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ADVANCED PROPULSION SYSTEM CONCEPT FOR HYBRID VEHICLES

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MECHANICAL TECHNOLOGY INCORPORATED

December 1980

Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Under Contract DEN 3-92

for
U.S. DEPARTMENT OF ENERGY
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NOTICE

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PREFACE

The Electric and Hybrid Vehicle Research, Development, and Demonstration Act of 1976 (Public Law 94-413) authorized a Federal program of research and development designed to promote electric and hybrid vehicle technologies. The Department of Energy (DOE), which has the responsibility for implementing the act, established the Electric and Hybrid Vehicle Research, Development, and Demonstration Program within the Office of Transportation Programs to manage the activities required by Public Law 94-413.

The National Aeronautics and Space Administration (NASA) was authorized under an interagency agreement (Number EC-77-A-31-1044) with DOE to undertake research and development of propulsion systems for electric and hybrid vehicles. The Lewis Research Center was made the responsible NASA center for this project. The study presented in this report is an early part of the Lewis Research Center program for propulsion system research and development for hybrid vehicles.

TABLE OF CONTENTS

TABLE OF CONTENTS (Cont'd)

TABLE OF CONTENTS (Cont'd)

LIST OF FIGURES

LIST OF FIGURES (Cont'd)

LIST OF TABLES

1.0 SUMMARY

This final report contains the results of a nine-month study contract awarded Mechanical Technology Incorporated (MTI) by the NASA-Lewis Research Center (NASA-LeRC) to identify and evaluate advanced propulsion systems for on-the-road hybrid vehicles. Two concepts, both utilizing Stirling engines, were evaluated for a number of reference mission/vehicles (i.e., the class of vehicle by size and general performance requirements).

All concepts and reference mission/vehicles reviewed during the study met the requirements specified by NASA-LeRC and defined in Section 2.3. Five reference mission/vehicles, ranging from a two-passenger commuter car to a large bus, were evaluated; each had its own performance specification. Selection of the "best" hybrid configuration and reference mission/vehicle was made against these requirements, as well as the major goals of reducing petroleum consumption and minimizing total energy consumption. The study evaluated both parallel and series hybrid systems, utilizing kinematic and free-piston Stirling engines, respectively. The Stirling engine was selected for use in the hybrid propulsion system because of its high-efficiency, multi-fuel, low-pollution and quiet-operation characteristics. Detailed discussion, results and conclusions of the initial parametric studies, the trade-off evaluations, the life-cycle cost studies and the conceptual design definition are contained in this report. A summary of the major results and conclusions is presented below.

1.1 Summary of Major Study Tasks

1.1.1 Parametric Analysis

The purpose of the parametric analysis was to review and evaluate five reference mission/vehicles utilizing either a parallel hybrid system with a kinematic Stirling engine or a series hybrid system with a free-piston Stirling engine-alternator. Based on this evaluation, a single reference mission/vehicle and concept (either a series or a parallel hybrid system) was selected for further study and definition.

It was concluded that either a parallel or series hybrid system, utilizing a Stirling engine, could meet the specified requirements, in addition to significantly reducing the petroleum consumption (the major goal) of all reference mission/vehicles. For the requirements given, however, the parallel system resulted in lower petroleum fuel consumption and lower vehicle weight than did the series system for all reference mission/vehicles.

Based on the parametric studies, reference mission/vehicle C, a large, full-size, 6-passenger vehicle utilizing a parallel Stirling-engine hybrid system, exhibited the greatest fuel savings (total gallons, not necessarily percentage) over any other reference mission/vehicle evaluated as part of the parametric analysis. Therefore, reference/vehicle C was selected for detailed trade-off evaluation and further definition.

1.1.2 Trade-Off and Life-Cycle Costing Studies

The purpose of the trade-off studies was to further define and evaluate the concept and vehicle type selected as part of the parametric analysis. The selected parallel system, utilizing a kinematic Stirling engine for a 6-passenger, full-size vehicle, was evaluated against the following trade-offs:

- With and without short-term energy storage
- Varying battery weight from an all-Stirling vehicle to an all-electric vehicle
- Stirling hybrid propulsion system versus internal combustion hybrid propulsion system

Life-cycle costing was performed on the various vehicle propulsion systems in order to make the final determination as to:

- Cost/benefit of short-term energy storage
- Selection of either lead-acid or nickel-zinc batteries
- Petroleum price at which the hybrid vehicle is cost-effective to the consumer.

1.2 Conclusions/Results

The major conclusions of the advanced hybrid vehicle study are given below, along with a brief discussion of the basis for the conclusion. Details of the study are given in later sections.

1) A parallel hybrid propulsion system utilizing a kinematic Stirling engine reduces petroleum consumption by 50% over an all-Stirling-powered vehicle.

   Specifically, the study concluded that if a battery and an electric propulsion system are added to the all-Stirling-powered vehicle to form a hybrid vehicle, the annual petroleum consumption drops from 530 to 265 liters (140 to 70 gallons).

2) The proposed Stirling hybrid vehicle propulsion system can reduce petroleum consumption to 32% of the petroleum required for a conventional, internal combustion, 6-passenger vehicle.

   The proposed propulsion system utilizes 265 liters (70 gallons) over an annual period compared to 854 liters (223 gallons) for a conventional, nonhybrid 6-passenger vehicle. A conventional vehicle utilizes an internal combustion engine (no batteries) with identical assumptions used for the Stirling-engine vehicle (i.e., same vehicle weight, on-off operating strategy, and engine operation at peak efficiency point.)

3) Short-term energy storage (hydraulic accumulator system) reduces petroleum usage by 76 liters (20 gallons) over the annual driving distribution, as compared to an identical vehicle without energy storage.

   A short-term energy storage system (hydraulic accumulator) was incorporated into the vehicle propulsion system not just because it saves 76 liters (20 gallons) over an annual driving distribution (reduced consumption of petroleum was the major goal of the study), but because it is cost-effective to the consumer. At $0.53/liter ($2.00/gal), the fuel savings over the life of the vehicle would pay for the hydraulic accumulator system.

4) The Stirling parallel hybrid propulsion system's life-cycle costs are comparable to an all-Stirling-powered vehicle when gasoline approaches $0.92/liter ($3.50/gallon).

   The initial cost of the Stirling hybrid propulsion system is greater than an all-Stirling-powered vehicle because of the cost of the "extra" battery system. Before a hybrid system is attractive to the consumer, the gasoline savings over the life of the vehicle must pay for the increased initial cost. Thus, a price of $0.92/liter ($3.50/gallon) is required before the hybrid system pays for itself.

   The high cost of gasoline required before the life-cycle cost of the hybrid vehicle becomes equal to an all-Stirling-powered vehicle is because of the small amount of fuel used by the all-Stirling-powered vehicle, and hence, the small amount of fuel saved by the hybrid vehicle (total gallons, not percentage). If the hybrid vehicle was compared to a conventional, internal combustion vehicle, life-cycle costs would become equal at a much lower cost of gasoline.

5) On-off engine operational strategy utilizes the least petroleum.

6) Lead-acid batteries cost less than nickel-zinc batteries over the life of the vehicle.

1.3 Conceptual Design

The conclusions summarized in Section 1.2 lead to the selection of the advanced parallel Stirling-engine/electric hybrid propulsion system for a full-size, 6-passenger vehicle. The system incorpprates a hydraulic accumulator for short-term energy storage and utilizes an on-off operational strategy for the kinematic Stirling engine.

A schematic of the conceptual system is shown in Figure 1-1. The components are as follows:

- Engine: kinematic Stirling
  - 10.5 kW average operating point (peak efficiency point)
- Batteries: 7 12-volt lead-acid
- Motor: ac 3-phase induction
  - 12.8 kW
  - 8,000 r/min (rpm)
  - 20 cm wide x 38 cm long (8 in. x 15 in.)
- Inverter: variable frequency controller
  45 cm x 50 cm x 20 cm (18 in. x 20 in. x 8 in.)
- Hydraulic Motor/Pump: 36.4 kW
  - 0.045 m^3 (1.6 ft^3) volume with accumulator
- Microprocessor Control: 15 cm x 15 cm x 8 cm (6 in. x 6 in. x 3 in.)

A conceptual layout of the advanced propulsion system integrated into a possible vehicle is shown in Figure 1-2. Integration and conceptual layout were prepared by AM General Corporation.

While Figure 1-2 depicts the conceptual layout of the individual components of the hybrid propulsion system, it is desirable to integrate the hydraulic short-term energy storage system into the transmission for the advanced system.

Major developmental tasks remaining for the successful advanced Stirling hybrid propulsion system are to:

- Gain operating experience with the Stirling engine in an on-off mode
- Develop a hydraulically augmented transmission (short-term energy storage)
- Monitor ongoing automotive Stirling engine development and adjust the engine design as necessary to match hybrid vehicle/mission requirements.

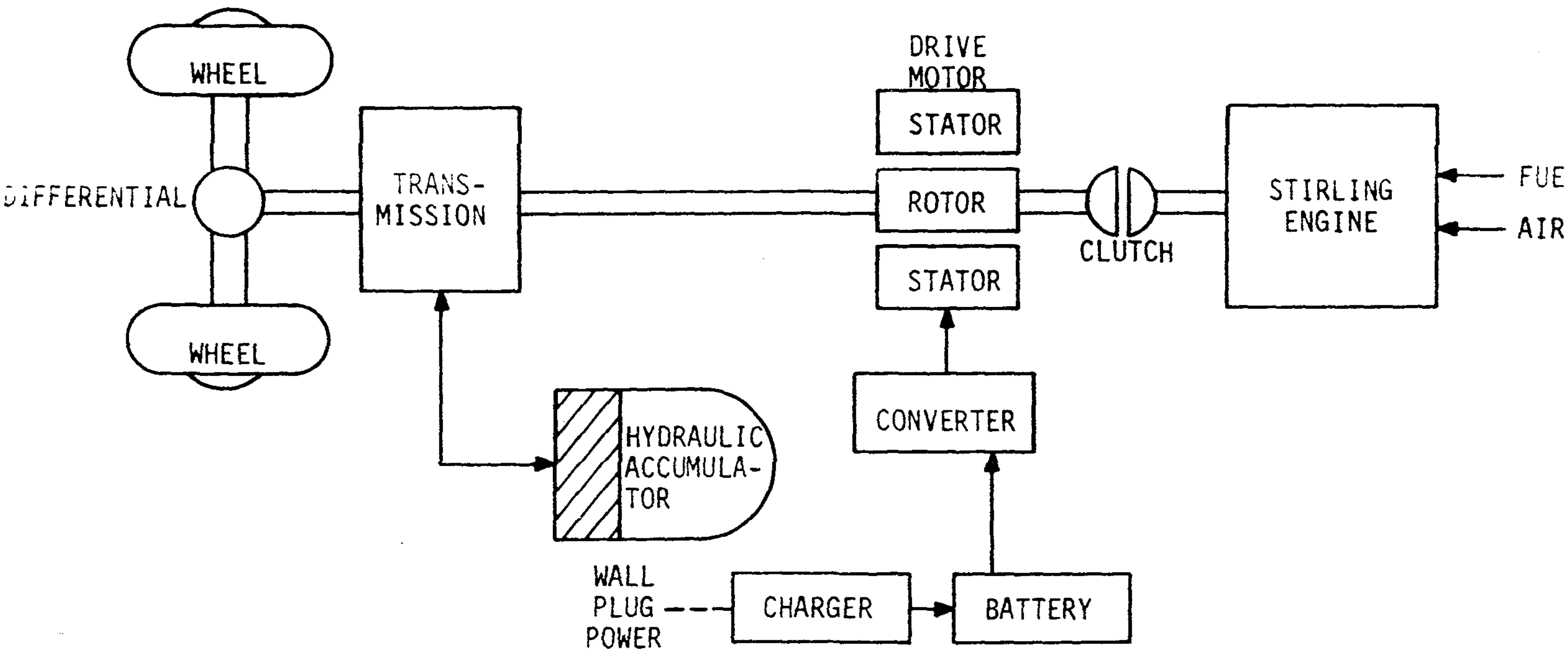

Fig. 1-1 Parallel System with Short-Term Storage

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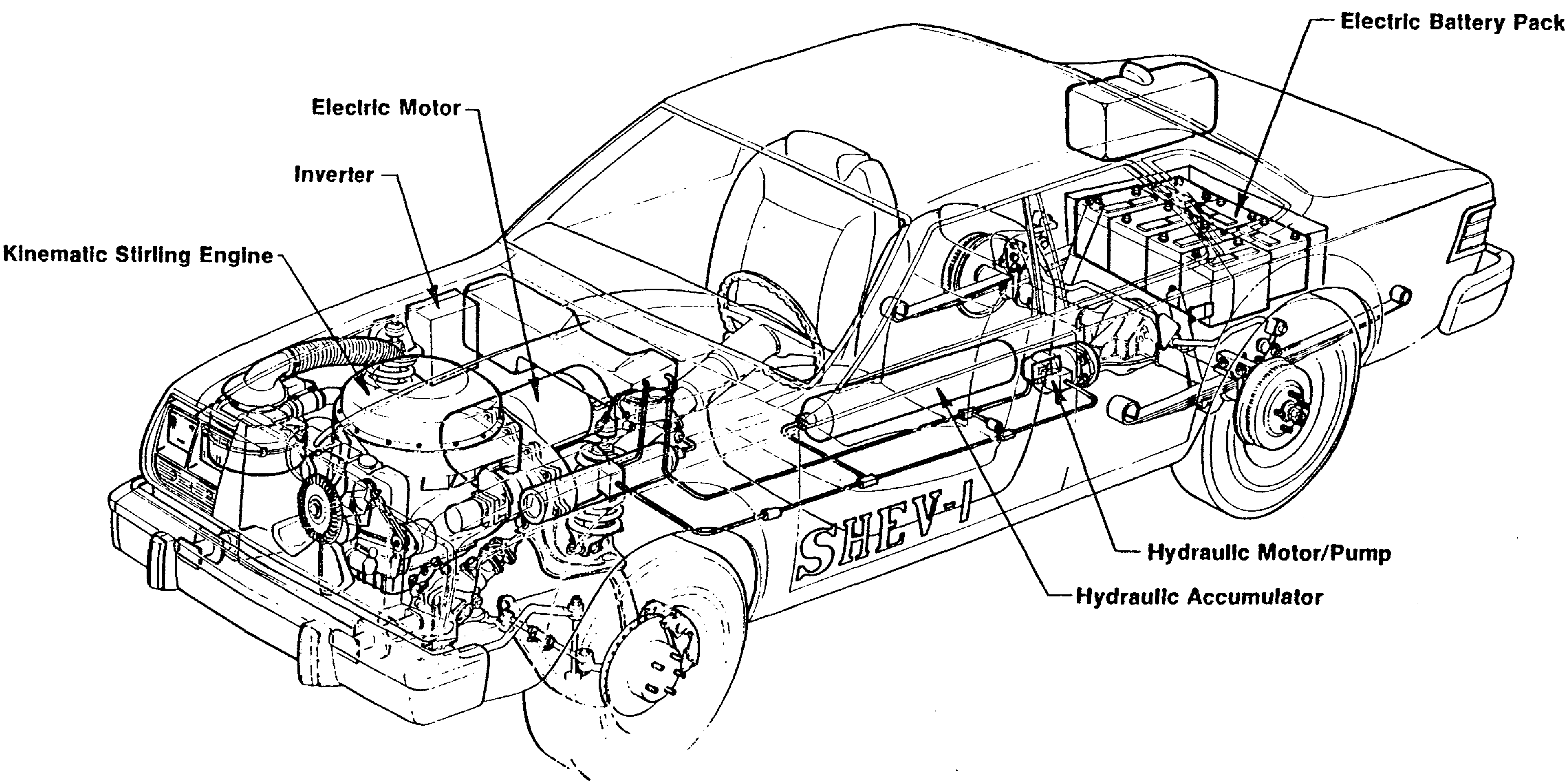

Fig. 1-2 Conceptual Layout of Advanced Propulsion System

2.0 INTRODUCTION

2.1 Background

The Electric and Hybrid Vehicle Program was initiated in 1975 within the U.S. Energy Research and Development Administration (ERDA), now the Department of Energy (DOE). In September of the following year Congress passed the Electric and Hybrid Vehicle Research, Development, and Demonstration Act of 1976 (Public Law 94-413). This act is intended to accelerate the integration of electric and hybrid vehicles into the nation's transportation system and to stimulate growth in the electric and hybrid vehicle industry.

The DOE is currently sponsoring efforts to design and build vehicles utilizing state-of-the-art technology for hybrid propulsion system and vehicle designs. Looking beyond that effort, the intent of this contract is to identify new hybrid propulsion system design concepts based on propulsion system conponents which can be developed in 5 years. Recent advances in heat engines, electronics, materials and fabrication techniques open the door to entirely new approaches and should be exploited under this program.

Potentially attractive concepts for advanced hybrid propulsion systems are to be identified that offer considerable performance and energy consumption improvement over existing propulsion systems with reduced petroleum fuel usage and little or no potential cost penalty.

2.2 Objective

The primary objective of this study was to identify advanced hybrid propulsion systems that can significantly reduce the petroleum usage of present vehicles with little or no cost impact. Such a propulsion system and vehicle had to meet all the performance requirements as established for this study by NASA-LeRC.

2.3 Requirements

The performance requirements for this study, as specified by NASA-LeRC, are given in Table 2-1. The vehicle was required to be evaluated over the Special Test Cycle (STC) (shown in Figure 2-1), which is a modified

TABLE 2-1

REFERENCE MISSION/VEHICLE DESIGN CONSTRAINTS AND GOALS*

A. Additional Basic Design Constraints

TYPE OF USE	COMMUTING	FAMILY USE (LOCAL)	FAMILY USE (INTERCITY)	VAN (VARIABLE ROUTE)	CITY BUS (VARIABLE ROUTE)
MISSION/VEHICLE DESIGNATION	A	B	C	D	E
PAYLOAD: NO. OF PASSENGERS —	2	4	6	8	50
CARGO kg (lb)	30 (66)	0 (0)	100 (220)	500 (1100)	225 (500)
TOTAL kg (lb)	166 (366)	272 (600)	508 (1120)	1043 (2300)	3629 (8000)
CARRIAGE: WEIGHT (SEE TABLES 3-1 AND 3-2)					
CHARACTERISTICS: AERO. COEF. C_OA m^2 (ft^2)	0.56 (6.0)	0.56 (6.0)	0.60 (6.5)	1.7 (18.3)	3.5 (37.7)
TIRE ROLL. RES. COEF —	$0.008 + 1 \times 10^{-5}V + 8 \times 10^{-8}V^2$ (V = km/h)				
TIRE ROLLING RADIUS m(ft)	0.27 (0.89)	0.29 (0.95)	0.31 (1.02)	0.34 (1.12)	0.518 (1.70)
ACCESSORY LOAD (MAXIMUM) WATTS	500	600	600	600	1200

YEARLY TRIP DISTRIBUTION AS NUMBER OF TRIPS VS. TRIP LENGTH	NO. TRIPS	TRIP LENGTH km (mile)	CITY BUS
	500	5 (3.1)	EXCEPTION:
	310	10 (6.2)	ASSUME 32,000 km
	200	20 (12.4)	PER YEAR ON
	100	40 (25.0)	SAE J227a
	10	80 (50.0)	SCHEDULE C
	10	160 (100.0)	

NOTE: ASSUME "SPECIAL TEST CYCLE," STC (FIG. 1-1), EXCEPT FOR CITY BUS, AND FOR 160 km TRIP ASSUME 5% DIST. ON STC, REMAINDER STEADY CRUISE AT 90 km/h (56 mph)

B. Additional Design Goals

	A	B	C	D	E
MIN. TOP SPEED ON LEVEL ROAD: km/h (mph)	105 (65)	105 (65)	105 (65)	96 (60)	80 (50)
MAX. ACCEL. TIME: seconds					
0-50 km/h (0-31 mph)	6	5	5	6	12
0-90 km/h (0-56 mph)	15	12	12	15	—
40-90 km/h (37-62 mph)	12	10	10	12	—
GRADEABILITY AT SPEED FOR SPECIFIED DISTANCE:					
3% GRADE, 90 km/h (56 mph) km (mile)	1.0 (0.62)	1.5 (0.93)	1.5 (0.93)	1.5 (0.93)	—
8% GRADE, 50 km/h (31 mph) km (mile)	0.3 (0.19)	0.5 (0.31)	0.5 (0.31)	0.5 (0.31)	0.2 (0.12)
15% GRADE, 25 km/h (16 mph) km (mile)	0.2 (0.12)	0.3 (0.19)	0.3 (0.19)	0.3 (0.31)	15% MAX. GRD.
MIN. RAMP SPEED ATTAINABLE FROM A STOP ON UPHILL 6% GRADE IN 300 m (984 ft): km/h (mph)	80 (50)	80 (50)	90 (56)	80 (50)	65 (40)
MIN. SUSTAINED SPEED UP 4% GRADE: km/h (mph)	90 (56)	90 (56)	90 (56)	90 (56)	70 (44)

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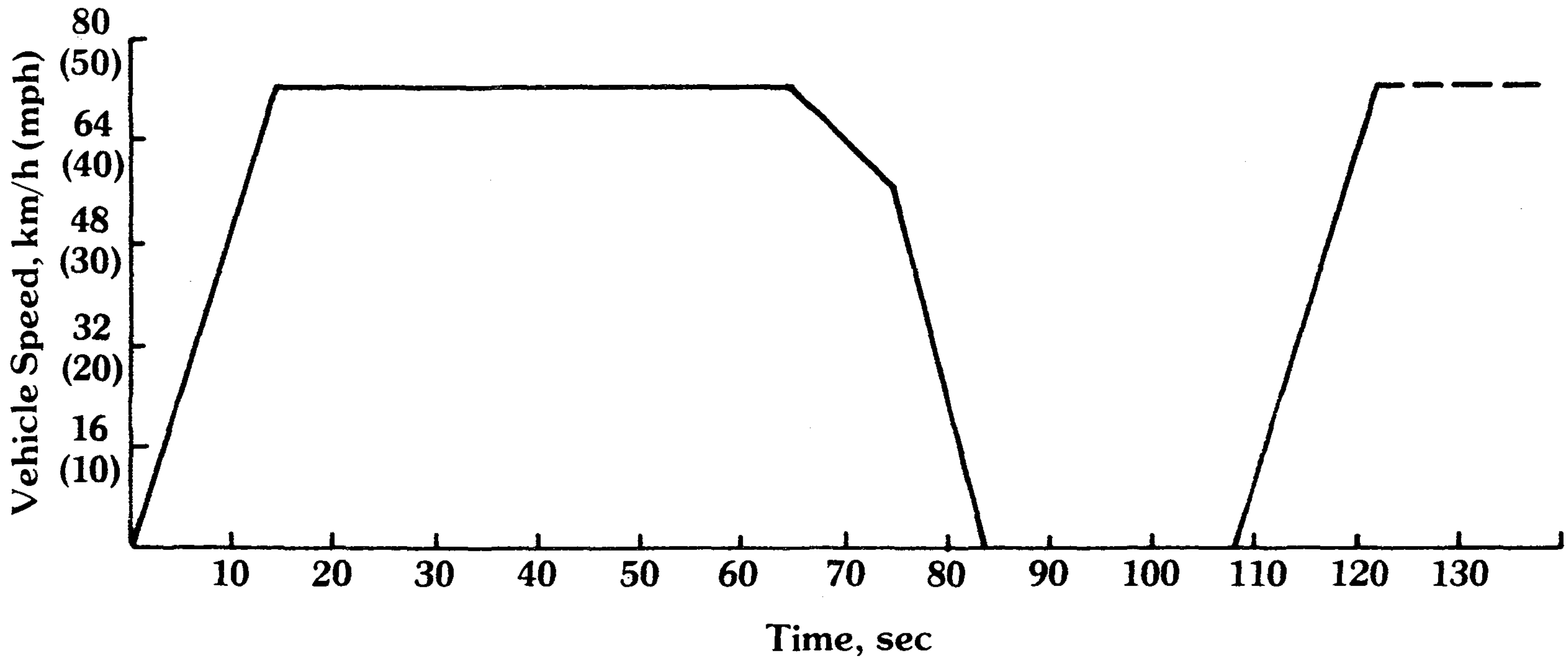
*This table has been reproduced from Contract DEN3-92, Exhibit A, Section 4.0 - Statement of Work, Table I.



Vehicle Cruise Speed, km/h (mph)	72 (45)
Acceleration Time, sec	14
Cruise Time, sec	50
Coast Time, sec	10
Braking Time, sec	9
Idle Time, sec	25

Fig. 2-1 Special Test Cycle (STC) (Modified SAE J227a, D Cycle)

SAE J227a, D cycle. The annual trip distribution, presented in Table 2-2, was required to determine total petroleum used and saved over an annual period, and operating cost over the life of the vehicle.

TABLE 2-2

DAILY RANGE FREQUENCY FOR ONE YEAR

DAILY RANGE		NO. DAYS OF THE YEAR	TOTAL RANGE	
km	mile		km	mile
0	0.0	16	0	0
10	6.2	130	1300	808
30	18.6	85	2550	1585
50	31.1	57	2850	1771
80	49.7	54	4320	2685
130	80.8	12	1560	970
160	99.4	7	1120	696
500	311.0	3	1500	932
800	497.0	1	800	497
TOTALS		365	16000	9944

Note: Use the above data to compute the yearly on-board fuel and wall plug energy consumption for all reference mission/vehicles except the city bus. For days with less than 80 km range, assume the Special Test Cycle (STC) shown in Figure 2-1. For days with 80 km range or more, assume that 10% of the distance is driven over the STC and that 90% of the distance is driven at a steady speed of 90 km/h (56 mph).

For the city bus, assume that its daily range is constant and use SAE J227a, Schedule C.

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3.0 TECHNICAL APPROACH

Petroleum resources are considered exhaustible in the foreseeable future. Electrical energy, however, can be generated by alternative sources such as nuclear power, solar power, coal, etc. In the development of a hybrid electric vehicle propulsion system,both total energy used (electrical and petroleum) and petroleum used must be taken into account.

The major philosophy and primary goal of this study was the reduction of petroleum consumption; reduction of total energy consumption was secondary. Meeting these goals, as well as providing the range and basic performance of a conventional vehicle, should be accomplished at a reasonable life-cycle cost to the consumer. This study did not consider that petroleum is required to generate electricity since electricity is assumed to be generated by alternative sources.

3.1 Scope of Study

The hybrid electric propulsion systems evaluated in this study included a free-piston Stirling engine combined with a linear alternator in a series configuration, and a kinematic Stirling engine in a parallel configuration. The conceptual hybrid propulsion systems conformed to the basic design constraints as specified by NASA-LeRC and given in Section 2.3. In achieving the study goals for these hybrid propulsion systems, the following vehicle standards also applied:

- Control characteristics similar to a conventional propulsion system vehicle
- Acceptable levels of pollution emissions
- Conformity to all applicable Federal Motor Vehicle Saftey Standards
- Cost goal of $0.05/km
- Maintenance and reliability equivalent to a conventional vehicle
- Capability to operate over a temperature range of -29°C to 52°C.

The study effort was divided into three work tasks. Each is discussed below.

Task 1.0 Parametric Studies

The purpose of this task was to perform a parametric study of each concept (series and parallel) to determine its suitability to each of the five reference mission/vehicles. The results of the parametric analysis were the basis for the recommendation to NASA-LeRC as to whether or not the configuration for a specific reference mission/vehicle should be continued for the remaining tasks.

Task 2.0 Design Trade-Off Studies

The objective of the design trade-off studies was to balance the performance and cost of the approved propulsion system approach for the approved reference mission/vehicle application.

Task 3.0 Conceptual Design

The major purpose of this task was to prepare a conceptual design of the selected propulsion system. The conceptual design presents, by means of layout drawings, the overall propulsion system configuration indicating location of all propulsion system components within a possible vehicle configuration.

3.2 Task Execution

The following sections give a detailed description of the actual work involved in the execution of each of the three study tasks.

3.2.1 Parametric Studies

The parametric studies were performed for each selected concept for each reference mission/vehicle specified by NASA-LeRC. The concepts considered were the series configuration and the parallel configuration. Both configurations included:

- A battery for primary energy storage
- An ac electric drive with an induction motor and a static inverter

- No energy buffer
- Electrical regeneration into the battery.

The series configuration used a Sealed Power Unit (SPU) containing a free-piston Stirling engine and an electric linear alternator. The parallel configuration differed from the series configuration in its use of a kinematic Stirling engine (rotating output shaft) as a power source, and in the possible addition of a transmission.

Flow charts for the parametric studies are shown in Figures 3-1 and 3-2. The concepts selected had to meet the specified performance criteria for each reference mission/vehicle in order to be considered acceptable.

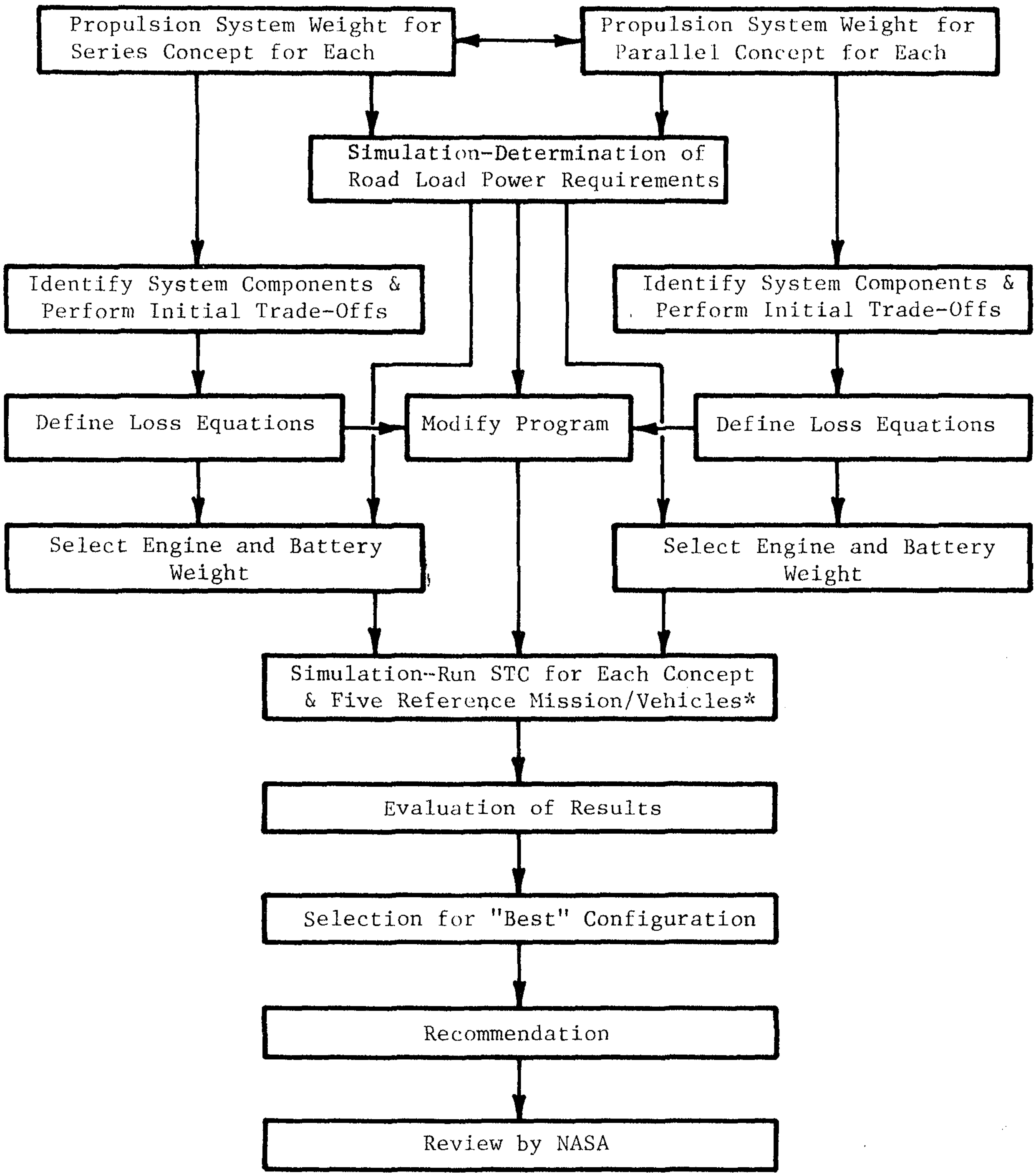
The total power required was assumed to be supplied by the engine or battery, or a combination of both. The total power is defined as the power to meet the highest peak transient load. This peak load could be two to four times the load required to meet the Special Test Cycle (STC) and/or cruise at 90 km/h.

The selection criterion used in sizing the engine and battery was to minimize the total petroleum consumption over the specified annual driving distribution. However, if this criterion was the only one used, then selection of an all-electric vehicle would result. Therefore, the criterion was more of a combination of minimizing total petroleum consumption, total energy consumption and vehicle weight.

The choice of the reference mission/vehicle and hybrid propulsion concept to be further evaluated through detail trade-off studies was based on the vehicle that showed the greatest savings in petroleum as compared to a conventional vehicle over an annual period.

3.2.2 Design Trade-Off Studies

Trade-off studies were performed within six areas. A short description of the work involved in each area follows.



* Note: This subroutine has been expanded and is given as Figure 3-2.

Fig. 3-1 Task 1.0 Parametric Studies of Series/Parallel Concepts for Each Reference Mission/Vehicle

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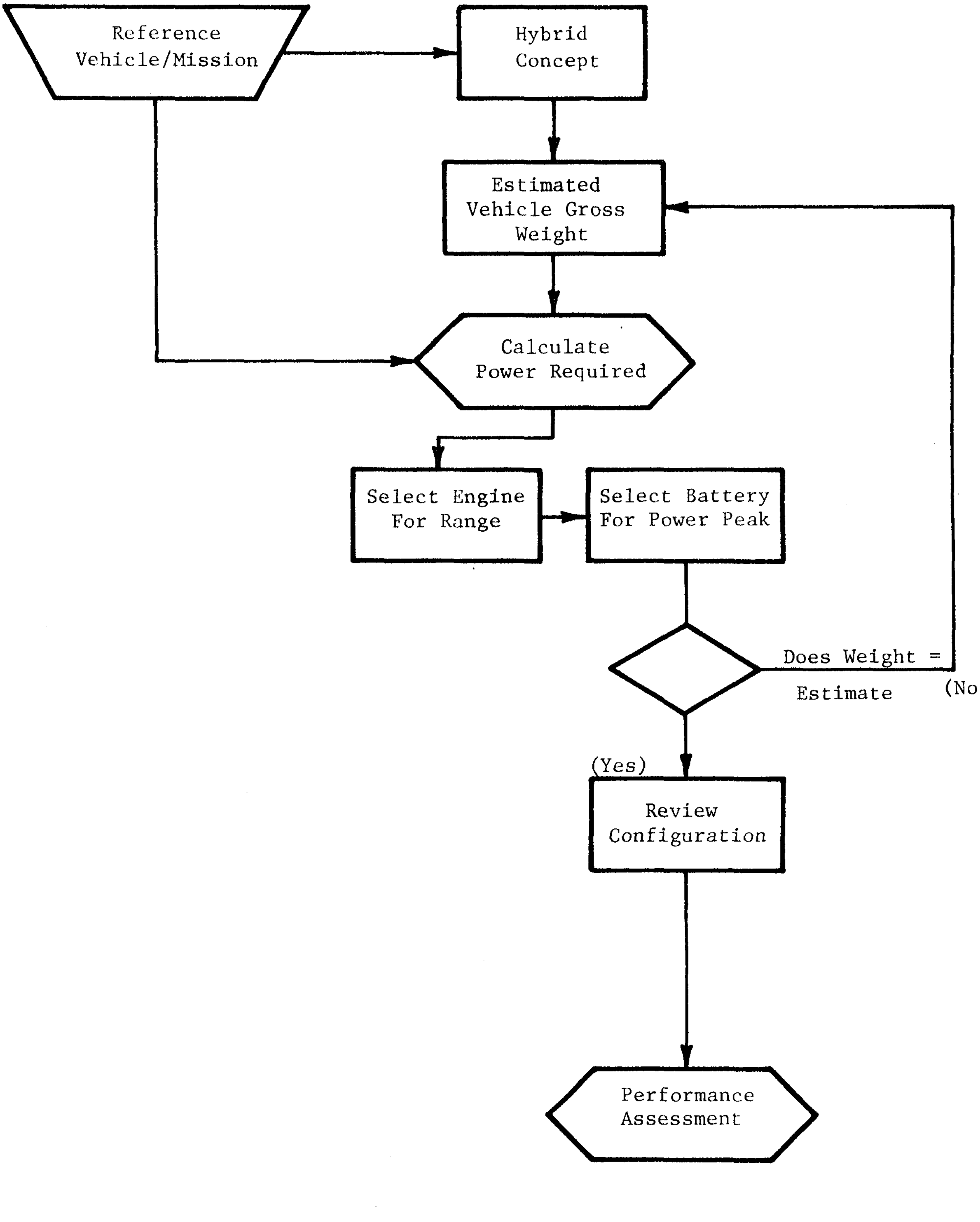

Fig. 3-2 Flow Chart for Analytical Simulation

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3.2.2.1 Energy Buffer System. The decision that an energy buffer may be desirable to improve overall system performance was based on the results from the parametric studies.

