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**Task 1
Tradeoff Studies**

MASTER

**ELECTRIC-POWERED PASSENGER
VEHICLE DESIGN STUDY PROGRAM**

76-13133

September 16, 1976

Prepared for

The Energy Research and Development Administration
Oakland, California

Contract E(04-3)-1213

AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

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Task 1 Tradeoff Studies

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September 16, 1976

Prepared by

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Program Manager

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SECTION 1

INTRODUCTION AND SUMMARY

INTRODUCTION

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

SUMMARY

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

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

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

CONCLUSIONS

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

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

SECTION 2

POWER SYSTEM BASELINE DESCRIPTION AND PERFORMANCE

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

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

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

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

Results are described in subsequent sections of this report.

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

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Figure 2-1. Baseline Schematic Power System

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SIDE VIEW

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UNLESS OTHERWISE SPECIFIED: BURR CONTROL PER SC683 STD INTERPRETATIONS PER P188 IDENTIFICATION MARKING PER MCT6			PREPARED BY	DSW 9-8-76	ELECTRIC VEHICLE TRANSMISSION (FRONT WHEEL DRIVE)	
MATERIAL			CHK			
FINISH PROCESS			DESIGN			
HEAT TREATMENT			VALUE ENGR			
			MATL			
			STRESS		SIZE CODE IDENT NO DWG NO	
			DESIGN SUPERVISOR	PROJECT ENGINEER	D 70210 SK-71642	
			GOVERNMENT APPD		SCALE 1/2 SHEET 1 OF 2	
REQD	NEXT ASSY	USED QN	APPLICATION			

SK-71642 5

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UNLESS OTHERWISE SPECIFIED: BURR CONTROL PER SC653 STD INTERPRETATIONS PER P155 IDENTIFICATION MARKING PER MC10		PREPARED BY DSW 9-9-76		ELECTRIC VEHICLE TRANSMISSION (REAR WHEEL DRIVE)	
MATERIAL		DESIGN		SCALE 1/2	
FINISH PROCESS		VALUE ENGR		D 70210 SK-71641	
HEAT TREATMENT		MATERIAL		LARGE IDENT NO	
RECD		APPRO		LUMP NO	
NEXT ASSY		DESIGN SUPERVISOR		PROJECT ENGINEER	
USED ON		GOVERNMENT APPD		SHEET 1 OF 2	
APPLICATION					

SK-71641

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

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

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

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

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

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

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

TABLE 2-1

MAJOR CHARACTERISTICS OF THE PRELIMINARY
BASELINE SYSTEM

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

Figure 2-2. Torque Characteristics of the Drive System

Figure 2-3. Power Characteristics of the Drive System

Figure 2-4. Full Power Vehicle Acceleration

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

S-8921

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

TABLE 2-2

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

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

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

SECTION 3

COMPUTER SIMULATION STUDIES

INTRODUCTION

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

Simulation models and program description

Simulation model inputs

Simulation model outputs

System performance studies

SIMULATION MODELS AND PROGRAM DESCRIPTION

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

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

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

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

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

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

SIMULATION MODEL INPUTS

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

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

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

Loss Models

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

Aerodynamic Drag

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

$$F_{aero} = C_D A \frac{1}{2} \rho V^2$$

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

A is the frontal area.

ρ is the density of air

V is the velocity of the vehicle

Rolling Resistance

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

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

where W_V is the gross weight of the vehicle

V is the velocity and

μ is the tire rolling resistance coefficient

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

TABLE 3-1

AERODYNAMIC COMPARISON OF SEVERAL COMPACT CARS

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

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

Motor and Generator Losses

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

Motor armature

Generator armature

Motor field

Generator field

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

1. Motor and Generator Armature Power Losses

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

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

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

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

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

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

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

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

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

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

2. Controller Power Losses

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

3. Motor and Generator Field Losses

The field losses are limited to copper losses as follows:

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

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

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

Motor and Generator Gears

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

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

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

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

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

Planetary Differential

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

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

where

T_F is the torque of the flywheel, and

N_F is the flywheel speed

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

Final Drive Differential

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

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

where

T_A is the torque at the axle, and

N_A is the speed of the axle

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

Flywheel Losses

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

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

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

N_F is flywheel speed, and

N_{FMAX} is maximum flywheel speed

TABLE 3-2

FLYWHEEL ASSEMBLY FRICTION LOSSES (AT 100 PERCENT RPM)

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

SIMULATION MODEL OUTPUTS

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

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

Battery Weight

The calculation of the required battery weight is as follows:

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

Overall System Efficiency

Overall system efficiency is determined as follows.

