Assistant Secretary for Conservation and Renewable Energy
Office of Transportation Programs
Under Contract No. DE-AC03-76CS51213

MASTER

Near-Term Electric Test Vehicle

ETV-2

Phase II — Final Report

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ETV-2

Phase II — Final Report

April 1981

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Under Contract No. DE-AC03-76CS 51213

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1. EXECUTIVE SUMMARY

1.1 PROGRAM ACCOMPLISHMENT

A unique battery-powered passenger vehicle has been developed that provides a significant improvement over conventional electric vehicle performance, particularly during stop-and-go driving. The vehicle is unique in two major respects: (1) the power system incorporates a flywheel that stores energy during regenerative braking and makes possible the acceleration capability needed to keep up with traffic without reducing range to unacceptable values; and (2) lightweight plastic materials are used for the vehicle unibody to minimize weight and increase range.

These features were analyzed and demonstrated in an electric test vehicle, designated ETV-2, which was designed, manufactured, and tested by the AiResearch Manufacturing Company, a division of The Garrett Corporation, before delivery to the Department of Energy (DOE) under Contract DE-AC03-76CS51213. Characteristics of this vehicle are summarized in Figure 1-1.

The work was accomplished as part of the DOE Near-Term Electric Vehicle (NTEV) Program. The overall objective of the program was to develop the performance potential and economic viability of an advanced electric vehicle that could be put into production in the 1980's. The objective of the Phase II effort, under which the vehicle was built and tested, was to proceed from the preliminary design of Phase I through a proof-of-concept test vehicle demonstration.

The ETV-2 project was supported by several major subcontracts:

- Vehicle Design--The Brubaker Group
- Vehicle Structural Design and Fabrication and Scale-Model Crash Testing--Budd Company
- Advanced Lead-Acid Battery Development--Eagle-Picher Industries
- Safety and Handling Analyses--Dynamic Sciences, Inc
- Selection of Suspension, Steering, and Brake Systems--All American Racers

1.2 FLYWHEEL AS A SOLUTION TO BATTERY LIMITATIONS

The driving range of various existing prototype electric vehicles that use lead-acid traction batteries has been claimed to be 161 km (100 mi) or

VEHICLE CHARACTERISTICS	
Overall Length	4.19 m (13.75 ft)
Width	1.68 m (5.5 ft)
Height	1.38 m (4.54 ft)
Wheelbase	2.41 m (7.92 ft)
Curb weight	1507 kg (3323 lb)
Gross weight with four passengers	1779 kg (3923 lb)
Vehicle effective drag area, $C_d A$	0.74 m ² (8.0 ft ²)
Rolling resistance coefficient	0.0075
Body/chassis type	Unibody, fiber-reinforced plastic

PERFORMANCE	
SPEED	
Cruise speed	89 km/h (55 mph)
Max. speed	100 km/h (62 mph)
Speed up 5-percent grade	80 km/h (50 mph)
ACCELERATION TIME	
0 to 48 km/h (30 mph)	8 s
0 to 80 km/h (50 mph)	16 s
40 to 89 km/h (25 to 55 mph)	10 s
RANGE (4 PASSENGERS)	
Federal Urban Driving Cycle	106 km (66 mi)
J227a(D) Driving Cycle	109 km (68 mi)

POWER SYSTEM CHARACTERISTICS

Number of batteries	18
Battery weight	484 kg (1066 lb)
Battery capacity (3-hr rate)	18.72 kW·h
Battery voltage	108 Vdc
Battery recharge time	8.3 h
Flywheel capacity	1.0 kW·h
Flywheel max. speed	24 000 rpm
Motor/generator rating	21 kW peak 11 kW continuous 0 to 11 650 rpm

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Figure 1-1. ETV-2 Summary

more, but only during constant low-speed cruising. When multiple accelerations in stop-and-go driving are included, the range is reduced to as low as 48 km (30 mi) due to the inherent characteristics of the lead-acid batteries, i.e., the usable capacity is an inverse function of the power demand. This limited range is a major obstacle to public acceptance of electric vehicles.

An equally important drawback to the acceptance of conventional electric vehicles is their limited acceleration capability, especially at cruising speeds of 48 km/h (30 mph) and above. Without sufficient acceleration, the electric vehicle will not be compatible with city traffic and may never be qualified for freeway driving. This limitation is aggravated by the inverse relationship between battery current and the terminal voltage, which limits maximum battery power. The available power also decreases as the battery is discharged.

Increased component efficiencies would increase range to a limited degree, but would not significantly improve acceleration. The lead-acid battery alone has not been able to provide acceptable acceleration as well as range.

New technology was used in applying the flywheel as a solution to the range vs acceleration dilemma. The successful development of a lightweight flywheel made from a high-strength composite material solved the weight problem that had hindered the use of energy storage flywheels in vehicles. A new transmission was conceived to match the power flow from a decelerating flywheel to the power required for accelerating the vehicle. A new power system was designed using the flywheel to supplement the battery during peak power conditions, allowing the battery to operate in its most efficient mode, i.e., providing a constant average sustaining current without power peaks. This approach offers the following advantages:

- (a) Extended range in stop-and-go driving
- (b) Range that is essentially independent of acceleration rates
- (c) Acceleration power that is independent of the number (weight) of batteries, state of battery discharge and battery temperature
- (d) Regenerative braking
- (e) Extended battery life

1.3 VEHICLE DESCRIPTION

The ETV-2 is similar to many conventional vehicles, but has significant features that distinguish it from any known electric or conventional internal-combustion-engine automobile. The vehicle power system includes a battery pack, a power unit, and various electronic components that provide system control based on operator commands and power system status. A unibody concept is used; all body panels and structure trim are fabricated from fiber-reinforced plastic. The curb weight of 1507 kg (3323 lb) is 272 kg (600 lb) less than

that of a comparable vehicle with a steel body. All Federal Motor Vehicle Safety Standards have been met except for light stability characteristics of the acrylic glazing in the side and rear windows. Visibility through the acrylic is more than adequate; therefore, minimizing weight was considered more important than this particular requirement.

1.3.1 Power System

The power system consists of a battery pack of eighteen 6-V batteries, a power unit, electronic controls, and a battery charger. The rear-mounted power unit, an assembly of the flywheel, two identical motor/generator units, and a planetary transmission, supplies power to the axle through a silent chain drive. A controller and a power conversion unit (PCU) perform the logic and control functions, and the onboard battery charger allows the batteries to be recharged by plugging into a 115-V, 60-Hz, ac electrical outlet. Figure 1-2 shows the interconnection of power system components.

Successful implementation of the regenerative braking, battery load leveling, and high acceleration capability features of the system hinges on the proper transfer of power among the battery pack, flywheel, generator, motor, and axle. The desired power flow is achieved by a controller that contains a microprocessor programmed to compare driver commands with vehicle and component status and to issue appropriate commands to the system. Power flow for various driving conditions is shown in Figure 1-3.

Figure 1-2. Interconnection of Power System Components

Figure 1-3. Power Flow Through the Power Unit

The power system is cooled by free convection and forced air: heat generated by the bearings and gears is dissipated from the transmission housing through free convection, and the PCU and motor/generator units are forced-air-cooled under all operating conditions. A fan cools the battery charger electronics and simultaneously ventilates the battery tunnel during the battery-charging mode. The controller is cooled by free convection through the built-in controller housing heat sink.

The two motor/generator units are nearly identical; each unit operates either as a motor or as a generator, depending on vehicle status and driving commands. Throughout this report, the terms "motor" and "generator" are used to designate the unit, not to designate a function. Accordingly, the unit geared directly to the planetary carrier shaft and the final drive is referred to as the "motor" although the unit may be generating electric power as it does during regenerative braking. The other unit, geared to the planetary ring gear, is referred to as the generator.

1.3.2 Battery Pack

Primary power is supplied by eighteen 6-V batteries with an energy density of 39.8 W·h/kg (18.1 W·h/lb). These golf-cart-sized batteries were developed by Eagle-Picher Industries specifically for the AiResearch ETV-2. The batteries are arranged in two rows in a removable tray that is housed in a tunnel extending from the front of the vehicle to a point between the rear seats (Figure 1-4).

Figure 1-4. Battery Installation

1.3.3 Power Unit

The vehicle is propelled by the power unit (Figure 1-5), which is a single assembly installed in the rear of the vehicle and connected to the rear axle through a silent chain drive. The flywheel and two motor/generator units are mechanically linked through a planetary transmission lubricated by an electric motor-driven pump. The flywheel operates in a near-vacuum ($1 \mu\text{m HgA}$) produced by a two-stage pumping arrangement: (1) an electric-motor-driven, vane-type fore pump mounted adjacent to the power unit, and (2) a drag-type molecular pump consisting of a smooth disc attached to the flywheel and a stator that is part of the housing.

1.3.4 Controls and Electronics

1.3.4.1 Controls

The automatic controls monitor and process vehicle operating conditions and driver inputs; perform computations and generate commands for the power unit, switchgear, and dashboard indicators; and monitor power system conditions to provide protective shutdown or signals in the event of excessive speeds, pressures, currents, or temperatures. The main elements of the system controls are shown in Figure 1-6.

1.3.4.2 Power Conversion Unit (PCU)

The PCU, installed behind the rear seat of the vehicle as shown in Figure 1-7, conditions battery power and applies it selectively to the power unit motors as commanded by the controller. The accessory power supply and lubrication and vacuum pump controls also are part of the PCU. The PCU monitors and protects the power unit, and feeds status information back to the controller.

1.3.4.3 Controller

A solid-state electronic controller provides all power system control. It senses the operating state of the vehicle and power unit and controls the flow of power among the flywheel, generator, motor, and battery in response to driver commands. It provides a smooth transition between operating modes and eliminates acceleration jerks and large current transients to enhance driving comfort and increase the power system switch gear service life. In addition, the controller automatically limits power unit speeds and temperatures, drives the dashboard display, and formats and outputs selected data parameters to the data recorder. The controller, the charger, and the data recorder are mounted in the vehicle front compartment (Figure 1-8).

1.3.4.4 Charger

The onboard battery charger will recharge the battery pack from either 15- or 30-A, 115-Vac household service, which is switch selectable at the charger. With the 30-A current, the battery pack may be recharged overnight.

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Figure 1-5. Power Unit Installation

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Figure 1-6. System Control Concept

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Figure 1-7. Power Conversion Unit Installation

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Figure 1-8. Front Compartment Electronics

1.3.4.5 Data Recorder

The data recorder provides the means for recording selected vehicle parameters such as speed, battery current and voltage, and several temperatures on cassette tape during vehicle operation. The recorded data are processed in a separate computer facility. The recorder is intended for vehicle demonstration and evaluation and would not be included as part of production vehicles.

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1.3.5 Body, Chassis, Suspension

The four-passenger ETV-2 styling is simple, clean, and attractive, and provides good aerodynamic characteristics as shown in Figure 1-9. The ample body width, along with the battery tunnel through the center of the vehicle, creates a feeling of interior roominess (Figure 1-10).

ETV-2 unibody structure is primarily fiber-reinforced plastic. The glazing is acrylic except for a conventional safety glass windshield.

With the all-plastic body, particular emphasis was placed on design for crashworthiness. In a crash situation, the battery tray will de-couple from the tunnel so that the vehicle structure will not be forced to absorb the kinetic energy of the batteries. Energy-absorbing, foam-filled, corrugated tubes are incorporated in the front and rear to provide crush resistance.

Suspension components are from the BMW 320i. The 320i is about the same size as the ETV-2, and its four-wheel suspension fits the packaging requirements of the battery tunnel.

1.3.6 Brakes

The ETV-2 has separate electrical regenerative and hydraulic braking systems. Signals from potentiometer pickups on the throttle and brake pedals apply regenerative braking, which transfers vehicle kinetic energy to the flywheel. Conventional hydraulic brakes are activated only when the brake pedal travel exceeds approximately 48 mm (1.9 in.). The parking brake operates mechanically on the rear wheels.

1.3.7 Driver Controls and Displays

Driver controls and displays, shown in Figure 1-11, are very similar to those in conventional vehicles, thus permitting a new driver to operate the ETV-2 with little or no instruction.

1.4 VEHICLE OPERATION

Operation of the ETV-2 is very similar to that of an internal combustion engine (ICE) vehicle with an automatic transmission. The driver selects the desired mode of operation, either drive, neutral, reverse, or park, using the selector lever. Acceleration and braking rates are controlled by the throttle and brake. The regenerative braking capability simulates the feel of compression braking when both the brake and throttle are off.

The principal differences between an ICE vehicle and the ETV-2 are in battery charging and in vehicle startup, which, for the ETV-2, involves spinning up the flywheel prior to driving.

Operation of the power system within the four basic conditions--nonoperation, prestart and idle, forward driving, and reverse--is governed by various automatic control modes within the controller, which are determined essentially by comparing motor and generator speeds with their base speeds. The driver will be generally unaware of the transition between automatic modes.

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Figure 1-9. Simple, Clean Appearance of the ETV-2

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Figure 1-10. ETV-2 Interior

① TURN SIGNAL SWITCH	⑪ HOOD RELEASE HANDLE (UNDER DASH)
② WARNING LIGHTS	⑫ HORN BUTTON
③ BATTERY POWER METER	⑬ BRAKE PEDAL
④ TURN SIGNAL/HAZARD WARNING INDICATOR LIGHT	⑭ THROTTLE PEDAL
⑤ HIGH BEAM INDICATOR LIGHT	⑮ PARKING BRAKE LEVER
⑥ SPEEDOMETER	⑯ HEATER AND VENTILATION CONTROL
⑦ FLYWHEEL POWER METER	⑰ DRIVE MODE SELECTOR LEVER
⑧ WARNING LIGHTS	⑱ ASH TRAY
⑨ WINDSHIELD WIPER AND WASHER SWITCH	⑲ GLOVE COMPARTMENT
⑩ AIR OUTLETS	

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Figure 1-11. Instruments and Controls

1.5 PERFORMANCE

ETV-2 performance was analyzed with the aid of a test-correlated computer simulation because all of the driving tests were eliminated from Phase II of the program to permit more extensive power system development. The computer simulation consisted of detailed performance models of each of the system components.

As shown in Table 1-1, all of the goals were met or exceeded except for the range for the J227a(D) driving cycle and the sustained speed on a 5-percent grade. This is largely offset by the range and acceleration performance under the more realistic Federal Urban Driving Schedule (FUDS). Although the FUDS requires higher accelerations and more stops, the range is approximately the same. This is a very important characteristic of the flywheel system: the range tends to be independent of the acceleration and power used.

One of the most important characteristics of the flywheel system is that the vehicle range is essentially independent of the acceleration demands; therefore, vehicle range will be maximized for a given battery capacity. This characteristic is demonstrated by the results of a simulated FUDS conducted on the vehicle dynamometer. Figure 1-12 shows the nearly random variation of vehicle speed with time that is typical of urban driving, together with the resulting flywheel speed and battery current. The nearly constant battery current indicates that the flywheel is providing the power for all of the acceleration demands and that deceleration and braking energy is continually utilized to reenergize (speed up) the flywheel by regenerative braking.

Figure 1-12. ETV-2 Power System Performance During FUDS

TABLE 1-1
ETV-2 PREDICTED PERFORMANCE

		Final Performance	
		Four Passengers (600 lb)	One Passenger (150 lb)
(English Units)	DOE Goal		
Range, J227a(D) cycle, mi	75	68	74
Range, FUDS, mi	*	66	72
Range, 25-mph cruise, mi	*	117	127
Range, 35-mph cruise, mi	*	105	114
Range, 45-mph cruise, mi	*	77	84
0 to 30 mph acceleration, s (19 000 rpm)**	9.0	8.3	7.4
25 to 55 mph acceleration, s (23 000 rpm)	18.0	12.9	11.2
0 to 50 mph acceleration, s (21 000 rpm)	*	17.1	15.2
Sustained speed on 5-percent grade, mph	50 for 1 mi	50 for 0.8 mi balance at 40	50 for 1 mi
(SI Units)	DOE Goal	Four Passengers (272 kg)	One Passenger (68 kg)
Range, J227a(D) cycle, km	121	104	119
Range, FUDS, km	*	106	116
Range, 25-mph cruise, km	*	188	204
Range, 35-mph cruise, km	*	169	183
Range, 45-mph cruise, km	*	124	135
Sustained speed on 5-percent grade, km/h	80 for 1.6 km	80 for 1.3 km, balance at 64	80 for 1.6 km

*Not specified

**Initial flywheel speeds

Acceleration capability is presented graphically in Figure 1-13, which shows that the vehicle has the traffic compatibility that was one of the primary objectives of the demonstration program.

*J227a(D) acceleration interpreted by JPL for minimum required power.

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Figure 1-13. Acceleration Performance Comparisons

Improvement in range can be expected in future vehicles over this proof-of-concept, experimental model. For example, a 10 percent reduction in weight and aerodynamic drag and a reduction in flywheel size could result in an increase in FUDS range from 106 to 122 km (66 to 76 mi).

2. INTRODUCTION

This document was prepared by AiResearch Manufacturing Company, a division of The Garrett Corporation, under Contract DE-AC03-76CS51213 for the Department of Energy (DOE). It presents the final report for Phase II of the Near-Term Electric Vehicle (NTEV) program.

2.1 PHASE I CONCLUSIONS AND RECOMMENDATIONS.

In Phase I of this contract, the NTEV goals were studied and preliminary designs of the vehicle and power systems were analyzed. Phase II was a continuation of the Phase I preliminary design study and was based on the development plan prepared during Phase I and on the following Phase I conclusions and recommendations.

Phase I Conclusion--A baseline design for an electric-powered passenger vehicle was defined that would effectively meet DOE near-term performance, weight, cost, and safety objectives. The baseline design reflected the following study conclusions:

- (a) A regenerative power system would be required to meet the near-term goals of the program.
- (b) A lightweight vehicle structure and body would be required to meet the cost and energy consumption goals of the program.
- (c) Using a flywheel for acceleration power would substantially increase the acceleration performance of the vehicle, leading directly to greater consumer acceptance.
- (d) Increasing the cycle life of the batteries would be necessary to ensure attainment of the predicted life-cycle cost of vehicle operation.
- (e) The cost goals would be attainable, provided the battery cost penalty could be reduced sufficiently.

Phase I Recommendations--The recommendations for future development activities included:

- (a) A 54-month development program was 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 was strongly recommended to ensure optimum utilization of the potential improvements in the critical area of the battery.

2.2 PHASE II PROGRAM OBJECTIVES

The overall objective of the DOE Near-Term Electric Vehicle program is to develop the performance potential and economic viability of an advanced electric vehicle that could be put into production in the 1980's. The objective of the Phase II effort was to proceed from the Phase I preliminary design through a proof-of-concept test vehicle demonstration. The primary technical objective was the development of an electric vehicle incorporating a flywheel regenerative power system and a body constructed from lightweight, fiber-reinforced, composite material.

The primary products of this effort were the completion and in-depth analysis and test of an integrated test vehicle, designated Electric Test Vehicle No. 2 (ETV-2), for delivery to DOE.

2.3 PROGRAM GOALS

The specific goals of the NTEV program are delineated in Table 2-1. The achievement of these goals would represent a significant improvement over existing electric vehicle performance, particularly during stop-and-go driving.

2.4 VEHICLE DESIGN CONCEPT

The ETV-2 is an all new design, highly optimized for the particular requirements of a small, four-passenger, electric-powered urban/suburban automobile. The power system, a new and unique design that uses flywheel energy to supplement battery power during peak power demands, incorporates regenerative braking to convert vehicle kinetic energy into retrievable flywheel energy during deceleration and braking. The process of energy storage and conversion to propulsive power is accomplished by the unique arrangement of a flywheel, two motor/generator units, and a power drive connected through a differential planetary gearset that functions as a fully automatic transmission with infinitely variable gear ratio. The result is a highly efficient electric vehicle that requires a minimum number of batteries while providing acceptable performance and driving characteristics that are similar to those of conventional, internal-combustion-engine-powered automobiles.

2.5 APPROACH

The performance characteristics of lead-acid traction batteries were fairly well known although detailed information on certain limiting functions was lacking. The actual driving range of various existing prototype electric vehicles was claimed to be 100 miles or more, but only during constant low-speed cruising. When multiple accelerations in stop-and-go driving were included, the typical range was reduced to a fraction of this value, or to as low as 30 miles. This is due to the inherent characteristics of the lead-acid batteries, i.e., the usable capacity is an inverse function of the power demand.

The driving range limitation has been considered the major obstacle to public acceptance of the electric vehicle. Of equal importance, however, is the acceleration power, especially at cruising speeds of 30 mph and above.

TABLE 2-1

DOE NEAR-TERM ELECTRIC VEHICLE PROGRAM OBJECTIVES

Parameter	Near-Term DOE Objectives
Minimum passenger capacity	4 adults
Maximum curb weight, lb	TBD
Minimum urban range (J227a(D)), mi	75
Maximum initial cost, projected, 1975 dollars	5000
Minimum life, mi	100,000
Minimum life, yr	10
Maximum life-cycle cost, projected in 1975 dollars/mi	0.15
Cost of energy in dollars/kw-hr	0.05
Maximum electric recharge energy in urban driving, kw-hr/mi	0.5
Maximum recharge time, hr (115-Vac, 30-amp service)	6
Minimum top passing speed, mph	60
Minimum top cruising speed, mph	55
Minimum accessories	Heater/defroster, onboard charger
Safety features	FMVSS* requirements at time of contract
Minimum unserviced park duration, days	7
Maximum years until production ready	5
Maximum critical materials required	Few
Minimum acceleration (0 to 30 mph), sec	9
Maximum merging time (25 to 55 mph), sec	18
Sustained speed on 5-percent one-mile grade, mph	50
Maximum scheduled maintenance cost, cents/mi	2
Minimum ambient temperature range, °F	-20 to +125
Interior noise	Minimum
Turning and braking	No power assist required

*Federal motor vehicle safety standards

Without sufficient acceleration power, the electric vehicle will not be compatible with city traffic and may never be qualified for freeway driving. This limitation is aggravated by another characteristic of the lead-acid battery, the inverse relationship between battery current demands and the terminal voltage, which limits the maximum power that can be provided by the battery. The available power also decreases as the battery is discharged.

With these battery limitations, the only means of increasing performance of the conventional battery-powered system was to increase component efficiencies, which could improve the driving range to a limited degree, but would not significantly improve acceleration performance. The improvement needed was to be able to achieve the 75-mi range while also providing acceleration compatible with normal city traffic. The lead-acid battery alone cannot provide this performance.

The flywheel often has been proposed as a solution to this problem. Two obstacles, however, have stood in the way of its use: (1) the excessive weight of the required flywheel assembly, and (2) the lack of a suitable transmission with infinitely variable ratio that could efficiently transform the kinetic energy stored in the flywheel into propulsive power delivered to the wheels of the car.

The NTEV program was based on newly demonstrated technology that provided solutions to each of the preceding problems. First, the successful development of a lightweight flywheel made from a high-strength composite material had been recently demonstrated in the laboratory. Second, a new power transmission was conceived to match the power flow from a decelerating flywheel to the power required for accelerating the vehicle. Both were proprietary developments of the Garrett Corporation that have been the subject of various patents and patent applications. The developments were sufficiently mature in early 1976 so that a vehicle utilizing them could be built and demonstrated as a part of the DOE NTEV program.

The primary function of the NTEV program was to design and demonstrate a flywheel auxiliary energy storage device that would increase electric vehicle performance potential beyond that which is obtainable from batteries alone. The approach taken in the program, therefore, was to design an entirely new power system using a flywheel to supplement the battery during peak power conditions, allowing the battery to operate in its most efficient mode: providing a constant average sustaining current without power peaks. This approach offers the following advantages:

- (a) Extended range in stop-and-go driving.
- (b) Range that is essentially independent of acceleration rates
- (c) Acceleration power that is independent of the number (weight) of batteries
- (d) Acceleration power that is independent of the battery state of discharge

- (e) Acceleration power essentially independent of low battery temperatures
- (f) Regenerative braking
- (g) Extended battery life

3. PROGRAM REVIEW

3.1 PROGRAM SCOPE

The original objectives of Phase II of this program were to build and test one "mule" car for engineering development and test, one frame and body for crash test demonstration, and one final vehicle for delivery and test evaluation and demonstration.

After the first power system and the first vehicle body were completed, it became apparent that increased development time and effort on the power system (primarily in the electronics) would be beneficial. Furthermore, it was realized that the most important objective of the program--to evaluate the advantage of the new power system--could be accomplished with a single vehicle.

Consequently, at DOE direction, the contract was amended to limit program costs while allowing more development effort on the power system by producing only one completed vehicle and a second, spare power system, complete with all electronics components. The test program also was amended to eliminate performance testing and road testing; final performance testing and evaluation will be done by JPL. The vehicle built by AiResearch was designated Electric Test Vehicle No. 2 (ETV-2) by DOE.

3.2 PROGRAM MANAGEMENT

3.2.1 Department of Energy Management

The activities described in this report were carried out by the AiResearch Manufacturing Company of California and its subcontractors under a contract with DOE. This work is part of the NTEV program being administered by DOE in accordance with Public Law 94-413, the Electric and Hybrid Vehicle Research, Development and Demonstration Act of 1976. Management responsibility for the program is within the Division of Transportation Energy Conservation under the Assistant Secretary for Conservation and Solar Applications (Figure 3-1).

Technical management of the AiResearch contract was the responsibility of the Jet Propulsion Laboratory (JPL) of Pasadena, California. Mr. Thomas A. Barber is Manager of the Electric and Hybrid Vehicle Systems Project at JPL. The Contract Technical Manager for the AiResearch contract was Mr. Marc Trummel. Mr. Steve DeGrey was the resident JPL representative.

3.2.2 AiResearch Management

Within AiResearch, the work was conducted by the engineering department of the Mechanical Power Systems Product Line, one of five major product lines

Figure 3-1. DOE Organizational Elements

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that are part of a matrix that combines product lines with functional departments. The relationship of the NTEV program to AiResearch management is shown in Figure 3-2.

Program tasks were assigned as indicated in the NTEV program chart of Figure 3-3.

The program was supported by several major subcontracts. The vehicle design was created by The Brubaker Group. The vehicle structural design, detail design, and fabrication were done by the Budd Company. The scale-model crash test program to verify the crashworthiness of the all-plastic structure also was done by the Budd Company at the Fort Washington Technical Center. The advanced lead-acid battery, which was especially designed to match the new vehicle and power system, was developed by Eagle-Picher Industries. Safety and handling analyses of the vehicle design were conducted by Dynamic Sciences, Inc. All American Racers participated in the design analysis and selection of the various components of the suspension, steering, and brake systems.

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Figure 3-2. Relationship of Near-Term Electric Vehicle Program to AiResearch Management

Figure 3-3. AiResearch Near-Term Electric Vehicle Program Organization

4.1 VEHICLE GENERAL CHARACTERISTICS

The ETV-2, summarized in Figure 4.1-1, is a unique passenger vehicle with a power system designed specifically for the variable-speed driving experienced in urban areas. Vehicle kinetic energy is stored in a flywheel through regenerative braking and, upon command, is used for vehicle acceleration.

Although the appearance of the ETV-2 is similar to that of many conventional vehicles, there are subtle features characterizing this electric vehicle that distinguish it from any known electric or conventional internal-combustion-engine automobile.

The heart of the vehicle propulsion system is its power system, which includes the rear-mounted power unit (flywheel, two motor/generator units, and a transmission), the battery pack, and various electronic components that provide system control based on operator commands and power system status.

A unibody concept is used; all body panels and structure trim are fabricated from fiber-reinforced plastic. The complete suspension, steering, and hydraulic brakes are from the BMW 320i to take advantage of available, proven hardware that is lightweight and efficient.

POWER SYSTEM CHARACTERISTICS	
Number of batteries	18
Battery weight	484 kg (1066 lb)
Battery capacity (3-hr rate)	18.72 kW·h
Battery voltage	108 Vdc
Battery recharge time	8.3 h
Flywheel capacity	1.0 kW·h
Flywheel max. speed	24 000 rpm
Motor/generator rating	21 kW peak 11 kW continuous 0 to 11 650 rpm

VEHICLE CHARACTERISTICS	
Overall Length	4.19 m (13.75 ft)
Width	1.68 m (5.5 ft)
Height	1.38 m (4.54 ft)
Wheelbase	2.41 m (7.92 ft)
Curb weight	1507 kg (3323 lb)
Gross weight with four passengers	1779 kg (3923 lb)
Vehicle effective drag area, $C_d A$	0.74 m ² (8.0 ft ²)
Rolling resistance coefficient	0.0075
Body/chassis type	Unibody, fiber-reinforced plastic

PERFORMANCE	
SPEED	
Cruise speed	89 km/h (55 mph)
Max. speed	100 km/h (62 mph)
Speed up 5-percent grade	80 km/h (50 mph)
ACCELERATION TIME	
0 to 48 km/h (30 mph)	8 s
0 to 80 km/h (50 mph)	16 s
40 to 89 km/h (25 to 55 mph)	10 s
RANGE (4 PASSENGERS)	
Federal Urban Driving Cycle	106 km (66 mi)
J227a(D) Driving Cycle	109 km (68 mi)

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Figure 4.1-1. ETV-2 SUMMARY

4.2 POWER SYSTEM DESIGN CONCEPT

The flywheel will improve the performance of the battery-powered vehicle when used in stop-and-go city driving by supplying the high power requirements during acceleration and leaving the battery to supply only the average or sustaining power. There are two primary advantages of this concept: (1) the vehicle range is improved because the battery does not have to operate at high power conditions where its capacity is low, and (2) better acceleration is available because the flywheel can provide more power than the maximum available from the battery.

The reason for this is illustrated in Figure 4.2-1, which shows the power and capacity of a 500-kg (1100-lb) battery pack designed for a typical electric vehicle. The slope of the curve can be varied somewhat from the typical one shown, but the inverse relation between battery power and capacity (vehicle range) is an inherent characteristic of the lead-acid battery. The flywheel provides higher power than the battery can supply and its use allows the battery to provide only the low average power, which maximizes the range of the vehicle.

An analysis was made to determine the acceleration power required for a typical 1500-kg (3300-lb) four-passenger car for various driving cycles. The SAE J227a(D) driving cycle is the standard used for electric cars and is shown in Figure 4.2-2. The Federal Urban Driving Schedule (FUDS) is the current standard for all internal-combustion-engine cars sold in the United States and is more typical of normal city traffic (Figure 4.2-3).

The battery power that is required to meet the acceleration requirements of these driving cycles is shown in Figure 4.2-4. The FUDS requires a peak power of a little over 50 kW (67 hp) at speeds between 30 and 40 mph, and this

Figure 4.2-1. Capacity as Function of Power for a Typical 500-kg (1100-lb) Battery Pack

Figure 4.2-2. SAE J227a(D) Driving Cycle

Figure 4.2-3. Federal Urban Driving Cycle

Figure 4.2-4. Acceleration Power Required for Various Driving Cycles

is the practical minimum for most cars today. The two lower curves in Figure 4.2-4 are for two interpretations of the J227a(D) cycle. If the acceleration from zero to 45 mph is constant, the peak power required is 44 kW (59 hp). The other interpretation of the J227a(D) cycle, which has been made by JPL, is to minimize the peak power by using reduced acceleration at speeds above 30 mph. This results in the third curve in Figure 4.2-4, which peaks at about 28 kW (37 hp). The design conditions for the flywheel system can be derived from the data shown in Figure 4.2-4. It is possible that the range goal of 75 mi might be met using the battery alone if the power and acceleration are limited to the modified J227a(D) curve. Although the flywheel might be helpful in minimizing the battery peak loads, it clearly is not warranted for the modified J227a(D) cycle. However, if the more realistic acceleration rates of the FUDS are to be used, then a reduced range below 75 mi will result. In this case, the flywheel can be used advantageously to supply the higher power for better acceleration and at the same time extend the range beyond that which the battery alone is capable of providing.

During the Phase I preliminary design study, it was determined that the flywheel system had the potential of achieving the range goals concurrently with the power and acceleration requirements of FUDS. This would result in approximately twice the acceleration capability of the modified J227a(D) cycle at the same range, or approximately twice the range at the same (or nearly the same) acceleration. The FUDS cycle, therefore, was adopted as the design goal for the flywheel system.

4. VEHICLE DESIGN AND DESCRIPTION

This section presents a detailed description of the ETV-2, beginning with an analysis (in Section 4.2) of several driving cycles and the application of a flywheel to these cycles. It is shown that the flywheel is particularly desirable for the stop-and-go driving of urban traffic.

Section 4.3 is an overall description of the ETV-2 power system, which includes the power unit, battery pack, and electronic controls. Each of these three major categories is discussed in detail in subsequent sections. The fiber-reinforced plastic body and other subsystems such as brakes, steering, suspension, heating/cooling, and driver controls are discussed in Section 4.7. A detailed breakdown of the vehicle's 1507-kg (3323-lb) curb weight is given in Section 4.8.

4.3 POWER SYSTEM DESCRIPTION

The power system consists of a battery pack, a power unit, electronic controls, and a battery charger. The battery pack consists of eighteen 6-V batteries developed specifically for the ETV-2. The power unit, an assembly of a flywheel, two identical motor/generator units, and a planetary transmission, supplies power to the axle through a silent chain drive. A controller and a power conversion unit (PCU) perform the logic and control functions, and the onboard battery charger allows the batteries to be recharged by plugging into a 115-V, 60-Hz, ac electrical outlet. Figure 4.3-1 shows the interrelation among power system components.

Figure 4.3-1. Power System Components

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The shafts of the flywheel, generator, and motor are mechanically linked through the planetary gearset and function together as an infinitely variable ratio, electromechanical transmission. The flywheel shaft has an operational speed range from 12 000 to 24 000 rpm and is connected to the sun gear of the planetary gearset. The generator, rotating in the same direction as the flywheel, is geared to the ring gear of the planetary gearset. The motor and the carrier shaft are both geared to the final drive.

The two motor/generator units are identical; each can function either as a motor or as a generator, depending on the vehicle mode of operation. For identification in the following discussions, the unit geared to the ring gear is called the generator and the unit geared to the final drive jackshaft is called the motor.

4.3.1 Power Flow and Control

Successful implementation of the regenerative braking, battery load leveling, and high acceleration capability features of the system hinges on the proper transfer of power among the battery pack, flywheel, generator, motor, and axle. The desired power flow is achieved by a controller that contains a microprocessor programmed to compare driver commands with vehicle and component status and to issue appropriate commands to the system.

During normal driving, the demand from the power system varies continuously. To maintain the battery current input to the power system at an essentially constant level, the controller was designed to command the flywheel to supply or to absorb the balance of the power demanded or available. The flow of energy into and out of the flywheel is controlled by the generator. The position of the throttle or brake specifies the torque required at the axle, which is a combined torque of the motor and the reaction torque of the generator through the planetary transmission to the carrier shaft. The controller monitors the armature chopper and the fields of both the motor and the generator and controls the armature current level and direction of the motor and generator to satisfy the torque required and maintain the proper battery current level.

4.3.1.1 Acceleration

During acceleration and hill climbing, the power required from the power system exceeds the power supplied by the battery as specified by the operational battery schedule, which essentially is set at a nearly constant level. As shown in Figure 4.3-2a, under command of the controller, the generator slows down the flywheel, converting flywheel kinetic energy into electric power. This electric power, combined with that supplied by the battery, is transmitted to the motor. The motor converts the combined electric power to mechanical torque and transmits it to the axles through the jackshaft gear. At the same time, the reaction torque of the generator is transmitted mechanically through the planetary gearbox, from the sun gear through the carrier shaft and jackshaft gear, to the axle. The effective efficiency of power transmission to the final drive exceeds the product of the motor and generator efficiencies because a substantial fraction of the total power is transmitted to the final drive through the more efficient mechanical path.

Under normal acceleration, the controller modulates the motor-generator circuit to maintain an essentially constant differential between the motor and generator armature currents, resulting in a nearly uniform battery current discharge rate. The controller also controls the armature current level of the generator, which, in turn, governs the amount of flywheel energy transmitted to the vehicle drive train to satisfy the driver's desired speed control.

Under full-throttle command, the controller demands maximum armature currents through both the motor and generator to obtain the maximum power from the system. Under this condition no electric power is taken from the battery. All energy comes from the flywheel. This unique feature provides a maximum power that exceeds the maximum provided by the battery in conventional electric vehicles.

POWER UNIT COMPONENTS

a. POWER FLOW FOR ACCELERATION

b. POWER FLOW FOR CRUISE

c. POWER FLOW FOR REGENERATIVE BRAKING

d. POWER FLOW FOR FLYWHEEL-LOW AND REVERSE

e. POWER FLOW FOR PRESTART

* NO BATTERY POWER FOR FULL-THROTTLE ACCELERATION

Figure 4.3-2. Power Flow Through the Power Unit

4.3.1.2 Cruise

In an extended cruise mode, the controller will schedule the battery current to the motor armature circuit so that the required road load power is provided by the battery. In this mode, illustrated in Figure 4.3-2b, the controller also switches the generator to operate as a motor; current from the battery is provided to overcome the gear and bearing losses and maintain flywheel speed.

4.3.1.3 Braking

As the controller detects the presence of a braking signal from the driver, the controller initiates the braking mode by switching the motor to operate as a generator and by directing the generator to operate as a motor. The reversal slows the vehicle. During braking, the vehicle kinetic energy in excess of that necessary to balance the road load is returned to the flywheel, as depicted in Figure 4.3-2c. Regenerative efficiency is high because a substantial fraction of the power is recovered through the direct mechanical path. The amount of the vehicle energy transmitted determines the braking rate, which is governed by the armature current level of the motor (operated as generator) as controlled in response to the braking signal from the driver. The controller continuously monitors the motor-generator circuit to maintain a constant armature current differential and an essentially constant battery discharge rate, even during the braking period. The additional electrical energy is directed through the generator (operated as motor) to the flywheel to make up for the system losses during other operating modes.