Initial trade-off studies were performed to evaluate various secondary energy storage systems and to select the one that was most beneficial to the proposed hybrid propulsion system. The energy storage system was selected based on limited energy storage required to aid in vehicle acceleration, rapid discharge (less than 15 seconds), system weight, and efficient recovery of energy during regenerative braking.

3.2.2.2 Engine. The engine selected is a modified design of an engine presently being developed for automotive applications. Because of its hybrid operation, the engine can be made to operate at its most efficient point (i.e. minimum petroleum consumption) most of the time.

3.2.2.3 Batteries. The battery type and performance were specified. Battery selection was based on the total life-cycle cost (first cost plus battery replacement) of the alternative configurations.

3.2.2.4 Inverter and Motor. The inverter and motor were selected based on the work performed in Reference [1]*.

3.2.2.5 Operating Strategy. As throughout the study program, the need to reduce petroleum consumption was also the major factor in selection of an operating strategy. The trade-offs included engine on-off versus continuous operation, batteries to supply base or peak power, battery weight versus range, and operation with or without an energy buffer.

3.2.2.6 System Selection and Life-Cycle Cost. Life-cycle cost studies were extremely important in the selection of the final design. An understanding existed that unless a clear benefit to the consumer could be shown, no hybrid or electric vehicles would be available in any number to significantly affect national petroleum usage. Therefore, the development of an advanced propulsion

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*References are located at the end of this report.

system with the lowest possible cost over the lifetime of the system was most desirable.

Because of the rapid escalation of petroleum fuel prices, the life-cycle costing was performed with various prices ranging from \$0.40 to \$1.19 per liter (\$1.50 to \$4.50 per gallon).

The system evaluation and life-cycle costing included the following:

- Hybrid versus all-electric versus all-Stirling power versus conventional internal combustion vehicles
- Lead-acid versus nickel-zinc batteries
- Hydraulic accumulator versus no hydraulic accumulator.

3.2.3 Conceptual Design

The conceptual design layout and integration within a possible vehicle configuration were provided by AM General Corporation. AM General Corporation also provided guidance and constructive criticism on the work performed throughout this study.

4.0 PARAMETRIC STUDIES

The required performance of different vehicles for various missions is well defined in Section 2.3. The objective of the parametric studies was to determine the most attractive reference mission/vehicle for a particular selected hybrid concept. Due to the lower power density of a battery as compared with an engine, a hypothesis can be formed that, while the vehicle weight, power requirement and total energy consumption increase as the percentage of the power supplied by the battery increases, the fuel consumption decreases. An optimum proportion may exist between the electric and engine power such that the total energy consumption as well as an acceptable vehicle weight is minimized. A computer simulation was used to find this optimum combination.

Two concepts, series and parallel, were simulated by computer program. Vehicle weight and energy consumption per mile were determined as a function of percentage of battery weight.

4.1 Definition of the Series Concept

The series concept, shown in Figure 4-1, consists of two sources of electric power, one from the battery and the other from an SPU. The SPU consists of a free-piston Stirling engine which powers a linear alternator in a hermetically sealed power unit. The SPU can operate in many different modes; i.e., power the vehicle all by itself, supplement the battery power, or just stand by. In this study, the battery was considered as the primary power source, and the SPU, a secondary power source.

4.2 Definition of the Parallel Concept

The parallel concept, represented in Figure 4-2, consists of a set of batteries as the primary energy source and a kinematic Stirling engine as a secondary power source. Differing from the series concept, the mechanical power output of the Stirling engine is coupled to the vehicle drive system in conjunction with the electric motor. The kinematic Stirling engine can also be operated in different modes; i.e., powering the vehicle alone, assisting the electric motor, or standing by, keeping the engine hot without idling.

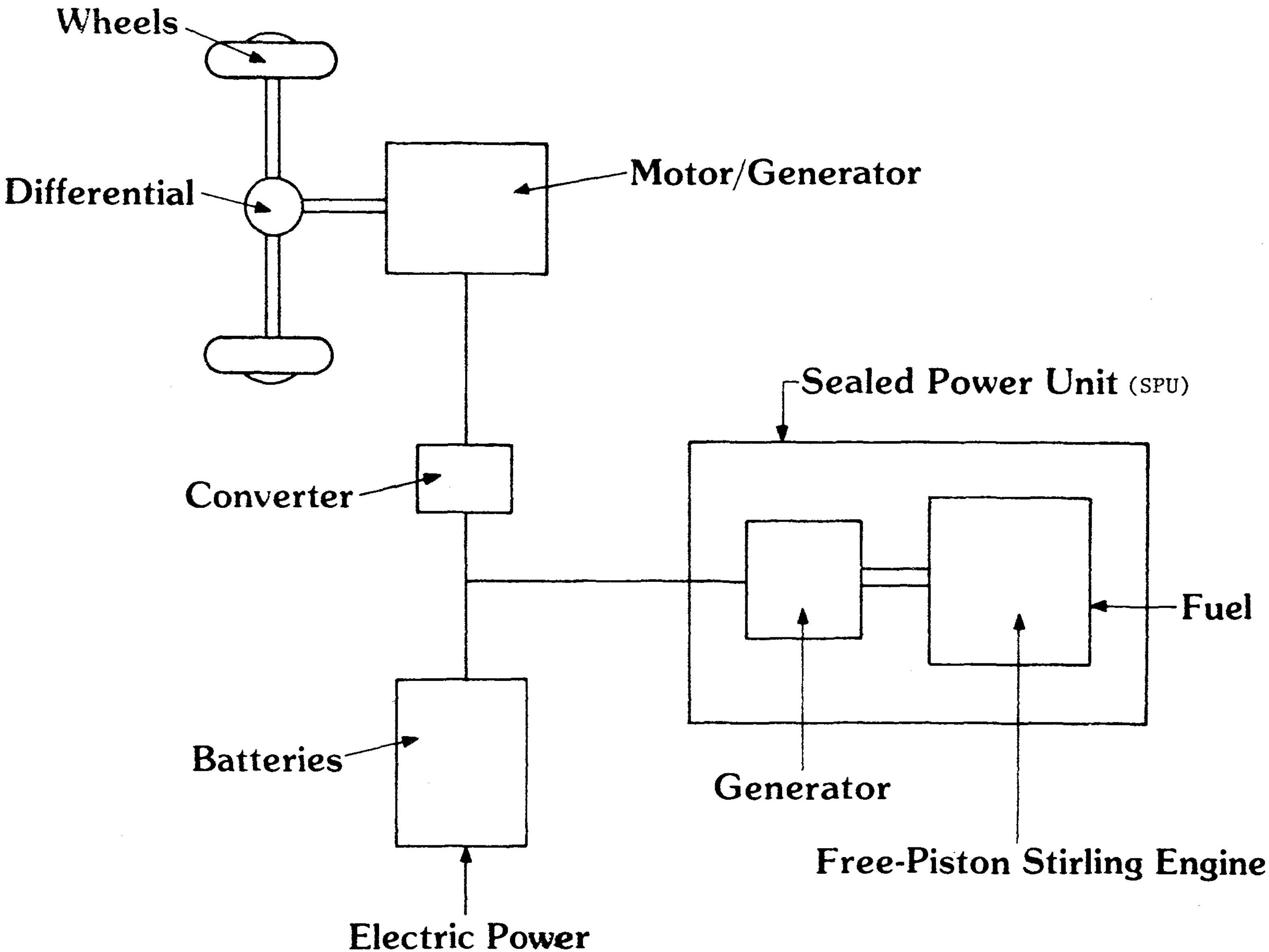

Fig. 4-1 Stirling Engine/Battery Series
Hybrid Propulsion System

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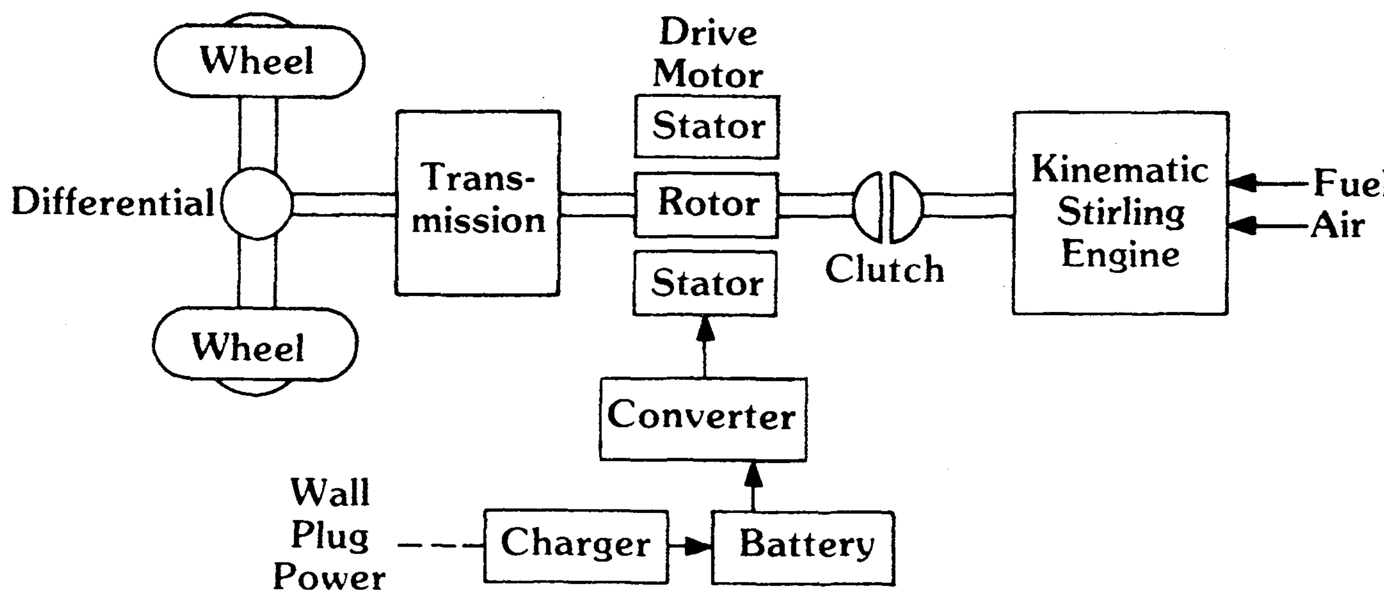

Fig. 4-2 Stirling Engine/Battery Parallel Hybrid Propulsion System

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4.3 Computer Simulation

To achieve the goal of reducing energy consumption and, in particular, petroleum fuel, vehicle weight had to be minimized. The parametric representation of weight is given in Table 4-1. For each mission/vehicle, the specific weight constants are given in Table 4-2. To minimize the vehicle weight is to minimize the propulsion system weight together with the weight of its supporting structure. Computer simulation was utilized to consider these interrelated weight factors as follows:

4.3.1 Vehicle Test Weight

The vehicle test weight is calculated in the computer simulation using four formulas specified by NASA/LeRC:

$$W_G = W_S + W_{PL} + W_P + W_F$$

$$W_C = W_G - W_{PL,\ max}$$

$$W_T = W_C + W_{TL}$$

$$W_S = 0.23\ W_G$$

These formulas are combined and the vehicle test weight is obtained as:

$$W_T = (\frac{1}{0.77} - 1)\ W_{PL} + W_{TL} + (W_F + W_P)/0.77$$

where

$$W_{PL} = W_{PL},\ max.$$

4.3.2 Propulsion System Weight

The propulsion system weight, W_P, is the sum of the weight of the propulsion system components. These components and their weight are described as follows:

- Battery - Constant specific power of 100 W/kg (22.05 lb/kW) is used in the simulation. (Variable specific power is used in the calculation described in Section 6.0.)

TABLE 4-1

PARAMETRIC REPRESENTATION OF WEIGHT*

Symbol	Definition	Formula
$W_{PL,\ max}$	Maximum design payload	-
W_{TL}	Test payload	-
W_F	Fixed weight	-
W_G	Gross vehicle weight	$W_G = W_S + W_{PL} + W_P + W_F$
W_C	Curb weight	$W_C = W_G - W_{PL,\ max.}$
W_T	Test weight	$W_T = W_C + W_{TL}$
W_S	Structure and chassis weight	$W_S = 0.23\ W_G$
W_P	Propulsion weight	Determined by contractor

*This table has been reproduced from the Contract's Statement of Work, Appendix A, Table A-1.

TABLE 4-2

MISSION/VEHICLE SPECIFIC WEIGHT CONSTANTS*

Constant	Units	Mission/Vehicle				
		A	B	C	D	E
$W_{PL,\ max.}$	kg (lb)	166(366)	272(600)	508(1120)	1043(2300)	3629 (8000)
W_{TL}	kg (lb)	83(183)	126(300)	254 (560)	522(1150)	1815 (4000)
W_F	kg (lb)	204(450)	408(900)	612(1350)	816(1800)	5200(11464)

*This table has been reproduced from the Contract's Statement of Work, Appendix A, Table A-2.

- Stirling Engine
  - SPU system weight, based on several previous MTI designs, is 43.55 + 5.12 kg/kW (96 + 11.288 lb/kW)
  - Kinematic Stirling engine weight - a novel variable-stroke, Z-crank engine, offering both simplicity and light weight, was selected for this study. This advanced Stirling engine was conceived and developed by MTI. The characteristics of this engine are reported in more detail in Section 4.3.1. For this simulation, the engine weight of 34.02 + 0.56 kg/kW (75 + 1.2418 lb/kW) is used.
- Fuel tank weight is assumed to be 34 kg (75 lb) for all vehicles
- Differential gear weight is assumed to be 45.5 kg (100 lb) for all vehicles
- Inverter weight is assumed to be 22.7 kg (50 lb) for all vehicles.

The weight of the components expressed in kg/kW is calculated based on maximum output power demanded from that component.

4.3.3 Numerical Integration of Velocity History

The power, P, to the wheel can be expressed in the following form:

$$P = q_1 V + q_2 V^2 + q_3 V^3 + q_4 V \frac{dV}{dt}$$

where q_1, q_2, q_3, and q_4 are constants and V is the instantaneous velocity of the vehicle. Using small time intervals (Δt), the velocity history can be integrated numerically as described here.

The values t_1 and V_1 are the known time and velocity at the beginning of the time interval and t_2 and V_2 are the unknown values at the end of the time interval. The following approximations are made:

$$V = (V_1 + V_2)/2$$

$$\frac{dV}{dt} = (V_2 - V_1)/\Delta t$$

Substituting these expressions for P in the previous equation and solving for V_2 yields a cubic equation:

$$P_0 + P_1V_2 + P_2V_2^2 + P_3V_3^2 = 0$$

where

$$P_0 = 4q_1V_1 + 2q_2V_1^2 + q_3V_1^3 + 4q_4V_1^2/\Delta t - 8P$$

$$P_1 = 4q_1 + 4q_2V_1 + 3q_3V_1^2$$

$$P_2 = 2q_2 + 3q_3V_1 + 4q_4/\Delta t$$

$$P_3 = q_3$$

Solving for the one real root of the above cubic equation yields V_2 which is then used as the initial velocity of the next time-interval calculations.

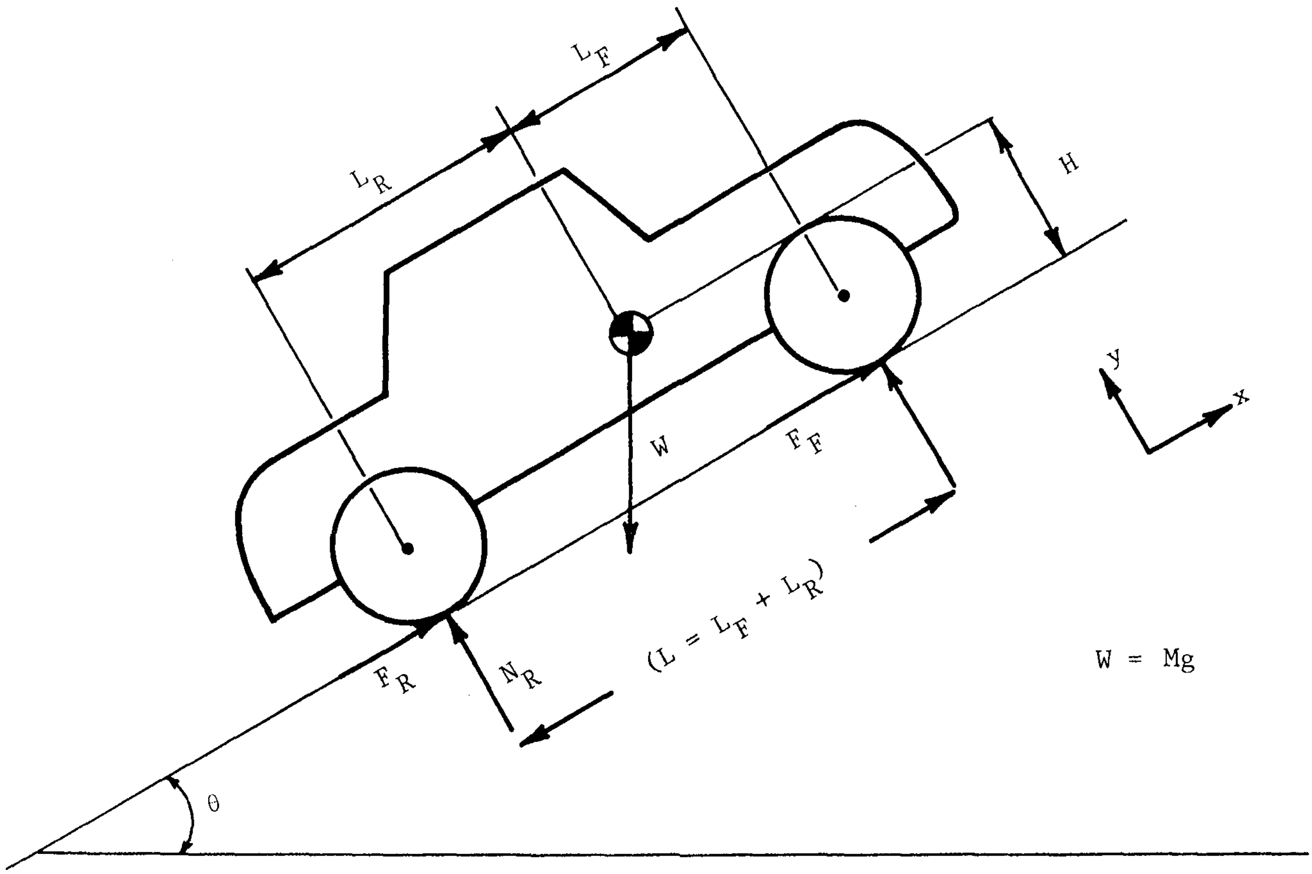
Constant power is used in the above calculation. However, at the initial stage during acceleration, the power supplied to the wheel may cause the wheels to slip. Therefore, when the calculated acceleration exceeds the traction limited acceleration, the integration is performed using constant acceleration equal to this limit. Figure 4-3 shows the equations used to calculate the traction-limited acceleration, a_{max}. In this case, V_2 is calculated from:

$$V_2 = V_1 + a_{max}\,\Delta t$$

The distance traveled by the vehicle, x, is integrated by:

$$x_2 = x_1 + V\Delta t$$

The above procedure is repeated until the desired velocity is achieved. If the time at the end of integration is more or less than the specified time, the power to the wheel is decreased or increased in order to minimize the error.



Equations of motion:

$$\Sigma F_x = F_F + F_R - W \sin\theta = M\ddot{x}$$

$$\Sigma F_y = N_F + N_R - W \cos\theta = M\ddot{y} = 0$$

$$\Sigma M_z = N_F L_F - N_R L_R + (F_F + F_R)\,H = I\ddot{\alpha} = 0$$

Assuming $F_F = 0$ and solving for N_F and N_R, then

$$N_F = W \cos\theta \frac{L_R}{L} - F_R \frac{H}{L}$$

$$N_R = W \cos\theta \frac{L_F}{L} + F_R \frac{H}{L}$$

By defining $\mu = F_{R,\,max}/N_R$ as the coefficient of static friction, the traction-limited acceleration is computed as follows:

$$a_{max} = g\left(\frac{\mu L_F/L}{1 - \mu\, H/L} \cos\theta - \sin\theta\right)$$

801893

Fig. 4-3 Calculations for Obtaining a_{max}

4.3.4 Iteration to Determine Power and Vehicle Weight

The power to the wheels, P, increases as the vehicle weight, W, increases. This weight versus power relationship can be expressed in linear form as:

$$W = A_1 + A_2 P$$

where A_1 and A_2 are constant coefficients.

Assuming that the power requirement per unit weight of vehicle is inversely proportional to some mth power of the required time, t (or distance x),

$$\frac{P}{W} \sim \frac{1}{t^m} ,$$

then an iterative algorithm can be easily devised in order to arrive at the solution. If P_i and W_i are the values used for the last iteration, and the simulation yields a time t_i, then the new iteration values (P_{i+1} and W_{i+1}) must satisfy the following two equations:

$$\frac{P_{i+1} W_i}{P_i W_{i+1}} = \left(\frac{t_i}{t}\right)^m$$

where $W_{i+1} = A_1 + A_2 P_{i+1}$.

These equations can be combined to yield:

$$P_{i+1} = \frac{P_i A_1 \left(\frac{t_i}{t}\right)^m}{W_i - P_i A_2 \left(\frac{t_i}{t}\right)^m}$$

When no vehicle weight variation is required, the above reduces to:

$$A_2 = 0$$

$$W_{i+1} = W_i = A_1$$

$$P_{i+1} = P_i \left(\frac{t_i}{t}\right)^m$$

If the desired maneuver is to reach a velocity within a specified distance, $(\frac{x_i}{x})$ should replace $(\frac{t_i}{t})$ in the above equation. The values of m = 1 for specified time and m = 0.5 for specified distance were found to be adequate in order to achieve convergence. Figure 4-4 is a schematic of different velocity histories during iterations towards a solution.

4.3.5 Battery and Engine Power Fraction Variation

To find the optimum proportion between peak power supplied by the battery and that of the engine, the fraction of battery power, x, is varied from x = 0 (no battery) to x = 1 (no engine).

4.3.6 Simulation Results

The computer outputs of vehicles A through D of the series concept are summarized as shown in Tables 4-3 through 4-6. The computer-generated resluts for the vehicle C, parallel concept, is shown in Table 4-7. For easy comparison within the same configuration as the power combination changes, or between two different concepts, the computer results of vehicle C in series and parallel concepts are plotted in Figures 4-5a and 4-5b. These figures clearly show the effect of battery power fraction on vehicle weight and annual specific energy consumption. Conclusions may be drawn as follows:

- The vehicle weight increases as the fraction of battery power increases.
- On an annual basis, the energy input to the charger increases as the battery power fraction increases until a point at which the battery power adequate for STC is reached. Energy to the charger is the energy required to replace energy used during the annual period including the efficiency of the charger system. After this point, the energy input to the charger increases slightly due to the increase of vehicle weight.
- The total specific energy input to the propulsion system decreases until it reaches a minimum at around 50 or 60 battery power fraction depending upon the concept, and then increases gradually due to the increase in vehicle weight.

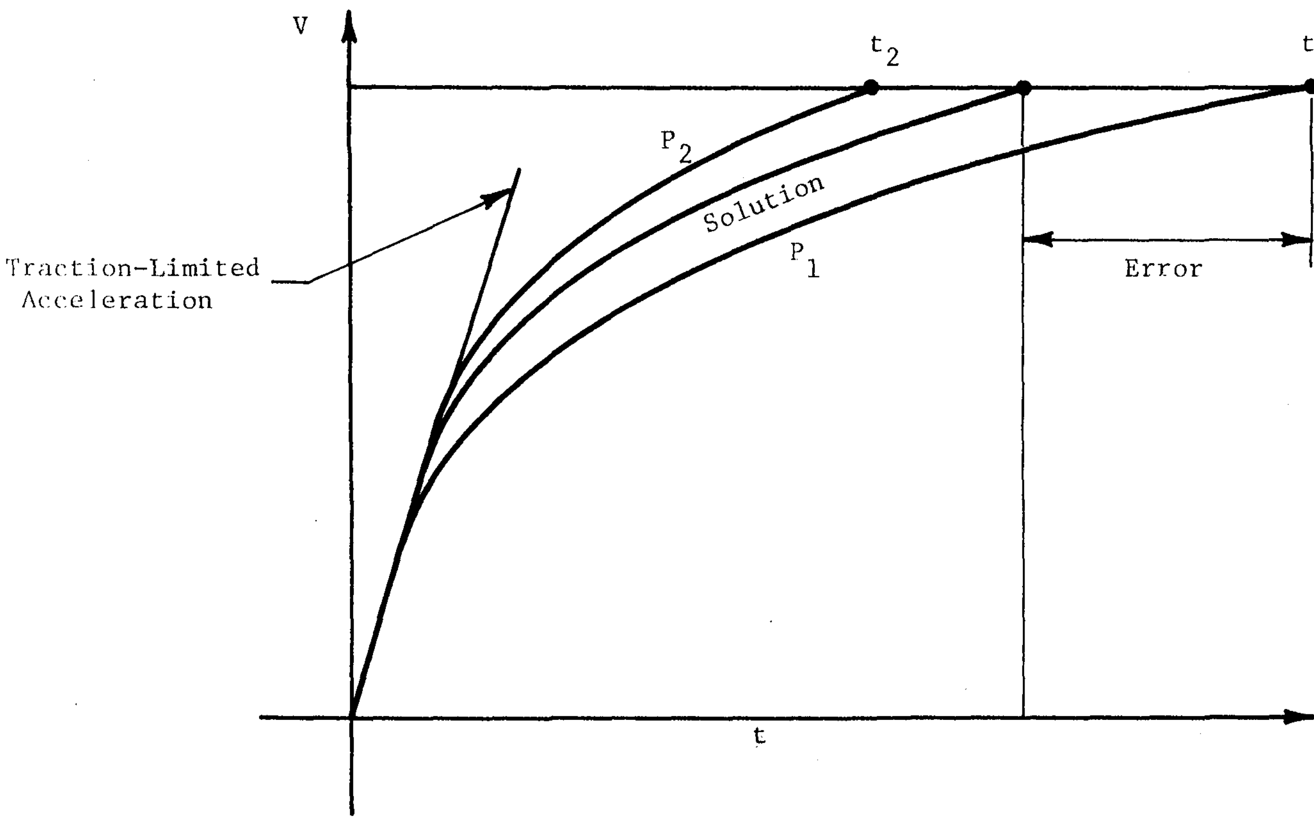


Fig. 4-4 Velocity Histories During Iterations

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TABLE 4-3

SIMULATED RESULTS FOR VEHICLE A/SERIES CONFIGURATION

VEHICLE/MISSION A		SERIES CONF.		5/09/79	
FRACTION SIZED ELECTRIC, X	0.0000	.2500	.5216	.7953	1.0000
WEIGHT OF BATTERY(LB)	0.0000	209.4600	454.6800	707.5700	900.0790
WEIGHT OF PROP.SYSTEM WP (LB)	1078.3619	1162.9926	1253.4522	1340.9781	1339.1224
TEST WEIGHT WT (LB)	2277.4700	2387.3800	2504.8600	2618.5300	2616.1200
MAXIMUM SPU OUTPUT (KW)	36.3500	28.4980	18.9100	8.7732	0.0000
MAX. BATTERY OUTPUT (KW)	0.0000	9.4993	20.6204	32.0893	40.8199
STC:					
PEAK WHEEL POWER (KW)	0.0000	0.0000	0.0000	0.0000	0.0000
ENERGY TO WHEELS (KJ/MI)	339.2285	346.7152	354.7128	359.6122	359.4522
ENERGY TO SPU (KJ/MI)	2015.5564	628.9501	196.6559	0.0000	0.0000
ENERGY TO CHARGER (KJ/MI)	0.0000	527.2670	742.3051	847.3100	846.8466
ENERGY TO SPU+CHARGER(KJ/MI)	2015.5564	1156.2171	938.9610	847.3100	846.8466
FUEL TO SPU (GAL/MI)	.0171	.0053	.0017	0.0000	0.0000
FUEL COST ($/MI)	.0257	.0080	.0025	0.0000	0.0000
ELECTRICITY COST ($/MI)	0.0000	.0088	.0124	.0141	.0141
TOTAL COST ($/MI)	.0257	.0168	.0149	.0141	.0141
STC RANGE TO .8 DISCHARGE(MI)	0.0000	23.0606	35.5569	48.4761	61.6988
56 MPH:					
WHEEL POWER (KW)	7.5246	7.6415	7.7665	7.8874	7.8848
ENERGY TO WHEELS (KJ/MI)	483.7263	491.2412	499.2736	507.0455	506.8807
ENERGY TO SPU (KJ/MI)	1838.3280	1865.4999	1894.5432	1566.6429	0.0000
ENERGY TO CHARGER (KJ/MI)	0.0000	0.0000	0.0000	158.2230	854.2439
ENERGY TO SPU+CHARGER(KJ/MI)	1838.3280	1865.4999	1894.5432	1724.8659	854.2439
FUEL TO SPU (GAL/MI)	.0156	.0159	.0161	.0133	0.0000
FUEL COST ($/MI)	.0234	.0238	.0242	.0200	0.0000
ELECTRICITY COST ($/MI)	0.0000	0.0000	0.0000	.0026	.0142
TOTAL COST ($/MI)	.0234	.0238	.0242	.0226	.0142
56MPH RANGE TO 0.8 DIS.(MI)					
COMBINED ANNUAL USAGE:					
ENERGY TO WHEELS (KJ/MI)	414.7960	422.2985	430.3155	436.7182	436.5557
ENERGY TO SPU (KJ/MI)	1922.8720	1275.6336	1084.6204	819.3371	0.0000
ENERGY TO CHARGER (KJ/MI)	0.0000	251.5200	354.0929	486.9250	850.7153
ENERGY TO SPU+CHARGER(KJ/MI)	1922.8720	1527.1536	1438.7133	1306.2621	850.7153
FUEL TO SPU (GAL/MI)	.0163	.0108	.0092	.0070	0.0000
FUEL COST ($/MI)	.0245	.0163	.0138	.0104	0.0000
ELECTRICITY COST ($/MI)	0.0000	.0042	.0059	.0081	.0142
TOTAL COST ($/MI)	.0245	.0205	.0197	.0186	.0142

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TABLE 4-4

SIMULATED RESULTS FOR VEHICLE B/SERIES CONFIGURATION

VEHICLE/MISSION B   SERIES CONF.   5/11/79

FRACTION SIZED ELECTRIC, X	0.0000	.2500	.5000	.7500	1.0000
WEIGHT OF BATTERY(LB)	0.0000	468.7980	1001.3290	1611.5370	2255.0190
WEIGHT OF PROP.SYSTEM WP (LB)	2050.5046	2271.3614	2522.2828	2809.8070	3019.7298
TEST WEIGHT WT (LB)	4311.9930	4598.8200	4924.6920	5298.1000	5570.7270
MAXIMUM SPU OUTPUT (KW)	79.9600	63.7300	45.4100	24.3600	0.0000
MAX. BATTERY OUTPUT (KW)	0.0000	21.2607	45.4117	73.0856	102.2684
STC:					
PEAK WHEEL POWER (KW)	32.7983	34.9002	37.2893	40.0267	42.0255
ENERGY TO WHEELS (KJ/MI)	473.6287	492.4839	513.9193	538.4647	556.4041
ENERGY TO SPU (KJ/MI)	3196.6452	1013.2493	99.9892	0.0000	0.0000
ENERGY TO CHARGER (KJ/MI)	0.0000	794.2104	1262.4690	1378.3457	1430.5307
ENERGY TO SPU+CHARGER(KJ/MI)	3196.6452	1807.4596	1362.4581	1378.3457	1430.5307
FUEL TO SPU (GAL/MI)	.0272	.0086	.0008	0.0000	0.0000
FUEL COST ($/MI)	.0408	.0129	.0013	0.0000	0.0000
ELECTRICITY COST ($/MI)	0.0000	.0132	.0210	.0230	.0238
TOTAL COST ($/MI)	.0408	.0262	.0223	.0230	.0238
STC RANGE TO .8 DISCHARGE(MI)	0.0000	34.2650	46.0423	67.8708	91.5066
56 MPH:					
WHEEL POWER (KW)	9.6885	9.9936	10.3401	10.7373	11.0273
ENERGY TO WHEELS (KJ/MI)	622.8319	642.4430	664.7237	690.2546	708.8942
ENERGY TO SPU (KJ/MI)	2359.1587	2430.0679	2510.6297	2602.9434	0.0000
ENERGY TO CHARGER (KJ/MI)	0.0000	0.0000	0.0000	0.0000	1186.8187
ENERGY TO SPU+CHARGER(KJ/MI)	2359.1587	2430.0679	2510.6297	2602.9434	1186.8187
FUEL TO SPU (GAL/MI)	.0201	.0207	.0213	.0221	0.0000
FUEL COST ($/MI)	.0301	.0310	.0320	.0332	0.0000
ELECTRICITY COST ($/MI)	0.0000	0.0000	0.0000	0.0000	.0198
TOTAL COST ($/MI)	.0301	.0310	.0320	.0332	.0198
56MPH RANGE TO 0.8 DIS.(MI)	0.0000	0.0000	0.0000	0.0000	110.2977
COMBINED ANNUAL USAGE:					
	0.0000	0.0000	0.0000	0.0000	0.0000
ENERGY TO WHEELS (KJ/MI)	551.6843	570.9372	592.8171	617.8805	636.1880
ENERGY TO SPU (KJ/MI)	2758.5144	1754.4790	1361.1872	1361.8476	0.0000
ENERGY TO CHARGER (KJ/MI)	0.0000	378.7074	601.9709	657.2018	1303.0194
ENERGY TO SPU+CHARGER(KJ/MI)	2758.5144	2133.1864	1963.1581	2019.0494	1303.0194
FUEL TO SPU (GAL/MI)	.0234	.0149	.0116	.0116	0.0000
FUEL COST ($/MI)	.0352	.0224	.0174	.0174	0.0000
ELECTRICITY COST ($/MI)	0.0000	.0063	.0100	.0110	.0217
TOTAL COST ($/MI)	.0352	.0287	.0274	.0283	.0217

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TABLE 4-5

SIMULATED RESULTS FOR VEHICLE C/SERIES CONFIGURATION

MISSION/VEHICLE C SERIES CONF. 5/09/79

FRACTION SIZED ELECTRIC, X	0.0000	.2500	.5216	.7853	1.0000
WEIGHT OF BATTERY (LB)	0.0000	740.0000	1666.0000	2717.0000	3642.0000
WEIGHT OF PROP.SYSTEM WP (LB)	3071.5300	3429.5800	3880.8000	4389.7700	4740.8900
TEST WEIGHT WT (LB)	6637.0000	7102.0000	7688.0000	8349.0000	8805.0000
MAXIMUM SPU OUTPUT (KW)	125.7000	100.6000	69.3500	33.7100	0.0000
MAX. BATTERY OUTPUT (KW)	0.0000	33.5601	75.5556	123.2200	165.1701
STC:					
PEAK WHEEL POWER (KW)	0.0000	0.0000	0.0000	0.0000	0.0000
ENERGY TO WHEELS (KJ/MI)	636.3267	673.0053	710.9005	754.5797	784.5403
ENERGY TO SPU (KJ/MI)	4539.8726	1491.2338	0.0000	0.0000	0.0000
ENERGY TO CHARGER (KJ/MI)	0.0000	1084.2325	1858.1327	1984.9436	2072.1559
ENERGY TO SPU+CHARGER (KJ/MI)	4539.8726	2575.4663	1858.1327	1984.9436	2072.1559
FUEL TO SPU (GAL/MI)	.0386	.0127	0.0000	0.0000	0.0000
FUEL COST ($/MI)	.0579	.0190	0.0000	0.0000	0.0000
ELECTRICITY COST ($/MI)	0.0000	.0181	.0310	.0331	.0345
TOTAL COST ($/MI)	.0579	.0371	.0310	.0331	.0345
STC RANGE TO .8 DISCHARGE (MI)	0.0000	39.6196	52.0474	79.4588	102.0277
56 MPH:					
WHEEL POWER (KW)	12.5865	13.0811	13.7043	14.4073	14.8923
ENERGY TO WHEELS (KJ/MI)	809.1328	840.9260	880.9923	926.1866	957.3645
ENERGY TO SPU (KJ/MI)	3032.7794	3147.7364	3292.6070	3456.0190	0.0000
ENERGY TO CHARGER (KJ/MI)	0.0000	0.0000	0.0000	0.0000	1586.1116
ENERGY TO SPU+CHARGER (KJ/MI)	3032.7794	3147.7364	3292.6070	3456.0190	1586.1116
FUEL TO SPU (GAL/MI)	.0258	.0268	.0280	.0294	0.0000
FUEL COST ($/MI)	.0387	.0401	.0420	.0441	0.0000
ELECTRICITY COST ($/MI)	0.0000	0.0000	0.0000	0.0000	.0264
TOTAL COST ($/MI)	.0387	.0401	.0420	.0441	.0264
56MPH RANGE TO 0.8 DIS. (MI)					
COMBINED ANNUAL USAGE:					
ENERGY TO WHEELS (KJ/MI)	726.7399	760.8669	799.9019	844.3753	874.9741
ENERGY TO SPU (KJ/MI)	3751.3522	2357.9697	1722.8734	1808.4081	0.0000
ENERGY TO CHARGER (KJ/MI)	0.0000	516.9269	885.8553	946.2954	1817.8233
ENERGY TO SPU+CHARGER (KJ/MI)	3751.3522	2874.8966	2608.7287	2754.7035	1817.8233
FUEL TO SPU (GAL/MI)	.0319	.0200	.0146	.0154	0.0000
FUEL COST ($/MI)	.0478	.0301	.0220	.0231	0.0000
ELECTRICITY COST ($/MI)	0.0000	.0086	.0148	.0158	.0303
TOTAL COST ($/MI)	.0478	.0387	.0367	.0388	.0303
Energy to SPU + Charger KW/MI	1.042	.7991	.7246	.7652	.5049

