) f^2 B^2 \lambda, \text{ where the terms are as defined under hysteresis loss above and } \lambda \text{ is the eddy loss coefficient.}$$

Thus, power loss in each of the armatures is the summation of P_{cu} , P_{BDP} , P_{BF} , P_W , P_{SL} , P_H , and P_E .

2. Controller Power Losses

For this initial study, a constant loss of 200 w was assumed.

3. Motor and Generator Field Losses

The field losses are limited to copper losses as follows:

$$P_{GF} = I_{GF}^2 R_{GF}$$

$$P_{MF} = I_{MF}^2 R_{MF}$$

where I_{GF} and I_{MF} are the field currents to the generator and motor, respectively, and R_{GF} and R_{MF} are resistances of the field coils.

Motor and Generator Gears

The expressions for power losses at the motor and generator gears are adapted from approximate available test data for power and windage dependent losses. Equations are

$$P_{GL} = [(3 \times 10^{-6}) T_G N_G + N_G^2 \times 10^{-10}], \text{ hp}$$

$$P_{ML} = [(2 \times 10^{-6}) T_M N_M + N_M^2 \times 10^{-10}], \text{ hp}$$

where T_G and T_M are the torque of the generator and motor, respectively, and N_G and N_M are the respective armature speeds

This model implies an approximate 2 percent power loss over the usable range of speed and torque.

Planetary Differential

The loss expression for the planetary differential gear set is similar to the power and windage dependent equation used for the generator and motor gears. Test data on a dry sump planetary gear set with a rotating ring gear suitable for predicting losses on the proposed set is not available. The current expression is based on an adjusted wet sump efficiency curve as follows:

$$P_{PL} = [(4 \times 10^{-6}) T_F N_F + N_F^2 \times 10^{-9}], \text{ hp}$$

where

T_F is the torque of the flywheel, and

N_F is the flywheel speed

This equation implies gear loss at idling of about 0.62 hp, increasing to over 2 hp at the high-power acceleration conditions.

Final Drive Differential

The power loss model for the final drive differential is consistent with industry standards for straight bevel geared differentials of modest gear ratios. The equation is

$$P_{FD} = [(7.6 \times 10^{-6})T_A N_A + (5.1 \times 10^{-9})N_A^2], \text{ hp}$$

where

T_A is the torque at the axle, and

N_A is the speed of the axle

This equation implies an approximate 4 percent power loss over the operating spectrum.

Flywheel Losses

The full speed flywheel power losses are estimated from the test data of past AiResearch flywheel programs. The run-down time results indicate that the power losses decrease as the square of the ratio of flywheel speed to maximum flywheel speed. Thus:

$$P_{FL} = 380 \left(\frac{N_F}{N_{FMAX}} \right)^2$$

where 380 is the power loss in watts at maximum flywheel speed (see Table 3-2)

N_F is flywheel speed, and

N_{FMAX} is maximum flywheel speed

TABLE 3-2

FLYWHEEL ASSEMBLY FRICTION LOSSES (AT 100 PERCENT RPM)

Item	Loss, watts
Windage	100
Rotor bearings (2)	120
Rotor seal	80
Lube pump and 2-stage vacuum pump	80
Total	380

SIMULATION MODEL OUTPUTS

There are several forms of the output for use in the studies. First, for all three simulation programs, a complete printout lists all of the significant parameters that describe the transient operation of each of the components for each interval of time (normally, 1 sec). Second, certain parameters can be automatically plotted as a function of the vehicle speed. Examples are those shown by Figures 2-2 through 2-4 in Section 2. Third, for the suburban driving cycle simulation, a summary printout of all losses in the cycle by category and by cycle segment is provided. The total energy tabulated is used for the calculation of the vehicle range (or the weight of batteries for a given range). The summary tabulation is especially useful for comparison of the relative efficiencies of different systems or components. It is also useful for the calculation of overall energy recovery efficiency during the regenerative braking segment.

An example is provided in Table 3-3, which is the summary tabulation for the baseline system operated over the J227A(d) cycle. Three calculations that are examples of the use of this summary are battery weight, system efficiency, and recovery efficiency. These are summarized below.

Battery Weight

The calculation of the required battery weight is as follows:

$$W_B = \frac{\text{Energy required/mile} \times \text{miles}}{\text{Battery energy density}} = \frac{209.8 \times 75}{18} = 874 \text{ lb}$$

Overall System Efficiency

Overall system efficiency is determined as follows.

$$\begin{aligned} \text{Eff}_{ic} &= \frac{\text{Ideal energy for acceleration + cruise}}{\text{Total energy required/cycle}} = \\ &= \frac{\text{Ideal kinetic energy + road load (for segments 1 and 2)}}{\text{Total energy required/cycle}} = \\ &= \frac{80.28 + 18.0 + 87.73}{204.69} = 0.91 \end{aligned}$$

Energy Recovery Efficiency

The energy recovery efficiency is demonstrated below for the 9-sec braking segment of the SAE suburban cycle as calculated by the program SUBURB. During regenerative braking, the kinetic energy of the vehicle is converted into flywheel energy and waste heat. The energy balance is the change in vehicle kinetic energy, which equals the change in flywheel/generator kinetic

TABLE 3-3

SUMMARY OF ENERGY LOSSES
FOR DRIVING CYCLE IN W-HR

Segment	Flwl-Gn KE	Vehicle KE	Road Load	Motor Armature	Generator Armature	Cntrlr	Field Losses	Final Drive	Motor Gear	Gen Gear	Flwl	Plnty Dif	Total Losses
1	-124.70	80.28	18.00	5.41	6.37	1.61	3.15	2.70	0.79	1.45	2.83	6.08	44.42
2	-111.91	-0.00	87.78	4.64	4.08	2.78	0.83	0.02	0.51	0.93	4.32	7.65	111.91
3	-3.70	-15.89	15.60	1.23	1.11	0.56	0.02	0.93	0.10	0.12	0.80	1.06	19.59
4	45.30	-64.38	4.45	3.58	3.42	0.50	1.22	1.86	0.44	0.54	0.75	1.89	19.08
5	-8.23	0.00	0.00	0.01	1.81	1.39	0.00	0.00	0.00	0.07	2.11	2.77	8.23
Totals	-203.24	-0.00	125.83	14.87	16.80	6.83	5.22	5.51	1.84	3.12	10.82	19.46	203.24
Total losses/mile = 209.80													

energy plus energy lost to road load, motor armature losses, generator armature, controller, motor and generator fields, motor gear, generator gear, flywheel, planetary differential, and final drive differential. The efficiency of the energy recovery operation thus can be written as:

$$\eta_R = \frac{\text{Change in vehicle kinetic energy} - \Sigma \text{ energy losses}}{\text{Change in vehicle kinetic energy}}$$

The energy summary at the end of the SUBURB output tabulates the energy balance for each driving segment (see Table 3-3). In the example shown for the design baseline, the regenerative braking operation occurs during segment four, and the recovery efficiency is calculated to be:

$$\eta_R = \frac{64.38 - (4.55 + 3.58 + 3.42 + .50 + 1.22 + 1.86 + .44 + .54 + 1.89)}{64.38}$$

$$\eta_R = 0.71$$

The recovery efficiency will vary with the distance traveled during braking and the deceleration rate. The energy dissipated due to road load is the sum of the aero and rolling forces integrated over the braking distance. The balance of the dissipated energy consists of the remaining power losses integrated over the braking time.

SYSTEM PERFORMANCE STUDIES

During the initial phase of the study, the vehicle performance simulations were used to focus on propulsion system tradeoffs and component sensitivity studies. The tradeoff investigations resulted in the identification and description of the critical parameters for each aspect of vehicle performance. The sensitivity analysis isolated the effect on overall efficiency of each loss element in the system.

Tradeoffs on Propulsion System

1. Evaluation Criteria

The objective of the propulsion system design tradeoff was to optimize the acceleration time, climbing distance, and driving cycle efficiency for various generator-motor, flywheel, gearbox, and final drive configurations.

2. Generator/Motor

The generator/motor power rating impacts acceleration and driving cycle efficiency, and strongly affects propulsion system weight and vehicle cost. The simulator permits moderate plus or minus scaling of a given motor design after it has been modeled. The power rating tradeoffs were conducted using a constant vehicle weight and final drive ratio. The typical variation of full power acceleration with power rating is shown in Figure 3-1. A 50-percent increase in motor power provides a 40 percent improvement in 0 to 60 acceleration time, while consuming the same amount of flywheel energy. In terms of suburban driving cycle efficiency, Figure 3-2 shows that the watt-hours

S-8911

Figure 3-1. Vehicle Acceleration as a Function of Motor Size

S-8914

Figure 3-2. Energy Required as a Function of Motor Size

consumed per mile decrease by 3 percent for the same 50 percent increase in power rating. The improved efficiency results from a smaller power loss in the larger armature operating at the same power level, as dictated by the driving cycle.

3. Flywheel

The effects of flywheel speed, torques, and losses were evaluated during various simulation runs on the computer. Various operational requirements were confirmed during these runs, such as the ability to efficiently extract flywheel energy at speeds as low as 50 percent of the design maximum. The flywheel capacity and size, however, were determined separately (as described in Section 4).

4. Final Drive Ratio

The final drive gear ratio affects all three criteria of vehicle performance. Higher final drive ratios produce higher acceleration rates, but degrade the uphill distance capability, as shown in Figures 3-3 and 3-4. Energy extraction from the flywheel is caused by a speed reduction (torque) in the generator. High final drive ratios produce higher vehicle accelerations by large decreases in the speed of the generator. Hill climbing also requires energy extraction from the flywheel, which must be caused by a further reduction in the speed of the generator. Therefore, for a given minimum speed of the generator, there is a balance between the speed reduction to accelerate to cruise and the speed reduction that must be reserved for hill climbing. Varying the final drive ratio accomplishes this balance. Obviously, a two speed final drive would improve the acceleration while maintaining hill climbing ability, or for the same performance, the sizes of the motor and generator could be reduced.

Available torque at the wheels as a function of the final drive gear ratio is shown in Figure 3-5 for the same cases shown in Figures 3-3 and 3-4. The effect of the final drive gear ratio on energy economy is very slight. Figure 3-6 shows this lack of sensitivity over the range of gear ratios studied.

Loss Sensitivity Studies

The simulation programs are useful in examining the effect of isolated variations in the component performance or loss factors. Some of the loss factors are somewhat unknown and have been estimated or scaled from related data. Because of the critical nature of some of these losses, it is prudent to examine separately the effect of these losses, especially on the critical performance objectives such as range or the required weight of the batteries to achieve the range.

The sensitivity study was therefore performed with the aid of the SUBURB program, where each of the losses can be perturbed separately to observe the resulting effect and energy requirements. Figures 3-7 through 3-15 show the change in the required energy or the change in the weight of the battery pack to maintain the required range for changes in each of the losses in the system. As shown in the various curves, the only large individual effect is for road load losses. This would be expected because the road load accounts for 59 percent of the total energy for the driving cycle.