4.3.1.4 Flywheel-Low and Reverse Modes

Flywheel energy augments battery power during acceleration, and vehicle kinetic energy is converted to flywheel energy during deceleration as described in paras. 4.3.1.1 and 4.3.1.3. The flywheel must contain a minimum available energy for the augmentation process to be of any significance. Hence, the flywheel speed must be maintained above a certain minimum as a prerequisite for use during the acceleration mode. In addition, generator speed also must be maintained above a certain minimum because the line voltage of the motor is limited during acceleration by that of the generator. As the generator speed drops below its base value, the line voltage between the generator and the motor will be reduced accordingly, and the power or torque contributed by the motor also will be reduced. As the generator speed drops below one half of its base value, the combined power output of the augmentation process becomes less than the maximum power the motor could provide alone without the restriction on line voltage by the generator.

To enable the power system to deliver its maximum available power at all times, a flywheel-low mode was created. In this mode, the generator is electrically disconnected from the power system, leaving the motor to operate only from the batteries, as shown in Figure 4.3-2d. The vehicle operates like a conventional electric vehicle without a flywheel (since the flywheel is free-wheeling), with maximum power output limited by the motor or the batteries. The criterion for entering the flywheel-low mode is a generator speed below 1500 rpm during normal or full-throttle acceleration modes. At the low component speeds, the power system will yield more power under the flywheel-low

mode than it would yield under either normal acceleration or in the full-throttle acceleration mode with the generator electrically connected. As soon as the operator starts braking, the power system automatically switches from the flywheel-low mode to the deceleration or idle mode, which tends to increase the generator speed and the flywheel speed.

The vehicle also operates exclusively on battery power in the reverse mode, using only the motor to produce the motive torque. In this mode, the generator is electrically disconnected from the power system, and the motor operates only from the battery. The difference between the reverse mode and the flywheel-low mode is that the reverser in the motor field is activated so that the torque obtained is in the reverse direction.

4.3.1.5 Prestart

To fully utilize the flywheel capability, the flywheel should be charged properly before operating the vehicle. The prestart mode was designed specifically for that purpose. During this mode, the motor is electrically disconnected from the power system, while the generator functions as a motor, charging the flywheel in accordance with a preset schedule as shown in Figure 4.3-2e. Ordinarily, it is suggested that the flywheel be charged to at least 20 000 rpm before operation, which takes about 4 to 5 min. This will prevent the power system from entering the flywheel mode unnecessarily. It takes about 5.5 min for the generator to charge the flywheel to its maximum speed of 24 000 rpm.

4.3.2 Power System Cooling

Heat generated by the bearings and gears is dissipated from the transmission housing through free convection. The power conversion unit and the motor/generator are forced-air-cooled under all operating conditions to provide maximum efficiency. As shown in Figure 4.3-3, an axial dc-motor-driven fan cools the power unit by drawing air from the rear of the vehicle near the tail lights. The PCU consists of two assemblies located on each side of the contactor compartment. The roadside PCU cooling air is ducted to the contactor plenum and the roadside PCU compartment. The curbside PCU cooling air is ducted directly into the curbside PCU compartment. The two streams of cooling air are mixed before entering the fan and the motor/generator set. The mixed air cools the commutator sections of the two motor/generators before it flows through their armature airgaps. It exits overboard from the motor/generator end bell sections.

A fan cools the battery charger electronics and simultaneously ventilates the battery tunnel during the battery-charging mode. The controller is cooled by free convection through the built-in controller housing heat sink.

4.3.3 Battery Pack

Primary power is supplied by eighteen 6-V batteries with an energy density of 39.8 W·h/kg (18.1 W·h/lb). These golf-cart-sized batteries were developed by Eagle-Picher Industries specifically for the ETV-2. The batteries are arranged in two rows in a removable tray that is housed in a tunnel extending from the front of the vehicle to a point between the rear seats (Figure 4.3-4).

4.3.4 Power Unit

The vehicle is propelled by the power unit (Figure 4.3-5), which is connected to the rear axle through a silent chain drive. The flywheel and two motor/ generator units in the power unit are mechanically linked through a planetary gearset. The motor/generator units are identical; each may operate either as a motor or a generator depending on the drive mode.

4.3.5 Power Conversion Unit

The power conversion unit (PCU) conditions battery power and applies it selectively to the power unit motors as commanded by the controller. The accessory power supply and lubrication and vacuum pump controls also are part of the PCU. The PCU monitors and protects the power unit, and feeds status information back to the controller. The PCU electronics are installed behind the rear seat of the vehicle as shown in Figure 4.3-6.

4.3.6 Controller

A solid-state electronic controller provides all power system control. It senses the operating state of the vehicle and power unit and controls the flow of power among the flywheel, generator, motor, and battery in response to driver commands. It provides a smooth transition between operating modes and eliminates acceleration jerks and large current transients to enhance driving comfort and increase the power system switch gear service life. In addition, the controller automatically limits power unit speeds and temperatures, drives the dashboard display, and delivers selected data parameters to the data recorder. The controller, the charger, and the data recorder are mounted in the vehicle front compartment (Figure 4.3-7).

4.3.7 Charger

The onboard battery charger will recharge the battery pack from either 15- or 30-A, 115-Vac household service, which is switch selectable at the charger. With the 30-A current, the battery pack may be recharged overnight.

4.3.8 Data Recorder

The data recorder provides the means for recording selected vehicle parameters such as speed, battery current and voltage, and several temperatures on cassette tape during vehicle operation. The recorded data are processed in a separate computer facility. The recorder is intended for vehicle demonstration and evaluation and would not be included as a part of production vehicles.

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Figure 4.3-5. Power Unit Installation

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Figure 4.3-6. Power Conversion Unit Installation

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Figure 4.3-7. Front Compartment Electronics

4.4.1 Motor/Generator

4.4.1.1 Basic Design and Rating

The two motor/generator units have essentially the same power and thermal requirements; therefore, one design is used for both applications. This design (Figure 4.4.1-1) is a 4-pole, dc, shunt motor, with interpole and pole-face windings, and is similar to a conventional traction-motor design. The rated motor output is 21 kW over a speed range of 3 000 to 10 000 rpm with an armature input of 108 V, 228 A. The control field maximum input is 80 V, 14.0 A. The operating speed range is from zero to 11 650 rpm.

The motor design meets the operating requirements for a typical traction motor application; however, the design deviates from conventional motors of this rating to minimize weight without lowering motor efficiency. The main differences in approach are as follows:

- (a) Aluminum end bell castings are used instead of steel to save weight.
- (b) Pole-face windings are used to improve control of armature reaction magnetomotive force (mmf), thus permitting higher armature current for a given motor size. This results in a net improvement in efficiency by reducing the main field losses.
- (c) The field frame is made of low-carbon steel tubing and serves a dual purpose: a structural member and a low-reluctance flux path between the poles.
- (d) Commutator design parameters such as tip velocity, brush current density, and interior stresses are the maximum allowable values in order to save weight.

A motor/generator cross section is shown in Figure 4.4.1-2. Design characteristics are presented in Table 4.4.1-1. The mounting of the units is shown in Figure 4.4.1-3.

4.4.1.2 Performance

Operation within the motor/generator torque-speed envelope (Figure 4.4.1-4) occurs at the point where the torque-speed line of the load intersects the motor/generator curve. Approximate operating points for the two units are shown in the figure. Cruise at 89 km/h (55 mph) is a steady-state condition, whereas the J277a(D) driving cycle points represent a weighted average of the various cycle conditions from the thermal viewpoint. A summary of performance characteristics and losses is presented in Figure 4.4.1-5.

4.4.1.3 Physical Characteristics

Four openings in the rear end bell, one over each brush, permit easy inspection and servicing of the brushes. A protective shroud covers the openings.

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Figure 4.4.1-1. Motor/Generator Components

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Figure 4.4.1-2. Motor/Generator Assembly

TABLE 4.4.1-1

MOTOR/GENERATOR DESIGN PARAMETERS

<u>Ratings</u>	
Power, kW	24 max. at 3000 to 10 000 rpm
Speed range, rpm	0 to 11,650
Power source voltage, V	108
Power source current, A	250 max. up to 10 000 rpm
<u>Dimensions,</u>	
Diameter	231 mm (9.1 in.)
Length	369 mm (14.5 in.)
Radial air gap (armature to field)	0.84 mm (0.33 in.)
<u>Circuit Resistance at 20°C, ohms</u>	
Armature winding	0.0215
Shunt (control) winding	3.488
Interpole winding	0.0060
Pole face winding	0.0095
<u>Circuit Inductances, mH</u>	
Armature winding	0.097
Shunt (control) winding	81.9 to 409.6
Interpole winding	0.125
Pole face winding	0.0469
<u>Electrical Loading, A</u>	
Maximum armature current	250.0
Maximum shunt field current	14.0
<u>Winding Descriptions</u>	
Armature:	75-bar commutator, 25 slots, 6 conductors/slot, wave winding, rectangular wire of 8.9 mm ² (0.0138 sq in.)
Shunt field:	144 turns No. 16 AWG wire
Interpole:	5 turns rectangular wire, 33.5 mm ² (0.0052 sq in.)
Pole face:	5 turns rectangular wire, 19.6 mm ² (0.0304 sq in.)
<u>Materials</u>	
Housing	AISI 1010 steel
Wire	Annealed copper, Kapton wrapped
Shunt poles	Silicon steel
Armature laminations	2V Permendur
Interpoles	Silicon steel
Commutator	Silver bearing copper
Brushes	Electrographitic*
End bells	Aluminum

*Union Carbide N-4 for the motor and Morganite H-100 for the generator.

Figure 4.4.1-3. Mounting Arrangement of Motor/Generator Units

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Figure 4.4.1-4. Motor/Generator Torque-Speed Performance Envelope

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Cooling air flows through a duct into the motor at the brush end at a volumetric rate of approximately $1.4 \text{ m}^3/\text{minute}$; then it flows over the commutator, along the armature and field structure, and finally exhausts through the holes in the drive end bell. Armature leads enter through a strain relief near one of the brushes, and terminate inside the motor, one on a brush box and the other directly on the interpole windings. This minimizes the number of junctions required and improves reliability. Similarly, control field leads enter through a strain relief and terminate on the shunt field coil winding. Two thermistors are located on the pole-face winding across from the armature hot-spot. The pole-face winding temperature, which tracks armature winding temperature, is monitored by the controller. If the temperature exceeds 204°C , the armature current is automatically and gradually reduced to 120 A. If the temperature exceeds 164°C as measured by the second thermistor, a dashboard warning light is illuminated.

At the drive end of the motor, a positive contact shaft seal (lip-type wiping seal) prevents oil and contaminants from entering the motor.

4.4.1.3.1 Armature

The armature winding is a one-turn-per-coil, three-coils-per-slot, retrogressive-wave winding, with four poles and four brushes. The lamination has 25 slots, and the commutator has 75 bars. The three coils are formed and taped as a group. To reduce stress at the hairpin turn, each turn consists of two rectangular wires. Each wire is insulated with 3-mil Kapton tape, plus one layer of 4-mil glass tape. Armature coils terminate in slots in the commutator riser, and are welded into place. A large-diameter, hollow shaft is used to provide maximum stiffness with minimum weight.

Two armature lamination materials were used in development test units: M-19 (composite of 3.5 percent silicone and 96.5 percent iron), and 2V Permendur (composite of 49 percent cobalt, 2 percent vanadium, and 49 percent iron). 2V Permendur laminations produce higher magnetic flux density than M-19 for given field ampere turns. 2V Permendur was selected for the ETV-2 motor/generator units because the higher flux density results in higher torque for the same current. 2V Permendur cost, which is higher than M-19 material, fluctuates depending on the availability of cobalt. Weight difference between M-19 and 2V Permendur is negligible.

The diametral interference between the shaft and the laminations maintains load capability under all operating conditions. Angular alignment of the laminations and the commutator is achieved by tooling during assembly. The thermal analysis, under a variety of load conditions, showed that the armature hot-spot is on the conductor at the hairpin curve where the cooling air is the hottest and there is no surrounding lamination material to act as a heat sink. To sense this temperature, without placing sensors on the rotating armature, a temperature sensor is located on the pole-face winding directly opposite the armature hot-spot.

Figure 4.4.1-5. Final Motor/Generator Performance Characteristics

4.4.1.3.2 Commutator and Brushes

The commutator incorporates silver-bearing copper segments and mica insulation. Because of the axial width of the brushes, each holder uses two side-by-side brushes. The shunts are joined in a common lug to facilitate insertion and removal as a pair. One brush is wider than the other; when inserted into the brush holder, they are staggered, so that the individual brush-wear paths on the commutator overlap and produce a more uniform commutator wear.

Thermal analyses under worst-case conditions of ambient temperature, cooling air temperature, and load cycle showed that commutator temperatures are within the design maximum of 177°C.

Brush material in both machines is an electrographic type material. During the development program, materials of two different grades were tested with differences in friction loss and commutation characteristics. For the final configuration, Union Carbide N-4 was selected for the carrier motor because of its better commutation characteristics at its high-speed condition. Morganite H-100, with a lower friction loss, is used in the ring generator, which has a lower maximum speed requirement. The only difference between the two machines is the brush material and the brush spring load (4 psi for the motor and 3 psi for the generator).

4.4.1.3.3 Interpole and Pole-Face Winding

Because the motor operates over a wide range of speed and armature current, pole-face windings and interpoles are used to facilitate commutation. Armature reaction flux, produced by the current-carrying armature conductors, produces crossmagnetizing and demagnetizing fields. Since the armature reaction is a function of armature current, the pole-face windings and the interpole windings are connected in series with the armature, and carry armature current. The pole-face windings produce a flux that counteracts the armature reaction flux produced by the current in the armature windings. The interpoles produce a flux that opposes the cross-magnetizing flux and also induce a voltage to counteract the reactance voltage.

A final adjustment of interpole strength, made during testing, consists of varying the reluctance of the interpole circuit by using a combination of magnetic and nonmagnetic shims between the pole and the field frame.

4.4.1.3.4 Control Field

The main field of the motor is a separately excited shunt field; it controls the motor in conjunction with armature voltage. Four field coils, connected in series, produce full-field capability with 80 V maximum. The coils are layer-wound with 16-gage, ML-insulated wire. Interlayer insulation is 0.08-mm (0.003-in.) Nomex. The entire coil is then wrapped with glass tape and Kapton tape.

4.4.1.4 Thermal Characteristics

The motor/generator insulation is rated at 220°C. Under all steady-state conditions, the armature, pole face, and interpole winding temperatures remain below this value. The typical transient thermal condition experienced during a hill climb is illustrated in Figure 4.4.1-6.

Results of an analysis of steady-state operation are presented in Table 4.4.1-2. Under transient conditions, the normal duration of the transient response is such that rated insulation temperature is not exceeded; however, short-time exposure to temperatures above 220°C will not damage the insulation.

Temperature sensors on the pole winding are used by the control system to track the hot spot temperature of the armature and protect it against overheating during prolonged transients (such as hill climb). Thermal analysis has shown that for an armature temperature of 220°C, the sensor temperature is 177°C.

Figure 4.4.1-6. Motor Transient Temperatures in Hill-Climb Mode

TABLE 4.4.1-2

THERMAL ANALYSIS OF
STEADY-STATE MOTOR/GENERATOR OPERATION

Motor/ Generator	Motor		Generator		Motor at 110 A		Motor at 120 A		Generator	
Operating condition	J227a(D) Cycle		J227a(D) Cycle		88.5 km/h (55 mph)		88.5 km/h (55 mph)		88.5 km/h (55 mph)	
Temperature	°C	°F	°C	°F	°C	°F	°C	°F	°C	°F
Armature	143	290	153	308	169	337	184	364	110	230
Interpole	107	225	114	237	126	259	131	268	96	205
Pole face	116	240	119	246	137	278	143	290	103	217
Main pole	124	256	119	247	115	239	116	241	215	419
Commutator	131	267	168	334	153	307	161	322	99	210
Ball bearing	126	259	141	286	162	324	164	328	117	243
Total heat generated, W	1087		1326		1508		1674		1157	

4.4 POWER UNIT

The power unit is a complete assembly installed in the rear of the vehicle. It consists of two motor/generator units, a planetary transmission and the flywheel. Although the vacuum forepump is mounted on the vehicle frame, it is considered, for description purposes, as part of the flywheel assembly.

The two motor/generator units are nearly identical units, each of which operates either as a motor or as a generator, depending on vehicle status and driving command. Throughout the report, the terms "motor" and "generator" are used to designate the unit, not to designate a function. Accordingly, the unit geared directly to the planetary carrier shaft and the final drive is referred to as the "motor" although the motor may be generating electric power as it does during regenerative braking. The other unit, geared to the planetary ring gear, is referred to as the generator.

4.4.2 Transmission Assembly

The transmission, shown in Figures 4.4.2-1 and 4.4.2-2, mechanically connects the axle, motor, generator, and flywheel and provides the speed ratios required of the drive. The key to obtaining maximum power transfer between the flywheel and the axle is the split powertrain differential planetary gearset that provides an infinitely variable gear ratio from standstill to maximum speed.

Figure 4.4.2-3 illustrates the transmission drive configuration, the main gear ratios, and the kinematic relationship of the generator, flywheel, and motor speeds.

Power is extracted from the flywheel and transmitted through a quill shaft to the planetary gearset, which functions as a differential power-splitting device. The sun gear, integral with the quill shaft, drives the planet gears, which are connected through the planet carrier and related gears to the final drive. The planet gears, in turn, drive the ring gear, which engages the generator drive gear through a fixed-ratio drive. The variation of the generator speed provides for an infinitely variable output speed to the final drive.

4.4.2.1 Construction

The entire transmission assembly is enclosed by an aluminum housing that provides the interconnecting and mounting structure for the various power train elements. The motor, generator, and flywheel are attached to the transmission case by bolted flanges. The lube pump and oil filter also are mounted on the transmission housing. The housing is a structural shell and a bearing bulkhead, which also forms the dividing wall between the transmission and the flywheel cavity.

The transmission housing structure dissipates all internal and external loads between the attached components (such as the motor/generators and the flywheel assembly) to the power unit mounts. The housing gear supports hold relative gear misalignment due to load deflection to a tolerable minimum; the loads are carried through basically cone-shaped elements to the dissipation points. Aluminum alloy A356-T6, used as a casting material, is well known for its high purity, strength, ductility, and vibration fatigue resistance. The gear bearings are supported by pressed-in, pin-retained steel sleeves for long life.

To limit friction, a major contributor to losses, the pitch-line velocity was established in the range of 10 to 30 m/s (2000 to 6000 ft/min), while the load factors (k) are 200 to 400. The resulting coefficient of friction is 0.03 to 0.04.

Each bearing was selected for the minimum mean diameter to carry the load and to provide 4000-hr B10 life.

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Figure 4.4.2-1. Transmission Assembly

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80201-1 F-32754

Figure 4.4.2-2. Transmission Components

$$N_G = 0.4 N_{FW} - 0.49 N_M$$

$$N_G = 0.4 N_{FW} - 57.53 V \text{ (IN km/h)}$$

$$N_G = 0.4 N_{FW} - 92.61 V \text{ (IN mph)}$$

$$N_M = 12.63 N_{AX}$$

$$N_M = 117.4 V \text{ (IN km/h)}$$

$$N_M = 189.0 V \text{ (IN mph)}$$

N_G = GENERATOR SPEED, rpm
 N_{FW} = FLYWHEEL SPEED, rpm
 N_M = MOTOR SPEED, rpm
 N_{AX} = AXLE SPEED, rpm
 V = VEHICLE SPEED, mph

BASED ON ROLLING RADIUS = 0.285 m (0.936 ft)

GEAR NAME	GEAR TYPE	NUMBER OF TEETH	PITCH DIAMETER	
			mm	in.
GENERATOR	HELIX	49	64.42	2.536
RING OUTSIDE GEAR	HELIX	82	107.81	4.245
RING INSIDE GEAR	SPUR	102	107.95	4.250
PLANET	SPUR	39	41.28	1.625
SUN	SPUR	24	25.40	1.000
CARRIER	HELIX	85	111.76	4.400
JACKSHAFT	HELIX	96	126.22	4.969
MOTOR	HELIX	20	26.30	1.035
CHAIN SPROCKET	-	19	-	-
CHAIN WHEEL	-	50	-	-

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Figure 4.4.2-3. Transmission Arrangement and Characteristics

4.4.2.2 Noise Considerations

Transmission noise has been minimized by three design steps. First, where possible, the gears are helical so the tooth-to-tooth contact ratio is greater than one, thereby producing quiet gear-tooth engagement. Second, the tips of the gears have been relieved to minimize noise on initial contact. Third, the critical tolerances for the gears have been selected as the best obtainable for an automotive application. The noise is controlled through accurate profile, tooth-to-tooth spacing, pitch-line runout, and lateral runout.

General transmission design follows automotive practices; case-hardened (carburized) material provides a controlled R_C -58 minimum surface hardness and a 35 to 38 core hardness to maximize fatigue strength.

4.4.2.3 Lubrication

The transmission lubrication system keeps churning losses and gear oil squeeze losses to a minimum. Jet lubrication is restricted to the planetary gear and the flywheel bearings; all other areas are splash lubricated.

The lubrication pump is a Gerotor pump delivering Mobil Jet II (MIL-L-23699) oil at 3.5 l/min and 1.51 atm with a power requirement of less than 25 W. The pump operates when the power system is energized through the key switch and continues to operate as long as the flywheel is rotating above 5000 rpm even after the key has been switched off. Predicted maximum oil temperature is 85°C.

The lubrication pump operation was extensively tested in the vehicle to validate its design characteristics, especially during the 3-hr maximum driving condition. Its performance is affected by battery voltage and oil temperature variations. Mobil Jet II oil viscosity changes from 28 to 11 centistokes as the oil temperature rises from 38° to 66°C. This change affects the pump characteristics, and also reduces significantly the transmission oil pressure drop. These characteristics are shown in Figure 4.4.2-4. If the pumping pressure drops below 20.7 kPa (3.0 psig), a pressure switch will cause the dashboard overheat light to illuminate.

The power unit must be ventilated during dynamometer testing to simulate the actual driving environment and prevent oil overheating. With proper filter and oil replacement at regular maintenance intervals, the vehicle will be free of any nuisance overheat indication caused by the lubrication system.

4.4.2.4 Transmission Losses

During development testing, transmission losses were categorized as: (1) planetary losses, which include sun, planet, ring, and generator gears and carrier shaft losses, and (2) final drive losses, including jackshaft, motor gear, drive chain sprocket, and chain wheel losses. Losses in these components depend on gear speeds and transmitted torque, which, in turn, are determined by the operating condition, i.e., low-speed cruise, high-speed cruise, acceleration or deceleration, and flywheel speed. If desired, the losses for any particular

Figure 4.4.2-4. Lubrication Pump Pressure vs Flow Characteristics

condition could be calculated from the computer model discussed in Section 7. As an example, the losses for a level road cruise at 56 km/h (35 mph) with fly-wheel speed at 20,000 rpm are 0.5 kW for the planetary geartrain and 0.2 kW for the final drive.

4.4.3 Flywheel Assembly

The flywheel assembly is a particularly unique and interesting part of the ETV-2 power system because of its central role in providing better vehicle performance than is obtainable from conventional lead-acid battery systems and also because it demonstrates a new technology that is expected to be useful in many transportation and energy-related applications. Previous attempts to incorporate an energy-storage flywheel in an energy-saving commuter car have failed, primarily because of excessive weight and excessive losses. In this program, an advanced-design, lightweight flywheel rotor made from composite material with a high strength-to-weight ratio was successfully developed and demonstrated. This new design also incorporates a high-vacuum environment for the flywheel, which has demonstrated acceptably low losses.

The flywheel and housing assembly consists of the lightweight composite-material flywheel, the housing, the vacuum pump, the molecular pump, the seal, the bearings, and the lubrication system. The two-stage vacuum pump system consists of a motor-driven vacuum pump and a molecular flow vacuum pump attached to and driven by the flywheel rotor. The relation of these parts can be seen on the cross-sectional drawing in Figure 4.4.3-1. Because the rotor must operate at high speeds, a very low pressure must be maintained around it to minimize frictional losses and to avoid overheating the composite material of the flywheel rim. Figure 4.4.3-1 identifies the two different vacuum pressure levels created within the housing. The small motor-driven vacuum pump evacuates the entire housing down to an intermediate pressure level (1 mm HgA), which is maintained by the vacuum and oil seal just outboard of the inboard bearing assembly. The molecular pump creates the high vacuum. The molecular pump rotor is bolted to the flywheel rotor and, in conjunction with the stator, effectively isolates the cavity (area) around the flywheel and expels the air from around the flywheel rotor, thus creating the second, very much lower ($1\text{ }\mu\text{m HgA}$) vacuum pressure level. The advantage of this arrangement is that the carbon-faced oil and vacuum seal has to maintain only the relatively modest intermediate pressure, and the motor-driven vacuum pump can adequately expel the leakage of the seal. The molecular flow vacuum pump can then reduce the pressure in the cavity around the rotor with no seal leakage to make up since the entire outboard portion of the housing has no openings to ambient pressure.

The flywheel is mounted in the rear of the vehicle with its rotational axis parallel to the rear wheel axis, and it rotates in the direction opposite to the vehicle wheels when driving forward. The gyroscopic forces in this arrangement will tend to resist the tendency of the vehicle to lean away from a turn.

4.4.3.1 Housing

The housing contains the flywheel and supports the bearings, seal, molecular pump stator, and outboard lubrication system. It is also a sealed chamber that is evacuated by the forepump and molecular pump so that the flywheel rotates in a near-vacuum. The housing, which provides safe flywheel containment, has 6061 aluminum walls and a 6.3-mm-thick (1/4-in.) steel ring around the outer circumference.

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Figure 4.4.3-1. Flywheel Assembly and Housing (View Looking Forward)

Figure 4.4.3-2. Composite Flywheel

4.4.3.2 Flywheel

The flywheel shown in Figure 4.4.3-2 is made of three components--the hub and two identical rims that mount on the hub. The hub is a single pancake forging of 7075 T3652 aluminum, which is stabilized by liquid nitrogen after it is bandsawed from the pancake and again before final machining. The contour of the hub ensures uniform stress throughout its total volume except for the hub ends of each of the four spokes. In this manner, all of the material is working, and the weight is held to a minimum. The spoke hub ends provide sufficient surface area so that the operational radial stresses on the rim from the spoke are below the allowable stresses for the rim. To further distribute the load on the rim at the spoke, thin pads of Melamine (epoxy glass) are bonded to each spoke tip, and thin pads of polyurethane are bonded to each Melamine pad. The rim is not bonded to the polyurethane.

Two rims are assembled onto each hub. Each rim consists of nine separate, radially stacked rings. Each ring is 53 mm (2.1 in.) wide and either 6.1 mm (0.24 in.) or 4.67 mm (0.184 in.) thick. Each rim consists of rings of this construction. The first ring is an S-2 fiberglass/epoxy composite. This ring is 6.10 mm (0.240 in.) thick. All composites in the rims use the four-part elastomerized epoxy consisting of Dow DER 7818 epoxy resin, DER 7575.02 epoxy resin (the elastomerized component), ERL 4206 diluent, and TONOX 60-40 hardener. Rings 2 through 4 are made of Kevlar 29/epoxy composite. Rings 5 through 9 are made of Kevlar 49/epoxy composite. The first ring is wound on the mandrel and cured. A mold release agent is placed over the first ring so that the second ring will be permanently separated from it. Each ring is subsequently wound upon and permanently separated from the previous ring by the mold release agent.

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This set of nine nested rings has a low radial stress when subjected to the radially graduated centrifugal force field. The low radial stress is necessary to prevent cracking and delamination in the epoxy composite transverse direction. Each ring works independently in the radial direction, and as a unit in hoop tension, taking best advantage of the filament-wound composite materials properties.

The two rims are mounted on the hub, separated from each other by an axial register. The rims are distorted to a subcircular shape by the amount the hub diameter exceeds the inside diameter of the rims. Thus, only the spring force in the rim, due to its attempt to return to its round free state, holds it onto the hub by friction force. No attachment or bonding flywheel devices and their attendant stress risers are employed. Upon spinning the flywheel to speed, the radial forces and resulting assembly friction forces between rims and hub are even greater than at rest; thus the rim is positively held in place.

Specific design characteristics are:

Total energy	1.0 kW·h
Rim OD	584 mm (23.0 in.)
Rim ID	489 mm (19.2 in.)
Design speed, maximum	26 250 rpm
Operating speed, maximum	24 000 rpm
Spoke radial force pressure	24 kN (24 MPa) (5300 lb)(3500 psi)
Power consumption (including bearings and outboard lubrication pump)	425 W at 24 000 rpm

Materials:

Inner ring (1)	6.1-mm (0.24-in.) S-2 fiberglass/epoxy
Mid-rings (3)	6.1-mm (0.24-in.) Kevlar 29
Outer rings (5)	4.67-mm (0.184-in.) Kevlar 49

4.4.3.3 Bearings and Outboard Lubrication

The flywheel rotor is supported by two bearing assemblies, one on either side of the rotor. The inboard bearing is a Conrad (deep-groove) type bearing that provides accurate axial positioning of the flywheel rotor. The bearing axial location must be as close to the molecular pump as possible to minimize any variation in pump face clearance due to rotor thermal expansion. These bearings are jet lubricated by the transmission lubrication pump.

The outboard bearing employs a roller bearing, which permits the rotor to expand axially. Since this bearing is located in the flywheel cavity and is exposed to a 1 μ m Hg absolute pressure, a self-contained lubrication concept was developed. The sump is filled with Fyrquel 150 oil. A liquid pump disc attached to the flywheel shaft transports the oil from the reservoir to a splash shield, providing adequate lubrication and cooling for the bearing. The size of the disc and the quantity of oil used for lubrication was determined by tests to minimize power consumption. The bearing arrangement is shown schematically in Figure 4.4.3-3.

Figure 4.4.3-3. Bearing Arrangement for Flywheel

4.4.3.4 Vacuum System

The vacuum system consists of two major components, a forepump and a molecular pump, which maintain a vacuum in the flywheel cavity so that the windage friction does not create excessive flywheel surface temperatures. (See arrangement in Figure 4.4.3-1).

The forepump is a vane-type pump that reduces flywheel cavity pressure to 1 mm HgA. It is driven by an electric motor, which is energized whenever the flywheel is rotating and the cavity pressure rises above 1 mm HgA. It is mounted as a separate component to the vehicle chassis.

The molecular pump is a mechanical drag type pump consisting of a smooth-surface disc attached to the flywheel rotor and a stator with spiral grooves cut to accommodate the induced flow of air molecules. Face clearance between the stator and rotor is 0.38 to 0.43 mm (0.015 to 0.017 in.). The spiral grooves, shown in Figure 4.4.3-4, curve in the direction of the disc rotation, allowing the disc to propel the molecules in an outward direction.

For evaluation purposes, molecular pump performance tests were conducted by varying the forepump pressure from 0.5 to 2 mm HgA (molecular pump outlet pressure) and the face clearance from 0.30 to 0.51 mm (0.012 to 0.020 in.). The pump inlet pressure (or cavity pressure) was measured as a function of pump rotational speed as shown in Figure 4.4.3-5. Pressure ratios in excess of 20 000 were obtained.

Figure 4.4.3-4. Molecular Pump Stator

Figure 4.4.3-5. Molecular Pump Performance Characteristics

The demonstrated capability to generate quickly and maintain low cavity pressure enhances the overall design goal of low flywheel losses, especially during flywheel startup. Heat during startup causes the flywheel surface temperature to rise until adequate vacuum is generated. Figure 4.4.3-6 combines the startup cavity pressure transient with the flywheel surface temperature transient analysis.

The curves show the forepump pressure response from an initial pressure of 760 mm HgA to the final pressure of 1 mm HgA. This response varies as a function of battery voltage condition. The corresponding flywheel cavity pressure response is also displayed for an initial cavity pressure of 1.0 mm HgA.

The forepump and motor are sized to generate a volumetric pump-down rate of approximately $683 \text{ cm}^3/\text{s}$ (2500 cu in./min), a minimum necessary to prevent excessive flywheel temperature overshoot during the startup.

Design characteristics of the molecular pump are summarized in Table 4.4.3-1.

TABLE 4.4.3-1

MOLECULAR PUMP SPECIFICATIONS

Rotor and stator OD	352 mm (13.88 in.)
Stator ID	118 mm (4.66 in.)
Channel depth	5.1 mm at ID to 1.3 mm at OD (0.200 in. at ID to 0.050 in. at OD)
Channel width	15.2 mm (0.60 in.)
Number of channels	6
Speed (max.)	25 000 rpm
Rotor disc and stator face taper	0.74 deg
Rotor-stator clearance	0.38 to 0.43 mm (0.015 to 0.017 in.)
Stator spiral angle	15 deg

4.4.3.5 Seal

A seal on the flywheel shaft with a pressure differential of nearly 101 kPa (ambient sea level pressure) separates the flywheel cavity from the transmission cavity. It incorporates the smallest practicable diameter to reduce sealing area, surface speed, and power loss. Only the small-diameter torque-carrying shaft protrudes through the seal. It is a carbon-faced seal, and is well lubricated and cooled by oil on both sides. The tested leakage rate is less than $0.47 \text{ cm}^3/\text{s}$ (0.001 cfm).

4.4.3.6 Flywheel Dynamic Characteristics

Vehicle maneuvering and flywheel gyroscopic effects were part of the original vehicle design considerations. The flywheel gyro stabilizing feature and the rotor bearing fatigue life design requirements were a part of these considerations. In addition, the dynamic balance of the rotor assembly was given considerable emphasis. In the course of hardware development, several

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Figure 4.4.3-6. Flywheel/Vacuum System Relationship.

design changes were made to achieve an optimum dynamic support for the high flywheel inertia design. Good suspension elasticity is desired for good flywheel dynamic balance and rotor resonance at low speeds. This elasticity is controlled by the resilient mount spring rate, which was set at 7.0 MN/m (40 000 lb/in.) to keep the rotor natural frequency and resonance below 8000 rpm.

As shown in Figure 4.4.3-7, the resilient mounts are designed with stop pads on both sides to accommodate the normally small vibration requirements with a low spring rate for good dynamic balance, and to limit the large amplitude movements (caused by the gyroscopic action) at 0.18 mm (0.007 in.) and avoid rubbing of rotating parts. The precession angle of the rotor is limited, which precludes any possibility of the molecular pump disc rubbing against the rotor. The characteristics of the mounts are shown in Figure 4.4.3-8.

The vehicle maneuvering diagram in Figure 4.4.3-9 shows that the design point for gyroscopic forces occurs for a vehicle turning at 20.8 km/h (12.95 mph) at a radius of 4.9 m (16 ft). This corresponds to an angular velocity of 1.19 rad/s. When this condition occurs along with a flywheel speed of 2513 rad/s (24 000 rpm), the resulting radial force on the bearing and mount is 11.0 kN (2474 lb). As shown in Figure 4.4.3-8, under this maximum radial load and with a worst-case bearing mount, no disc-to-stator contact will occur. The stress life fatigue characteristics are shown in Figure 4.4.3-10.

4.4.3.7 Flywheel Dynamic Balancing Procedure and Characteristics

The flywheel balancing is accomplished in two steps:

- (a) Preliminary balancing of the flywheel rotor assembly; the rotor is balanced in two planes to within 1.7×10^{-3} N·m (7.0 gm-in.) at approximately 1000 rpm.
- (b) Dynamic balancing of the complete flywheel assembly: the flywheel is balanced in two planes to within 1.2×10^{-3} N·m (5.0 gm-in.) at the resonant (critical) speed, which is between 7500 and 8500 rpm. If necessary the moment balance correction is also made to maintain the vibration level of each plane below 0.076 mm (0.003 in.) in the range from 8000 to 25 000 rpm.

The balance corrections are made by applying lead tape on the inner surface of the rim.

Figure 4.4.3-11 shows the peak-to-peak displacement measured on the two ends of the rotating assembly (inboard and outboard) from zero to 25 000 rpm.

Figure 4.4.3-7. Resilient Mount Used for Both Roller and Ball Bearings in the Flywheel Assembly

$$\text{RADIAL FORCE, } F = \frac{I_p \omega \Omega}{\delta} = 11 \text{ kN (2474 lb)}$$

FOR

WHERE

I_p , POLAR MOMENT OF INERTIA = 9.1 IN.-LB-S²

ω , FLYWHEEL ANGULAR VELOCITY = 2513 rad/s (24 000 rpm)

Ω , VEHICLE ANGULAR (TURNING) VELOCITY

δ , SEPARATION BETWEEN FLYWHEEL BEARINGS = 11.0 IN.

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Figure 4.4.3-8. Resilient Mount Movement Characteristics

Figure 4.4.3-9. Vehicle Maneuvering Envelope

Figure 4.4.3-10. Resilient Mount Stress Life Fatigue Characteristics

Figure 4.4.3-11. Typical Flywheel Shaft Displacement

4.4.3.8 Highlights of Development Program for Flywheel Assembly

The development of the flywheel and the various parts of the assembly progressed generally according to the proposed and planned program. The program was very successful and resulted in a design that achieved all structural and operational requirements. The final design was essentially the same as the design that was originally proposed; no significant deviations from or alterations to the original design were required. One of the reasons for this was the conservative approach used throughout the program. For this first demonstration of the flywheel concept, it was considered more important to show durability rather than minimum size and weight.

The development of the flywheel rotor represented a significant part of the total ETV-2 development primarily because of the long fabrication time required and the necessity to build one unit at a time and to acquire test confirmations before proceeding to fabrication of subsequent assemblies. Following the development work, the final design configuration was supported by additional-bearing accelerated endurance testing and by maneuvering tests on the entire flywheel assembly. The endurance and cycling tests were followed by teardown and inspection. Although the composite rotor design concept had been demonstrated in the laboratory, data on the stress characteristics of the new Kevlar materials were limited, particularly for creep and long-term fatigue.

4.4.3.8.1 Endurance Test

After the 1000-cycle endurance test, a permanent gap was discovered between the fiberglass ring and the first Kevlar ring. This was attributed to an initial strain and permanent set of the Kevlar. This effect was researched by a separate test program during which cyclic and creep tests were conducted. The tests confirmed that the growth stopped after the first few cycles and did not progress beyond an acceptable value. The material creep tests also confirmed values within acceptable limits for the life of the rotor.

4.4.3.8.2 Burst Test

Unlike failure of a conventional metal-rim flywheel, which may result in sharp fragments with significant amounts of kinetic energy, failure of the composite-material flywheel typically is much less severe because the material deteriorates into a soft cotton-like mass. This was demonstrated by test during the flywheel development.