TABLE 4-6

SIMULATED RESULTS FOR VEHICLE D/SERIES CONFIGURATION

VEHICLE/MISSION D SERIES CONF. 5/11/79

FRACTION SIZED ELECTRIC, X	0.0000	.2500	.5000	.7500	1.0000
WEIGHT OF BATTERY (LB)	0.0000	736.7870	1538.2300	2413.2270	3334.6116
WEIGHT OF PROP.SYSTEM WP (LB)	3131.0695	3417.2623	3731.6318	4074.8670	4359.2341
TEST WEIGHT WT (LB)	8240.3240	8612.0030	9020.2750	9466.0350	9835.3430
MAXIMUM SPU OUTPUT (KW)	128.3930	100.2400	69.7600	36.4800	0.0000
MAX. BATTERY OUTPUT (KW)	0.0000	33.4144	69.7610	109.4434	446.0473
STC:					
PEAK WHEEL POWER (KW).	64.1150	66.8855	69.9287	73.2513	76.0050
ENERGY TO WHEELS (KJ/MI)	1113.8153	1138.6995	1166.0153	1195.8255	1220.5303
ENERGY TO SPU (KJ/MI)	6604.0392	2198.8643	798.4502	0.0000	0.0000
ENERGY TO CHARGER (KJ/MI)	0.0000	1598.1413	2299.3842	2740.3041	2811.6099
ENERGY TO SPU+CHARGER (KJ/MI)	6604.0392	3797.0056	3097.8343	2740.3041	2811.6099
FUEL TO SPU (GAL/MI)	.0561	.0187	.0068	0.0000	0.0000
FUEL COST ($/MI)	.0842	.0280	.0102	0.0000	0.0000
ELECTRICITY COST ($/MI)	0.0000	.0266	.0383	.0457	.0469
TOTAL COST ($/MI)	.0842	.0547	.0485	.0457	.0469
STC RANGE TO .8 DISCHARGE (MI)	0.0000	26.7625	38.8338	51.1211	62.8479
56 MPH:					
WHEEL POWER (KW)	24.3264	24.7217	25.1560	25.6301	26.0229
ENERGY TO WHEELS (KJ/MI)	1563.8422	1589.2548	1617.1695	1647.6472	1672.8977
ENERGY TO SPU (KJ/MI)	5761.6338	5853.5200	5954.4528	6064.6533	0.0000
ENERGY TO CHARGER (KJ/MI)	0.0000	0.0000	0.0000	0.0000	2735.9793
ENERGY TO SPU+CHARGER (KJ/MI)	5761.6338	5853.5200	5954.4528	6064.6533	2735.9793
FUEL TO SPU (GAL/MI)	.0490	.0498	.0506	.0516	0.0000
FUEL COST ($/MI)	.0735	.0746	.0759	.0773	0.0000
ELECTRICITY COST ($/MI)	0.0000	0.0000	0.0000	0.0000	.0456
TOTAL COST ($/MI)	.0735	.0746	.0759	.0773	.0456
56MPH RANGE TO 0.8 DIS. (MI)	0.0000	0.0000	0.0000	0.0000	70.7511
COMBINED ANNUAL USAGE:					
	0.0000	0.0000	0.0000	0.0000	0.0000
ENERGY TO WHEELS (KJ/MI)	1349.1765	1374.3400	1401.9720	1432.1343	1457.1267
ENERGY TO SPU (KJ/MI)	6163.4665	4110.2499	3495.0745	3171.8946	0.0000
ENERGY TO CHARGER (KJ/MI)	0.0000	762.3131	1096.7907	1307.0885	2772.0537
ENERGY TO SPU+CHARGER (KJ/MI)	6163.4665	4872.5630	4591.8651	4478.9831	2772.0537
FUEL TO SPU (GAL/MI)	.0524	.0349	.0297	.0270	0.0000
FUEL COST ($/MI)	.0786	.0524	.0446	.0404	0.0000
ELECTRICITY COST ($/MI)	0.0000	.0127	.0183	.0218	.0462
TOTAL COST ($/MI)	.0786	.0651	.0628	.0622	.0462

TABLE 4-7

SIMULATED RESULTS FOR VEHICLE C/PARALLEL CONFIGURATION

MISSION VEHICLE C   PARALLEL CONFIGURATION PERFORMANCE   5/24/79

FRACTION SIZED ELECTRIC, X		0.0000	.2500	.5000	.7500	1.0000
WEIGHT OF BATTERY	(LB)	0.0000	490.9400	1203.2400	2328.4200	4248.7300
WEIGHT OF PROP.SYSTEM WP	(LB)	312.8649	1325.2162	2132.0376	3406.1643	5492.2175
TEST WEIGHT WT	(LB)	3054.3180	4369.0600	5416.8800	7071.5900	9780.7500
MAXIMUM SPU OUTPUT	(KW)	50.6240	52.9000	43.2200	27.8800	0.0000
MAX. BATTERY OUTPUT	(KW)	0.0000	22.2649	54.5687	105.5973	192.6862
STC:						
PEAK WHEEL POWER	(KW)	23.7229	33.3162	40.9962	53.1289	73.0001
ENERGY TO WHEELS	(KJ/MI)	404.1480	493.2162	562.1274	670.9531	849.1945
ENERGY TO SPU	(KJ/MI)	1950.5087	697.5374	17.5362	.0000	.0000
ENERGY TO CHARGER	(KJ/MI)	0.0000	874.0437	1515.5745	1870.0432	2432.5964
ENERGY TO SPU+CHARGER	(KJ/MI)	1950.5087	1571.5811	1533.1107	1870.0432	2432.5964
FUEL TO SPU	(GAL/MI)	.0166	.0059	.0001	.0000	.0000
FUEL COST	($/MI)	.0249	.0089	.0002	.0000	.0000
ELECTRICITY COST	($/MI)	0.0000	.0146	.0253	.0312	.0405
TOTAL COST	($/MI)	.0249	.0235	.0255	.0312	.0405
STC RANGE TO .8 DISCHARGE	(MI)	0.0000	32.6059	46.0867	72.2787	101.3887
56 MPH:						
WHEEL POWER	(KW)	8.7761	10.1744	11.2888	13.0487	15.9301
ENERGY TO WHEELS	(KJ/MI)	564.1756	654.0679	725.7100	838.8468	1024.0791
ENERGY TO SPU	(KJ/MI)	1627.0189	1774.4652	1968.8281	2275.7645	0.0000
ENERGY TO CHARGER	(KJ/MI)	.0000	47.6190	47.6190	47.6190	1779.9383
ENERGY TO SPU+CHARGER	(KJ/MI)	1627.0189	1822.0843	2016.4471	2323.3836	1779.9383
FUEL TO SPU	(GAL/MI)	.0138	.0151	.0167	.0193	0.0000
FUEL COST	($/MI)	.0207	.0226	.0251	.0290	0.0000
ELECTRICITY COST	($/MI)	.0000	.0008	.0008	.0008	.0297
TOTAL COST	($/MI)	.0207	.0234	.0259	.0298	.0297
56MPH RANGE TO 0.8 DIS.	(MI)	0.0000	598.4781	1466.3041	2838.4495	138.5654
COMBINED ANNUAL USAGE:						
ENERGY TO WHEELS	(KJ/MI)	487.8424	577.3631	647.7110	758.8006	940.7077
ENERGY TO SPU	(KJ/MI)	1781.3235	1260.9142	1038.4184	1190.7546	.0000
ENERGY TO CHARGER	(KJ/MI)	.0000	441.7134	747.5655	916.4911	2091.0750
ENERGY TO SPU+CHARGER	(KJ/MI)	1781.3235	1702.6276	1785.9839	2107.2458	2091.0750
FUEL TO SPU	(GAL/MI)	.0151	.0107	.0088	.0101	.0000
FUEL COST	($/MI)	.0227	.0161	.0132	.0152	.0000
ELECTRICITY COST	($/MI)	.0000	.0074	.0125	.0153	.0349
TOTAL COST	($/MI)	.0227	.0234	.0257	.0305	.0349

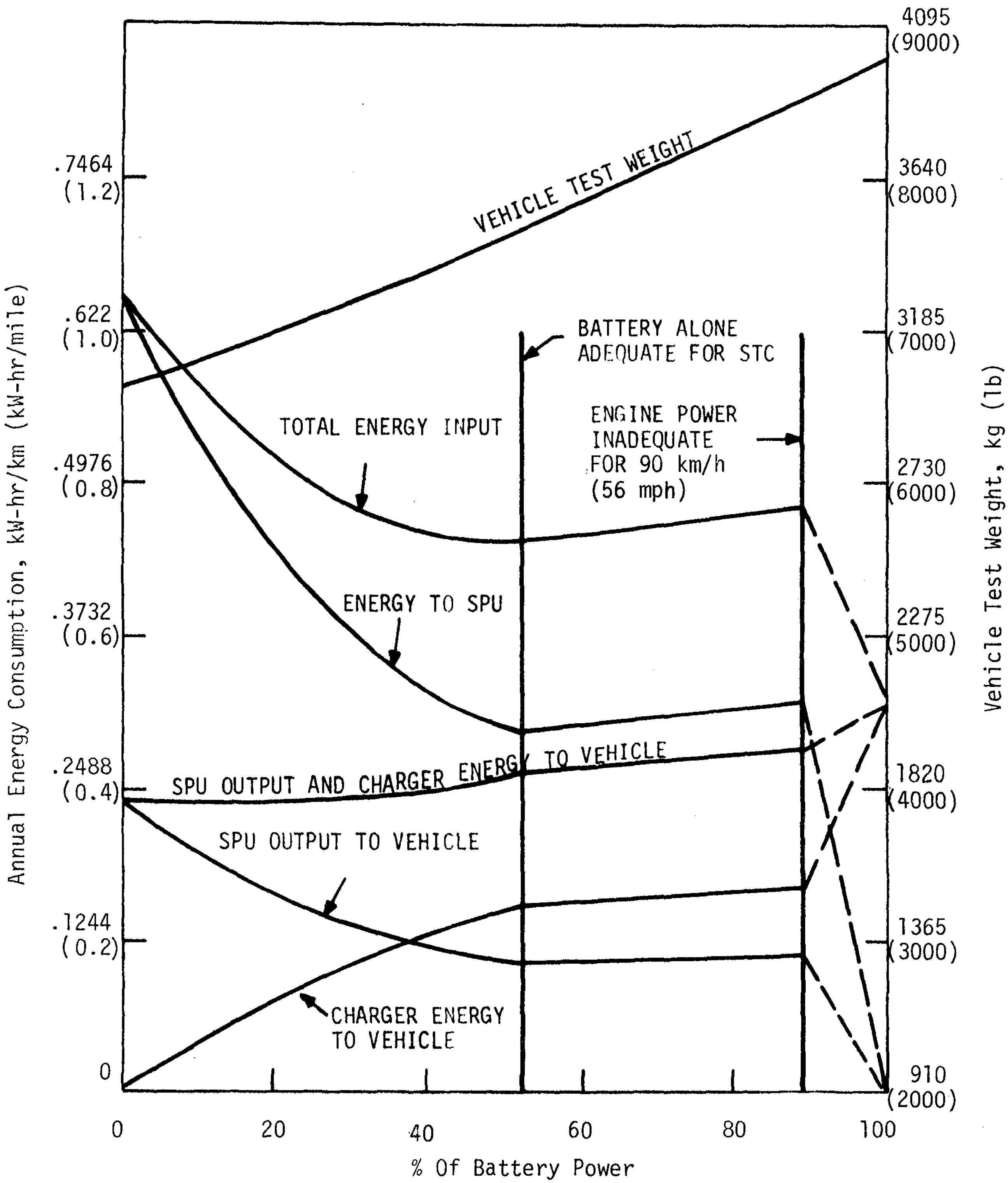

Fig. 4-5a Annual Specific Energy Consumption, Vehicle C Series System

793779-1

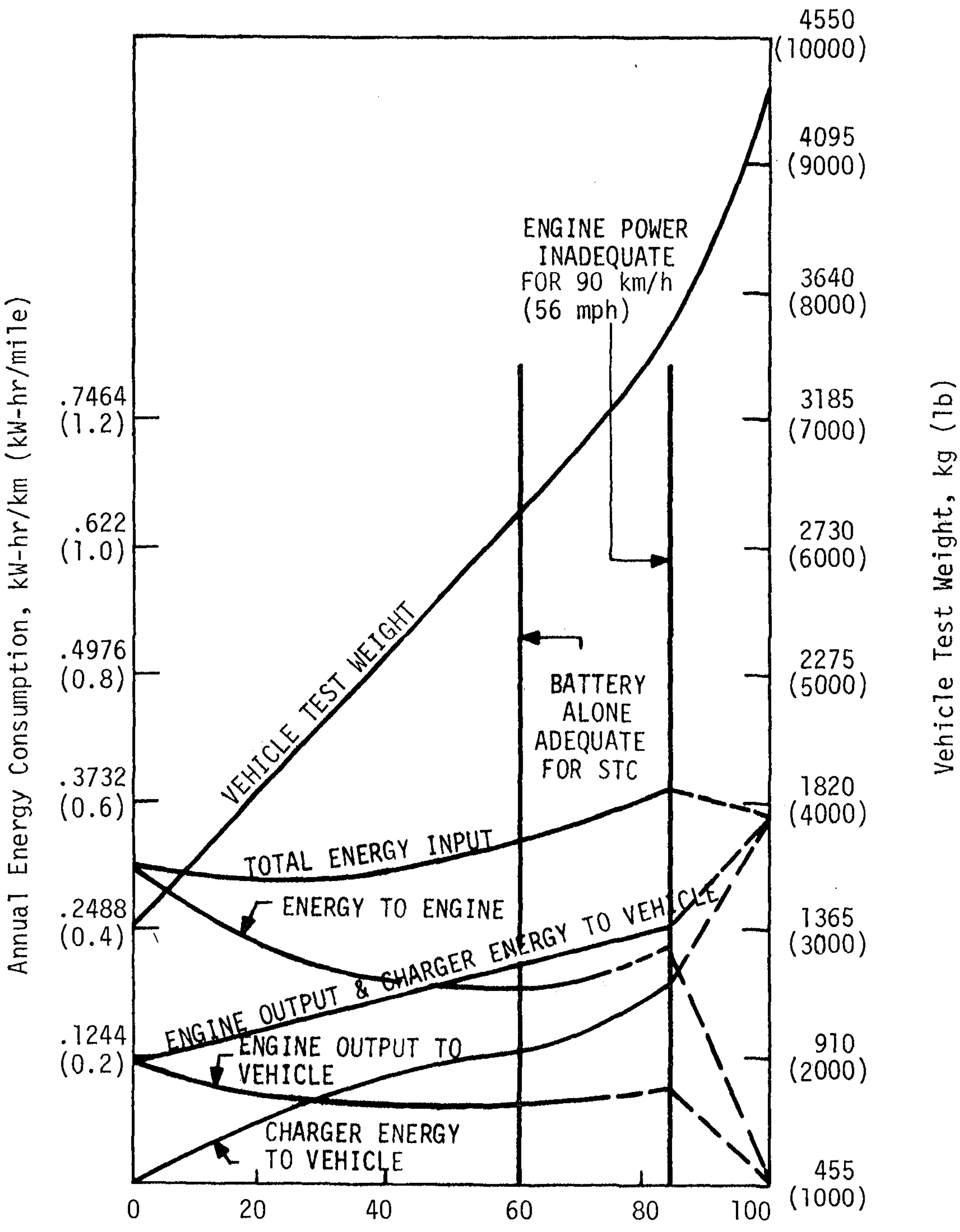

Fig. 4-5b Annual Specific Energy Consumption
Vehicle C Parallel System

4.4 Selection of Battery Power Fraction

As a compromise between vehicle weight, gasoline consumption, and total energy consumption, the battery power fractions for each mission/vehicle were selected as follows:

Vehicle	Battery Power
A	60%
B	50%
C	50%
D	60%

4.4.1 Optimum Battery and Engine Power Combination Results

Based on the selected battery power fraction, the vehicle weight and specific energy consumption were recalculated for each of the vehicles in both the series and parallel concepts. The computer outputs are summarized as shown in Tables 4-8 and 4-9. A summary comparison of series versus parallel systems for vehicles A and C is presented in Figure 4-6. Any annualized kilometers per liter of fuel (mpg) included in this report did not include petroleum used by utilities to generate electricity.

4.4.2 Energy Flow Charts

The detailed figures of the computer output were utilized to generate energy flow charts, power variations with time, and specific energy consumptions. Vehicles A and C in the series and parallel configurations are plotted as shown in Figures 4-7 through 4-10. The energy flow diagrams presented in these figures represent the energy losses during a complete STC; that is, 360 degrees represent pictorially the 108 seconds of the STC. Energy given is the total energy expended during the different phases of the STC; i.e., acceleration, cruise, coast, brake, idle.

4.5 Conclusions and Recommendations

From the parametric studies performed, the following conclusions were determined.

- Both series and parallel systems can meet requirements.
- The parallel system utilizes less fuel and weighs less overall.

TABLE 4-8

SERIES CONFIGURATION PERFORMANCE 7/24/79

MISSION/VEHICLE	A	B	C	D
FRACTION SIZED ELECTRIC, X	.60	.50	.50	.60
WEIGHT OF BATTERY(LB)	502.	861.	1337.	1714.
WEIGHT OF PROP.SYSTEM WP (LB)	1163.	1970.	2882.	3197.
TEST WEIGHT WT (LB)	2387.	4207.	6391.	8326.
MAXIMUM SPU OUTPUT (KW)	15.19	39.05	60.62	51.82
MAX. BATTERY OUTPUT (KW)	22.79	39.05	60.62	77.73
STC				
PEAK WHEEL POWER (KW)	19.38	32.24	48.32	66.49
AVERAGE POWER TO WHEELS (KW)	2.87	3.85	5.16	9.31
ENERGY TO WHEELS (KJ/MI)	344.	462.	619.	1118.
ENERGY TO SPU (KJ/MI)	48.82	53.17	1686.	131.4
ENERGY TO CHARGER (KJ/MI)	833.	1154.	1558.	2445.
ENERGY TO SPU+CHARGER(KJ/MI)	881.82	1207.	3244.	2576.
FUEL TO SPU (GAL/MI)	.000415	.000452	.01324	.001117
FUEL COST ($/MI)	.000622	.000678	.01986	.001676
ELECTRICITY COST ($/MI)	.01388	.01922	.02597	.04075
TOTAL COST ($/MI)	.01450	.01990	.04583	.04243
STC RANGE TO .8 DISCHARGE(MI)	34-37	44-49	57-65	40-45
56 MPH				
WHEEL POWER (KW)	7.641	9.58	12.33	24.42
ENERGY TO WHEELS (KJ/MI)	491.	615.	792.	1570.
ENERGY TO SPU (KJ/MI)	2139.	2639.	3320.	6315.
ENERGY TO CHARGER (KJ/MI)	0.	0.	0.	0.
ENERGY TO SPU+CHARGER(KJ/MI)	2139.	2639.	3320.	6315.
FUEL TO SPU (GAL/MI)	.01812	.02243	.02822	.05368
FUEL COST ($/MI)	.02727	.03365	.04233	.08051
ELECTRICITY COST ($/MI)	0.	0.	0.	0.
TOTAL COST ($/MI)	.02727	.03365	.04233	.08051
56MPH RANGE TO 0.8 DIS.(MI)	INF.	INF.	INF.	INF.
COMBINED ANNUAL USAGE				
AVERAGE POWER TO WHEELS (KW)	5.37	6.85	8.91	17.21
ENERGY TO WHEELS(KJ/MI)	420.90	542.56	709.50	1354.45
ENERGY TO SPU (KJ/MI)	1142.25	1405.89	2540.79	3366.22
ENERGY TO CHARGER (KJ/MI)	397.23	550.31	742.96	1165.95
FUEL TO SPU+CHARGER(KJ/MI)	1539.49	1956.12	3283.76	4531.98
FUEL TO SPU (GAL/MI)	.00968	.01195	.02108	.02861
FUEL COST ($/MI)	.01456	.01793	.03162	.04292
ELECTRICITY COST ($/MI)	.00662	.00917	.01238	.01943
TOTAL COST ($/MI)	.02118	.02709	.04399	.06235

TABLE 4-9

PARALLEL CONFIGURATION PERFORMANCE 7/24/79

MISSION/VEHICLE	A	B	C	D
FRACTION SIZED ELECTRIC, X	.60	.50	.50	.60
WEIGHT OF BATTERY(LB)	434.	661.	1088.	1469.
WEIGHT OF PROP.SYSTEM WP (LB)	854.	1155.	1695.	2183.
TEST WEIGHT WT (LB)	1986.	3150.	4849.	7009.
MAXIMUM SPU OUTPUT (KW)	10.65	24.49	38.61	36.69
MAXIMUM BATTERY OUTPUT (KW)	19.67	29.98	49.35	66.61
STC:				
PEAK WHEEL POWER (KW)	16.54	24.77	36.98	54.31
AVERAGE POWER TO WHEELS (KW)	2.65	3.28	4.33	8.38
ENERGY TO WHEELS (KJ/MI)	318.	394.	520.	1006.
ENERGY TO SPU (KJ/MI)	0.	0.	0.	0.
ENERGY TO CHARGER (KJ/MI)	784.	997.	1310.	2199.
ENERGY TO SPU+CHARGER(KJ/MI)	784.	997.	1310.	2199.
FUEL TO SPU (GAL/MI)	0.	0.	0.	0.
FUEL COST ($/MI)	0.	0.	0.	0.
ELECTRICITY COST ($/MI)	.0131	.0166	.0218	.0367
TOTAL COST ($MI)	.0131	.0166	.0218	.0367
STC RANGE TO .8 DISCHARGE(MI)	30-32	38-42	51-57	39-42
56MPH				
WHEEL POWER (KW)	7.21	8.45	10.69	22.59
ENERGY TO WHEELS (KJ/MI)	464.	543.	687.	1452.
ENERGY TO SPU (KJ/MI)	1198.	1404.	1775.	3751.
ENERGY TO CHARGER (KJ/MI)	0.	0.	0.	0.
ENERGY TO SPU+CHARGER(KJ/MI)	1198.	1404.	1775.	3751.
FUEL TO SPU (GAL/MI)	.01019	.01193	.01509	.03188
FUEL COST ($/MI)	.01528	.0179	.02263	.0478
ELECTRICITY COST ($/MI)	0.	0.	0.	0.
TOTAL COST ($/MI)	.01528	.0179	.02263	.0478
56MPH RANGE TO 0.8 DIS. (MI)				
COMBINED ANNUAL USUAGE				
AVERAGE POWER TO WHEELS (KW)	5.04	5.98	7.66	15.81
ENERGY TO WHEELS (KJ/MI)	394.	472.	607.	1239.
ENERGY TO SPU (KJ/MI)	627.	735.	929.	1962.
ENERGY TO CHARGER (KJ/MI)	374.	475.	625.	1049.
ENERGY TO SPU+CHARGER(KJ/MI)	1001.	1210.	1553.	3011.
FUEL TO SPU (GAL/MI)	.00533	.00624	.00789	.01668
FUEL COST ($/MI)	.00799	.00936	.01184	.02501
ELECTRICITY COST ($/MI)	.00625	.00792	.01040	.01750
TOTAL COST ($/MI)	.01424	.01728	.02223	.04251

PARAMETER	SERIES	PARALLEL
Selected % Batteries for Fuel Conservation	60	60
"STC" Range on Batteries Only, km (miles)	55-60 (34-37)	48-51 (30-32)
Annualized Kilometers per Liter of Fuel (miles per gallon)	166 (103)	301 (187)
Energy Per Year		
Fuel kW-hr	3155	1731
Wall Plug-Elec kW-hr	1097	1033
Total kW-hr	4252	2764
Vehicle Weight kg (lb)	1083 (2387)	901 (1986)

801707-1

a) Vehicle A

PARAMETER	SERIES	PARALLEL
Selected % Batteries for Fuel Conservation	50	50
"STC" Range on Batteries Only, km (miles)	92-105 (57-65)	82-92 (51-57)
Annualized Kilometers per Liter of Fuel (miles per gallon)	20 (47)	54 (126)
Energy per Year		
Fuel kW-hr	7018	2564
Wall Plug-Elec kW-hr	2053	1725
Total kW-hr	9070	4289
Vehicle Weight, kg (lb)	2908 (6391)	2206 (4849)

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b) Vehicle C

Fig. 4-6 Comparison of the Series Parallel Systems

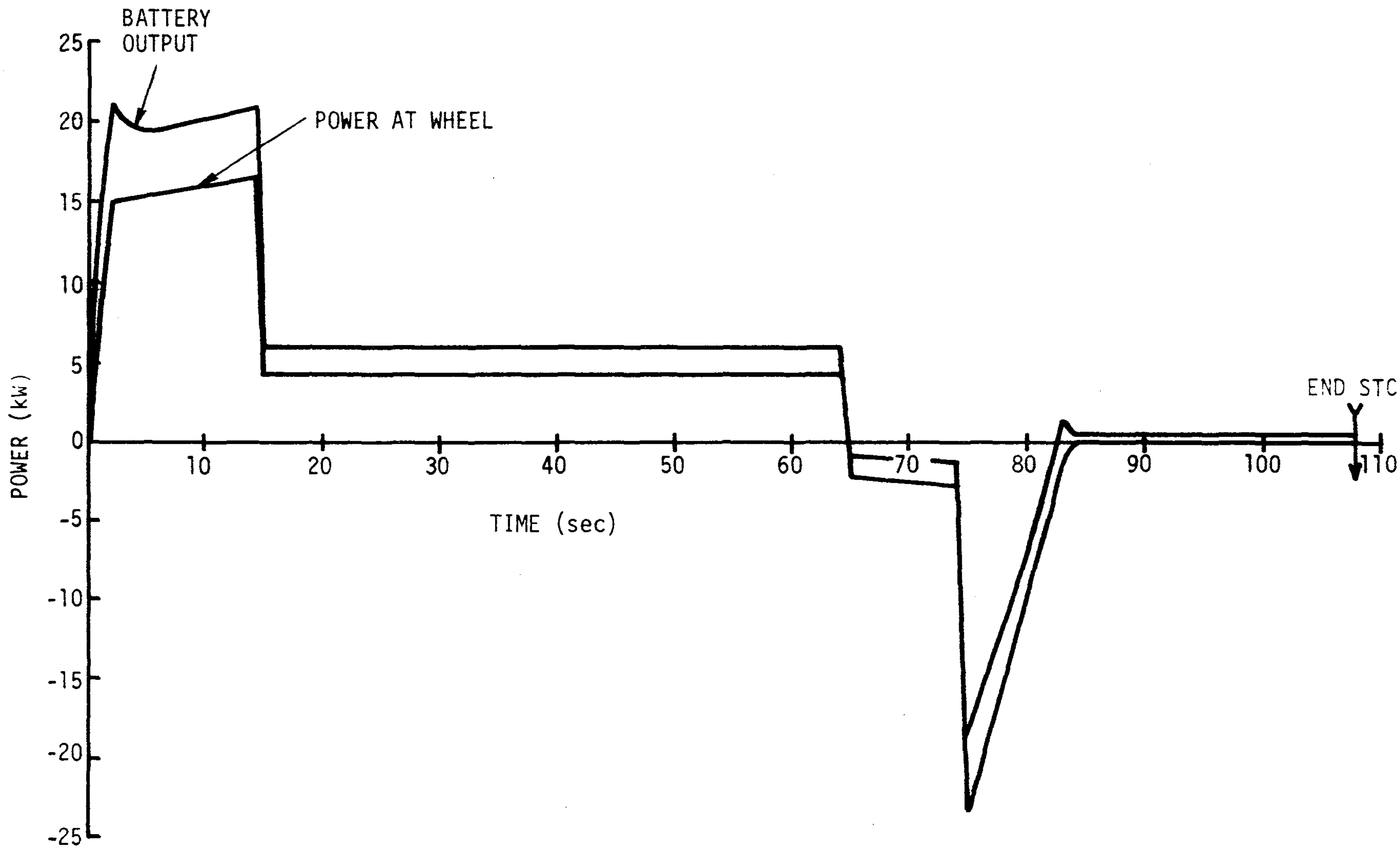

Fig. 4-7a Vehicle A, Parallel STC, Power versus Time

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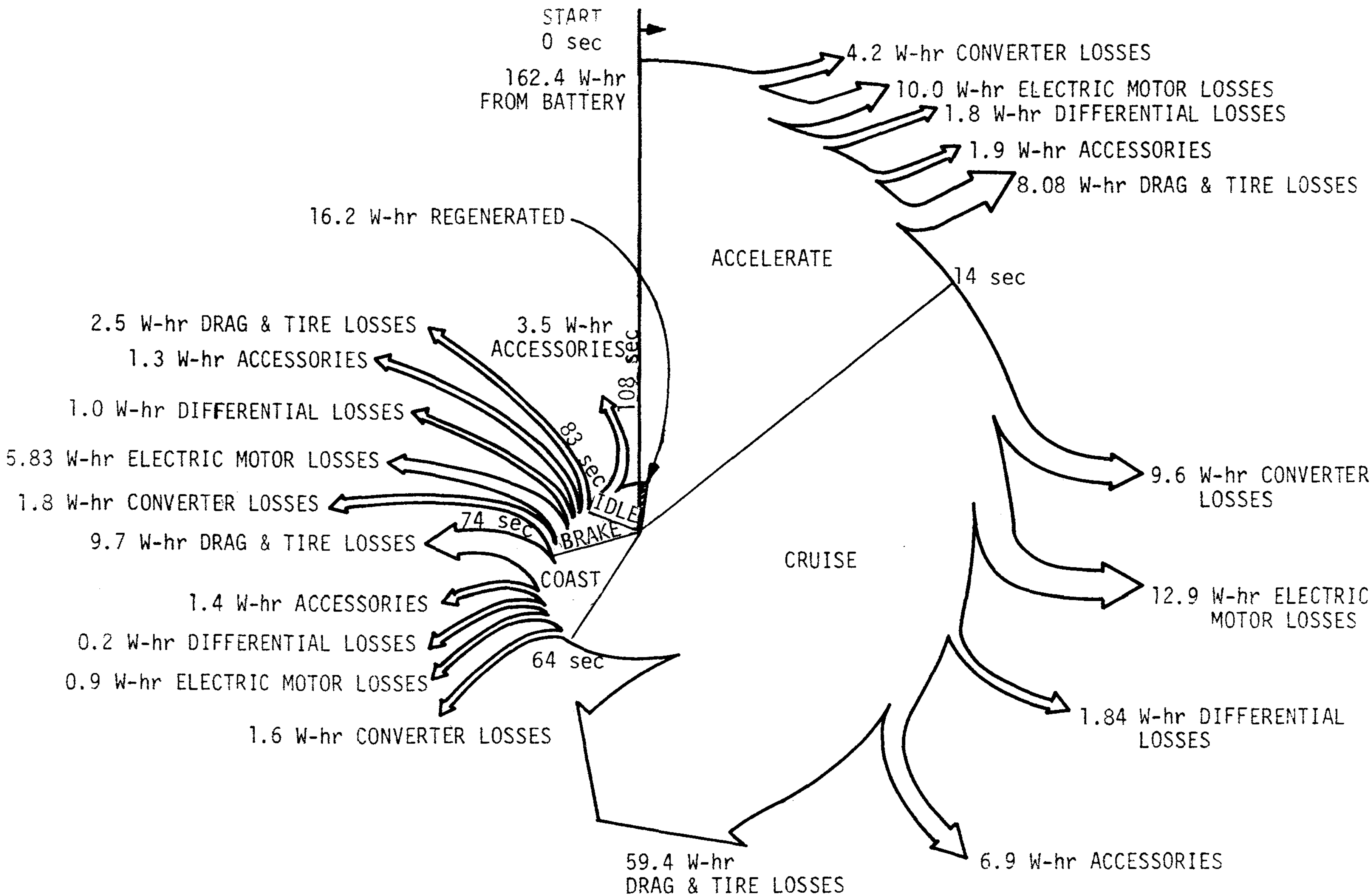
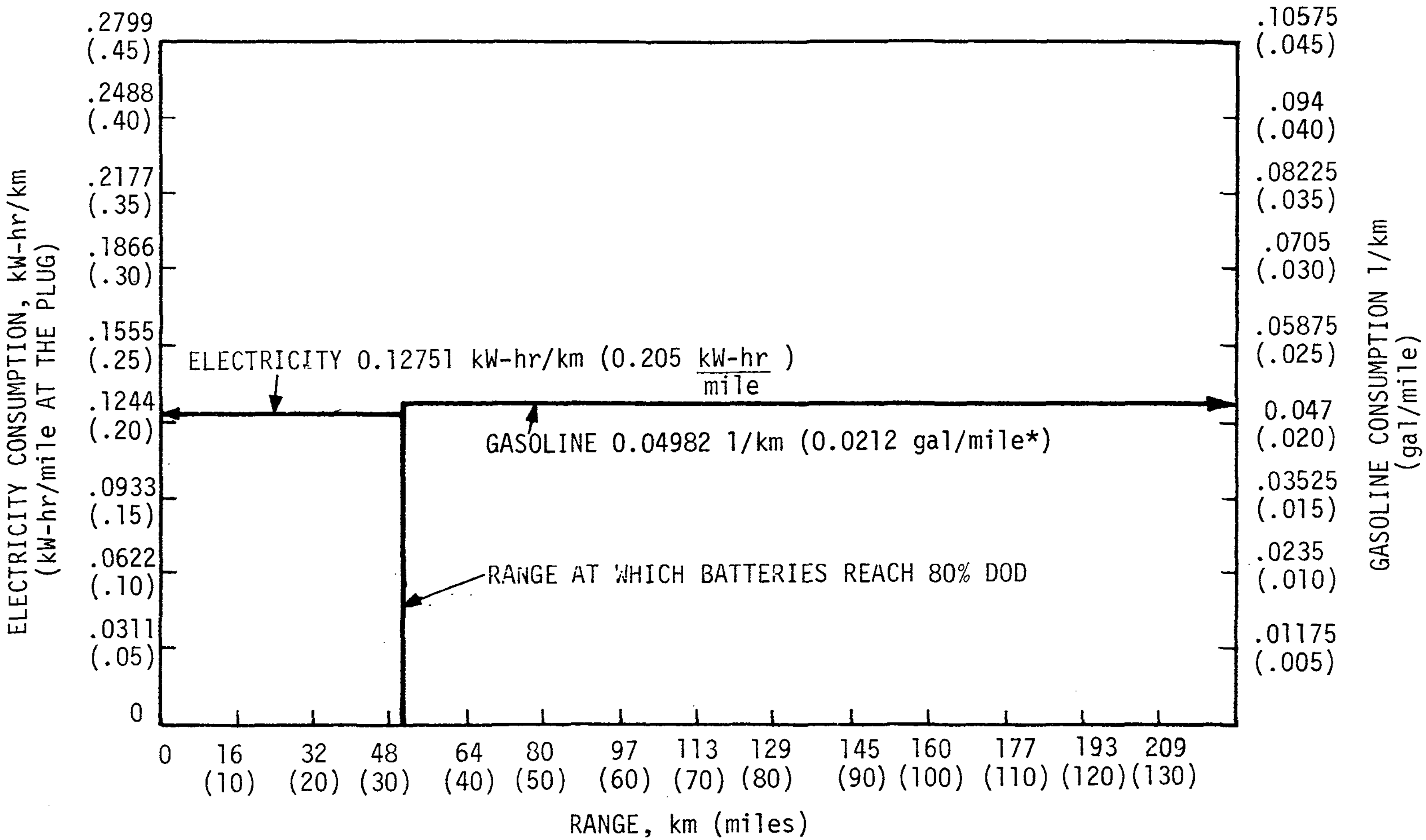


Fig. 4-7b Vehicle A, Parallel STC, Energy Flow Diagram

793781



*GASOLINE CONSUMPTION CALCULATION MADE WITHOUT
KNOWING MAXIMUM POWER OF ENGINE. η TAKEN AS 0.39

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Fig. 4-7c Vehicle A, Parallel STC, Specific Energy Consumption