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

Energy Recovery Efficiency

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

TABLE 3-3

SUMMARY OF ENERGY LOSSES
FOR DRIVING CYCLE IN W-HR

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

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

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

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

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

$$\eta_R = 0.71$$

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

SYSTEM PERFORMANCE STUDIES

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

Tradeoffs on Propulsion System

1. Evaluation Criteria

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

2. Generator/Motor

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

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

S-8914

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

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

3. Flywheel

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

4. Final Drive Ratio

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

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

Loss Sensitivity Studies

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

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

S-8910

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

S-8916

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

s-8917

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

s-8898

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

s-8897

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

S-8901

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

S-8902

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

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

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

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

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

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

S-8900

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

SECTION 4

COMPONENT AND POWER SYSTEM TRADEOFF STUDIES

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

FLYWHEEL DESIGN TRADEOFFS

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

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

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

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

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

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

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

TABLE 4-1
FLYWHEEL CHARACTERISTICS

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

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

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

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

S-8913

Figure 4-4. Rotational Speed for Different Flywheel Diameter

MOTOR AND GENERATOR DESIGN TRADEOFFS

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

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

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

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

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

BATTERY DESIGN TRADEOFFS

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

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PERFORMANCE PLOT ELECTRIC CAR MOTOR

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

Figure 4-5. Baseline Motor/Generator Characteristics

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

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

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

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

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

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

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

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

TABLE 4-2

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

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

*Corresponding to 70 to 80 percent depth of discharge

TABLE 4-3

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

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

*Corresponding to 70 to 80 percent depth of discharge

TABLE 4-4

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

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

TABLE 4-5

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

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

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

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

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Eagle-Picher Industries, Inc.
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Figure 4-7. Nickel-Iron Electric Vehicle Batteries, Power Density Versus Energy Density

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

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

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

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

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

GEARBOX

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

ALTERNATE TRANSMISSION CONCEPTS

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

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

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

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

ALTERNATE ELECTRICAL TRANSMISSION

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

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

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

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

CONTROLLER DESIGN TRADEOFFS

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

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

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

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

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

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

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

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

SECTION 5

VEHICLE DESIGN TRADEOFFS

INTRODUCTION

Vehicle design tradeoffs conducted under this task include:

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

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

VEHICLE GROSS WEIGHT, EMPTY WEIGHT, AND RANGE

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

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

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

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

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

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

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TABLE 5-1
VEHICLE EMPTY WEIGHTS
WITHOUT ENGINE AND FUEL SYSTEM

	1975 Vega Weight*, lb (Curb Weight, 2487 lb)	1976 Chevette Weight*, lb (Curb Weight, 1997 lb)	Detroit Average Weight* for 1980 to 1985, lb (Curb Weight, 2400 lb)	Target Objective Weight, lb (Curb Weight, 2400 lb)
Frame and body structure	432	347	440	316
Body and panels	858	670	535	476
Drive system	110	97	116	96
Suspension system	138	124	142	110
Brake system	150	118	144	106
Steering system	37	35	39	33
Wheels and tires	160	145	174	139
	<hr/> 1885	<hr/> 1536	<hr/> 1590	<hr/> 1276

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

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

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

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

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

VEHICLE LAYOUT STUDIES OF THE POWER SYSTEM AND BATTERY LOCATIONS

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

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

S-8920

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

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SIZE		CODE IDENT NO.		DRAWING NUMBER		REV.	
				PRE 3008			

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

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

OPERATIONAL TRADEOFFS AND CHARACTERISTICS

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

Energy Loss During Parking

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

Startup After Extended Parking

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

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

Braking Characteristics

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

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

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

S-8924

Figure 5-5. Split Transmission Schematic

SECTION 6

ECONOMIC EVALUATION/ANALYSIS MODEL

INTRODUCTION

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

OBJECTIVES

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

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

MODEL STRUCTURE

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

COST ESTIMATING RELATIONSHIPS (CER's)

RDT&E Cost

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

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

The general calculational algorithm for RDT&E costs is:

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

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

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

Cost of Manufacturing R&D Vehicles =

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

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

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

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

where

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

X = cumulative output

a = cost of first item

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

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

Acquisition Cost

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

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

*References presented at end of section.