A burst test on the flywheel rotor was conducted to demonstrate the containment of a rotor failure and also to measure the momentum transfer to the supporting structure during the rapid deceleration of the rotor parts. The rotor failure was induced by admitting air into the evacuated housing while the flywheel rotor was operating at maximum rotational speed. The failure occurred as a tensile stress failure in the outer two Kevlar rings. The resultant debris was thrown outward where it formed a hard-packed wedge between the rotor and the housing and stopped the rotor. The deceleration time was about 2 sec. The measured torque reaction on the structure peaked at 1491 N·m (1100 lb-ft). The test was considered very successful and demonstrated that future designs could utilize a much lighter weight housing assembly. A photograph of the burst test rotor is shown in Figure 4.4.3-12.

During the entire development program there were only two inadvertent rim failures. One of these was the direct result of a bearing failure that permitted the rim to rub against the housing. The second failure was attributed to an outer ring that was damaged during assembly of the rotor into the housing. The assembly procedures were revised; assembly tooling was modified; and there were no further rim problems during the program.

4.4.8.3.3 Inboard Bearing Selection

The rotor inboard bearing selected for the flywheel is a single-row, angular-contact bearing. The initial design utilized a matched pair of angular-contact bearings, but this duplex bearing was subsequently found to be too stiff radially.

4.4.8.3.4 Flywheel Outboard Bearing and Lube System

The flywheel outboard bearing requires its own separate lubrication system and oil because the inboard bearing is lubricated at ambient pressure in the gearbox and the outboard bearing is at the low pressure of 1 μ m Hg. During the development of this closed-loop oil circulating system, several changes were made to ensure that the flow was positive and that the heat

Figure 4.4.3-12. Flywheel Burst Test Results

transfer to the case was sufficient to keep the oil temperature within limits. Although the development was fairly straightforward, several versions and improvements were tested before selecting the final design.

4.4.8.3.5 Molecular Pump Development

The molecular pump initial design based on published theory and practice was found to be inadequate.

The use of previous theory required clearances between the rotor and stator of 0.10 mm or less. These close clearances would be unacceptable in an automotive application from both economic and dynamic points of view. Using modifications of existing theory, a design was achieved that permitted more realistic tolerances. The clearance used in the final design was 0.4 to 0.5 mm, and satisfactory performance was maintained. The carbon-face seal used to provide the vacuum in the rotor housing originally had been a point

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of concern. However, the initial seal selection and design performed as required and never had to be replaced. The seal was replaced several times during the program when disassembly would occur for other reasons.

4.4.3.8.6 Flywheel Losses

The flywheel losses are significant in this kind of a power system and must be kept as low as possible, otherwise the flywheel weight and energy consumption can obviate all of the advantages of the concept. In this Phase II program, the primary effort and priority was on the demonstration of the concept and the determination of the losses by actual measurement. There was insufficient time and resources to optimize this part of the system by repeated build and test cycles. However, the losses in the flywheel assembly were measured throughout the bearing development program. The losses measured included both bearings, both lubrication systems, windage losses, and lubrication losses. The losses at the maximum speed varied from 425 to 650 W and dropped off exponentially at the lower speeds. The lowest loss configuration utilized angular-contact ball bearings for both the inboard and outboard bearings. The design selected utilized a roller bearing for the outboard bearing because less development time was required although the losses were higher. Subsequent designs will utilize the lower-loss design supported by the data achieved during this development program.

4.5 BATTERY PACK

4.5.1 Description

The unique feature of the AiResearch flywheel-regenerative power system is the capability to create a load-leveled duty cycle for the battery. This eliminated the high-specific-power requirement and made possible a design emphasis on specific energy and cycle life. The load-leveled duty cycle allows greater use of the active battery materials, and use of long-life, tubular positive-plate construction. Compared to the golf-cart batteries, the active material is reduced by 20 to 25 percent, corresponding to a specific-energy improvement of approximately 2 W·hr/kg. In addition, the low current demand allows a significant reduction in the weight of the current-carrying parts. The weight of the internal current conductors for the load-leveled battery can be reduced by 45 percent compared to those for golf-cart batteries. This reduction corresponds to a specific-energy improvement of approximately 6 W·hr/kg.

The battery pack contains eighteen 6-volt batteries manufactured by Eagle-Picher Industries (Model EP200AH), a standard GC-2 size electric vehicle battery (Figure 4.5-1). This battery features tubular positive plate design, and grids containing no antimony for both the positive and the negative plates. Tubular plates retain the active materials, prevent shedding, and provide long cycle life. Eliminating antimony from the grid prevents antimony contamination of the negative plate, thereby providing long wet life, improving charge retention, and reducing maintenance. It utilizes the lead/lead dioxide electrochemical system with sulfuric acid as the electrolyte. Connections between batteries are lead plated, copper braided cable straps soldered to the terminals.

Figure 4.5-2 shows the construction of the tubular plate battery cell. The component parts for the complete 6-V module are shown in Figure 4.5-3. Dimensions and battery pack characteristics are shown in Figure 4.5-4.

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Figure 4.5-1. Eagle-Picher EP200 AH Battery

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Figure 4.5-2. Tubular-Plate Battery Construction

Figure 4.5-3. Tubular-Plate Battery Components

BATTERY PACK CHARACTERISTICS

NUMBER OF 6-V UNITS	18
WEIGHT OF ONE 6-V UNIT	26.9 kg (59.2 lb)
WEIGHT OF BATTERY PACK	484 kg (1066 lb)
CAPACITY, 6-V UNIT	
3-hr RATE	185 A·h 1071 W·h
TOTAL CAPACITY, 3-hr RATE	19.28 kW·h
SPECIFIC ENERGY, 3-hr RATE	39.8 W·h/kg (18.1 W·hr/lb)

S-28583 -C

Figure 4.5-4. Battery Outline Dimensions and Battery Pack Characteristics

The 18 batteries are located in a central longitudinal tunnel in the vehicle. The batteries are mounted in two rows in a lightweight, fiber-reinforced plastic tray, and each pair of batteries is strapped firmly to the tray and connected as shown in Figure 4.5-5.

The battery pack also includes a solid-state fault detection subsystem, which provides an LED signal on the dashboard when the first 6-V unit drops below 4.50 V. Fault detection wiring can be seen in Figure 4.5-5.

Battery temperature is monitored in the controller by thermocouples attached to the sides of two batteries. If either battery case temperature exceeds 57°C, the controller will illuminate a dashboard warning light.

4.5.2 Battery Performance

The voltage of a battery cell depends upon the potentials of the positive and negative plates with respect to the electrolyte solution. The potentials vary with the concentration of the electrolyte (at a fixed discharge rate and temperature), which in turn is a function of the discharge time. Hence, as the battery discharges, the voltage at the terminals falls gradually from its open-circuit value, until the end of the discharge is approached, when it begins to fall much more rapidly. Figure 4.5-6 shows the typical discharge curves for the EP200AH battery at 25°C.

Figure 4.5.5. Battery Interconnection

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Figure 4.5-6. Effects of Discharge Rate on Battery Voltage Profile

Low temperatures increase the resistivity and the viscosity of the electrolyte, reducing conductivity and impairing its circulation in the pores of the plates. The effect of temperature on battery voltage is shown in Figure 4.5-7.

The most commonly used method of rating battery capacity is in terms of ampere-hours for the specified rate of discharge and end voltage. Figure 4.5-8 shows the typical variations in the ampere-hour capacity of the EP200AH battery, as a function of discharge rate and temperature.

Another commonly used battery-performance indication is the relation between energy density and power, shown for the EP200AH battery in Figure 4.5-9.

The battery cycle life depends on a number of factors. If the battery has been operated and maintained properly, the cycle life becomes a function of environmental temperatures and the depth of discharges, as shown in Figure 4.5-10. The recommendation is made, therefore, to limit the environmental temperature to 38°C, and the depth of discharge to 80 percent during normal operation.

The effect of the battery cycle life on the total mileage available over the life of the battery pack is examined in Figure 4.5-11 for the 25°C ambient temperature case. Maximum utilization of battery stored energy is achieved by regular cycling to 50-percent depth of discharge.

The results of Eagle Picher life cycle tests are presented in the appendix to this report.

Figure 4.5-7. Effect of Temperature on Battery Voltage Profile

Figure 4.5-8. EP200AH Battery Discharge Time vs Discharge Rate at Various Temperatures

Figure 4.5-9. EP200AH Battery Energy Density vs Power Density at Various Temperatures

Figure 4.5-10. Cycle Life vs Depth of Discharge at Various Temperatures

Figure 4.5-11. Total Battery Pack Miles vs Nominal Miles per Charging Cycle

4.5.3 Battery Service Concept

As shown in Figure 4.5-5, the batteries are mounted in two rows in a light-weight, fiber-reinforced plastic tray. There are two runners on the bottom of the tray, covered with a Teflon-like material. These runners move on tracks (also made from Teflon-like material) along the bottom of the battery tunnel. The coefficient of friction between the runners and the tracks is approximately 0.10. Thus, a 474-N (107-lb) pulling force will be sufficient to slide the 484-kg (1066-lb) battery pack out of the vehicle.

The concept for removing and installing the battery pack is shown in Figure 4.5-12. The vehicle is lifted at its four jacking points until the wheels are off the ground, which ensures a fixed position for the battery tunnel during the battery-removal process. A battery cart, specially designed for the servicing operation, has Teflon-like tracks on its bed and a jackscrew for moving the tray.

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Figure 4.5-12. Battery Removal/Installation Concept

4.6 CONTROLS AND ELECTRONICS

The vehicle electrical system, through its interconnecting cabling and wiring, integrates all system components, and connects the power system with the vehicle controls, displays, and accessories. The locations of these components in the vehicle are shown in Figure 4.6-1.

The controller, battery charger, and data recorder are installed in the vehicle front compartment, while the power conversion unit and the switchgear are installed in the rear compartment. The battery power outlet is adjacent to the power conversion unit to minimize cabling and provide maximum protection. The battery high-voltage circuits are located in the rear of the vehicle, except for the charger interface, which is located in the front compartment (under the hood) for convenient access. All dashboard controls and displays and vehicle accessories are powered by a solid-state, 12-Vdc power supply for maximum safety--consistent with current automotive design practices.

Figure 4.6-1. Electrical/Electronic Component Arrangement

4.6.1 System Controls

The automatic system controls integrate the power system with the vehicle controls; monitor and process vehicle operating conditions and driver inputs; perform computations to generate commands for the power unit, switchgear, and dashboard indicators; and monitor power system conditions to provide protective controls. System controls effectively utilize stored battery and flywheel energy to maximize the vehicle range.

The main elements of the system controls are shown in Figure 4.6.1-1.

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Figure 4.6.1-1. System Control Concept

The digital processor includes the battery current schedule, the throttle torque schedule, the braking torque schedule, the armature command generator, and the compensated integrating control loops. The protective controls and vacuum and lubrication pump control modules (not shown in Figure 4.6.1-1) also are included in the processor.

The battery current schedule receives shift position, flywheel speed, and vehicle speed inputs and schedules the battery current demand based on these inputs. The throttle or braking torque schedule specifies the torque demand at the carrier by the position of the throttle or brake and the vehicle

speed. The armature command generator, which is the heart of the control system, calculates and generates proper motor and generator armature current commands that satisfy the battery current demand and the carrier torque demand simultaneously. Through the compensated, integrating-control loops, a set of commands to the armature chopper and the field choppers is determined by the differences between the generated motor and generator armature current commands and the armature current feedback signals. This set of output chopper commands from the digital processor then directs the respective choppers to control the power system to provide the carrier torque as requested and maintain the battery current discharge rate as scheduled.

4.6.1.1 Battery Current Schedule

Battery current is programmed according to the operational control mode. There are three different battery current schedules: prestart (PBS), reverse (RBS), and operational (OBS). During vehicle prestart, the battery current is programmed as a function of the flywheel speed and the relationship between generator speed and its base speed according to the prestart battery schedule shown in Figure 4.6.1-2. The PBS allows the flywheel to be charged to maximum speed within the shortest possible time during the prestart mode.

During vehicle reverse, the battery current is programmed as a function of throttle pedal position as shown in the reverse battery schedule, Figure 4.6.1-3. The RBS allows the motive power of the vehicle to be controlled directly by the throttle during reverse.

During normal forward driving modes, the battery current to the power system is regulated according to the operational battery schedule shown in Figure 4.6.1-4. As shown in the figure, the battery current is programmed as a function of flywheel speed. A typical urban driving schedule is shown in line a-b-c, which also is the initial base schedule. If the flywheel operates continuously below 20 000 rpm for more than one minute, the controller would shift the battery schedule upward to a new schedule that parallels the original schedule. The increment is determined by the difference between the battery current demand at that instant and the current demand at 20 000 rpm of flywheel speed on the original schedule. The new battery current schedule remains operative until the flywheel speed again operates continuously below 20 000 rpm or above 22 500 rpm for more than one minute. If the flywheel continuously operates above 22 500 rpm for more than one minute, the battery schedule would shift downward by the difference between the instantaneous battery current demand and the current demand at 22 500 rpm of flywheel speed on the schedule. These adjustments are limited to the urban driving schedule a-b-c, which is considered to be the minimum average current demand during normal driving, and the high-speed, 89-km (55-mph), cruise schedule a-d-c, which is considered to be the maximum average current demand.

A minimum limit for battery current, c-e-f, is imposed on the schedule, which increases the battery current demand rapidly whenever the flywheel speed drops below 18 800 rpm. Two maximum limits for battery current also are imposed, which limit the current to 140 A when both the motor and generator are above their base speeds and limit the current to 120 A when either the motor or the generator is below its base speed.

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Figure 4.6.1-2. Prestart Mode Battery Schedule

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Figure 4.6.1-3. Reverse Mode Battery Schedule

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Figure 4.6.1-4. Operational Battery Schedule

The operational battery schedule, through time-delayed self adjustment, can always gradually adjust itself to yield a battery current demand corresponding to the average road load under all driving conditions. It also has the ability to respond quickly by sharply increasing the current demand in the case of a sudden increase in average road load. This avoids exhausting the flywheel energy and placing the vehicle into the flywheel-low mode.

The OBS of Figure 4.6.1-4 was designed for use during driving typical of the Federal Urban Driving Schedule, which consists of many stop-and-go cycles and a variety of accelerations, speeds, and decelerations over a 23-min period. It will perform its function over other schedules such as constant-cruise, J227a(D)-type schedules.

Note that the battery current demand by OBS is always positive, which means that the battery is always in a discharge mode regardless of whether the vehicle is in an acceleration or regeneration mode.

When the battery is nearly discharged (less than 95 V) the OBS is modified to call for a reduced current of 45 A so that the flywheel energy will be used at a more rapid rate. This minimizes the residual energy remaining in the flywheel at system shutdown. The control system will shut down the vehicle when the battery voltage falls to 90 V. In normal driving, the time for the batteries to drop from 95 to 90 V is approximately 7 min.

If the flywheel rotational speed drops below the minimum flywheel capacity necessary to cover transient driving conditions, the control system will override the basic battery control current schedule as explained in para. 4.3.1.

4.6.1.2 Throttle Torque Schedule

Vehicle acceleration is a function of torque delivered at the axle, road load, and vehicle weight; however, maximum acceleration capability and smooth vehicle response depend on the control concept, computation, and quality of implementation. Figure 4.6.1-5 shows the envelope of the required throttle torque command as a function of throttle position for various vehicle speeds.

4.6.1.3 Braking Torque Schedule

The vehicle braking system includes regenerative (electrical) and friction (hydraulic) brakes as indicated by Figure 4.6.1-6. During regenerative braking, the rear wheels drive the flywheel (or charge the batteries) through the power unit, which creates the deceleration effect. The power from regenerative braking is stored in the vehicle flywheel during this period, when the drive motor functions as a generator, providing electrical energy to accelerate the flywheel. If the flywheel is saturated when electrical energy is being generated, the excess energy is used to charge the batteries. The braking torque schedule, implemented by the processor under various pedal positions and vehicle speeds, is shown in Figure 4.6.1-7.

Whenever the vehicle is in a driving mode, the regenerative braking is activated; its magnitude is a function of vehicle speed and pedal position. In an off-throttle driving mode, compression braking feel is created. In the neutral and reverse positions the regenerative braking is inoperative.

Figure 4.6.1-5. Throttle Torque Schedule

REGENERATIVE BRAKING LOGIC IS IN SOFTWARE

Figure 4.6.1-6. ETV-2 Brake System

Figure 4.6.1-7. Electrical Braking Torque Schedule

4.6.1.4 Armature Command Generator

The armature command generator performs the most important function of the digital processor by receiving the battery current demand and the carrier torque demand and calculating and generating the proper motor and generator armature current commands to satisfy the two demands simultaneously. Its logics are extremely simple. The motor armature current command is set proportional to the torque demand, and the generator armature current command is set to be equal to the motor current command minus the battery current demand. A positive armature current command denotes that the motor functions as a motor and the generator functions as a generator. Negative current command denotes that the motor functions as a generator and the generator functions as a motor.

4.6.1.5 Vacuum and Lube Pump Controls

The vacuum and lube pump controls operate independently of the controller, since flywheel vacuum and system lubrication must be maintained whenever the flywheel is above 5000 rpm. As shown in the block diagram (Figure 4.6.1-8), the oil pump motor is energized whenever the control system is operating, and is not deenergized until flywheel speed drops below a safe minimum speed, even if the main controller fails or is shut down. The cavity forepump also operates whenever the control system is operating and the flywheel is above minimum speed. The cavity pressure switch energizes and deenergizes the forepump motor in order to provide on-off control of cavity pressure to 1.0 mm HgA.

4.6.1.6 Protective Controls

Protective controls are incorporated to protect the passengers and prevent a major power system component failure or a combination of failures. As shown in Figure 4.6.1-9, design is based on the single-failure and hardware redundancy concept. Protective limits are listed in Table 4.6.1-1, and implementation is shown in Figure 4.6.1-10.

Figure 4.6.1-8. Vacuum and Lubrication Pumps Control

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Figure 4.6.1-9. Protective Control Functions

TABLE 4.6.1-1

PROTECTIVE CONTROL METHODS AND LIMITS

Parameter	Description	Control Limit
Flywheel overspeed	One sensor, independent of controller, dedicated to overspeed and pump control (1-s delay).	>25 750 rpm (1-s delay)
Vacuum overpressure	Pressure transducer (activated only after flywheel speed has exceeded 5000 rpm for 5 min).	>3.5 mm HgA
Generator overcurrent and motor overcurrent	Three current sensors for: generator armature current, I_G ; motor armature current, I_M ; and battery current, I_B . Protective system monitors I_G , I_G' , I_M and I_M' : $I_G' = I_M - (I_B - 30)$ $I_M' = I_G + (I_B - 30)$ <p>A 1-s delay prevents inadvertent trips.</p>	>270 A
Motor overspeed	One sensor, independent of speed limit control sensor.	>11 732 rpm
Motor and generator overtemperature	One thermistor in each unit provides an analog signal.	>204°C (400°F)
PCU overtemperature	A Klaxon switch on the armature chopper provides an overtemperature signal.	>121°C (250°F)
Lubricating oil temperature	Lube pressure is sensed by a switch. If pressure is below limit, switch closes indicating that an over-temperature condition will occur.	<3.0 psig
Battery failure	Fault detection system monitors each 6-V unit.	<4.5 V
Hydraulic brake failure	Sensor monitors fluid level in reservoir.	Low level

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EPPV PROTECTIVE CONTROLS BLOCK DIAGRAM

Figure 4.6.1-10. Protective Control Implementation

4.6.4.7 Ground Fault Protection

A ground fault interruptor and a circuit breaker are included for the 115-Vac, 60-Hz power.

4.6.4.8 Charge Indication

During charging, the charger provides the input to the dashboard battery power meter to indicate the status of the charge. At 110 V, the meter will indicate zero percent and at 151 V, the meter will read 100 percent. Also, a CHARGE-COMPLETE light on the charger will come on when the charger shuts down.

4.6.2 Controller

The controller (Figure 4.6.2-1) is a high-speed, 16-bit microprocessor-based computer that performs the functions of propulsion control including acceleration, cruise, regenerative braking, mode sequencing, speed and temperature limiting. The controller adjusts the power from the battery pack by substituting the flywheel energy source to optimize use of the battery energy. The controller adapts the battery current to changes in the average power requirement over the successive segments of the drive cycle, while the power for acceleration and braking is provided or absorbed by the flywheel.

The overall control strategy for vehicle propulsion is depicted in Figure 4.6.2-2. As shown in this figure, the controller loop and dynamics computation is based on commands derived from operator inputs and the status of the vehicle and power system variables. These commands are compared with feedback information provided by current, voltage, and speed sensors in a closed-loop fashion. The controller also contains a tape deck control interface that is used to format vehicle test data and periodically store it on a tape cassette for subsequent data reduction (not shown in Figure 4.6.2-2). Figure 4.6.2-3 shows the controller modules and circuit boards, which provide ease of functional checkout, troubleshooting, and maintenance. There are five plug-in boards: a signal-conditioning board, an input/output (I/O) control board, a central processor unit (CPU), data recorder buffer board, and a memory board, as well as an electrical interconnect board (motherboard).

4.6.2.1 Signal Conditioning Board

The signal conditioning board processes the signals from Hall-effect sensors for battery current, armature current, and generator armature current. Field current feedback for the motor and generator is sensed by measuring the voltage drop across precision resistors. Additional input sensors include the throttle (accelerator) sensor and brake sensor. The signal conditioning board conditions the sensor inputs in interface with a single analog-to-digital (A/D) converter (12 bits).

Motor speed, generator speed, and flywheel speed are sensed by variable-reluctance monopoles, providing a frequency output that is proportional to speed. The input conditioning board receives the analog, discrete, and monopole signals and shapes the raw signals to be compatible with the digital processor.

4.6.2.2 Input/Output (I/O) Control Board

The I/O control board is organized into several I/O ports, and each port is assigned a number for the instruction to reference. This circuit decodes the data bus for the specific port address. If a match is found, the CPU control lines are decoded to the input and output states. If an input instruction is found, the transfer buffer connected to the data bus is opened. If an output instruction is detected, the external latch is enabled to retain the content of the data bus.

Figure 4.6.2-1. Controller and Circuit Boards

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Figure 4.6.2-2. Control System Signal Flow Diagram

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Figure 4.6.2-3. Controller Modules

The system clock, generated by the CPU and the control lines, performs the system timing. Transfer of addresses and data takes place across the full-word parallel-data transfer bus. All analog voltages are transmitted through an analog multiplexer, an A/D converter, and a data buffer before entering the data bus. All speed signals are transmitted through a digital multiplexer, and are processed by use of a fixed time base and a counter.

4.6.2.3 CPU Board

The RF- μ 16 computer is a third-generation Garrett CPU design, based on more than seven years of development, testing, and production of a general-purpose processor. It is an engineer-oriented processor for medium-sized, dedicated processor applications, based on contemporary LSI (microprocessor) device technology. The heart of the processor is its arithmetic-logic unit (ALU). The ALU is incorporated on microprocessor-slice chips. It provides for logical operations, summation, and shifting of data words. Each device includes four bits of the ALU; the devices cascade to provide full-word, arithmetic-data-processing capability.

Like most modern processors, the RF- μ 16 uses a general register file for working accumulators and index registers. Arithmetic, logic, and data-transfer instructions can operate, using these registers as data sources, saving substantial time over similar memory-referenced instructions.

Control over the selection and processing of data is provided within the processor by the microprogram control section. Processor data-bus interfaces are fully microprogram controlled.

Input-output is controlled, with three separate control lines, to indicate (1) presence on the processor bus of a peripheral address, (2) presence on the processor bus of data being output to a peripheral, and (3) availability of the processor bus for data input from the last previously addressed peripheral. Finally, the processor provides an external test line, available to the input-output system, which can be used as a peripheral bus and tested under software control. This last feature enables the programmer to ensure that the addressed peripheral is ready to accomplish the intended input-output transfer function.

4.6.2.4 Memory Board

The major characteristics of the memory board are:

- Storage capability of 8000 word x 16-bit EPROM (expandable in 1000-word modules); 1000 word x 16-bit N-Channel RAM; and 512 word x 16 BIT CMOS RAM

- Provision for nonvolatile CMOS RAM operation

- Power switching on EPROM memory for minimum power dissipation

- Designed for 1.5- μ sec operation time (read or write)

- External memory enabled by single control line

4.6.2.5 Data Recorder Buffer Board

The buffer board uses a tape deck control interface to format data for storage and to generate hand-shake signals between the tape deck and controller. The onboard data recorder provides storage means for recording performance data during vehicle test periods.

4.6.2.6 Motherboard

The motherboard forms the base for the four plug-in boards and for the interconnects to the control box connector.

4.6.2.7 Software Design Description

The software architecture has a top-down hierarchical arrangement as shown in Figures 4.6.2-4 through 4.6.2-7. The software functional diagram is depicted Figure 4.6.2-8.

The software structure has been partitioned into approximately 27 modular subroutines to facilitate design refinements or modifications. The software is entirely in Garrett-designed RF- μ 16 assembly language with control applications and ease of use in mind. The simplified mode operation is depicted in Figure 4.6.2-9. The main flow chart in Figure 4.6.2-10 depicts intermodule structuring and the linkage between modules.

In general, the modules have been structured to keep each one functionally simple and isolated from the others. The main flow chart also depicts the simple in-line programming of software subsystems. The program execution time is less than 4 ms (worst case).

The RF- μ 16 has a unique branch-on-external-low (BXL) instruction set, which is used to test the internal single discrete input to the processor under software control. It provides a 5-ms clock to be the fundamental time interval to the real-time schedule and operation.

The BXL instruction also is used to monitor the health of the computer control system. During vehicle operation, the processor samples the heartbeat signal every 5 ms, and if it finds the signal not responsive on three successive samples, the backup electronics in the controller interprets this as "program being lost" or computer hardware failure. The backup electronics then disengages the motor/generator contactor to put the vehicle in a fail-safe mode. To engage the contactors, the system would require a reinitialization process. Functions associated with the important modules are described in the following paragraphs.

4.6.2.7.1 LOOP Module

The LOOP module, shown in Figure 4.6.2-11, computes and updates the state equations to generate current commands for vehicle propulsion drive. In every 5-ms time frame I_{fmc} (motor field current command), I_{fgc} (generator field current command), and I_{cc} (armature chopper current command) are implemented in terms of three different equations for solution by this software.

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Figure 4.6.2-4. Software Design Tree (Top Level)

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Figure 4.6.2-5. Software Design Tree (ARM CMD Sublevel)

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Figure 4.6.2-6. Software Design Tree (MODE TRANS Sublevel)

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Figure 4.6.2-7. Software Design Tree (I/O Sublevel)

Figure 4.6.2-8. Software Control Functions Block Diagram

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Figure 4.6.2-9. Simplified Mode Operation

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Figure 4.6.2-10. Control System Main Flow Chart

Figure 4.6.2-11. Block Diagram for Generating Current Commands to PCU

From the LOOP diagram of Figure 4.6.2-11:

$$I_{fgc} = \frac{K_3}{S} (I_{gerr} + I_{os})$$

where $I_{gerr} = I_{gc} - I_g$

$$\dot{I}_{fgc} = K_3(I_{gerr} + I_{os})$$

By Euler, $Y_{n+1} = Y_n + \dot{Y} \Delta t$

$$I_{fgc_{n+1}} = I_{fgc_n} + K_3 (I_{gcn} + I_{os}) \Delta T$$

For $\Delta t = 0.005$, $K_1 = K_3 = 1$, and $K_2 = 40$

$$I_{fgc_{n+1}} = I_{fgc_n} + 0.005 \left[(I_{gcn} - I_{gn}) + I_{os_n} \right]$$

$$I_{fmc_{n+1}} = I_{fmc_n} - 0.005 \left[(I_{mc_n} - I_{m_n}) - I_{os_n} \right]$$

$$I_{cc_{n+1}} = I_{cc_n} + 0.2 \left[(I_{mc_n} - I_{m_n}) - (I_{gcn} - I_{gn}) \right]$$

The three difference equations above represent the physical relationships between parameters and are not scaled for the computer, which in this case employs fixed-point fractional number computation.

4.6.2.7.2. Read Speed Timing

The read-speed timing diagram, shown in Figure 4.6.2-12, provides a means of describing the task graphically.

As seen in the diagram, the complete sample time for all speed channels is 265 ms. During initialization, INIT flag = 1 (see Figure 4.6.2-10) prevents the program from computing armature current command based on erroneous speed inputs.

The initialization process is complete when a complete sample of speed channels is obtained. The program is then allowed to follow through the main routine and compute torque commands, armature current commands, or any pre-scribed schedule.

Figure 4.6.2-12. READ-SPEED Timing Diagram

4.6.2.7.3 Armature Command

The fundamental scheme for generating the armature current command for both machines is shown in Figure 4.6.2-13. As seen from the diagram, I_{mc} (motor armature current command) is essentially a function of torque command. The torque command is a prescribed schedule that depends on throttle or brake input and vehicle velocity.

The armature command routine also includes the derating of I_{mc} and I_{gc} (generator armature current command) according to the computation based on power limit, flywheel overspeed, motor overspeed, and battery overvoltage.

The appropriate battery schedule is selected by the driver and the state-of-the-vehicle input. The functional concept and system implementation is explained in detail in Section 4.6.1, System Controls.

4.6.2.7.4 Transition Routines (MODE Sequencing Function)

The controller provides interlock and smooth transition between various modes and submodes of vehicle operation. The controller, as a sequential machine, operates the electrical subsystem and shifts between different modes and submodes. The controller examines operator input and vehicle parameters to initiate the appropriate transition between the available modes.

Fundamentally, the transition process is a logical control. The computation task is effectively kept separated from it. The sequential controller detects a possible new mode candidate, and the associated mode transition is executed or else it loops back to the neutral state. The new mode is entered after it passes all the decision tests and after the report-back signal is verified as being true. If report-back is not true, then the transition is aborted and the control system stays in the previous mode configuration.

Figure 4.6.2-13. Armature Current Commands

Figure 4.6.2-14 represents the essential structure of the mode transition process.

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Figure 4.6.2-14. Structure Chart of Mode Sequencing

4.6.3 Power Conversion Unit

The power conversion unit (PCU) conditions vehicle battery power and applies power selectively to the power unit motors as commanded by the vehicle controller. Power unit monitoring and protection is also provided, and status information is fed back to the controller.

The PCU consists of the functional groups listed below and interconnected as shown in Figure 4.6.3-1.

- 120-A transistorized armature chopper to control armature voltage
- Two 15-A motor/generator field choppers to control field current
- Five contactor drivers:
 - Main line contactor
 - Motor contactor
 - Generator contactor
 - Armature chopper bypass contactor
 - Reverser contactor
- 620-W accessory power supply to provide 12-V power for all vehicle accessories
- Vacuum and lubrication pump controls to maintain an appropriate environment whenever the flywheel is spinning
- Protective controls to provide independent backup monitoring and disconnect in the event of a controls system failure

Figure 4.6.3-1. Power Conversion Unit Interconnections

The power conversion unit is designed to be installed in an electronics compartment behind the vehicle rear seat support structure (Figures 4.6.3-2, -3, and -4). The PCU electronics are contained in two assemblies, one on each side of the switchgear (contactor) compartment. The unit is contained in a shielded enclosure, which contains any generated electromagnetic interference.

Cooling is provided by air drawn by a PCU fan and ducted from the aft end of the vehicle through the switchgear compartment, both PCU's, and on to the motor/generator units. Approximately $0.028 \text{ m}^3/\text{s}$ (60 cfm) of air flows through each PCU, which under worst-case conditions limits maximum air temperature rise to 13°C (23°F). With an ambient air temperature of 28°C , the PCU heat sink temperature will be approximately 41°C .

The power electronics subunit mechanizations, loads, and power requirements are described in the following paragraphs.

4.6.3.1 Armature Chopper

The armature chopper converts the 108-V battery voltage to a lower average voltage, via a pulse-width-modulation/regulation scheme. Its functional diagram is shown in Figure 4.6.3-5.

The chopper output drive is rated at a current of 120 A. This is achieved by operating seven matched lower-rated transistors in parallel.

A single comparator is used to generate a 2000-Hz reference signal and clock. The error signal, generated as the difference between the armature chopper current command and the current feedback signals, is compared to a reference signal to generate a duty-cycle control. An inhibit signal turns off the drive signal in an overcurrent condition.

The armature chopper also contains on-board fast-acting current-limit circuits that provide virtually instantaneous protection against maximum power and overcurrent for the power transistors. Several key parameters, listed below, are sensed to ensure proper chopper operation.

- Loss of 5-V power for base drive
- Peak armature chopper current $>180 \text{ A}$
- Average armature chopper current $>140 \text{ A}$
- Power transistor saturation ON time $>10 \mu\text{s}$
- Greater than 260 A in generator current sensor
- Greater than 260 A in motor current sensor

If one of the parameters above is active (chopper malfunctioning), the local current limit control holds off the power transistor drive independent of the controller command until the armature current falls and returns to the safe region, at which point the current will follow the armature current command from the controller.

81287-1

Figure 4.6.3-2. PCU Components

80988-2

80988-4

F-32756

Figure 4.6.3-3. PCII Assembly

A-6366

F-32757

Figure 4.6.3-4. PCU Installation

Figure 4.6.3-5. Armature Chopper Functional Diagram

S-20431-c

Figure 4.6.3-6. Field Chopper Functional Diagram

S-20435-c

When the chopper is full ON (i.e., operating at its maximum duty cycle, ON time/period >99 percent), the armature chopper provides a discrete maximum chopper status to the controller, which issues chopper by-pass contactor closure command and further sets the armature chopper command to zero voltage to keep the power drawn by base drive circuitry to minimum.

4.6.3.2 Field Choppers

The motor and generator use identical field choppers, with a rated operating current of 15 A. The field chopper (Figure 4.6.3-6) is a self-oscillating current regulator. Both frequency and duty cycle vary as a function of input command, load variations, and supply voltage.

Current feedback is supplied via a $10\text{-}\Omega$ ground-leg shunt. The feedback is scaled and is compared to the input command at a summing integrating amplifier. Accompanying transistors form a bang-bang switch with hysteresis. Driven by the integrator output, this switch performs the necessary regulation functions with the field current following the input command.

4.6.3.3 Contactors Drivers

The contactor drivers receive a discrete command from the controller. In response to this command, the circuit applies a starting pulse to the contactor coil and, after a fixed period of time, the current/voltage is dropped to a lower average value to reduce power dissipation.

The contactor drive circuit, Figure 4.6.3-7, employs a one-shot, 0.25-s pulse to provide the initial starting pulse; thereafter, the lower current/voltage is maintained by pulse-width frequency modulation, with the drive transistors using a current-bandwidth-regulating loop.

4.6.3.4 Accessory Power Supply

The accessory power supply is a lightweight, high-efficiency dc-to-dc converter that converts the 108-Vdc battery power into 12-Vdc power used by the vehicle accessory loads listed in Table 4.6.3-1. However, not all of the loads listed in the table will be imposed simultaneously. The maximum design load is 542 W, as shown in Figure 4.6.3-8. The power supply circuit is shown in Figure 4.6.3-9.

TABLE 4.6.3-1

ACCESSORY POWER SUPPLY LOAD

Item	Load, W
Controller	47.2
Data recorder	4.1/5.3 (without/with motor running)
Wiper	22.4/40.1
Annunciator	1.8
Defog/vent/heat fan	23.6
Headlights	106.2/54.1 (high/low)
Parking and tail lights	37.7
Panel and under-dash lights	17.4
Turn signal	35.4
Back-up light	38.3
Brake light	33.1
Horn	47.2
Windshield washer	11.8
Wiring	1.8/40.0
Heater glow plug	227.1 (up to 4 min only during startup)

ECONOMIZER CIRCUIT - FULL LINE VOLTAGE FOR 0.25 s
800 mA PULL-IN, 0.12-A HOLDING CURRENT (14 V)

S-20437.c

Figure 4.6.3-7. Contactor Drivers Functional Diagram
 (Four Drivers per Unit)

A-7044

Figure 4.6.3-8. Accessory Power Supply Voltage-Load Characteristic

A-5934 -A

Figure 4.6.3-9. Accessory Power Supply Functional Diagram

4.6.4 Battery Charger

An onboard battery charger (Figure 4.6.4-1), designed to operate in an ambient temperature range of -20° to $+60^{\circ}\text{C}$, will recharge the battery pack overnight from a 30-A, 115-Vac, 60-Hz household service. A switch on the charger will allow it to operate on a 15-A household circuit; however, charging time would be longer. Automatic functions are included to monitor the state of charge and the temperature of the batteries and to shut down the charger when charging is complete or an overtemperature condition occurs. Functional elements are shown in Figure 4.6.4-2.

4.6.4.1 Charging Modes

The charger operates in two modes. Charging begins in a constant-current (21- to 25-A) mode and continues until the charging voltage reaches 151 Vdc. At that point, charging will switch to a constant-voltage mode with voltage at 151 V (+0, -2). Current will drop quickly at the switch point to a new level and decrease slowly or hold steady during constant-voltage charging.

The rate of change of voltage during the constant-current mode and the current values during the constant-voltage mode will vary depending on the original state of charge and temperature of the battery. Typical battery charger characteristics in Figure 4.6.4-3 show the variation of input and output currents and efficiency as a function of the battery voltage.

Charging duration is determined by a 3-h timer, which starts when the charging voltage (during constant-current mode) reaches a trip point that is a function of battery temperature as noted below:

<u>Temp, $^{\circ}\text{C}$</u>	<u>Timer Trip Point, Vdc</u>
0	143.9 to 146.8
25	137.7 to 141
50	131.1 to 134

At the end of the 3-h period, the charger will shut down. Approximately 8-1/3 h is required for a full charge.

A temperature-sensing function on the charger protects the battery against overheating during charging. If the battery case temperature reaches $57 \pm 3^{\circ}\text{C}$, the charger will slow the charging rate so as not to exceed this limit.

Two thermistors attached to a battery case at two representative locations provide the temperature input. The higher of the two temperatures is determinant.

In the event of an external ac power failure longer than one second, the charger will restart the charging cycle when power returns.