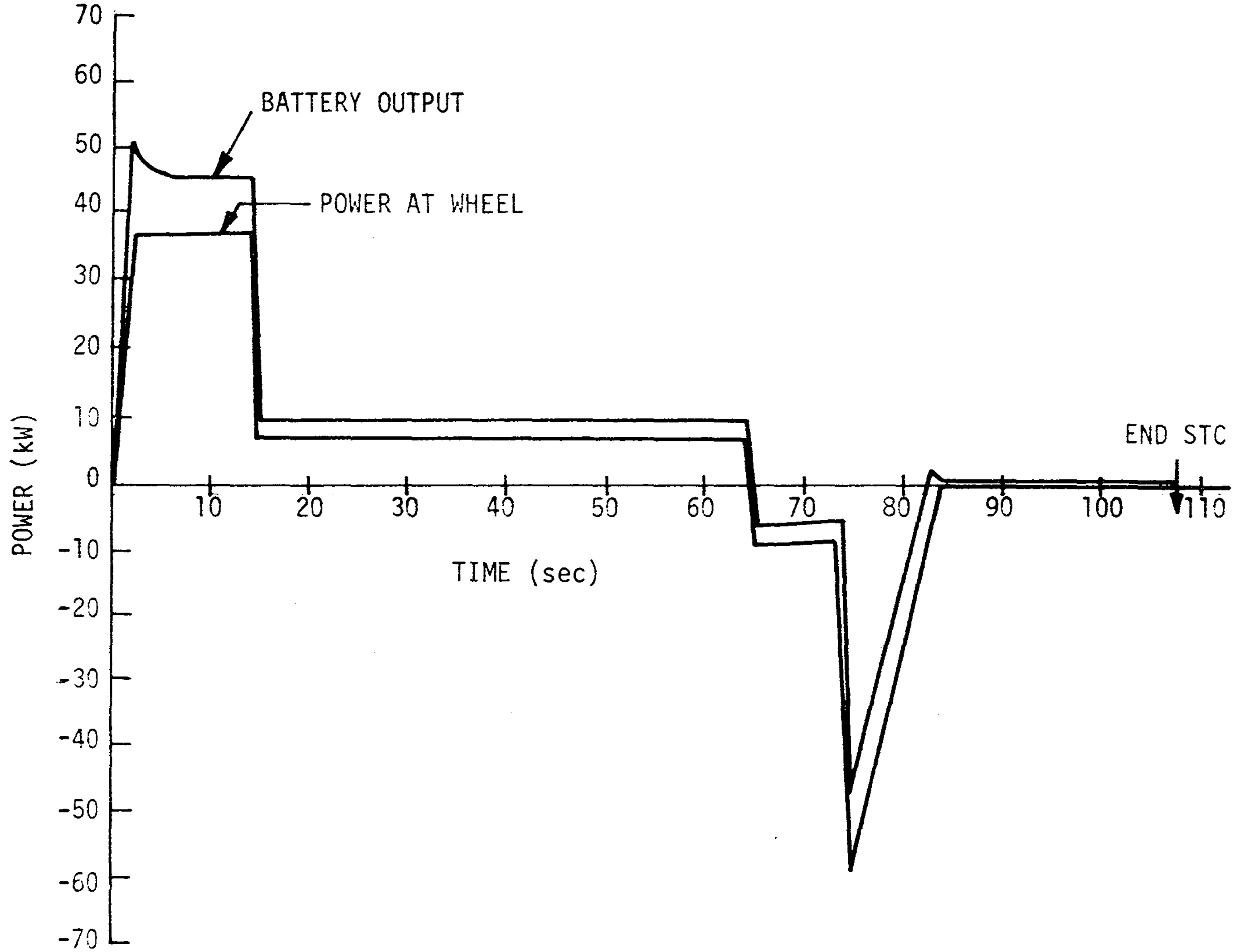


Fig. 4-8a Vehicle C, Parallel STC, Power versus Time

793749

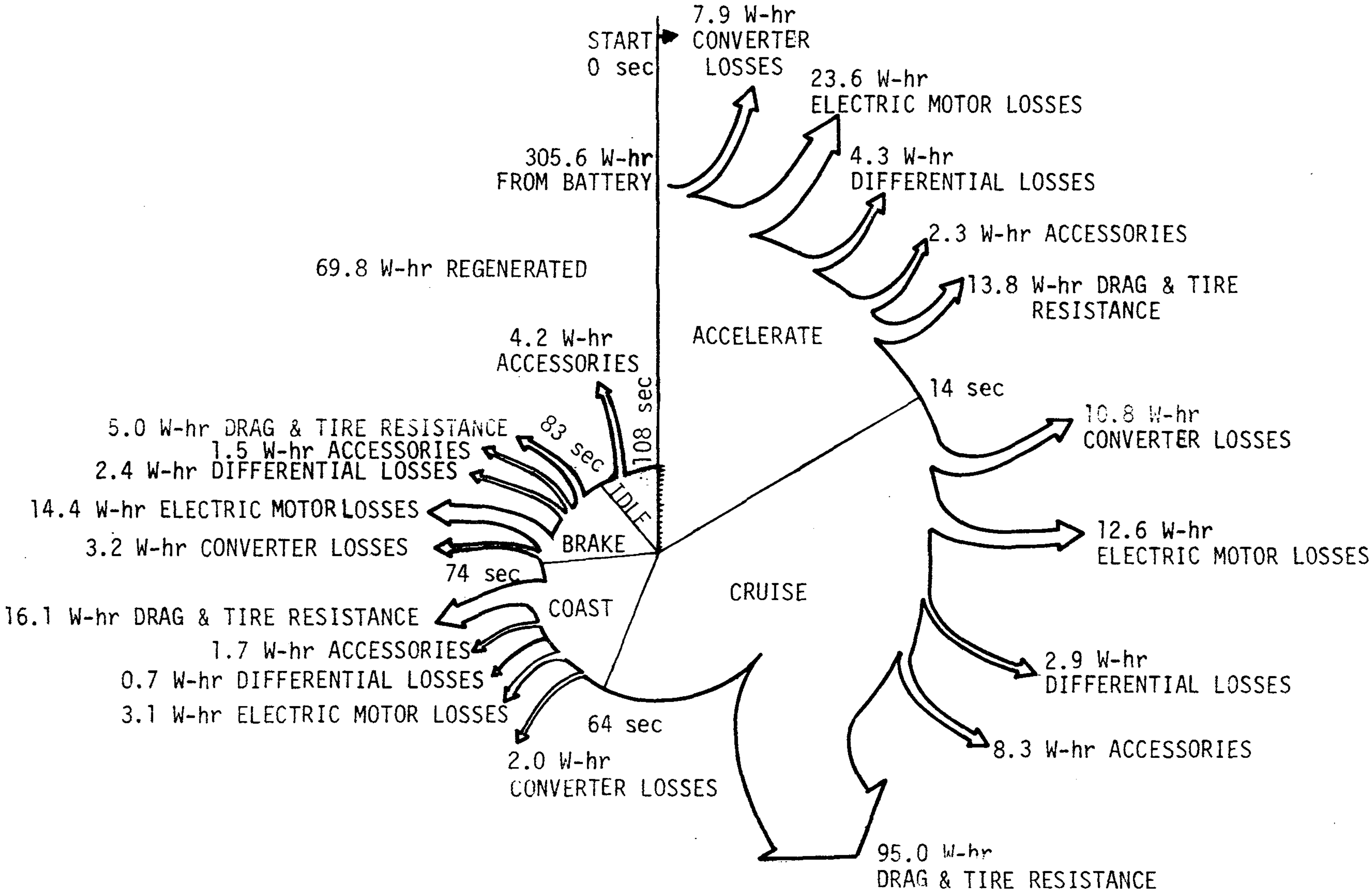
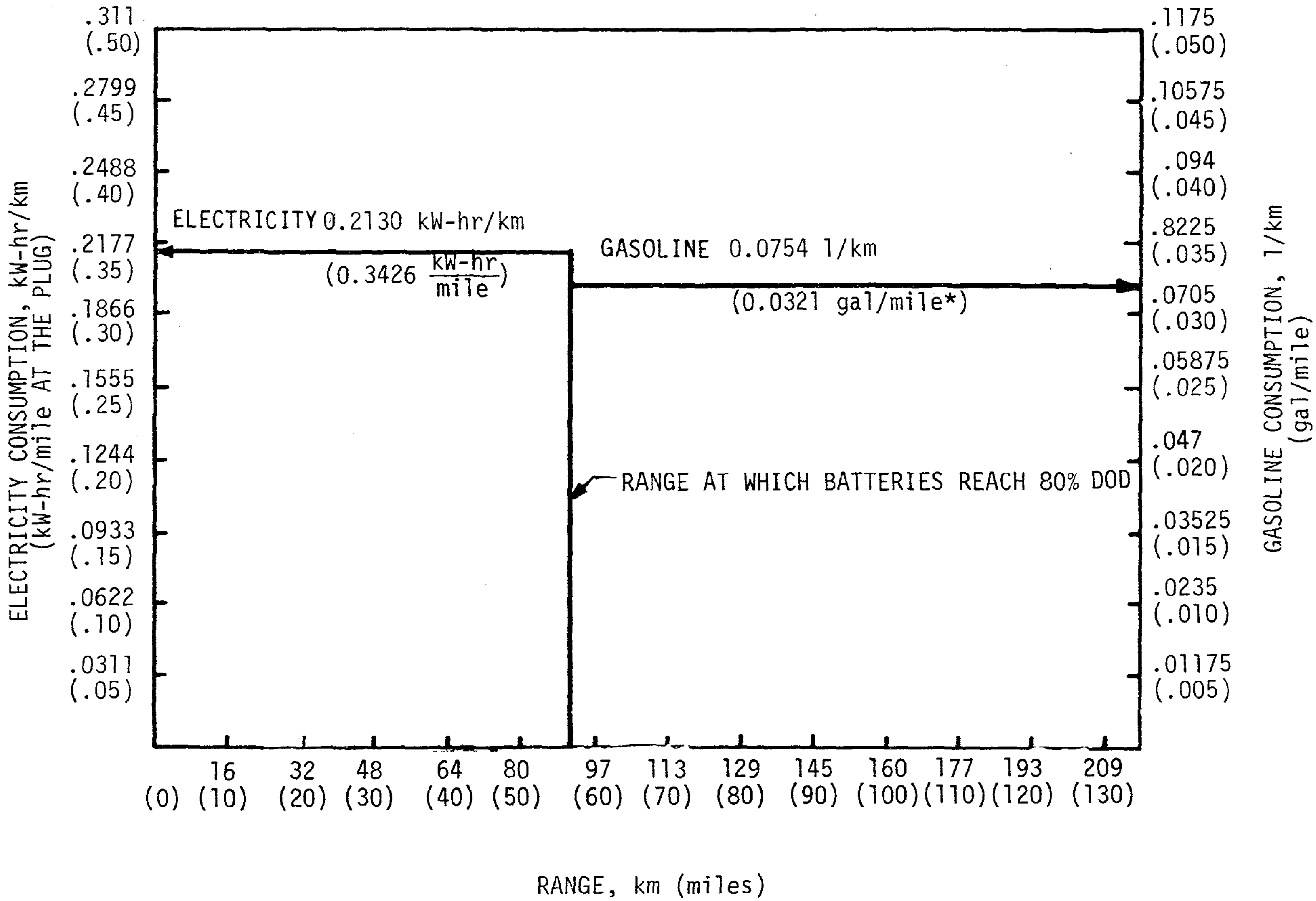


Fig. 4-8b Vehicle C, Parallel STC, Energy Flow Diagram

793782



*GASOLINE CONSUMPTION CALCULATION MADE WITHOUT KNOWING MAXIMUM POWER OF ENGINE. η TAKEN AS 0.39.

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Fig. 4-8c Vehicle C, Parallel STC, Specific Energy Consumption

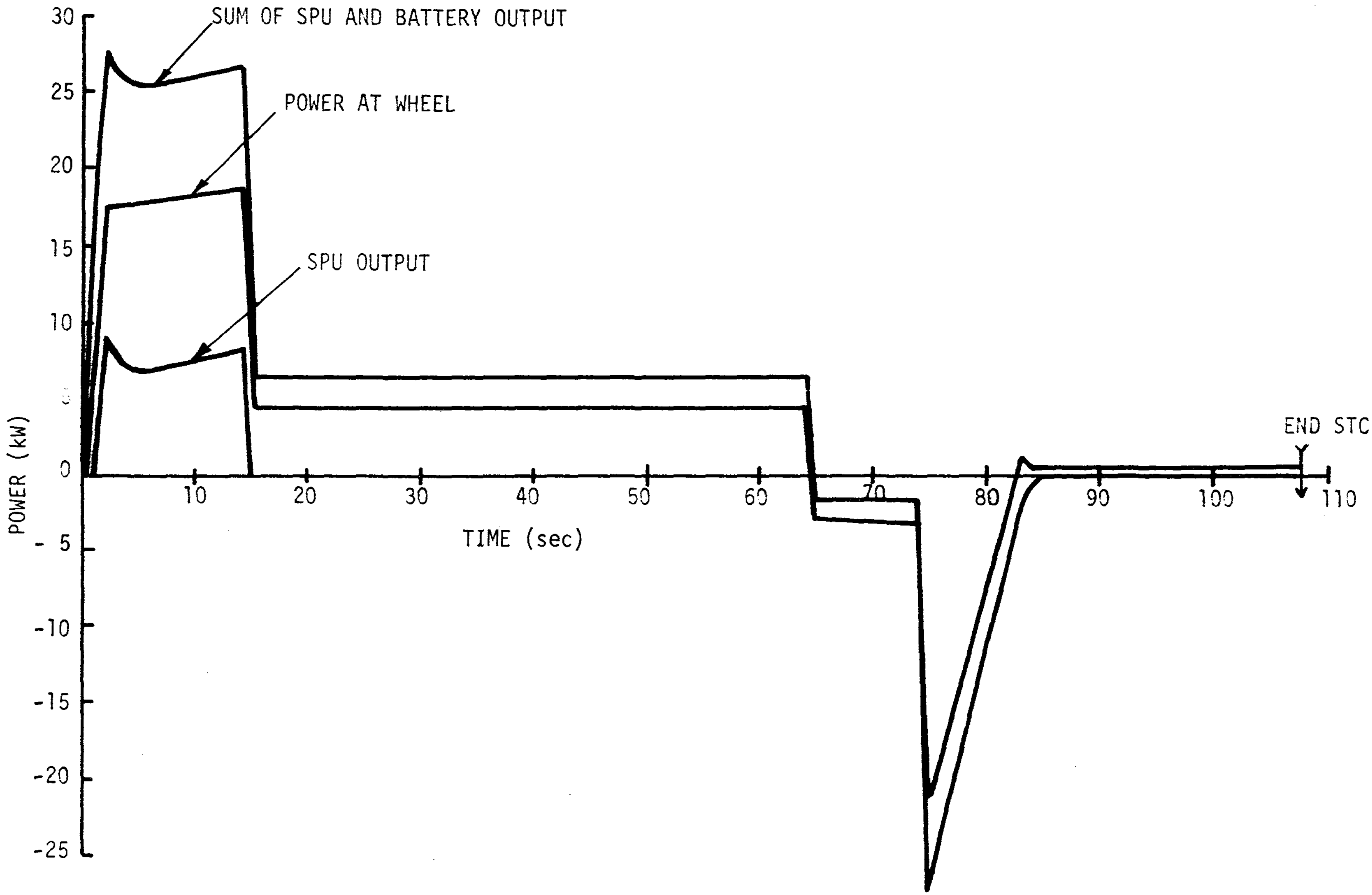

Fig. 4-9a Vehicle A, Series STC, Power versus Time

793776

ENERGY FLOW DIAGRAM VEHICLE A - SERIES - STC

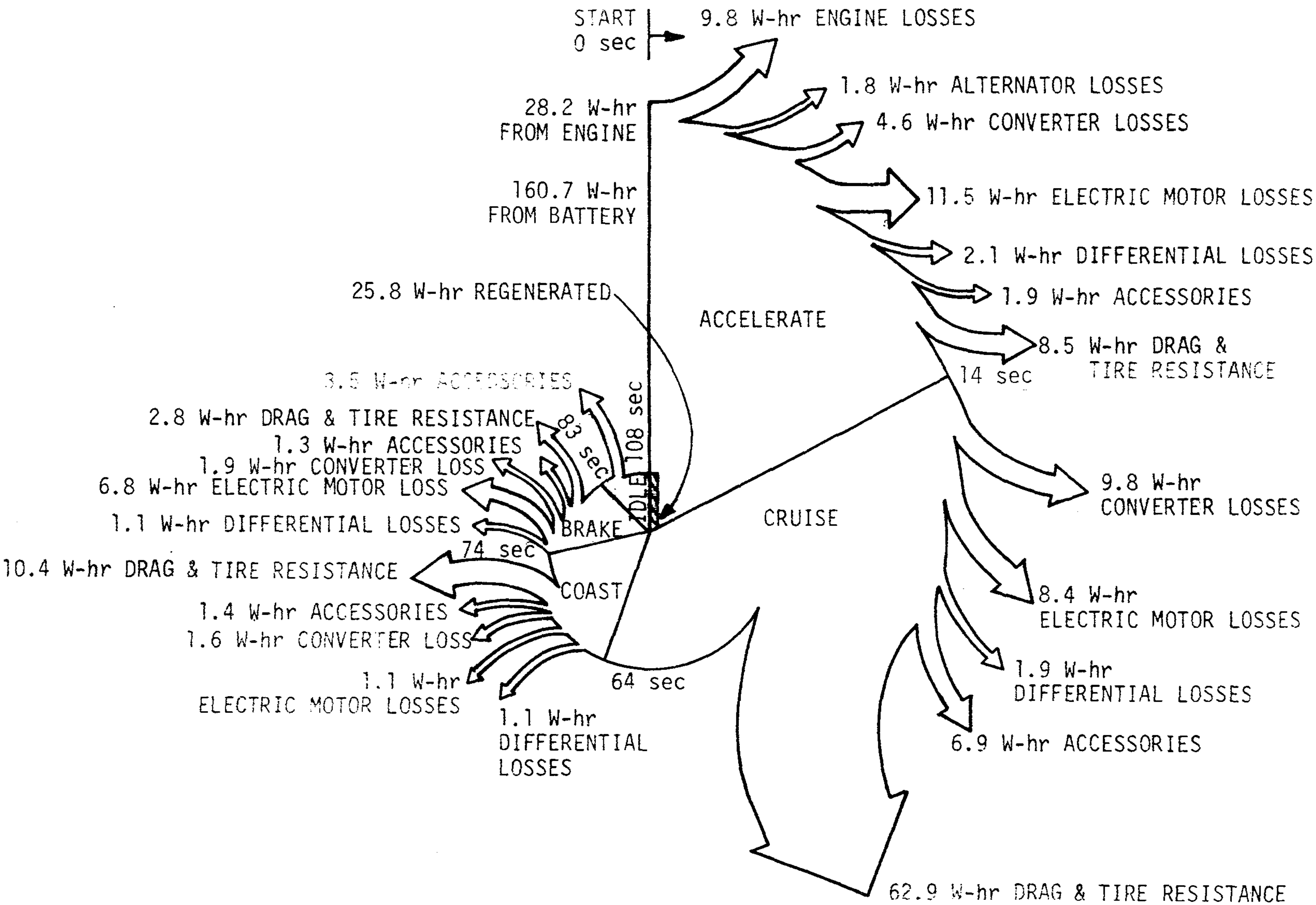

Fig. 4-9b Vehicle A, Series STC, Energy Flow Diagram

793750

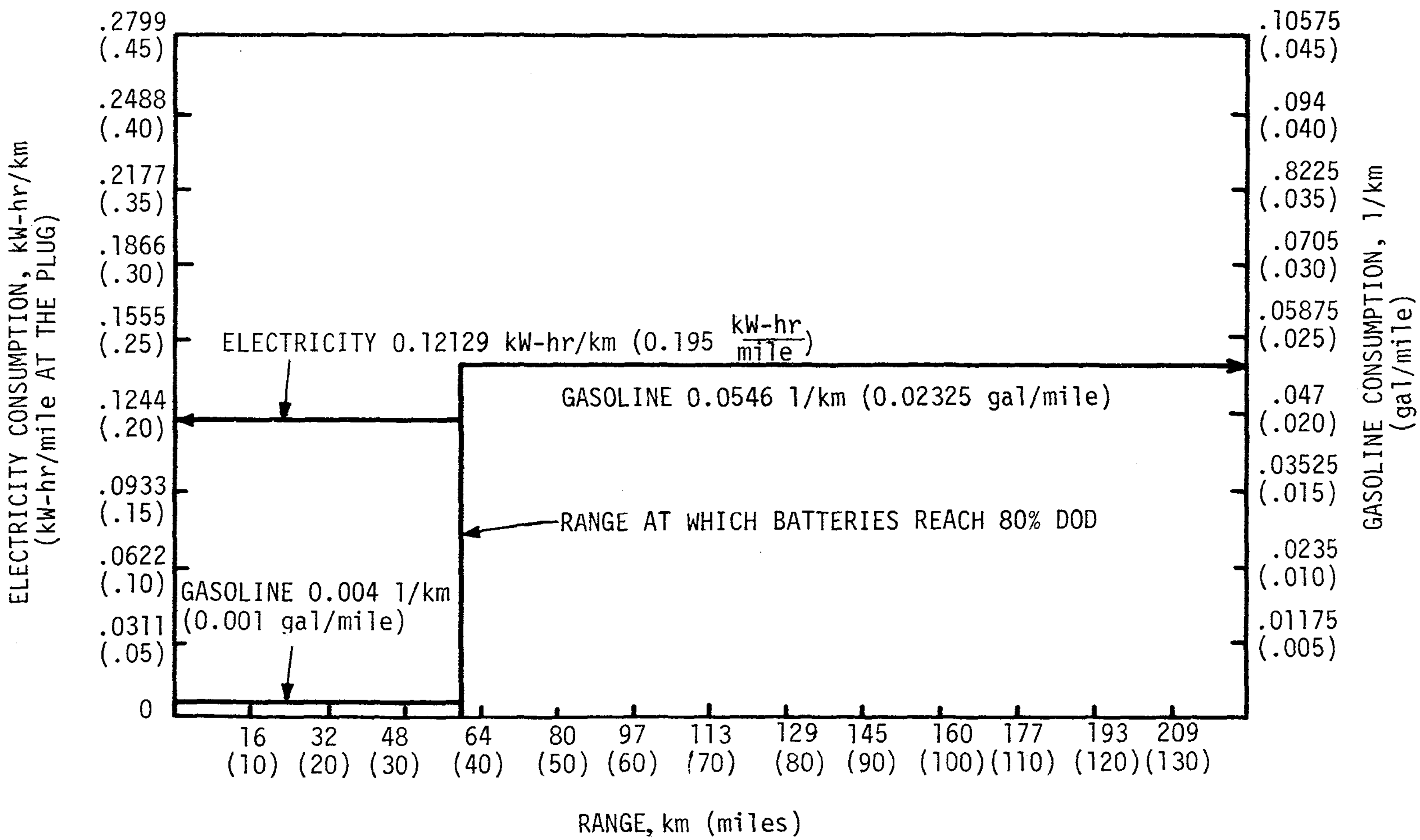

Fig. 4-9c Vehicle A, Series STC, Specific Energy Consumption

793777-1

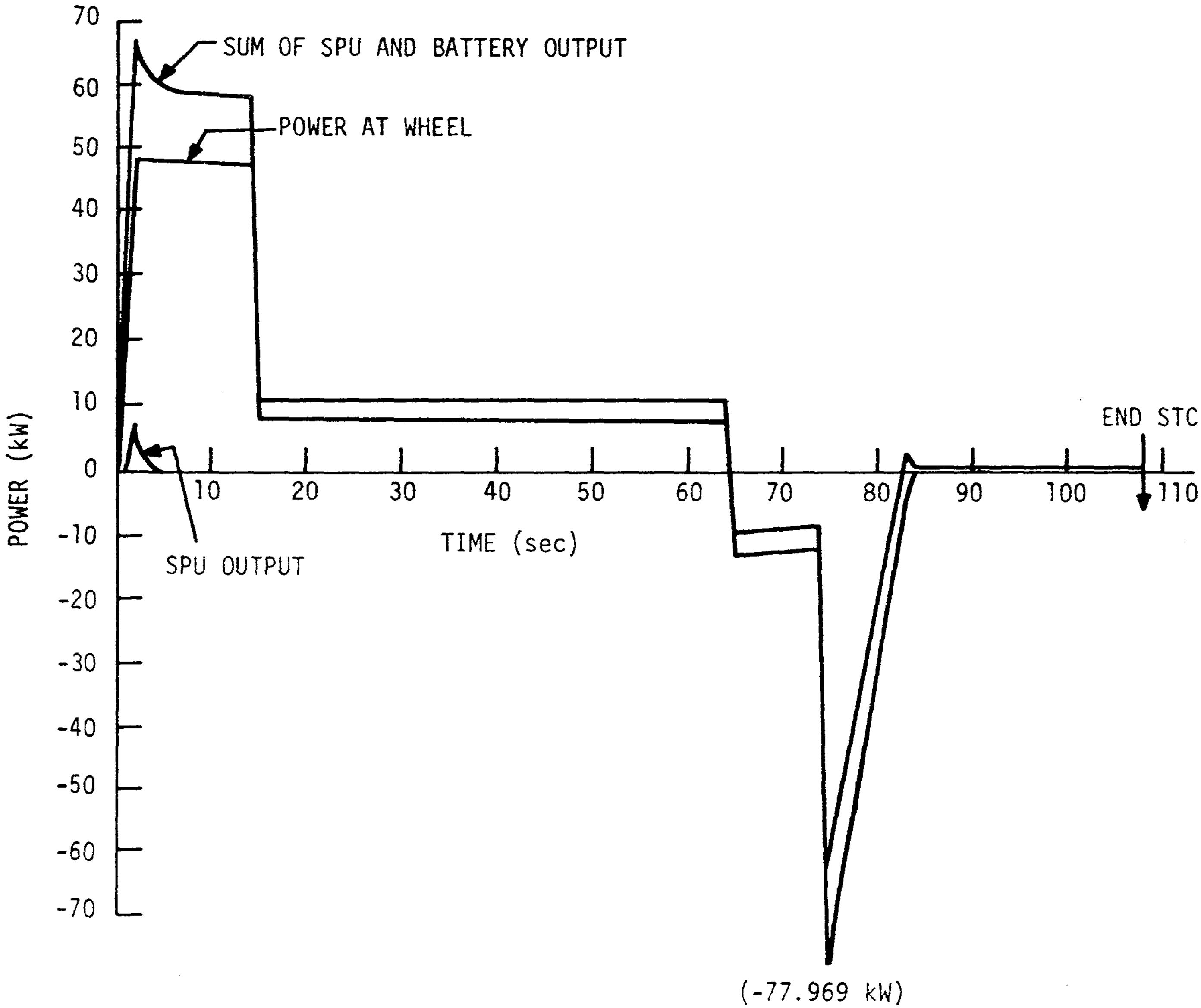

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Fig. 4-10a Vehicle C, Series STC, Power versus Time

ENERGY FLOW DIAGRAM VEHICLE C - SERIES STC

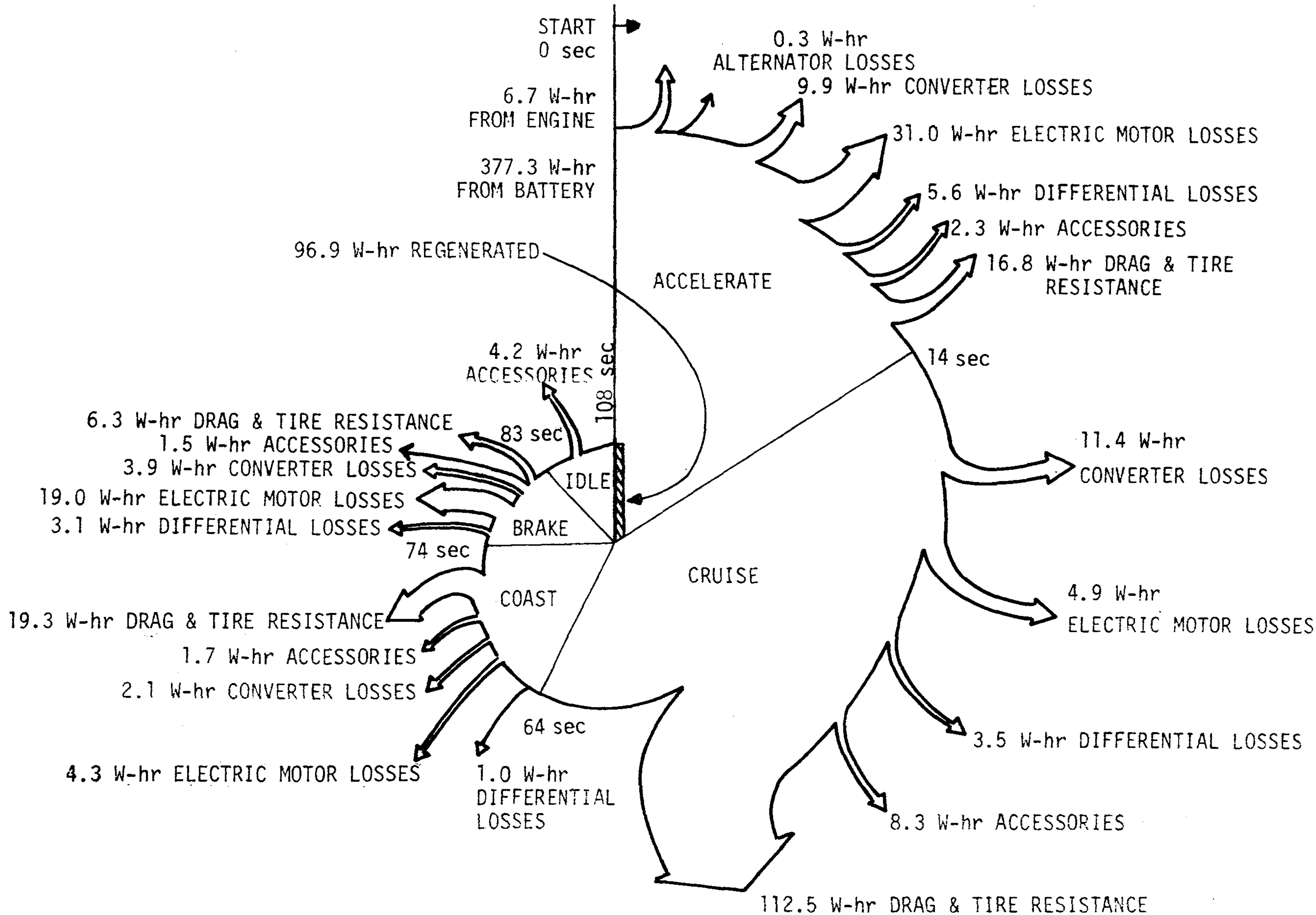
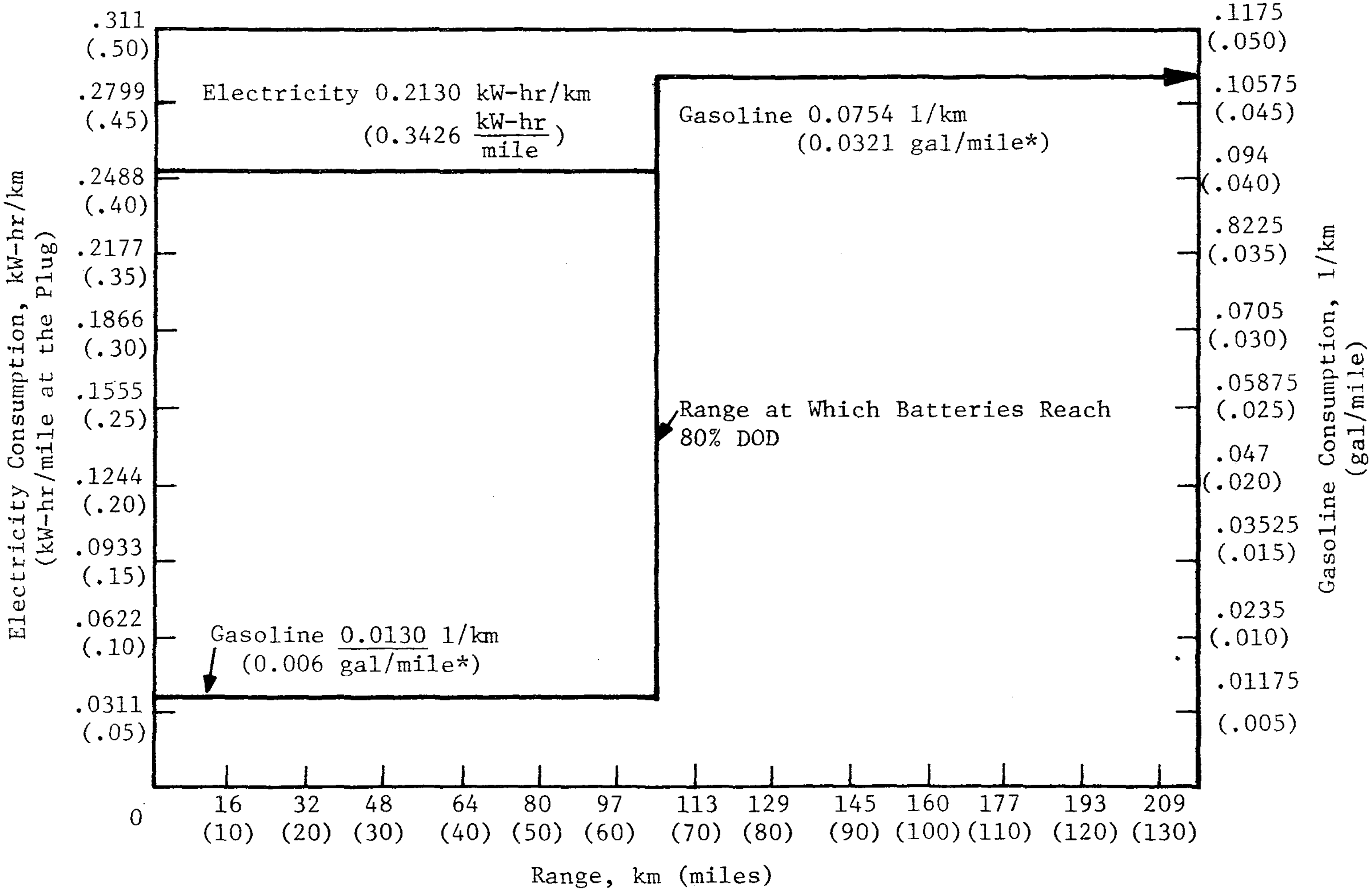


Fig. 4-10b Vehicle C, Series STC, Energy Flow Diagram

793783



*Gasoline consumption calculation made without knowing maximum power of engine. η taken as 0.39.

Fig. 4-10c Vehicle C, Series STC, Specific Energy Consumption

801670

- A high vehicle weight is incurred to meet peak power demand.
- The parallel system is more attractive than the series system for the given requirements because of its higher transfer efficiency (i.e., engine mechanical energy at wheel versus engine mechanical to electrical and then back to mechanical). Also, since a parallel system requires no alternator, overall weight is reduced.

Based on these conclusions, a recommendation was made that the remainder of the study evaluate reference mission/vehicle C utilizing a parallel hybrid system. Reference mission/vehicle C was selected because it represents the vehicle which, in this parametric study, resulted in the greatest savings of petroleum fuel over an annual period on a per vehicle basis. Because of the present automobile fleet distribution, reference mission/vehicle C also represented the greatest amount of petroleum fuel conservation for the nation.

Although not contemplated during the initial studies, a recommendation was also made that an evaluation of short-term energy storage be conducted, since it may benefit this parallel system when the high transient loads are required.

Both of these recommendations were approved by NASA-LeRC and the study proceeded with detailed evaluation of a parallel system with and without short-term energy storage for reference mission/vehicle C.

For a vehicle with lower performance requirements than those specified within this study, a series hybrid system may be attractive. Discussion/characterization of a Stirling series hybrid system is presented in Reference [1].

5.0 EVALUATION OF THE HYBRID VEHICLE COMPONENTS/PROPULSION SYSTEM

5.1 Selection of an Energy Buffer System

The parametric studies revealed that high power demand during acceleration and peak transients results in heavy propulsion system weight which, in turn, requires heavy vehicle structure to support the system. Since the STC acceleration requires large power but comparatively small energy, a high power density but low energy density buffer system would reduce vehicle weight, and, consequently, its energy consumption.

An efficient system which would capture the vehicle kinetic energy during braking and convert this energy at high power during acceleration would be highly desirable for the conservation of energy. The idea for such a system led to the investigation of the energy buffer system.

Various means of energy storage for automobiles have been under investigation by many industries and researchers. Among the numerous objectives of these studies are:

- Storing enough energy to carry out a given vehicle mission
- Improving the acceleration performance of a given vehicle with a given power plant
- Using energy storage devices as load-levelling devices which deliver bursts of high power for a short time as required during acceleration (results in reduced size of the main power plant).

The design philosophies vary according to the intended use of these energy storage devices.

For the hybrid vehicle study under consideration, the objective was to employ an energy buffer system to recover energy normally lost or inefficiently recovered during braking, and to reduce power drawn from power plants during STC acceleration

The addition of an energy buffer system imposes a weight penalty on the vehicle which results in more energy consumption per kilometer (mile) of travel. Therefore, one of the significant parameters in the choice of a buffer is the added weight.

The weight of the buffer system consists of two major components: a) the weight of the storage device and b) the weight of the power system that charges and discharges the storage device and matches to the propulsion system. The storage device weight is directly proportional to the amount of energy required. The weight of the power system depends on the peak power requirement.