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

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

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

(a) Cost of Financing

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

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

where P_s = sticker (sales) price, \$

t = financing term, yr

i = financing interest rate, decimal

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

(b) Salvage Value

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

2. Total Acquisition Cost

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

Ownership Cost

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

TABLE 6-1

VEHICLE COST BREAKDOWN CATEGORIES

Item	Category	Determination Method	Units
1	Frame and body structure (unitized frame)	$C_{FB} = 0.441$, where C (in general) is in \$/lb or \$/mi and subscripts refer to appropriate individual components	\$/lb
2	Removable panels (2 doors)	$CRP2D = 2.256 - 0.0000403 W_c$, where W_c = vehicle curb weight (lb)	\$/lb
3	Removable panels (4 doors)	$CRP4D = 2.292 - 0.0000310 W_c$	\$/lb
4	Basic body (includes interior and exterior trim, accessories, lights, glass, heater, wiper/washer, crash and panel pads)	$C_{BB} = 1.356$	\$/lb
5	Drive system, rear drive	$C_{DSR} = 0.997 - 0.0000719 W_c$	\$/lb
6	Drive system, front drive	$C_{DSF} = 1.0246 - 0.0000639 W_c$	\$/lb
7	Suspension system, coil/leaf	$CSUS c/l = 0.654 - 0.0000170 W_c$	\$/lb
8	Suspension system, coil/coil	$CSUS c/c = 0.706 - 0.0000179 W_c$	\$/lb
9	Brake system, drum/drum	$C_{BR DRM/DRM} = 0.536 - 0.0000213 W_c$	\$/lb
10	Brake system, disk/drum	$C_{BR DSK/DRM} = 0.676 - 0.0000379 W_c$	\$/lb
11	Steering system, front drive, ball and rack and pinion	$C_{STR,BALL,R\&P,F} = 1.489 - 0.0000379 W_c$	\$/lb
12	Steering system, rear drive, ball and rack and pinion	$C_{STR,BALL,R\&P,R} = 1.522 - 0.0000368 W_c$	\$/lb
13	Wheels, 4 or 5	$C_{WH} (4),(5) = 0.109 + 0.000109 W_c$	\$/lb
14	Tires, 5 bias ply	$C_{TIRE BIA} (5) = 0.404$	\$/lb
15	Tires, 5 radial ply	$C_{TIRE RAD} (5) = 0.572$	\$/lb
16	Tires, 4 run flat	$C_{TIRE RUN FLAT} (4) = 0.850 + 0.000115 W_c$	\$/lb
17	Restraints	$C_{RES} = 2.042$	\$/lb
18	Air Conditioning	$C_{A/C} = 4.427$	\$/lb
19	Propulsion System, Motor	$C_{MOT} = 3.85$	\$/lb
20	Generator	$C_{GEN} = 3.85$	\$/lb
21	Flywheel	$C_{FW} = 4.90$	\$/lb
22	Flywheel Housing	$C_{FWH} = 1.44$	\$/lb
23	Gearbox	$C_{GB} = 1.92$	\$/lb
24	Controller	$C_{CONTR} = 12.02$	\$/kw
25	Battery Charger	$C_{BC} = 12.02$	\$/kw
26	Battery	$C_{BAT} = 0.55$	\$/lb
27	Vehicle Assembly	$C_{ASSEM} = 0.0668$	\$/lb curb wt
28	Total Manufacturing Cost (includes variable and fixed material and labor costs; engineering tooling costs; and quality control costs)	$C_{MANF, TOT} = \sum \text{Items 1 through 27}$	\$
29	Markup Factor (includes factory profit; warranty costs; marketing costs; facilities equipment costs; corporate profit; and dealer markups)	$MF = 0.0002918$ (manufacturing cost) + 1.4083	\$
30	Sticker Price (suggested retail selling price; does not include transportation, dealer preparation, or sales taxes)	$P_c = (MF)(C_{MANF, TOT})$	\$

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

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

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

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

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

TABLE 6-2

OWNERSHIP COST BREAKDOWN

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

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

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

TOTAL OWNERSHIP COST

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

TOTAL ELECTRIC VEHICLE SYSTEM (LIFETIME) COST

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

REFERENCES

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

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

PAGE 1

RUN NO. 1

DATE 09 SEP 76

VEHICLE DESIGN I.D. 1

COST ELEMENT	HOURS (HRS)	COMPOSITE RATE (\$/HR)	TOTAL COST(\$)	TOTAL COST INCLUDING CONTRACTOR PROFIT
INITIAL ENGINEERING	100000.00	25.00	2525000.00	2777500.00
DEVELOPMENT TOOLING	3000.00	20.00	160000.00	176000.00
DEVELOPMENT SUPPORT	100000.00	21.00	2150000.00	2364999.90
MANUFACTURE OF R+D VEHICLES			98495.07	108344.58
TEST FACILITIES			100000.00	110000.00
TEST OPERATIONS	10000.00	22.50	230000.00	253000.00
TOTAL, RDT+E			5263495.10	5789844.30