80988-7

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F-32746

Figure 4.6.4-1. Battery Charger and Components

Figure 4.6.4-2. Battery Charger Functional Elements

Figure 4.6.4-3. Typical Charger Input-Output Characteristic and Efficiency

4.6.4.2 Basic Charging Operation

The charger operates on the basic scheme of a full-wave rectifier driving current through an inductor, as shown in Figure 4.6.4-4.

A-6085

Figure 4.6.4-4. Charger Operating Principle

The wide ranges of battery and line voltages during the charging cycle generate the requirement for a variable-inductor control to vary the conduction angle of L . With no control circuit, current may increase in L only when $E_{in} > V_B$. To overcome this restriction, switch S is incorporated, which grounds the inductor and increases the voltage across it from $|E_{in}| - V_B$. Since this represents an increase of 90 to 140 V across L , the potential current in L is dramatically increased ($E = L (di/dt)$).

The control circuit implemented in the charger grounds L just prior to the $E_{in} > V_B$ point, allowing the inductor current to ramp up and then transfer to the battery at a point where natural conduction will continue. The basic voltage and current waveforms of Figure 4.6.4-5 show how the current control is accomplished by adjusting t_{on} , which controls the initial battery current I_o .

The actual current is monitored, averaged, and compared to the desired current by an error amplifier. The error amplifier shown in Figure 4.6.4-6 produces a signal that is proportional to the desired turn-on point of the input waveform.

In addition to the current control loop, there is a battery voltage monitor, which allows the charger to supply current until the battery voltage has reached a temperature-dependent limit or the 3-h timer expires.

4.6.4.3 Cooling

A fan on the charger provides cooling air at 0.019 to 0.028 in m^3/s (40 to 60 cfm), which also ventilates the battery tunnel. The fan motor is designed for a nominal 125-Vdc, 0.47-A input, but will operate over a battery voltage range of 90 to 151 Vdc.

A-6052 -A

Figure 4.6.4-5. Charger Voltage and Current Waveforms

Figure 4.6.4-6. Charger Control Logic

4.6.4.4 Flasher Control

A flasher power supply is included to flash the parking and tail lights in an emergency when all other electronics are inactivated. The unit is operated by an external switch on the charger. Operating parameters are as follows:

Input: 108 Vdc nominal, 90 V min. to 151 V max.

Output: 12 Vdc, 8 A to flasher assembly at 108-Vdc input; range is 11 to 16 Vdc. Flasher duty cycle is 50 percent.

4.6.4.5 EMI Suppression

A filter is included to suppress 1 μ V maximum at a distance of 3 m at frequencies from 30 to 3000 MHz.

4.6.4.6 Startup Logic

When the operator turns the key ON, the charger supplies 12-V power to close the main battery contactor. This provides power to the PCU, which supplies the 12-V power to the controller.

4.6.6 Data Recorder

The data recorder is a digital cassette tape deck, with read/write electronics, used for recording various performance parameters during vehicle operation. It is a standard Teac unit, PN MT-06, chosen for its low power requirements and compact and lightweight structure. The block diagram (Figure 4.6.6-1) shows the major elements of the system and how they are implemented.

The recorder provides the storage means for recording data during test, and the cassette format simplifies data transmittal to a computer center for subsequent data reduction. A tape control interface is built inside the controller for formatting data and generating a handshake signal between the data recorder and the controller. The recorder uses standard Philips type cassettes (certified data recorder cassettes). Cassette capacity depends on the number of parameters sampled, the parameter sampling rate, and the tape recording density.

Figure 4.6.6-1. Data Reduction Block Diagram

Following are the parameters to be stored during vehicle test.

- Flywheel speed 0 to 30,000 rpm
- Vehicle speed 0 to 90 mph
- Battery voltage 90 to 150 V
- Battery current -250 to +250 A
- Battery temperature 0° to 150°F
- Motor temperature 250° to 450°F
- Generator temperature 250° to 450°F
- Altitude (pressure) 12.5 to 15.0 psia

These eight parameters are sampled once per second and are transferred to the data recorder once every eight seconds. With this data recording format, tape capacity is four hours per side.

The onboard recorder was intended for use during actual city driving tests and was to provide sufficient data to permit the analysis of vehicle performance during various traffic conditions. Because this part of the testing was eliminated from the program, the recorder was not used.

4.6.7 Electromagnetic Interference (EMI)

The power conversion unit incorporates four choppers that are a source of electrical noise. The dc motor commutators and contactors also generate electromagnetic noise that must be suppressed. The design is based on the following:

- (a) Metallizing nonmetallic vehicle body parts for noise containment (Figure 4.6.7-1); i.e., power conversion and switchgear enclosures, and adjacent ducting
- (b) Suppressing dc motor commutators with capacitors in the motor housings (motor/generator and cooling fan)
- (c) Shielding all 108-Vdc cables (braided insulation)
- (d) Filtering all auxiliary 108-Vdc motor power supplies
- (e) Filtering all 12-Vdc power supply outputs
- (f) Separating all control cabling and power supply cabling from other cabling; shielding and grounding them as appropriate
- (g) Isolating the 108-Vdc power circuits from the 12-Vdc, ± 12 -Vdc, and 5-Vdc circuits
- (h) Circuit layout shielding and detail design to provide adequate EMI protection and suppression of internally generated EMI noise

A-5979

Figure 4.6.7-1. EMI Suppression

4.7 VEHICLE

The vehicle description begins with the structural evaluation and dynamic analysis (para. 4.7.1 and 4.7.2), which led to the final body design described in para. 4.7.3. It is interesting to note that the all-plastic unibody concept saved approximately 272 kg (600 lb) over a steel sheet metal design. The suspension, steering, braking, passenger heating and cooling, and driver controls are covered in para. 4.7.4 through 4.7.7. A brief discussion of vehicle noise is given in para. 4.7.8.

4.7.1 Structural Design Evaluation

Design of the fiber-reinforced plastic body and vehicle structures was based on scale model static crush tests and the finite-element structural analyses listed below:

Scale Model Tests

- Single battery
- Battery column
- Battery tunnel
- Energy absorber
- Front end

Structural Analyses

- Static bending and torsion loads on body structure
- Dynamic bending and torsion loads
- Power unit supports load
- Crashworthiness

4.7.1.1 Single Battery Tests

The first phase of component testing was to determine the crush characteristics of the batteries. The primary concern was to establish the ability of the 484 kg (1066 lb) of batteries to absorb their own energy in a crash situation. Existing plate-type batteries were used throughout the test and design program because the new tubular-plate batteries developed specifically for the ETV-2 were not available during the structural design period. The impact and crush characteristics of the tubular type are not expected to be significantly different.

The first tests performed on the batteries were static crush tests. Exide EV-106 golf-cart batteries were procured, filled with water, and subjected to both static and dynamic crush tests. For the dynamic tests, batteries were drop-tested at 9 to 12 m/s (30 to 40 fps). Tests were performed with the force applied in the direction parallel to, and then perpendicular to, the battery plates as indicated by Figure 4.7.1-1.

Based on the results shown in Figures 4.7.1-2 through 4.7.1-4 and packaging considerations in the vehicle tunnel structure, the decision was made to orient the battery with the plates perpendicular to the frontal-barrier crash direction.

4.7.1.2 Battery Column Crush Tests

A full-scale battery-tunnel structure was fabricated of fiberglass-reinforced polyester to accommodate a single row of 11 Exide EV-106 batteries. Figure 4.7.1-5 shows the tunnel after a 48.3-km/h (30-mph) impact. The first five batteries were crushed, the next four showed some damage, and the last two were intact. The tunnel was undamaged behind the simulated firewall, which was located 838 mm (33 in.) from the front. There would have been no acid spillage or intrusion into the passenger area. The battery-column crush distance was about 533 mm (21 in.).

S-20451

Figure 4.7.1-1. Battery Test Orientation

S-47602

Figure 4.7.1-2. Single-Battery Static Crush Force Characteristics

Figure 4.7.1-3. Crushed Battery with Force Applied Parallel to Plates

Figure 4.7.1-4. Single-Battery Dynamic Crush Force Characteristics

F-32760

Figure 4.7.1-5. Crushed Battery Column

The crush-force characteristic results are shown in Figure 4.7.1-6. With the exception of the high initial peak that is attributable to the battery stack, the results were approximately as anticipated, demonstrating that the battery configuration was feasible from an energy-management standpoint. The aggressiveness of the battery column does not exceed that of the rest of the front-end structure, and the batteries, when decoupled from the structure, absorb their own kinetic energy.

4.7.1.3 Battery Tunnel Crush Tests

The high peak force from the front end of the tunnel (note the peak in Figure 4.7.1-6) was undesirable. The objective was to provide a trigger mechanism in the forward section that would reduce the peak and provide for a controlled crushing reaction. Three approaches were tested to obtain this effect: lead-in slits, incremental slits, and a corrugated front section. The corrugated section (Figure 4.7.1-7) was the most successful. The force levels were well within acceptable values on a full-scale model (Figure 4.7.1-8), so this technique was chosen for the final design.

4.7.1.4 Energy-Absorbing Element Tests

Two full-scale energy-absorbing elements were constructed, one with incremental slits and one with corrugation. Both were filled with 32 kg/m^3 (2 lb/ft^3) rigid polyurethane foam and were impact tested at 48.3 km/h (30 mph). Both controlled crush with very little initial peaking, but the corrugated

F-32761

Figure 4.7.1-6. Battery Column Crush Force Characteristics

Figure 4.7.1-7. Corrugated Tunnel End Trigger Mechanism

F-32762

Figure 4.7.1-8. Corrugated End Crush Force Characteristics

design was chosen, since it would integrate better with the corrugated battery tunnel. Its force-deflection curve is shown in Figure 4.7.1-9.

4.7.1.5 Front-End Crash Tests

Following the tests on the battery tunnel and the energy absorber tubes, the front-end structure was assembled to determine how it behaved as a system. Three half-scale front-end models were built. The first two consisted of only the basic structural elements. The third model was more elaborate, including fenders, hood, cowl, A-post, toe board, and side sills.

The first model included a battery tunnel, energy-absorbing tubes, a front bulkhead, a bumper system, and a battery tray, with motorcycle batteries and honeycomb material to simulate battery stiffness. After impact at 48.3 km/h (30 mph), failure was not as intended and was unsymmetrical. The battery tunnel buckled as opposed to crushing evenly. In addition, the model crushed only 67 percent of the amount required for the desired force vs deceleration.

In the second model, bumper strength was reduced to fail at impacts over 16 km/h (10 mph), bumper position was corrected, and the eccentricity of the first few corrugations on the energy absorbers and tunnel was increased. In a 48.3-km/h (30-mph) impact, its crush distance was slightly greater than 250 mm (10 in.). The force-displacement curves are shown in Figures 4.7.1-10 and 4.7.1-11, projected to full scale.

Figure 4.7.1-9. Corrugated Foam-Filled Energy Absorber Force Characteristics

Figure 4.7.1-10. Front End Model 2 Tunnel Characteristics (Model results converted to full scale)

Figure 4.7.1-11. Front End Model 2 Energy Absorber Force Characteristics
(Model results converted to full scale)

The following numerical example shows how close test results from model 2 came to prediction.

Vehicle kinetic energy at 48.3 km/h (30 mph) with 2 passengers (less batteries), J (ft-lb)	85 958	(63 400)
Predicted average force for 0.6-m (2-ft) crush, N (lb)	141 008	(31 700)
Actual average force for structural elements, N (lb)		
Tunnel	39 785	(8 944)
Absorbers	<u>103 278</u>	<u>(23 218)</u>
	143 063	(32 162)

Thus, the test average force was only 1.5 percent higher than predicted. This does not mean that all future models and vehicles will be within this precision, but it does indicate that the basic mechanisms and techniques used in the ETV-2 energy management are effective, understood, and predictable.

The third half-scale front-end model was much more elaborate and detailed. Figure 4.7.1-12 shows the front of the unit and the cowl, wheel houses, energy absorbers, and bumper system. The model has a battery tray and motorcycle batteries. Figure 4.7.1-13 is the interior of the model and shows the tunnel, A-posts, seat riser, and toe board.

Before-and-after test photos of model 3 are shown in Figures 4.7.1-14 and 4.7.1-15. The crush distance was slightly over 305 mm (12 in.) as desired, and there was no damage or intrusion into the passenger compartment. The force-displacement curve is shown in Figure 4.7.1-16, projected to full scale. The test was a complete success.

4.7.1.6 Finite-Element Static Analysis

The structure of the electric vehicle was modeled using the beam and panel elements as shown in Figure 4.7.1-17. Except for the location of the power unit, the vehicle is symmetrical about the longitudinal axis. Consequently, only one-half of the structure was considered in the analysis. The effects of various bending and torsion loadings were examined by varying the boundary conditions along the plane of symmetry. The static response of the vehicle was examined under the loading conditions discussed in the following paragraphs.

A 3.0-g* vertical load, acting on the structure, was simulated as a vertically applied loading condition. The weight of the batteries was distributed along the tunnel, and the passenger weights were applied at the front- and rear-seat locations. The weight of the power unit, acting at the rear, also was included. The deformation of the structure under this loading condition is shown in Figure 4.7.1-18. For the analysis, the structure was supported at the top of the front- and rear-wheel housings. Under this loading condition, the maximum deflection of 2.3 mm (0.091 in.) occurred in the tunnel. The side sill in the door opening deflected 0.99 mm (0.039 in.) The maximum stresses occurred at the front shock tower designated as Area 3. The locations and magnitudes of the stresses are shown in Figure 4.7.1-19. Fatigue load was 3.0 g while 6.0 g was for the ultimate condition.

A torsional loading condition was simulated to evaluate the torsional rigidity of the vehicle. For this loading condition, the rear end of the vehicle was held rigid, while the front end was twisted. The result is shown in Figure 4.7.1-20.

A 6.0-g end load was applied to the front energy-management area of the structure to study its deformation and determine how the load is distributed into the structure ahead of the passenger compartment.

A 6.0-g horizontal load also was applied to simulate the rear impact on the vehicle. It is important to realize that, in practice, such a peak load is of very short duration, and the severity is localized; however, the static analysis will be useful in identifying potential problems in the rear structure and load-distribution paths.

*g = acceleration of gravity at sea level

Figure 4.7.1-12. Half-Scale Front End Model 3 Front View

Figure 4.7.1-13. Half-Scale Front End Model 3 Passenger Compartment View

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Figure 4.7.1-14. Half-Scale Front End Model 3 Before Crash Test

Figure 4.7.1-15. Half-Scale Front End Model 3 After Crash Test

F-32765

Figure 4.7.1-16. Total Force Characteristic Scale Model 3

Figure 4.7.1-17. Finite-Element Static Analysis Model
(363 Nodes, 154 Beams, 366 Plates)

Maximum Deflections:
Side sill 0.99 mm (0.039 in.)
Tunnel 2.31 mm (0.091 in.)
Rear sill 0.58 mm (0.023 in.)

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Figure 4.7.1-18. Finite-Element Static Analysis for Deflection Under 3.0-g Vertical Load

Stress In										
Load	Panel 164 - 172		Beam 166 - 170		Beam 88 - 170		Beam 170 - 98		Beam 170 - 68	
	kPa	psi	kPa	psi	kPa	psi	kPa	psi	kPa	psi
3 g	7171	1040	16340	2370	8652	1255	9445	1370	15341	2225
6 g	14341	2080	32681	4740	17306	2510	18891	2740	30682	4450

Figure 4.7.1-19. Finite-Element Static Analysis Locations and Magnitudes of Maximum Stress Due to 3.0-g and 6.0-g Vertical Loads

TOTAL APPLIED TORQUE AT FRONT
WHEEL OF 2011 N · m (1483 ft-lb)

TOTAL ANGULAR DEFLECTION AT THE
FRONT WHEEL OF 0.163 deg OR A
TORSIONAL STIFFNESS OF 12 340 N · m
(9100 ft-lb) PER DEGREE

A-6375

Figure 4.7.1-20. Static Torsional Mode.

4.7.1.7 Finite-Element Dynamic Analysis

A dynamic analysis was performed to obtain the bending and torsional modes of the vehicle for the following cases:

- (a) A stripped vehicle, containing only the structural elements
- (b) A completely loaded vehicle, including doors, hood, and windshield

The finite-element dynamic analysis procedure is outlined in Figure 4.7.1-21. The finite-element model used for the dynamic analysis was nearly identical to the one used for the static analysis, and is shown in Figure 4.7.1-22. The whole structure was assumed to be symmetrical about the center-plane, and the analysis was divided into a symmetric and an asymmetric case of the half-structure. The battery weights were distributed as lumped masses along the length of the tunnel. The weights of the passengers and the power unit also were included in the model as lumped masses, attached at appropriate locations. The dynamic model of the vehicle was supported on a soft suspension, approaching a free-free condition.

Owing to the large number of nodes (276), and the resulting degrees of freedom (1656), a condensation of dynamic degrees of freedom was necessary. The procedure essentially involved the designation of certain degrees of freedom as "masters". The remaining degrees of freedom were termed "slaves" and were eliminated, thus reducing the size of the problem. The elimination technique, sometimes called Guyan reduction, preserves the potential energy of the system, but modifies to some extent the kinetic energy. The kinetic energy

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Figure 4.7.1-21. Flow Diagram for Finite-Element Dynamic Analysis

FINITE ELEMENT MODEL
 TOTAL NUMBER OF ELEMENTS = 425
 TOTAL NUMBER OF NODES = 276

S-22563

Figure 4.7.1-22. Dynamic Analysis Model (AiResearch)

of the lower-frequency modes is less sensitive to the condensation than the kinetic energy of the higher-frequency modes. The selection of the "masters" for dynamic degrees of freedom is somewhat comparable to the method of selecting lumped masses for a dynamic model. In general, only dynamic degrees of freedom at the points of significant mass and moment of inertia are needed.

The first bending mode and the first torsional mode of the basic structure are shown in Figures 4.7.1-23 and 4.7.1-24. A vehicle body frequency interference diagram is shown in Figure 4.7.1-25.

4.7.1.8 Power Unit Supports

The electric vehicle power unit supports were designed to ensure an uncoupled system with respect to other vehicle frequencies, as well as a load path for the large gyroscopic moments and forces. With respect to frequencies, the recommended isolators result in a vibration between 10 and 20 Hz. This avoids coupling with the vehicle suspension (2 to 3 Hz) and the anticipated vehicle local mount structure first-mode frequency of 30 Hz. Large loads and deflections were minimized by locating the supports as far away from the power unit center of gravity as possible. The selected locations result in a maximum load of 4702 N (1057 lb) and an un-snubbed displacement of 22.3 mm (0.876 in.) for the limit-load yaw condition of 1.42 radians per second. The snubbing effect of the mounts limits deflection to less than 13 mm (0.5 in.).

4.7.1.8.1 Support Geometry

The locations of the supports (numbered 1 through 4) are shown in Figure 4.7.1-26. The center of gravity of the unit (c.g.) is assumed as the coordinate system origin, and it lies on the centerline of the vehicle; however, the c.g. does not lie on the rotational axis of the flywheel, being slightly forward and below. Also shown in the figure are the pitch, roll, and yaw axes with respect to the vehicle body.

4.7.1.8.2 Loads

The loads shown in Table 4.7.1-1 are given with respect to the power unit coordinate system. The limit load condition was used to determine reaction for this analysis.

4.7.1.8.3 Isolators

The isolators are Barry Mount series 507. These meet the stiffness and load requirements and are available as off-the-shelf items. They are also available with snubber washers that minimize the deflection. The isolator characteristics are as follows:

Mount Number	Stiffness $KX = KY = KZ$,	
	kN/m	(lb/in.)
1 and 2	403	2300
1 and 3	175	1000

Figure 4.7.1-23. Dynamic Results for First Bending Mode
(Frequency = 16.077 Hz)

Figure 4.7.1-24. Dynamic Results for First Torsion Mode
(Frequency = 23.893 Hz)

S-47360

Figure 4.7.1-25. Vehicle Body Frequency Interference Diagram

Figure 4.7.1-26. Location of Power Unit Supports

TABLE 4.7.1-1

POWER UNIT PRELIMINARY LOAD FACTORS

Load	Y, g	X, g	Z, g	Roll, rad/s	Pitch, rad/s	Yaw, rad/s
Design	1.0	0.7	2.0	0.3	0.3	0.5
Limit	6.0	2.5	3.0	0.6	0.6	1.42
Crash	10.0	10.0	10.0	6.0	-	6.0

NOTE 1. All loads and rates assumed to be applied independently.

2. An additional 1-g vertical, static, downward load was applied in all above cases.

$$M_{GYRO} = 2752 \Omega \text{ N}\cdot\text{m} \text{ (2030 } \Omega \text{ ft}\cdot\text{lb)}$$

where Ω is the roll or yaw angular velocity in rad/sec

4.7.1.8.4 Method of Analysis

To eliminate strong model coupling, a computer program was developed to compute the required isolator stiffness for an uncoupled system, given a specific set of support locations. Standard isolators, which closely resemble the required stiffness, were used in a computer program developed to compute support loads and natural frequencies of a rigid body mounted on a redundant spring suspension. The program includes the gyroscopic effect of the flywheel in computing the support reactions, but it does not include the gyroscopic stiffening in computing the natural frequencies. The gyroscopic effects on the natural frequencies have been calculated and added to the frequency summary.

4.7.1.8.5 Natural Frequencies

Table 4.7.1-2 summarizes the natural frequencies and mode shapes. It should be noted that the gyroscopic effects are calculated from an equation that assumes an uncoupled system. The mode shapes indicate coupling in the YY and ZZ directions. Thus the values representing the gyroscopic effect are an upper limit. The actual frequency is somewhere between the two.

4.7.1.8.6 Deflections of Supports

The deflections at the supports are due to the limit load condition. Limit load displacements are tabulated in Table 4.7.1-3. No consideration has been given to snubbing, which is available on the isolators. The installation will include deflection-limiting devices.

TABLE 4.7.1-2

FREQUENCY SUMMARY

Natural Frequency, Hz	8.7	11.3	11.8	12.5	15.5	18.4
With Gyro Effects	23.0	--	--	--	26.0	--
X	0.006	0.038	0.997	0.067	0.036	0.013
Y	0.087	0.994	0.044	0.094	0.963	0.260
Mode shape (Motion of c.g.)	Z	0.982	0.095	0.063	0.992	0.104
XY	0.020	0.0	0.0	0.007	0.038	0.962
YY	0.151	0.002	0.0	0.043	0.028	0.051
ZZ	0.015	0.026	0.0	0.005	0.241	0.063

TABLE 4.7.1-3

LIMIT LOAD DISPLACEMENTS

Mount Number	Load Direction and Magnitude	Displacement (Unsnubbed)					
		X		Y		Z	
		mm	in.	mm	in.	mm	in.
1	Fx = 2.5 g	4.45	0.175	0.03	0.001	0.05	0.002
	Fy = 6.0 g	1.78	0.070	11.66	0.459	0.20	0.008
	Fz = 2.0 g	0.13	0.005	0.03	0.001	4.22	0.166
	Roll $\dot{\theta}_y = 0.6$ rad/s	3.15	0.124	1.60	0.063	0.08	0.003
	Yaw $\dot{\psi}_z = 1.42$ rad/s	1.52	0.060	0.36	0.014	8.38	0.330
	Torque Mx = (4000 lb-in.)	0.03	0.001	0.15	0.006	1.27	0.050
2	Fx	4.50	0.177	0.03	0.001	0.08	0.003
	Fy	1.68	0.066	11.43	0.450	0.03	0.001
	Fz	0.03	0.001	0.05	0.002	4.42	0.174
	Roll	2.95	0.116	1.27	0.050	0.38	0.015
	Yaw	0.33	0.013	0.61	0.024	9.80	0.386
	Torque	0.03	0.001	0.13	0.005	1.37	0.054
3	Fx	4.50	0.177	0.05	0.002	0.13	0.005
	Fy	1.37	0.054	8.86	0.349	0.33	0.013
	Fz	0.05	0.002	0.08	0.003	2.03	0.080
	Roll	2.41	0.095	3.30	0.130	0.38	0.015
	Yaw	0.53	0.021	1.12	0.044	19.56	0.770
	Torque	0.03	0.001	0.05	0.002	0.84	0.033
4	Fx	4.45	0.175	0.05	0.002	0.15	0.006
	Fy	1.12	0.044	8.86	0.349	0.23	0.009
	Fz	0.18	0.007	0.08	0.003	1.75	0.069
	Roll	1.98	0.078	3.30	0.130	0.66	0.026
	Yaw	2.21	0.087	1.12	0.044	22.25	0.876
	Torque	0.05	0.002	0.05	0.002	1.07	0.042

4.7.1.8.7 Support Reactions

The support reactions are based on the limit load condition. Gyroscopic effects of the flywheel due to vehicle motions have been included. The limit load reactions are tabulated in Table 4.7.1-4.

4.7.1.9 Crashworthiness Analysis

The vehicle structure was designed to meet the current Federal Motor Vehicle Safety Standards. Applicable standards, related to structures, are tabulated below.

- FMVSS 204, Steering Column Rearward Displacement
- FMVSS 208, Occupant Protection
- FMVSS 214, Side Door Strength
- FMVSS 215, Exterior Protection
- FMVSS 216, Roof Crush Resistance
- FMVSS 301, Fuel System Integrity

A computer-simulation study was performed to provide design guidance in occupant and structural crashworthiness for the electric vehicle. The study determined load paths and stiffness capable of absorbing the energy of a 48 km/h (30-mph), flat-barrier, front impact. Side impact and rear impact also were investigated. The initial curb weight of 1082 kg (2386 lb) was used in the study.

A lumped-parameter, nonlinear-spring, planar, computer code was developed to obtain the structural dynamic response of a modeled representation of the vehicle. The mathematical idealization of the front, side, and rear-impact models is illustrated in Figures 4.7.1-27 through 4.7.1-29. The batteries are located along the vehicle centerline, and are decoupled from the vehicle structure longitudinally. Motion is limited to the planar attitude only. Springs have nonlinear characteristics, including reversal, hysteresis, and viscous damping. Damping is assumed to be zero in the simulations. Based on the classical rigid-body-dynamics theory, the equations of motion are generated internally in the computer program through input descriptions. The program then integrates the equations of motion by an Adams-Moulton, variable-step, corrector-predictor routine.

Program outputs are specified through input descriptions and contain the following:

- Displacement time histories
- Velocity time histories
- Acceleration time histories
- Spring force time histories
- Spring deformation time histories
- Energy balance time histories

TABLE 4.7.1-4

LIMIT LOAD REACTIONS

Mount Number	Load Direction and Magnitude	Reactions					
		X		Y		Z	
		N	lb	N	lb	N	lb
1	Fx = 2.5 g	1793	403	9	2	22	5
	Fy = 6.0 g	721	162	4702	1057	85	19
	Fz = 2.0 g	49	11	9	2	1695	381
	Roll $\theta_y = 0.6$ rad/sec	1263	284	645	145	27	6
	Yaw $\dot{\psi}_z = 1.42$ rad/sec	614	138	142	32	3376	759
	Torque Mx = 452 N·m (4000 lb-in.)	9	2	62	14	512	115
2	Fx	1815	408	4	1	27	6
	Fy	676	152	4608	1036	13	3
	Fz	9	2	22	5	1775	399
	Roll	1188	267	507	114	151	34
	Yaw	133	30	249	56	3946	887
	Torque	13	3	49	11	556	125
3	Fx	787	177	9	2	22	5
	Fy	240	54	1548	348	58	13
	Fz	9	2	13	3	356	80
	Roll	423	95	578	130	67	15
	Yaw	93	21	196	44	3425	770
	Torque	4	1	9	2	142	32
4	Fx	778	175	9	2	27	6
	Fy	196	44	1548	348	40	9
	Fz	31	7	13	3	307	69
	Roll	347	78	578	130	111	25
	Yaw	387	87	196	44	3897	876
	Torque	9	2	9	2	187	42

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Figure 4.7.1-27. Frontal Impact Model

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Figure 4.7.1-28. Lateral Impact Model

Figure 4.7.1-29. Rear Impact Model

4.7.1.10 Parametric Studies and Summary Results

The mass distribution of each model was sized and located to best reflect the vehicle geometry and location of major substructures, and to provide simulation output at desired specific locations within the model. The numerical values for the front- and side-impact models are given in Table 4.7.1-5, while those of the rear-impact model are given in Table 4.7.1-6. The load paths within the model are represented by the spring locations and directions. The baseline load-deflection characteristics are estimated values, based on past experience. The characteristics were then scaled ± 25 percent to obtain the "stiff" and the "soft" cases to complete one series of cases. Based on the results, certain load-deflection characteristics were then further modified for the next series of cases to improve the vehicle crashworthiness. A total of eight series, consisting of more than 60 cases, was investigated. The results are summarized in Table 4.7.1-7.

The stiffness requirements for the 600 series are similar to those proposed by the Budd Company. Further modifications of the load-deflection characteristics do not significantly improve the crashworthiness, as seen by results of the 700 series. Even though the drive accelerations of 45.2/53.0 g are on the high side, they are considered acceptable because of lack of exact load-deflection characteristics of fiberglass vehicle components.

TABLE 4.7.1-5

WEIGHT DISTRIBUTION FOR FRONT AND LATERAL IMPACT MODELS

Mass	Component	Component Weight		Total Mass per Car	
		kg	lb	kg	lb
M ₁ , M ₃	Wheel	4.99	11		
	Tire	11.34	25		
	Suspension	13.61	30		
	Brake	11.34	25		
	Frame and body	22.68	50	127.92	282
M ₂	Frame and body	13.61	30		
	Steering	13.61	30	27.22	60
M ₄ , M ₈	Door	53.30	117.5		
	Frame and body	11.34	25	129.28	285
M ₅ , M ₆	Seat (body)	11.34	25		
	Restraint	1.36	3		
	Driver/pass.	68.04	150	161.48	356
M ₇ , M ₉	Frame and body	47.63	105		
	9 Batteries	235.87	520	567.00	1250
M ₁₀	Wheels	9.98	22		
	Tires	22.68	50		
	Brakes	22.68	50		
	Suspension	27.22	60		
	Frame and body	53.87	118.75		
	Propulsion	160.35	353.5	296.77	654.25
Total (includes 136 kg (300 lb) of occupants)				1309.66	2887.25

TABLE 4.7.1-6

WEIGHT DISTRIBUTION FOR REAR IMPACT MODELS

Mass.	Component	Component Weight		Total Mass per Car	
		kg	lb	kg	lb
M ₁ , M ₃	Wheel	4.99	11		
	Tire	11.34	25		
	Suspension	13.61	30		
	Brake	11.34	25		
	Frame and body	22.68	50	127.92	282
M ₃	Frame and body	13.61	30		
	Steering	13.61	30	27.22	60
M ₄ , M ₈	Door	53.30	117.5		
	Frame and body	11.34	25	129.28	285
M ₅ , M ₆	Seat (body)	11.34	25		
	Restraint	1.36	3		
	Driver/pass.	68.04	150	161.48	356
M ₇ , M ₉	Frame and body	47.63	105	567.00	1250
M ₁₀	Wheels	9.98	22		
	Tires	22.68	50		
	Brakes	22.68	50		
	Suspension	27.22	60		
	Frame and body	35.72	78.75		
	Propulsion	160.35	353.5	278.62	614.25
M ₁₁	Rear frame and body	9.07	20		
	Rear bumper	9.07	20	18.14	40
M ₁₂	Moving barrier			1309.43	2886.75
Total (includes 136 kg (300 lb) of occupants)				1309.66	2887.25

TABLE 4.7.1-7

SUMMARY OF FRONT IMPACT ANALYSIS
FOR 500, 600, AND 700 SERIES CASES

Case	Driver Accel. 1st/2nd Peaks, g	Front Deformation		% Energy Absorbed by Bat- teries (1)	% Energy Absorbed by Front Side Struc- ture(2)	% Energy Absorbed by Front and Rear Side Struc- ture (3)
		mm	in.			
500	66.6/52.3	510.5	20.1	30.0	28.5	41.0
501	66.6/52.4	510.5	20.1	33.0	28.5	41.1
502	68.4/53.7	457.2	18.0	29.1	22.6	40.5
503	68.4/74.4	457.2	18.0	32.0	23.5	41.3
504	69.4/75.7	416.6	16.4	26.9	31.7	57.8
505	69.5/76.9	416.6	16.4	30.2	33.6	58.4
600	45.2/53.0	655.3	25.8	33.1	32.6	39.8
601	60.2/56.1	574.0	22.6	32.5	36.0	51.3
602	50.6/64.9	683.3	26.9	32.6	37.1	51.3
700	44.9/52.2	657.9	25.9	31.7	33.0	39.5
701	45.0/52.7	657.9	25.9	32.9	32.8	39.7
702	44.9/52.2	657.9	25.9	31.7	33.0	39.5

Notes: (1) Represented by K_3 and K_6 - % of initial total kinetic energy.

(2) Represented by K_1 and K_5 - % of initial total kinetic energy.

(3) Represented by K_1 , K_5 , K_9 , and K_{12} - % of initial total kinetic energy.

4.7.2 Vehicle Dynamics Analysis

4.7.2.1 Handling Analysis

A unique feature of the ETV-2 is the flywheel, mounted at the rear-right of the car, with its spinning axis parallel to the axle, and rotating in the opposite direction. This configuration was selected because:

- (a) Vehicle pitch occurs more frequently, and generally with higher rate, than vehicle roll and yaw. Thus, the gyro-moment produced by pitch could be largest. To eliminate this moment, the flywheel axis must be parallel to the pitch axis.
- (b) After the flywheel axis is selected, the flywheel rotation can be either the same as or opposite to axle rotation. Vehicle yaw rate is generally higher than roll rate, such as in accident-avoidance maneuvers. It was decided to rotate the flywheel in the opposite direction to the axle, so that gyro-moment might offset the centrifugal force to some extent.

Figure 4.7.2-1 shows a simplified vehicle model, subjected to the gyro-moment and the centrifugal force. The moment produced by the flywheel is given by

$$M_g = I_f \omega \Omega$$

where M_g = gyro-moment, $N \cdot m$

I_f = flywheel inertia* = $1.05 N \cdot m \cdot s^2$ ($0.772 \text{ ft-lb-sec}^2$)

ω = flywheel rotating speed at 25 000 rpm = 2618 rad/s

Ω = precession (yaw) rate in rad/sec

Substitution yields

$$M_g = 2749 \Omega N \cdot m \quad (2021 \Omega \text{ ft-lb})$$

The moment produced by the centrifugal force is given by

$$M_c = \frac{W}{g} R \Omega^2 H$$

where M_c = moment caused by centrifugal force in $N \cdot m$

W = sprung weight = $12\,010 \text{ N}$ (2700 lb)

R = turning radius, m

H = distance between the c.g. and rolling center = 0.40 m (1.3 ft)

Ω = yaw rate in rad/sec

Substitution yields

$$M_c = 490 R \Omega^2$$

*Includes generator and gear inertia.

Figure 4.7.2-1. Vehicle Model on a Curve

Figure 4.7.2-2 shows the stabilizing effect of the gyroscopic moment as a function of vehicle speed and rate of turning. Also shown for comparison are values of the rolling moments without the gyro effect; i.e., the moments encountered in the conventional vehicle without the flywheel stabilizing effect. The data show that the tendency of the vehicle to roll or lean during all speeds and all possible turning rates is significantly reduced, particularly at the lower speeds. At freeway speeds of 80 km/h (50 mph), the rolling tendency is reduced by 25 percent and at 32 km/h (20 mph) the tendency is reduced by over 60 percent. At still lower speeds, the rollover tendency is almost completely eliminated, even at turning rates at the limit of tire adhesion.

Figure 4.7.2-2. Gyro Stabilizing Effect

The effect of the flywheel-gyro forces on vehicle-handling response was investigated. Steady-state and transient yaw responses are used as measures of the vehicle lateral-directional response. The yaw response requirements, based on the Research Safety Vehicle Specification*, specify that in response to a steering input for which the lateral acceleration is 0.4 ± 0.02 g, the vehicle must:

- (a) Maintain a steady-state yaw response at forward velocities of 11.2 m/s (25 mph), 22.3 m/s (50 mph), and 31.3 m/s (70 mph) within the envelope defined by Figure 4.7.2-3.
- (b) Within this envelope, the specific vehicle-response curve must be concave downward at all points. Similar response curves, obtained with different lateral accelerations, will exhibit the same characteristic shapes. In relation to the 0.4-g yaw-response curve on Figure 4.7.2-3 coordinates, as lateral acceleration increases from 0.4 g, the yaw-response curve will progressively move downward; as lateral acceleration decreases from 0.4-g, the yaw-response curve will progressively move upward.
- (c) With a steering input applied at a rate no less than 500 deg/s, and held constant at a value that produces 0.4-g steady-state, lateral acceleration with test velocities of 40 and 113 km/h (25 and 70 mph), the transient yaw response will be as described for Figure 4.7.2-4. The initial time is the time at which one-half the steering-wheel input is complete. In this figure, the upper curve is the upper limit for a test speed of 113 km/h (70 mph), while the lower curve is a lower limit for a test speed of 40 km/h (25 mph), and does not apply to the 113 km/h test.

Figure 4.7.2-3. Steady-State Yaw Response Versus Tangential Velocity

*Final Handling Analysis Report, Dynamic Science, Inc. Report No. 4927-78-4, January 1978.

Figure 4.7.2-4. Transient Yaw Response Versus Time

4.7.2.1.1 Vehicle Model

A vehicle handling model for the ETV-2 was established. Figure 4.7.2-5 presents a schematic representation of the mathematical model.

The model has the following characteristics:

- (a) 3 degrees of freedom for the sprung mass (roll, yaw, and sideslip)
- (b) The flywheel gyro forces are applied as D'Alembert Forces on the vehicle body

Certain assumptions are inherent in this model. These include:

- (a) Constant vehicle speed during a maneuver
- (b) No significant pitching of the vehicle
- (c) Weight shift is proportional to lateral acceleration
- (d) Small angles
- (e) Rigid spring mass

Figure 4.7.2-5. Analytical Representation of Vehicle

A Dynamic Science Inc. computer program was modified to include the effect of the flywheel gyro forces.

Table 4.7.2-1 presents the data input parameters and the values used for the NTEV. The values for I_E (flywheel inertia) and ω_E (flywheel angular velocity) are input values for the spinning gyro. These two parameters are set equal to zero for the case of a vehicle without a flywheel.