Four alternative means of energy buffer systems were comparatively evaluated. For a given energy storage requirement, these alternatives were ranked from the lightest to heaviest storage device listed.

1) Nickel-zinc (Ni-Zn) batteries
2) Lead-acid batteries
3) A flywheel
4) A hydropneumatic accumulator.

For associated power system weights, the alternatives were ranked as follows:

1) Hydraulic accumulator
2) Flywheel
3) Ni-Zn batteries
4) Lead-acid batteries.

In the hydropneumatic accumulator and flywheel alternatives, the rate of charge or discharge is limited only by the associated power system. However, batteries have an upper limit on the rate at which they can be charged or discharged. Thus, the weight of the batteries is also determined by the power requirements.

If the rate of discharge is assumed to be constant as:

$$\text{power} = \frac{\Delta E}{t}$$

where

ΔE = amount of stored energy

t = time for depletion of ΔE,

then the weight of the energy buffer system can be determined by ΔE and t. Since the ranking of weights for ΔE is different from the ranking based on power requirements, different systems would be expected to result in least overall weight for different periods of discharge.

Figure 5-1 shows the effect of storage system weight versus discharge time. In this figure, total weight of the energy buffer system is plotted against the time of discharge. The calculations for the plot, given in the Appendix A, are for a vehicle of 1589 kg (3500 lb) before the addition of an energy buffer system. The energy storage device is sized to be able to store energy equivalent to the kinetic energy of the vehicle at 72.5 km/h (45 mph). This speed was chosen as it is the cruising speed during STC and the speed from which braking is repetitively done, with the specified mission profile. (Storable braking energy is less than the vehicle kinetic energy by the drag and friction effects.) This choice of speed ensures that most of the consistently available energy from the braking can be stored.

Figure 5-1 shows that, if discharge time is very large, batteries provide the least overall weight since the energy densities of batteries are high as compared to those of the flywheel and hydropneumatic accumulator. At very small discharge times, the high power density of the hydropneumatic system more than compensates for the low energy density of the accumulator and results in the lightest system. A conclusion that may be reached from the curves in Figure 5-1 is that for a discharge time of less than about 30 seconds, the hydropneumatic system is lighter than the flywheel system. The curves also indicate that for discharge times greater than 30 seconds, the battery system is the lightest.

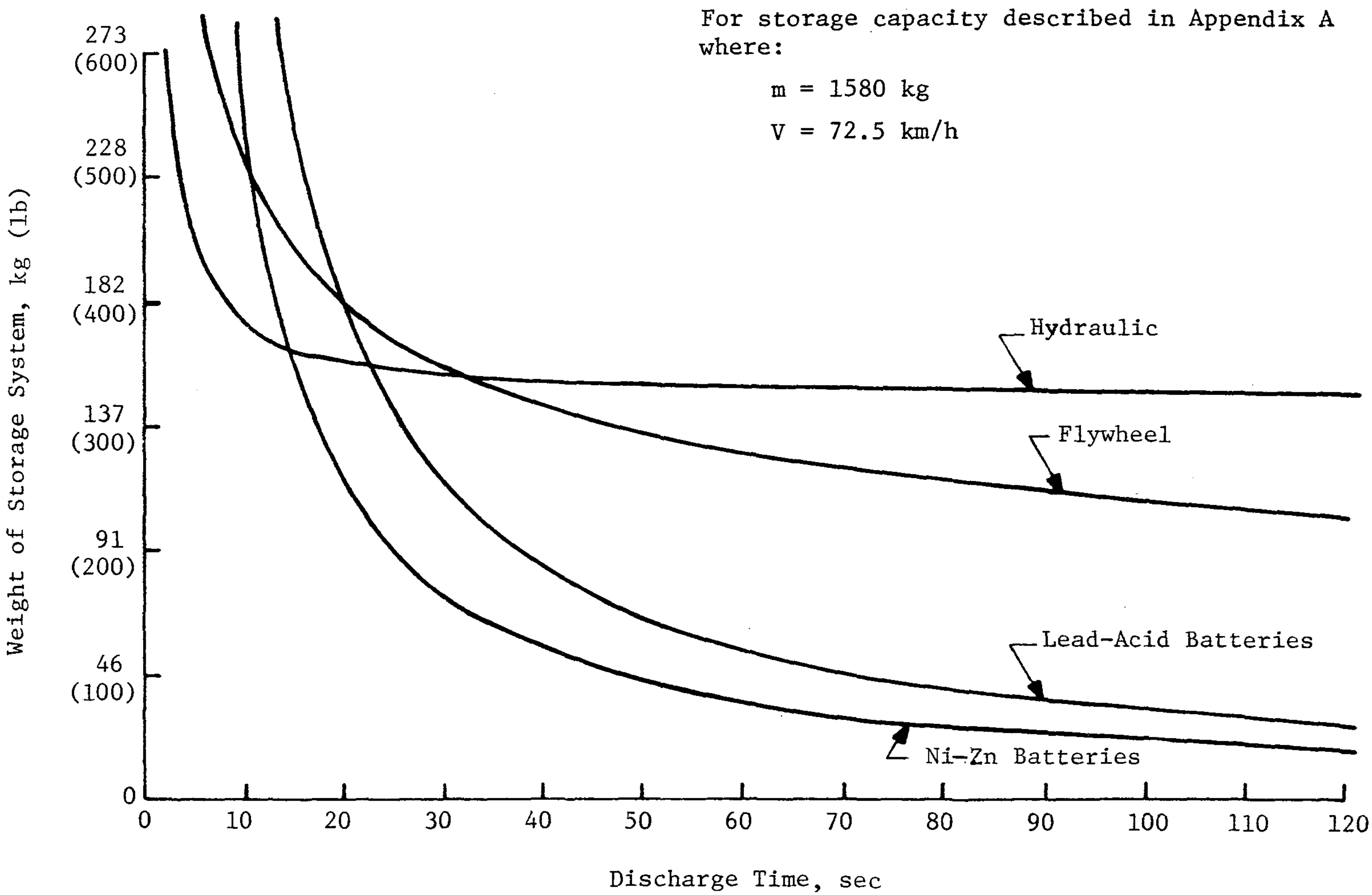


Fig. 5-1 Storage System Weight Versus Discharge Time

801570 -1

An intermediate range of discharge time may exist, where the flywheel system is the lightest of all the four alternatives considered, particularly if the energy to be stored is high. However, for the study of the hybrid vehicle under consideration, a regenerative braking period of nine seconds (absorption of the vehicle kinetic energy) and the discharge of that energy during subsequent STC acceleration are the most important criteria. At these conditions, the hydraulic system is the lightest system.

Another important consideration in the choice of an energy buffer system is the charge/discharge efficiency during an STC cycle and, to a lesser degree, the ability to retain energy storage during longer time periods such as continuous driving or periods of nonoperation (overnight, weekends, etc.). At high charge/discharge rates, the efficiencies of the battery systems are considerably lower than those of either the hydropneumatic or flywheel systems. Therefore, a hydropneumatic buffer system was selected for further study.

5.2 Hydropneumatic Energy Buffer System

The hydropneumatic system consists of a hydraulic accumulator pressurized with gas to a pressure usable in a hydraulic element or module. A schematic of a proposed system is shown in Figure A-1 of Appendix A.

Energy usually wasted or inefficiently regenerated by the electric system during braking would be stored in the hydropneumatic accumulator and later used by the hydraulic element or module (hydraulic transmission) to deliver power to the drive train and accelerate the vehicle.

5.2.1 The Accumulator

- Capacity - The accumulator is sized to absorb the vehicle kinetic energy less drag and tire friction during the STC braking period with power train and buffer system efficiency considered

- Size Approximation - The size of the accumulator is determined by its kinetic energy as described under "capacity". The energy stored in the accumulator can be expressed as follows:

$$\frac{1}{n-1} P_1 V_2 \quad (R^{-1} - R^{-n})$$

where n = specific heat ratio of gas (1.4 is used in the calculation)

P_1 = maximum system pressure

V_2 = volume of accumulator before the energy is stored

R = volume expansion ratio

- Weight - For the weight approximation, the accumulator length to diameter ratio is assumed to be equal to 1.5. The accumulator weight can then be expressed as

$$\frac{\pi d^3 \; P_1 \rho}{S} \quad \text{lb}$$

where d = diameter of accumulator in inches

P_1 = maximum system pressure in psi

ρ = specific weight of material to be used for the accumulator in lb/in.^{3}

S = working stress of the material in psi

- Specific Energy Density - With the accumulator energy and weight expressions, the specific energy density may be expressed as follows:

The gravitative specific energy density

$$= \frac{1}{32\ (n-1)} \; \frac{R^{n-1} -1}{R^n} \; \frac{S}{\rho} \; \frac{\text{ft-lb}}{\text{lb}} \quad .$$

For $S = 40{,}000$ psi, $\rho = 0.2832$ lb/in.^{3}
and $R = 2$,

the gravitative specific energy density
= 3982.6 J/kg (1335 ft-lb/lb).

The volumetric specific energy density

$$= \frac{1}{12} \frac{P_1}{n-1} \frac{R^{n-1} - 1}{R^n} \frac{\text{ft-lb}}{\text{in.}^3} .$$

For $P_1 = 3000$ psi, $R = 2$ and $n = 1.4$,

the volumetric specific energy density
= 6.26 MJ/m^3 (75.67 $\frac{\text{ft-lb}}{\text{in.}^3}$) .

- Mass - Based on the specific energy density, the accumulator mass and energy storage capacity versus the accumulator volume (in SI units) are calculated and plotted as shown in Figures 5-2 and 5-3.

5.2.2 Energy Transfer (Hydraulic) Transmission

The energy stored in the accumulator can be transferred to the automobile drive train by using a hydraulic transmission. The element is a variable displacement axial piston pump/motor; with the proper controls, the element acts as a pump when storing energy in the accumulator, and then as a motor when accepting energy from the accumulator and delivering the energy to the drive train. Transferring all of the power through the hydrostatic elements, however, reduces the effectiveness of the system because of the inefficiency of the hydraulic pump/motor.

A second and more flexible energy transfer system uses a continuously variable hydromechanical transmission (CVT) or a hydraulically augmented transmission. The CVT operates in three modes: pure mechanical, where all power is transferred through the gear train; power-split, where a low percentage of the power is transferred through the hydraulic module, and the remaining power, through the gear train; and pure hydrostatic, where all the power is

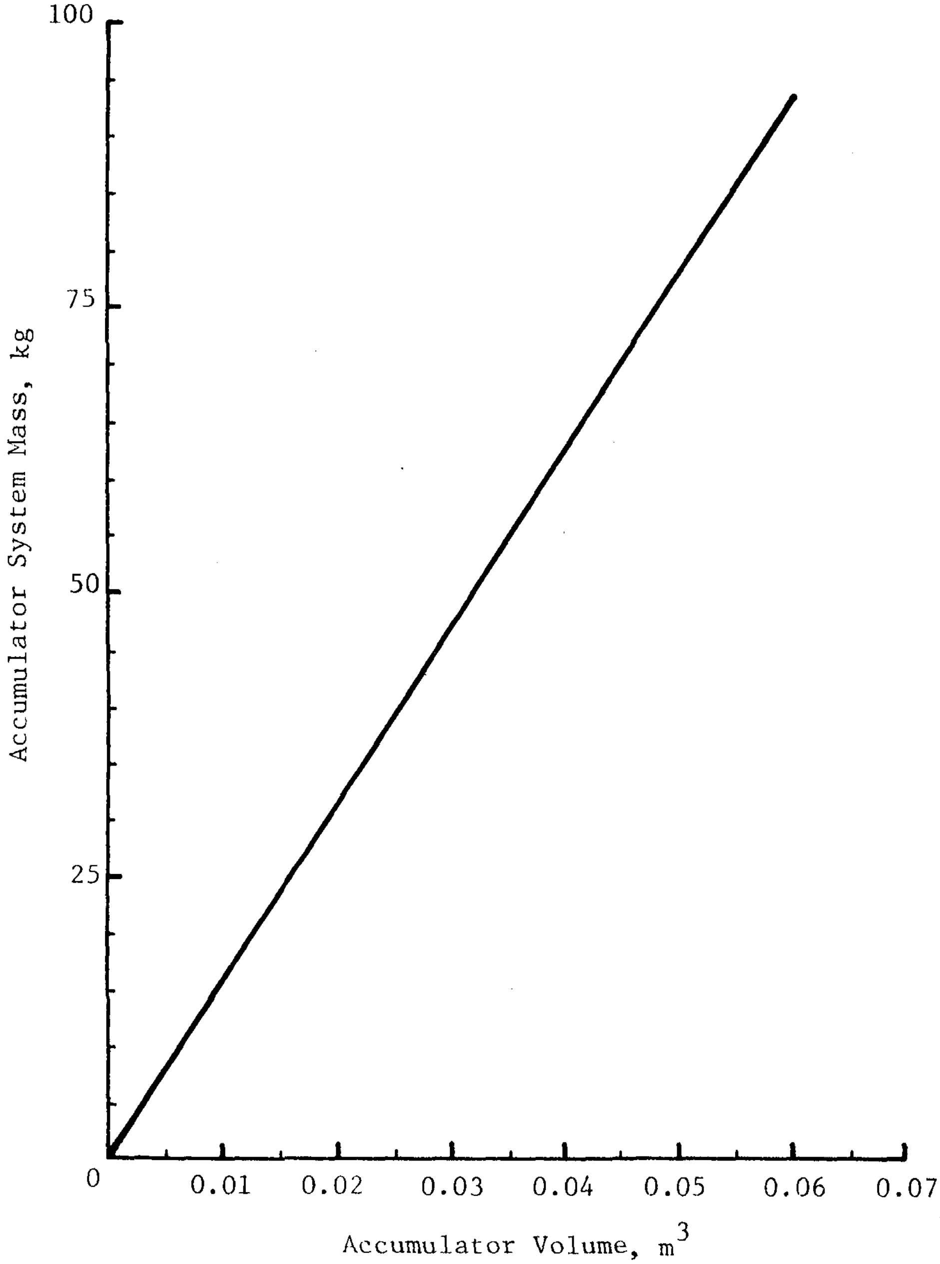


Fig. 5-2 Hydropneumatic Accumulator System, Mass versus Accumulator Volume

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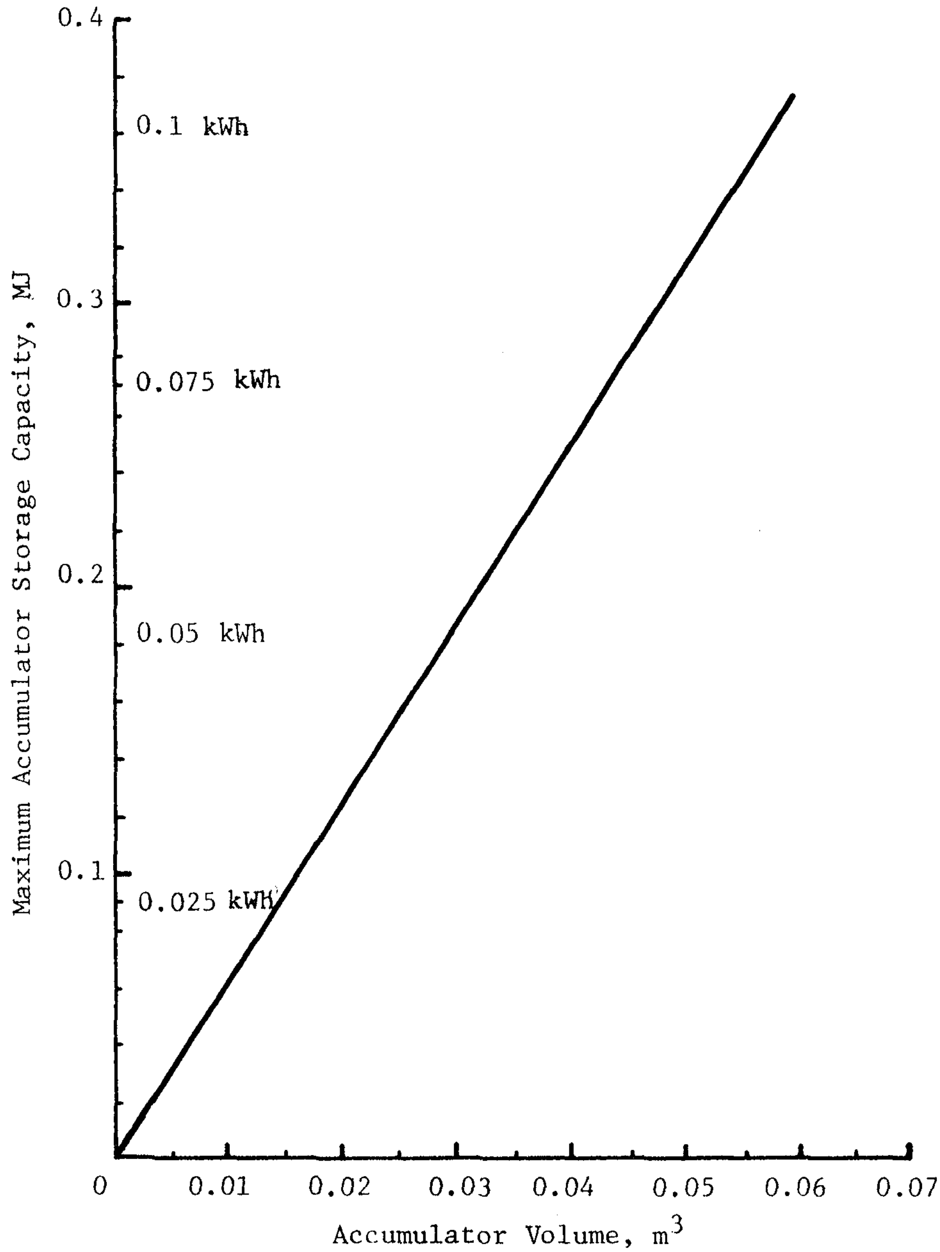

Fig. 5-3 Maximum Accumulator Storage Capacity as a Function of Accumulator Volume for the Hydropneumatic/Electric Vehicle

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transferred through the pump/motor module. Proper design of the CVT insures that most of the power is transferred through the high-efficiency mechanical gear train path.

The flexibility of the hydromechanical CVT and accumulator provides an efficient method of storing and delivering the regenerative energy of braking, an effective means of managing the Stirling-engine/electric propulsion system energy, and a means of providing overdrive for the automobile.

Although a CVT for this specific application has not been developed, several companies, including Orshansky Corp., Sunstrand Corp., and MTI, have built and tested CVTs which incorporate all of the features required for such a transmission.

The hydraulic motor weight was assumed to be 1.22 kg/kW (2 lb/hp). The above completes the specifications of the hydraulic energy buffer system.

5.3 Kinematic Stirling Engine Design and Operating Strategies

The design and operating strategies of the Stirling engine were extensively investigated to further minimize energy consumption. Various aspects of this engine are described in the following sections.

5.3.1 Variable Angle Z-Crank Engine

A variable angle Z-crank Stirling engine was selected for this study. The Z-crank drive offers several advantages over the present U-crank drive Stirling engine, particularly for automotive applications. These advantages include:

- Light weight
- Fewer drive system parts and lower manufacturing cost
- Less drive system friction power, about half of the U-crank drive system
- Longer piston seal life, no side load

- Simple power control, varying piston stroke versus mean pressure power control which requires a hydrogen compressor and its associated hydraulic-actuated valve system. The present hydrogen compressor consumes 700 watts of power even if the compressor is not in the operating mode (short-circuited)
- No gear, no gear noise and backlash.

Because the Z-crank engine is in its development stage, the cycle analysis of such an engine requires specific considerations. Therefore, for this study, the performance of the Stirling engine used for fuel consumptions (as listed in Table 5-2, page 5-28) was based on a conventional "U" crank engine with charge pressure power control.

5.3.2 Engine Design Strategy

The power output of a Stirling engine can be easily increased several times by increasing the charge pressure. Likewise, the power output of an internal combustion engine may be increased by increasing the intake pressure with a supercharger. However, the internal combustion engine with a supercharger may have detonation or pre-ignition problems and a high-pressure exhaust results in energy loss unless an additional system is added to the engine exhaust to recover some of the energy. Therefore, the potential for power increase of an internal combustion engine is limited to below a factor of two.

The pressure fluctuation of a Stirling engine is about ± 30 to 35% of the charge pressure over the cycle. In comparison, the internal combustion engine pressure fluctuation is 40 to 70 times the intake pressure. When cycling life must be considered, a small amplitude over a higher mean pressure is preferred to a high amplitude over a lower mean pressure.

To demonstrate the power increase capability of the Stirling engine, a sample calculation was conducted; the results are summarized and plotted in Figure 5-4. This engine was designed for 11.12 kW indicated output under the following operating conditions:

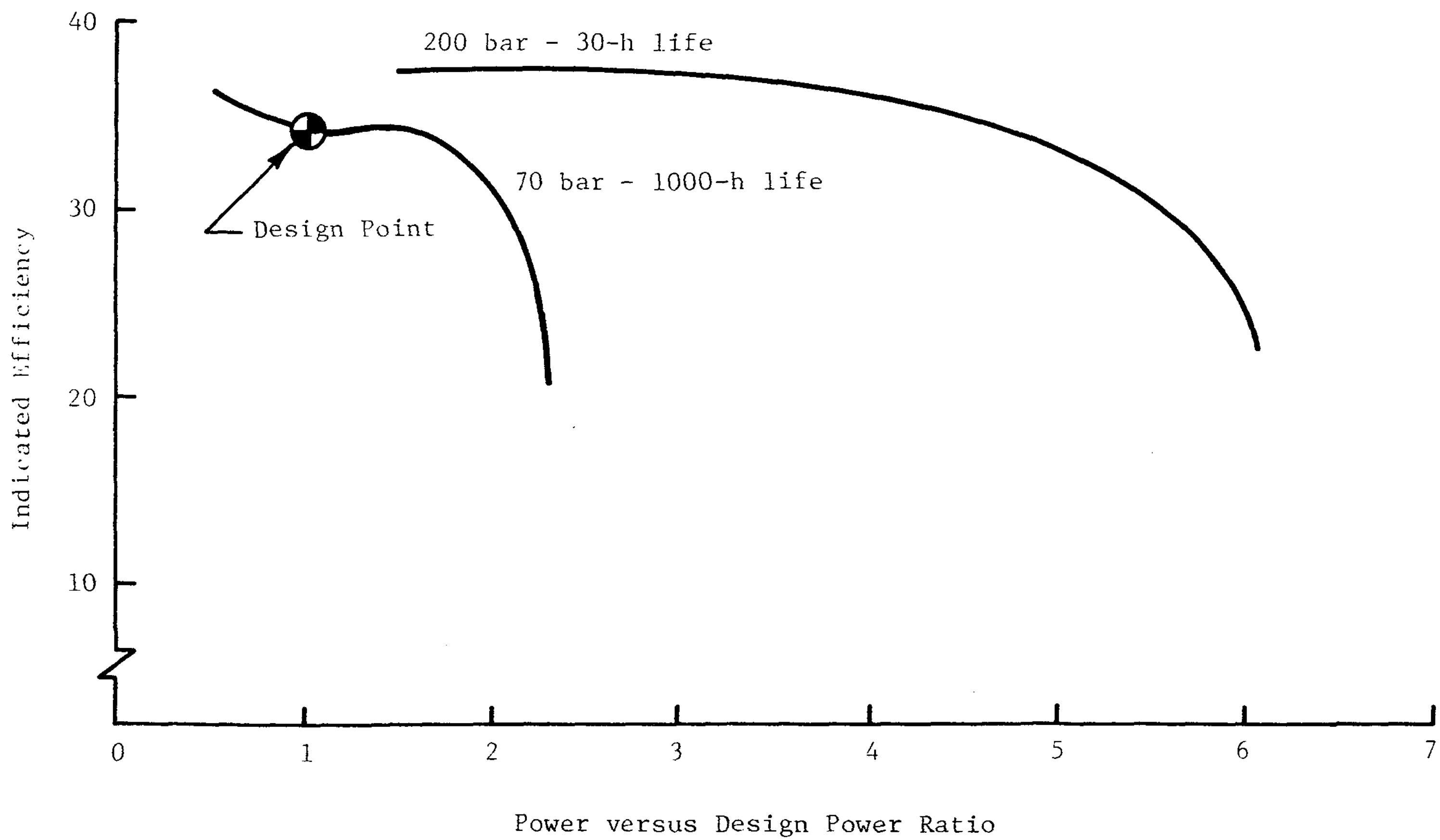

Fig. 5-4 Cycle Efficiency versus Power Increase at Two Charge Pressures

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- Heater head temperature: 1088°K
- Cooler temperature: 330°K
- Charge pressure: 70 bar
- Engine speed: 1000 r/min
- Life expectancy: 1000 h

As shown in Figure 5-4, the same engine is capable of developing five times its design power without losing the cycle efficiency, but at a cost of dropping its life expectancy to 30 hours. This is acceptable for automotive application because of the infrequent demand for high power. This attractive characteristic of a Stirling engine was utilized in the hybrid vehicle application in two ways. First, a smaller engine with the highest efficiency point coincides with the power level most frequently used, yet it is capable of providing high power if demanded, but at a cost of life expectancy. This results in a lighter engine weight; thus, lighter overall vehicle weight. Second, its high-cycle efficiency spread over a large range of power level contributes to fuel economy.

Due to the lack of experimental verification of life prediction, a conventional U-crank Stirling engine was designed for the reference mission/vehicle C in this study; its performance is shown in Figure 5-5. This engine is designed for 42% cycle efficiency at 10.33 kW with a life expectancy of 4000 hours. This power level is adequate to propel vehicle C at 90 km/h (56 mph) and to supplement STC acceleration. At a high power of 53 kW (70 hp), it is adequate to accelerate vehicle C up a 6% grade to 90 km/h (56 mph) in 300 m (984 ft). At this power level, the life expectancy drops to 200 hours.

5.3.3 Engine Operating Strategy

The Stirling engine tends to rotate, unless mechanically locked, as long as the temperature differential between the hot and cold ends is maintained at a level at which the power output is greater than the frictional power and heat conduction losses. When power is not needed, the Stirling engine may be locked up (no rotation) while maintaining the hot and cold ends at the design operating temperature. This characteristic was utilized to minimize the energy consumption in the hybrid vehicle application. Since the hybrid

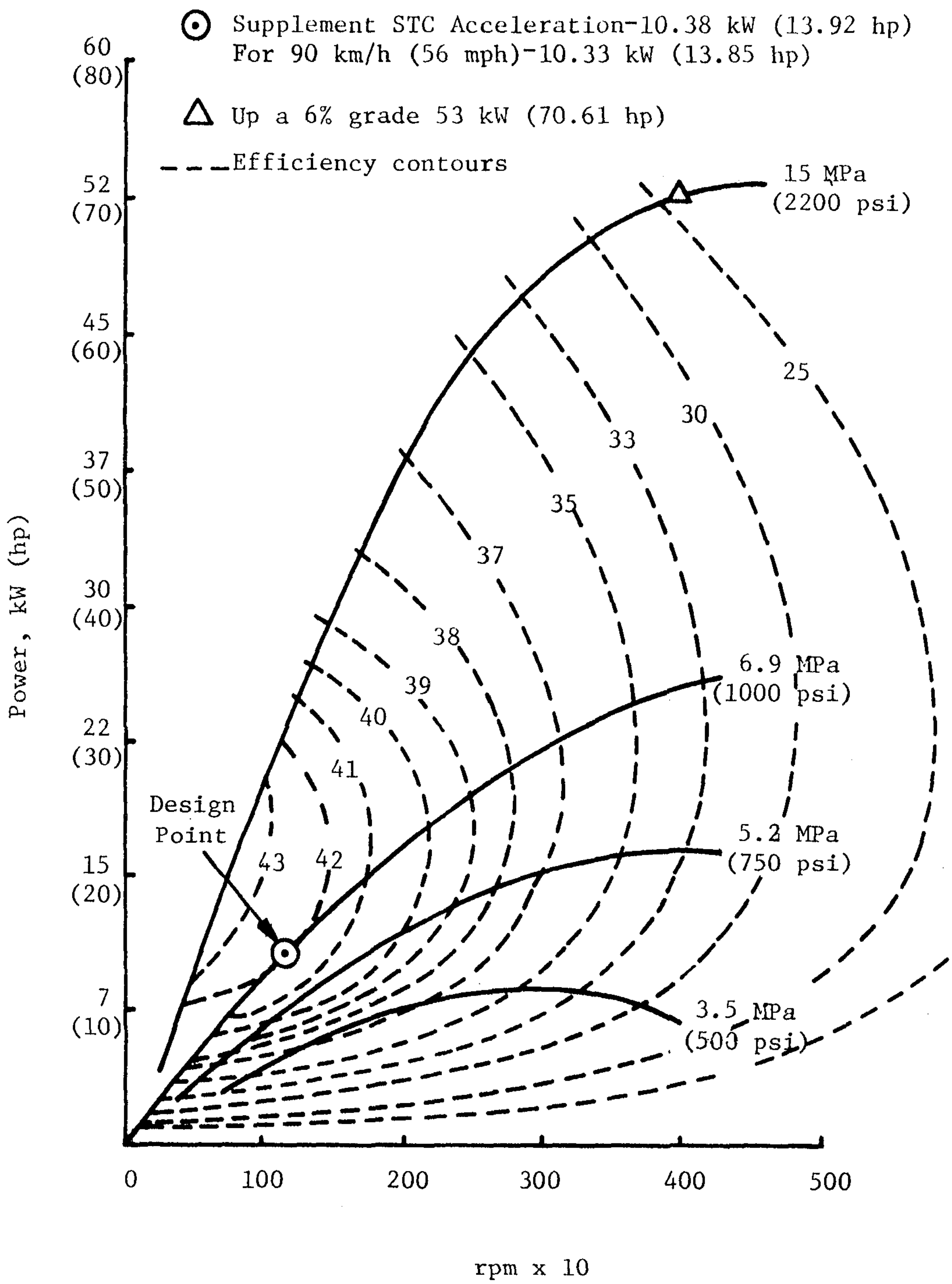

Fig. 5-5 Vehicle C - Engine Power versus rpm as a Function of Charge Pressure

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vehicle has only intermittent engine power demands, the Stirling engine power is readily available to meet its demand simply by releasing the engine's locking mechanism.

Keeping the hot and cold ends of a Stirling engine at the design operating temperature is equivalent to the idling of an internal combustion engine. The difference, however, is that an idling internal combustion engine consumes energy to overcome all the engine friction and auxiliary power losses. When mechanically locked, a Stirling engine consumes energy only to make up the heat conduction losses through the walls between the hot and cold ends, and some of the convection losses at the hot surfaces. The Stirling engine, as represented in Figure 5-5, consumes only 0.19 l/hr (0.05 gal/hr) of fuel when locked. This amount is extremely low as compared with the idling of an internal combustion engine capable of the same maximum power.

5.4 Propulsion System Operating Strategy

To reduce the petroleum consumption and to minimize total energy usuage, the strategy for operating various power plants was selected as follows:

- First, energy available in the hydropneumatic accumulator (if one is provided) is used
- Second, the energy from the battery is used, up to a power limit, until 80% DOD
- Finally, the engine is used.

In order to meet the goal of reducing petroleum consumption, the engine should be operated in an on-off mode and only when the power required exceeds the power capabilities of the hydraulic accumulator system and batteries.

The strategy for batteries was to only use their power up to a maximum value; this value was set at the power to drive the vehicle at 90 km/h (56 mph) on a level road. Power demand greater than this value would be supplied by the engine; that is, the batteries would be load leveled while the heat engine meets peak requirements. The operating strategy is presented in Figure 5-6. It should be noted that the batteries by themselves are not adequate to

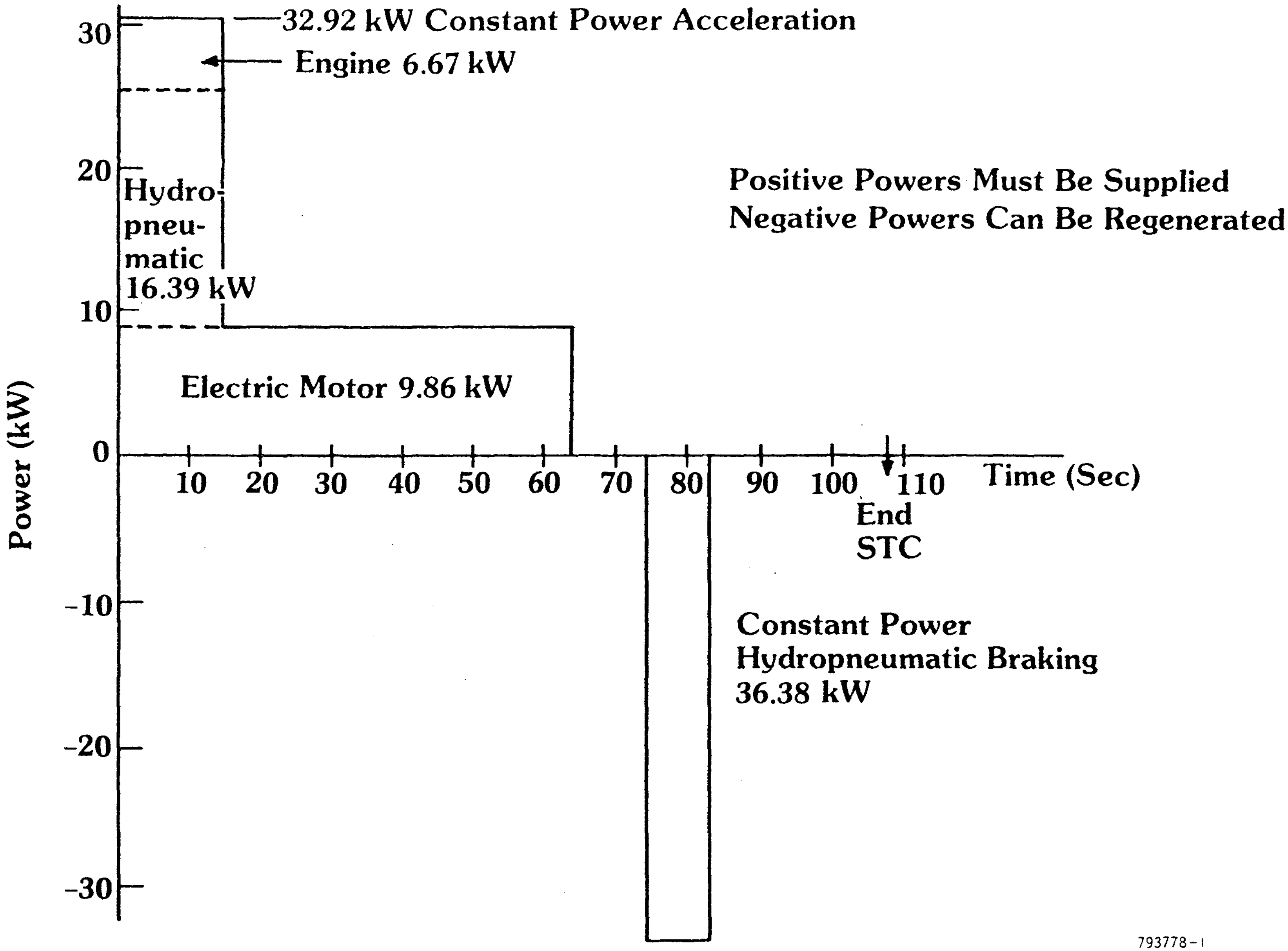

Fig. 5-6 Composition of Power at Wheel During Special Test Cycle (STC)
Vehicle C, Test Weight 1855.94 kg (4079 lb)

meet the STC acceleration requirements. The peak power is supplied by the hydraulic system and/or the engine.

To ensure that the engine power is available to meet peak transients and/or emergency maneuvers, as needed, it is necessary to keep the external combustion Stirling engine warm. If a vehicle is performing continuous STCs, the heat loss between cycles is negligible. If a vehicle is not performing continous STCs, it is necessary to keep the engine warm by igniting the combustor of the Stirling engine for x seconds every y minutes; this ensures that the engine is available to instantly supply power on demand. The values of x and y need to be determined during on-off evaluation of the particular Stirling engine design chosen for the hybrid operation. Preliminary calculations indicated that 0.2 liters (0.06 gallons) of gasoline per hour could keep the Stirling engine combustor hot during an extended period when engine power is not demanded.

When operating in an on-off mode, the external combustion Stirling engine offers advantages over the present internal combustion engine by maintaining the engine hot (ready for use); hence, peak power is always available on immediate demand.

5.5 Total Propulsion System - Evaluation and Results

An extensive evaluation was made to determine the vehicle and component weight, as well as the annual energy consumption under a series of combinations of engine and battery power conditions with and without a hydropneumatic system. Since there were three power systems (electrical, heat engine, hydraulic), they were judiciously sized and used in the load-sharing manner to optimize the performance and fuel consumption. Eight different cases were considered to reflect different operating strategies and design criteria. To assist the reader in following the text, the strategies of operating each system are listed in Tables 5-1a through 5-1c. Terms used in the tables are as follows:

DOD - Depth of Discharge

STC - Special Test Cycle

P_{STC} - Power required for STC acceleration (constant power acceleration assumed)

P_{72} - Power required at the wheels to cruise at 72 km/h (45 mph) (STC cruising)

P_{90} - Power required at wheels to cruise at 90 km/h (56 mph) (highway cruising)

P_{brake} - Power available at wheels during STC braking

P_{hydro} - Rated power of the hydropneumatic system provided at the wheels

P_{motor} - Rated power of the electric motor provided at the wheels

$P_{battery}$ - Rated power output of the battery provided at the wheels

P_{engine} - Power of the engine provided at the wheels

NOTE: Actual ratio of the power of various of the above components will be higher to account for the efficiency of the components in between the wheel and the output of the components under consideration.