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

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EPDA ELECTRIC VEHICLE PROGRAM PHASE I AUGUST 1976

VEHICLE ACQUISITION COST SUMMARY

(1975 \$)

ANNUAL PRODUCTION RATE 100000, VEHICLES/YR

PAGE 2

RUN NO. 1

DATE 09 SEP 76

VEHICLE DESIGN I.D. 1

VEHICLE COMPONENT/SYSTEM	WEIGHT (LBS)	COST (\$)	COST/LB (\$/LB)	% OF STICKER PRICE
FRAME + BODY STRUCTURE (UNITIZED FRAME)	280.00	123.48	.44	1.93
REMOVABLE PANELS (2 DOORS)	220.00	475.04	2.16	7.42
BASIC BODY	210.00	284.76	1.36	4.45
DRIVE SYSTEM (FRONT DR)	96.00	83.64	.87	1.31
SUSPENSION SYSTEM (COIL/COIL)	110.00	67.45	.61	1.05
BRAKE SYSTEM (DRK/DRK)	106.00	66.21	.62	1.03
STEERING SYSTEM (RACK + PINION, FRONT DR)	33.00	46.14	1.40	.72
WHEELS (4)	69.00	25.97	.37	.40
TIRES (4 RUN FLAT)	70.00	78.82	1.13	1.23
RESTRAINTS (4)	6.00	12.25	2.04	.19
AIR CONDITIONING	0.00	.00	.00	.00
PROPULSION SYSTEM				
MOTOR	80.00	308.00	3.85	4.81
GENERATOR	50.00	192.50	3.85	3.01
FLYWHEEL	25.00	122.50	4.90	1.91
FLYWHEEL HOUSING	25.00	36.00	1.44	.56
GEAR BOX	20.00	38.40	1.92	.60
CONTROLLER	10.00	120.20	12.02	1.88
BATTERY CHARGER	10.00	120.20	12.02	1.88
SYSTEM INSTALLATION	20.00	.00	.00	.00
BATTERY	900.00	495.00	.55	7.73
BATTERY INSTALLATION	60.00	.00	.00	.00
TOTAL PROPULSION SYSTEM	240.00	937.80	3.91	14.65
TOTAL BATTERY SYSTEM	960.00	495.00	.52	7.73
VEHICLE ASSEMBLY		160.32	.07	2.50
VEHICLE CURR WEIGHT	2400.00		.00	.00
TOTAL MANUFACTURING COST		2856.48	1.19	44.61
FACTORY + DEALER MARKUPS		3547.00	1.48	58.39
SUGGESTED RETAIL STICKER PRICE		6403.48	2.67	100.00

Figure 6-1. Page 2 of 4

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

PAGE 3

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

AVERAGE ANNUAL MILEAGE 10000. (MI)

TOTAL VEHICLE LIFE 10.00 (YRS)

COMPONENT	SCHEDULED MAINT INTERVAL (MI)	MEAN TIME TO REPAIR/ REPLACE/INSPECT (HRS)	LABOR RATE (\$/HR)	PARTS COST/MAINT INTERVAL (\$)	LIFETIME TOTAL COST (\$)	COST/MILE (\$/MI)
1. MOTOR						
BRUSHES	25000.0	.50	12.00	10.00	48.00	.000480
BEARINGS	100000.0	1.50	12.00	20.00	.00	.000000
2. GENERATOR						
BRUSHES	25000.0	.50	12.00	10.00	48.00	.000480
BEARINGS	100000.0	1.50	12.00	18.00	.00	.000000
3. FLYWHEEL						
SEALS	50000.0	1.00	12.00	10.00	22.00	.000220
BEARINGS	50000.0	2.00	12.00	25.00	49.00	.000490
4. GEARBOX						
BEARINGS	50000.0	2.00	12.00	25.00	49.00	.000490
5. CONTROLLER						
	100000.0	1.25	12.00	15.00	.00	.000000
6. BATTERY CHARGER						
	100000.0	1.30	12.00	12.50	.00	.000000
7. BATTERY						
	800.0	.25	12.00	.10	384.40	.003844
TOTAL MAINTENANCE COST					600.40	.006004