Figure 3-3. Effect of Final Drive Gear Ratio on Acceleration

Figure 3-4. Hill Climbing Capability as a Function of Final Drive Gear Ratio

S-8916

Figure 3-5. Total Output Torque at the Wheel Axle as a Function of Final Drive Gear Ratio

Figure 3-6. Energy Requirements as a Function of Final Drive Gear Ratio (Suburban Driving Cycle)

Figure 3-7. Sensitivity of Total Energy Requirements to Flywheel Losses (Suburban Driving Cycle)

Figure 3-8. Sensitivity of Total Energy Requirements to Change in Motor Losses

Figure 3-9. Sensitivity of Total Energy Requirements to Change in Generator Losses

Figure 3-10. Sensitivity of Total Energy Requirements to Change in Planetary Gear Losses

Figure 3-11. Sensitivity of Total Energy Requirements to Change in Final Drive Losses

Figure 3-12. Sensitivity of Total Energy Requirements to Change in Motor Gear Losses.

Figure 3-13. Sensitivity of Total Energy Requirements to Change in Generator Gear Losses

Figure 3-14. Sensitivity of Total Energy Requirements to Change in Aero Drag Losses

Figure 3-15. Sensitivity of Total Energy Losses to Change in Rolling Resistance

SECTION 4

COMPONENT AND POWER SYSTEM TRADEOFF STUDIES

This section describes the various design tradeoffs or options with respect to the major components of the power system. This includes the flywheel, the motor and generator, the gearbox, the battery pack, and the controller.

FLYWHEEL DESIGN TRADEOFFS

The primary variables for the design of the flywheel rotor are material choice, shape, peripheral tip speed, and rotational speed. Other design restraints enter into the design of the flywheel assembly, such as the vacuum environment, containment housing design, and the operational environment (such as the cycle life and the generation of gyroscopic loads induced by external angular velocities of the vehicle).

Steel disc flywheels are currently in use in railcar propulsion systems and in some experimental cars. The successes of these experiences has stimulated the development of higher energy density wheels using composite materials. A cylindrical composite flywheel will be very much lighter, particularly considering the weight required for the containment shroud.

The proposed flywheel design is based on a new concept which has been proven in full-scale laboratory testing, including 1000 full-speed deep discharge cycles. The laboratory testing was done with a flywheel made from E-glass and epoxy material and was cycled to 1920 ft/sec. This flywheel had a weight of 11.08 lb and stored 0.242 kwh of energy. The flywheel was made by conventional winding techniques, and no special problems or failures were encountered in the entire test program.

During the past two years, AiResearch has been developing flywheels with still higher strength materials such as S-glass and Kevlar. The use of S-glass requires the same winding and processing techniques as used with the successful E-glass flywheel, and the material properties are well known. However, the higher strength of S-glass permits it to be designed for operation at 2500 ft/sec, for a 63 percent increase in energy density over the E-glass with the same long fatigue life.

Kevlar 49 has the potential for still higher speeds and energy density. Test rings made from Kevlar and epoxy have been repeatedly burst-tested at 3800 ft/sec, which would indicate that operational speeds of 3000 ft/sec should be possible. However, some creep at high stress levels has been observed, which may require that lower speeds be used for long-life applications such as the automobile. It is currently estimated that 2500 ft/sec be used as a design maximum for long-life application, so that at this time no special benefit exists in the use of this material. This may change as more experience is obtained. In view of the above, and until more experience is gained with the Kevlar, it appears that the use of S-glass is the better choice. It is less expensive, and the properties and winding and curing techniques are well established; at the present time, the energy density is approximately the same as Kevlar if 2500 ft/sec is taken as the maximum for long-term cycle life.

This is particularly true with respect to the cycle life, which ultimately determines the cost of ownership. Also, the battery size and total number of cells must be considered because these determine the line voltage. The battery types being considered are the lead acid, nickel iron, nickel-zinc, and lithium iron sulfide. All of these have advantages, but it is not yet known which will be the most cost effective in the final analysis.

The concept of the flywheel hybrid power system is based on the premise that regenerative braking in conjunction with load leveling will result in a sufficient improvement in vehicle range and performance to achieve the near-term goals of the program. With the possible exception of cycle life and ownership cost, this is true for all of the batteries being considered, even the near-term lead acid battery. For this reason, it is recommended that the baseline design be based on the use of the near-term lead acid battery so that the development and demonstration program can be initiated as soon as possible. Later improvements can only improve the weight and life-cycle cost aspects.

The vehicle battery goals established by the Argonne National Laboratory are given in tables 4-2 through 4-5. The tables show both the (improved) near-term goals and the advanced versions. Figures 4-6 through 4-8 show data from one manufacturer, Eagle Picher, for both current, state-of-the-art and improvements expected by 1980.

From the design studies of the hybrid power system and the vehicle gross weight variations, it is possible to determine the required battery weights as a function of energy density of the battery. Figure 4-9 shows the required weights of the various batteries for the 75-mile range.

For the lead acid battery, the data from Table 4-2 and Figure 4-6 have been used to establish the battery ratings for the baseline system. These ratings are:

Specific energy (C3 rate)	40 w-hr/kg
Specific power (sustained)	15 w/kg
Cycle life (80% discharge)	800 cycles
Cost	\$50/kwh

The cycle life of the batteries is based on the conventional traction service duty cycle. However, for the hybrid system (where the load is to be nearly constant at the C3 rate), the cycle life of a given battery should be significantly increased. The battery manufacturers generally agree, but do not have any data to support or to quantify this premise.

The voltage of the battery pack will be determined by the number of 2-v cells. At the present time, most of the high-energy traction batteries available are 6-v golf cart batteries of a standard size that weigh about 65 lb. If the design is to be based on these standard-size batteries, then the total battery weight will determine the number of batteries, and consequently the voltage.

PERFORMANCE PLOT ELECTRIC CAR MOTOR

12 JUL 76

MAGNET : L = .812 BGAP = 7000 G
 ARMATURE : 3.0 T/C AWG 12.0 R = .0361
 STACK : L = 6.00 PITCH = 8.0

Figure 4-5. Baseline Motor/Generator Characteristics

MOTOR AND GENERATOR DESIGN TRADEOFFS

The major variables in the design of the dc motor and the generator are steady-state and transient power ratings, design duty cycle, speed, voltage, and type of cooling system. The desired transient performance established in the vehicle simulation studies is described in Section 3, which shows that approximately 30 hp is desired for the acceleration and braking power.

The 108-v system was selected tentatively as a compromise between a system based on existing battery size (which would have to be 84-v for the battery pack weight objective) and still higher voltage systems that would imply much smaller cell sizes than those currently existing. Therefore, it is planned to defer the actual tradeoff on armature voltage until more information is obtained from the battery manufacturers regarding the implications of small cell sizes. Higher voltages will give higher efficiencies primarily because of the influence of the brush voltage losses. This loss is twice the loss from this effect in a normal system because the armature circuit has two sets of brushes in series.

Tentatively, the two motors are identical. Although the operational duty cycle is somewhat different, it has not yet justified using different designs. However, this will be reviewed during the conceptual design task comparisons. They are conventional, brush-type traction motors with separately excited fields. The maximum speed has been chosen as 7000 rpm, which is a compromise between the conventional, slower speed traction motor and higher speed motors used in many aircraft applications. The base speed of 1800 rpm was established to keep the maximum speed to base speed ratio less than 4:1. In the conceptual design (Task 2), these motors will be compared to higher speed motors to determine the overall gain (if any) in system efficiency and cost. Lighter-weight motors will result from the higher speeds, and this should also reduce the cost. However, higher gear and windage losses may offset these gains. Similarly, the effect of a higher speed for the generator only will be investigated.

Air cooling has been used for the baseline design. Separate, external fans have been assumed for each machine so that they will only be turned on when necessary. It is assumed that the fans are not required during normal city driving, such as in the suburban cycle, but that they may be required during a steady-state cruise at maximum speed. The thermal design to confirm this is currently in progress. Although oil cooling of the brush-type motor is somewhat difficult, it will be considered during the Task 2 design comparisons.

The characteristics of the baseline motor and generator are shown in Figure 4-5.

BATTERY DESIGN TRADEOFFS

The tradeoffs for the battery system for the vehicle have several important considerations. First, selection of the type of battery is important because recent development emphasis has accelerated the technology for some of the advanced batteries. Next, the choice of what ratings to use is important because conflicting data exist regarding the important characteristics.

S-8912

Figure 4-3. Volume Requirements for 1-kwh Flywheel

S-8913

Figure 4-4. Rotational Speed for Different Flywheel Diameter

Figure 4-2. Distance and Speed Capability for a 5-Percent Grade as a Function of Flywheel Capacity

S-8915

Figure 4-1. Total Flywheel Capacity for a 5-Percent Grade

Another major option is the size or capacity of the wheel. Ideally, the size should be the minimum consistent with the goal of eliminating all peak loads from the battery. The peak loads occur during acceleration, maximum speed passing maneuvers, and hill climbing. The hill climbing requirement is the most severe. If only acceleration to 55 mph (or 70 for passing) were required, the flywheel could be sized for an available capacity of about 250 w-hr. However, if the flywheel is to furnish the power to maintain 50 mph up a 5 percent grade for a distance of one mile, the required available capacity is about 650 w-hr, for a total capacity of about 900 w-hr. Figure 4-1 shows the total capacity as a function of distance up a 5 percent grade. The baseline system selected includes a total capacity of 1-kwh, which is sufficient to accelerate the car to 50 mph and then go up a 5 percent grade for a total distance of 1.26 miles before having to rely on the battery only. Figure 4-2 shows the distance and speed capability for a 5 percent grade for various flywheel capacities.

The rotational speed of the flywheel is determined by the maximum diameter that can be installed and by the recommended gear speeds in the planetary differential. The volume requirements for a flywheel of 1-kwh total capacity are shown in Figure 4-3 in terms of diameter and axial length. The rotational speeds for the same range of diameters are shown in Figure 4-4. Tentatively, a 23-in.-dia flywheel operating at a maximum speed of 25,000 rpm has been selected as a compromise between installation volume and reasonable planetary gear speeds. Smaller-diameter flywheels would have less windage losses, but the higher rotational speeds produce higher bearing seals and planetary gear losses. The characteristics for the baseline flywheel are summarized in Table 4-1.

TABLE 4-1
FLYWHEEL CHARACTERISTICS

Characteristic	Value
Outside diameter, in.	23
Inside diameter, in.	18.40
Axial length, in.	3.00
Rotational speed, rpm	25,000
S-glass weight, lb	33.5
Hub weight, lb	12.0
Housing weight, lb	16.0

TABLE 4-2

PRELIMINARY GOALS FOR LEAD-ACID ELECTRIC VEHICLE BATTERY
(Argonne National Laboratory)

	Improved State-of-the-Art	Advanced
Fiscal year	1978	1980
Duty cycle	2 to 4 hr discharge 4 to 8 hr charge	
Energy efficiency (round trip)	> 60 percent	
Cycle life*	800 cycles	1000 cycles
Installed cost	\$50/kwh	\$40/kwh
Specific energy	40 w-hr/kg (C3 rate)	60 w-hr/kg (C3 rate)
Specific power		
Sustaining	20 w/kg	30 w/kg
Peak	100 w/kg	150 w/kg
Typical installation size	20 to 30 kwh	
Environmental impact	Minimal	

*Corresponding to 70 to 80 percent depth of discharge

TABLE 4-3

PRELIMINARY GOALS FOR NICKEL-IRON OXIDE ELECTRIC VEHICLE BATTERY
(Argonne National Laboratory)

	Improved State-of-the-Art	Advanced
Fiscal year	1977	1979
Duty cycle	2 to 4 hr discharge 4 to 8 hr charge	
Energy efficiency (round trip)	> 60 percent	
Cycle life*	1500 cycles	1500 cycles
Installed cost	\$70/kwh	\$50/kwh
Specific energy	50 w-hr/kg (C3 rate)	60 w-hr/kg (C3 rate)
Specific power		
Sustaining	40 w/kg	50 w/kg
Peak	150 w/kg	250 w/kg
Typical installation size	20 to 30 kwh	
Environmental impact	Minimal	

*Corresponding to 70 to 80 percent depth of discharge.