The design criteria for the selection of the components are described in Tables 5-1d and 5-1e. The engine is designed to meet two conditions as follows:

a. Best cycle efficiency to occur at a power/speed point corresponding to the requirement of driving the vehicle at 90 km/h (56 mph) by the engine alone.

b. Peak power adequate to accelerate the vehicle up the hill (6% grade) with minimum speed of 90 km/h (56 mph) in 300 m (984 ft)

The procedures for the calculation of component energy and power requirements are as follows:

a. For a given vehicle, the test weight, W_T, is expressed as a function of the propulsion system weight, W_P, which is determined by this calculation.

TABLE 5-1a

OPERATING STRATEGY DURING STC ACCELERATION
FOR BATTERY DOD < 80%*

Case No.	Amount of Power Provided at Wheels by:		
	Engine E	Electric System M	Hyraulic System H
I	0	$P_{STC} - H$	P_{hydro}
II	$P_{STC} - H-P_{90}$	P_{90}	P_{hydro}
III **	$P_{STC} - H-P_{90}$	P_{90}	P_{hydro}
IV	$P_{STC} - P_{90}$	P_{90}	0
V	$P_{STC} - H$	0	P_{hydro}
VI	P_{STC}	0	0
VII	$P_{STC} - H-P_{72}$	P_{72}	P_{hydro}
VIII	$P_{STC} - H-P_{72}$	P_{72}	P_{hydro}

*Battery DOD > 80% means the battery is discharged.

**Case III selected for conceptual design.

TABLE 5-1b

OPERATING STRATEGY DURING STC ACCELERATION
FOR BATTERY DOD > 80%*

Case No.	Amount of Power Provided at Wheels by:		
	Engine E	Electric System M	Hydraulic System H
I	$P_{STC} - H$	0	P_{hydro}
II	$P_{STC} - H$	0	P_{hydro}
III**	$P_{STC} - H$	0	P_{hydro}
IV	P_{STC}	0	0
V	$P_{STC} - H$	0	P_{hydro}
VI	P_{STC}	0	0
VII	$P_{STC} - H$	0	P_{hydro}
VIII	$P_{STC} - H$	0	P_{hydro}

*Battery DOD > 80% means the battery is discharged.

**Case III selected for conceptual design.

TABLE 5-1c

OPERATING STRATEGY DURING STC CRUISE AND 90-km/h (56-mph) CRUISE FOR BATTERY DOD < 80%*

	STC Cruise: Amount of Power Provided at Wheels by:			90-km/h (56-mph) Cruise: Amount of Power Provided at Wheels by:		
Case No.	Engine E	Electric System M	Hydraulic System H	Engine E	Electric System M	Hydraulic System H
I	0	P_{72}	0	0	P_{90}	0
II	0	P_{72}	0	0	P_{90}	0
III**	0	P_{72}	0	0	P_{90}	0
IV	0	P_{72}	0	0	P_{90}	0
V	P_{72}	0	0	P_{90}	0	0
VI	P_{72}	0	0	P_{90}	0	0
VII	0	P_{72}	0	$P_{90}-P_{72}$	P_{72}	0
VIII	0	P_{72}	0	$P_{90}-P_{72}$	P_{72}	0

*Battery DOD > 80% means the battery is discharged.

**Case III selected for conceptual design.

TABLE 5-1d

BATTERY DESIGN CRITERIA

Case No.	Weight of Battery Determined by:		Remarks
	Power	Energy	
I	$P_{battery} = P_{STC} - P_{hydro}$		
II	$P_{battery} = P_{90}$		
III*	$P_{battery} \geq P_{90}$	To provide electric range = 50 km (31.1 miles)	Whichever is bigger.
IV	$P_{battery} \geq P_{90}$	To provide electric range = 50 km (31.1 miles)	Whichever is bigger.
V	No Electric System	No Electric System	
VI	No Electric System	No Electric System	
VII	$P_{battery} = P_{72}$		
VIII	$P_{battery} = P_{72}$	To provide electric range = 50 km (31.1 miles)	Whichever is bigger.

*Case III selected for conceptual design.

TABLE 5-1e

MOTOR DESIGN CRITERIA

Case I	$P_{motor} = P_{90}$, $(P_{STC} - P_{hydro})$, whichever is bigger.
Case II	$P_{motor} = P_{90}$
Case III*	$P_{motor} = P_{90}$
Case IV	$P_{motor} = P_{90}$
Case V	No Electric System
Case VI	No Electric System
Case VII	$P_{motor} = P_{72}$
Case VIII	$P_{motor} = P_{72}$

*Case III selected for conceptual design.

b. The total energy and power at the wheel for STC acceleration are expressed as a function of W_T.

c. The energy during deceleration less drag and tire friction is then expressed as a function of W_T.

d. Hydropneumatic accumulator size and weight can now be expressed as a function of W_T with the specific energy density as derived in Section 5.2.1.

e. Hydraulic system power level is determined based on the energy stored in the accumulator and specified time period for deceleration. The weight of the system is assumed to be 1.22 kg/kW (2 lb/hp). Therefore the hydraulic motor weight is a function of W_T.

f. The battery weight, expressed as a function of W_T, varies depending on the power or energy level set as indicated in Table 5-1d. A constant 100 watts/kg and 40 watt-hour/kg are used in this expression.

g. Depending on the condition set in Tables 5-1a-e, the electric motor weight can also be expressed as a function of W_T. A constant 2.73 kg/kW (6 lb/kW) is used in this expression.

h. The engine is designed to provide the power for the vehicle up the ramp. The engine weight in kilograms (pounds) is determined, based on 34.02 kg + 0.6 kg/kW (75 lb + 0.926 lb/hp), where p is the maximum engine power which is a function to W_T.

i. Other propulsion components' weights are assumed to be constant for all vehicles as follows:

Transmission	45.36 kg (100 lb)
Differential	45.36 kg (100 lb)
Inverter	22.68 kg (50 lb)

Summing up all the propulsion system component weights, W_p, in terms of W_T, the vehicle test weight can be solved. With the W_T known, the power and energy at different driving modes are then calculated. In these calculations, constant efficiency for each power train component, except the heat engine thermal efficiency, is used. The overall thermal efficiency at a power level other than the design point of a Stirling engine is shown in Figure 5-7. For fuel consumption of an internal combustion engine, the efficiency is based on Figure 5-8, which has been scaled from a Jet Propulsion Laboratory Technical Report [5].

The results of this evaluation are summarized as shown in Tables 5-2a (metric) and 5-2b (U.S. equivalent).

5.6 Conclusions

The detailed results of the various trade-off evaluation are presented in Table 5-2. A summary table of the highlights of Table 5-2 is presented in Table 5-3.

Examining Table 5-2, Case III is the preferred selection for the following reasons:

1. Case III consumes the least amount of gasoline except for Case I, which consumes more total annual energy and is also heavier in vehicle weight.
2. Case III consumes the least amount of total energy, except for Case VIII, which consumes more gasoline.
3. Case III has a vehicle electric range of 50 km, which covers 42% of the total mileage and 75% of driving days.

Observing Tables 5-2 and 5-3, it is concluded that:

- Use of a parallel hybrid vehicle with a Stirling engine, as compared to total-powered Stirling vehicles, results in significant petroleum savings (50%)
- Addition of a hydraulic accumulator on a Stirling hybrid system reduces petroleum consumption by 76 liters (20 gallons) per year.

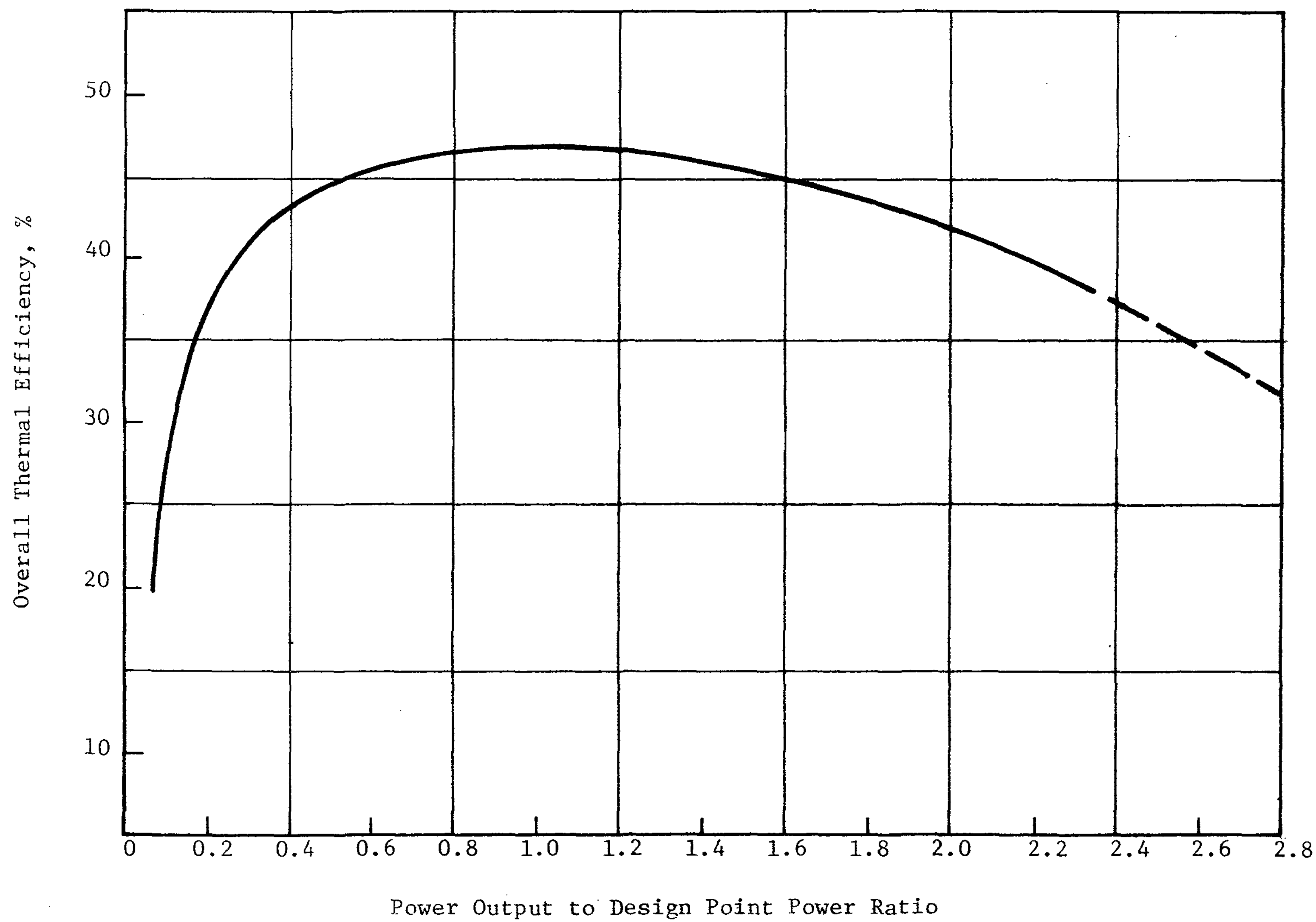

Fig. 5-7 Overall Stirling Engine Thermal Efficiency versus Power Output Ratio

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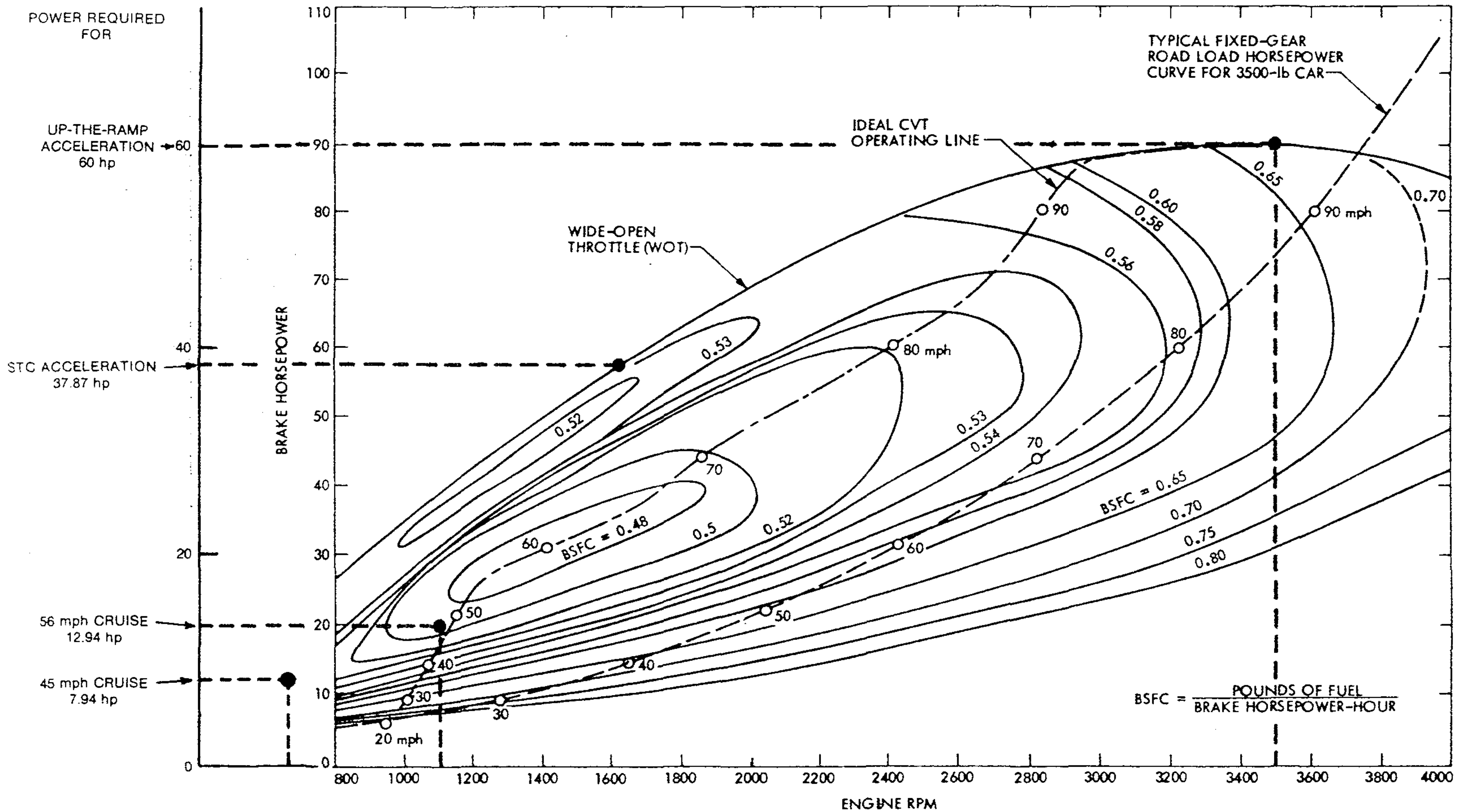

IDEAL TRANSMISSION ASSUMED TO DETERMINE OTTO-CYCLE FUEL ECONOMY

Fig. 5-8 Typical Fuel Consumption Map for Otto-Cycle Engine

TABLE 5-2a

SUMMARY OF HYBRID VEHICLE COMPONENT WEIGHT AND PERFORMANCE, SI UNITS

VEHICLE C		CASE I	CASE II	* CASE III	CASE IV	CASE V	CASE VI	CASE VII	CASE VIII
Battery Weight	kg	225.68	126.26	175.40	158.58	0	0	77.35	153.33
Electric Motor Power	kW	20.23	11.36	12.83	11.09	0	0	6.95	7.15
Electric Motor Weight	kg	50.05	31.031	31.53	30.26	0	0	19.05	19.56
Accumulator Volume	m^3	.04648	.04217	.04393	0	.03685	0	.04021	.0427
Accumulator Weight	kg	74.16	67.06	69.88	0	58.24	0	64.15	67.79
Hydraulic Motor Pump (for Braking)	kW	38.59	34.9	36.38	0	30.2	0	33.29	35.35
Hydraulic Motor Pump	kg	46.86	42.31	44.36	0	36.85	0	40.49	43.22
Engine Power	kW	60.0	54.91	56.96	51.93	48.39	44.75	52.67	55.53
Engine Weight	kg	68.25	65.15	66.29	63.45	62.33	59.15	63.7	65.52
Transmission Weight	kg	45.50	45.50	45.50	45.50	45.50	45.50	45.50	45.50
Differential Weight	kg	45.50	45.50	45.50	45.50	45.50	45.50	45.50	45.50
Inverter Weight	kg	22.75	22.75	22.75	22.75	22.75	22.75	22.75	22.75
Total Propulsion System	kg	583.76	446.35	501.41	365.82	271.18	172.9	385.84	463.19
Vehicle Test Weight	kg	1962.87	1789.51	1855.94	1680.31	1557.01	1429.61	1706.25	1805.89
kW Required at Wheel for STC Acceleration		34.65	31.77	32.92	30.08	28.09	26.03	30.5	32.11
• Supplied by Hydropneumatic		17.38	15.72	16.39	0	13.61	0	15.0	15.92
• Supplied by Electric Motor		17.26	9.698	9.86	9.46	0	0	5.93	6.11
• Supplied by Engine		0	6.344	6.67	20.62	14.48	26.03	9.57	10.08
Vehicle Range (80% Battery Depletion)	km	41.0	37.0	50.0	50.0	—	—	26.0	50.0
Annual Electricity Consumption	kW-hr	1847.0	1258.0	1542.0	1419.0	0	0	862.0	13.46
Annual Gasoline Energy Input to Stirling Engine Gasoline Consumption	kW-hr	2112.0	2704.0	2309.0	3073.0	4599.0	5233.0	2964.0	2390.0
Total Annual Energy Consumption	kW-hr	3959.0	3962.0	3851.0	4492.0	4599.0	5223.0	3826.0	3736.0
Annual Gasoline Consumption	liters								
• Stirling Engine		243.0	310.0	268.0	352.0	530.0	602.0	341.0	275.0
• Conventional I.C. Engine		—	—	—	—	780.0	844.0	—	—
Annual Average km/l									
• Stirling Engine		65.8	51.0	60.4	45.0	30.0	26.4	46.8	58.2
• Conventional I.C. Engine		—	—	—	—	20.5	18.7	—	—

*Selected for conceptual design

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TABLE 5-2b

SUMMARY OF HYBRID VEHICLE COMPONENT WEIGHT AND PERFORMANCE, U.S. EQUIVALENT

VEHICLE C		CASE I	CASE II	CASE III*	CASE IV	CASE V	CASE VI	CASE VII	CASE VIII
Battery Weight	lb	496.0	278.3	385.5	348.53	0	0	170.0	337.0
Electric Motor Power	kW	20.23	11.36	12.83	11.09	0	0	6.95	7.15
Electric Motor Weight	lb	121.0	68.2	69.3	66.52	0	0	42.0	43.0
Accumulator Volume	ft³	1.66	1.506	1.569	0	1.316	0	1.436	1.525
Accumulator Weight	lb	163.0	147.4	153.6	0	128.0	0	141.0	149.0
Hydraulic Motor Pump (for Braking)	kW	38.59	34.9	36.38	0	30.2	0	33.29	35.35
Hydraulic Motor Pump	lb	103.0	93.6	97.5	0	81.0	0	89.0	95.0
Engine Power	kW	60.0	54.91	56.96	51.93	48.39	44.75	52.67	55.53
Engine Weight	lb	150.0	143.2	145.7	139.46	137.0	130.0	140.0	144.0
Transmission Weight	lb	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
Differential Weight	lb	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
Inverter Weight	lb	50.0	50.0	50.0	50.0	50.0	50.0	50.0	50.0
Total Propulsion System	lb	1283.0	981.0	1102.0	804.0	596.0	380.0	848.0	1018.0
Vehicle Test Weight	lb	4314.0	3922.0	4079.0	3693.0	3422.0	3142.0	3750.0	3969.0
kW Required at Wheel for STC Acceleration		34.65	31.77	32.92	30.08	28.09	26.03	30.5	32.11
• Supplied by Hydropneumatic		17.38	15.72	16.39	0	13.61	0	15.0	15.92
• Supplied by Electric Motor		17.26	9.698	9.86	9.46	0	0	5.93	6.11
• Supplied by Engine		0	6.344	6.67	20.62	14.48	26.03	9.57	10.08
Vehicle Range (80% Battery Depletion)	miles	25.52	22.87	31.1	31.1	—	—	16.0	31.1
Annual Electricity Consumption	kW-hr	1847.0	1258.0	1542.0	1419.0	0	0	862.0	13.46
Annual Gasoline Energy Input to Stirling Engine Gasoline Consumption	kW-hr	2112.0	2704.0	2309.0	3073.0	4599.0	5223.0	2964.0	2390.0
Total Annual Energy Consumption	kW-hr	3959.0	3962.0	3851.0	4492.0	4599.0	5223.0	3826.0	3736.0
Annual Gasoline Consumption	gal								
• Stirling Engine		64.3	82.0	70.7	93.0	140.0	159.0	90.0	72.76
• Conventional I.C. Engine		—	—	—	—	206.0	223.0	—	—
Annual Average Miles per Gallon									
• Stirling Engine		155.0	120.0	142.0	106.0	71.0	62.0	110.0	137.0
• Conventional I.C. Engine		—	—	—	—	48.0	44.0	—	—

*Selected for conceptual design

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TABLE 5-3

HYBRID VEHICLE PERFORMANCE

	CASE I	CASE II	* CASE III	CASE IV	CASE V	CASE VI	CASE VII	CASE VIII
Battery Weight (lb)	496.0	278.3	385.5	348.53	0	0	170.0	337.0
Accumulator Volume (ft^3)	1.66	1.506	1.569	0	1.316	0	1.436	1.525
Vehicle Test Weight (lb)	4314.0	3922.0	4079.0	3693.0	3422.0	3142.0	3750.0	3969.0
Vehicle Range Miles (80% Battery Depletion)	25.52	22.87	31.1	31.1	0	0	16.0	31.0
Total Annual Energy Consumption (kWh)	3959.0	3962.0	3851.0	4492.0	4599.0	5223.0	3826.0	3736.0
Annual Gasoline Consumption (gal)								
• Stirling Engine	64.3	82.0	70.7	93.0	140.0	159.0	90.0	72.76
• Conventional I.C. Engine	—	—	—	—	206.0	223.0	—	—

*Selected for conceptual design

- Addition of a hydraulic accumulator to a Stirling hybrid system results in a greater percentage fuel savings than applying an accumulator to an internal combustion system.
- Case I (refer to Table 5-2) provides the least petroleum consumption.
- Case III (refer to Table 5-2) provides similar petroleum consumption but also results in less total vehicle weight and less total energy consumption.

Based on the above conclusions, the selected system is a parallel kinematic Stirling hybrid propulsion system with the engine sized for best performance at 90 km/h (56 mph) and batteries sized for a maximum range of 50 km (31.1 miles) (Case III or IV). This vehicle will have low petroleum consumption, minimum total vehicle weight and near-minumum total energy consumption.

Final determination regarding the addition of a hydraulic accumulator and incorporation of nickel-zinc batteries (as compared to lead-acid batteries), was made dependent upon the results of the life-cycle cost studies. Section 6.0 includes a detailed discussion of the reasons for the eventual, final selection of a hydraulic system and lead-acid batteries.

6.0 LIFE-CYCLE COST/TRADE-OFF STUDIES

In Section 5.0, various comparisons of different propulsion system configurations were performed and evaluated with respect to total fuel consumption and total energy consumption. The life-cycle cost/trade-off studies were performed to make the final selection of the proposed configuration. The life-cycle cost/trade-off studies provided the answers to the following considerations:

- Cost/benefit of short-term energy storage (hydraulic accumulator)
- Selection of either lead-acid or nickel-zinc batteries, based on least life-cycle cost
- Price of gasoline at which the proposed hybrid vehicle configuration is cost-effective to the consumer.

In the design of a hybrid vehicle, the weight of the battery is an important parameter. The procedure used to determine various trade-offs such as reduction in annual gasoline consumption, weight, and cost of the vehicles with battery weight treated as an independent variable are discussed in this section. Figure 6-1 is a flow chart describing this procedure, and the following subsections discuss each of the elements therein.

6.1 Determination of Total Vehicle Weight

The weight of the battery impacts on the power rating and, consequently, on the weight of the various components of the propulsion system, the structural design, and the overall vehicle weight.

Using the iterative procedure in Figure 6-2, the total vehicle weight, W_t, may be determined to meet the given mission requirements once the battery weight is specified. Assuming simple linear relationships between the power/energy ratings and the weights of the various propulsion system components shown in Table 6-1, the iterative procedure can be replaced by an explicit procedure. Details are given in Appendix B.

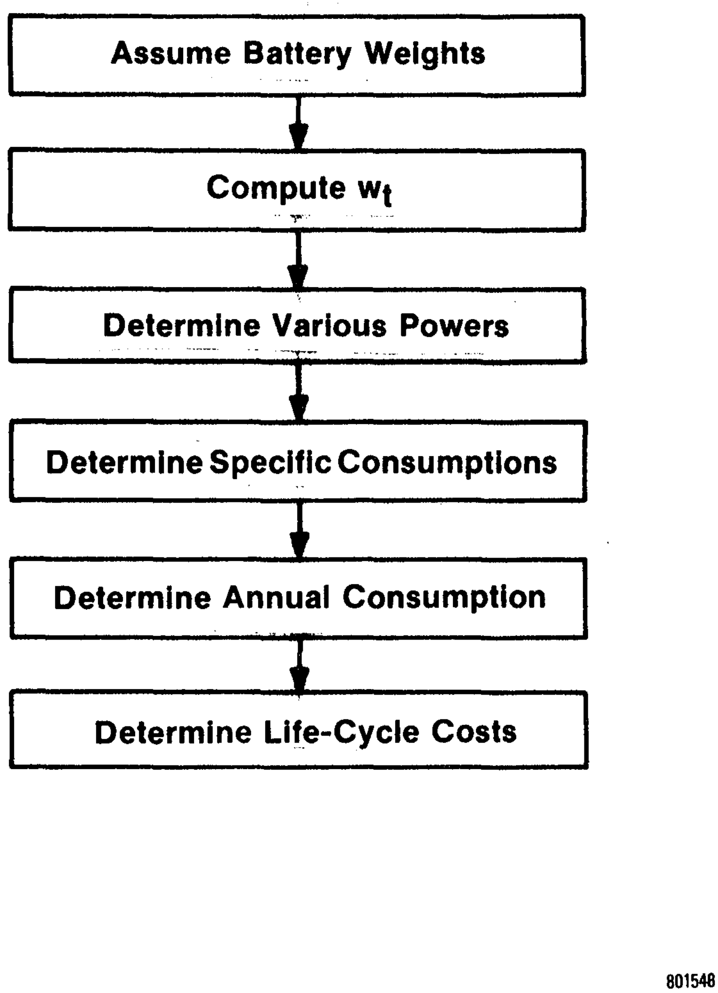

Fig. 6-1 Major Blocks in the Flow Chart for the Computation of Life-Cycle Costs

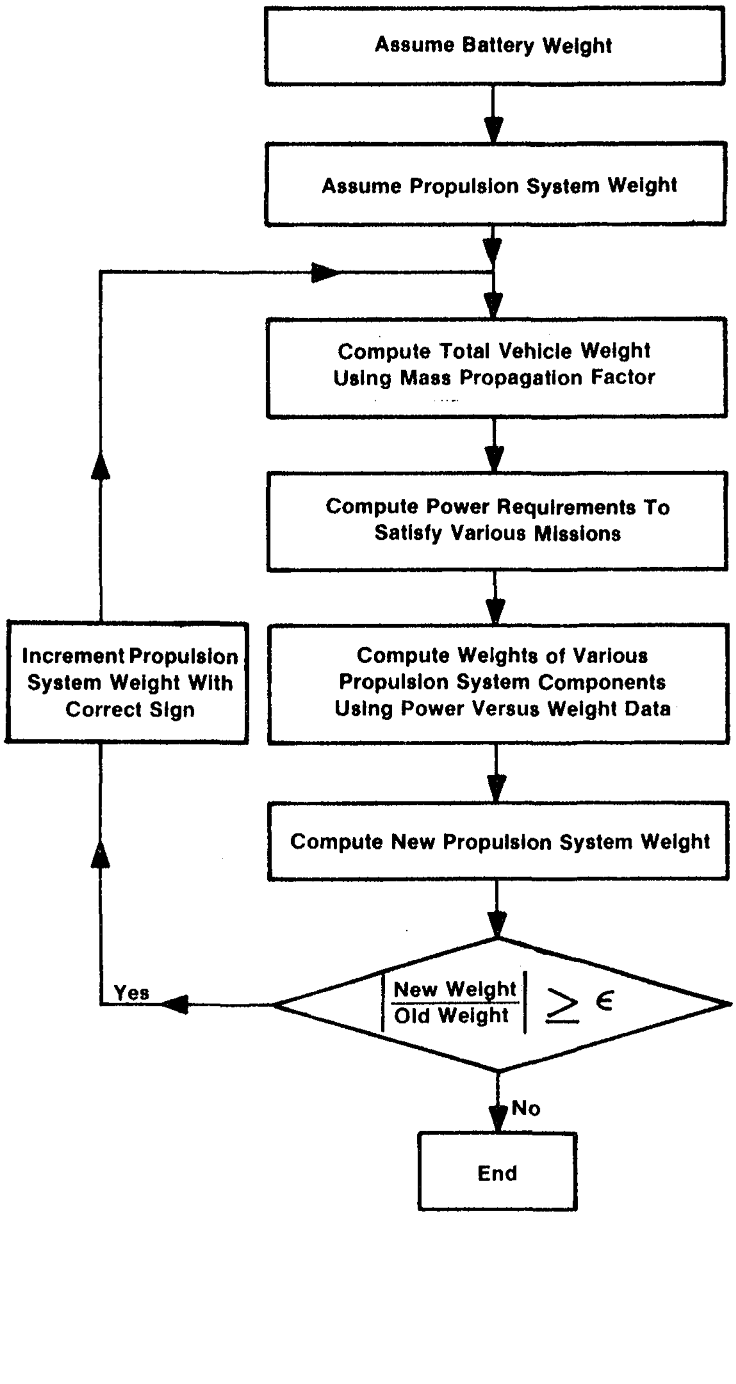

Fig. 6-2 Conceptual Design to Determine Total Vehicle Weight, W_t, in Terms of Battery Weight

TABLE 6-1

COMPONENT WEIGHTS OF THE PROPULSION SYSTEM

W_{engine} (lb) = (0.926 x hp) + 75

W_{motor} (lb) = 4.48 x hp

$W_{hydro\ motor/pump}$ (lb) = 2 x hp

$W_{accumulator}$ (lb) = 1.8 x watt-hours

6.2 Determination of Various Powers

Once the total vehicle weight, W_t, is known, power required at the wheels to satisfy various missions can be calculated by explicit analytical expressions. In the case of accelerations, a constant acceleration followed by acceleration at constant power is assumed. Using the weight (W_t) and the effective frontal area, the effects of rolling friction and aerodynamic drag can be analytically considered. Once the power required at the wheels is known, the power flowing through each component in the power train is determined as a simple linear expression in W_t by considering the efficiencies of various components, their relative place in the power train, and the operating strategy.

The power available from the battery is determined by its power density and its weight.

6.3 Determination of Specific Consumptions

Various specific consumptions of components shown in Table 6-2 are determined by using operating strategy (defined in Section 5.0), the power demands under different mission requirements, and the ratings of various power components. Details of this procedure are discussed in Appendices B, C and D.

6.4 Determination of Annual Consumption

The annual electricity consumption, gasoline consumption, and number of 80% Depth of Discharge (DOD) cycles of the battery are determined using the specific consumptions, the operating strategy, and the annual trip distribution statistics. Appendix E gives the details of the procedure used and the associated data.

6.5 Determination of Life-Cycle Costs

Considerations for total life-cycle cost are shown in Figure 6-3. The costs in terms of constant 1976 dollars are computed over a vehicle life of 10 years and 100,000 miles. The cost of fuel is computed at $0.40, $0.66, $0.92 and $1.19 per liter ($1.50, $2.50, $3.50 and $4.50 a gallon) in constant 1976 dollars, thus generating four total life-cycle cost curves. Appendix F gives the details of the implementation of this scheme.

TABLE 6-2

SPECIFIC CONSUMPTIONS REQUIRED IN THE CALCULATION OF ANNUALIZED CONSUMPTIONS

- Electric energy depletion from the battery during STC
- Gasoline consumption during STC
  - a) Before 80% DOD
  - b) After 80% DOD
- Electric energy depletion from the battery per kilometer at 90 km/h (per mile at 56 mph) cruise (if any)
- Fuel consumption per kilometer at 90 km/h (per mile at 56 mph)
  - a) Before 80% DOD
  - b) After 80% DOD

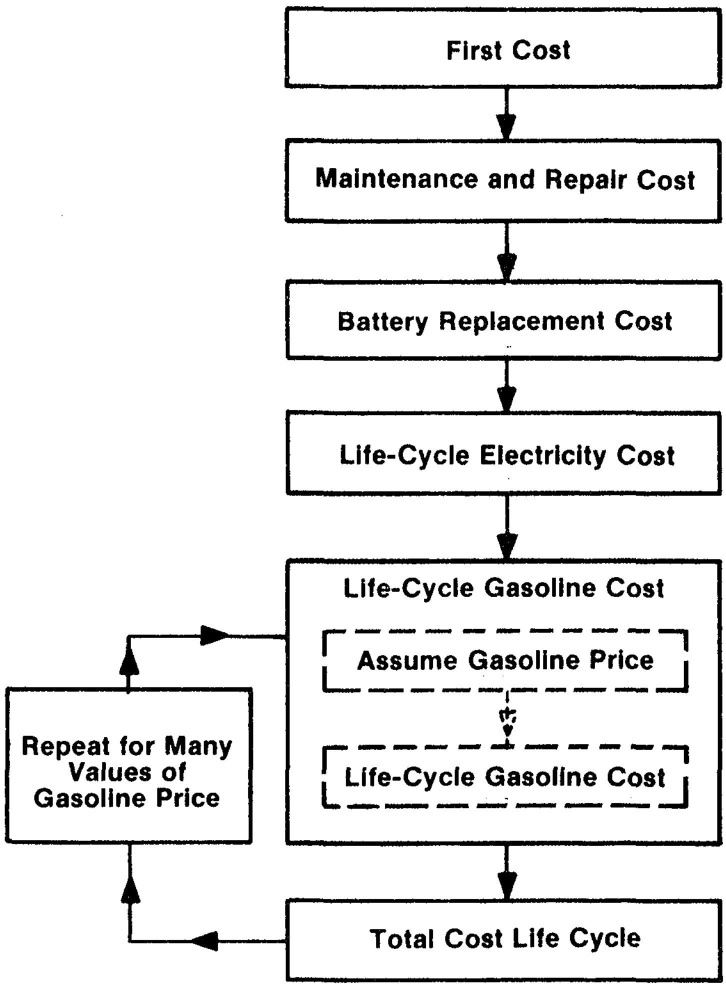

Fig. 6-3 Elements of Life-Cycle Cost and Flow Chart Used in Trade-Off Studies

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6.6 Life-Cycle Cost/Trade-Off Studies Results

Figures 6-4 through 6-8 show the results of the life-cycle cost studies with lead-acid batteries. In these figures, various parameters are plotted with battery weight as an independent variable. Figure 6-4 reveals an interesting point - as battery weight is increased, the total cost of the propulsion system (over the lifetime of 10 years) increases until the fuel price approaches \$0.92/liter (\$3.50/gal). Therefore, the added cost of a second power plant (electric) is not balanced by the cost of fuel saved until the fuel approaches a cost of \$0.92/liter (3.50/gal). The first cost, the life and the replacement cost of the battery, are important elements in the overall economic picture.

Figure 6-4 also shows that for a constant weight of batteries, the life-cycle cost of a vehicle with hydraulic short-term energy storage is greater than the life-cycle cost of a similar vehicle without short-term energy storage until the price of gasoline reaches \$0.66/liter (\$2.50/gallon). Case III (the configuration selected in section 5.0) is highlighted on the figures.

Figure 6-5 is plotted to show annual gasoline consumption as an independent variable. Annual gasoline consumption is related to battery weight (i.e., all-electric vehicle equals zero gasoline consumption). The curves show how the total propulsion system life-cycle costs would vary as the propulsion system is designed to limit annual gasoline consumption. Obviously, the major selection criterion is to design a hybrid vehicle that minimizes gasoline consumption with little or no impact on life-cycle cost.

As shown in Figure 6-5 (also in 6-4), the life-cycle costs of an all-gasoline-powered Stirling engine vehicle is less than the life-cycle cost of a hybrid vehicle until the price of gasoline approaches \$0.92/liter (\$3.50/gallon).