Figure 6-1. Page 3 of 4

AIRESEARCH MANUFACTURING COMPANY
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ERDA ELECTRIC VEHICLE PROGRAM PHASE I AUGUST 1976

TOTAL SYSTEM COST SUMMARY

(1975 \$)

PAGE 4

RUN NO. 1

DATE 09 SEP 76

VEHICLE DESIGN I.D. 1

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

SYSTEM COST CATEGORY	LIFETIME COST (\$)	ANNUAL COST (\$/YR)	LIFETIME COST PER VEHICLE MILE (\$/MI)	% OF TOTAL
RDT+E	11.58	1.16	.00012	.1
ACQUISITION				
PURCHASE PRICE	6403.48	640.35	.06403	39.3
FINANCE CHARGE AT INTEREST RATE 12.00%				
AND TERM 40.0 MOS	1738.04	173.80	.01738	10.7
SALVAGE VALUE 7.0% OF ACQ.	448.24	44.82	.00448	2.8
TOTAL ACQUISITION	7693.28	769.33	.07693	47.3
OWNERSHIP				
MAINTENANCE + REPAIR				
PROPULSION + BATTERY SYSTEMS	600.40	60.04	.00600	3.7
OTHER	937.20	93.72	.00937	5.8
TOTAL M+R	1537.60	153.76	.01538	9.4
TIRE REPLACEMENT	321.65	32.16	.00322	2.0
TIRE LIFE 30000.0 MI.				
ACCESSORIES	274.00	27.40	.00274	1.7
INSURANCE	3750.00	375.00	.03750	23.0
REGISTRATION + LICENSE	170.00	17.00	.00170	1.0
GARAGING, PARKING + TOLLS	7.06	.71	.00007	.0
ENERGY	1500.00	150.00	.01500	9.2
RATE .050 \$/KWH				
BATTERY REPLACEMENT				
BATTERY	1021.18	102.12	.01021	6.3
FINANCE CHARGE AT INT. RATE 12.00%				
AND TERM 12.0 MOS	122.54	12.25	.00123	.8
TOTAL BATTERY REPLACEMENT	1143.72	114.37	.01144	7.0
TOTAL OWNERSHIP	8574.63	857.46	.08575	52.7
TOTAL SYSTEM COST	16279.49	1627.95	.16279	100.0

Figure 6-1. Page 4 of 4

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Page 6-13

SECTION 7

RELIABILITY AND SAFETY

INTRODUCTION

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

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

RELIABILITY ANALYSIS

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

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

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S-8922

Figure 7-1. Safety/Fire Fault Tree Analysis

s-8888

Figure 7-2. Powertrain System Fault Tree Analysis

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

RELIABILITY PREDICTION AND REPORTING

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

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

where N = sample size and
X = number of failures

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

s-8890

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

S-8889

Figure 7-4. Brake System Fault Tree Analysis

S-8895

Figure 7-5. Body System Fault Tree Analysis

DESIGN VERIFICATION AND STATUS

SYSTEM <u>12.000</u>		ASSEMBLY <u>LIGHTING</u>		PROGRAM <u>ELECTRIC CAR</u>				
SUBSYSTEM <u>12.200</u>		COMPONENT <u>EXTERIOR LIGHT</u>		DATE <u>8/12/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Side Turn Signal Lamp Spec.	SAE J514			20		P1.0 = .83		
Turn Signal Flasher	SAE J590 Life & Performance, 36 hours at 75°F J1055			10		P1.0 = .67		
Hazard Lamp Flasher	SAE J545 Life & Performance, 36 hours at 75°F J1056			10		P1.0 = .67		
Hazard Lamp Switch	SAE J510 Life Test 10,000 Cycles			20		P1.0 = .83		
Parking Lamp Spec.	SAE J522			20		P1.0 = .83		
Back-up Lamp Spec.	SAE J593			20		P1.0 = .83		
Back-up Lamp Switch	SAE J1076 1 Hour Each 75°, 225° and -25°F -- 10 Cycles Each Temperature			20		P1.0 = .83		

All Lighting Tests at 95% Confidence

Figure 7-6. Sample Design Verification and Status Form

APPENDIX

COMPONENT RELIABILITY PREDICTION AND REPORTING INDEX AND FORMS

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SYSTEM INDEX FOR THE COMPONENT
DESIGN VERIFICATION AND STATUS RECORDS

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

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

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

DESIGN VERIFICATION AND STATUS

SYSTEM <u>00.000</u>		ASSEMBLY <u>Vehicle</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE _____				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Electric Car	SAE J 227 Durability Equivalent to 100,000 Customer Miles			10		P.S = .54

[illegible]

ENGINE TEST AT 95 PERCENT CONFIDENCE LEVEL .