TABLE 4-4

PRELIMINARY GOALS FOR ZINC/NICKEL OXIDE ELECTRIC VEHICLE BATTERY
(Argonne National Laboratory)

Duty cycle	2 to 4 hr discharge 4 to 8 hr charge
Energy efficiency (round trip)	>60 percent
Cycle life	1000 cycles
Installed cost	50/kwh
Specific energy	110 w-hr/kg (C5 rate)
Sustaining	50 w/kg
Peak	200 w/kg
Typical installation size	20 to 30 kwh
Environmental impact	Minimal

TABLE 4-5

PROGRAM GOALS FOR THE LITHIUM/IRON SULFIDE ELECTRIC VEHICLE BATTERY
(Argonne National Laboratory)

	Mark I 1977	Mark II 1979	Mark III 1981	Potential for Future
Specific energy, w-hr/kg				
Cell	100	125	160	180 to 200
Battery	75	100	130	145 to 160
Peak power, w/kg				
Cell	100	150	200	240
Battery	75	120	160	200
Lifetime				
Deep discharges	400	700	1,000	>1,000
Automobile cycles	1,200	2,100	3,000	>3,000
Equivalent miles	40,000	85,000	150,000	>250,000

Eagle-Picher Industries, Inc.
August 1976, Earl S. Carr

Figure 4-6. Lead-Acid Electric Vehicle Batteries, Power Density Versus Energy Density

Eagle-Picher Industries, Inc.
August 1976, Earl S. Carr

Figure 4-7. Nickel-Iron Electric Vehicle Batteries, Power Density Versus Energy Density

Eagle-Picher Industries, Inc.
August 1976, Earl S. Carr

Figure 4-8. Nickel-Zinc Electric Vehicle Batteries, Power Density Versus Energy Density

S-8919

Figure 4-9. Required Battery Weights for 75-mi range.

For the baseline vehicle system, the requirement for 874 lb of batteries would dictate the use of 13 or 14 standard batteries, for a maximum voltage of either 78 or 84 v. Generally, the dc traction motors will be more efficient at higher voltages because of the brush voltage drop. In the hybrid system, the armature circuit includes two sets of brushes that are in series during transient performance and parallel during cruise. It would therefore be desirable to have smaller, 6-v batteries so that the voltage could be at least 108 v, and possibly higher.

This problem has been discussed with the battery manufacturers, who feel that as long as the battery proportions do not change much from the present models, there is no reason for not having the more ideal size, except for the tooling that would be required to make them. Therefore, for the purposes of this design study, 14 standard-size golf cart batteries have been chosen as the preliminary model for use in the vehicle layouts, but assuming that smaller batteries could be made available later. The electrical system will therefore be designed for 108 v for the higher efficiency.

The major problem with the present battery technology is the effect on total vehicle costs. The initial battery costs are a penalty, and the heavy weight that is required adds to the cost of the vehicle chassis, either in additional size and weight to carry the batteries or in additional sophistication in the attempt to minimize the penalty of additional weight. Furthermore, the cycle life of the batteries is such that frequent replacement is required. Some of the advanced batteries have promise for increasing the cycle life, but only at the expense of increased cost. A certain amount of increased cost may be justified, but only if the performance or cycle life is sufficiently improved. The replacement cost of batteries is shown in Figure 4-10 in terms of cost per mile as a function of battery cost, performance, and cycle life.

GEARBOX

Various alternate versions of the planetary differential gearbox have been previously studied. During this part of the study, the analysis has shown that the primary variable in the transmission is the speed ratios of the three shafts. As long as the gear arrangement will accommodate the most desirable speed ratios of the flywheel, the generator, and the motor, then no further improvement can be made. It is apparent, of course, that additional speed changing devices would enhance the overall operational capability of the system (i.e., it would extend the speed ranges of the two motors, which have electrical commutation limitations that restrict their speed ratios). However, because of the added weight, cost, and complexity of the gear changing mechanisms, it is not recommended at the present time.

ALTERNATE TRANSMISSION CONCEPTS

Several alternate transmission concepts on hybrid drive systems have been treated in the literature. The use of an all-mechanical variable-speed drive such as the traction roller concept or the variable ratio belt drive is shown in Figure 4-11. Several of these devices are currently in various stages of development. Principal limitation is in the range of ratios permissible, which require either two stages or perhaps a fluid clutch. At the present stage of

Figure 4-10. Battery Replacement Cost as a Function of Battery Cost, Performance, and Cycle Life

Figure 4-11. All-Mechanical, Infinitely Variable Transmission Schematic

development, none of these concepts are competitive with the simple planetary differential gearbox, which is lightweight, efficient, and requires no additional clutch or speed changing mechanism.

ALTERNATE ELECTRICAL TRANSMISSION

The all-electrical concept is shown on Figure 4-12. This system has been developed by AiResearch for the NYTA R-32 railcars and also for the DOT advanced concept train (ACT-1). These systems have demonstrated substantial energy savings by energy recovery during deceleration, as well as other operational advantages. However, the overall efficiency of the system is too low to be considered for the battery-powered car. Furthermore, since all of the power must be transmitted through both of the motors, the weight (and cost) becomes excessive.

A new and interesting variation of this system considered by AiResearch is based on the use of a high-speed, Samarium-cobalt, permanent-magnet generator. This system is shown schematically in Figure 4-13. There are several advantages to this concept. The generator can be very small and lightweight, which permits it to be driven at very high speeds (such as the flywheel speed). This eliminates the gearbox entirely. The flywheel and generator housing can be hermetically sealed, which eliminates the vacuum seals. Also, very high generator efficiencies (such as 98 percent) are possible.

A disadvantage of this system is that the hermetically sealed flywheel bearings must have a lubrication system developed that will operate in the vacuum environment. Another disadvantage is the complex electronics required to rectify the high-frequency power from the generator and convert the dc voltage from the traction motor to ac for the generator. Also, the traction motor size is large because all of the power must be transmitted electrically.

Although this system has merit and should be investigated, an evaluation of the system relative to the baseline system will require an extensive analysis that is outside the scope of the present study.

CONTROLLER DESIGN TRADEOFFS

The controller will be very similar to the one developed for use on the DOT advanced concept train, and is a modification of the Ward Leonard system for the variable speed control of dc motors. In this system, separately excited motors are used, and the two armatures are connected directly. The controller adjusts the two field excitations to determine the armature current and direction in response to the command signal from the throttle or brake. The controller will be an all-solid-state device with a maximum current requirement of 15 amp.

The controller will include a microprocessor that will act to regulate the current from the battery to be used for cruise power. The microprocessor will receive speed signals from the flywheel and the traction motor; and through the use of suitable algorithms, it will determine the required battery current to maintain the sum of the vehicle and flywheel kinetic energies at a constant value. However, this control must either lag behind or include a current limiter so that the battery current demands do not reflect the transient energy consumption

Figure 4-12. All-Electrical Regenerative Power System With Two DC Motors

Figure 4-13. All-Electrical Regenerative Power System With Permanent-Magnet Motor and AC to DC Controller

rates. The objective of the controller design will be to fix the current from the battery at an average value for the driving cycle because this would result in the maximum capacity available from the battery. There may be a tradeoff between a drooping control system that controls current as a reflection of the previous time period and a simpler current limiter that may need an adjustable setting.

It is also possible that performance could be improved in some cases by cutting off the battery current entirely. The maximum acceleration rate occurs when the maximum current is taken from the generator. If the battery is supplying some of the current to the motor, then the generator armature current will be reduced and the acceleration will be reduced. It may be that for maximum acceleration throttle setting, the battery should be turned off. However, this results in a penalty in battery capacity because the average current during the total "on time" must then be increased. Therefore, there is a tradeoff between the maximum acceleration and the effect on the available battery capacity. This effect would be very small if it is only used rarely for the highest acceleration, but it can be large if frequently used. Figure 4-14 shows the variation in range as a function of the "on time" fraction.

The analysis and preliminary design of the controller will be started after the complete spectrum of motor and generator performance is more firmly established.

Figure 4-14. Variation in Range as a Function of Battery "On-Time".

SECTION 5

VEHICLE DESIGN TRADEOFFS

INTRODUCTION

Vehicle design tradeoffs conducted under this task include:

- Vehicle gross weight, empty weight, and range
- Power system and battery locations

These are discussed in this section. In addition, operational considerations are being studied throughout the program. Operational tradeoffs are discussed at the end of this section.

VEHICLE GROSS WEIGHT, EMPTY WEIGHT, AND RANGE

It is important to determine the vehicle gross weight at an early stage. The proposed maximum 3000 lb with four passengers, or a curb weight of 2400 lb, is seen as an ambitious goal when converted to the vehicle empty weight without the batteries. The overall weight must be minimized because of its effect on initial cost and the cost of battery replacement. With the present sell price of compact automobiles at approximately \$2/lb and with a target production sell price of the the electric car of \$5000, it is apparent that the tradeoff between both cost and weight is heavily biased toward cost, and even large reductions in weight can only justify minor increases in the cost/lb of the vehicle parts.

The energy requirements as a function of the gross vehicle weight (GVW), as predicted by the computer simulation, are shown in Figure 5-1. For the baseline 3000-lb GVW, a 15.7-kwh battery pack is required for the 75-mile range. As shown, a 10 percent increase in the GVW will require a 6 percent increase in the battery weight. Figure 5-2 shows the range (J227A,d) as a function of GVW and battery weight. This is based on the use of lead acid batteries with an energy density of 18 w-hr/lb (60-amp rate). This figure shows the penalty in battery weight for excessive GVW. It also shows the reduction in the range for a given battery weight. For the baseline system with 874 lb of batteries, a 10 percent increase in the GVW would cause the range to be reduced to 70 miles. The same data are shown in Figure 5-3 as a function of the empty weight of the car without the batteries and without the engine. These weights clearly show the challenge to the vehicle designer to achieve the 1276-lb objective. As an example of this, the empty weight of the VW Rabbit without the engine and fuel supply is 1508 lb.

Empty vehicle weights of several production vehicles in the same weight class are listed in Table 5-1. These data are taken from the DOT report, "Estimated Weights and Manufacturing Costs: Automobiles, Vehicles and Parts," by Rath & Strong Inc. The table for the 2400-lb curb weight car was derived by plotting the tabulated data as a function of the curb weights and then taking average values for a hypothetical 2400-lb car. The column for the baseline objectives shows a tentative target weight allocation to achieve the overall 20 percent weight reduction required.