The maximum annual fuel consumption is 530 liters (140 gal) for the all-Stirling design with hydraulic accumulator, and 606 liters (160 gal) without

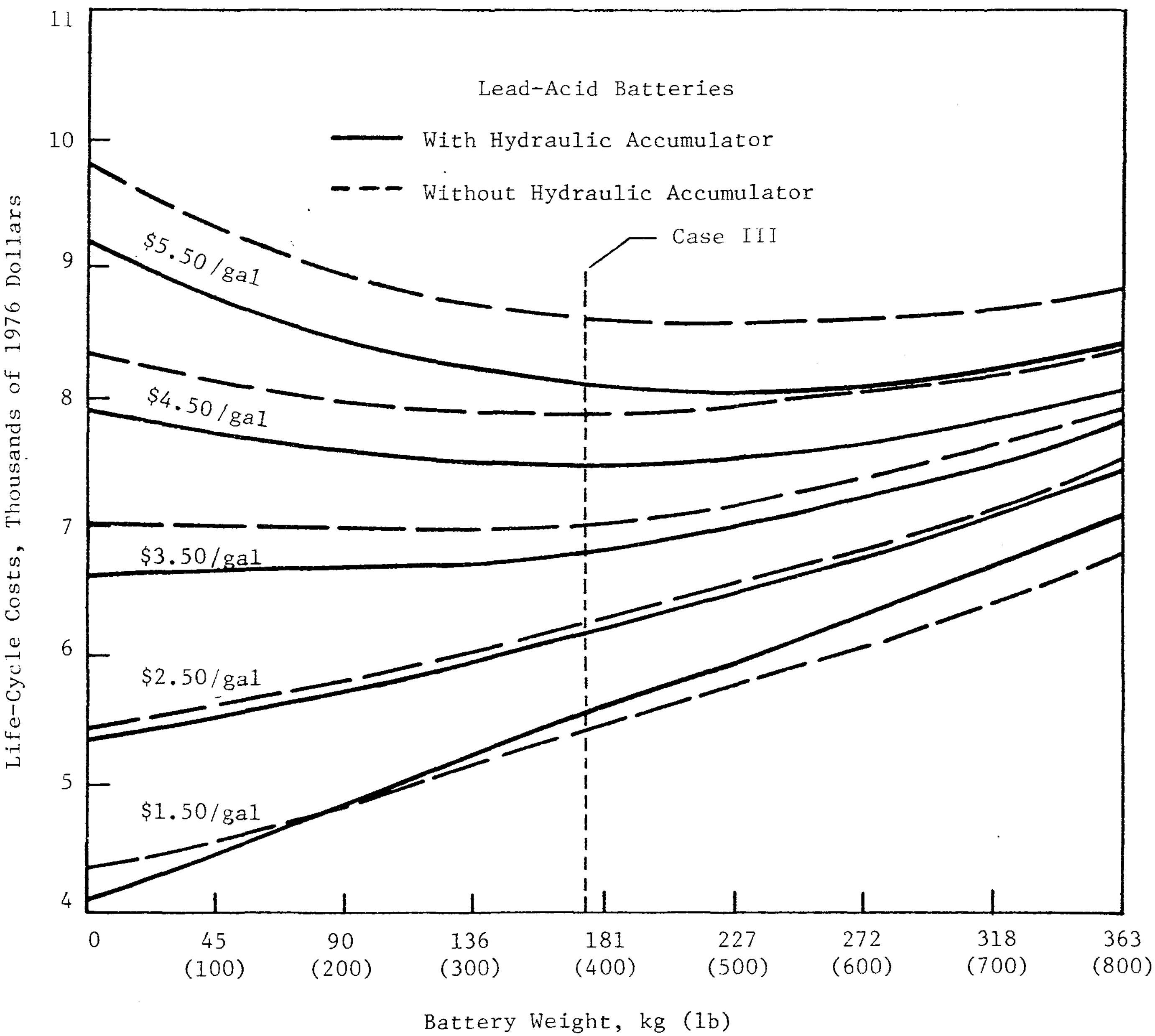

Fig. 6-4 Life-Cycle Costs of Propulsion System versus Battery Weight at Varying Gasoline Prices

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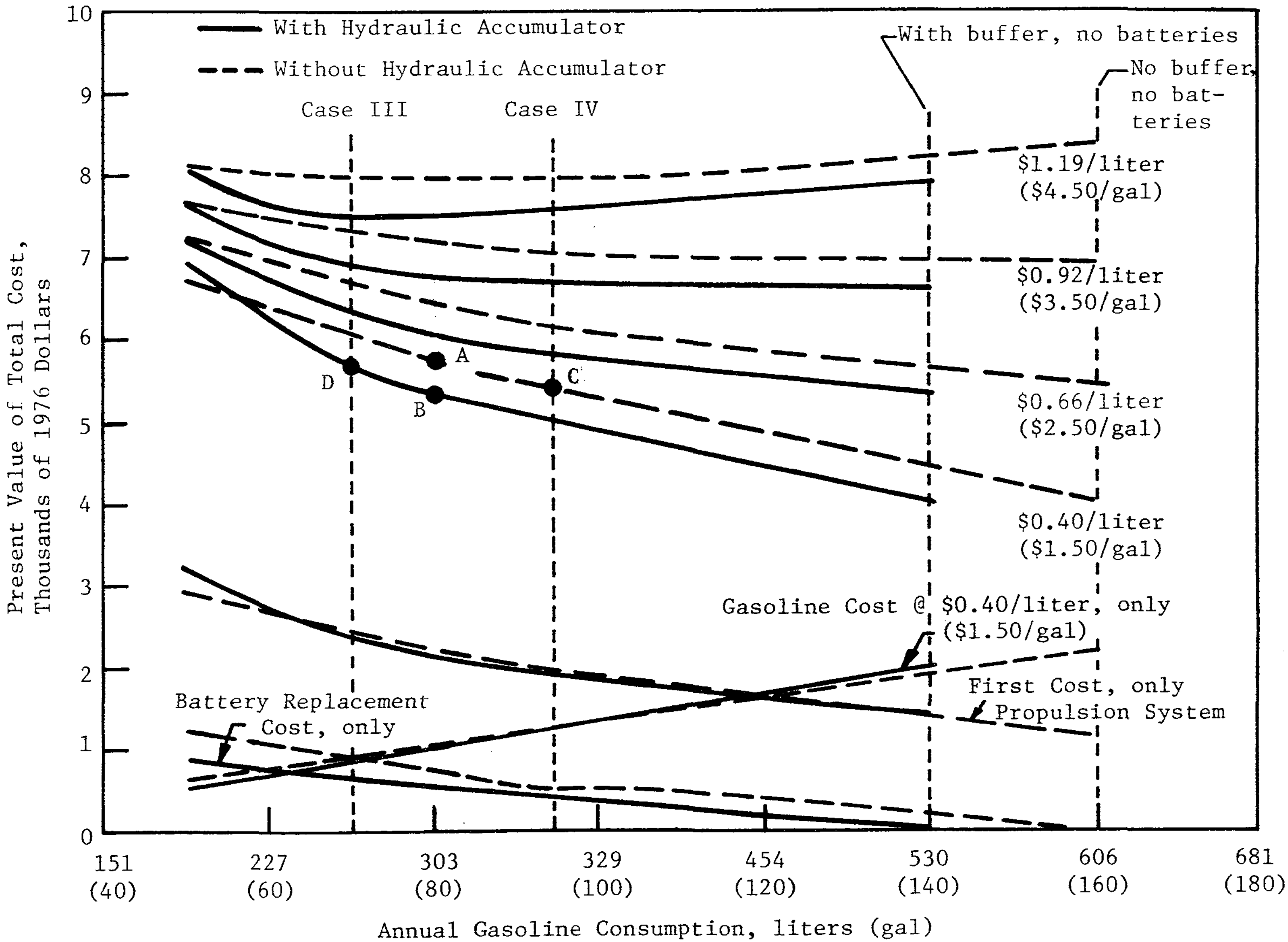

Fig. 6-5 Life-Cycle Cost of Propulsion System versus Annual Gasoline Consumption, Lead-Acid Batteries

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it. These points correspond to a Stirling-cycle, heat-engine-only design without batteries. As batteries are added, the fuel consumption decreases (to the left on the curve). Points A and B represent the total cost for the two different designs which satisfy identical performance specifications. Each of the two design vehicles consumes 303 liters (80 gal) of fuel annually. Thus, design B is more economical than design A (i.e., hydraulic accumulator is beneficial). Weight of battery for design A and B are, of course, different.

Figure 6-5 also shows another approach to evaluating the benefit/cost trade-off of hydraulic storage, which is to examine the difference between a similar system with and without the energy system. Case III and Case IV propulsion configurations represent the vehicle with and without the hydraulic system, respectively. The addition of the hydraulic accumulator saves an additional 87 liters (23 gallons) on an annual basis; however, the propulsion system without a hydraulic accumulator has less life-cycle cost (point C) than a system with a hydraulic accumulator (point D) when gasoline is priced at $0.40/liter ($1.50/gallon). This conclusion changes when gasoline is priced at $0.66/liter ($2.50/gallon). Since the hybrid configuration does not "break even" with an all-Stirling-powered vehicle until gasoline reaches $0.92/liter ($3.50/gallon), hydraulic short-term energy storage is beneficial since it "pays for itself" when gasoline reaches $0.66/liter ($2.50/gallon).

Figure 6-5 also shows the following trends: first cost of the propulsion system, battery replacement cost, and gasoline cost as a function of annual gasoline consumption, which is a function of battery weight. Other parameters to determine life-cycle cost (defined in Figure 6-3) are not shown.

The difference between the two identical performance designs with identical annual fuel consumption, with and without the hydraulic accumulator, is the battery weight and electric range. These parameters are shown in Figure 6-6, plotted against fuel consumption. In the design with a hydraulic accumulator, as little as 175 kg (385 lb) of battery weight causes almost 50% reduction in the annual fuel consumption. This figure stands out in strong contrast with the more than 454 kg (1000 lb) of battery weight required by a low-performance,

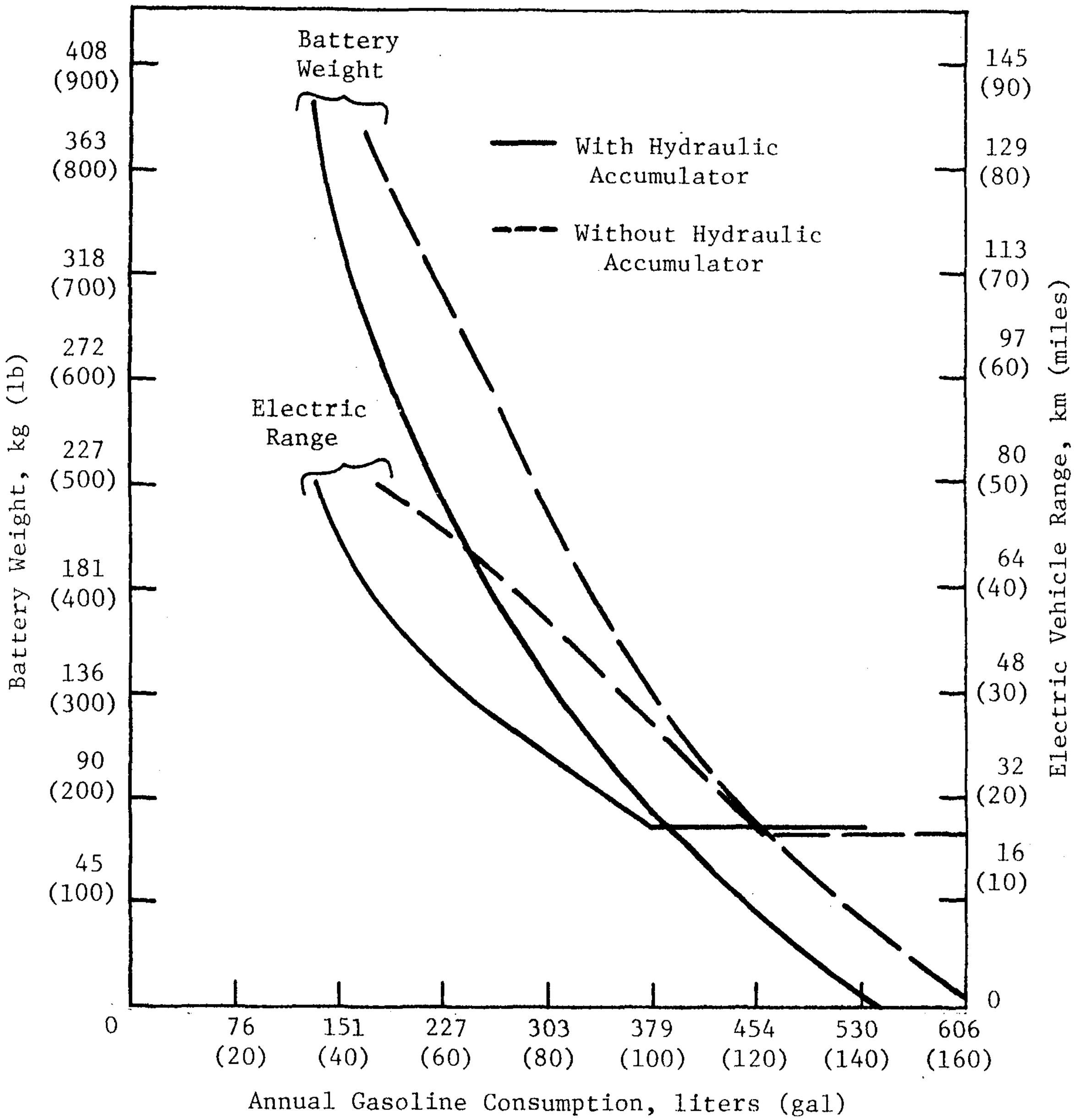

Fig. 6-6 Battery Weight and Electric Range versus Annual Gasoline Consumption

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two-passenger, all-electric car. In this curve, the electric range is defined as the range until batteries reach 80% DOD; that is, the batteries are limited to a specific power dependent on battery weight, and the STC cycle is performed by both the heat engine and the batteries. Even at 400 kg of battery weight, the engine supplements the power required during acceleration of the STC. Until the battery weight exceeds ∿170 kg and, therefore, the battery specific power exceeds that required at 45 mph cruise, the so-called electric range is flat, as shown in Figure 6-6.

Figure 6-7 shows life-cycle costs versus annual gasoline consumption for designs using Ni-Zn batteries, which have higher power and energy densities than lead-acid batteries. These batteries are more expensive and have fewer discharge cycles to replacement. Overall costs for identical performance and fuel consumption are higher than those for the lead-acid batteries. The battery weights, however, are smaller than those of lead-acid batteries in equivalent cars, as shown in Figure 6-8. The battery characteristics used are given in Table 6-3. The details of a sample design with 175 kg (385 lb) of lead-acid batteries with Stirling engine and hydraulic accumulator are shown in Section 5.0, Tables 5-2a and b, Case III.

6.7 Conclusions

The detailed computer life-cycle cost/trade-off studies substantiated the conclusions reached in Section 5.0. In addition, the results presented in Figures 6-4 through 6-8 support the following conclusions:

- The life-cycle cost of the parallel hybrid system becomes equal to the life-cycle cost of an equivalent all-Stirling heat engine vehicle when fuel cost approaches $0.92 per liter ($3.50/gallon)
- A parallel Stirling hybrid propulsion system can reduce fuel consumption to 30% over a present conventional (non-hybrid) 6-passenger, internal combustion vehicle.
- At $0.92 per liter ($3.50/gallon) fuel cost, the life-cycle cost (Figure 6-4) is constant with battery weight between 0.0-181 kg (0-400 lb). Gasoline consumption, however, is

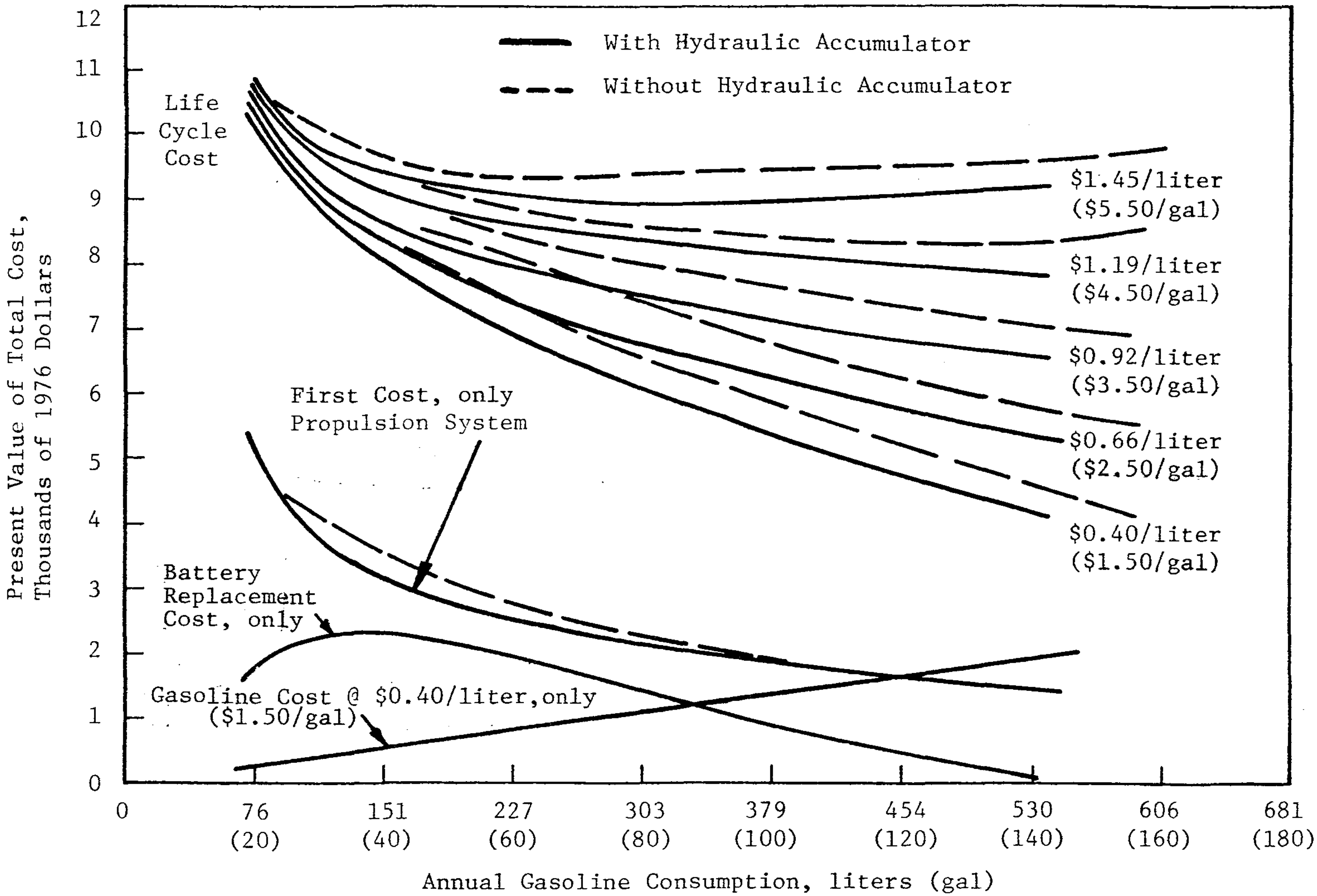

Fig. 6-7 Life-Cycle Costs of Propulsion System versus Annual Gasoline Consumption, Ni-Zn Batteries

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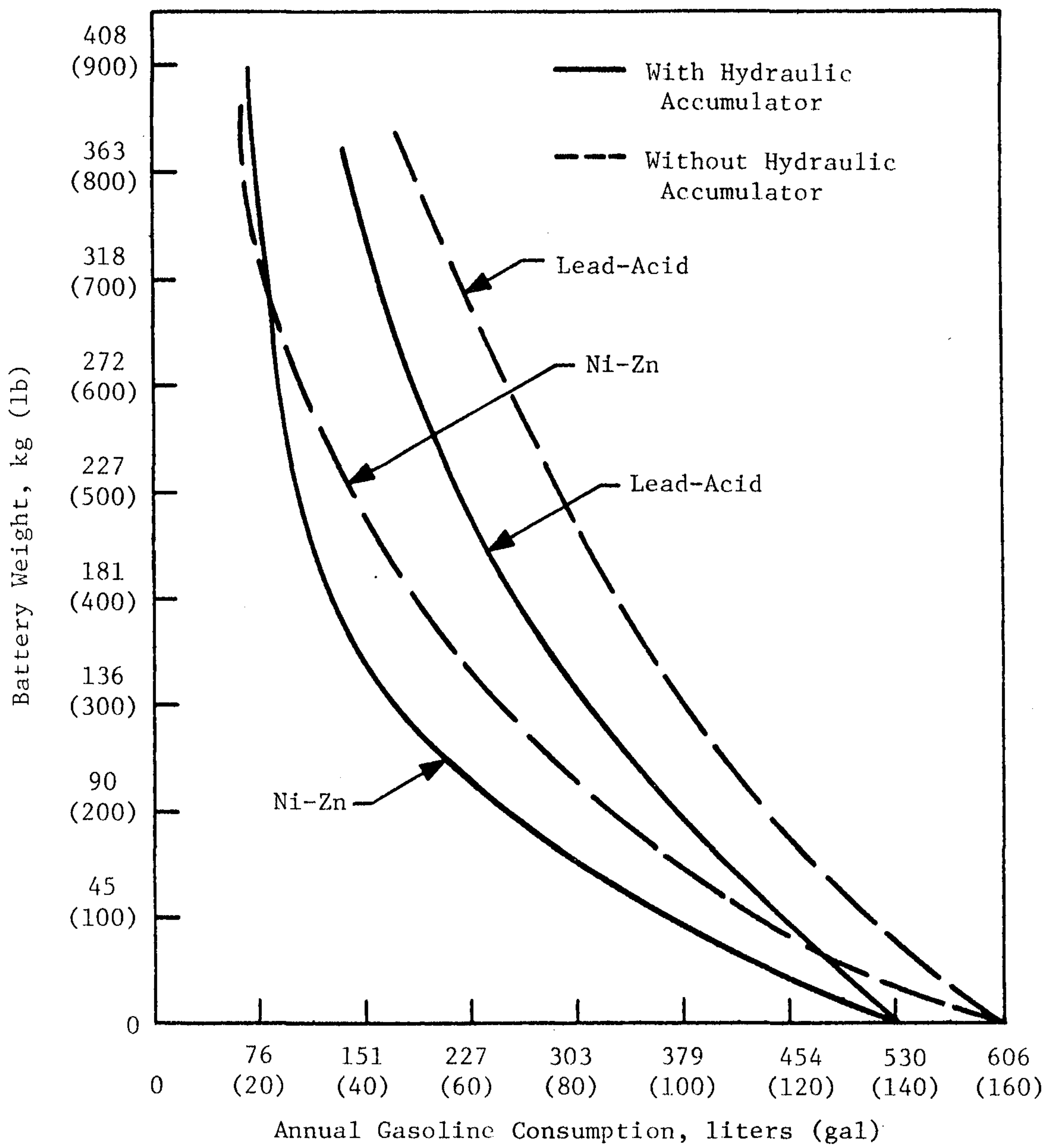

Fig. 6-8 Comparison of Battery Weights with Ni-Zn and Lead-Acid Batteries for Equivalent Designs

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TABLE 6-3

BATTERY CHARACTERISTICS

	Battery Types	
	ISOA Lead-Acid	Nickel-Zinc
Specific Energy[a], W-h/kg	40	80
Specific Power [b], W/kg	100	150
Cycle Life[c]	800	500
Cost, $/kW-h	50	75
Energy Efficiency	>.6	0.7

(a) 3 hr discharge rate and an 8 hr charge rate

(b) 15 sec discharge rate

(c) 3 hr discharge rate and an 8 hr charge rate correspondingly reduced from 606 liters (160 gallons) to 303 liters (80 gallons); therefore, a Stirling hybrid system utilizing 181 kg (400 lb) of batteries significantly reduces fuel consumption by 50% over an equivalent all-Stirling heat engine vehicle.

- At \$0.92 per liter (\$3.50/gallon) fuel cost (Figure 6-4), hydraulic accumulators are attractive.
- Lead-acid batteries have lower life-cycle cost than nickel-zinc batteries for equal performance.

7.0 SUMMARY/VEHICLE CONFIGURATION

7.1 Conceptual Vehicle Configuration

The detailed evaluation and life-cycle cost/trade-off studies led to the selection of the advanced parallel Stirling-engine/electric, hybrid propulsion system as the most attractive configuration for reducing petroleum consumption and total energy consumption. The vehicle is a full-size, full-performance, 6-passenger vehicle. Its propulsion system incorporates a hydraulic accumulator for short-term energy storage and utilizes engine on-off operational strategy to minimize petroleum consumption. Such a vehicle can significantly reduce petroleum consumption (greater than 70%) over present internal combustion vehicles, and the proposed hybrid system becomes cost-effective (i.e., no cost penalty to the consumer over the life of the vehicle) when the price of petroleum approaches $0.92 per liter ($3.50/gallon).

Major components of the proposed advanced hybrid propulsion system are as follows:

- Engine
  - kinematic Stirling
  - design point - 10.5 kW
  - peak engine power - 52 kW
  - efficiency at design point - 42%
- Batteries
  - 181 kg (400 lb) of lead-acid batteries
  - 7 12-volt batteries
- Hydraulic system*
  - accumulator - 0.045 m^3 (1.6 ft^3)
  - motor/pump - 36.4 kW

---

*The alternate approach of using a hydromechanical CVT was not sized for this study.

- Electric motor
  - ac 3ϕ induction
  - 12.8 kW
  - 8000 r/min (rpm) at no load
  - 20 cm wide x 38 cm long (8" x 15")
- Inverter: variable frequency controller
  - 46 cm x 50 cm x 20 cm (18" x 20" x 8")
- Microprocessor control
  - 15 cm x 15 cm x 8 cm (6" x 6" x 3")

The definitions of the electric motor, inverter and microprocessor control are based primarily on Reference [1] and are modified to the power requirements of the proposed hybrid system. A conceptual layout of the advanced propulsion system integrated into a possible vehicle configuration is presented in Figure 7-1; various components and their relative size and location within the vehicle are indicated.

The major development effort for the proposed hybrid system is based on the incorporation of a hydraulic accumulator, coupled with a hydraulic pump/motor or hydromechanical transmission, to provide ease of interface with the vehicle and to ensure the benefits of short-term energy storage.

The major development of automotive Stirling engines is ongoing under separate contract. As envisioned for hybrid operation, the Stirling engine is smaller than those presently under development and has a lower power operating design point. A major feature of the hybrid system is the on-off operating strategy of the Stirling engine; such a capability may be inherent in the design of the Stirling engine but must be developed.

7.2 Conclusion

A well-designed and well-developed parallel Stirling hybrid system can significantly reduce petroleum consumption (greater than 70%) over present internal combustion, conventional (nonhybrid), 6-passener vehicles.

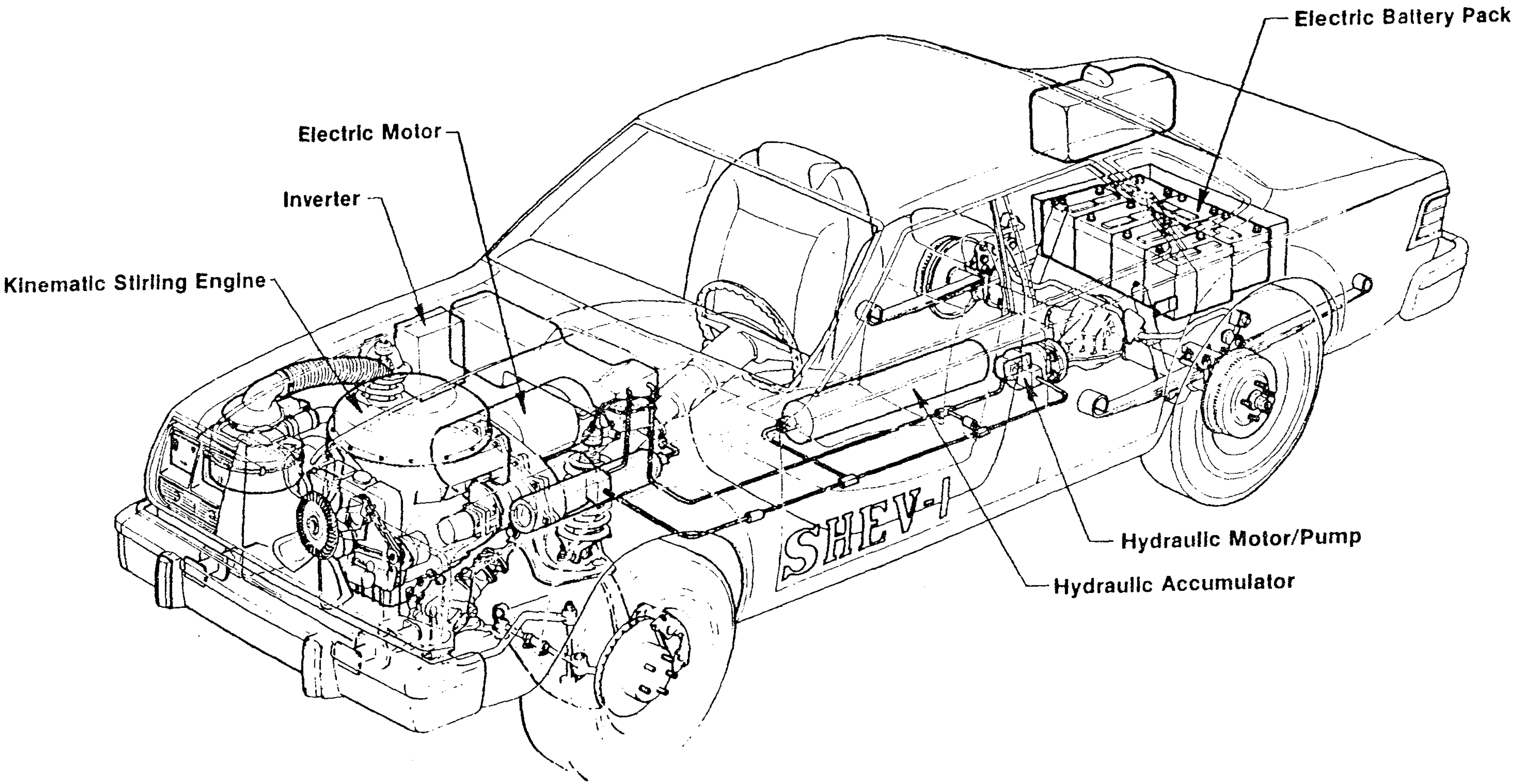

Fig. 7-1 Conceptual Layout of Advanced Propulsion System

APPENDIX A

EVALUATION AND COMPARISON OF SHORT-TERM ENERGY STORAGE SYSTEM WEIGHTS

Weight of Battery Buffers

$$m_B = \frac{\Delta E}{e}$$

where

m_B = mass of the buffer system

ΔE = energy to be stored

e = specific energy density

Now

$$\Delta E = \frac{1}{2} (m_B + m)V^2$$

where

m = mass of vehicle without buffer system

V = cruising speed of STC

Hence,

$$m_B = \frac{1}{2} (m + m_B) \frac{V^2}{e}$$

$$m_B \left(1 - \frac{1}{2} \frac{V^2}{e}\right) = \frac{1}{2} \frac{mV^2}{e}$$

Hence

$$m_B = \left(\frac{1}{2} \frac{mV^2}{e}\right) / \left(1 - \frac{V^2}{2e}\right)$$

Data used in computation

m = 1588 kg (3500 lb)

V = 72 km/h (45 mph)

e = 100 watts/kgm (lead-acid battery)

e = 150 watts/kgm (Ni-Zn battery)

The values of e for the batteries are specified by NASA-Lewis in the contract document.

Weight of Hydraulic System

Reference [2]* gives the following equation for the weight of the hydraulic buffer systems shown in Figure A-1.

$$W_h = 1.1 \left[\Delta E \left(\frac{1}{e_h} + 1 \right) + \frac{HP_p}{\eta_p} \right] ,$$

where W_h is the weight of hydraulic system in pounds, ΔE is the energy storage requirement in W_h, e_h is the specific energy of the hydraulic accumulator in W_h/lb, HP_p is the peak horsepower requirement, and η_p is the efficiency of the hydraulic pump/motor. The equation assumes that 10% of the weight is piping, valves, and assorted hardware, that the weight of the hydraulic fluid is numerically equivalent in pounds to the specific energy in watt-hours, and that hydraulic pump motors with a power-to-mass ratio of about 1 hp/lb can be obtained.

Now

$$\Delta E = \frac{1}{2} (m + W_h) V^2$$

$$HP_p = \frac{\Delta E}{T}$$

where

T = time of discharge

Hence

$$W_h = 1.1 \, \Delta E \left[\left(\frac{1}{e_h} + 1 \right) + \frac{1}{T\eta_p} \right]$$

$$= 1.1 \left[\frac{1}{2} (m + W_h) V^2 \right] \left[\left(\frac{1}{e_h} + 1 \right) + \frac{1}{T\eta_p} \right]$$

*References are located at the end of this report.

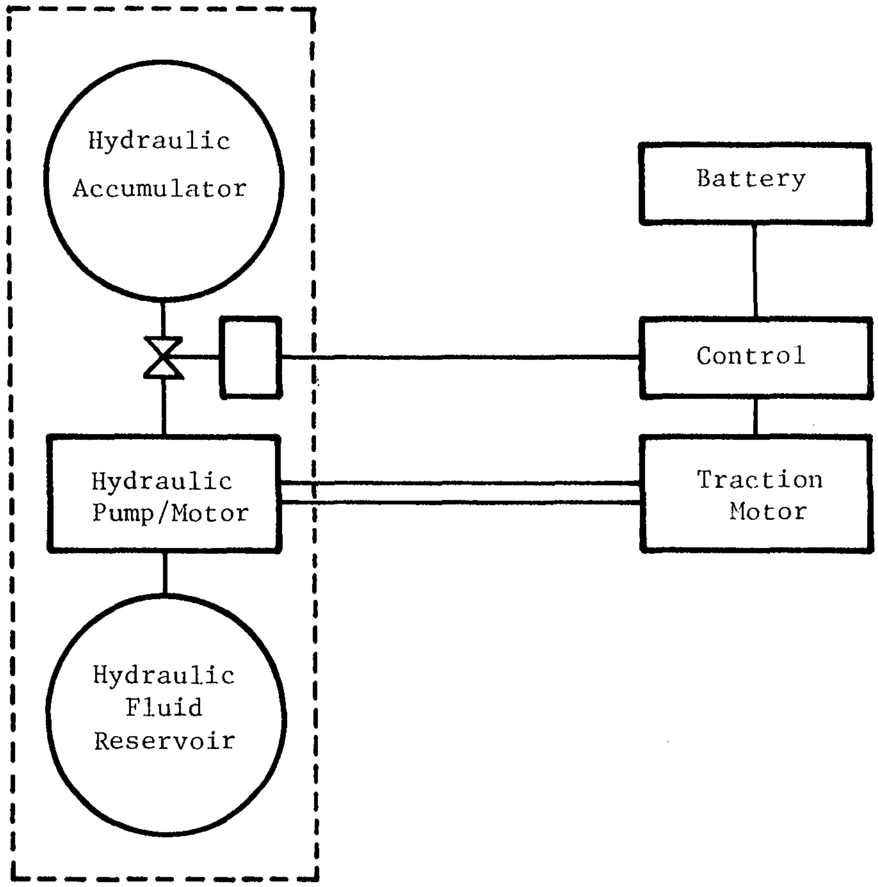

Fig. A-1 Candidate Hydraulic Accumulator Energy Storage System

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$$W_h \left\{1 - 1.1 \frac{V^2}{2} \left[\left(\frac{1}{e_h} + 1\right) + \frac{1}{T\eta_p}\right]\right\}$$

$$= 1.1 \frac{mV^2}{2} \left[\left(\frac{1}{e_h} + 1\right) + \frac{1}{T\eta_p}\right]$$

Hence

Let

$$x = 1.1 \frac{V^2}{2} \left[\left(\frac{1}{e_h} + 1\right) + \frac{1}{T\eta_p}\right]$$

$$W_h = \frac{xm}{1 - x}$$

Data for the computation are:

m = 1588 kg (3500 lb)

V = 72 km/h (45 mph)

e_h = 1.1 watt-hours/kg (0.5 watt-hours/lb)

η_p = 0.9

Weight of the Flywheel System

Reference [2] gives an expression for computing the weight of a flywheel buffer system similar to the one shown in Figure A-2. This weight is computed using the expression for the weight of the Garrett Near Term Electric Vehicle as described in Reference [3]. The actual weight of the flywheel system used in that vehicle was found to be approximately 2.75 times the above computed weight. The curve plotted in Figure 5-1 on page 62 for the flywheel is computed by taking the expression for the determination of the weight of the flywheel system from Reference [2] as exhibiting the trend. Actual numerical value is obtained by multiplying the weight computed from the above referred expression by 2.75. Thus, the computed weights are made compatible with one point corresponding to the Garrett vehicle of Reference [2].