4.7.2.1.2 Handling Analysis Conditions

Conditions used to evaluate the effect of the flywheel gyro forces are derived from the Research Safety Vehicle Specification for yaw response. Two criteria were used for evaluation:

The predicted yaw responses should be within the boundaries of Figures 4.7.2-3 and 4.7.2-4.

The response parameter values for the NTEV, with and without a rotating flywheel, would be compared to evaluate the effect of the flywheel.

In essence, the transient-yaw-response condition used the vehicle's steering gain (yaw velocity divided by front-wheel angle) to evaluate vehicle turning performance. The steering gain represents the ratio of how much the vehicle turns to how much steering input is required. The larger the steering gain, the tighter the vehicle turns (has more oversteer) for a given steering input.

TABLE 4.7.2-1

DATA INPUT FOR ETV-2 HANDLING ANALYSIS

Variable	Definition	Units*
<u>Card 1:</u>		
$W_f = 1571$	Total weight on front wheels of the vehicle (with 4 occupants)	lb
$W_r = 1885$	Total weight on rear wheels of the vehicle (with 4 occupants)	lb
$W_{uf} = 150$	Front unsprung weight	lb
$W_{ur} = 160$	Rear unsprung weight	lb
$l = 95.0$	Wheelbase	in.
$h_f = 1.04$	Front roll center height above road plane	in.
$h_r = 5.65$	Rear roll center height above road plane	in.
$h_{uf} = 11.30$	Front unsprung weight cg height above road plane	in.
$h_{ur} = 11.30$	Rear unsprung weight cg height above road plane	in.
$h_t = 18.41$	Total vehicle cg height above road plane (with 4 occupants)	in.
$I_{xx}^s = 469.0$	Sprung mass roll inertia about origin on roll axis below cg of total vehicle	lb-sec ² -ft
$I_{zz}^s = 2261.0$	Sprung mass yaw inertia about origin on roll axis below cg of total vehicle	lb-sec ² -ft
$I_{zz}^u = 247.6$	Unsprung mass yaw vehicle about origin on roll axis below cg of total vehicle	lb-sec ² -ft
$I_{xz}^s = 10.59$	Sprung mass product of inertia about origin of roll axis below cg of total vehicle	lb-sec ² -ft
$D_f = 1.91$	Front roll damping coefficient	ft-lb-sec/deg
$D_r = 3.48$	Rear roll damping coefficient	ft-lb-sec/deg
$K_f = 993$	Front roll stiffness	ft-lb/deg
$K_r = 558$	Rear roll stiffness	ft-lb/deg
$\Gamma_{\theta f} = 0.725$	Front roll camber coefficient (+ understeer)	
$\Gamma_{\theta r} = 0.0$	Rear roll camber coefficient (+ understeer)	
$E_{\theta f} = 0.10$	Front roll steer coefficient (+ understeer)	
$E'_{\theta f} = 0.09$	Rear roll steer coefficient (+ understeer)	
<u>Card 2:</u>		
$E_{Nf} = 0.016$	Front aligning torque deflection steer per wheel (+ understeer)	deg/ft-lb
$E_{yf} = 0.0007$	Front lateral force deflection steer per wheel (+ understeer)	deg/lb
$N_f = -0.0002$	Front aligning torque deflection camber per wheel (+ understeer)	deg/ft-lb

*Program input was in the English system of units as shown in this table.

TABLE 4.7.2-1. (Continued)

Variable	Definition	Units
$\Gamma_{Yf} = 0.0022$	Front lateral force deflection camber per wheel (+ understeer)	deg/deg/lb
$\Gamma_{Nr} = 0.002$	Rear aligning torque deflection steer per wheel (+ understeer)	deg/ft-lb
$\Gamma_{Yr} = 0.0003$	Rear lateral force deflection steer per wheel (+ understeer)	deg/lb
$\Gamma_{Nr} = 0.0001$	Rear aligning torque deflection camber per wheel (+ understeer)	deg/ft-lb
$\Gamma_{Yr} = 0.0019$	Rear lateral force deflection camber per wheel (+ understeer)	deg/lb
$C_{ab} = 109.4$	Front tire cornering stiffness for one tire (always positive)	lb/deg
$C_{ar} = 113.6$	Rear tire cornering stiffness for one tire (always positive)	lb/deg
$C_{Yf} = 4.8$	Front tire camber stiffness for one tire (positive)	lb/deg
$C_{Yr} = 5.4$	Rear tire camber stiffness for one tire (positive)	lb/deg
<u>Card 3:</u>		
$N_{zf} = 12.2$	Aligning torque per unit slip angle for one front tire (positive)	ft-lb/deg
$N_{ar} = 13.4$	Aligning torque per unit slip angle for one rear tire (positive)	ft-lb/deg
$N_{Yf} = 0.81$	Aligning torque per unit camber angle for one front tire (positive)	ft-lb/deg
$N_{Yr} = 0.90$	Aligning torque per unit camber angle for one rear tire (positive)	ft-lb/deg
$I_E = 0.7754$	Flywheel inertia	lb-sec ² -ft
$\omega_E = 2618^*$	Flywheel angular velocity	rad/sec
$A_{Yss} = **$	Steady-state lateral acceleration selected by user	ft-sec ²
$x = 51.81$	Distance from front wheel center to lateral acceleration measurement point (positive for points behind wheel center) (with 4 occupants)	in.
$z = 18.45$	Distance from road plane to lateral acceleration measurement point (positive for points above road plane) (with 4 occupants)	in.
$u = ***$	Forward velocity	mph
$\Delta t = .10$	Printing time increment for transient response (not to exceed 0.1 sec)	sec
$t_{max} = 5.0$	Total time interval for transient response	sec

*0.0 for cases with flywheel stationary.

**Steady-state lateral accelerations to be used include 0.2 g (6.44 ft/sec²), and 0.7 g (22.54 ft/sec²).

***Vehicle forward velocities to be used include 25, 40, 50, 60, and 70 mph.

4.7.2.1.3 Handling Analysis Results

The results for the handling analysis are presented in Figures 4.7.2-6 through 4.7.2-8. Figure 4.7.2-6 illustrates that the steady-state yaw response is within the acceptable boundary, and that there is no significant difference in response, with or without the flywheel.

Figure 4.7.2-7 illustrates that the transient-yaw responses at 40 km/h (25 mph) are well above the required lower limit. This figure also shows no significant difference in transient-yaw response, with or without the flywheel.

Figure 4.7.2-8 illustrates that the transient response at 113 km/h (70 mph) is within the required upper boundary, except for a small time period around the 1.6-s point. The response for the spinning-flywheel case is slightly worse than the response for the stationary-flywheel case.

Table 4.7.2-2 presents a comparison of selected response parameters for cases with and without a rotating flywheel. Table 4.7.2-3 provides a glossary of the response parameters. The effects of the spinning flywheel are as follows:

(a) Yaw velocity

The peak yaw velocity is slightly larger and occurs earlier.

The difference between a spinning and stationary flywheel increases with vehicle speed.

The steady-state yaw velocity is independent of flywheel-spinning conditions.

(b) Roll Angle

The peak roll angle is considerably less at the lower vehicle speeds, and slightly less at the higher vehicle speeds.

(c) Sideslip Angle

The peak sideslip angle is slightly less at lower vehicle speeds, but slightly more at higher vehicle speeds.

(d) Steering Input

Less steering input is required to achieve the same lateral-acceleration turn.

In summary, the effect of the spinning flywheel is:

A slightly quicker response to the turn

A slightly increased overshoot of the turn

A slightly increased oversteer response

An improved roll response during the turn

Figure 4.7.2-6. Electric Vehicle Steady-State Condition at 0.4 g

Figure 4.7.2-7. Electric Vehicle Transient Yaw, 40 km/h (25 mph) at 0.4 g

TABLE 4.7.2-2

COMPARISON OF SELECTED HANDLING RESPONSE PARAMETERS AT 0.4 g

Flywheel status	40 km/h (25 mph)		64 km/h (40 mph)		80 km/h (50 mph)		97 km/h (60 mph)		113 km/h (70 mph)	
	Spinning	Stationary	Spinning	Stationary	Spinning	Stationary	Spinning	Stationary	Spinning	Stationary
Yaw velocity, deg/sec	20.152	20.128	12.688	12.617	10.365	10.210	8.906	8.696	7.947	7.690
Sideslip angle, deg	-1.547	-1.582	-2.765	-2.783	-3.069	-3.071	-3.263	-3.244	-3.415	-3.370
Roll angle, deg	-0.701	-1.177	-0.846	-1.146	-0.910	-1.147	-0.959	-1.152	-1.003	-1.161
Lateral acceleration m (ft/sec)	(12.930)	(12.913)	(12.922)	(12.890)	(12.979)	(12.927)	(13.136)	(13.003)	(13.337)	(13.125)
	3.948	3.936	3.939	3.929	3.956	3.940	4.004	3.963	4.065	4.001
Yaw velocity response time, s	0.521	0.553	0.677	0.695	0.693	0.716	0.672	0.702	0.635	0.671
Lateral acceleration response time, s	0.850	0.857	1.075	1.123	1.217	1.266	1.275	1.337	1.307	1.380
Sideslip response time, s	0.948	0.966	1.202	1.240	1.270	1.321	1.313	1.380	1.337	1.416
Steady-state yaw velocity, deg/s	20.126	20.126	12.579	12.579	10.063	10.063	8.386	8.386	7.186	7.186
Steady-state sideslip, deg	-1.553	-1.576	-2.752	-2.779	-3.035	-3.057	-3.189	-3.207	-3.281	-2.297
Reference steer angle, deg	4.729	4.758	2.092	2.110	1.484	1.499	1.155	1.167	0.951	0.967
Roll gain, deg/g	-1.619	-2.823	-2.071	-2.823	-2.221	-2.823	-2.321	-2.823	-2.393	-2.823
Sideslip gain, deg/g	-3.829	-3.936	-6.875	-6.942	-7.503	-7.636	-7.968	-8.013	-8.202	-8.240
Control sensitivity, m/s ² /deg (ft/sec ² /deg)	(2.724)	(2.707)	(6.157)	(6.104)	(8.676)	(8.593)	(11.152)	(11.037)	(13.466)	(13.322)

TABLE 4.7.2-3

GLOSSARY OF RESPONSE PARAMETER TERMS

Terms	Definitions
Yaw velocity	The angular velocity about the z-axis (deg/sec)
Sideslip angle	The angle between the traces on the x-y plane of the vehicle x-axis and the vehicle velocity vector at some specified point in the vehicle (deg)
Roll angle	The angle between the vehicle y-axis and the ground plane (deg)
Lateral acceleration	The component of the vector acceleration of a point in the vehicle perpendicular to the vehicle x-axis and parallel to the road plane (m/s^2) (ft/sec^2)
Yaw velocity response time	The time required for the vehicle's yaw velocity to reach at least 90 percent of the steady-state yaw velocity
Lateral acceleration response time	The time required for the vehicle's lateral acceleration to reach at least 90 percent of the steady-state lateral acceleration (sec)
Sideslip response time	The time required for the vehicle's sideslip angle to reach at least 90 percent of the steady-state sideslip angle (sec)
Steady-state yaw velocity	The final yaw velocity obtained that remains relatively constant over a long period of time (deg/sec)
Steady-state sideslip angle	The final sideslip angle obtained that remains relatively constant over a long period of time (deg)
Reference steer angle	The steer angle required to obtain a given lateral acceleration based on a known velocity and turning radius (deg)
Roll gain	The rate of change in vehicle roll angle with respect to change in steady-state lateral acceleration on a level road at a given trim* and test conditions (deg/g)
Sideslip gain	The rate of change in vehicle sideslip angle with respect to change in steady-state lateral acceleration on a level road at a given trim and test conditions (deg/g)
Control sensitivity	The change in steady-state lateral acceleration on a level road with respect to change in steering wheel angle at a given trim and test conditions ($m/s^2/deg$) ($ft/sec^2/deg$)

*Trim may be defined as the steady-state condition of the vehicle with constant input that is used as the reference point for analysis of dynamic vehicle stability and control characteristics.

Figure 4.7.2-8. Electric Vehicle Transient Yaw, 113 km/h (70 mph) at 0.4 g

4.7.2.2. Ride Analysis

The main factors that affect the ride of a vehicle are front suspension, rear suspension, seat suspension, weight distribution, structural stiffness, and tire characteristics (Figure 4.7.2-9). In the following discussion, the term "suspension system" will be used in a broad sense to include all these factors.

In designing the suspension system, the objective was to minimize the acceleration levels transmitted to the car floor and to the driver and passengers. Acceleration levels primarily indicate the ride comfort of the vehicle. These criteria are given in Figure 4.7.2-10, and are recommended by the International Standards Organization. Because of the range limitation of the ETV-2 the 2.5-hr curve, shown in Figure 4.7.2-10, will be used for the design goal.

The motion of a vehicle in a path is treated as a system that consists of mass points interconnected by springs and dashpots. Since the large number of degrees of freedom makes the computer simulation complicated and time-consuming, the vehicle model generally adopted by analysts is to treat the car-body weight as having six degrees of freedom. Also, the bending mode of the car may be significant, because of the heavy battery load. For a majority of highway-travel conditions, a vehicle is traveling on a relatively flat and straight road at a constant forward speed. The excitation of the road profile causes vertical motion of the axles, and consequently, pitch and bounce of the vehicle. Assume that the waviness is equal in the left and right tracks, and that the vehicle and rotational modes are not coupled to the lifting and pitching oscillations. A ride-analysis model can be constructed as shown in Figure 4.7.2-9, showing the individual components that are important in vibration investigations.

The car is treated as a flexible member. Each wheel with a pneumatic tire is modeled as an unsprung mass, plus a spring and dashpot. A suspension system, with springs and shock absorbers, connects the wheels and the spring mass. It should be noted here that the spring rates and the damping characteristics can be linear or nonlinear, such as friction and hysteresis. The model has a total of six degrees of freedom. The various road excitations can be thought of as inputs, or forcing functions, to the system.

The BMW 320i suspension was selected for the ETV-2. Typical bounce frequencies for the 320i system are 1.0 to 1.5 Hz in the front, and 1.3 to 1.8 Hz in the rear.

Figure 4.7.2-9. Vehicle Model With Vertical Plane of Symmetry

Figure 4.7.2-10. Vertical Ride Quality, ISO Proposed Recommendation

4.7.3 Body and Chassis

Design and construction details of the ETV-2 were based on the following guidelines:

- Requirement to meet applicable Federal Motor Vehicle Safety Standards, with particular emphasis on crashworthiness requirements because of the use of an all-plastic (fiber-reinforced plastic) for the unibody concept.
- Achievement of low aerodynamic drag and weight that are important to performance.
- Product acceptance from a styling viewpoint.

4.7.3.1 Exterior Design

The four-passenger ETV-2 styling is simple, clean, and attractive, and provides good aerodynamic characteristics as shown in Figure 4.7.3-1. The body styling reflects a lightweight-vehicle appearance. The wraparound glass, thin roof section, and character lines high on the body sides and ends enhance this effect. The tucked-under rocker panels contribute to the removal of "visual weight" from the area low between the wheels, while small flares have been incorporated to prevent mud and stones from damaging the body finish.

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 outer mold line (OML) panels--soft nose, soft tail, hood, left door, right door, and left and right rear quarter panels.

Lightweight, deformable-bumper, fascia panels have been incorporated at the vehicle nose and tail. 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.

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 low headlamp profile and location well aft of the bumper permit a steeply raked hood and low nose for improved aerodynamics and vision. Bumper corners are well rounded to improve aerodynamic performance and to maximize pedestrian safety. The tail light and the backup/license-plate light system are positioned well forward of the rear impact area, and away from the corners of the vehicle.

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Figure 4.7.3-1. Simple, Clean Appearance of the ETV-2

4.7.3.2 Interior Design

The width of the car is strongly influenced by the battery tunnel, which extends through the center of the vehicle. The ample body width provides additional room between passengers and creates a general feeling of roominess. The ETV-2 interior is presented in Figure 4.7.3-2. Every attempt has been made to have the ETV-2 appear and operate similar to a conventional internal combustion engine (ICE) car. Controls and operation are identical to those for an ICE car equipped with an automatic transmission. There is no clutch, and a shift selector lever governs the operating modes. Instruments are conventional except for one gage indicating flywheel energy and another indicating remaining battery charge. Ample package storage is provided in the passenger dash area and on top of the battery tunnel console.

Maximum use is made of available parts. The steering column, brake and throttle pedals, and hand brake are from the BMW 320i. Front seats are Stylex Grande Luxe Recliners, which are available aftermarket seats made in England. These seats are lightweight and comfortable. Window riser mechanisms are from the VW Rabbit and are efficient and lightweight.

A standard passenger restraint system combines the three-point, continuous-loop system from the 1977 Mustang in the front with an inertia locking retractor that is buried in the B post. A standard Type 1 lap belt is available in the rear seat.

4.7.3.3 Glazing

Glass always has been the preferred glazing material for automobiles. It is hard, abrasion resistant, has good optical qualities, and is very stiff. It is also very heavy, 2451 kg/m^3 (0.089 lb/cu in.). Since American car manufacturers have become weight conscious, plastics are being seriously considered for glazing. New coatings have improved abrasion resistance, optical qualities are good, and their density is half that of glass at 1190 kg/m^3 (0.043 lb/cu in.). The potential weight savings makes plastic glazing very attractive for future electric vehicles. Currently plastic glazing does not meet all FMVSS requirements; however, it is expected that plastic glazing and the requirements will be compatible in the near future.

In the ETV-2, coated acrylics are used for all glazing except the windshield, which is conventional laminated safety glass. Characteristics of the plastic ETV-2 glazing are presented in Table 4.7.3-1. Stretched acrylic, because of its superior ductility, is suitable for use in the roll-up windows and the back light, while the as-cast material is sufficient for the fixed rear quarter-windows. Though the acrylic must be thicker than its glass counterpart because of its low bending modulus, plastic glazing offers a 9.7-kg (21.5-lb) weight savings.

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Figure 4.7.3-2. ETV-2 Interior

TABLE 4.7.3-1

CHARACTERISTICS OF PLASTIC GLAZING

Glazing	Material	Acrylic				Glass			
		Thickness,		Weight,		Weight		Thickness	
		mm	in.	kg	lb	kg	lb	mm	in.
Driver's window	Stretched acrylic	3.17	0.125	1.70	3.75	3.3	7.3	3.0	0.118
Passenger window	Stretched acrylic	3.17	0.125	1.70	3.75	3.3	7.3	3.0	0.118
Rear light	Stretched acrylic	6.35	0.250	7.18	15.84	11.2	24.7	3.0	0.118
Right quarter-window	As-cast acrylic	3.17	0.125	1.35	2.97	2.6	5.8	3.0	0.118
Left quarter-window	As-cast acrylic	3.17	0.125	2.35	2.97	2.6	5.8	3.0	0.118
		Totals		13.28	29.28	23.0	50.9		

4.7.3.4 Structure Concept

The ETV-2 body and structure are primarily fiber-reinforced plastic (FRP). The only metal parts in the body (except for the running gear) are metal inserts for fastenings, reinforcements, and glazing. The latter is also plastic (acrylic) except for a conventional safety glass windshield. The major body structure elements are shown in Figure 4.7.3-3. The basic floor assembly components form subassembly 1 as shown in Part A of the figure. Additional bulkheads, energy tubes, and dash structure are added to form the complete floor assembly, subassembly 2. Inner and outer side rails and roof components complete the assembly. Elements are joined in special assembly fixtures and are either bonded with epoxy adhesives, or riveted, or both. An assembly tree is shown in Figure 4.7.3-4. The three major subassemblies are the floor, the right-hand side panel, and the left-hand side panel. A total of 174 separate parts makes up the body assembly.

More detailed assembly drawings of the body are shown in Figures 4.7.3-5 through 4.7.3-10. The part numbers shown are the last three digits of the AiResearch six-digit part number. The first three digits are 574 in all cases.

Figure 4.7.3-3. Body Structures

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Figure 4.7.3-4. Body Overall Breakdown

Figure 4.7.3-5. Roof, Hood, and Tunnel Details

Figure 4.7.3-6. Tunnel and Energy Absorbers

Figure 4.7.3-7. Side Panel Details

Figure 4.7.3-8. Front End, Dash, and Rear End Details

S-41717

Figure 4.7.3-9. Bumper and Fender Details

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Figure 4.7.3-10. Door Assembly

The battery tunnel is a primary structural member; it carries most of the normal running and operating loads and the front and rear suspension loads. The passenger compartment provides additional structural rigidity to meet roll-over requirements and to maintain integrity under crash conditions.

The batteries are mounted in a double row running the length of the battery tunnel. This locates all the batteries in a common pack, simplifies wiring, simplifies servicing, and provides desirable weight distribution. In addition, the approach offers desirable crash-energy management characteristics.

Each pair of batteries is strapped to a continuous battery support tray that is attached to the structure from the underside with six bolts. These connections will restrain the pack in the tunnel under loads of up to about 10 g. Under major front-end-impact conditions, the connections will fail and allow the batteries to move relative to the tunnel and absorb their own kinetic energy. The upper portion of the tunnel is integral with the floor structure to provide a continuous barrier to exclude battery acids and fumes from the passenger compartment. The bottom section is bonded into position and stiffened to carry the battery loads back into the side walls.

Figure 4.7.3-11 shows a cross-section of the ETV-2 body through its center-line. The flexible foam bumpers will absorb the energy of an 8 km/h (5 mph) collision and rebound undamaged. In a 16 km/h (10 mph) collision, front or rear, the fiberglass backup beam will absorb the impact and may be damaged, but the remaining structure will be totally intact. In a 48-km/h (30-mph) collision, the structure will crush and absorb energy in a manner that limits passenger g-loading to safe levels.

Figure 4.7.3-11. Body Structure, Centerline Cross Section

A section through the front axle is shown in Figure 4.7.3-12. The energy management front-crush elements are located on both sides of the battery tunnel. A metal front cross-member picks up the lower control arms and is in turn bolted to the battery tunnel.

The rear structure is shown in Figure 4.7.3-13. Rear crash loads travel forward on the rear sills from the rear bumper, around the wheel wells, and into the side sills. The rear sills also contain energy absorbing elements.

A side view of the rear sill is shown in Figure 4.7.3-14. The forward part of the rear sill is spoon shaped and bonded onto the side sill. The power unit is suspended on four Barry-500 series motor mounts. These are all-angle isolators utilizing elastomers in compression. Power unit installation is accomplished by raising the unit with a special fixture and installing four motor-mount bolts. The power unit and the installation fixture are shown in Figure 4.7.3-15.

The door concept is shown in Figure 4.7.3-16. A fiberglass corrugated-type beam runs the length of the door to absorb side impact loads. Deflection tests conducted on this beam verify its stiffness and strength.

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Figure 4.7.3-12. Body Structure, Front Axle Section

Figure 4.7.3.13. Body Structure, Rear, Top View

Figure 4.7.3-14. Body Structure, Rear, Side View

Figure 4.7.3-15. Power Unit Installation

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Figure 4.7.3-16. Door Construction

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The hood concept is shown in Figure 4.7.3-17. It is a rear-opening, two-part construction with conventional hinges and latches.

Figure 4.7.3-17. Hood Construction

The rear hatch concept is shown in Figure 4.7.3-18. The hatch is a formed piece of 6.4-mm (0.25-in.) stretched acrylic. The hinges and latches are bolted directly to the acrylic.

Figure 4.7.3-18. Hatch Construction

4.7.3.5 Crash Management

The design of the crash-energy-management system for the electric vehicle meets the overall crushing resistances required of the vehicle structure. The weight of the electric vehicle, with four occupants, is approximately 1780 kg, (3923 lb) including 484 kg (1066 lb) of batteries.

The vehicle design requires the batteries to decouple from the vehicle structure during a crash situation, so that the vehicle structure will not be forced to absorb the kinetic energy of the batteries. This concept was demonstrated in battery crash testing performed by The Budd Company.

Drop-tower tests, conducted by The Budd Company on its 48-km/h (30-mph) test facility, indicate that the major structural elements used in the vehicle will provide adequate crush resistance. The test results on the battery tunnel indicate an average crush force of 44.5 kN (10,000 lb); the two energy-absorbing elements (foam-filled tubes) yield an average crush force of 129 kN (29,000 lb). The combined crush resistance is 173 kN (39,000 lb). Based on the 173-kN figure, the crush distance would be approximately 0.50 m (1.64 ft).

Since these crush distances are for an ideal square-wave energy absorber, additional distance should be provided to take into account the actual performance of the front-end structure. Based on past experience, approximately 0.9 m (3 ft) of crush distance will be provided in the front-end design.

The crash-energy data are summarized in Table 4.7.3-2.

TABLE 4.7.3-2
CRASH ENERGY MANAGEMENT

Vehicle kinetic energy at 48 km/h (30 mph) less batteries and with two passengers, J	86,000 J	63,400 ft-lb
Average force for 0.6-m (2-ft) crush, kN	141 kN	31,700 lb
Average force for structural elements, kN		
Tunnel	44.5 kN	10,000 lb
Absorbers	<u>129.0</u>	<u>29,000</u>
	173.5 kN	39,000 lb

4.7.3.6 Sectional Properties

Figure 4.7.3-19 presents a preliminary study of the sectional properties of the vehicle, as conceptually designed. Shown for comparison purposes on the figure are the properties for a metal vehicle structure of the same size.

The torsional rigidity of the design is shown in Table 4.7.3-3 and is compared with values for other vehicle designs.

Section	ETV-2			BUDD-SWIFT			EI_x		EI_y	
	Area, sq in.	I_x , in. ⁴	I_y , in. ⁴	Area, sq in.	I_x , in. ⁴	I_y , in. ⁴	NTEV lb-in. ² x10 ⁶	Swift lb-in. ² x10 ⁶	NTEV lb-in. ² x10 ⁶	Swift lb-in. ² x10 ⁶
A-A	2.17	9.69	10.48	0.46	1.24	1.09	14.54	37.20	15.75	32.70
B-B	2.07	1.25	17.11	0.28	0.90	0.20	1.88	27.00	25.67	6.00
C-C	1.97	13.18	1.35	0.36	0.57	0.29	19.77	17.10	2.03	8.70
D-D	0.97	0.94	0.74	0.47	0.77	0.73	1.41	23.10	1.11	21.90
E-E	0.99	0.13	1.65	0.19	0.06	0.07	0.20	1.80	2.48	2.10
F-F	0.95	0.84	0.74	0.34	0.15	0.16	1.26	4.50	1.11	4.80
G-G	1.13	1.48	1.04	0.28	0.13	0.06	2.22	3.90	1.56	1.80
H-H	2.21	6.97	1.85	0.62	1.03	1.01	10.46	30.90	17.78	30.30
Tunnel	6.47	178.81	305.38	0	0	0	268.22	0	458.07	0

Section	ETV-2			BUDD-SWIFT			EI_x		EI_y	
	Area, cm ²	I_x , cm ⁴	I_y , cm ⁴	Area, cm ²	I_x , cm ⁴	I_y , cm ⁴	NTEV N·cm ² x10 ⁶	Swift N·cm ² x10 ⁶	NTEV N·cm ² x10 ⁶	Swift N·cm ² x10 ⁶
A-A	14.00	403	436	2.97	52	45	417	1068	451	938
B-B	13.35	52	712	1.81	37	8	54	775	737	172
C-C	12.71	549	56	2.32	24	12	567	491	58	250
D-D	6.26	39	31	3.03	32	30	40	663	32	628
E-E	6.39	5	69	1.23	2	3	6	52	71	60
F-F	6.13	35	31	2.19	6	7	36	129	32	138
G-G	7.29	62	43	1.81	5	2	64	112	45	52
H-H	14.26	290	493	4.00	43	42	300	887	510	870
Tunnel	41.74	7443	12711	0	0	0	7697	0	13146	0

Figure 4.7.3-19. Comparison of Section Properties

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TABLE 4.7.3-3

COMPARISON OF TORSIONAL BODY STIFFNESS

Vehicle	Torsional Stiffness,	
	$\frac{\text{N}\cdot\text{m}}{\text{degree}}$	$\frac{\text{lb}\cdot\text{ft}}{\text{degree}}$
ETV-2	12,300	9,100
Chevrolet plastic-foam body - XP-898	4,390	3,240
Budd Company-designed Swift vehicle	7,600	5,600
British Leyland vehicle	8,720	6,430
Fiat 128	11,800	8,700

4.7.3.7 Materials

Primary body materials are predominantly fiber-reinforced polyester (FRP) with metal inserts in high-stress areas such as suspension attachment points. Most of the body structure uses glass-mat-reinforced polyester with glass content of 35 percent (referred to as Class 1). Some high-stress areas use glass-cloth-reinforced polyester with glass content from 50 to 65 percent (referred to as Class 2).

Material properties (except the fatigue strengths) presented in Table 4.7.3-4 are test results from three or more flat-test specimens, machined from test slabs 2.3 to 3.2 mm thick (0.090 to 0.125 in.). All values presented are those at room temperature.

Secondary materials are nonstructural in nature: glazing materials, polyurethane, foam, rubber, adhesive materials, etc. Suspension components are BMW 320i parts, made of high-strength steel. Table 4.7.3-5 lists materials uses.

TABLE 4.7.3-4

FRP MATERIAL PROPERTIES

Property	Units	SI Unit Values		Test Method	Units	English Unit Values	
		Class 1	Class 2			Class 1	Class 2
Tensile strength, min.	MPa	103	230	ASTM-D-638	ksi	15	34
Tensile modulus, min.	MPa	9,700	15,000	ASTM-D-638	msi	1.4	2.2
Flexural strength, min.	MPa	160	200	ASTM-D-790	ksi	23	30
Flexural modulus, min.	MPa	8,300	6,900	ASTM-D-790	msi	1.2	1.0
Compressive strength, min.	MPa	150	200	ASTM-D-695	ksi	22	30
Fatigue flux (10 cycles)	MPa	26	42	--	ksi	3.8	6.1
Impact strength, min.	N/m	880	1,000	ASTM-D-256	lb/in.	5	6
Specific gravity, min.	--	1.35	1.5	ASTM-D-792	--	--	--
Flammability, min.	--	Pass	Pass	FMVSS 571.302	--	--	--

TABLE 4.7.3-5

NTEV MATERIALS USES

Material	Use
Glass mat/reinforced polyester: 1.5 mm (0.060 in.) thick 2.3 mm (0.090 in.) thick	Chassis Nonstructural Structural
Glass cloth/reinforced polyester	High-stress areas
Glazing: Laminated glass, EZ3-eye tinted, 6.4 mm (0.25 in.) thick Abrasion-resistant acrylic, EZ-eye tinted 3.2 mm (0.125 in.) thick 3.2 mm (0.125 in.) thick 6.4 mm (0.25 in.) thick	Windshield Door windows Rear quarter windows Rear window
Polyurethane foam: 2 PCF rigid Flexible 6 PCF, flexible	Energy absorption Cushions Bumpers
Vinyl, cloth backing	Upholstery
Neoprene foam, butyl rubber	Door and window seals
Epoxies, urethanes	Adhesives
Steel	Suspension components

4.7.4 Suspension, Steering, and Tires

The design for suspension and steering components adapts existing components to the ETV-2. Following a study of candidate vehicles, the total system from the BMW 320i was selected. The 320i is about the same size as the ETV-2, and its four-wheel independent suspension fits the packaging requirements dictated by the central battery tunnel.

A comparison of ETV-2 and 320i physical characteristics is shown in Table 4.7.4-1. From the suspension standpoint, the only change is the addition of extensions on the steering links to accommodate the 137-mm (5.4-in.) greater track width of the ETV-2. Although the ETV-2 is heavier than the 320i, it has about the same weight on the front wheels, which ensures reasonable steering effort without power steering.

TABLE 4.7.4-1

NTEV/320i PHYSICAL CHARACTERISTICS

	ETV-2	BMW 320i
Curb weight, kg (lb)	1507 (3323)	1182 (2605)
Gross weight, kg (lb)	1779 (3923)	1454 (3205)
Weight distribution front/rear, percent (no passengers)	47/53	54/46
Wheelbase, m (in.)	2.41 (95)	2.56 (100.9)
Track, front/rear, m (in.)	1.52/1.50 (60/59)	1.39/1.40 (54.6/55.1)
Length, m (in.)	4.19 (165)	4.51 (177.5)
Width, m (in.)	1.68 (66)	1.61 (63.4)
Curb height, m (in.)	1.39 (54.5)	1.38 (54.3)

4.7.4.1 Suspension and Steering

Characteristics of the ETV-2 suspension and steering system are presented in Table 4.7.4-2. Views of the selected system are presented in Figures 4.7.4-1 through 4.7.4-3.

TABLE 4.7.4-2

ETV-2 SUSPENSION AND STEERING DATA

Front Suspension

Upward wheel travel	84 mm
Downward wheel travel	109 mm
King-pin inclination	10 deg, 54 min

Steering

Maximum wheel lock, inside wheel	40.4 deg
Maximum wheel lock, outside wheel	35.3 deg
Overall steering ratio	21.1:1
Turns lock-to-lock	4.1

Rear Suspension

Upward wheel travel	102 mm
Downward wheel travel	122 mm

Figure 4.7.4-1. Front Suspension Underside View Showing Steering

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Figure 4.7.4-2. Rear Suspension Underside View

Figure 4.7.4-3. Front Suspension

The ETV-2 uses lightweight wheels and tires with low rolling resistance. The three-piece spun aluminum wheels weigh only 4.8 kg (10.6 lb). They are made in France by GOTTI and are completely street-compatible. The wheel rims were modified to obtain the negative 16-mm (0.625-in.) offset required for the BMW suspension.

The tires selected for the ETV-2 are steel-belted radials made from special compounds that will permit higher operating pressures while retaining an acceptable ride. The size selected is the P175/70R13, which has a load rating of 500 kg (1100 lb) at the manufacturer's recommended pressure of 241 kPa (35 psi). Maximum ETV-2 tire load is 474 kg (1046 lb) on the rear wheels.

4.7.5 Brakes

The ETV-2 has both electrical regenerative and hydraulic braking systems that are completely separate. Potentiometer pickups on the throttle and brake pedals communicate with the controller. Signals from the throttle pedal apply regenerative braking to simulate the compression braking experienced in the conventional ICE auto. Regenerative braking is normally accomplished by transferring the vehicle's kinetic energy directly into the flywheel. At the flywheel saturation speed of 24 000 rpm, the excess current directly charges the battery.

As shown in Figure 4.7.5-1, the hydraulic brakes are separate. Direct mechanical linkage from the brake pedal operates the master cylinder, which controls separate fluid paths to front and rear brakes. The parking brake operates mechanically on the rear wheels.

The front disc brakes require relatively high actuation force, and the regenerative braking requires artificial feel; both increase pedal pressure. The brake pedal arrangement and braking mode vs-pedal travel are shown in Figure 4.7.5-2. A special spring applies the artificial feel during the regenerative mode; then, due to its overcenter anchor point, it relaxes to avoid building up pedal force. Clearance is provided between the brake pedal rod and the master cylinder to lock out the hydraulics during regenerative braking.

The braking pedal mechanism offers three distinct regions of brake control, as shown in Figure 4.7.5-3. The first 25.4 mm (1.0 in.) of pedal travel controls the regenerative braking torque from an initial value of 203 N·m to a maximum regeneration torque of 542 N·m (150 to 400 lb-ft). The driver can "feel" the transition region from 25.4 mm to 46 mm (1.9 in.) of steady regenerative braking followed by the combined hydraulic and regenerative braking. The combined braking mode is limited to 89 mm (3.5 in.) of pedal full movement.

The regenerative braking characteristics are also tabulated in Table 4.7.5-1.

ETV-2 braking torque/pedal force characteristics are shown in Figure 4.7.5-3. All requirements of FMVSS 571.105 are met. Emergency braking is accomplished with 516 N (116 lb) of pedal force, and the front failure emergency is met with 667 N (150 lb) of pedal force, all within 571.105 requirements.

The most severe failure condition is with the front hydraulics inoperative, which reduces braking torque 67 percent. Because of the regenerative braking on the rear wheels, that failure case is less severe than with the standard 320i, where a front hydraulics failure reduces braking capability by 80 percent.

REGENERATIVE BRAKING LOGIC IS IN SOFTWARE

Figure 4.7.5-1. Brake System

S-46940 -A

Figure 4.7.5-2. Brake Pedal Arrangement

TABLE 4.7.5-1

REGENERATIVE BRAKING CHARACTERISTICS
(from 30 and 45 mph Initial Vehicle Speeds)

Pedal Position, mm (in.)	Stopping Time, s		Braking Torque, N·m (lb-ft)		Deceleration Rate, g	
	48 km/h (30 mph)	72 km/h (45 mph)	48 km/h (30 mph)	72 km/h (45 mph)	48 km/h (30 mph)	72 km/h (45 mph)
0	22	32	203 (150)	203 (150)	0.062	0.064
6.4 (0.25)	14	19	363 (268)	389 (287)	0.098	0.108
25.4 (1.0)	10	14	531 (392)	556 (410)	0.137	0.146

(Note: Data derived from dynamometer tests)

Figure 4.7.5-3. Braking Performance

Figure 4.7.5-4. Braking Characteristics

In Figure 4.7.5-4, regenerative braking is included as a constant $0.54 \text{ kN}\cdot\text{m}$ ($400 \text{ lb}\cdot\text{ft}$) torque. As shown in Figure 4.7.5-5, maximum regenerative braking varies as a function of vehicle velocity. Peak braking is $0.73 \text{ kN}\cdot\text{m}$ at 32 km/h (20 mph) and it is above $0.54 \text{ kN}\cdot\text{m}$ ($400 \text{ lb}\cdot\text{ft}$) to beyond 97 km/h (60 mph). At low speeds, regenerative braking is effective to about 6.4 km/h (4 mph) and drops linearly to zero at standstill. When the flywheel is saturated, and the energy is being put back into the batteries, regenerative braking is reduced by about one-half and is not effective below 40 km/h (25 mph).

Figure 4.7.5-6 shows the potential usefulness of regenerative braking. Various parameters of the Federal Urban Driving Schedule (FUDS) are shown. The lower curve shows the braking duty cycle required for the FUDS; most of the braking can be handled regeneratively.