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DESIGN VERIFICATION AND STATUS

SYSTEM 01.100		ASSEMBLY Flywheel		PROGRAM Electric Car				
SUBSYSTEM		COMPONENT		DATE 8-17-76				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Flywheel Fatigue	Equivalent to 200,000 Customer Miles or 8,000 hr			10		P1.0 = .68		
Pump Fatigue	Equivalent to 200,000 Customer Miles			10		P1.0 = .68		
Seals	Equivalent to 50,000 Customer Miles			10		P1.0 = .68		
Bearings	Equivalent to 50,000 Customer Miles			10		P1.0 = .68		

ENGINE TEST AT 95 PERCENT CONFIDENCE

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OF CALIFORNIA

DESIGN VERIFICATION AND STATUS

SYSTEM <u>02.000</u>		ASSEMBLY <u>Traction Motor</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE _____				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Fatigue	Equivalent to 200,000 Customer Miles							
Brush Fatigue	Equivalent to 25,000 Customer Miles			50		P1.0 = .93		

MOTOR TEST AT 95 PERCENT CONFIDENCE LEVEL

SYSTEM <u>03.000</u>		ASSEMBLY <u>Transmission</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE _____				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Transmission Performance	SAE J 651 J 985			3				
Fatigue	Equivalent to 100,000 Customer Miles			5		P1.0 = .63		
Gear Noise				10		P1.0 = .79		

TRANSMISSION TESTS AT 80 PERCENT CONFIDENCE LEVEL

[illegible]

DRIVELINE TEST AT 95 PERCENT CONFIDENCE LEVEL

SYSTEM		ASSEMBLY		PROGRAM					
SUBSYSTEM		COMPONENT		DATE		8717775			
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT	
				REQ.	TESTED	REQ.	ACTUAL		
Fatigue Noise	Equivalent to 100,0C0 Customer Miles			5		P1.0 = .63			

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[illegible]

STEERING COLUMN TEST AT 95 PERCENT CONFIDENCE LEVEL

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DESIGN VERIFICATION AND STATUS

SYSTEM 06.203		ASSEMBLY Steering Gear		PROGRAM Electric Gear				
SUBSYSTEM		COMPONENT		DATE 8/17/76				
TEST NAME	ACCEPTANCE CRITER A	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Sector Shaft Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Housing	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		

STEERING TEST AT 95 PERCENT CONFIDENCE LEVEL

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DESIGN VERIFICATION AND STATUS

SYSTEM 05.300		ASSEMBLY Steering Linkage		PROGRAM Electric Car				
SUBSYSTEM		COMPONENT		DATE 8/17/76				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Center-link Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Tierod Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Pitman Arm Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Idler Arm Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		

STEERING TESTS AT 95 PERCENT CONFIDENCE LEVEL

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DESIGN VERIFICATION AND STATUS

SYSTEM <u>07.100</u>		ASSEMBLY <u>Front Suspension</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8/17/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Spring Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Ball Joint Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Drag Strut Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
"A" Arm Fatigue	Equivalent to 200,000 Customer Miles			20		P1.0 = .82		
Shock Fatigue	Equivalent to 20,000 Customer Miles			10		P.3 = .43		
Spindle Fatigue	Equivalent to 200,000 Customer Miles			30		P1.0 = .88		

SUSPENSION TESTS AT 95 PERCENT CONFIDENCE LEVEL

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DESIGN VERIFICATION AND STATUS

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

SUSPENSION TEST AT 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>08,000</u>		ASSEMBLY <u>Brake Assembly</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8/12/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS:	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Fluid Test	SAE J 291 J 1702 J 1703 FMVSS 116							
Master Cylinder	SAE J 66			10		P1.0 = .68		
Brake Tubes and Lines	SAE J 1401 J 1043			100		P1.0 = .96		
Brake Lining Test	SAE J 160 J 379.380 J 660,866 J 998,840			100		P1.0 = .96		
Brake Tube and Line	Performance FMVSS 106			100		P1.0 = .96		