Figure 5-1. Energy Requirements as a Function of Vehicle Gross Weight

Figure 5-2. Range as a Function of Vehicle Gross Weight

Figure 5-3. Range as a Function of Empty Vehicle Weight (Excluding Batteries and Engine)

TABLE 5-1

VEHICLE EMPTY WEIGHTS
WITHOUT ENGINE AND FUEL SYSTEM

	1975 Vega Weight*, lb (Curb Weight, 2487 lb)	1976 Chevette Weight*, lb (Curb Weight, 1997 lb)	Detroit Average Weight* for 1980 to 1985, lb (Curb Weight, 2400 lb)	Target Objective Weight, lb (Curb Weight, 2400 lb)
Frame and body structure	432	347	440	316
Body and panels	858	670	535	476
Drive system	10	97	116	96
Suspension system	38	124	142	110
Brake system	150	118	144	106
Steering system	37	35	39	33
Wheels and tires	160	145	174	139
	<hr/> 1385	<hr/> 1536	<hr/> 1590	<hr/> 1276

*"Estimated Weight and Manufacturing Costs for Automobiles, Vehicles, Systems, and Parts,"
by Rath & Strong, Inc., 1976

The entire problem of vehicle and battery weights is a direct function of the drive efficiency, and although efforts will continue to optimize the component efficiencies, the already high overall system efficiency would indicate that any further gains will be small. This problem is summarized by the data in Figure 5-4, which show the empty weight design objective as a function of the power system energy requirements and, as a reference, the ideal energy requirements for a no-loss power system.

The data of Figure 5-4 depict the principal challenge of the program--to achieve the weight goal within cost objectives. In determining the cost, in either development or tooling, to reduce the weight to the target of 1276 lb, a certain amount of nonrecurring costs are clearly justified. However, if the resultant product would have to be sold for much greater than \$2/lb (\$1 at cost), the cost objectives for the commuter car would not be achieved.

The industry in Detroit has been working on the problem of providing a lightweight frame and body. Only with the use of higher strength-to-weight ratio materials will it be possible to eliminate much of the weight of the frame. The experimental Chevrolet XP-898 project is an example. In this project, Chevrolet designed and built a prototype sports car that has the entire frame and body made from reinforced plastics. In a discussion with the designer of this vehicle, he stated his belief that it will be possible within a few years to put this concept into production without the extensive use of high-cost materials.

Because of the importance of the overall cost and weight relationships, this subject will be emphasized during all of the remaining studies.

VEHICLE LAYOUT STUDIES OF THE POWER SYSTEM AND BATTERY LOCATIONS

Many layouts were prepared to study the effect of different locations for the power components and the batteries. Several examples of these layouts are shown on the Brubaker Group drawings PRE 3001 through 3008. As explained in Section 4, the preliminary model for the batteries would be 14 standard-size golf cart batteries, even though this is not an optimum model. This is slightly more weight and volume than is needed, and therefore the final configurations should be somewhat easier to accommodate. Preliminary sketches of the various power system arrangements were used for these early layouts.

The location of the batteries has conflicting requirements. It would be desirable to have the batteries near the center of the vehicle to minimize the vehicle polar moment of inertia. However, center tunnel locations (as shown on Drawings 3001 and 3002) would appear to be the most difficult for servicing, and probably would require battery removal. This would be difficult without special handling equipment. The center tunnel arrangement could have important structural advantages if the tunnel also were designed as a box section to act as the principal torsional structure of the frame and body. Also, the center tunnel is the only arrangement that would accommodate all of the batteries into one module. All other versions require that the batteries be separated into two or more modules to achieve the proper weight distribution.

Figure 5-4. Vehicle Empty Weight as a Function of Energy Consumption for Different Vehicle Gross Weights

Copyright © 1971 by The Brubaker Group All rights reserved. No part of this publication may be reproduced, stored in a retrieval system, or transmitted, in any form or by any means, electronic, mechanical, photocopying, recording, or by any information storage and retrieval system, without the prior written permission of The Brubaker Group.		DRAWN BY		DATE CHECKED BY		DATE	
		DESIGNED BY		DATE		PROCESS DATE	
		DESIGN ENGR		DATE		FINAL APPROVAL DATE	
		MATERIAL					
NEXT ASSEMBLY		WEIGHT		DO NOT SCALE DRAWING		SCALE	
				SIZE		CODE IDENT NO.	
				DRAWING NUMBER		REV.	
				PRE 3001			
				SHEET		OF

<small>Copyright notice and permission of use. This drawing is the property of The Brubaker Group and is not to be reproduced without written permission. The Brubaker Group is not responsible for any errors or omissions in this drawing. The Brubaker Group is not responsible for any damage or injury resulting from the use of this drawing. The Brubaker Group is not responsible for any loss of profit or business resulting from the use of this drawing. The Brubaker Group is not responsible for any other consequences resulting from the use of this drawing.</small>		DRAWN BY		DATE CHECKED BY		DATE		
		DESIGNED BY		DATE		PROCESS		DATE
		DESIGN ENGINE		DATE		FINAL APPROVAL		DATE
		MATERIAL						
© The Brubaker Group		MATERIAL						
		MATERIAL		CONNECTION				
NEXT ASSEMBLY		WEIGHT		DO NOT SCALE DRAWING		SCALE		
						SHEET OF		

The Brubaker Group 10785 W. Plinn Boulevard Los Angeles, California 90064 (213) 277-6251			
TITLE			
SIZE	CODE IDENT NO.	DRAWING NUMBER	REV.
		PRE 3002

<small>By the use of this drawing, the Designer warrants that the design is original and that the Designer is not infringing on the rights of others. The Designer warrants that the design is not a copy of any other design and that the Designer is not aware of any other design that may be infringing on the rights of others. The Designer warrants that the design is not a copy of any other design and that the Designer is not aware of any other design that may be infringing on the rights of others.</small>		DESIGN BY	DATE	DESIGNED BY	DATE	The Brubaker Group 10315 W. Pico Boulevard Los Angeles, California 90064 (213) 277-6251	
		BY OTHER	DATE	PROCESSOR	DATE		
		DESIGN ENGINEER	DATE	FINAL APPROVAL	DATE		
		MATERIAL					
NEXT ASSEMBLY		WEIGHT		DO NOT SCALE DRAWING		SCALE	SHEET OF

TITLE			
SIZE	CODE	JOINT NO.	DRAWING NUMBER
			PRE 3003
			REV.

11-5-10

<small> The Brubaker Group 10315 W. Pico Boulevard Los Angeles, California 90064 (213) 277-6251 </small>		DRAWN BY: _____ DATE: _____ CHECKED BY: _____ DATE: _____ DESIGNED BY: _____ DATE: _____ APPROVED BY: _____ DATE: _____	
THE BRUBAKER GROUP		TITLE: _____	
MATERIAL: _____		SIZE: _____ CODE IDENT. NO.: _____ DRAWING NUMBER: _____	
NEXT ASSEMBLY: _____		PRE 3005	
DO NOT SCALE DRAWING		SCALE: _____

THE BRUBAKER GROUP 10315 W. PICO BOULEVARD LOS ANGELES, CALIFORNIA 90064 (213) 277 6251	DESIGNED BY	DATE	CHECKED BY	DATE	TITLE
	STRESS	DATE	PROCESS	DATE	
	DESIGN ENGR	DATE	FINAL APPROVAL	DATE	
	MATERIAL				
© The Brubaker Group	MATERIAL				SIZE CODE IDENT NO. DRAWING NUMBER
	MATERIAL CONDITION				
NEXT APPROVAL	WEIGHT				DO NOT SCALE DRAWING SCALE
PRE 3006					SHEET OF

<small>This drawing and any information contained herein are the property of The Brubaker Group, Inc. and are to be used for the purpose specified only. No part of this drawing may be reproduced or transmitted in any form or by any means electronic or mechanical, including photocopying, recording, or by any information storage and retrieval system, without the prior written permission of The Brubaker Group, Inc.</small>		DRAWN BY		DATE		CHECKED BY		DATE		The Brubaker Group 10315 W. Pico Boulevard Los Angeles, California 90064 (213) 277-6251	
		STATION		DATE		REVISION		DATE			
		DESIGN ENGINEER		DATE		FINAL APPROVAL		DATE			
		MATERIAL									
© The Brubaker Group		DATE		CONDITION						TITLE 	
NEXT ASSEMBLY		WEIGHT		DO NOT SCALE DRAWING		SCALE		SHEET		OF	
								PRE 3007		REV.

11-5-13

Mounting the batteries along the two side sills (as shown in Drawings PRE 3003 and 3004) would be somewhat better for servicing accessibility and would still minimize the vehicle polar moment of inertia. However, the seat width and ingress and egress would be severely compromised. Dividing the batteries between the front (engine) compartment and the rear (trunk) compartment sacrifices the lower polar moment of inertia, but provides the most straightforward solution. This concept provides the most design flexibility in location of the power components and also in the ultimate shape and size of the batteries. Accessibility for servicing and removal is good, and allowances for a reasonable luggage compartment can be provided. Drawings PRE 3005 through 3008 show various versions of this concept. No practical version of mounting the batteries under the seats or across the width of the vehicle was found, primarily because of the height limitations of the standard golf cart batteries. In view of the above, future work will be concentrated on the fore and aft mounting of the batteries in conjunction with either front or rear drive systems.

Drawings PRE 3005 and 3006 show the results of the layout study for the front wheel drive version. Drawing 3006 has been selected for more detail study in Task 2. Drawings 3007 and 3008 show two arrangements for rear wheel drive or split (4-wheel) drive. These versions have also been selected for more detailed study.

OPERATIONAL TRADEOFFS AND CHARACTERISTICS

Tradeoffs related to the operational characteristics that have been considered are the loss of the flywheel energy during extended parking and the subsequent need to reenergize the flywheel for startup. Also, some options in the design of the electrical braking system have been considered.

Energy Loss During Parking

Normally, when the car is parked for an extended time, the full energy capacity of the flywheel will be lost as the flywheel spins down. However, for the normal commuter trip, the battery current could be turned off 3 to 4 miles from the destination. The car would then be parked with a nearly depleted energy supply in the flywheel. The spin down loss would then be as low as 250 w-hr instead of the full capacity of 1000 w-hr.

Startup After Extended Parking

The normal procedure to start the system after the flywheel has run down is to energize the flywheel by the battery, which will require about 8 to 10 minutes for full speed. For normal commuter use, a timer might be satisfactory, which would energize the system a few minutes before the vehicle was to be used. However, if this is desired, it is not necessary to wait the full 10 minutes to fully energize the system. Two options are available to the driver. First, the car can be operated by the traction motor until the flywheel is energized, which would be within the first few minutes of driving. This would take higher-than-normal current levels, and therefore, there would be some penalty in range. This penalty is estimated to be about 5 miles. The second option would be to partially energize the flywheel before starting. If the flywheel is energized

for only 2 or 3 minutes before starting, the consequent penalty for driving while recharging the flywheel would be reduced to about 2 miles.

Braking Characteristics

A possible disadvantage of the regenerative braking concept could be the use of only two wheels for braking. Although the full-power braking effect is modest, as shown in Figure 2-5, Section 2, it is sufficient for all normal driving conditions. However, the minimum coefficient of tire adhesion required to prevent a skid is approximately twice the value required for four-wheel braking. This may be objectionable during wet or icy driving conditions.

The split-power characteristics of the power system permit the design to be arranged for four-wheel braking. This would be accomplished by driving one axle with the traction motor and one axle with the planetary carrier output shaft. The final drive gear ratio would be different for each shaft to compensate for the different shaft speeds. This problem has been analyzed and layouts have been made to examine the relative benefits of the concept. Figure 5-5 is a schematic of one possible arrangement. In this case, the traction motor has been mounted on the front axle because there is more braking effect from the motor than from the carrier shaft.