The 320i hydraulic system functions on the ETV-2 without change. Characteristics of the system result in applying 80 percent of the braking to the front wheels on the 320i. Figure 4.7.5-7 shows the overall braking on the rear wheels as a function of deceleration rate. At very low rates, for regenerative braking only, braking is 100 percent on the rear. When the hydraulics are brought in, more and more braking shifts to the front wheels.

Also shown in Figure 4.7.5-7 is the ETV-2 dynamic weight distribution. At standstill, 54 percent of the ETV-2 weight is on the rear wheels. The effect of deceleration is to dynamically shift more weight to the front. At low decelerations there is more braking than weight on the rear wheels, resulting in a tendency to lock the rear wheels on poor road surfaces. At about 3.4 m/s^2 the brakes and car are matched and all four wheels are equally effective. At higher deceleration rates, which include the FMVSS 571.105 emergency stops, enough braking shifts to the front to lock the front wheels during a skid.

Figure 4.7.5-5. Flywheel Saturation Provision

Figure 4.7.5-6. Brake Characteristics During Federal Urban Driving Cycle

Figure 4.7.5-7. Braking Distribution

Future replacement of the front disc brakes with drum brakes offers significant advantages for electric vehicles; it would (a) reduce weight, (b) reduce pedal force requirement, and (c) reduce brake drag; however, replacing the disc requires considerable design and development, not consistent with the NTEV program objectives. Use of solid brake discs, which are standard on the 1978 320i, reduces the weight of the ETV-2.

4.7.6 Passenger Heating and Cooling

The vehicle heating and ventilation airflow is shown in Figure 4.7.6-1. Fresh air, collected in the inlet plenum at the base of the windshield, can be directed into the passenger compartment or it can be mixed with the passenger compartment recirculated air. The operator can select maximum fresh air mode, maximum recirculated air mode, or a combination of the two by adjusting the recirculation lever. To increase the ventilation rate, the operator can turn on the fan, which is part of the heater assembly. When the heater is turned on, the air pumped by the fan will be reheated. The reheated air is used either to warm the passenger compartment or to defrost the windshield.

The heater is an ESPAR DIL diesel-fueled hot-air heater that has long been in production for European air-cooled as well as water-cooled automobiles. Characteristics of the heater are shown below:

Capacity 6124 kJ/h (5800 Btu/hr)

Dimensions 368 x 127 x 178 mm (14.5 x 5 x 7 in.)

Air delivery 1.13 m³/min (40 cfm)

Heater performance is shown in Figure 4.7.6-2.

Figure 4.7.6-1. Passenger Heating and Ventilation Airflow

Q - 6124 kJ/h (5,800 BTU/hr)
BODY LOSS COEFFICIENT - 2.3 kJ/min/°C
(1.2 BTU/MIN/°F)

A-5983

Figure 4.7.6-2. ESPAR DIL Heater Performance

4.7.7 Driver Controls and Displays

Driver controls and displays, shown in Figures 4.7.7-1 and 4.7.7-2, are very similar to those in conventional vehicles, thus permitting a new driver to operate the ETV-2 with little or no instruction. Controls and displays that require special comment are described in the following paragraphs. All other controls and displays are conventional in appearance and usage.

4.7.7.1 Key Switch

The key switch is comparable to the conventional ignition switch. There are four positions:

ACCES--Battery power is provided to the accessories and the power system. In addition, the battery line contactor will close to provide battery power to the power unit, but the other three main contactors will remain open to prevent the power unit from operating.

OFF--Battery power is disconnected from the power unit and the accessories; however, if the flywheel is coasting down at a speed of 5000 rpm or higher, the pump control will continue to operate the vacuum and lubrication pumps until flywheel speed drops below 5000 rpm.

ON--Battery power is provided to the power system and the accessories. The vehicle will operate in all modes after the power system has been properly started.

START--Battery power is provided to the power system. In addition, the START signal causes the controller to begin functioning. Without the START signal, the controller will not process information from the vehicle or commands from the driver. The starting procedure requires that the switch simply make contact in the START position. When the key is released, a spring returns it to the ON position.

4.7.7.2 Selector Lever

The drive mode selector lever, similar in appearance and usage to an automatic transmission selector in a conventional vehicle, is used to select one of the four operating modes:

P (park)--In the P position (with key ON and after having been turned to START as noted above), the vehicle is in the prestart mode, which is used to charge the flywheel prior to driving the vehicle. In this position the battery supplies power to the generator, which, operating as a motor, drives the flywheel to the desired operating speed. The motor and throttle are electrically disconnected. Prestart is terminated whenever the selector is shifted to another position. There is no mechanical lock on the wheels in P; therefore, the parking brake should be used to lock the wheels whenever the vehicle is parked.

① TURN SIGNAL SWITCH	⑪ HOOD RELEASE HANDLE (UNDER DASH)
② WARNING LIGHTS	⑫ HORN BUTTON
③ BATTERY POWER METER	⑬ BRAKE PEDAL
④ TURN SIGNAL/HAZARD WARNING INDICATOR LIGHT	⑭ THROTTLE PEDAL
⑤ HIGH BEAM INDICATOR LIGHT	⑮ PARKING BRAKE LEVER
⑥ SPEEDOMETER	⑯ HEATER AND VENTILATION CONTROL
⑦ FLYWHEEL POWER METER	⑰ DRIVE MODE SELECTOR LEVER
⑧ WARNING LIGHTS	⑱ ASH TRAY
⑨ WINDSHIELD WIPER AND WASHER SWITCH	⑲ GLOVE COMPARTMENT
⑩ AIR OUTLETS	

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Figure 4.7.7-1. Instruments and Controls

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Figure 4.7.7-2. Dashboard Instruments

R (reverse)--The vehicle can be driven in reverse in a conventional manner with a maximum speed of 18 km/h (11 mph). The generator is disconnected electrically and the flywheel is coasting. The armature chopper regulates the motor armature current, as commanded by the throttle.

N (neutral)--In the N position, the power unit is energized, but power will not be applied to the motor or the generator. The vehicle is electrically free-wheeling; no electrical regenerative braking is applied to the rear wheels and the flywheel is coasting. The neutral position is the safest for start and shut-down sequences because no power is applied to the wheels.

D (drive)--In the D position, the vehicle can be driven forward in a conventional manner. When the throttle is depressed, the vehicle will accelerate and cruise at a rate depending on throttle position. Regenerative braking will be automatically applied when the throttle is closed and also when the brake pedal is depressed.

4.7.7.3 Instruments and Warning Lights

The following instruments and warning lights are provided.

Speedometer/Odometer--This is an electrically driven instrument using conventional format.

Flywheel Power Meter--This meter informs the driver, in broad terms, of the vehicle acceleration capability. The meter indicates flywheel speed and is driven by the controller. It is divided into three zones: (1) red indicates flywheel speed is too low to be used for acceleration (0 to 10 000 rpm); (2) yellow indicates moderate acceleration capability (flywheel speed of 10 000 to 18 000 rpm); and (3) green indicates design-level acceleration capability (flywheel speed of 18 000 to 24 000 rpm).

Battery Power Meter--The battery power meter indicates: (1) the battery voltage while charging, and (2) the battery capacity in percent when the vehicle is operating (key ON and charger disconnected).

When the battery voltage drops to 90 V, the indicated capacity is momentarily 0 percent. Immediately following, the indication changes to 100 percent because the controller automatically shuts down the system when the battery charge is too low and there is no load on the battery pack.

Brake Fluid Warning Light--The BRAKE FLUID light will glow following a loss of hydraulic fluid pressure.

Temperature Warning Light--The TEMP light functions in case of a motor, generator, or power conversion unit overtemperature. It will also glow when the transmission lubricating oil pressure drops below 3.0 psig indicating a potential transmission oil overheat condition.

Pressure Warning Light--The flywheel operates in a sealed cavity in which the pressure is reduced to lessen windage losses. The PRESS light functions when the flywheel forepump inlet pressure is greater than 3.5 torr; however,

a delay is built into the circuit so that after start, the forepump has 5 min to reduce pressure to less than 3.5 torr before the light comes on. If this light glows during the flywheel startup but before 5 min have elapsed, the flywheel should not be charged. If, after the first 5 min of the startup, the light comes ON, indicating the forepump pressure is too high, the generator will be disconnected automatically, and flywheel charging will stop until the forepump pressure is reduced to 3.5 torr. If the generator is automatically disconnected, the vehicle must be restarted before flywheel charging will continue. Typically, this condition would not occur unless the flywheel cavity were repressurized immediately before driving due to repair or the vehicle had not been operational for more than one week so that the normal leakage rate caused cavity repressurization.

Battery Fail Light--The BATT FAIL light indicates when the voltage of any 6-V battery drops below 4.5 ± 0.1 V.

4.6.5 Switch Gear

The battery-line contactor and the chopper contactor are part of the main power loop and are required to carry 200 A for several minutes, a current which is treated as a continuous-rating requirement. The selected contactors have a capability of interrupting fault currents in excess of 1200 A. These contactors have blowouts and arc shutes, and are rated at 72 Vdc at 0.62 A, closing and holding, at 20 percent of rated voltage.

The motor/generator contactors are of the same configuration with a continuous-rating of 228 A. Current may flow in either direction. When interrupting rated currents, arcing will be contained within the shutes.

The reverser switch carries a nominal 15-A, and makes and breaks at zero A. In the interest of reliability and safety, the unit has load-braking capability, with magnetic blowouts, but with no arc shutes. Its coil requires 2 W, closing and holding. Its auxiliary contacts are SPDT, and operate the backup lights.

The start-run relay is at the heart of the on-off control for the vehicle, supplying coil power to the battery contactor. This relay is a Siemens unit, weighing 12 ounces, and carrying 12 V, 7 W closing and holding. The range-meter relay is SPDT at relatively low current levels, with coil power of 2.2 W, and operates from the battery contactor at 108 Vdc.

All fuses of the power circuits are 250 V, rated to ensure clearing of transients. Current sensors in the main power loop will accept fault-current transients of 2000 A, recovering without loss of accuracy in microseconds.

4.7.8 Noise Characteristics

The significant noise characteristic exhibited by the vehicle is that its interior noise level is relatively constant whether the vehicle is stationary or moving. The major noise sources are the generator, transmission geartrain, and the flywheel. Superimposed on the broadband noise from these sources is the pure tone flywheel noise of the frequency of one per revolution. The interior noise level was measured at the driver's ear level while the vehicle was driven on the dynamometer. Table 4.7.8-1 shows the measured noise for different vehicle and flywheel speeds.

TABLE 4.7.8-1

ETV-2 MEASURED NOISE LEVEL DATA

Vehicle speed, km/h	Interior Noise Level (On Dynamometer), dBa				Exterior Noise Level (3 m away while vehicle moving), dBa
	0	48	48	72	
Flywheel speed, rpm	6 000	18 000	20 300	20 000	20 000
Noise level, dBa	65	72	73	78	76

The peak noise level of 78 dBa was measured at a vehicle speed of 72 km/h and a flywheel speed of 20 000 rpm. The maximum predicted stationary noise level at a flywheel speed of 24 000 rpm is 71 dB in preferred speech interference level (PSIL).

When the vehicle is moving, the noise sources are the transmission geartrain, motor, tires, and aerodynamic drag noise. This noise is essentially a broadband type. A measurement was made 10 ft away from the power unit while the vehicle was moving at 20 mph with the flywheel rotating at 20 000 rpm. The exterior noise level was 76 dBa. The power unit noise could not be perceived. It is estimated based on this test that the power unit will not be a significant exterior noise source at 89 km/h.

Available ICE noise data (Table 4.7.8-2) are limited to data taken on a freeway at 55 mph.

TABLE 4.7.8-2
ICE NOISE LEVEL DATA

Vehicle	Noise Level, dBa		Remarks
	Window Closed	Window Open	
CDA*	77	81.5	--
VW Bug	73	73.5	With bad window Seal - 1969
VW Bug	71	--	1977 model
Cordoba	64	68.5	A/C on, 1977
Granada	63	--	1977 model

*Copper Development Association Electric Vehicle

4.8 WEIGHT

The measured curb weight of the ETV-2 is 1507 kg (3323 lb). A detailed weight breakdown is presented in Table 4.8-1.

TABLE 4.8-1

ETV-2 WEIGHT

<u>Body Assembly</u>		<u>Wt, kg</u>	<u>Power unit (Con t)</u>		<u>Wt, kg</u>
Body		301.	Transmission and housing assembly		41.0
Operating hardware		15.9	Gears	5.1	
Lamps, switches, instrument panel control	4.1		Bulkhead	2.2	
Exterior ornaments	7.7		Housing	10.6	
Trim panels	29.9		Carrier and differential	4.8	
Seats	36.7		Housing differential	3.2	
Seals, weather strip, insulation	5.4		Chain and sprockets	7.7	
Glazing	34.0		Lubrication pump, filter tubes, adapter	2.9	
Convenience items	7.7		Nuts, bolts, shafts, bearings, oil	4.5	
Interior molding and ornaments	0.5		Motor/generators (2)	54.3 ea	108.6
Instrument panel and console	2.3		Armature laminations	6.9	
Paint	5.4		Armature winding	4.0	
Suspension		121.9	Commutator	6.2	
Front suspension	39.7		Shaft and bearings	3.0	
Rear suspension	25.9		Main field and pole laminations	5.2	
Tires	37.2		Interpole laminations	2.4	
Wheels	19.1		Housing field frame	10.5	
Axles and constant-velocity joints		11.8	Pole face winding	2.0	
Brakes		49.3	Main field wiring	3.1	
Front brakes	24.3		Interpole winding	1.8	
Rear brakes	19.2		Insulation	1.2	
Master cylinder, lines	5.9		Brushes and holders	1.2	
Steering		15.0	End bells (2)	5.2	
Chassis indirect materials		1.4	Miscellaneous hardware	1.6	
Heater		3.4	Fore pump		10.0
Accessory equipment		12.7	Mounting brackets and shims		1.4
Total		666.1	Solenoid valve, delay timer, vacuum tubing		2.3
<u>Electronics and Controls</u>		<u>Wt, kg</u>	Total		241.8
Controller		5.4	<u>Weight Summary</u>		
Power conversion unit		32.4		<u>Weight, kg</u>	<u>lb</u>
Charger		13.1	Body assembly	666.1	1468.5
Contactors, reverser and sensors		15.2	Shell, structural, interior components, suspension, steering, brakes		
Electrical wires, cables, and clamps		2.5	Power unit	241.8	533.1
Total		68.6	Electronics and controls	68.6	151.2
<u>Power Unit</u>		<u>Wt, kg</u>	Battery pack	483.5	1065.9
Flywheel and housing assembly		78.5	Batteries and tray		
Rotor	27.4		Miscellaneous components	47.3	104.3
Rims (2)	12.5		Cooling fan		
Hub and shaft	14.9		Air ducts		
Outer cover	9.3		Supports, cables, clamps		
Inboard housing	14.9				
Stator (molecular pump)	2.7		Total curb weight	1507.3	3323
Containment ring	20.1		Total with 4 passengers	1779.5	3923
Nuts, bolts, bearings, seals, oil	4.1				
			Data recorder	1.3	2.8
			Recorder battery	3.7	8.1

4.9 PRODUCIBILITY

The producibility study originally planned for Phase II of the program was not conducted. This task was deleted from the contract when it was determined that it would be premature to freeze the design at this early stage of development. The technical feasibility and the many advantages have been demonstrated during this phase, but the economic feasibility must be determined based on further development.

5. VEHICLE OPERATION

5.1 DRIVING PROCEDURES

From the driver's point of view, the operation of the ETV-2 is very similar to that of an internal combustion engine (ICE) vehicle with an automatic transmission. The driver selects the desired mode of operation, either drive, neutral, reverse, or park, using the drive mode selector lever as in the ICE vehicle. The rates of acceleration and braking are controlled by the throttle and brake in a similar manner. The vehicle regenerative braking capability is used to simulate the feel of compression braking when both the brake and throttle are off.

The principal differences between an ICE vehicle and the ETV-2 are in battery charging and vehicle startup. To recharge the batteries, the ETV-2 key switch is turned off, the selector is placed in the P (park) position, and the parking brake is applied before plugging the cord for the onboard charger into a 30-A, 115-Vac power source. The significant difference in vehicle starting is flywheel charging. To begin vehicle operation (prestart procedure) the flywheel should first be brought up to a minimum speed. A typical sequence of events for a driver leaving his garage in the morning is as follows:

- (a) Unplug charger from garage wall receptacle if vehicle is being recharged.
- (b) Verify that parking brake is set and drive mode selector is in P (park) position.
- (c) Turn key ON and check dashboard warning lights.
- (d) Check battery power meter on dash to determine battery state of charge. When batteries are fully charged meter reading will be close to 100 percent.
- (e) Turn key momentarily to START; then release key, the flywheel charging cycle will begin.

CAUTION: During prestart procedure or after starting when the selector is in P (park) and the flywheel is being charged, the brakes must be applied to prevent the vehicle from moving backward. This is due to the reactive torque transmitted through the planetary geartrain in the transmission. While the vehicle is at a stop with the selector in driving mode D, the brakes also should be applied until ready to drive on, as in conventional automatic transmission ICE vehicles.

- (f) Check warning lights on dash during flywheel charging. If TEMP light is illuminated, system should be shut down and cause of light determined. If PRESS light is illuminated, system should be shut down and cause of light determined.
- (g) Wait until flywheel is charged to proper speed (approximately 20 000 rpm) as indicated by flywheel power meter (tachometer) on dash. This normally will take 4 min.
- (h) Shift selector to N (neutral) and release parking brake. If there are no warning lights illuminated, vehicle is now ready to be driven. Driving procedures are essentially the same as for a conventional, automatic-transmission, ICE vehicle. The D and R selector lever positions are used in the usual manner. N is used whenever electrical free-wheeling is desired, such as for pushing or coasting without electrical regenerative braking.

If the driver desires to obtain the maximum range of the vehicles, the sequence of events should be modified at step (g). The flywheel should be charged to full speed (approximately 24 000 rpm) instead of only 20 000 rpm. After the flywheel is charged to full speed using battery power, the charger should be connected again to recharge the batteries to their fully charged state while maintaining the flywheel at its full speed. This startup procedure is applicable to the experimental ETV-2; however, a future vehicle would incorporate a 220-Vac charger and other system modifications to achieve shorter charge times for the battery and flywheel as well as a preprogrammed startup procedure that would result in a fully charged flywheel and battery at the beginning of a drive.

The driver should occasionally note the flywheel power meter on the dash. If flywheel speed becomes too low (red zone), the vehicle will not have the normal acceleration capability. If desired the vehicle could be stopped and the selector placed in P (park). This would recharge the flywheel.

The battery power meter on the dash (equivalent to a fuel gage) shows the state of battery charge. When the battery voltage drops below 91 V under a load of 60 A (typically, 30-mph cruise condition on a level road), the batteries are considered to be fully discharged, and the system will automatically shut down. The normal urban driving duration for the batteries to drop from 95 V to 90 V is estimated to be 7 min.

Whenever the vehicle is parked and the key turned off, the flywheel will continue rotating as it coasts down. The coast-down period from 20 000 to zero rpm is approximately 1 hr. The vehicle may be left unattended during this period. The controller monitors flywheel speed and causes the lubrication and vacuum pumps to operate until flywheel speed is below 5000 rpm.

Whenever possible, complete discharge of the battery should be avoided to extend battery life. However, frequent "topping-off" of the battery is not necessary and may be detrimental.

5.2 OPERATING CONDITIONS

Operating conditions include nonoperation, prestart and idle, forward driving, and reverse. The conditions are implemented in the power system by ten basic automatic control modes as shown in Table 5-1.

TABLE 5-1
CONTROLLER SET DEFINITION

Condition	Mode(s)	Notes*
Nonoperation	Mode 0	
Prestart and idle	Mode 1	$N_G < N_B$
	Mode 6	$N_G > N_B$
Driving	Mode 2	$N_M > N_B$ $N_G < N_B$
	Mode 3	$N_M < N_B$ $N_G > N_B$
	Mode 4	$N_M > N_B$ $N_G > N_B$
	Mode 5	$N_M < N_B$ $N_G < N_B$
	FWLO (2)	$N_M > N_B$
	FWLO (5)	$N_M < N_B$
Reverse	Mode 7	$N_M < N_B$

* N_M = Motor speed

N_G = Generator speed

N_B = Base speed: the motor base speed is characterized by the maximum speed the motor can achieve at full field. It is the normal-armature-voltage, full-field speed of the motor.

5.2.1 Nonoperation

Whenever the key is in the OFF position or the driving mode selector is set in the N (neutral) position with the key at the ON position, the vehicle is deenergized in the nonoperational condition with both motor and generator off line. If the vehicle is moving, it will coast in a freewheeling mode. If the vehicle is parked, it is ready for the battery charging process as described.

With the vehicle deenergized (key in OFF position), the driving mode selector in the P (park) position, and the hand brake applied, the operator may at any time connect the vehicle battery charger to residential 115-Vac, 60-Hz power. When the vehicle charger is connected to residential power, the charging current is controlled by the charger current-control circuits. A dashboard voltmeter gives an approximate indication of battery charge. The charger should be disconnected before the vehicle is started to avoid operation of the generator in the prestart mode, i.e., charging the flywheel, at the charging voltage, which could be as high as 151 V. (The system also provides a separate electrical circuit, permitting vehicle battery charging by use of a 220-Vac, 60-Hz charger separate from the vehicle. This provides a faster battery charge rate.)

With the charger connected to residential power, the controller monitors the key, charger, and selector positions to inhibit any drive-mode command by the driver. The inhibit is removed when the driver disconnects the battery charger from the residential power source.

CAUTION: When the charger on the ETV-2 is connected to a residential ac source, the vehicle ground is electrically "hot" with respect to the residential ground. A ground fault interrupter is included in the charger as a protection against a short circuit to residential ground. However, as a matter of proper safety procedure, it is recommended that a person not touch vehicle metal parts while in contact with an electrical ground such as a metal pipe, water on the floor, etc.

5.2.2 Startup

A prerequisite for the most effective operation of the ETV-2 is charging the flywheel to a minimum speed, which can be achieved by the prestart procedure or in the idle mode during driving.

5.2.2.1 Prestart

With the selector in P (park) or N (neutral), when the key is turned ON, the battery contactor closes, providing electrical power to the controller central processor. The 12-V, key-switch supply is activated, closing the battery-contactor relay, which applies 108-V battery power to the battery-contactor solenoid.

Turning the key momentarily to START energizes the system. With the selector in the P (park) position, the flywheel is accelerated to operational speed, using the generator as a motor, powered by the battery, while the motor

and wheels are held stationary (brake ON). During normal operation, before starting the first trip of the day, the flywheel is allowed to reach operating speed using power provided by the battery, a cycle requiring approximately 5 min.

During daily use, which includes parking events of short duration, the vehicle can be operated without the need for additional flywheel startup cycles. If the flywheel is at 85 to 92 percent of maximum speed when the vehicle is parked and the driver resumes operation in less than one hour, the flywheel could still contain sufficient energy to provide acceleration power for the initial vehicle acceleration. The flywheel is then accelerated back to operational speed during the subsequent 5 to 10 min of driving, using battery power.

5.2.2.2 Idle

During vehicle idle with selector in D (drive) and no throttle input, the flywheel also is recharged by battery power like it is in the prestart procedure. However, the charging rate of the flywheel is governed by the operational battery schedule (OBS) in this mode, instead of by the prestart battery schedule (PBS).

5.2.3 Drive

After the battery is recharged and the flywheel is charged to the proper speed, the vehicle is ready for driving. With the selector in the D (drive) position the vehicle enters the drive mode when the controller senses that a throttle input is greater than 5 percent or vehicle speed is greater than 3 km/hr (2 mph). The driver controls the vehicle speed by the throttle and brake.

In normal driving with proper flywheel speed and energy, both the motor and the generator are in operation. The controller monitors the power flow as described in Section 4.3.1 to satisfy the power demand specified by the throttle or brake and to maintain a load-leveled battery discharge (operational battery schedule (OBS)).

When the maximum power is requested by a full-throttle command, all of the power is extracted from the flywheel--none from the battery. This occurs only for a full-throttle command.

Flywheel prestart is a recommended procedure for normal vehicle operation, but is not a prerequisite. If desired, the driver can bypass the flywheel prestart procedure and enter the driving condition directly with the understanding that maximum power will be limited to that available from the motor alone, with no flywheel assistance.

Normal driving, including full-throttle acceleration, is accomplished by four automatic control modes (modes 2, 3, 4, and 5) in the controller; the flywheel-low driving condition is implemented by the FWLO(2) and FWLD(5) modes. The modes are described in para. 5.3.

5.2.4 Reverse

The vehicle enters reverse when the selector is put in the R (reverse) position with the key ON. The generator is switched off line with the flywheel freewheeling. The throttle position commands the amount of the reverse power. Vehicle speed is limited to less than 31 km/h (19 mph), because the control armature current level is achieved by modulating the armature chopper, which is effective only when the motor is below base speed. No regenerative braking is available in this condition.

5.3 AUTOMATIC CONTROL MODES

The four operating conditions are implemented by ten automatic control modes as illustrated in Figure 5-1 and as described in the following paragraphs.

The operational state of the power system is determined by the vehicle and the flywheel energy level. In combination with the drive mode selector, throttle, and brake commands, this operational state effectively defines the speed relationship between the motor and generator, and thus the mode of control required. The logic for controlling intermode transitions is shown in Figure 5-2.

A simplified system schematic (Figure 5-3) shows the configuration of the battery, generator and switch (RS), motor and switch (CS), battery chopper, chopper bypass switch (CBPS), planetary differential transmission, and flywheel. The switch designations used in the following paragraphs refer to this figure.

5.3.1 Mode 0--Nonoperation

In mode 0, all control switches are open; neither motor nor generator are operating; the chopper is OFF.

5.3.2 Mode 1--Prestart and Idle

In mode 1, the motor must be nonrotational by braking and is switched out of the circuit (control loop) (switch CS opens). The generator is operated as a motor with switch RS closed to charge the flywheel. Since the generator (operating as a motor) is below base speed in mode 1, full field current is demanded by the controller. The opening of switch CBPS allows the armature chopper to control the armature current as required by the prestart battery schedule (PBS) or by the operational battery schedule (OBS).

When the operator designates the prestart mode by turning the key to the START position momentarily while the selector is in the P (park) position, the armature current will be controlled by the PBS. The operator also may designate this mode by braking the vehicle to a full stop and sending no throttle input while the selector is in D (drive) position; the armature current will then be controlled by the OBS. This vehicle idle situation makes up a fundamental segment of the load-leveling cycle.

MODE 0
 TYPE: NONOPERATING
 SPEEDS: ANY
 BATTERY SCHED: NONE
 CONTROL: NONE

MODE 1
 PRESTART, IDLE
 $N_G < N_B$ $N_G < N_B$
 N_M ANY N_M ANY
 PBS OBS
 CHOPPER CHOPPER

MODE 2
 DRIVE
 $N_G < N_B$ $N_G < N_B$
 $N_M > N_B$ $N_M > N_B$
 OBS CHOPPER, IFM

MODE 3
 DRIVE
 $N_G > N_B$ $N_G > N_B$
 $N_M < N_B$ $N_M < N_B$
 OBS CHOPPER, IFG

FWLO 5
 DRIVE
 $N_H < N_B$ $N_H < N_B$
 N_G ANY N_G ANY
 OBS CHOPPER

MODE 4
 TYPE: DRIVE
 SPEEDS: $N_G > N_B$ $N_G > N_B$
 $N_M > N_B$ $N_M > N_B$
 BATTERY SCHED: OBS
 CONTROL: IFG, IFM

MODE 5
 DRIVE
 $N_G < N_B$ $N_G < N_B$
 $N_M < N_B$ $N_M < N_B$
 OBS CHOPPER, IFB OR IFM

MODE 6
 PRESTART, IDLE
 $N_G > N_B$ $N_G > N_B$
 N_M ANY N_M ANY
 PBS OBS
 IFG IFG

MODE 7
 REVERSE
 N_G ANY N_G ANY
 $N_M < N_B$ $N_M < N_B$
 RBS CHOPPER

FWLO 2
 DRIVE
 $N_H > N_B$ $N_H > N_B$
 N_G ANY N_G ANY
 OBS CHOPPER
 IFM

G = GENERATOR
 M = MOTOR

OBS = OPERATING BATTERY SCHEDULE
 PBS = PRESTART BATTERY SCHEDULE
 RBS = REVERSE BATTERY SCHEDULE

N_G = GENERATOR SPEED
 N_H = MOTOR SPEED
 N_B = BASE SPEED
 IFG = FIELD CURRENT, GENERATOR
 IFM = FIELD CURRENT, MOTOR

Figure 5-1. Controller Mode Definition

S-40794-A

T_{ij} IS A TRANSITION PATH CONNECTING TWO OPERATING MODES. ELEMENTS i REPRESENT THE PARAMETERS THAT CAUSE THE TRANSITION AND j DEFINES THE SUBSEQUENT SET OF THE CONTROL SYSTEM.

- UNDER FAULT CONDITION, EACH OPERATING MODE TRANSITIONS TO MODE 0 (NONOPERATING MODE). WHENEVER OPERATOR DESIRES A SET CHANGE, THE SYSTEM GOES TO DESIRED SET THROUGH MODE TRANSITION. FOR SAKE OF BREVITY, MODE 0 TRANSITION FOR SET CHANGE AND FAULT ARE NOT SHOWN.

SET DEFINITION

- ```

0 NONOPERATING MODE, COASTING
1 PRESTART, IDLE
2 DRIVE
3 REVERSE

```

## MODE CONFIGURATIONS

S-40777-A

Figure 5-2. ETV-2 Mode Transition Diagram

Figure 5-3. Power System Simplified Switching Diagram

#### 5.3.3 Mode 6--Prestart and Idle

In mode 6, the nonrotational motor is switched out of the circuit (switch CS opens), and the generator is operated as a motor with switch RS closed to charge the flywheel as in mode 1; however, in mode 6 the generator is above base speed and the field weakening of the generator is used to control armature current as required by the PBS or the OBS. Switch CBPS also must be closed and the armature chopper must be turned off.

This mode is designated by the operator in the same manner as for mode 1.

#### 5.3.4 Mode 3--Driving

In mode 3, both motor and generator are operating with switches CS and RS both closed. The motor is below its base speed and its field is set to full flux by the controller. The generator is above its base speed. Field weakening of the generator is used to control the armature current through the motor and generator as required by the throttling or braking command. The opening of switch CBPS allows the armature chopper to control the battery current as required by the OBS.

During normal vehicle maneuvers, while the selector is in the D (drive) position, the vehicle automatically enters mode 3 when its speed is below 31 km/h (19 mph) and while the generator is operating above its base speed. Mode 3 is one of the most common operating modes of the system. The vehicle may be

under acceleration, cruise, or braking, depending on the throttle and brake positions, while in mode 3 or in any other normal driving modes.

#### 5.3.5 Mode 4--Driving

In Mode 4, both motor and generator are operating above their base speeds with switches CS and RS both closed. The armature chopper is bypassed by the closing of switch CBPS. Field weakening of the motor and generator controls armature current level and direction to satisfy the torque required by the throttling or braking commands and simultaneously maintains the proper battery current level as specified by the OBS.

During normal maneuvering, the vehicle automatically enters mode 4 when accelerating above 31 km/h (19 mph) and while the generator is above its base speed. Mode 4 is one of the most common operating modes and also is the most efficient operating mode of the system.

#### 5.3.6 Mode 2--Driving

In mode 2, both motor and generator are operating, with switches CS and RS both closed. The generator is below base speed, and its field is set to full flux by the controller. The motor is above base speed, and field weakening of the motor is used to control the armature current through the motor and generator as required by the throttling or braking commands. The opening of switch CBPS allows the armature chopper to control the battery current as required by the OBS.

During normal vehicle maneuvers, the vehicle automatically enters mode 2 when its speed is above 31 km/h (19 mph) and while the generator drops below its base speed due to the variation of the flywheel speed.

#### 5.3.7 Mode 5--Driving

In mode 5, both motor and generator are operating below their base speeds with switches CS and RS both closed. Whichever has the lower speed is set to full flux by the controller. Field weakening of the other and the modulation of the armature chopper controls the output power of the system and the battery current. The opening of switch CBPS allows the armature chopper to function.

During driving, the vehicle enters mode 5 only when both the motor and the generator speeds drop below their base speeds, an event that rarely occurs.

#### 5.3.8 FWLO Mode 5--Driving

In FWLO mode 5, the generator is switched off line by opening switch RS. Battery power provides the motive torque through the motor with switch CS closed. The motor is below its base speed, and full field current is requested by the controller. Opening of switch CBPS allows the armature chopper to control the armature current as required by the throttle command.

During driving under mode 5, the controller automatically enters the FWLO mode 5 when it detects that the flywheel speed is too low or the generator is far below its base speed.

#### 5.3.9 FWLO Mode 2--Driving

In FWLO mode 2, the motor is on line with switch CS closed, and the generator is off line by the opening of switch RS as in FWLO mode 5. However, the motor is above its base speed, and field weakening is used to control armature current as required by the throttle command. Switch CBPS is closed and the armature chopper is turned off.

During driving under mode 2, the controller enters the FWLO mode 2 when it detects that the flywheel speed is too low or the generator speed is too low.

#### 5.3.10 Mode 7--Reverse

In mode 7, the generator is switched off line by the opening of switch RS. Battery power provides motive torque through the motor with switch CS closed. The reverser in the motor field is activated so that the torque obtained is in the reverse direction. Vehicle speed is limited to less than 31 km/h (19 mph), so that the motor is always operated below its base speed and at full field flux. The armature chopper is used to control the motor armature current, by opening switch CBPS, as required by the demand from the throttle.

The operator activates mode 7 by shifting the selector to R (reverse) and depressing the throttle.



## 6. VEHICLE PERFORMANCE

This section presents the performance of the ETV-2 and describes some of the unusual characteristics and sensitivities that result from the use of the flywheel. The many rotating elements in the ETV-2 power system create additional degrees of freedom and more operational modes to analyse than in the simpler, conventional systems, complicating the description of the performance and the laboratory test verifications.

The flywheel speed, which varies throughout a driving cycle, is influenced by the frequency and magnitude of the accelerations, the number of stops, and the overall average speeds. The Phase II program plan, in recognition of this, originally included several weeks of vehicle test driving under various conditions of city traffic, and an onboard recorder was designed and built to record performance parameters. It was recognized that actual driving in city traffic would be necessary to evaluate all of the advantages of the system. This is particularly true in the subjective evaluation of the increased acceleration power, which cannot be done on the dynamometer. However, all of the driving tests were eliminated from Phase II of the program to permit more extensive power system development. Consequently, it was necessary to analyze ETV-2 performance with the aid of a computer simulation.

The computer simulation consisted of very detailed performance models of each of the system components. The component performance data from the laboratory system test program were used to provide correlation of the simulation program. The results of these tests have been included in the component descriptions of this report. The performance data presented in this section, therefore, represent test-correlated computer predictions.

### 6.1 PREDICTED PERFORMANCE

The tabulation of predicted performance and the specified goals of the NTEV program in Table 6-1 shows that all of the goals were met or exceeded except for the range for the J227a(D) driving cycle and the sustained speed on a 5-percent grade. The primary reason for not meeting the goal for the J227a(D) cycle is that the system was not designed specifically for the J227a(D); instead it was designed to satisfy the much more stringent (but more representative) Federal Urban Driving Schedule (FUDS). In fact, the modest demands of the J227a(D) cycle do not warrant the complexities of the flywheel system.

TABLE 6-1  
ETV-2 PREDICTED PERFORMANCE

|                                            |                  | Final Performance                     |                              |
|--------------------------------------------|------------------|---------------------------------------|------------------------------|
|                                            |                  | Four<br>Passengers<br>(600 lb)        | One<br>Passenger<br>(150 lb) |
| (English Units)                            | DOE<br>Goal      |                                       |                              |
| Range, J227a(D) cycle, mi                  | 75               | 68                                    | 74                           |
| Range, FUDS, mi                            | *                | 66                                    | 72                           |
| Range, 25-mph cruise, mi                   | *                | 117                                   | 127                          |
| Range, 35-mph cruise, mi                   | *                | 105                                   | 114                          |
| Range, 45-mph cruise, mi                   | *                | 77                                    | 84                           |
| 0 to 30 mph acceleration, s (19 000 rpm)** | 9.0              | 8.3                                   | 7.4                          |
| 25 to 55 mph acceleration, s (23 000 rpm)  | 18.0             | 12.9                                  | 11.2                         |
| 0 to 50 mph acceleration, s (21 000 rpm)   | *                | 17.1                                  | 15.2                         |
| Sustained speed on 5-percent grade, mph    | 50 for<br>1 mi   | 50 for<br>0.8 mi<br>balance<br>at 40  | 50 for<br>1 mi               |
| (SI Units)                                 | DOE<br>Goal      | Four<br>Passengers<br>(272 kg)        | One<br>Passenger<br>(68 kg)  |
| Range, J227a(D) cycle, km                  | 121              | 104                                   | 119                          |
| Range, FUDS, km                            | *                | 106                                   | 116                          |
| Range, 25-mph cruise, km                   | *                | 188                                   | 204                          |
| Range, 35-mph cruise, km                   | *                | 169                                   | 183                          |
| Range, 45-mph cruise, km                   | *                | 124                                   | 135                          |
| Sustained speed on 5-percent grade, km/h   | 80 for<br>1.6 km | 80 for<br>1.3 km,<br>balance<br>at 64 | 80 for<br>1.6 km             |

\*Not specified

\*\*Initial flywheel speeds

The J227a(D) driving cycle is a modest test cycle specifically prepared for the low performance expected of battery-powered cars. It is intended as a comparison criterion for vehicles that do not have the acceleration capability of gasoline-powered cars and, therefore, cannot be tested over the more representative FUDS. The FUDS, used for all gasoline-powered cars sold in the United States, specifies the normal speeds and accelerations that occur in city traffic. As such, it reflects more stringent speed and acceleration demands than the J227a(D) driving cycle. The most important aspects of the FUDS which typify the normal flow of city traffic are 11 accelerations per mile, 5.6 of which exceed the J227a(D) acceleration of 1.6 mph/s, an average of 2.0 stops per mile, an average speed of 19.7 mph and accelerations as high as 3.3 mph/s. By contrast, the J227a(D) driving cycle has 1 acceleration and 1 stop per mile, an average speed of 28 mph, and an average acceleration of 1.6 mph/s.