ALL BRAKE SYSTEM COMPONENTS AT 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM 8.200		ASSEMBLY Brake Assembly		PROGRAM Electric Car				
SUBSYSTEM		COMPONENT		DATE 8/12/76				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Service Brake Test	SAE J 658 J 135 J 299 J 843 J 345 J 229 J 937			100		P1.0 = .96		
Service Brake Performance	FMVSS 105			3		P1.0 = .52		

ALL BRAKE SYSTEM COMPONENTS AT 95 PERCENT CONFIDENCE

SYSTEM <u>8.200</u>		ASSEMBLY <u>Service Brake Assembly</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM <u>8.210</u>		COMPONENT <u>Front Brake</u>		DATE <u>8-12-76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Disc Test	SAE J 02 Seal Test Life			100		P1.0 = .96		

ALL BRAKE SYSTEM COMPONENTS AT 95 PERCENT CONFIDENCE LEVEL

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DESIGN VERIFICATION AND STATUS

SYSTEM 08.200		ASSEMBLY Service Brake Assembly		PROGRAM Electric Car				
SUBSYSTEM 03.220		COMPONENT Rear Brake		DATE 8-12-76				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Brake Drum Test	SAE J 65 Seal Life J 101 Wheel Cylinder			100		P1.0 = .96		

ALL BRAKE SYSTEM COMPONENTS AT 95 PERCENT CONFIDENCE

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

ALL BRAKE COMPONENTS AT 95 PERCENT CONFIDENCE LEVEL

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DESIGN VERIFICATION AND STATUS

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

ALL BRAKE COMPONENTS AT 95 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>09.000</u>		ASSEMBLY <u>Controller</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE _____				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Controller Fatigue	Equivalent to 200,000 Customer Miles or 6,000 MTBF			6		P1 = .60		

CONTROLLER TEST AT 95 PERCENT CONFIDENCE LEVEL

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DESIGN VERIFICATION AND STATUS

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

ACCELERATOR CONTROL TEST AT 95 PERCENT CONFIDENCE LIMITS

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DESIGN VERIFICATION AND STATUS

SYSTEM <u>11.000</u>		ASSEMBLY <u>Elect. Dist.</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8-12-76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
EMI RFI	SAE J 551 J 1113 Canada FMVSS			5		P1.0 = .55		
Fuses Circuit Breaker Life	SAE J 553 J 554 6 hr Vibration 4 hr Life 50,000 Cycles			100		P.6 = .53		
Wiring, P.C.	SAE J 556 SAE J 771 Life and Durability Test			10		P.80 = .55		
On-Off Switch Life	SAE J 259 Life Test, 1 hr Each 75°F, 165°F, -5°F -25°F, 0°F 10 Cycles Each Temp			10		P.80 = .55		

FMVSS TEST AT 90 PERCENT CONFIDENCE

DESIGN VERIFICATION AND STATUS

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

BATTERY TEST AT 95 PERCENT CONFIDENCE

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DESIGN VERIFICATION AND STATUS

SYSTEM <u>12.000</u>		ASSEMBLY <u>Lighting</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM <u>12.200</u>		COMPONENT <u>Exterior Light</u>		DATE <u>0/12/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Lamp Retention	SAE J 567 Socket Load			50		P1.0 = .93		
Lamp Connector	SAE J 856							
Lamp Switch Life	SAE J 253 J 564 Life Test 100,000 Cycles			20		P1.0 = .89		
Lamp Spec.	SAE J 571,760 573,1049 575,256 576, 29 578,577 579, 32 580,583 582 FMVSS -108			50		P1.0 = .93		
Stop and Tail Lamp Spec.	SAE J 585 J 586			20		P1.0 = .89		
Stop Switch	SAE J 249 Life 250,000 Cycles			20		P1.0 = .89		
License Plate Lamp Spec.	SAE J 587			20		P1.0 = .89		
Turn Lamp Spec.	SAE J 588			20		P1.0 = .89		
Turn Signal Switch Spec.	SAE J 589 Life 165,000			20		P1.0 = .89		

ALL LIGHTING TEST AT 95 PERCENT CONFIDENCE

DESIGN VERIFICATION AND STATUS

SYSTEM 12.000		ASSEMBLY Lighting		PROGRAM Electric Car				
SUBSYSTEM 12.200		COMPONENT Exterior Light		DATE 8-12-76				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Side Turn Signal Lamp Spec.	SAE J 914			20		P1.0 = .83		
Turn Signal Flasher	SAE J 590 Life and Performance J 1055 36 hr @ 75°F			10		P1.0 = .67		
Hazard Lamp Flasher	SAE J 945 Life and Performance J 1056 36 hr at 75°F			10		P1.0 = .67		
Hazard Lamp Switch	SAE J 910 Life Test 10,000 Cycles			20		P1.0 = .83		
Parking Lamp Spec.	SAE J 222			20		P1.0 = .83		
Back-Up Lamp Spec.	SAE J 593			20		P1.0 = .83		
Back-Up Lamp Switch	SAE J 1076 1 hr Each 75°F, 225°F, and -25°F; 10 Cycles Each Temp			20		P1.0 = .83		