The disadvantage of the two-wheel braking system may be eliminated by the four-wheel hydraulic braking system. This system will require a control system that will provide the smooth transition from electrical braking to the hydraulic braking system. It is probable that this control system can include a positive anti-skid feature with a minimum of additional complexity. This feature would override the electrical braking command in the event of an incipient skid, and thus minimize the disadvantage of the two-wheel braking concept. A more detailed study of this concept will be conducted to determine the feasibility of this approach.

Figure 5-5. Split Transmission Schematic

SECTION 6

ECONOMIC EVALUATION/ANALYSIS MODEL

INTRODUCTION

This section describes the objectives and structure of a computerized economic analysis model and presents some of the initial results obtained using preliminary input data generated in the early stages of the Phase I program activity. The model designation, EVECOM, is an acronym for electric vehicle economic analysis model, and has been programmed and used on a Univac-1108 digital computer system currently operational at AiResearch.

OBJECTIVES

The specific objectives of the economic modeling activities during Phase I of the electric vehicle program are:

- (a) The generation of economic and cost data concurrently and in parallel with the engineering design of the electric vehicle to ensure that the principal economic goals in the areas of vehicle investment (purchase) and lifetime operating costs are both feasible and achievable.
- (b) To assist, through sensitivity and tradeoff analysis, in determining the preferred or optimum set of vehicle technical design parameters leading to a vehicle design that meets or exceeds the required performance and economic criteria.
- (d) To assess the impact of major program variables that, while perhaps not directly under the control of the vehicle designer, could have a significant bearing in the areas of public acceptance and market penetration potentials for the electric vehicle.

MODEL STRUCTURE

EVECOM is basically a deterministic model (i.e., it utilizes analytical relationships derived largely from historical data to estimate the individual cost categories and elements making up the system or total system cost of an electric vehicle. Its three principal system cost categories are (1) research, development, testing, and evaluation (RDT&E), (2) acquisition; and (3) ownership costs. These are, in turn, further broken down into cost elements where required; the sum of all cost elements within the three principal categories comprises the total system, or lifetime, cost. The cost estimating relationships used in the model for each cost element are described below. These relationships are subject to refinement and revision throughout the duration of Phase I as more representative basic input data are collected and generated by AiResearch and as further guidance is received from the ERDA program office. All monetary values are in 1975 dollars.

COST ESTIMATING RELATIONSHIPS (CER's)

RDT&E Cost

This cost category includes the research and development costs that would be incurred by an electric vehicle manufacturer prior to undertaking a large-scale production program for a specific electric vehicle model or models. This is distinct from, but not independent of, U.S. Government programs such as the current ERDA electric vehicular program and other possible future government-sponsored electric vehicle programs. It represents the manufacturer's in-house development effort prior to committing the large capital investments required for mass production and incorporates all the knowledge and expertise gained from outside programs such as the on-going ERDA program and the manufacturer's own background and experience. In estimating the total RDT&E costs, the following cost elements have been defined:

- (a) Initial engineering
- (b) Development tooling
- (c) Development support
- (d) Manufacture of the R&D vehicles
- (e) Test facilities
- (f) Test operations
- (g) Contractor profit

The general calculational algorithm for RDT&E costs is:

$$\text{Total Cost} = (\text{element hours}) (\text{composite rate}) + (\text{man-hour related costs}) + (\text{profit}), (\$). \quad (6-1)$$

Equation 6-1 is applicable to all RDT&E cost elements. At this stage in the model development, CER's have not yet been formulated for engineering, tooling, support, and test operations hours and the related man-hour costs; therefore, these values are presently treated as inputs to the program. An analysis of historical data for similar programs will be made and should provide a basis for the construction of appropriate CER's, which will then be incorporated into the model.

The manufacturing cost of the R&D vehicles is a function of the number of such vehicles selected by the program user and is based on employment of the production learning curve concept. The total manufacturing cost of the vehicle, as derived in the next subsection, is assumed to be the cumulative average cost at some large production quantity. The cost of the R&D vehicles is computed based on a conventional learning curve, the slope of which is an input specified by the user. The applicable formula is:

Cost of Manufacturing R&D Vehicles =

$$\frac{(\text{cum avg cost at quantity N})(\text{No. of R\&D vehicles})^{1.4427 \ln (\text{slope})+1}}{(\text{annual production rate})^{1.4427 \ln ("slope")}} \quad (6-2)$$

where the slope (S) of the learning curve (a log-linear relationship is assumed) is defined as follows:

$$\bar{Y}_N = aX^b \quad (6-3)$$

$$S = 2^b \quad (6-4)$$

where

$\bar{Y}_N$ = cumulative average cost of N items

X = cumulative output

a = cost of first item

b = actual slope of the logarithmic curve (a straight line on log grids)

An example of a summary of RDT&E costs is shown on page 1 of Figure 6-1 presented at the end of this section.

Acquisition Cost

This cost category, second of the three principal categories, includes all costs associated with the manufacture, distribution, and marketing of the vehicle as reflected in the sticker price, and the costs of financing incurred by the owner and any salvage value remaining after its useful life. It is assumed that the purchase price is equal to the sticker price of the vehicle.

In estimating the manufacturing costs of the vehicle, the vehicle was broken down into components and subsystems, and a cost estimating relationship (CER) was developed for each element by correlation with some physical characteristic of the element, such as weight. In many cases, the available cost data correlated well with vehicle curb weight, and this was therefore used as a parameter in the CER's. The CER's discussed below, derived largely from Reference 6-1*, are based on 1975 dollars and assume economic lot sizes, (i.e., production runs on the order of at least 100,000 vehicles annually). As previously discussed, the manufacturing cost estimated is assumed to be the cumulative average cost (that is, if the annual output is N vehicles, the total program cost can be derived by simply multiplying the cumulative average cost of the nth vehicle by the total output N). Also, the

*References presented at end of section.

manufacturing cost relationships are based on present-day conventional automotive design and manufacturing practices. Should nonstandard or significantly different design, material, or manufacturing standards be employed in the electric vehicle (to reduce curb weight, for example), the present CER's would have to be reexamined for applicability.

1. Vehicle Acquisition, Cost CER's in 1975 Dollars

Determination of vehicle acquisition cost directly related to material acquisition, manufacture, and markup is delineated in Table 6-1. Since it is assumed in the model formulation that the sticker price is the actual sales price paid by the customer, the items not included in the sticker price formulation given in Table 6-1, such as transportation and dealer preparation, can be considered equivalent to any discount negotiated between the dealer and the customer.

(a) Cost of Financing

The cost of financing the vehicle purchase is calculated in the economic model as follows:

$$\text{Cost of financing} = C_{\text{FIN}} = P_s \left(\frac{ti}{1 - (1+i)^{-t}} - 1 \right), \$ \quad (6-5)$$

where P_s = sticker (sales) price, \$

t = financing term, yr

i = financing interest rate, decimal

For the purpose of generating some early runs with the model, a term of 3.33 yr and an interest rate of 12 percent have been assumed.

(b) Salvage Value

The salvage value, in 1975 dollars, is calculated as a fraction of the purchase price (sticker price). For the initial use of the model, a value of 7 percent was assumed. The salvage value includes any remaining life in batteries, tires, and components in the propulsion system such as motors and generators which can be reconditioned and resold.

2. Total Acquisition Cost

The total lifetime vehicle acquisition cost is defined as the sum of the purchase price (suggested retail sticker price) and the financing charges, minus the vehicle salvage value. An example showing a summary of the acquisition cost for an illustrative case is on page 4 of Figure 6-1. The associated total manufacturing cost summary is presented on page 2 of Figure 6-1.

Ownership Cost

The third and final cost category in the vehicle total system cost is the ownership cost. This category also has been described variously as annual

TABLE 6-1

VEHICLE COST BREAKDOWN CATEGORIES

Item	Category	Determination Method	Units
1	Frame and body structure (unitized frame)	$C_{FS} = 0.67$ where C (in general) is in \$/lb or \$/mi and subscripts refer to appropriate individual components	\$/lb
2	Removable panels (2 doors)	$CRP2D = 1.17 - 0.0000403 W_c$, where W_c = vehicle curb weight (lb)	\$/lb
3	Removable panels (4 doors)	$CRP4D = 1.17 - 0.0000310 W_c$	\$/lb
4	Basic body (includes interior and exterior trim, accessories, lights, glass, heater, wiper/washer, crash and panel pads)	$C_{BB} = 1.45$	\$/lb
5	Drive system, rear drive	$C_{DSR} = 0.79 - 0.0000719 W_c$	\$/lb
6	Drive system, front drive	$C_{DSF} = 0.83 - 0.0000639 W_c$	\$/lb
7	Suspension system, coil/leaf	$C_{SUS\ c/l} = 0.61 - 0.0000170 W_c$	\$/lb
8	Suspension system, coil/coil	$C_{SUS\ c/c} = 0.67 - 0.0000179 W_c$	\$/lb
9	Brake system, drum/drum	$C_{BR\ DRM/DRM} = 0.536 - 0.0000213 W_c$	\$/lb
10	Brake system, disk/drum	$C_{BR\ OSK/DRM} = 0.60 - 0.0000379 W_c$	\$/lb
11	Steering system, front drive, ball and rack and pinion	$C_{STR,BALL,R\&P,F} = 1.32 - 0.0000379 W_c$	\$/lb
12	Steering system, rear drive, ball and rack and pinion	$C_{STR,BALL,R\&P,R} = 1.35 - 0.0000368 W_c$	\$/lb
13	Wheels, 4 or 5	$C_{WH}(4),(5) = 0.109 + 0.000109 W_c$	\$/lb
14	Tires, 5 bias ply	$C_{TIRE\ BIA}(5) = 0.404$	\$/lb
15	Tires, 5 radial ply	$C_{TIRE\ RAD}(5) = 0.572$	\$/lb
16	Tires, 4 run flat	$C_{TIRE\ RUN\ FLAT}(4) = 0.850 + 0.000115 W_c$	\$/lb
17	Restraints	$C_{RES} = 2.042$	\$/lb
18	Air Conditioning	$C_{A/C} = 4.427$	\$/lb
19	Propulsion System, Motor	$C_{MOT} = 1.35$	\$/lb
20	Generator	$C_{GEN} = 1.35$	\$/lb
21	Flywheel	$C_{FW} = 3.57$	\$/lb
22	Flywheel Housing	$C_{FWH} = 1.10$	\$/lb
23	Gearbox	$C_{GB} = 1.10$	\$/lb
24	Controller	$C_{CONTR} = 20.00$	\$/kw
25	Battery Charger	$C_{BC} = 8.00$	\$/kw
26	Battery	$C_{BAT} = 0.42$	\$/lb
27	Vehicle Assembly	$C_{ASSEM} = 0.0668$	\$/lb curb wt
28	Total Manufacturing Cost (includes variable and fixed material and labor costs; engineering tooling costs; and quality control costs)	$C_{MANF,TOT} = \sum \text{Items 1 through 27}$	\$
29	Markup Factor (includes factory profit; warranty costs; marketing costs; facilities equipment costs; corporate profit; and dealer markups)	$MF = 0.0002918 (\text{manufacturing cost}) + 1.4083$	\$
30	Sticker Price (suggested retail selling price; does not include transportation, dealer preparation, or sales taxes)	$P_s = (MF)(C_{MANF,TOT})$	\$

operating cost, operating and maintenance (O&M) cost, and road costs. For purposes of this study, ownership cost is defined as including the following cost elements:

- (a) Maintenance and repair cost for the complete vehicle
- (b) Tire replacement cost
- (c) Accessories cost such as expenditures for floor mats, seat covers, radio/stereo/tape equipment, etc.
- (d) Insurance, including comprehensive coverage (fire, theft, glass), collision, and liability (including uninsured motorist coverage)
- (e) Registration and licensing (title certification), including personal property taxes if applicable
- (f) Garaging, parking, and toll costs
- (g) Battery recharging energy cost
- (h) Battery replacement cost, including financing charges

State and Federal taxes on gasoline and oil and tires are not broken out separately as they are assumed to be either not applicable for the electric car at this stage of its development or, as in the case of tires, are included in the appropriate cost element.