The range goal for the J227a(D) driving cycle might possibly be achieved without the flywheel system; however, the actual range in city traffic will be critically dependent on the acceleration power demanded from the battery and will drop to unacceptably low levels during the normal accelerations in stop-and-go city driving.

With the flywheel system, the range is largely independent of the acceleration demands. As stated in Section 4.2, Power System Design Concept, the performance goals of the more complex flywheel power system should include the full power capability to perform the FUDS. This added performance must, of course, be balanced against the added cost and complexity of the final system.

The maximum acceleration performance shown in Table 6-1 is an indication of the acceleration capability of the new power system. The acceleration performance exceeds the specified goals. This is shown more completely by Figures 6-1 and 6-2. Figure 6-1 shows the usual curve of the time required to accelerate to maximum speed. This curve is, however, misleading when compared with conventional characteristics of gasoline-powered cars because "jackrabbit" starts are used in such tests. The electric car is not capable of such starts although it provides initial, low-speed acceleration that is equal to the normal startup of traffic.

A more meaningful plot (Figure 6-2) shows the acceleration as a function of speed. Here, the acceleration capability can be compared to the requirements of the FUDS. Also shown in the figure are the acceleration characteristics of production gasoline-powered automobiles. As seen on the curves, the ETV-2 has excellent acceleration characteristics, particularly in the mid-to-high-speed range, and will easily achieve the traffic compatibility that was one of the primary objectives of the demonstration program.

The output power capability of the ETV-2 power system is shown in Figure 6-3a, which displays three curves--the two components of power and the combined output. The two components of power are the motor output and the additional mechanical power supplied to the carrier shaft by reason of the decelerating torque of the generator. It can be seen that the maximum output power reaches 65 hp, which accounts for the good mid-range acceleration of the ETV-2. Figure 6-3b shows the torque available at the rear wheels and also the two components of torque that make up the total output. The battery is off for this max. power case.

Figure 6-1. Maximum Acceleration Capability in Terms of Speed and Time

\*J227a(D) acceleration interpreted by JPL for minimum required power.

Figure 6-2. Acceleration Performance Comparisons

a. FULL-THROTTLE POWER OUTPUT

b. FULL-THROTTLE TORQUE OUTPUT

Figure 6-3. ETV-2 Power System Power Capability and Torque Output

Figure 6-4 shows one of the most important characteristics of the power system. This chart shows the combined characteristics of the power unit and the electronic control strategy to balance the energy usage between the flywheel and the battery current. The chart shows three curves: flywheel speed, battery current as dictated by the controller, and vehicle speed throughout the cycle. An interesting feature of the overall control strategy can be seen on the chart between  $t = 200$  and  $t = 300$  s. This is the short freeway driving segment. During this short acceleration and high-speed cruise, the flywheel furnishes the additional power and the controller limits the battery current that would otherwise need to be increased at this speed. The curve of battery current shows only a slight rise; however, if the high speed were sustained, the flywheel speed would continue to decelerate and the battery current would continue to approach the higher steady-state requirements of that speed.

From the data in Figure 6-4, it would appear that the control strategy is very nearly optimum as indicated by the nearly constant battery current throughout the entire cycle. However, other driving conditions also must be accommodated, and although it is believed that the same control strategy will perform well under a wide variety of conditions, it is still subject to further analysis and development, which were beyond the scope of this contract.

Table 6-2, column 1, shows the total energy consumption during operation over the FUDS as depicted in Figure 6-4. Columns 2, 3, 4 and 5 are additional data derived from the first column to show the average energy consumption per kilometer (mile), the average power used, and the individual losses as a percentage of the total. Table 6-3 shows the same type of data for the J227a(D) cycle.

Figure 6-4. ETV-2 Power System Performance During FUDS

TABLE 6-2

ENERGY LOSSES DURING  
FEDERAL URBAN DRIVING SCHEDULE  
(Range: 106 km (66 mi) per battery charge)

| Component               | Energy Loss   |        |        | Power Loss, W | Portion of Total, Percent |
|-------------------------|---------------|--------|--------|---------------|---------------------------|
|                         | W·h/schedule* | W·h/km | W·h/mi |               |                           |
| Planetary geartrain     | 218.74        | 18.2   | 29.2   | 572           | 9.2                       |
| Motor gearset           | 22.35         | 1.9    | 3.0    | 59            | 0.9                       |
| Generator gearset       | 17.99         | 1.5    | 2.4    | 47            | 0.8                       |
| Flywheel                | 164.49        | 13.6   | 21.9   | 429           | 6.9                       |
| Final drive             | 121.25        | 10.1   | 16.2   | 318           | 5.1                       |
| Motor armature          | 345.63        | 28.7   | 46.1   | 904           | 14.5                      |
| Motor field             | 110.53        | 9.1    | 14.7   | 288           | 4.6                       |
| Generator armature      | 379.47        | 31.4   | 50.6   | 992           | 15.9                      |
| Generator field         | 42.51         | 3.5    | 5.7    | 112           | 1.8                       |
| Controller and choppers | 132.08        | 10.9   | 17.6   | 345           | 5.5                       |
| Fan and pump            | 53.36         | 4.4    | 7.1    | 139           | 2.2                       |
| Road load               | 777.23        | 64.4   | 103.6  | 2031          | 32.6                      |
| Total                   | 2385.58       | 197.7  | 318.1  | 6236          | 100.0                     |

\*A complete Federal Urban Driving Schedule is 12.1 km (7.5 mi).

TABLE 6-3

ENERGY LOSSES DURING J227a(D) CYCLE  
(Range: 109 km (68 mi) per battery charge)

| Component               | Energy Loss |        |        | Power Loss, W | Portion of Total, Percent |
|-------------------------|-------------|--------|--------|---------------|---------------------------|
|                         | W·h/cycle   | W·h/km | W·h/mi |               |                           |
| Planetary geartrain     | 18.89       | 12.1   | 19.5   | 525           | 7.1                       |
| Motor gearset           | 2.73        | 1.7    | 2.8    | 79            | 1.0                       |
| Generator gearset       | 1.63        | 1.1    | 1.7    | 47            | 0.6                       |
| Flywheel                | 12.83       | 8.2    | 13.2   | 370           | 4.8                       |
| Final Drive             | 11.07       | 7.1    | 11.4   | 320           | 4.1                       |
| Motor armature          | 37.01       | 23.8   | 38.2   | 1068          | 13.9                      |
| Motor field             | 6.26        | 4.0    | 6.5    | 181           | 2.4                       |
| Generator armature      | 33.37       | 21.5   | 34.4   | 963           | 12.5                      |
| Generator field         | 7.99        | 5.1    | 8.2    | 231           | 3.0                       |
| Controller and choppers | 11.11       | 7.1    | 11.5   | 321           | 4.2                       |
| Fan and pump            | 4.78        | 3.0    | 4.9    | 138           | 1.8                       |
| Road load               | 119.02      | 76.2   | 122.7  | 3436          | 44.6                      |
| Total                   | 266.69      | 170.9  | 275.0  | 7679          | 100.0                     |

The overall efficiency (propulsion) is higher during the J227a(D) cycle because of the higher average speed. The average speed of the J227a(D) cycle is 28 mph, but it is 19.6 mph for the FUDS. This is characteristic of the flywheel system; since many of the losses are a strong function of time, the slower speeds will have higher losses (per mile) and the higher speeds will have lower losses per mile. However, the difference in efficiency is almost totally compensated by the variation in the battery capacity as a function of discharge rate. At the slower speeds the increased battery capacity at the lower average discharge rate compensates for the decreased efficiency, as shown by the range, which is nearly the same in each case. This also is consistent with the characteristic that the vehicle range is largely independent of the acceleration or the type of driving cycle.

## 6.2 RANGE SENSITIVITY

Throughout the program, performance and performance sensitivities were analyzed to guide design priorities and decisions. The range sensitivities presented in Figure 6-5 were analyzed on the basis of the J227a(D) cycle, except as noted; however, the results are also essentially applicable to FUDS.

Figures 6-5 and 6-6 show the effect of the more important design variables plotted as functions of vehicle range over the driving cycle. These parametric sensitivities show where emphasis should be placed when evaluating the system concept or when considering subsequent future developments. These data are particularly meaningful now that a baseline system and actual component test data exist.

To demonstrate the usefulness of these data, it can be seen that reducing the vehicle aerodynamic drag effective area by 10 percent would increase vehicle range by 4.8 km (3.0 mi), and reducing vehicle weight by 10 percent would increase range by 5.3 km (3.3 mi), resulting in a total increase in range of 10.1 km (6.3 mi). Improvements greater than 10 percent are believed possible when considering future design possibilities.

The sensitivity of the vehicle range as a function of the storage capacity of the flywheel is also an interesting variation. The flywheel for the ETV-2 was sized somewhat arbitrarily for 1.0 kW·h energy storage capacity based on the 80 km/h (50 mph) sustained speed up a 1.6 km (one mile) grade. If the battery is to be used for the hill climbing capability, then the optimum flywheel capacity for maximum range and acceleration would be substantially smaller. As the data of Figure 6-6 show, a reduction in the capacity to about 500 W·h would increase the vehicle range by 9.7 km (6 mi). No reduction in the acceleration capability would occur. The primary reason for this improvement is the reduction in the internal losses associated with the flywheel. The lower capacity flywheel also would be much smaller and lighter, and this effect is included in the sensitivity data.

A-9074

Figure 6-5. ETV-2 Range Sensitivity Analysis Data

Figure 6-6. Effect of Flywheel Energy on Range

### 6.3 PERFORMANCE IN EXTREME TEMPERATURE ENVIRONMENTS

#### 6.3.1 Range at Low Temperatures

The effect on the range of the vehicle after being subjected to extreme temperature soak conditions is shown by Figure 6-7. This curve shows that the battery capacity and thus the range of the vehicle is reduced to about 40 percent of its basic value when the battery soak temperature is  $-20^{\circ}\text{C}$ . This will occur after 2 or 3 days of  $-20^{\circ}\text{C}$  ambient temperatures. Studies have shown that insulation could delay the descent of the battery temperature, but excessive insulation weights (and volume) would be required for the seven days of exposure to this temperature. Continuous or intermittent trickle charging of the battery has been recommended as a partial solution and may be considered a practical compromise but would require a relaxation from the specification, which states seven days "unattended" exposure. If it is considered necessary to maintain the same range as for normal battery temperatures, the use of the fuel-fired heater appears to result in the least penalty to the system. The heater could be combined with the passenger heater or a separate unit could be provided as an option that would only be required in geographic areas with severe winters.

Figure 6-7. EP200AH Battery Relative Capacity as a Function of Temperature

### 6.3.2 Acceleration at Low Temperatures

A prime feature of the flywheel system is that the acceleration performance of the vehicle is essentially independent of the battery state of charge. Acceleration of conventional battery-powered vehicles normally drops off as a function of the remaining capacity of the battery. Therefore, acceleration is poor after cold soak of the battery. The ETV-2 depends on the flywheel, not the battery, for acceleration power, so full acceleration is available throughout the discharge cycle, independent of the soak temperature of the battery.

### 6.3.3 Performance at High Temperatures

Operation at temperatures as high as 52°C is not expected to be a problem with the ETV-2. As shown by Figure 6-7, there will be a small increase in the expected range of the vehicle due to slightly higher battery capacity at higher temperature. Acceleration performance will remain essentially the same.

An expected adverse effect of high battery temperature is a reduction in battery cycle life due to an increase in the rate of corrosion of the positive plate. Recent indications, however, are that this increase in corrosion rate may be less than previously thought, at least for temperatures below 50°C.

With the flywheel system, battery self heating will be less of a problem than for conventional electric vehicle systems because of the load leveling characteristic of the flywheel system. It is estimated that the normal temperature rise during driving will be about 6°C, which is not expected to be a significant additional penalty to the cycle life of the battery.

## 7. COMPUTER MODELS

A major achievement of the NTEV program was the development of two computer programs: one for predicting vehicle performance, and the other for determining battery pack characteristics. Listings for these programs were submitted to JPL, who documented the programs in JPL Report 5030-233\*. Section 7.1 presents the final listing, based on actual test data, of all of the various loss coefficients required in the computer program. A summary of the steps in using the battery model is presented in Section 7.2.

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\*Garrett/AiResearch Electric-Flywheel Simulation Program Documentation, JPL Report 5030-233, July 1978.



## 7.1 VEHICLE LOSSES

Vehicle losses are presented in terms of the performance parameters listed in Table 7-1 and the component characteristics and loss coefficients shown in Table 7-2.

### 7.1.1 Motor and Generator Model

The motor/generator can be modeled by the following equations. The voltage constant of the motor is a function of its field current:

$$(V/N)_m = f(I_{fm}) \quad (7-1)$$

where  $I_{fm}$  is the motor field current. This function also can be expressed by a saturation curve as shown in Figure 7-1. The internal generated voltage is proportional to the motor speed:

$$E_{gm} = (V/N)_m(N_m) \quad (7-2)$$

where  $N_m$  is the motor speed. The brush voltage drop is a function of the armature current:

$$V_{bdm} = 1.0 + 0.0139 I_m/A_b \quad (7-3)$$

where  $I_m$  is the armature current and  $A_b$  is the area of the brushes. The armature current is proportional to the voltage differential:

$$I_m = (V_{mg} - E_{gm} - V_{bdm})/R_m \quad (7-4)$$

where  $R_m$  is the resistance of the armature and  $V_{mg}$  is the line voltage. The net output torque of the motor is calculated by:

$$T_m = \frac{(V_{mg} I_m - P_{ml} 7.04)}{N_m} - T_{ml} \quad (7-5)$$

where  $P_{ml}$  is the power losses in the motor armature and  $T_{ml}$  is motor gear friction torque. The equations for the generator are similar:

$$(V/N)_g = f(I_{fg}) \quad (7-6)$$

TABLE 7-1.  
PERFORMANCE PARAMETERS

| Symbol    | Name                                  | Units   |
|-----------|---------------------------------------|---------|
| $V_{mc}$  | Line voltage                          | volts   |
| $V_{bdm}$ | Motor, brush voltage drop             | volts   |
| $V_{bdg}$ | Generator, brush voltage drop         | volts   |
| $E_{gm}$  | Motor, internal generated voltage     | volts   |
| $E_{gg}$  | Generator, internal generated voltage | volts   |
| $I_m$     | Motor, armature current               | amperes |
| $I_g$     | Generator, armature current           | amperes |
| $I_{fm}$  | Motor, field current                  | amperes |
| $I_{fg}$  | Generator, field current              | amperes |
| $R_m$     | Motor, armature resistance            | ohms    |
| $R_g$     | Generator, armature resistance        | ohms    |
| $R_{fm}$  | Motor, field resistance               | ohms    |
| $R_{fg}$  | Generator, field resistance           | ohms    |
| $B$       | Flux density                          | gauss   |
| $f$       | Frequency                             | Hertz   |
| $N_m$     | Motor, rotational speed               | rpm     |
| $N_g$     | Generator, rotational speed           | rpm     |
| $N_f$     | Flywheel rotational speed             | rpm     |
| $N_{ar}$  | Armature rotational speed             | rpm     |
| $N_a$     | Axle speed                            | rpm     |
| $T_m$     | Motor, output torque                  | lb-ft   |
| $T_g$     | Generator, output torque              | lb-ft   |
| $T_{ml}$  | Motor, gear friction torque           | lb-ft   |

TABLE 7-1 (Continued)

| Symbol    | Name                              | Units |
|-----------|-----------------------------------|-------|
| $T_{gl}$  | Generator, gear friction torque   | lb-ft |
| $T_f$     | Flywheel torque                   | lb-ft |
| $T_a$     | Axle torque                       | lb-ft |
| $N_b$     | Number of brushes                 | ----  |
| $P_{cu}$  | Copper losses                     | watts |
| $P_{bdp}$ | Power loss due to voltage drop    | watts |
| $P_{bf}$  | Brush friction power loss         | watts |
| $P_w$     | Armature windage power loss       | watts |
| $P_{sl}$  | Stray load losses                 | watts |
| $P_s$     | Shaft power                       | watts |
| $P_h$     | Hysteresis power loss             | watts |
| $P_e$     | Eddy losses                       | watts |
| $P_{mf}$  | Motor, field loss                 | watts |
| $P_{gf}$  | Generator, field loss             | watts |
| $P_{fl}$  | Flywheel power loss               | watts |
| $P_{pl}$  | Planetary transmission power loss | watts |
| $P_{fd}$  | Final drive loss                  | watts |
| $P_{ml}$  | Motor gear loss                   | watts |
| $P_{gl}$  | Generator gear loss               | watts |

TABLE 7-2

## COMPONENT CHARACTERISTICS AND LOSS COEFFICIENTS

|                                                     |             |                                                     |
|-----------------------------------------------------|-------------|-----------------------------------------------------|
| Effective vehicle drag area                         | $C_{dA}$    | 8.0 sq ft                                           |
| Rolling resistance coefficient                      | M           | 0.0075                                              |
| Area of motor/generator brushes                     | $A_b$       | 3.47 sq in.                                         |
| Motor armature resistance                           | $R_m$       | 0.0379 ohm                                          |
| Generator armature resistance                       | $R_g$       | 0.0379 ohm                                          |
| Number of brushes                                   | $N_b$       | 4                                                   |
| Brush pressure                                      | $P_{sp}$    | 3.0 psi                                             |
| Motor brush coefficient of friction                 | $C_{fm}$    | 0.19                                                |
| Generator brush coefficient of friction             | $C_{fg}$    | 0.16                                                |
| Diameter of commutator                              | $D_{com}$   | 4.25 in.                                            |
| Diameter of armature                                | $D_a$       | 5.0 in.                                             |
| Weight of armature teeth                            | $W_{th}$    | 6.86 lb                                             |
| Weight of armature core                             | $W_{er}$    | 7.48 lb                                             |
| Density of core material                            | D           | 0.295 lb/cu in.                                     |
| Hysteresis loss coefficient                         | n           | $6.788 \times 10^{-2}$                              |
| Hysteresis loss exponent                            | X           | 1.138                                               |
| Thickness of core lamination                        | d           | 0.014 in.                                           |
| Eddy loss coefficient                               | $\lambda$   | 31674                                               |
| Generator field resistance                          | $R_{gf}$    | 3.81 ohm                                            |
| Motor field resistance                              | $R_{mf}$    | 3.81 ohm                                            |
| Flywheel loss constant                              | $C_{fl}$    | 600 w                                               |
| Flywheel loss exponent                              | $x_{fl}$    | 2.33                                                |
| Maximum flywheel speed                              | $N_f$       | 25,000 rpm                                          |
| Loss constant of generator gear                     | $C_{gla}$   | $1.119 \times 10^{-3} \text{ w}/(\text{ft-lb-rpm})$ |
| Loss constant of generator gear                     | $C_{glb}$   | $0.726 \times 10^{-7} \text{ w}/\text{rpm}^2$       |
| Loss constant of motor gear                         | $C_{mla}$   | $1.491 \times 10^{-3} \text{ w}/(\text{ft-lb-rpm})$ |
| Loss constant of motor gear                         | $C_{mlb}$   | $0.746 \times 10^{-7} \text{ w}/\text{rpm}^2$       |
| Loss constant of planetary transmission             | $C_{pla}$   | $1.119 \times 10^{-2} \text{ w}/(\text{ft-lb-rpm})$ |
| Loss constant of planetary transmission             | $C_{plb}$   | $2.401 \times 10^{-3} \text{ w}/\text{rpm}$         |
| Loss constant of final drive differential/jackshaft | $C_{fda}$   | $6.488 \times 10^{-3} \text{ w}/(\text{ft-lb-rpm})$ |
| Loss constant of final drive differential/jackshaft | $C_{fdb}$   | $4.329 \times 10^{-6} \text{ w}/\text{rpm}^2$       |
| Loss constant of final drive differential/jackshaft | $C_{fda}^1$ | $5.190 \times 10^{-3} \text{ w}/(\text{ft-lb-rpm})$ |
| Loss constant of final drive differential/jackshaft | $C_{fdb}^1$ | $1.622 \times 10^{-6} \text{ w}/\text{rpm}^3$       |

Note: English units only are shown in this table because these are the actual units and values used in the computer models.



$$E_{gg} = (V/N)_g (N_g) \quad (7-7)$$

$$I_g = (E_{gg} - V_{mg} - V_{bdg})/R_g \quad (7-8)$$

$$T_g = \frac{(V_{mg} I_g + P_{g1})}{N_g} 7.04 - T_{gl} \quad (7-9)$$

S-47700

Figure 7-1. No-Load Saturation Curve

#### 7.1.2 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 due to hysteresis and eddy losses. The following equations are used to calculate these losses. Copper losses are given by:

$$P_{cu} = I^2 R \quad (7-10)$$

where  $I$  is the armature current and  $R$  is the resistance of the armature, adjusted for the predicted operating temperature. The power loss due to brush voltage drop is given by:

$$P_{bdp} = 1.0 I + 0.0139 I^2 / A_b \quad (7-11)$$

where  $A_b$  is the area of the brushes and  $I$  is the armature current. The brush friction loss is given by:

$$P_{bf} = N_b \times A_b \times P_{sp} \times C_f \times D_{com} \times N_a / 168.9 \quad (7-12)$$

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 rotational speed of the armature. Windage losses are given by:

$$P_w = (4.83 \times 10^{-5})(\pi D_a N_a)^2 \quad (7-13)$$

where  $D_a$  is the diameter of the armature and  $N_a$  is the rotational speed of the armature. Stray load losses are given by:

$$P_{sl} = 0.01 P_s \quad (7-14)$$

where  $P_s$  is the shaft power. Hysteresis loss is given by:

$$P_h = \left( \frac{W_{er}}{10^7 D} \right) f \eta B^X \quad (7-15)$$

where  $W_{er}$  is the weight of the core,  $f$  is the frequency,  $B$  is the flux density,  $D$  is the density of the core material,  $X$  is the hysteresis loss exponent for  $B$ , and  $\eta$  is the hysteresis loss coefficient. Eddy losses are given by:

$$P_e = \frac{1.645 d^2 W_{er} f^2 B^2 \lambda}{10^{16} D} \quad (7-16)$$

where the terms are as defined for hysteresis loss above,  $d$  is the thickness of core laminations, and  $\lambda$  is the eddy loss coefficient.

Thus, total power loss in each of the armatures is the summation of  $P_{cu}$ ,  $P_{bdp}$ ,  $P_{bf}$ ,  $P_w$ ,  $P_{sl}$ ,  $P_h$ , and  $P_e$ .

### 7.1.3 Motor and Generator Field Losses

The field losses are limited to copper losses as follows:

$$P_{gf} = I_{fg}^2 R_{fg} \quad (7-17)$$

$$P_{mf} = I_{fm}^2 R_{fm} \quad (7-18)$$

where  $I_{fg}$  and  $I_{fm}$  are the field currents to the generator and motor, respectively, and  $R_{fg}$  and  $R_{fm}$  are resistances of the field coils.

#### 7.1.4 Motor and Generator Gears

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

$$P_{gl} = C_{gla} T_g N_g + C_{glb} N_g^2 \quad (7-19)$$

$$T_{gl} = P_{gl} / N_g \quad (7-20)$$

$$P_{ml} = C_{mla} T_m N_m + C_{mlb} N_m^2 \quad (7-21)$$

$$T_{ml} = P_{ml} / N_m \quad (7-22)$$

where  $C_{gla}$ ,  $C_{glb}$ ,  $C_{mla}$ , and  $C_{mlb}$  are the loss constants of the generator and motor, 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.

#### 7.1.5 Flywheel

The flywheel power loss equation is determined by the flywheel spin-down test:

$$P_{fl} = C_{fl} \left( \frac{N_f}{N_{f \max}} \right)^{X_{fl}} \quad (7-23)$$

where  $C_{fl}$  is the flywheel loss constant,  $N_f$  flywheel speed, and  $X_{fl}$  flywheel loss exponent.

#### 7.1.6 Planetary Transmission

The loss expression for the planetary transmission gear set is similar to the power- and windage-dependent equation used for the generator and motor gears:

$$P_{pl} = C_{pla} T_f N_f + C_{plb} N_f^2 \quad (7-24)$$

where  $C_{pla}$  and  $C_{plb}$  are the loss constants for the gearset,  $T_f$  is the torque of the flywheel, and  $N_f$  is the flywheel speed.

#### 7.1.7 Final Drive Differential and Jackshaft

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

$$P_{fd} = C_{fda} T_a N_a + C_{fdb} N_a^2 \quad (7-25)$$

$$T_a = P_{fd} / N_a \quad (7-26)$$

where  $C_{fda}$  and  $C_{fdb}$  are the loss constants for the final drive and jackshaft,  $T_a$  is the torque at the axle, and  $N_a$  is the speed of the axle.

#### 7.1.8 Electronic Component Losses

Electronic losses are shown in Figure 7-2. Losses that vary between a minimum and a maximum are determined from the equations listed in Table 7-3.

Figure 7-2. Electrical System Power Losses



TABLE 7-3  
POWER CONVERSION UNIT  
POWER DISSIPATION MODEL

Armature chopper

$$P_{\text{LOSS}} \text{ in watts} = 1.05 I_L + I_L D + 100 D$$

$D$  = Duty cycle, 0 - 1.0

$I_L$  = Load current, 0 - 120 amp

Field choppers

$$P_{\text{LOSS}} \text{ in watts} = 3.98 I_F$$

$I_F$  = Field current, 0 - 15 amp

Accessory power supply

$$P_{\text{LOSS}} \text{ in watts} = 20 + 0.35 (75 D + 12 I_L)$$

$D$  = Duty cycle, 0 - 1.0

$I_L$  = Accessory load current, 0 - 60 amp

|                        |                      |
|------------------------|----------------------|
| Pump controls          | 13 watts             |
| Protective controls    | 26 watts             |
| Contactors and drivers | 3.6 watts/contactors |

## 7.2 BATTERY MODEL

The battery model as used in the vehicle performance simulation computer program is a dynamic performance model based on the steady-state battery performance program developed by Eagle-Picher and the principle of fractional utilization.

Inputs to the model are current discharge rate (amperes), ambient battery temperature, time increment, and initial battery capacity used (or depth of discharge). Outputs are battery terminal voltage and present battery capacity used. The program logic sequence of the model is shown in Figure 7-3.

Figure 7-4 shows the relationship between the battery's discharge rate and discharge time. Given the discharge rate, the corresponding discharge time can be identified from the curve. The available capacity of the battery is the product of the discharge rate and discharge time. A capacity modification factor due to the ambient temperature is obtained from Figure 7-5. Multiplying the available capacity by this factor gives the actual capacity. Ampere-hours used during an increment of time is the product of the discharge rate and the time increment. According to the principle of fractional utilization, the fractional capacity used during this increment of time is the ampere-hours used divided by the actual capacity at that discharge rate. The accumulation of this fractional capacity used becomes total capacity used.

Figure 7-6 shows the relationship between the discharge time and the cutoff voltage. The terminal voltage is a function of the cutoff voltage and the total capacity used as expressed by the following equations:

$$B = \log (3900 - 1900 \times \bar{V}_C) - 2.89835$$

$$\bar{V} = \left[ 3900 - e^{(0.089835 \times \text{DOD} + B)} \right] / 1900$$

where  $\bar{V}_C$  = cut-off voltage

DOD = depth of discharge (capacity used) in percent

$\bar{V}$  = terminal voltage.

A typical voltage profile is shown in Figure 7-7.

The fractional utilization method is not the best dynamic model, especially if the load demands on the batteries vary considerably. But because of the load-leveled feature of the ETV-2, the error created by using this method becomes negligible and is usually on the conservative side. With the simplicity of its application, AiResearch chose to use the fractional utilization method in the simulation of the battery dynamic model.

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Figure 7-3. Dynamic Battery Performance Model ; Program Logic Sequence

Figure 7-4. EP200AH Battery Discharge Time vs Discharge Rate

Figure 7-5. EP200AH Battery Relative Capacity as a Function of Temperature

Figure 7-6. EP200AH Battery Cutoff Voltage vs Discharge Time

Figure 7-7. EP200AH Battery Voltage vs Discharge



## 8.1 COMPONENT AND SYSTEM TESTING

Component and system testing on breadboard equipment demonstrated the feasibility of the unique battery-motor-flywheel propulsion system for the ETV-2. These tests culminated in a number of acceleration/deceleration runs and driving cycles under both open- and closed-loop control against a vehicle load on a dynamometer. Figures 8.1-1 and 8.1-2 are reproductions of oscillograph traces recorded during two of the runs.

In addition to demonstrating concept feasibility, a principal objective was to obtain component loss data for verifying or altering the component loss equations used in the computer model of the system. The comparisons in Figures 8.1-3 and -4 show very close agreements between predicted vehicle and flywheel speeds and speeds recorded during actual runs. This correlation provided confidence in predictions of the vehicle performance, characteristics, and effectiveness of future improvements or modifications.

### 8.1.1 Controls and Electronics

Development efforts and checkout tests were conducted on the controller, power conversion unit, charger, and data recorder to verify functional performance under actual driving conditions. These efforts demonstrated reliable operation and achievement of the functional design requirements for the controls and electronics components in their intended environment.

#### 8.1.1.1 Controller

Initial component testing was done using a brassboard controller and a test panel, which simulated all vehicle inputs while the controller was tested to verify that component functional requirements were met. The first step in controller checkout consisted of testing all hardware functions before implanting the operating software in the controller memory. Hardware checkout included testing the controller central processor section for proper full-speed operation, then proceeding to all input and output circuits and testing for scaling and accuracy. The controller bench check in the laboratory was continued until all circuit functions were verified to specification limits.

A typical input circuit test involved the simulation of vehicle speed, motor speed, flywheel speed, and generator speed with external frequency oscillators. Appropriate signals were applied to each channel over the full range of intended operation. The system development unit (SDU) was connected to the breadboard controller and used to verify that the speed signals were processed by the controller.

All software subsystems were tested using the brassboard controller, test panel, and SDU. After subsystem checkout, the complete software control program was assembled and tested until all software discrepancies (bugs) were removed.

Finally, a hardware and software functional check breadboard was completed and the package was ready for the integrated breadboard power system development test and prototype hardware implementation.

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IM - MOTOR ARMATURE CURRENT  
IFM - MOTOR FIELD CURRENT  
IFMC - MOTOR FIELD CURRENT COMMAND  
IG - GEN. ARMATURE CURRENT  
IFG - GEN. FIELD CURRENT

IFGC - GEN. FIELD CURRENT COMMAND  
VA - ARMATURE VOLTAGE  
NM - MOTOR SPEED  
NF - FLYWHEEL SPEED  
ICC - CHOPPER CURRENT COMMAND

Figure 8.1-1. ETV-2 Closed-Loop Control

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Figure 8.1-3. Comparison Between a Computer Simulation and an Actual Acceleration-Cruise-Deceleration Test Run

a. INPUT CURRENTS TO MOTOR AND GENERATOR MATCH ACTUAL TEST CURRENTS (CURVES COINCIDE)

b. AXLE TORQUE

c. FLYWHEEL SPEED AND VEHICLE SPEED

S-41442 -A

Figure 8.1-4. Comparison Between a Computer Simulation and an Actual Acceleration-Deceleration Test Run

Some of the significant problems encountered during development involved electrical noise and thermal limitations. Some layout rework was required to eliminate signal coupling and crosstalk on the low-level, high-frequency memory and CPU boards to ensure reliable operation. The most damaging noise source was found to be the controller switching power supply module, which also required re-layout as well as the addition of increased filter capacitors at the power supply inputs and outputs.

Intermittent high-temperature shutdowns were corrected by changing vendors for several components and making minor circuit changes. The controller was then successfully tested at 125°F for 16 hr and at -40°F for 1.5 hr.

#### 8.1.1.2 Power Conversion Unit

Component testing was accomplished with a four-motor test rig and dissipative load, which simulated the flywheel and road load. A manual control box was constructed to simulate the closed-loop dynamics and output command signals of the controller. Several hundred operating hours of motor/generator running time were accumulated, during which all modes of power flow and intermode transitions were verified, and approximately 100 deep flywheel cycles (24 000 to 10 000 to 24 000 rpm) were simulated.

Two specific developments deserve comment. The original design for the 120-A armature chopper utilized four parallel power transistors arranged so that the transistor switch was between the battery and the motors (i.e., high-side switching). Consequently, the transistor base drive circuitry was transformer-coupled to the output stage to accommodate the 108-V differential between the drive transistor and the PCU ground potential, while maintaining efficient control of the power stage devices. During initial testing, the high-speed drive pulses required to control the output power transistor could not be coupled across the transformer at speeds high enough to prevent excessive chopper device dissipation. To resolve the difficulty, the armature chopper was redesigned so that the power switching devices were relocated between the motor return leg and PCU ground. This allowed a direct-coupled base drive circuit, eliminating the coupling transformer, since the chopper transistor base circuits are always at ground potential. At the same time, the four parallel single transistor devices were replaced with seven parallel Darlington transistors, which achieved two benefits. First, the cost of the power stage was reduced by 82.5 percent, and second, the Darlington devices are inherently extremely well-matched so that direct parallel hookup achieved excellent device current sharing without additional drive circuit complexity.

The 15-A field choppers initially were operated as conventional ripple regulators with a high-side transistor switch. A ripple regulator works by continuously measuring actual field current and turning the chopper transistor on or off whenever the actual current deviates from the desired current level by more than a predefined threshold. During motor testing, it was found that current and voltages applied to the armature circuit were interfering with the field current control because of transformer-type coupling between the motor armature and field windings. This problem was resolved by applying a sampling technique to the field chopper current control circuit synchronous with the armature chopper clock but out of phase with armature chopper transitions so that armature transients were not coupled to the field control circuits.

The remaining problems in power electronics circuit design were generally related to making the circuitry immune to EMI signal coupling. These packaging problems were compounded and made more difficult than normal because the final design was very close coupled in order to fit the power conversion unit heat-sink subassemblies into a small volume behind the rear seat support structure. The PCU contains all of the low-level control circuitry as well as the power switching stages, creating a "dirty" EMI environment for the electronics.

#### 8.1.1.3 Charger

A breadboard charger was fabricated and subjected to a number of charging cycles--first on a set of standard golf cart type batteries and later on several sets of the Eagle-Picher EP 200 AA batteries for the ETV-2. Both the 15-A and 30-A circuits were used.

An initial breadboard charger provided the constant-current control with high voltage cutoff and then constant-voltage operation with trickle charging current. During the final development test phase, the constant-voltage mode of operation was raised to 151 V, and a timer function of 3 hr and a temperature override were introduced for constant-voltage operation. Either the 3-hr timer or the temperature override terminates the charger cycle. The changes were made to reduce the charging time and increase the ampere-hours put into the battery in one charge.

#### 8.1.1.4 Data Recorder

A short software subroutine was developed to store dummy data parameters from the vehicle controller in the data recorder cassette tape in the laboratory. The recorded tape was then played back through the tape-read interface box and displayed on a CRT terminal to give a visual check of the accuracy of the recorded data.

#### 8.1.2 Motor/Generator

The categories of tests described below were conducted on motor/generator units with configuration variations shown in Table 8.1-1. Characteristics of the final units are shown in Figure 4.4.1-5.

Torque Speed--Results show that a torque of 72.8 N·m (53.7 lb-ft) and 2627 rpm could be achieved using 2V Permendur armature laminations and an air gap of 0.84 mm (0.033 in.). At a 1.52-mm (0.060-in.) air gap, M-19 and Permendur machines produced nearly equal torque; however, at 0.84 mm, Permendur exceeded M-19 by about 40 percent (Figure 8.1-5).

Efficiency--Efficiency tests were conducted on an M-19, 1.52-mm air gap machine, and the results were adjusted as required for the 2V Permendur, 0.84-mm air gap machine. Predicted peak efficiency is 90 percent.

Mechanical Losses and Commutation Characteristics--The results of these two tests were used to select the specific brush load and material for the motor and generator. Coast-down tests provided data for brush, bearing, and windage losses. As expected, a brush load of 27.6 kPa (4 psi) resulted in approximately 33 percent more loss than a load of 20.7 kPa (3 psi).

TABLE 8.1-1

## MOTOR/GENERATOR TEST CONFIGURATIONS

| Characteristics               | Configuration                                                                          |
|-------------------------------|----------------------------------------------------------------------------------------|
| Armature lamination materials | 2V Permendur and M-19 steel                                                            |
| Nominal air gap               | From 1.52 mm to 0.84 mm (0.060 in. to 0.033 in.) in increments of 0.23 mm (0.009 in.). |
| Brush grade material          | Morganite H-100, Union Carbide N-4, and advanced EG 7097                               |
| Brush pressure                | 21 kPa and 28 kPa (3 psi and 4 psi)                                                    |

Figure 8.1-5. Effect of Armature Material and Air Gap on Torque Output

In addition, the loss for Union Carbide N-4 material was 28 percent greater than for the Morganite H-100 at the same load. Commutation tests conducted for both brush materials and loads for speeds up to 11 700 rpm and maximum armature current showed that the lower load would be acceptable for both materials but the H-100 material was not suitable at the highest speeds. Therefore, the final brush configuration is N-4 at 27.6 kPa (4 psi) for the motor and H-100 at 20.7 kPa (3 psi) for the generator, which does not operate as fast as the motor.

Brush Life--Wear tests were conducted on Union Carbide H-4 and Advanced Carbon EG 7079 materials, both at a load of 27.6 kPa (4 psi). The extrapolated results (assuming linear wear with time) showed a wear life from 1500 to 2500 hr, which far exceeds the recommended 500-hr maintenance interval.

No-Load Saturation--No-load saturation curves were determined for a field current up to 15 A. The volts/rpm value for a 2V Permendur, 0.84-mm (0.33-in.) air gap machine was 0.032 as opposed to 0.028 for the M-19, 1.52-mm (0.060-in.) air gap machine at a field current of 14 A.

Brush Voltage Drop--Tests were conducted up to 60 A armature current and a straight-line extrapolation to 200 A (390 W) is used for the computer model.