ALL LIGHTING TEST AT 95 PERCENT CONFIDENCE

DESIGN VERIFICATION AND STATUS

SYSTEM 13 000		ASSEMBLY Wiper/Washer		PROGRAM Electric Car				
SUBSYSTEM 13.300		COMPONENT Switch		DATE 8/12/76				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Wiper Switch Spec.	SAE J 112 Life Test 10,000 Cycles			20		P1.0 = .83		
Washer Switch Spec.	SAE J 234 Life Test 10,000 Cycles			20		P1.0 = .83		
Wiper Performance	1,500,000 Cycles SAE J 9C3			20		P1.0 = .83		
Washer Performance	SAE J 9E2 8000 Cycles			20		P1.0 = .83		

WIPER/WASHER TEST AT 95 PERCENT CONFIDENCE

SYSTEM 14.000		ASSEMBLY Climate Control		PROGRAM Electric Car				
SUBSYSTEM		COMPONENT Switch		DATE 8/12/76				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Blower Motor Switch	SAE J 225 Life Test 10,000 Cycles			20		P.7 = .53		
Defroster Test	SAE J 982 Performance FMVSS 103			1				
Backlight Test	SAE J 953 Performance			1

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DESIGN VERIFICATION AND STATUS

SYSTEM <u>16.000</u>		ASSEMBLY <u>Body</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8/12/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Body Test	SAE	J 963,224		5		P1.0 = .63		
		138,885 211,972 850,857 996,1089 374,367 136,384 921,346 944, 52 338						
	FMVSS 215 FMVSS 203 FMVSS 216			3				

BODY TEST AT 80 PERCENT CONFIDENCE LEVEL

DESIGN VERIFICATION AND STATUS

SYSTEM <u>17.000</u>		ASSEMBLY <u>Seat Belts</u>		PROGRAM <u>Electric Car</u>		DATE <u>8/17/76</u>			
SUBSYSTEM _____		COMPONENT _____							
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT	
				REQ.	TESTED	REQ.	ACTUAL		
Seat Belt Test	SAE J 14C J 141 J 117 J 787 J 114 J 128			20		P1.0 = .68			
Seat Measure	SAE J 826			20		P1.0 = .68			
Seat Test	SAE J 879 J 383 J 954 J 815			20		P1.0 = .68			
Seat Belt Test	FMVSS 207 FMVSS 208 FMVSS 209 FMVSS 210			20		P1.0 = .68			

SEAT AND SEAT RESTRAINT TESTS AT 95 PERCENT CONFIDENCE LEVEL

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DESIGN VERIFICATION AND STATUS

SYSTEM 18.003		ASSEMBLY Locks and Latches		PROGRAM Electric Car		DATE 8/17/76		
SUBSYSTEM		COMPONENT						
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Hinge Sysytem	SAE J 934 2500 lb Load FMVSS - 206			20		P1.0 = .82		
Door Latch	SAE J 839 Static Load FMVSS -206			20		P1.0 = .82		
Hood Latch	SAE J 362 Static Load			20		P1.0 = .82		

LATCH SYSTEM TEST AT 95 PERCENT CONFIDENCE LEVEL

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DESIGN VERIFICATION AND STATUS

SYSTEM <u>20.000</u>		ASSEMBLY <u>Glass and Mirrors</u>		PROGRAM <u>Electric Car</u>				
SUBSYSTEM _____		COMPONENT _____		DATE <u>8/17/76</u>				
TEST NAME	ACCEPTANCE CRITERIA	TEST RESULTS	DESIGN LEVEL	SAMPLE SIZE		RELIABILITY		RISK ASSESSMENT
				REQ.	TESTED	REQ.	ACTUAL	
Glass Drop Test	SAE J 938			10		P.8 = .68		
Mirror Test	SAE J 964 J 985 J 834			3		P1.0 = .80		

GLASS AND MIRROR TESTS AT 50 PERCENT CONFIDENCE LEVEL