The CER's for the items listed above have been formulated using data from several sources, particularly References 6-2, 6-3, and 6-4. Many of the cost elements comprising ownership costs are highly dependent on geographical area; insurance cost is an example of this, where depending on the particular area of the country being considered, the same insurance coverage can vary in cost from \$150 to \$500 annually. In this connection, the CER's given in Table 6-2 incorporate, where appropriate, 1975 cost data for the Chicago area on the assumption that this area is more representative of a national average than the Baltimore area, which is used as a basis in Reference 6-2. Recent discussion with ERDA on this subject has lead to an understanding that guidance in the area of standardization of certain ownership costs will be forthcoming to the study contractors. Determination of ownership costs is delineated in Table 6-2.

Maintenance and repair cost for the propulsion system (Item 2) has been analyzed for maintenance and repair requirements by breaking it down into its constituent components and estimating the wearout or replacement life of the critical parts and assemblies in each principal component. In the case of the traction motor, for example, the critical items identified were the motor brushes and the rotor bearing assembly. It was then estimated that, based on previous history with similar components and duty cycle environments, brush replacement would be required after 25,000 miles of vehicle operation; bearings will be designed for a life of at least 100,000 miles of operation. The results of this type of analysis applied to the entire propulsion system,

TABLE 6-2
OWNERSHIP COST BREAKDOWN

Item	Category	Determination Method	Units
1	Maintenance and Repair Cost, other than Propulsion System	$C_{MCR, \text{ other}} = \frac{65.4 + 0.0118 W_c}{\text{annual mileage}}$ <p style="text-align: center;">where W_c = vehicle curb weight, lb</p>	\$/mi
2	Maintenance and Repair Cost, Propulsion System	$\text{Total Cost} = \left(\frac{\text{total vehicle life (mi)}}{\text{Scheduled maint. interval (mi)}} - 1 \right) [(mean \text{ time to repair})(labor \text{ rate}) + (\text{parts cost per maint. interval})]$	\$
3	Tire Replacement Cost	$C_{TIRE \text{ REPL}} = \left(\frac{\text{total vehicle life (mi)}}{\text{tire life (mi)}} \right) \left[\frac{(\text{OEM cost of tire})(\text{retail markup factor})}{\text{total vehicle life (mi)}} \right]$	\$/mi
4	Accessories Cost	$C_{ACCESS} = \frac{27.40}{\text{annual mileage}}$	\$/mi
5	Insurance Cost	$C_{INSUR} = \frac{375.00}{\text{annual mileage}}$	\$/mi
6	Registration and License Cost	$C_{REG \ \& \ LIC} = \frac{17.00}{\text{annual mileage}}$ <p>This formulation assumes that no personal property tax is levied against the vehicle and that no taxes in lieu of the State and Federal gasoline taxes are imposed.</p>	\$/mi
7	Garaging and Parking Cost (includes tolls)	$C_{GAR \ \& \ PARK} = \frac{213.00}{(\text{annual mileage})}$	\$/mi
8	Battery Recharging Energy Cost	$C_{ENERGY} = (\text{recharge energy requirement, kwh/mi}) (\text{energy cost, \$/kwh})$	\$/mi
9	Battery Replacement	$C_{BAT \text{ REPL}} = \left[\frac{\text{total vehicle life, yr} - 1}{\text{battery life, yr}} \right] \left[\frac{(\text{OEM cost of battery})(\text{retail markup factor})}{(\text{total vehicle life, mi})} \right]$	\$/mi

together with estimated labor and parts cost requirements, were summarized in a maintenance matrix, an example of which can be found on page 3 of Figure 6-1. The general computational algorithm used in this matrix is presented in Table 6-2, Item 3.

Financing charges for the battery replacement purchases are calculated in a manner similar to that employed in financing the vehicle purchase, Equation 6-5, with the term P_S replaced by the battery replacement cost $C_{BAT\ REPL}$, and the length of the loan and interest rate entered as judged appropriate by the program user.

TOTAL OWNERSHIP COST

Total ownership cost is the sum of individual ownership cost elements discussed above. An example of ownership costs calculated for a representative but tentative set of inputs in the early stages of the design study is shown on page 4 of Figure 6-1.

TOTAL ELECTRIC VEHICLE SYSTEM (LIFETIME) COST

The aggregate sum of the RDT&E, acquisition, and ownership costs presented above is the total system, or lifetime (life-cycle), cost of the electric vehicle. This total cost can be calculated on a system (lifetime) basis, an annual basis, or a per-mile basis; all three calculation bases are shown in the example printout on page 4 of Figure 6-1, together with a percentage breakdown for each cost category and element.

REFERENCES

- 6-1. Fitzgibbons, R.G., and L.H. Lindgren, "Estimated Weights and Manufacturing Cost, Automobiles, Vehicles, Systems, and Parts," Goals Beyond 1980, U.S. Department of Transportation, Rath and Strong, Inc., Lexington, Massachusetts, March 15, 1976.
- 6-2. Liston, L.L., and R.W. Sherrer, Cost of Operating an Automobile, U.S. Department of Transportation, Federal Highway Administration, Washington, D.C., April 1974.
- 6-3. Kirkwood, T.F., and A.D. Lee, A Generalized Model for Comparing Automobile Design Approaches to Improved Fuel Economy, Report R-1562-NSF, The Rand Corporation, Santa Monica, California, January 1975.
- 6-4. Personal communication with Runzheimer and Company, Inc., Runzheimer Park, Rochester, Wisconsin. Mr. Don Stielow, Senior Field Consultant.

ERDA ELECTRIC VEHICLE PROGRAM PHASE I AUGUST 1976
RESEARCH, DEVELOPMENT, TESTING, & EVALUATION (RDT+E) COST SUMMARY
(1975 \$)

PAGE 1

RUN NO. 1

DATE 09 SEP 76

VEHICLE DESIGN I.D. 1

NUMBER OF R + D VEHICLES 3.

COST ELEMENT	HOURS (HRS)	COMPOSITE RATE (\$/HR)	TOTAL COST(\$)	TOTAL COST INCLUDING CONTRACTOR PROFIT
INITIAL ENGINEERING	100000.00	25.00	2525000.00	2777500.00
DEVELOPMENT TOOLING	3000.00	20.00	160000.00	176000.00
DEVELOPMENT SUPPORT	100000.00	21.00	2150000.00	2364999.90
MANUFACTURE OF R+D VEHICLES			98495.07	108344.58
TEST FACILITIES			100000.00	110000.00
TEST OPERATIONS	10000.00	22.50	230000.00	253000.00
TOTAL: RDT+E			5263495.10	5789844.10

Figure 6-1. Electric Vehicle Economic Analysis Computer Output
(Page 1 of 4)

ERDA ELECTRIC VEHICLE PROGRAM PHASE I AUGUST 1976

VEHICLE ACQUISITION COST SUMMARY

(1975 \$)

ANNUAL PRODUCTION RATE 100000, VEHICLES/YR

PAGE 2

RUN NO. 1

DATE 09 SEP 76

VEHICLE DESIGN I.D. 1

VEHICLE COMPONENT/SYSTEM	WEIGHT (LBS)	COST (\$)	COST/LB (\$/LB)	% OF STICKER PRICE
FRAME + BODY STRUCTURE (UNITIZED FRAME)	280.00	123.48	.44	1.93
REMOVABLE PANELS (2 DOORS)	220.00	475.04	2.16	7.42
BASIC BODY	210.00	284.76	1.36	4.45
DRIVE SYSTEM (FRONT DR)	96.00	83.64	.87	1.31
SUSPENSION SYSTEM (COIL/COIL)	110.00	67.45	.61	1.05
BRAKE SYSTEM (DRK/DRM)	106.00	66.21	.62	1.03
STEERING SYSTEM (RACK + PINION, FRONT DR)	33.00	46.14	1.40	.72
WHEELS (4)	69.00	25.57	.37	.40
TIRES (4 RUN FLAT)	70.00	78.82	1.13	1.23
RESTRAINTS (4)	6.00	12.25	2.04	.19
AIR CONDITIONING	.00	.00	.00	.00
PROPULSION SYSTEM				
MOTOR	80.00	308.00	3.85	4.81
GENERATOR	50.00	192.50	3.85	3.01
FLYWHEEL	25.00	122.50	4.90	1.91
FLYWHEEL HOUSING	25.00	36.00	1.44	.56
GEAR BOX	20.00	38.40	1.92	.50
CONTROLLER	10.00	120.20	12.02	1.88
BATTERY CHARGER	10.00	120.20	12.02	1.88
SYSTEM INSTALLATION	20.00	.00	.00	.00
BATTERY	900.00	495.00	.55	7.73
BATTERY INSTALLATION	60.00	.00	.00	.00
TOTAL PROPULSION SYSTEM	240.00	937.80	3.91	18.65
TOTAL BATTERY SYSTEM	960.00	495.00	.52	7.73
VEHICLE ASSEMBLY		160.32	.07	2.50
VEHICLE CURB WEIGHT	2400.00		.00	.00
TOTAL MANUFACTURING COST		2856.48	1.19	44.61
FACTORY + DEALER MARKUPS		3547.00	1.48	55.39
SUGGESTED RETAIL STICKER PRICE		6403.48	2.67	100.00

Figure 6-1. Page 2 of 4

11-6-11

11-6-12

PAGE 3

ERDA ELECTRIC VEHICLE PROGRAM PHASE I AUGUST 1976
 PROPULSION SYSTEM MAINTENANCE COSTS
 (1975 \$)

RUN NO. 1
 DATE 09 SEP 76
 VEHICLE DESIGN I.D. 1

AVERAGE ANNUAL MILEAGE 10000. (MI) TOTAL VEHICLE LIFE 10.00 (YRS)

COMPONENT	SCHEDULED MAINT INTERVAL (MI)	MEAN TIME TO REPAIR/ REPLACE/INSPECT (HRS)	LABOR RATE (\$/HR)	PARTS COST/MAINT INTERVAL (\$)	LIFETIME TOTAL COST (\$)	COST/MILE (\$/MI)
1. MOTOR						
BRUSHES	25000.0	.50	12.00	10.00	48.00	.000480
BEARINGS	100000.0	1.50	12.00	20.00	.00	.000000
2. GENERATOR						
BRUSHES	25000.0	.50	12.00	10.00	48.00	.000480
BEARINGS	100000.0	1.50	12.00	20.00	.00	.000000
3. FLYWHEEL						
SEALS	50000.0	1.00	12.00	10.00	22.00	.000220
BEARINGS	50000.0	2.00	12.00	25.00	49.00	.000490
4. GEARBOX						
BEARINGS	50000.0	2.00	12.00	25.00	49.00	.000490
5. CONTROLLER	100000.0	1.25	12.00	15.00	.00	.000000
6. BATTERY CHARGER	100000.0	1.30	12.00	15.60	.00	.000000
7. BATTERY	800.0	.25	12.00	.10	384.40	.003844
TOTAL MAINTENANCE COST					400.40	.004004