Armature Iron Loss--Tests were conducted for conditions corresponding to field currents up to 13 A, where the loss was 270 W.

### 8.1.3 Flywheel Spindown

Flywheel spindown tests were conducted on the flywheel component and on the power unit to improve dynamic characteristics and reduce power losses. Variations in configuration included bearing type; bearing resilient mount characteristics, outboard bearing oil, outboard lube pumping scheme, and transmission lube oil pressure. Final tests show a flywheel loss of 425 W at 24 000 rpm that decreases approximately linearly with speed.

### 8.1.4 Vacuum System

The development of the vacuum system was divided into two areas: the forepump, and the molecular pump. The forepump must be capable of evacuating the flywheel cavity to a pressure of 1.0 torr ( $\sim 1.0$  mm HgA) or better within 5 min in order to generate the operating conditions under which the molecular pump can further evacuate the flywheel cavity to an absolute pressure of 0.001 to 0.0015 mm HgA.

For the forepump, the requirements first favored a Gerotor-type positive-displacement pump element. The initial testing, however, showed that this type of pump could not be refined to satisfy the low pressure design goal life requirement in time to be utilized. Lubrication and close clearances required careful development in order to produce a satisfactorily performing unit.

An alternative to the Gerotor pump was a commercially available vane pump. Tests were conducted to increase the capacity of the pump and to match it with



a dc motor. The final configuraton was a dc motor with a vane pump on each end. One vane pump was modified for rotation in the reverse direction.

A molecular pump consists of a smooth-surface rotating disc in close proximity to a stator with spiral channels. When the disc rotates in the direction of the spiral channels, the high vacuum (low absolute pressure) is at the center of the disc and the pump works against the low vacuum (fore pressure) at the rim. In order for the molecular pump to generate the desired low pressure level a forepump is required to maintain a certain reduced pressure level at the discharge (i.e., at the rim) of the molecular pump.

The test program involved variations in rotor-to-stator clearance, rotor speed, fore pressure, and channel geometry.

#### 8.1.5 Battery Tests

Battery tests, conducted to establish guidelines for a charging schedule, confirmed that the 18-battery pack could be charged in slightly over 8 hr. For a pack that has been discharged by 90-percent or more, the basic schedule will consist of a constant-current mode (21 to 25 A) for approximately 5-1/2 hr followed by a constant-voltage mode (151 V) up to 2-3/4 hr.

#### 8.1.6 System Tests

##### 8.1.6.1 Laboratory Layout

All system tests were conducted in a laboratory setup simulating vehicle operation. The laboratory layout is shown in Figures 8.1-6 through 8.1-8. The power unit was mounted on a Volkswagen chassis and connected to the rear wheels, which drove a Clayton dynamometer. The output shafts from the power unit to the wheels included the wheel bearings, brake drums, tires, and a torque meter, which measured the torque to each wheel. The dynamometer simulated the aerodynamic, rolling, and inertia loads.

Operational control was accomplished with breadboard electronic components. For closed-loop operation, throttle and brake position signals were imposed from the test console. For manual operation, a manual command box was used to perform the control function of the controller. A charger charged the batteries as required. When batteries were excessively low, an external dc power supply was used to supplement battery output.

##### 8.1.6.2 Subsystem Checkout

These test results verified that all electrical components were correctly interconnected and that they operated in all modes during the power system testing.

##### 8.1.6.3 Operational Modes

The operational mode checkouts consisted of prestart, reverse, and full driving modes, including tests of steady-state operation and mode transition checks. Closed-loop operation of the control system was verified over all vehicle automatic control modes.

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Figure 8.1-6. Breadboard Power System Test Facility and Vehicle

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Figure 8.1-7. Breadboard Power System Installation on Volkswagen Chassis

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Figure 8.1-8. Laboratory Layout for Breadboard System Tests

#### 8.1.6.4 Final Drive Losses

For this test only, the power unit transmission was modified so that there was no mechanical connection between the planetary carrier shaft and the final drive. All power output from the motor went to the final drive; none went to the planetary. The system was then operated at various motor speeds against a dynamometer load measured at the axle. The final drive power loss is the difference between the measured motor input and the sum of the dynamometer load plus motor losses (determined from component tests). Results showed a final drive loss varying nonlinearly from 100 W at 31 km/h (19 mph) to 700 W at 101 km/h (63 mph) (Figure 8.1-9).

#### 8.1.6.5 Planetary Transmission Losses

With the transmission in its normal configuration and the final drive locked, the generator was used to drive the flywheel through the planetary geartrain. Planetary losses were determined by subtracting calculated generator losses and flywheel windage/friction losses from the electrical input to the generator. Results showed the planetary loss to be approximately linear with flywheel speed and equal to 660 W at 25 000 rpm (Figure 8.1-10).

#### 8.1.6.6 Flywheel Acceleration/Deceleration

Several acceleration and deceleration runs were conducted at generator armature currents of 120 and 200 A. Results for two runs are shown in Figure 8.1-11.

A-6364

Figure 8.1-9. Final Drive Loss Curve

Figure 8.1-10. Planetary Transmission Losses

a. FLYWHEEL DECELERATION AT 120 A

S-41115-A

b. FLYWHEEL ACCELERATION AT 200 A

S-41114-A

Figure 8.1-11. Transmission and Flywheel Losses During Deceleration and Acceleration

## 8.2 FINAL CHECKOUT TEST DEMONSTRATIONS

The final test effort, which completed the NTEV contract, was a series of operational checkout tests to demonstrate proper functioning of the components throughout the various control modes and transitions. The tests were conducted to demonstrate that the vehicle was completely functional and ready for delivery to JPL for further test evaluations. (All performance evaluation testing and driving tests had been eliminated from the contract.)

The checkout tests were run on a dynamometer with a minimum of instrumentation. Figure 8.2-1 shows that for simulated vehicle speeds up to 64 km/h (40 mph) the dynamometer load, which simulated the road load, was higher than would be imposed on this particular vehicle during actual driving. This means that actual acceleration and range below 64 km/h will be higher than measured during these tests.

Initially the vehicle was operated over the various control modes to confirm satisfactory operation of the automatic controls during steady-state and transient conditions.

### 8.2.1 Power Conversion Unit Cooling Test

For this test, the vehicle was operated at a constant 24 km/h (15 mph) with the maximum dynamometer load applied and an accessory load of 330 W. The PCU maximum armature chopper current was not measured, but was assumed to be at or near the maximum steady-state current. The heat sink and transistor temperatures were monitored until steady-state conditions were achieved.

The ambient cooling air temperature was about 21°C (70°F). The steady-state heat sink temperatures were 41°C and 31°C (105°F and 88°F) for the roadside and curbside assemblies, respectively, indicating that cooling would be adequate even at the maximum ambient temperatures.

### 8.2.2 Braking Evaluation

Tests were conducted to check braking capability as a function of brake pedal position and pedal force.

A mechanical stop was used to control pedal travel to 13-mm (0.5-in.) increments. With the flywheel at 15 000 to 20 000 rpm, the vehicle was brought to a stop from 48 km/h and from 72 km/h (30 and 45 mph) at various pedal positions. Sufficient runs were conducted to measure and to plot stopping time vs brake pedal position. The test results are shown in Figure 8.2-2. The regenerative braking alone decelerates the car from 72 km/h to zero in 13 s. This is an average deceleration of over 5.5 km/h/s, which is greater than is normally used in city traffic. The hydraulic brakes will only be needed for faster stops or abnormally high deceleration.

### 8.2.3 Interior Noise Level Measurement

While the ETV-2 was operated at a cruise condition, interior noise levels were measured at the driver's ear level. The results are summarized in

Figure 8.2-1. Comparison of Dynamometer Load and Calculated Road Load

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Figure 8.2-2. Braking Evaluation



Table 8.2-1. However, due to the unusually noisy laboratory area and confined laboratory structure, the measured noise level may be slightly higher than the actual vehicle noise level.

TABLE 8.2-1

MEASURED INTERIOR NOISE LEVEL

|                     |    |      |        |        |        |
|---------------------|----|------|--------|--------|--------|
| Vehicle speed, km/h | 0  | 0    | 48     | 48     | 72     |
| Flywheel speed, rpm | 0  | 6000 | 18 000 | 20 300 | 20 000 |
| Noise Level,<br>dBa | 67 | 65   | 72     | 73     | 78     |

#### 8.2.4 Simulated Driving

Tests were conducted to confirm that the power system and vehicle accessories were ready for operation under the variety of driving situations that could be imposed by a driver. Accelerations, decelerations, and cruise conditions were operated under various combinations of battery states and flywheel speeds. These were the first tests during which a driver determined the status of the vehicle and reacted from the information presented on dashboard gages and lights in addition to the laboratory instrumentation.

A series of acceleration tests at different initial conditions of vehicle speed and flywheel speed is summarized in Table 8.2-2. Generally, the measured accelerations agreed with the expected results although the dynamometer loads are higher than would occur in actual driving.

A series of tests was run to demonstrate constant-speed driving, beginning with a fully charged battery and continuing until the system shut down at 90 V. These tests were to confirm steady-state operational temperatures of the motors and to show system characteristics when approaching the fully discharged condition of the battery.

Two constant-speed runs at 56 km/h (35 mph) were conducted. In the first, the battery performance was unexpectedly poor and delivered only 107.8 A·h at a nearly constant discharge of 60 A. Eagle-Picher recommended that the battery be heated during recharge to approximately 38°C (100°F), which more nearly equals the end-of-charge temperature under normal ambient conditions. When this was done, however, the delivered capacity was still only 142.6 A·h instead of the required 180 A·h.

TABLE 8.2-2

## ACCELERATION TEST SUMMARY

| Speed,<br>km/h | Flywheel<br>speed, rpm | Acceleration<br>time, s |
|----------------|------------------------|-------------------------|
| 0 to 48        | 21 700 to 20 400       | 8.4                     |
| 0 to 48        | 22 000 to 20 430       | 9.7                     |
| 0 to 72        | 23 250 to 21 100       | 13.0                    |
| 0 to 89        | 23 000 to 20 500       | 17.0                    |
| 0 to 96        | 23 000 to 19 500       | 23.5                    |
| 0 to 96        | 20 500 to 15 580       | 28.0                    |
| 48 to 80       | 19 350 to 17 300       | 9.05                    |
| 48 to 96       | 21 700 to 18 200       | 16.5                    |
| 0 to 48        | 14 100 to 12 300       | 8.8                     |
| 0 to 48        | 14 000 to 11 600       | 9.0                     |
| 0 to 76        | 16 500 to 12 750       | 19.3                    |
| 48 to 80       | 17 400 to 14 800       | 9.1                     |
| 48 to 80       | 16 000 to 14 200       | 10.6                    |

Note: Speed conversion:      km/h      mph

|    |    |
|----|----|
| 48 | 30 |
| 72 | 45 |
| 76 | 47 |
| 80 | 50 |
| 89 | 55 |
| 96 | 60 |

Another battery test was conducted in which, at the end of the normal charging cycle of 10.1 hr, the charger was restarted, adding another 2.4 hr of charge time. The prolonged charge period at the high-voltage condition shown in Figure 8.2-3 illustrates the charge current and voltage schedule. At the end of the 12.45-hr charge, the electrolyte specific gravity was slightly above 1.32 and the total input was 191.7 A·h. Immediately after the charge period, the vehicle was driven at a 60-A battery discharge current. The battery pack delivered 168 A·h (17.6 kW·h) as shown in Figure 8.2-4. It was therefore concluded that the rated capacity still could not be achieved and that the normal charging procedure would be used for the remainder of the vehicle test.

Several acceleration and constant-speed runs at 72 and 86 km/h (45 and 55 mph) were made. It was determined that a minimum initial flywheel speed of 21 000 rpm was required for full-throttle acceleration to 72 km/h. It was also determined that the maximum vehicle speed at which the flywheel could be maintained at a constant speed was 80 km/h (50 mph). A test run was made by accelerating to 72 km/h, cruising at constant speed for approximately 40 min and then cruising at a constant speed of 50 mph until the system shutdown. Operation was successful and mode transitions were smooth. A summary of the data is shown in Figure 8.2-5.

#### 8.2.5 SAE J227a(D) Cycle Tests

The vehicle was operated over the J227a(D) driving cycle repeatedly until the system shut down at 51 cycles. Three representative cycles are shown in Figures 8.2-6 through 8.2-8. The similarity of the data in each of the cycles is representative of the predicted characteristics. The endurance of only 51 cycles, however, is less than predicted, and is attributed to the low battery performance as well as the excessive dynamometer load.

#### 8.2.6 Federal Urban Driving Schedule (FUDS)

The vehicle was operated over one complete FUDS and all operations were normal. The acceleration profiles were achieved. A summary of the data from this schedule is shown in Figure 8.2-9.

| BATT.<br>CELL<br>NO. | SP.<br>GR. | TEMP,<br>°C |
|----------------------|------------|-------------|
| 8                    | 1.145      | 19          |
| 9                    | 1.155      | 19          |
| 20                   | 1.145      | 19          |
| 26                   | 1.145      | 19          |
| 35                   | 1.135      | 19          |
| 53                   | 1.145      | 19          |

NOTE: ELAPSED TIME = 12.45 h  
TOTAL INPUT = 191.7 A·h  
23.3 kW·h

| BATT.<br>CELL<br>NO. | SP.<br>GR. | TEMP,<br>°C |
|----------------------|------------|-------------|
| 8                    | 1.312      | 48          |
| 9                    | 1.310      |             |
| 20                   | 1.302      | 46          |
| 26                   | 1.314      | 48          |
| 35                   | 1.312      | 47          |
| 53                   | 1.310      | 42          |

| BATT.<br>CELL<br>NO. | SP.<br>GR. | TEMP<br>°C |
|----------------------|------------|------------|
| 8                    | >1.32      | 45         |
| 9                    | >1.32      |            |
| 20                   | >1.32      | 44         |
| 26                   | >1.32      | 43         |
| 35                   | >1.32      | 44         |
| 53                   | >1.32      | 39         |

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Figure 8.2-3. Battery Charge Profile for Constant 56 km/h Test

| CELL NO. | SP. GR. | TEMP °C |
|----------|---------|---------|
| 8        | >1.320  | 45      |
| 9        | >1.320  | 45      |
| 20       | >1.320  | 46      |
| 26       | >1.320  | 43      |
| 35       | >1.320  | 46      |
| 53       | >1.320  | 39      |

NOTE: ELAPSED TIME = 2.83 h  
TOTAL OUTPUT  
168.0 A·h  
17.59 kW·h

| CELL NO. | SP. GR. | TEMP °C |
|----------|---------|---------|
| 8        | 1.143   | 44      |
| 9        | 1.142   | 44      |
| 20       | 1.140   | 44      |
| 26       | 1.140   | 42      |
| 35       | 1.140   | 44      |
| 53       | 1.140   | 42      |

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Figure 8.2-4. Battery Current and Voltage during Constant 56-km/h Test

A-6314

Figure 8.2-5. Combined Acceleration and Constant Speed Run Results

A-6091

Figure 8.2-6. J227a(D) Driving Cycle No. 1

A-6093

Figure 8.2-7. J227a(D) Driving Cycle No. 25

A-6092

Figure 8.2-8. J227a(D) Driving Cycle No. 47

Figure 8.2-9. Federal Urban Driving Schedule Test Run

## 9. CONCLUSIONS AND RECOMMENDATIONS

### 9.1 CONCLUSIONS

The development and preliminary testing of the unique AiResearch Near-Term Electric Vehicle (NTEV) included several new component developments, each of which is considered an advancement in the technology base for future developments in electric vehicles. These are:

- A composite-material energy storage flywheel
- An electromechanical transmission with infinitely variable gear ratio
- A crashworthiness design technique for plastic bodies and frames
- A high-energy-density battery with tubular positive plates and antimony-free grid materials

Each of these elements was developed to the degree necessary to complete and test the final vehicle. Component losses were measured during the test program, and although they were satisfactorily low, it is evident that further development would improve the overall system efficiency. The vehicle has been tested on a dynamometer for the equivalent of approximately 2414 km (1500 mi) through all of the various modes of power system operation.

The operational tests were successful in demonstrating the principles involved in the vehicle subsystems and the advantages of the concept.

Final performance testing and evaluation by JPL will show the degree of compliance with the goals of the program.

### 9.2 RECOMMENDATIONS

Further development on the vehicle subsystems would be advantageous to realize the full performance potential of the new concept. With further technical development work to improve some of the component efficiencies, the full advantage of the energy-saving flywheel concept can be realized. Improvement in the design and fabrication procedures for the plastic body and frame will allow significant weight reduction by taking better advantage of the potential weight savings of the materials. Further confirmation is required of the expected energy capacity and extended life of the tubular-plate battery.

In addition to the recommendation for further technical development, it is recommended that variations in the system concept be explored and analyzed to simplify the system and reduce the size, weight, and potential costs.

During the latter part of the program, several variations of the flywheel system were conceived, which should be given consideration for follow-on work. One variation is particularly interesting because it would substantially increase the mid-range acceleration power without requiring any change in the hardware. It requires only a small change in the controller software, which alters the existing control strategy.

The increase in power occurs in the mid-speed range between 16 and 42 mph, where both the motor and generator are above base speed and operating at full battery voltage of 108 V. As described in the previous sections, maximum vehicle acceleration is achieved by cutting off the battery current so that the generator current (and consequently the torque) is maximized. The existing control strategy limits the generator armature voltage to that of the battery, which is in parallel with the armature chopper bypass contactor in the closed position. However, the armature chopper also can be deactivated with the bypass contactor remaining in the open position if there is an additional instruction from the controller setting the chopper duty cycle to zero. With this modification, the armature voltage is completely disassociated from the battery voltage and can be permitted to increase in response to the demand for increased acceleration power or torque. The armature voltage can increase from 108 V at 16 mph to approximately 200 V at 30 mph, with a consequent increase in output power. There is no change in the armature current. The curves in Figure 9-1 show the output power of the existing system under this condition; the added power shown by the shaded area is the increase available by the change in controller software. Computer simulations of the modification were made to evaluate the effect of the change on acceleration. The results show that the time to accelerate from 0 to 50 mph is reduced from 17 to 14 s and the merging time from 25 to 55 mph is reduced from 13 to 10.5 s. Figure 9-2 shows the effect of the change on the acceleration capability of the vehicle.

Another alternate that should be considered is the flywheel system concept that was used on the DOT-sponsored Advanced Concept Train (ACT-1), which eliminated all of the power electronics in the armature circuit. This would greatly simplify the system, making it much less costly to produce.

Figure 9-1. ETV-2 Power Output and Increased Power Possible through Modified Controller Software

Figure 9-2. ETV-2 Acceleration Performance Showing Increased Acceleration Possible through Modified Controller Software

APPENDIX A

BATTERY LIFE CYCLE TEST RESULTS REPORT  
BY EAGLE-PICHER INDUSTRIES, INC.



## 1.0 Purpose

The purpose of the Performance Test Program was to demonstrate the capability of the selected battery design and also, to provide data for comparison and correlation with the analytical computer performance model. The purpose of the life cycle test program is to demonstrate that the improved lead-acid battery adapted for The Garrett Corporation, AiResearch Manufacturing Company of California has suitable life for use in electric vehicles.

## 2.0 Description of the Battery

The battery for the near term Garrett AiResearch electric vehicle consists of eighteen (18) six volt units. Each unit contains three cells with an ampere hour capacity of approximately 180 AH at the 60 ampere discharge rate. The size of each 6 volt unit is the standard GC-2 size electric vehicle battery. The battery utilizes the lead/lead dioxide electro-chemical system with sulphuric acid as the electrolyte. Design parameters include tubular positive plates and non-antimonial grid materials. These design variables were utilized to provide long cycle life and low maintenance.

Connections between each six volt unit are lead plated copper braided cable straps. Connection is made by soldering these straps to the terminal post of the 6 volt unit.

## 3.0 Battery Adaptation

The battery adaptation project consisted of an evolutionary design program. The objective of the program was to reduce battery weight by improving active materials' utilization and reduction of plate grids, current carrying parts, case and cover. Six battery packs were manufactured and delivered during the contract. Table I presents the improvement demonstrated during the adaptation program.

## 4.0 Test Specimens

Two 6 volt batteries were selected for the Performance Test Program and six

Table I

NTEV

Eagle-Picher  
6V200 Tubular Positive Plate Battery  
Energy Density Improvement

The following table presents performance,  
weight and energy density for battery  
packs no. 1 through 6.

|                         | <u>Packs<br/>1 &amp; 2</u> | <u>Packs<br/>3 &amp; 4</u> | <u>Packs<br/>5 &amp; 6</u> |
|-------------------------|----------------------------|----------------------------|----------------------------|
| Capacity (AH)           | 167 AH                     | 176 AH                     | 185 AH                     |
| Capacity (WH)           | 986 WH                     | 1025 WH                    | 1071 WH                    |
| Weight (Lbs.)           | 61.2 Lbs.                  | 60.3 Lbs.                  | 59.2 Lbs.                  |
| Weight (KG)             | 27.8 KG                    | 27.4 KG                    | 26.9 KG                    |
| Energy Density (WH/LB.) | 16.1 WH/LB.                | 17.0 WH/LB.                | 18.1 WH/LB.                |
| Energy Density (WH/KG)  | 35.5 WH/KG                 | 37.4 WH/KG                 | 39.8 WH/KG                 |

All data is for 6 volt monoblocks in GC-2 configuration

Above capacities are for 3 hour discharge rate

6 volt batteries were selected for the Life Cycle Test Program. These batteries were constructed at the same time as deliverable battery packs #5 and #6 and represent the final design for the Adaptation Program.

## 5.0 Test Results

### 5.1 Acceptance Testing

Two batteries, serial numbers 39 and 40, were subjected to acceptance testing. These tests consisted of three charge-discharge tests at 60 amperes. Capacity data is presented in Table II for the results of the acceptance testing.

Charging was performed as follows:

- a) The open circuit voltage of the power supply was set at 2.60 volts per cell.
- b) The current limit was 15 amperes.
- c) The units were allowed to charge 15 to 18 hours at this constant potential.
- d) Toward the end of the charge the voltage rose to 2.6 volts per cell and the current tapered to one ampere or less.
- e) At this time the charge mode was changed to 10 amperes constant current for 2½ additional hours charge.

Discharge was performed at 60 amperes to a 4.95 end voltage.

The average voltage during discharge was 5.86 volts. Weight of each battery was 27.0 kilograms. Therefore, calculated energy density on cycle 2 is:

$$\text{S/N 39} \quad \frac{187 \times 5.86}{27} = 40.6 \text{ WH/kg}$$

$$\text{S/N 40} \quad \frac{180 \times 5.86}{27} = 39.1 \text{ WH/kg}$$

### 5.2 Conditioning Cycles

Testing included ten 100% depth of discharge conditioning cycles. The test method performed was the same as the procedure outlined in Section 5.1 above. Results are presented in Table III for serial numbers 39 and 40. Table IV presents conditioning cycle data for serial numbers 53 through 58.

Table II

## Acceptance Tests

S/N 39 &amp; 40

S/N 39

| <u>Cycle</u> | <u>Capacity</u> | <u>Charge</u><br><u>S. G.</u> | <u>End of</u><br><u>Charge Temp.</u> | <u>Discharge</u><br><u>S. G.</u> | <u>Average</u><br><u>Discharge Temp.</u> |
|--------------|-----------------|-------------------------------|--------------------------------------|----------------------------------|------------------------------------------|
| 1            | 183 AH          | 1.333                         | 46°C                                 | 1.167                            | 42°C                                     |
| 2            | 187 AH          | 1.322                         | 38°C                                 | 1.166                            | 43°C                                     |
| 3            | 180 AH          | 1.323                         | 40°C                                 | 1.176                            | 39°C                                     |

S/N 40

|   |        |       |      |       |      |
|---|--------|-------|------|-------|------|
| 1 | 178 AH | 1.335 | 47°C | 1.167 | 42°C |
| 2 | 180 AH | 1.320 | 39°C | 1.166 | 42°C |
| 3 | 175 AH | 1.319 | 42°C | 1.174 | 41°C |

Table III

Capacity During  
Conditioning Cycles

S/N 39 &amp; 40

| <u>Cycle</u> | <u>S/N 39</u>   |              | <u>S/N 40</u>   |              |
|--------------|-----------------|--------------|-----------------|--------------|
|              | <u>Capacity</u> | <u>Temp.</u> | <u>Capacity</u> | <u>Temp.</u> |
| 1            | 183 AH          | 42°C         | 178 AH          | 42°C         |
| 2            | 187 AH          | 43°C         | 180 AH          | 42°C         |
| 3            | 180 AH          | 40°C         | 175 AH          | 41°C         |
| 4            | 188 AH          | 44°C         | 181 AH          | 44°C         |
| 5            | 175 AH          | 35°C         | 171 AH          | 35°C         |
| 6            | 175 AH          | 36°C         | 170 AH          | 35°C         |
| 7            | 178 AH          | 38°C         | 175 AH          | 38°C         |
| 8            | 176 AH          | 37°C         | 172 AH          | 36°C         |
| 9            | 174 AH          | 36°C         | 170 AH          | 35°C         |
| 10           | 176 AH          | 38°C         | 173 AH          | 37°C         |



Table IV

Battery Capacity  
Conditioning Cycles  
Discharge Rate 60 Amperes

| <u>Cycle</u><br><u>No.</u> | <u>Serial Numbers</u> |           |           |           |           |           | <u>Avg. Battery</u><br><u>Discharge Temp.</u> |
|----------------------------|-----------------------|-----------|-----------|-----------|-----------|-----------|-----------------------------------------------|
|                            | <u>53</u>             | <u>54</u> | <u>55</u> | <u>56</u> | <u>57</u> | <u>58</u> |                                               |
| 1                          | 180 AH                | 186 AH    | 184 AH    | 170 AH    | 170 AH    | 187 AH    | 43°C                                          |
| 2                          | 181 AH                | 184 AH    | 183 AH    | 175 AH    | 173 AH    | 186 AH    | 43°C                                          |
| 3                          | 182 AH                | 185 AH    | 184 AH    | 176 AH    | 173 AH    | 186 AH    | 43°C                                          |
| 4                          | 182 AH                | 186 AH    | 184 AH    | 177 AH    | 175 AH    | 187 AH    | 45°C                                          |
| 5                          | 178 AH                | 182 AH    | 179 AH    | 175 AH    | 174 AH    | 185 AH    | 40°C                                          |
| 6                          | 177 AH                | 180 AH    | 178 AH    | 172 AH    | 171 AH    | 183 AH    | 37°C                                          |
| 7                          | 180 AH                | 184 AH    | 181 AH    | 177 AH    | 175 AH    | 185 AH    | 41°C                                          |
| 8                          | 182 AH                | 186 AH    | 183 AH    | 178 AH    | 176 AH    | 186 AH    | 43°C                                          |
| 9                          | 182 AH                | 185 AH    | 182 AH    | 177 AH    | 174 AH    | 187 AH    | 42°C                                          |
| 10                         | 183 AH                | 186 AH    | 183 AH    | 177 AH    | 175 AH    | 187 AH    | 43°C                                          |

Note: Ambient temperature was room temperature (approximately 18°C to 35°C).  
Battery temperature of 37°C to 45°C resulted from the overcharge  
portion of the recharge (paragraph 5.1 step e) above).

### 5.3 Performance Testing

Table V presents a summary of capacity, rate and temperatures for the performance test results. The temperatures listed are temperatures at the start of discharge. All charging was performed at room ambient. Charging was performed in accordance with the schedule listed in section 5.1 above.

In all cases for the  $25^{\circ}\text{C} \pm 3^{\circ}\text{C}$  discharges, batteries were above the allowable temperature range after charging. The batteries were allowed to stand open circuit until the temperature dropped within the allowable range. Typically the batteries were allowed to stand open circuit for 20 to 24 hours before discharge.

For discharges scheduled at the  $38^{\circ}\text{C} \pm 3^{\circ}\text{C}$  temperature, the batteries were discharged immediately after charge. The ten ampere constant current (step e) overcharge portion of the charge was used to heat the battery to the specified  $38^{\circ}\text{C} \pm 3^{\circ}\text{C}$  temperature.

Discharges at  $0^{\circ}\text{C} \pm 3^{\circ}\text{C}$  were conducted in an ice bath. After charge (at room ambient) the batteries were placed in an ice bath. Discharge was initiated when temperatures were in the  $0^{\circ}\text{C} \pm 3^{\circ}\text{C}$  range.

Figures 1 through 9 present voltage versus ampere hours capacity data. A representative discharge for each test condition is plotted. Also plotted is the computer performance prediction for each test condition.

### 5.4 Cycle Testing

The batteries were set to cycle at an 80% depth of discharge, that is,  $0.8 \times 180 \text{ AH}$  (the three hour rated capacity). The ampere-hour discharge each cycle therefore was 144 AH. At a 60 ampere constant current discharge this corresponds to a discharge period of 2 hours and 24 minutes.

Initially it was attempted to recharge the batteries using a constant potential charge at 2.60 volts per cell with a maximum current of 20 amperes.

Table V

Summary of  
Performance Test Results

| Test No. | Temp.* | Rate  | Cut-Off<br>Voltage | Capacity |        |
|----------|--------|-------|--------------------|----------|--------|
|          |        |       |                    | S/N 39   | S/N 40 |
| 1        | 25°C   | 40 A  | 5.00 V             | 191 AH   | 190 AH |
| 2        | 28°C   | 40 A  | 5.00 V             | 193 AH   | 192 AH |
| 3        | 24°C   | 60 A  | 4.95 V             | 174 AH   | 172 AH |
| 4        | 25°C   | 60 A  | 4.95 V             | 173 AH   | 170 AH |
| 5        | 25°C   | 130 A | 4.70 V             | 124 AH   | 124 AH |
| 6        | 25°C   | 130 A | 4.70 V             | 124 AH   | 122 AH |
| 7        | 37°C   | 40 A  | 5.00 V             | 200 AH   | 195 AH |
| 8        | 38°C   | 40 A  | 5.00 V             | 202 AH   | 198 AH |
| 9        | 41°C   | 60 A  | 4.95 V             | 184 AH   | 178 AH |
| 10       | 38°C   | 60 A  | 4.95 V             | 182 AH   | 177 AH |
| 11       | 38°C   | 130 A | 4.70 V             | 130 AH   | 128 AH |
| 12       | 35°C   | 130 A | 4.70 V             | 128 AH   | 125 AH |
| 13       | 0°C    | 40 A  | 5.00 V             | 140 AH   | 132 AH |
| 14       | +2°C   | 40 A  | 5.00 V             | 142 AH   | 135 AH |
| 15       | 0°C    | 60 A  | 4.80 V             | 117 AH   | 115 AH |
| 16       | +3°C   | 60 A  | 4.80 V             | 122 AH   | 118 AH |
| 17       | 0°C    | 130 A | 4.60 V             | 90 AH    | 87 AH  |
| 18       | +3°C   | 130 A | 4.60 V             | 94 AH    | 90 AH  |

\* Temperature at start of discharge

Figure 2  
PERFORMANCE TESTS  
TASK 5D  
VOLTAGE vs. CAPACITY  
DISCHARGE RATE: 60 AMPS  
TEMPERATURE: 25°C  
SERIAL NUMBER: 40

10

CAPACITY - AMPERE HOURS

Figure No. 4  
PERFORMANCE TESTS  
TASK 5D

VOLTAGE vs. CAPACITY  
DISCHARGE RATE: 40 Amps  
TEMPERATURE: 37°C  
SERIAL NUMBER: 39

Figure No. 6  
PERFORMANCE TESTS

TASK 5D  
VOLTAGE vs. CAPACITY  
DISCHARGE RATE: 130 Amps  
TEMPERATURE: 35°C  
SERIAL NUMBER: 40

VOLTAGE

Figure No. 7

PERFORMANCE TESTS

TASK 5D

VOLTAGE vs. CAPACITY

DISCHARGE RATE: 40 Amps

TEMPERATURE: 0°C

SERIAL NUMBER: 40

6.0

5.5

5.0

4.5

Actual Performance

Computer Prediction

25

50

75

100

125

150

175

200

CAPACITY - AMPERE HOURS



Figure No. 8  
PERFORMANCE TEST  
TASK 5D  
VOLTAGE vs. CAPACITY  
DISCHARGE RATE: 60 Amps  
TEMPERATURE: 0°C  
SERIAL NUMBER: 39

VOLTAGE

6.0

5.5

5.0

4.5

25

50

75

100

125

150

175

200

CAPACITY - AMPERE HOURS

Actual Performance

Computer Prediction

Figure No. 9  
PERFORMANCE TESTS  
TASK 5D  
VOLTAGE vs. CAPACITY  
DISCHARGE RATE: 130 Amps  
TEMPERATURE: 0°C  
SERIAL NUMBER: 40

VOLTAGE

6.0

5.5

5.0

4.5

25

50

75

100

125

150

175

200

CAPACITY - AMPERE HOURS

Actual Performance

Computer Prediction

The charge period was 9 hours and 36 minutes. After one week of cycling it was noted that the end of discharge voltage during cycling was declining noticeably. At this time the recharge method was changed to constant current recharge at 16.5 amperes. This corresponds to an ampere hour return of 110%. The end of discharge voltage during cycling was stabilized by carefully adjusting the ampere charging rate. Cycling was continued using this recharge procedure at two charge-discharge cycles each day.

The batteries were cycled in room air ambient. Water baths were not used, nor any other active cooling method. Battery temperature was significantly higher than room temperature. Measured electrolyte temperatures varied between 40°C and 50°C. This method of cycling is similar to that described in federal specification for Battery, Storage (Lead-Acid Industrial Portable Service) W-B-00133B, paragraph 4.5.4. This paragraph allows a maximum electrolyte temperature of 125°F (52°C).

Approximately once per month the batteries were removed from cycle test to measure full capacity. This corresponds to a capacity test approximately each 50 cycles. Table VI presents the capacity of each of the six batteries measured during these full depth discharges. Table VII presents cubic centimeters water addition per cell during cycle testing. During the first 203 cycles a capacity and voltage fade is indicated. Figure 10 presents the change in the voltage curve as a function of cycles. Figure 10 presents average capacity versus cycle number.

During cycle testing there was several instances of equipment malfunction or power failure. These occurred on cycle 30, cycle panel off for two days; cycle 46, relay malfunction; cycles 131 and 143, power outage; and cycle 210, lightning damage. The lightning storm caused the panel to continue charging throughout cycle 210, omitting the discharge. This amount of over-charge caused

Table VI

Summary  
 Cycle Test Results  
 Capacity during 100% DOD

| <u>Cycle No.</u> | <u>53</u> | <u>54</u> | <u>55</u> | <u>56</u> | <u>57</u> | <u>58</u> |
|------------------|-----------|-----------|-----------|-----------|-----------|-----------|
| 1                | 180 AH    | 186 AH    | 184 AH    | 170 AH    | 170 AH    | 187 AH    |
| 10               | 183 AH    | 186 AH    | 183 AH    | 177 AH    | 175 AH    | 187 AH    |
| 46               | 170 AH    | 171 AH    | 170 AH    | 166 AH    | 165 AH    | 177 AH    |
| 47               | 175 AH    | 175 AH    | 175 AH    | 172 AH    | 170 AH    | 178 AH    |
| 98               | 171 AH    | 171 AH    | 171 AH    | 167 AH    | 167 AH    | 175 AH    |
| 99               | 175 AH    | 175 AH    | 175 AH    | 171 AH    | 170 AH    | 178 AH    |
| 150              | 167 AH    | 169 AH    | 167 AH    | 163 AH    | 162 AH    | 170 AH    |
| 151              | 169 AH    | 170 AH    | 168 AH    | 166 AH    | 164 AH    | 173 AH    |
| 202              | 162 AH    | 163 AH    | 163 AH    | 160 AH    | 160 AH    | 167 AH    |
| 203              | 164 AH    | 164 AH    | 165 AH    | 162 AH    | 161 AH    | 167 AH    |

Table VII

Water Addition  
During Cycle Testing

| <u>Cycle No.</u> | <u>Cubic Centimeters</u><br><u>Water Added Per Cell</u> |
|------------------|---------------------------------------------------------|
| #15              | 50 cc                                                   |
| #37              | 25 cc                                                   |
| #49              | 75 cc                                                   |
| #63              | 50 cc                                                   |
| #84              | 50 cc                                                   |
| #105             | 75 cc                                                   |
| #115             | 50 cc                                                   |
| #145             | 75 cc                                                   |
| #161             | 75 cc                                                   |
| #171             | 50 cc                                                   |
| #175             | 25 cc                                                   |

Figure No. 11

CYCLE TESTS

CAPACITY vs. CYCLES

DISCHARGE RATE: 60 Amps

TEMPERATURE: 35°C - 45°C

AVERAGE CAPACITY: 6 units

CAPACITY

180

170

160

150

140

50

100

150

200

250

300

350

400

Cycle No.

severe over-heating of the batteries. After this charging malfunction the batteries would not deliver 144 AH and the test was terminated.

#### 6.0 Observations and Conclusions

The test series describes the performance capabilities of the battery for various discharge rates and temperatures. These data were used to improve the computer performance prediction model. Figures 1 through 9 show a comparison between actual voltage and the computer performance prediction.

The battery is capable of providing 39.6 WH/kg (18 WH/lb.) energy density at 60 amperes. The battery can also provide 44.4 WH/kg (20.2 WH/lb.) energy density at 40 amperes.

A capacity fade and voltage fade was observed during cycling. The capacity and voltage fade may be inherent in the battery as new plate manufacturing technology was used. However, it is believed that these phenomena may be attributable to the high temperature of the batteries during test. As described in Section 5.4 the batteries varied between 40°C and 50°C during the cycle test.

It is planned during 1980 to build additional batteries of this type and to perform life cycle testing on these batteries. It is planned to use a water bath for temperature control in order to determine temperature effects on voltage and capacity fade.

Task 5I

Summary Report  
On All Performance And  
Life Cycle Test Results

Battery Adaptation Program  
Subcontract No. 425-85111-7

Summary Report  
June 16, 1980

Eagle-Picher Industries, Inc.  
Electronics Division  
Commercial Products Department  
Seneca, Missouri 64865

Figure 4.4.1-5. Final Motor/Generator Performance Characteristics



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DEPARTMENT OF ENERGY  
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