ERDA ELECTRIC VEHICLE PROGRAM PHASE I AUGUST 1976

PAGE 4

TOTAL SYSTEM COST SUMMARY

RUN NO. 1

(1975 \$)

DATE 09 SEP 76

VEHICLE DESIGN I.D. 1

TOTAL VEHICLE LIFE 10, (YRS) AVERAGE ANNUAL MILEAGE 10000, (MI)
 ANNUAL PRODUCTION RATE 100000, (VEHICLES/YR) ROI+E AMORTIZATION PERIOD 5, (YRS)

SYSTEM COST CATEGORY	LIFETIME COST (\$)	ANNUAL COST (\$/YR)	LIFETIME COST PER VEHICLE MILE (\$/MI)	% OF TOTAL
ROI+E	11.58	1.16	.00012	.1
ACQUISITION				
PURCHASE PRICE	6403.48	640.35	.06403	39.3
FINANCE CHARGE AT INTEREST RATE 12.00% AND TERM 40.0 MOS	1738.04	173.80	.01738	10.7
SALVAGE VALUE 7.0% OF ACQ.	448.24	44.82	.00448	2.8
TOTAL ACQUISITION	7693.28	769.33	.07693	47.3
OWNERSHIP				
MAINTENANCE + REPAIR				
PROPULSION + BATTERY SYSTEMS	600.40	60.04	.00600	3.7
OTHER	937.20	93.72	.00937	5.8
TOTAL M+R	1537.60	153.76	.01538	9.4
TIRE REPLACEMENT	321.65	32.16	.00322	2.0
TIRE LIFE 30000.0 MI.				
ACCESSORIES	274.00	27.40	.00274	1.7
INSURANCE	3750.00	375.00	.03750	23.0
REGISTRATION + LICENSE	170.00	17.00	.00170	1.0
GARAGING, PARKING + TOLLS	7.06	.71	.00007	.0
ENERGY	1500.00	150.00	.01500	9.2
RATE: .050 \$/KWH				
BATTERY REPLACEMENT				
BATTERY	1021.18	102.12	.01021	6.3
FINANCE CHARGE AT INT. RATE 12.00% AND TERM 12.0 MOS	122.54	12.25	.00123	.8
TOTAL BATTERY REPLACEMENT	1143.72	114.37	.01144	7.0
TOTAL OWNERSHIP	6574.63	657.46	.06575	52.7
TOTAL SYSTEM COST	16279.49	1627.95	.16279	100.0

Figure 6-1. Page 4 of 4

SECTION 7

RELIABILITY AND SAFETY

INTRODUCTION

AiResearch considers reliability a major design goal of all programs. A reliability method that will guarantee this goal for this program is based on the technical and program requirements of safety, warranty, weight, performance, and cost. Because of the safety and reliability impact on design, test, failure, and value analysis, AiResearch will design a vehicle that exceeds the 100,000 customer mile requirement (infinite life of 200,000 customer miles for structural safety related items). The methods used to achieve high reliability at minimum cost are:

- Engineering Review Request System--Provides a means of reporting problems or deficiencies uncovered during the design and value analysis reviews.
- Trouble Reporting System--Provides a means of reporting problems, deficiencies and failures uncovered during testing, development and/or durability.
- Reliability Analysis--Provides fault tree and failure mode and effect analysis, which includes all components and systems having a critical or major failure mode, the effect, and hazard classification.
- Reliability Prediction and Reporting--Provides a system approach to total reliability prediction using a minimum sample size to estimate the minimum populating mean (worst-case).
- Reliability of Parts--Provides a means of continuing reliability through selection, application, and control. The part selection is from preferred parts and supplier lists. The part application is analyzed for stress, environmental, operational, and value conditions. The part control is maintained through inspection, installation and in-process testing. In addition, other test methods--durability (in vehicle), fatigue (in laboratory), performance and metallurgical--are used to obtain and maintain part and vehicle reliability.

RELIABILITY ANALYSIS

All failures and their effects will be evaluated for impact on reliability, warranty, and/or safety versus the cost and value required to reduce the probability of failure. A systems approach for vehicle to component level reliability has been initiated. The fault tree comprises the following five major systems:

- Safety/fire (Figure 7-1)
- Powertrain system (Figure 7-2)

Figure 7-1. Safety/Fire Fault Tree Analysis

Figure 7-2. Powertrain System Fault Tree Analysis

- Steering/suspension system (Figure 7-3)
- Brake system (Figure 7-4)
- Body system (Figure 7-5)

RELIABILITY PREDICTION AND REPORTING

The system approach to reliability is continued to the component level. The system reliability is predicted for SAE test standards, FMVSS standards, and fatigue test on the respective components. The component reliability prediction is based on the minimum sample size to closely estimate the population mean for the confidence level indicated on each component sheet (Design Verification and Status Form). The calculation is:

$$P = \frac{N-X}{N}$$

where N = sample size and
 X = number of failures

For the indicated sample size and P, the minimum population mean expected to pass the particular test is shown on the design verification and status form. In general, the maximum population mean expected to pass the particular test is 100 percent. A sample of the Design Verification and Status Form is shown in Figure 7-6. Individual systems and subsystem forms are included in the Appendix.

Figure 7-3. Steering/Suspension System
Fault Tree Analysis

Figure 7-4. Brake System Fault Tree Analysis

Figure 7-5. Body System Fault Tree Analysis

DESIGN VERIFICATION AND STATUS

SYSTEM <u>12.000</u>		ASSEMBLY <u>LIGHTING</u>		PROGRAM <u>ELECTRIC CAR</u>				
SUBSYSTEM <u>12.200</u>		COMPONENT <u>EXTERIOR LIGHT</u>		DATE <u>8/12/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Side Turn Signal Lamp Spec.	SAE J914			20		P1.0 = .83		
Turn Signal Flasher	SAE J590 Life & Performance, 36 hours at 75°F J1055			10		P1.0 = .67		
Hazard Lamp Flasher	SAE J945 Life & Performance, 36 hours at 75°F J1056			10		P1.0 = .67		
Hazard Lamp Switch	SAE J910 Life Test 10,000 Cycles			20		P1.0 = .83		
Parking Lamp Spec.	SAE J222			20		P1.0 = .83		
Back-up Lamp Spec.	SAE J593			20		P1.0 = .83		
Back-up Lamp Switch	SAE J1076 1 Hour Each 75°, 225° and -25°F -- 10 Cycles Each Temperature			20		P1.0 = .83		

All Lighting Tests at 95% Confidence

Figure 7-6. Sample Design Verification and Status Form

APPENDIX
COMPONENT RELIABILITY PREDICTION AND REPORTING INDEX AND FORMS

SYSTEM INDEX FOR THE COMPONENT
DESIGN VERIFICATION AND STATUS RECORDS

System Number	System Description
01.000	Engine assembly
01.100	Flywheel
01.200	Shaft
01.300	Bearing assembly
02.000	Traction motor assembly
03.000	Transmission assembly
03.100	Transmission case
03.200	Gear assembly
03.300	Bearing assembly
04.000	Driveline assembly
04.100	Drive shaft
04.200	"U" joints
04.300	Slip joint
05.000	Rear axle assembly
05.100	Gear assembly
05.200	Side gear assembly
05.300	Axle shaft
05.400	Housing
06.000	Steering system
06.100	Steering column assembly
06.200	Steering gear assembly
06.300	Steering linkage
07.000	Suspension system

System Number	System Description
07.100	Front suspension assembly
07.200	Rear suspension assembly
08.000	Brake system
08.100	Electric brake assembly
08.200	Service brake assembly
08.210	Front brake assembly
08.220	Rear brake assembly
08.300	Parking brake assembly
09.000	Controller system
10.000	Controls
10.100	Accelerator assembly
10.200	Brake assembly
11.000	Electrical distribution
11.100	Battery system
11.200	Alternator system
11.300	Charging system
12.000	Lighting
12.100	Interior lighting
12.200	Exterior lighting
13.000	Wiper/washer assembly
13.100	Wiper
13.200	Washer
13.300	Switch
14.000	Climate control

System Number	System Description
14.100	Heater
14.200	Defroster
15.000	Instruments and instrument panel
16.000	Body in white assembly
17.000	Seats, restraints, and interior trim
18.000	Locks, latches, and mechanisms
19.000	Bumpers and exterior ornamentation
20.000	Glass and mirrors

SYSTEM <u>00.000</u>		ASSEMBLY <u>Vehicle</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE _____				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Electric Car	SAE J 227 Durability Equivalent to 100,000 Customer Miles			10		P.9 = .94		

111-A-4

SYSTEM		ASSEMBLY		PROGRAM	
SUBSYSTEM	TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	RISK ASSESSMENT
01.000	Case Explosion Proof	Retain Flywheel Energy			
	Case Durability	Equivalent to 200,000 Customer Miles			

11-A-5

DESIGN VERIFICATION AND STATUS

SYSTEM <u>01.100</u>		ASSEMBLY <u>Flywheel</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8-17-76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Flywheel Fatigue	Equivalent to 200,000 Customer Miles or 8,000 hr			10		P1.0 = .58		
Pump Fatigue	Equivalent to 200,000 Customer Miles			10		P1.0 = .58		
Seals	Equivalent to 50,000 Customer Miles			10		P1.0 = .58		
Bearings	Equivalent to 50,000 Customer Miles			10		P1.0 = .58		

ENGINE TEST AT 95 PERCENT CONFIDENCE

DESIGN VERIFICATION AND STATUS

SYSTEM <u>02.080</u>		ASSEMBLY <u>Traction Motor</u>		PROGRAM <u>Electric Car</u>		DATE _____		
SUBSYSTEM _____		COMPONENT _____						
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Fatigue	Equivalent to 200,000 Customer Miles							
Brush Fatigue	Equivalent to 25,000 Customer Miles			50		P1.0 = .93		

MOTOR TEST AT 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>01.000</u>		ASSEMBLY <u>Transmission</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE _____				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Transmission Performance	SAE J 651 J 915			3				
Fatigue	Equivalent to 100,000 Customer Miles			5		P1.0 = .63		
Gear Noise				10		P1.0 = .79		

TRANSMISSION TESTS AT 80 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>04.000</u>		ASSEMBLY <u>Driveline</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE _____				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Fatigue	100,000 Cycles (Ref. J 901)			30		P1.0 = .88		
Resonant Bending	None in Drive Range			3		P1.0 = .40		

DRIVELINE TEST A" 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>C5.000</u>		ASSEMBLY <u>Rear Axle</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8/17/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Fatigue Noise	Equivalent to 100,000 Customer Miles			5		P1.0 = .63		

REAR AXLE TEST AT 80 PERCENT CONFIDENCE

DESIGN VERIFICATION AND STATUS

SYSTEM <u>06.00</u>		ASSEMBLY <u>Steering Column</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8/17/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Shaft Fatigue	Equivalent to 200,000 Customer Miles			20		P1. = .82		
Rag Joint Fatigue	Equivalent to 200,000 Customer Miles			20		P1 = .82		

STEERING COLUMN TEST AT 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>06.200</u>		ASSEMBLY <u>Steering Gear</u>		PROGRAM <u>Electric Gear</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8/17/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Sector Shaft Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Housing	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		

STEERING TEST AT 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>06.300</u>		ASSEMBLY <u>Steering Linkage</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8/17/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Center-link Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Tierod Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Pitman Arm Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Idler Arm Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		

STEERING TESTS AT 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>07.100</u>		ASSEMBLY <u>Front Suspension</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8/17/75</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Spring Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Ball Joint Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Drag Strut Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
"A" Arm Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Shock Fatigue	Equivalent to 20,000 Customer Miles			10		P.8 = .43		
Spindle Fatigue	Equivalent to 200,000 Customer Miles			30		P1.0 = .88		

SUSPENSION TESTS AT 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>07.200</u>		ASSEMBLY <u>Rear Suspension</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM <u>1</u>		COMPONENT _____		DATE <u>8/18/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Spring Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Fixed Eye Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Shackle Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Shocks	Equivalent to 20,000 Customer Miles			20		P.8 = .43		

SUSPENSION TEST AT 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>08.000</u>		ASSEMBLY <u>Brake Assembly</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM <u></u>		CONFIGURE <u>4T</u>		DATE <u>8/2/75</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Fluid Test	SAE J 291 J 1722 J 1733 FMVSS 116							
Master Cylinder	SAE J 66			10		P1.0 = .6E		
Brake Tubes and Lines	SAE J 1401 J 104E			100		P1.0 = .96		
Brake Lining Test	SAE J 160 J 379, 380 J 660, 866 J 998, 840			100		P1.0 = .96		
Brake Tube and Line	Performance FMVSS 106			100		P1.0 = .96		

ALL BRAKE SYSTEM COMPONENTS AT 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>8.200</u>		ASSEMBLY <u>Brake Assembly</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8/12/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Service Brake Test	SAE J 658 J 135 J 299 J 843 J 345 J 229 J 537			100		P1.0 = .96		
Service Brake Performance	FMVSS 105			3		P1.0 = .52		

ALL BRAKE SYSTEM COMPONENTS AT 95 PERCENT CONFIDENCE

DESIGN VERIFICATION AND STATUS

SYSTEM <u>8.200</u>		ASSEMBLY <u>Service Brake Assembly</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM <u>8.210</u>		COMPONENT <u>Front Brake</u>		DATE <u>8-12-76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Disc Test	SAE J 02 Seal Test Life			100		$P_{1.0} = .96$		

ALL BRAKE SYSTEM COMPONENTS AT 95 PERCENT CONFIDENCE LEVEL.

DESIGN VERIFICATION AND STATUS

SYSTEM <u>08.203</u>		ASSEMBLY <u>Service Brake Assembly</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM <u>08.220</u>		COMPONENT <u>Rear Brake</u>		DATE <u>8-12-76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Brake Drum Test	SAE J 65 Seal Life J 101 Wheel Cylinder			100		P1.0 = .96		

ALL BRAKE SYSTEM COMPONENTS AT 95 PERCENT CONFIDENCE

SYSTEM <u>8.300</u>		ASSEMBLY <u>Parking Brake</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8-12-76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Parking Brake Test	SAE J 360 J 293			20		P1.0 = .82		

111-A-20

DESIGN VERIFICATION AND STATUS

SYSTEM <u>08.400</u>		ASSEMBLY <u>Wheel and Tires</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8-12-76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN- LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Wheel Test	SAE J 175 J 267 J 328 FMVSS 110			36		P1.0 = .89		
Tires Test	SAE J 332,345 J 918,966 J 1060 FMVSS 109			36		P1.0 = .89		

ALL BRAKE COMPONENTS AT 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>09.000</u>		ASSEMBLY <u>Controller</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM <u></u>		COMPONENT <u></u>		DATE <u></u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Controller Fatigue	Ecuivalent to 200,000 Customer Miles or 6,000 MTBF			6		P1 = .50		

CONTROLLER TEST AT 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>10.000</u>		ASSEMBLY <u>Controls</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8-17-76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Accelerator Shutdown	FHVSS 124							

ACCELERATOR CONTROL TEST AT 95 PERCENT CONFIDENCE LIMITS

DESIGN VERIFICATION AND STATUS

SYSTEM <u>11.000</u>		ASSEMBLY <u>Elect. Dist.</u>		PROGRAM <u>Electric Car</u>		DATE <u>8-12-76</u>		
SUBSYSTEM _____		CONFONE _____						
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
EMI RFI	SAE J 55 J 113 Canada FMVSS			5		P1.0 = .55		
Fuses Circuit Breaker Life	SAE J 553 J 554 6 hr Vibration 4 hr Life 50,000 Cycles			100		P.6 = .53		
Wiring, P.C.	SAE J 556 SAE J 771 Life and Durability Test			10		P.80 = .55		
On-Off Switch Life	SAE J 259 Life Test, 1 hr Each 75°F, 165°F, -25°F -25°F, 0°F 10 Cycles Each Temp			10		P.80 = .55		

FMVSS TEST AT 90 PERCENT CONFIDENCE

DESIGN VERIFICATION AND STATUS

SYSTEM <u>11.000</u>		ASSEMBLY <u>Electrical</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM <u>11.100</u>		COMPONENT <u>Battery</u>		DATE <u>8-12-76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESLTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Battery Acceptance	Life Test SAE J 240 Vibration Test J 537			10 10		P1.0 = .79 P1.0 = .79		
Battery Grounding	V _d < .2v @ 68°F SAE J 538			10		P1.0 = .79		
Battery Discharge Life	1500 Cycles			50		P1.0 = .93		

BATTERY TEST AT 95 PERCENT CONFIDENCE

11-A-26

DESIGN VERIFICATION AND STATUS

SYSTEM <u>12.000</u>		ASSEMBLY <u>Lighting</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM <u>12.200</u>		COMPONENT <u>Exterior Light</u>		DATE <u>D/12/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Lamp Retention	SAE J 567 Socket Load			50		P1.0 = .93		
Lamp Connector	SAE J 856							
Lamp Switch Life	SAE J 253 J 564 Life Test 100,000 Cycles			20		P1.0 = .89		
Lamp Spec.	SAE J 571, 50 573, 249 575, 36 576, 29 578, 77 579, 32 580, 33 582 FMVSS -108			50		P1.0 = .93		
Stop and Tail Lamp Spec.	SAE J 585 J 586			20		P1.0 = .89		
Stop Switch	SAE J 249 Life 250,000 Cycles			20		P1.0 = .89		
License Plate Lamp Spec.	SAE J 587			20		P1.0 = .89		
Turn Lamp Spec.	SAE J 588			20		P1.0 = .89		
Turn Signal Switch Spec.	SAE J 589 Life 165,000			20		P1.0 = .89		

ALL LIGHTING TEST AT 95 PERCENT CONFIDENCE

DESIGN VERIFICATION AND STATUS

SYSTEM <u>12.000</u>		ASSEMBLY <u>Lighting</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM <u>12.250</u>		COMPONENT <u>Exterior Light</u>		DATE <u>8-12-76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Side Turn Signal Lamp Spec.	SAE J 514			20		P1.0 = .83		
Turn Signal Flasher	SAE J 590 Life and Performance J 1055 36 hr at 75°F			10		P1.0 = .67		
Hazard Lamp Flasher	SAE J 945 Life and Performance J 1056 36 hr at 75°F			10		P1.0 = .67		
Hazard Lamp Switch	SAE J 910 Life Test 10,000 Cycles			20		P1.0 = .83		
Parking Lamp Spec.	SAE J 222			20		P1.0 = .83		
Back-Up Lamp Spec.	SAE J 593			20		P1.0 = .83		
Back-Up Lamp Switch	SAE J 1C76 1 hr Each 75°F, 225°F, and -25°F; 10 Cycles Each Temp			20		P1.0 = .83		

ALL LIGHTING TEST AT 95 PERCENT CONFIDENCE

DESIGN VERIFICATION AND STATUS

SYSTEM 13.000		ASSEMBLY Wiper/Washer		PROGRAM Electric Car				
SUBSYSTEM 13.300		COMPONENT Switch		DATE 8/12/76				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Wiper Switch Spec.	SAE J 112 Life Test 10,000 Cycles			20		F1.0 = .83		
Washer Switch Spec.	SAE J 234 Life Test 10,000 Cycles			20		F1.0 = .83		
Wiper Performance	1,500,000 Cycles SAE J 903			20		P1.0 = .83		
Washer Performance	SAE J 942 8000 Cycles			20		P1.0 = .83		

WIPER/WASHER TEST AT 95 PERCENT CONFIDENCE

DESIGN VERIFICATION AND STATUS

SYSTEM <u>14.000</u>		ASSEMBLY <u>Climate Control</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT <u>Switch</u>		DATE <u>8/12/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Blower Motor Switch	SAE J 235 Life Test 10,000 Cycles			20		P.7 = .53		
Defroster Test	SAE J 902 Performance FMVSS 103			1				
Backlight Test	SAE J 953 Performance			1				

BLOWER SWITCH TEST AT 80 PERCENT CONFIDENCE

ARESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

DESIGN VERIFICATION AND STATUS

SYSTEM <u>16.000</u>		ASSEMBLY <u>Eody</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM <u></u>		COMPONENT <u></u>		DATE <u>8/12/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Body Test	SAE J 963, 22#			5		P1.0 = .63		
	138, 88#							
	211, 97#							
	850, 85#							
	996, 104#							
	374, 36#							
	136, 98#							
	921, 34#							
	944, 5#							
	338							
	FMVSS 215			3				
	FMVSS 203							
	FMVSS 216							

BODY TEST AT 80 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>17.000</u>		ASSEMBLY <u>Seat Belts</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8/17/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Seat Belt Test	SAE	J 140		20		P1.0 = .68		
		J 141						
		J 117						
		J 787						
		J 114						
		J 128						
Seat Measure	SAE	J 826		20		P1.0 = .68		
Seat Test	SAE	J 879		20		P1.0 = .68		
		J 388						
		J 954						
		J 815						
Seat Belt Test	Ft-VSS 207			20		P1.0 = .68		
	Ft-VSS 208							
	Ft-VSS 209							
	Ft-VSS 210							

SEAT AND SEAT RESTRAINT TESTS AT 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM 18.000		ASSEMBLY Locks and Latches		PROGRAM Electric Car				
SUBSYSTEM		COMPONENT		DATE 8/17/76				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Hinge Sysyten	SAE J 934 2500 lb Load FMVSS - 206			20		P1.0 = .82		
Door Latch	SAE J 339 Static Load FMVSS -206			20		P1.0 = .82		
Hood Latch	SAE J 352 Static Load			20		P1.0 = .82		

LATCH SYSTEM TEST AT 95 PERCENT CONFIDENCE LEVEL

[illegible]

11-A-33

DESIGN VERIFICATION AND STATUS

SYSTEM <u>20.000</u>		ASSEMBLY <u>Glass and Mirrors</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8/17/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Glass Drop Test	SAE J 933			10		P.3 = .68		
Mirror Test	SAE J 954 J 955 J 834			3		P1.C = .80		

GLASS AND MIRROR TESTS AT 50 PERCENT CONFIDENCE LEVEL