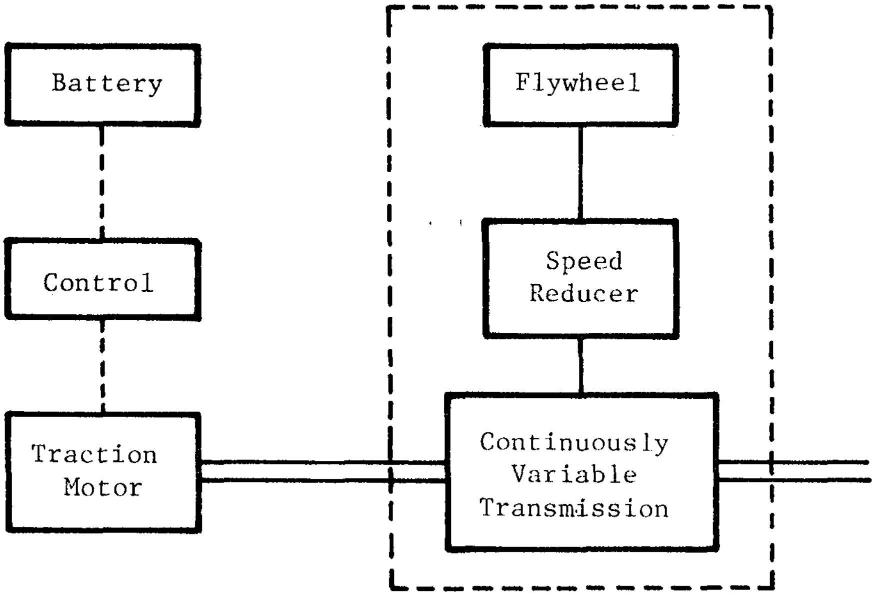

Fig. A-2 Candidate Flywheel Energy Storage System

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The weight W_f of the flywheel system can be expressed by

$$W_f = 2.75 \left[\frac{\Delta E}{e_f \eta_f} + 10 \left(\frac{HP_p}{\eta_{sr}} \right)^{0.232} + 20 \left(\frac{HP_p}{\eta_t} \right)^{0.475} \right]$$

where ΔE is the energy storage requirement in W_h, e_f is the specific energy of the flywheel, η_f the efficiency of the flywheel, HP_p the peak power requirement, and η_{sr} and η_t are the efficiencies of the speed reducer and the CVT, respectively.

The effect of the additional buffer system weight on ΔE is considered in a manner similar to the hydraulic system. The only exception is that an explicit expression for the weight of the hydraulic is possible, whereas for the flywheel, the expression is computed by the solution of the nonlinear equation above.

Data used for the computation include:

e_f = 9.9 watt-hours/kg (4.5 watt-hours/lb)

η_f = 0.7

η_{sr} = 0.9

η_t = 0.85

APPENDIX B

DETERMINATION OF THE TOTAL VEHICLE WEIGHT, W_t FOR A GIVEN BATTERY WEIGHT

Table B-1 gives the specifications and equations for accounting for the mass propagation effect (i.e., the increase in the structural weight of the vehicle due to an increase in the propulsion system component weight).

Replacing W_s in Equation (B.1) by Equation (B.4), from Table B-1,

$$W_G = 0.23\ W_G + W_{PL} + W_P + W_F \quad .$$

Hence,

$$(1 - 0.23)\ W_G = W_{PL} + W_P + W_F \quad .$$

Hence,

$$W_G = \frac{W_{PL} + W_P}{0.77} + \frac{W_F}{0.77} \quad . \tag{B.5}$$

Using Equations (B.2), (B.3) and (B.5),

$$\begin{aligned} W_T &= W_G - W_{PL,max} + W_{TL} \\ &= \frac{W_{PL} + W_P}{0.77} + \frac{W_F}{0.77} - W_{PL,max} + W_{TL} \\ &= \left(\frac{W_{PL} - 0.77\ W_{PL,max}}{0.77} + \frac{W_F}{0.77} + W_{TL}\right) + \frac{W_P}{0.77} \quad . \end{aligned}$$

Hence,

$$W_T = W_{CONSTANT} + \frac{W_P}{0.77} \quad , \tag{B.6}$$

where

$$W_{CONSTANT} = \left(\frac{W_{PL} - 0.77\ W_{PL,max} + W_F}{0.77} + W_{TL}\right) \quad . \tag{B.7}$$

The values of W_{TL}, W_F and $W_{PL,max}$ for various mission/vehicles are specified of the contract document. That table is reproduced here as Table B-2 for convenience.

TABLE B-1

PARAMETRIC REPRESENTATION OF WEIGHT*

Symbol	Definition	Formula
$W_{PL,\ max}$	Maximum design payload	-
W_{TL}	Test payload	-
W_F	Fixed weight	-
W_G	Gross vehicle weight	$W_G = W_S + W_{PL} + W_P + W_F$ (B1)
W_C	Curb weight	$W_C = W_G - W_{PL,\ max.}$ (B2)
W_T	Test weight	$W_T = W_C + W_{TL}$ (B3)
W_S	Structure and chassis weight	$W_S = 0.23\ W_G$ (B4)
W_P	Propulsion weight	Determined by contractor

*This table has been reproduced from the Contract's Statement of Work, Appendix A, Table A-1, and also appears as Table 4-1 in the main report.

TABLE B-2

MISSION/VEHICLE SPECIFIC WEIGHT CONSTANTS*

Constant	Units	Mission/Vehicle				
		A	B	C	D	E
$W_{PL,\ max.}$	kg (lb)	166(366)	272(600)	508(1120)	1043(2300)	3629 (8000)
W_{TL}	kg (lb)	83(183)	126(300)	254 (560)	522(1150)	1815 (4000)
W_F	kg (lb)	204(450)	408(900)	612(1350)	816(1800)	5200(11464)

*This table has been reproduced from the Contract's Statement of Work, Appendix A, Table A-2, and also appears as Table 4-2 in the main report.

In these calculations the following has been assumed

$$W_{PL} = W_{PL,max} \quad .$$

Thus Equation (B.7) reduces to:

$$W_{CONSTANT} + \left(\frac{0.23\ W_{PL,max} + W_F}{0.77}\right) + W_{TL} \tag{B.8}$$

Using values from Table B-2 for vehicle C the following is obtained.

$$W_{CONSTANT} = 1200 \text{ kg } (2647 \text{ lb})$$

The propulsion weight, W_p, is the sum of the weights of the individual components in the propulsion train. Thus,

$$W_P = W_{ENGINE} + W_{MOTOR} + W_{BATT} + W_{HYDROM} + W_{ACCUM} + W_M \tag{B.9}$$

where,

W_{ENGINE} = heat engine weight

W_{MOTOR} = electric motor weight

W_{BATT} = battery weight

W_{HYDROM} = hydraulic motor/pump weight

W_{ACCUM} = hydraulic accumulator weight

W_M = total weight of other power train components (such as differential, transmission, battery charger/controller for the electric motor, etc.)

For this study, W_M is assumed to be 113 kg (250 lb).

The weight of the hydraulic accumulator is dependent on the energy storing capability. The weights of the heat engine, motor, and hydraulic motor/pump are strongly dependent on their power handling capability. In turn, the power/energy handling capability required of these components depends on the total weight, W_t of the vehicle. Thus the weight, W_t, can be determined by an iterative procedure indicated in Section 6.0, Figure 6-2.

However, an explicit procedure can be developed by making certain assumptions about the relationship between the weight of the components and their ratings.

Appendix C shows that by making such assumptions the following relationships result:

$$W_{MOTOR} = C_1 W_T + P_1 \tag{B.10}$$

$$W_{HYDROM} = C_2 W_T + D_2 \tag{B.11}$$

$$W_{ACCUM} = C_3 W_T + D_3 \tag{B.12}$$

$$W_{ENGINE} = C_4 W_T + D_4 \tag{B.13}$$

where C_i, D_i $(i = 1,\ldots,4)$ are constants. (Their values are derived in Appendix C.)

Substituting Equations (B.10) through (B.13) in Equation (B.9),

$$W_P = (C_1 + C_2 + C_3 + C_4)W_T + (D_1 + D_2 + D_3 + D_4)$$

$$+ W_{BATT} + W_M$$

$$W_P = C W_T + D$$

where

$$C = C_1 + C_2 + C_3 + C_4$$

$$D = D_1 + D_2 + D_3 + D_4 + W_{BATT} + W_M$$

Substituting this value for W_P in Equation (B.6),

$$W_T = W_{CONSTANT} + \left(\frac{C}{0.77}\right)W_T + \frac{D}{0.77} \quad .$$

Hence,

$$W_T \left(1 - \frac{C}{0.77}\right) = W_{CONSTANT} + \frac{D}{0.77}$$

Hence,

$$W_T = \frac{W_{CONSTANT} + \frac{D}{0.77}}{\left(1 - \frac{C}{0.77}\right)} \quad . \tag{B.15}$$

When the battery is too small, it may not have enough power capability to absorb all the power available during regenerative braking. Under such circumstances, it is futile to have the motor power rating equal to the regenerative power. Instead, motor power rating should match the battery peak power capability. Then, the motor weight is known explicitly in terms of battery weight, as can be seen from Appendix C, Equation (C.11). In such a case, the above algorithm for computing W_T is modified to:

$$C = C_2 + C_3 + C_4$$

$$D = D_2 + D_3 + D_4 + W_{BATT} + W_M + W_{MOTOR}$$

$$W_T = \frac{W_{CONSTANT} + \frac{D}{0.77}}{\left(1 - \frac{C}{0.77}\right)} \tag{B.16}$$

APPENDIX C

DETERMINATION OF COMPONENT WEIGHT RELATIONSHIPS

Engine Weight

Table 2-1 on page 10 specifies different missions that a vehicle must be able to perform. A total of 13 different power levels are defined by those missions. These power levels include:

- Special Test Cycle (STC) cruising at 72 km/h (45 mph)
- 90 km/h (56 mph) cruising
- STC acceleration
- STC braking
- Nine conditions specified in Table 2-1 B. Additional Design Goals.

The power required at the wheels to satisfy these missions is dependent on the mission specification, total vehicle weight, W_t, the coefficient of tire rolling friction, and the effective aerodynamic frontal area, $C_D A$. Table 2-1 specifies all of these parameters except the vehicle weight, W_t. Thus, the 13 power levels may be computed as functions of the total vehicle weight, W_t. In general, these functions are linear in W_t and have the following form:

$$\text{Power} = K_1 W_t + K_2 \qquad \text{(C.1)}$$

where K_1 and K_2 are constants. Their values are determined by the mission requirements and $C_D A$.

Using the data for Vehicle C from Table 2-1 over a wide range of vehicle weights, W_t, the highest power requirements were determined to be for the mission that requires a speed of 90 km/h (56 mph) to be attained in 300 m (984 ft) from a full stop on a 6% grade. The constants K_1 and K_2 required in Equation (C.1) are evaluated for this case, assuming constant power acceleration for performing this mission.

Now, the peak power available from the engine and a given engine weight, W_{ENGINE}, are related by:

$$W_{ENGINE} = 0.926\, P + 75 \qquad \text{(C.2)}$$

where

P = peak available power (hp)
W_{ENGINE} = Stirling engine weight (lb) .

Using Equations (C.1) and (C.2) with the appropriate constants, then,

$$W_{ENGINE} = 0.0162\ W_t + 79.7 \tag{C.3}$$

Hydraulic Accumulator Weight

The hydraulic accumulator is designed to store all the energy available during the braking period of the STC.

This energy is given by:

$$E_{Braking} = [KE_1 - (E_{Drag} + E_{Rolling})] \tag{C.4}$$

where

KE_1 = kinetic energy at the end of coasting in STC

E_{Drag} = energy lost during the braking period in overcoming aerodynamic resistance

$E_{Rolling}$ = energy lost during the braking period in overcoming rolling friction

For Vehicle C the effective aerodynamic area C_DA is specified. The velocity at the beginning of the braking period, and the time period during braking is specified in the STC specification. Thus, the quantities on the right hand side of Equation (C.4) can be written as:

$$KE_1 = K_3\ W_t$$

$$E_{Drag} = K_4$$

$$E_{Rolling} = K_5\ W_t$$

where K_3, K_4, K_5 are constants evaluated from STC specification and value of C_DA for Vehicle C.

Thus,

$$E_{Braking} = K_6\ W_t + K_7 \tag{C.5}$$

The weight of the hydraulic accumulator to store this much energy is determined by the specific energy density of 3982.6 J/kg (1335 ft-lb/lb). This density is derived and described in Section 5.2.1. Evaluating the constants K_6 and K_7 for Vehicle C and utilizing this specific energy for the hydraulic accumulator the following is obtained

$$W_{ACCUM} = 0.0396\ W_t\ 8.05 \qquad (C.6)$$

where

W_{ACCUM} = hydraulic accumulator weight (lb)

Hydraulic Motor/Pump Weight

In the final system recommended in Section 1.3, a split-path, hydraulic-augmented transmission is proposed to be used. The hydraulic part of such a transmission will have the equivalent of a hydraulic motor/pump. This motor/pump will allow the energy stored in the hydraulic accumulator to be used for acceleration, and the braking energy to be stored in the accumulator. The split-path feature transfers as much energy directly to the wheels as possible during normal driving i.e., except during acceleration and braking.

The power rating of the hydraulic system is determined by the power during the braking period, assuming constant power braking. Thus,

$$P_{HYDRAULIC} = \frac{E_{BRAKING}}{9} \qquad (C.7)$$

where

$P_{HYDRAULIC}$ = power rating of the hydraulic system

$\frac{E_{BRAKING}}{9}$ = average power during braking

and 9 in the denominator = the braking period of 9 sec specified in the STC.

The weight of the hydraulic motor/pump can then be determined by assuming a specific power density of 1.22 kg/kW (2 lb/hp).

Thus,

$$W_{HYDROM} = 2 \; P_{HYDRAULIC\ hp} \tag{C.8}$$

where W_{HYDROM} = hydraulic motor/pump weight (lb)

Now from Equations (C.5), (C.7) and (C.8)

$$W_{HYDROM} = K_8 \; W_t + K_9 \tag{C.9}$$

where K_8, K_9 are constants.

When constants K_8 and K_9 are evaluated for Vehicle C, then

$$W_{HYDROM} = 0.0252 \; W_t - 5.11 \tag{C.10}$$

<u>Electric Motor Weight</u>

Electric motor power rating is the smallest of the values determined from two different considerations as follows.

1) Motor power is matched to deliver maximum power available from the battery.

   $Power_1$ = 0.88 x 0.9 x Battery Peak Power

   where

   0.9 is the inverter efficiency

   0.88 is the motor efficiency.

   Assuming a weight of 2.73 kg/kW (6 lb/kW) for the electric motor,

   $$W_{MOTOR1} = (6 \times 0.88 \times 0.9) \text{ (Battery Peak Power)} \tag{C.11}$$

   where

   W_{MOTOR1} = weight of first motor in (lb)

   Battery Peak Power = power in (kW)

2) Motor power is adequate to deliver braking power in the regenerative mode.

Hence

$$\text{Power 2} = 0.97 \times P_{BRAKING}$$

where

0.97 is the efficiency of the differential

Now, as in Equation (C.7)

$$P_{BRAKING} = \frac{E_{BRAKING}}{9}$$

Hence

$$\text{Power 2} = 0.97 \frac{E_{BRAKING}}{9}$$

Hence utilizing Equation (C.5)

$$W_{MOTOR2} = K_{10} W_t + K_{11}$$

Evaluationg K_{10} and K_{11} for Vehicle C

$$W_{MOTOR2} = [6.0(0.0242 \times \frac{3.6}{9.0})0.97] W_t [6.0(4.91 \times \frac{3.6}{9.0})0.97] \qquad (C.12)$$

where W_{MOTOR2} and W_t are in lb.

APPENDIX D

SPECIFIC CONSUMPTIONS REQUIRED IN THE COMPUTATION OF ANNUALIZED CONSUMPTIONS

ELECTRIC ENERGY DEPLETION FROM THE BATTERY DURING STC

The total electric energy depletion is the sum of the energy depletion during three stages:

- STC acceleration
- STC cruise
- STC braking (a negative quantity)

The algorithms for computing each of these stages are given in the following subsections.

During STC Acceleration

According to the operating strategy defined in Section 5.4, the hydraulic accumulator is applied first. As constant power acceleration is assumed, the power available from the hydraulic system at the wheels is simply the energy stored in the accumulator divided by the time for acceleration - allowances are made for the efficiencies of the series components involved. (The hydraulic system power is zero when the case without the hydraulic accumulator is applied.)

After the hydraulic system power is subtracted from the total power required for constant power acceleration, the remaining power has to be supplied by the electric system. The rate of depletion of energy from the battery then is this remaining power required for STC acceleration (reflected at the battery terminals).

Thus, let

STCAWP = power required at wheels during STC acceleration (constant power acceleration assumed)

HYDRWP = power available at wheels from the hydraulic system during STC acceleration

E_{STORED} = energy stored in the hydraulic accumulator at the beginning of STC acceleration

T_a = acceleration period

η_H = efficiency of the hydraulic power system

x_1 = rate of energy depletion from the battery during STC acceleration

Then

$$HYDRWP = \frac{E_{STORED}}{T_a} \times \eta_H$$

$$x_1 = (STCAWP - HYDRWP)/(\eta_D \eta_M \eta_C)$$

where

η_D = efficiency of the differential (assumed 0.97)

η_M = efficiency of the motor (assumed 0.88)

η_C = efficiency of the power conditioning unit (assumed 0.9)

x_1 may be greater than the peak power available from the battery as determined by the battery weight and its peak power density. In that case, the rate of depletion of the battery energy is equal to the peak power capability of the battery.

As constant power acceleration is assumed, the battery energy depletion during STC acceleration is given by

BESTCA = BPSTCA x 14

where

BPSTCA = rate of energy depletion at battery terminals during STC acceleration

14 seconds = the duration of acceleration.

<u>During STC Cruise</u>

As the hydraulic accumulator is now completely depleted, the power required at wheels for STC cruise is provided entirely by the electric system. The rate of energy depletion of the battery energy is given by:

$$x_2 = STCCP/(\eta_D \eta_M \eta_C)$$

where STCCP is the power required at wheels to propel the vehicle at 72 km/h (45 mph) (STC cruise speed).

As stated earlier, the possibility exists that

$$x_2 > B_{PAVLB}$$

where

$$B_{PAVLB} = W_{BATT} \times P_{PB}$$

and

W_{BATT} = battery weight

P_{PB} = battery power density

In this instance, the battery energy depletion rate is B_{PAVLB} itself.

The energy depletion of the battery during the STC cruise period is then given by:

$$BESTCC = BPSTCC \times 40$$

where

BPSTCC = power at battery terminals (lesser of B_{PAVLB} and x_2)

40 seconds = the duration of STC cruise

The depletion of energy from the battery per mile of driving can be calculated using the same procedure as used for STC cruising at 72 km/h (45 mph). The only difference is that the duration for driving one mile at 90 km/h (56 mph) is (3600/56) seconds.

During STC Braking

If a hydraulic system is provided, the accumulator is devoid of any energy at the beginning of the STC braking period. The accumulator is designed to store all the available energy from braking. Thus, neither depletion of, nor addition to, the battery energy will occur.

When the hydraulic accumulator is not employed, some of the braking energy can be returned to the battery. In this mode, the rate of addition to battery energy is:

$$x_3 = \text{BRAKEP}\ (\eta_D \eta_G \eta_C)$$

where

BRAKEP = the breaking power available at wheels during STC braking

η_G = electric motor efficiency when operating as a generator (assumed 0.88)

Other efficiencies have been defined earlier.

As in earlier cases, the possibility exists that, due to inadequate batteries,

$$x_3 > \text{BPAVLB}$$

In that case, the rate of charging the battery will be limited to BPAVLB. Total energy addition during regenerative braking is therefore given by

$$\text{BEBR} = \text{BPBR} \times 9 \times 0.8$$

where

BPBR = the rate of energy addition to the battery (smaller of x_3 and BPAVLB)

9 seconds = the duration of braking

0.8 = the charging efficiency of the batteries (assumed)

Hence, total energy depletion from the battery during one STC is given by:

$$\text{BESTC} = \text{BESTCA} + \text{BESTCC} - \text{BEBR} \quad .$$

GASOLINE CONSUMPTION DURING STC

The gasoline consumption during one STC is made up of two components:

- During STC acceleration
- During STC cruising

During STC Acceleration

When combined, the power provided by the hydraulic and electric systems at the wheels may be adequate for STC accleration. During this period, the engine is not turned on and as a result no gasoline comsumption occurs. When the combined power is not adequate, the engine is turned on to provide the balance power. The gasoline consumption for such a case is computed as follows:

$$EPSTCA = \frac{STCAWP - (HYDRWP + WPSTCA)}{(\eta_D \eta_T)}$$

where

EPSTCA = power at the engine output shaft during STC acceleration

STCAWP = power required at wheels for STC acceleration

HYDRWP = power provided by the hydraulic system at wheels during STC acceleration

WPSTCA = power provided by the electric system at wheels during STC acceleration

η_D = efficiency of differential (assumed 0.97)

η_T = efficiency of the transmission (assumed 0.95)

Then, the energy consumption from gasoline during STC acceleration is given by:

$$EESTCA = (EPSTCA \times 14)/(\eta_E \ \eta_{COMB})$$

where

η_E = engine efficiency (assumed 0.4)

η_{COMB} = combustor efficiency (assumed 0.9)

and the volume of gasoline consumption during STC acceleration is given by:

$$GSTCA = EESTCA \ / \ H$$

where H is the heat value of a gallon of gasoline (assumed 19029.5 Btu/lb, 5.89 lb/gallon).

Another case that must be considered involves the batteries when they are below 80% Depth of Discharge and hence not available for providing power. Then,

$$WPSTCA = 0$$

and the rest of the algorithm remains the same as above.

During STC Cruise

If the power provided by the electric system is not adequate to propel the vehicle at the STC cruise speed of 72 km/h (45 mph), the engine is turned on to make up the balance power. (If the engine has to be turned on during STC cruise, it will invariably have to be turned on during STC acceleration. Thus, the engine will be on almost all the time during STC except during standstill, coasting and braking periods.) The power at the output shaft of the engine during STC cruise is given by:

$$EPSTCC = (STCCWP - WPSTCC)/(\eta_T \ \eta_D)$$

where

STCCWP = power required at wheels during STC cruise
WPSTCC = power provided by electric system at wheels during STC cruise.

The energy consumption from gasoline during STC cruise is given by:

$$EESTCC = (EPSTCC \times 40)/(\eta_E \ \eta_{COMB})$$

where

40 seconds = the cruise period in STC

The gasoline consumption during STC cruise is similarly

$$GSTCC = EESTCC/H$$

When the batteries are below 80% DOD, the gasoline consumption during STC cruise is computed similarly by setting

$$WPSTCC = 0 \quad .$$

ELECTRIC ENERGY DEPLETION AT 90 km/h (56 mph) cruise

If the battery size is large enough, the power available from the battery may be adequate to propel the vehicle at 90 km/h (56 mph) cruise. If the battery size is not large enough, the operating strategy defined in Section 5.4 requires that whenever batteries are above 80% DOD, electric power should be utilized to its full extent. Thus, under such circumstances, the engine will be turned on to provide the balance of power.

FUEL CONSUMPTION PER MILE AT 90 km/h (56 mph)

Fuel consumption per mile at 90 km/h (56 mph) is computed by a similar procedure as used in computing the fuel consumption at 72 km/h (45 mph) with the exception that the time duration for driving one mile at 90 km/h (56 mph) is (3600/56) seconds.

APPENDIX E

DETERMINATION OF ANNUAL CONSUMPTIONS

This appendix outlines the procedure for computing annual consumptions of gasoline and electricity, and the procedure for computing the number of annual deep discharge cycles (to 80% Depth of Discharge). These quantities are required in the life-cycle cost calculations and are figures of merits in the evaluation of any given design.

The annual consumption is derived by summing the daily consumptions over the year. The daily consumptions depend on the trip length and type of driving. For the purposes of computing annual consumptions of gasoline and electricity in this study, the annual driving is specified in terms of two specifications:

1. The Special Test Cycle (STC) shown in Figure 2-1, page 11
2. Continuous driving at 90 km/hr (56 mph)

Table 2-2 on page 13 gives the details of the annual trip distribution statistics used in this study.

The trip distribution is divided into two cases:

- Trips less than or equal to 50 km (31.1 miles) a day, assuming the STC
- Trips more than or equal to 80 km (49.7 miles) a day. The total mileage for each day is made up as follows.
  - First 10% of the distance is driven over the STC
  - 90% of the distance is driven at a continuous speed of 90 km/h (56 mph).

The operating strategy of the vehicle requires that the various energy sources on board the vehicle be used in the following order.

1. Hydraulic accumulator until depleted
2. Batteries up to given power level or until depleted to a level of 80% discharge
3. Engine to make up the needs of power/energy (engine is turned off when not needed for providing power).

Using this strategy the following quantities can be calculated:

- For one STC
  - When battery is available:

    BESTC - battery energy depletion

    GSTC - gallons of gasoline spent

    STCRNG - number of miles comprising only the STC's that can be travelled before battery is depleted to 80% depth of discharge using BESTC energy from the battery

  - When battery is not available:

    ENTSTG - gallons of gasoline spent

- For cruising at 90 km/hr (56 mph)
  - When battery is available:

    BATE56 - battery energy depletion per mile
    GPM56E - gallons of gasoline consumption per mile

  - When battery is not available:

    GPM56 - gallons of gasoline consumption per mile with engine alone.

The procedure for computing these quantities is given in Appendix D.

Figures E-1 through E-4 show the logic diagram for calculating daily consumptions of battery energy and gasoline for each row of Table 2-2. The procedure is written to faithfully follow all the implications presented by the operating strategy cited above. The total annual consumption is then calculated from Table 2-2 as follows:

Let

E_i = total daily battery energy consumption for the i_{th} row in Table 2-2

G_i = total daily gasoline consumption for the i_{th} row in Table 2-2

D_i number of days in i_{th} row of Table 2-2.

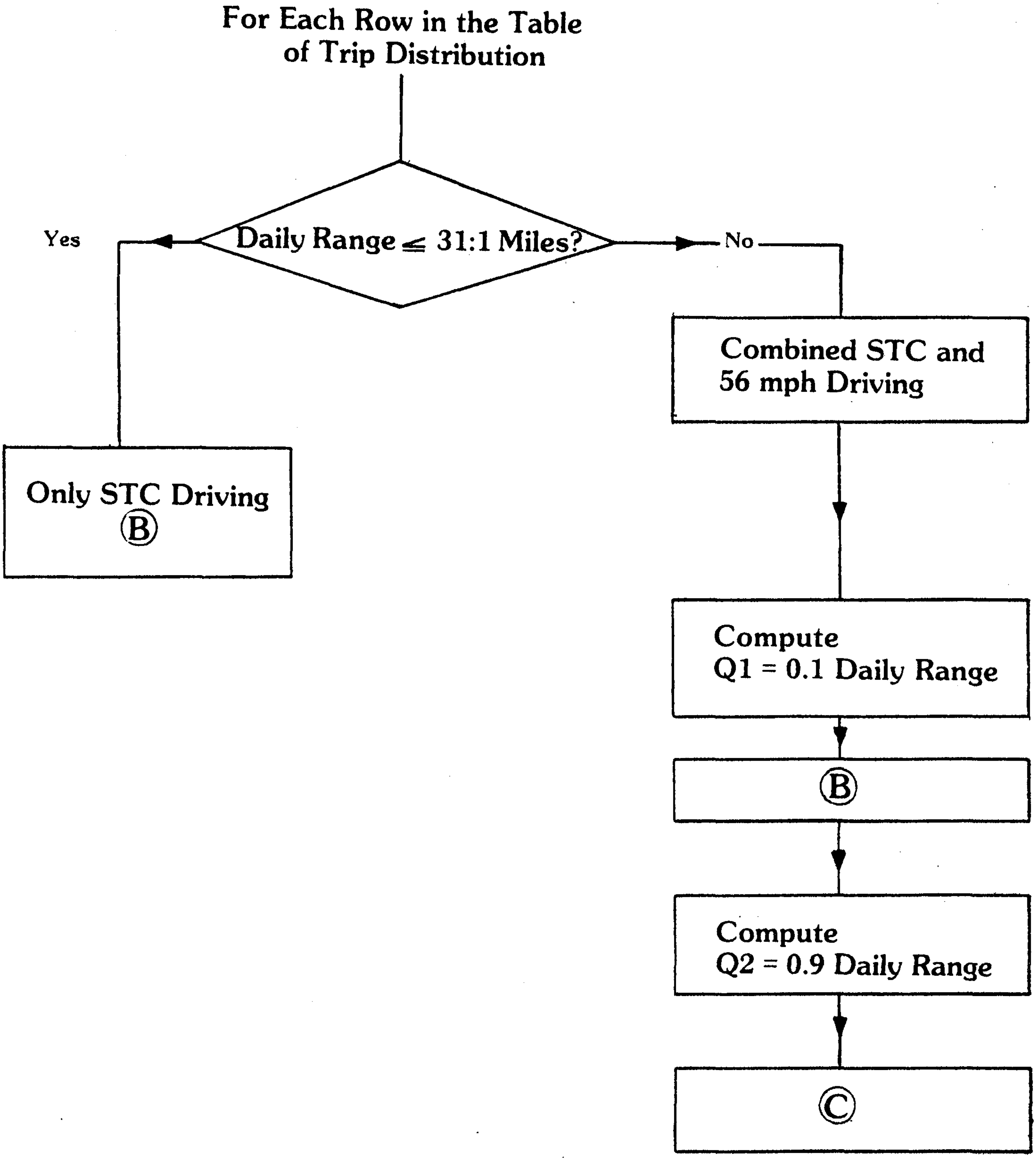

Fig. E-1 Logic Diagram A

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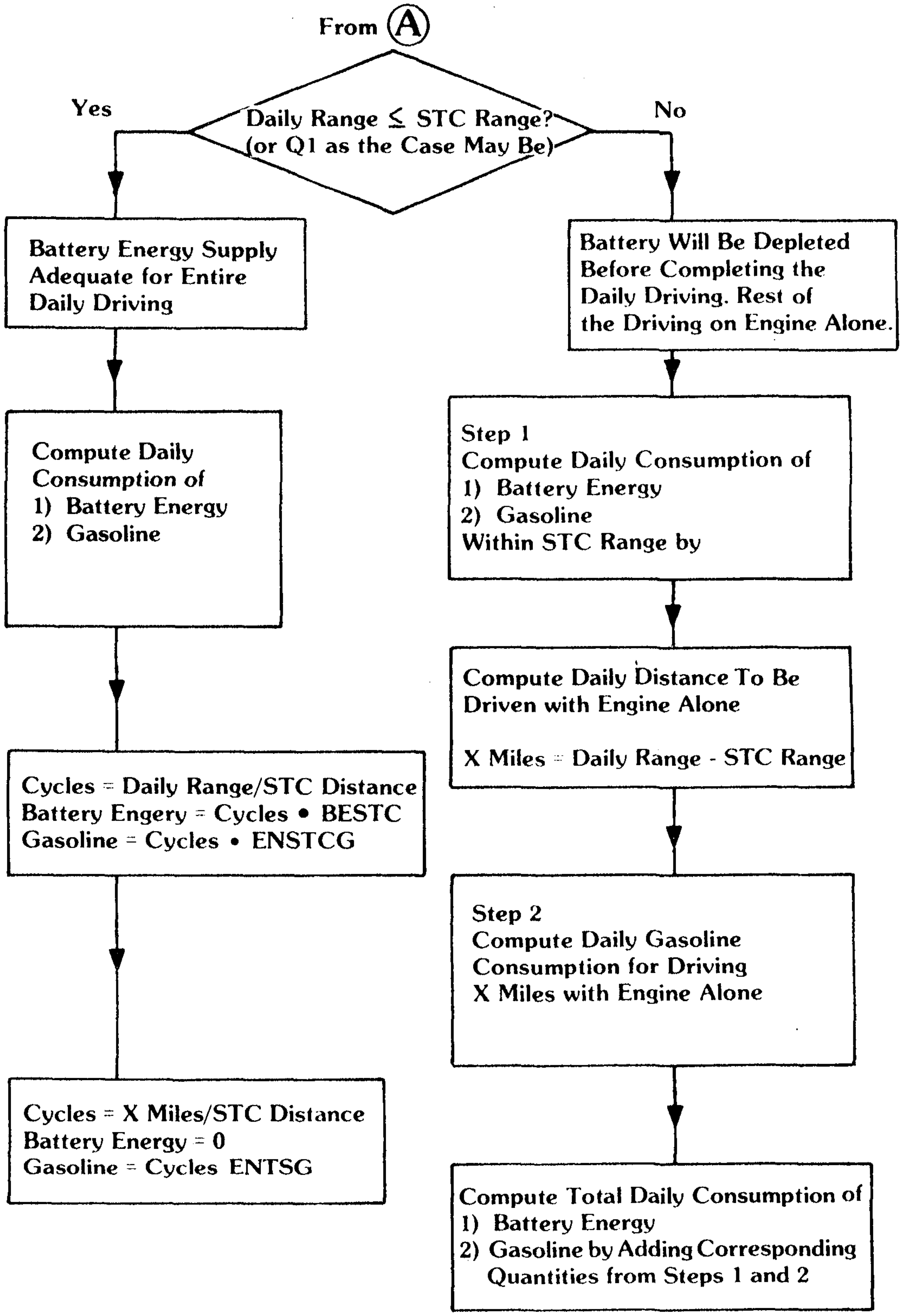


Fig. E-2 Logic Diagram B

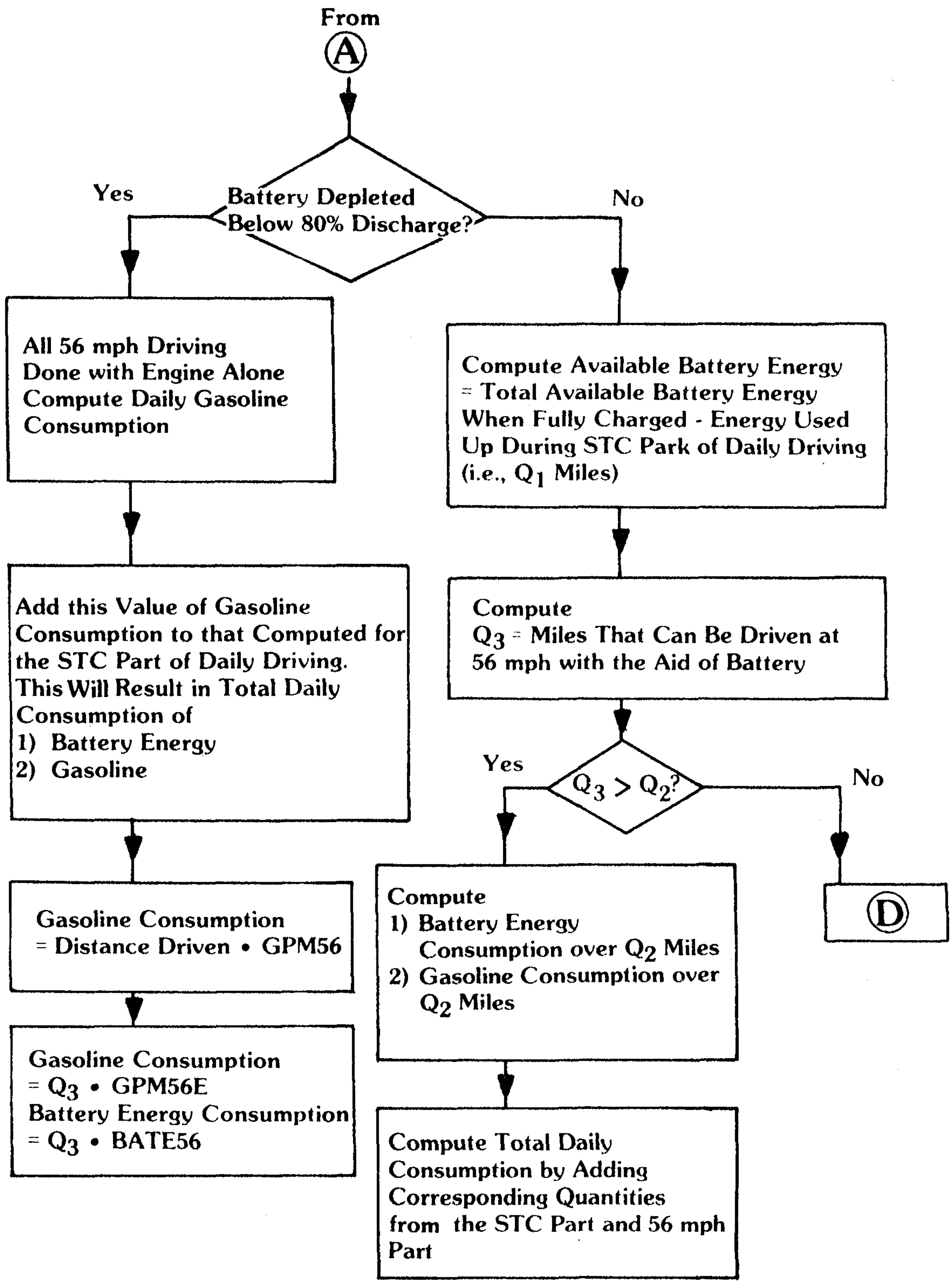


Fig. E-3 Logic Diagram C

793800-2

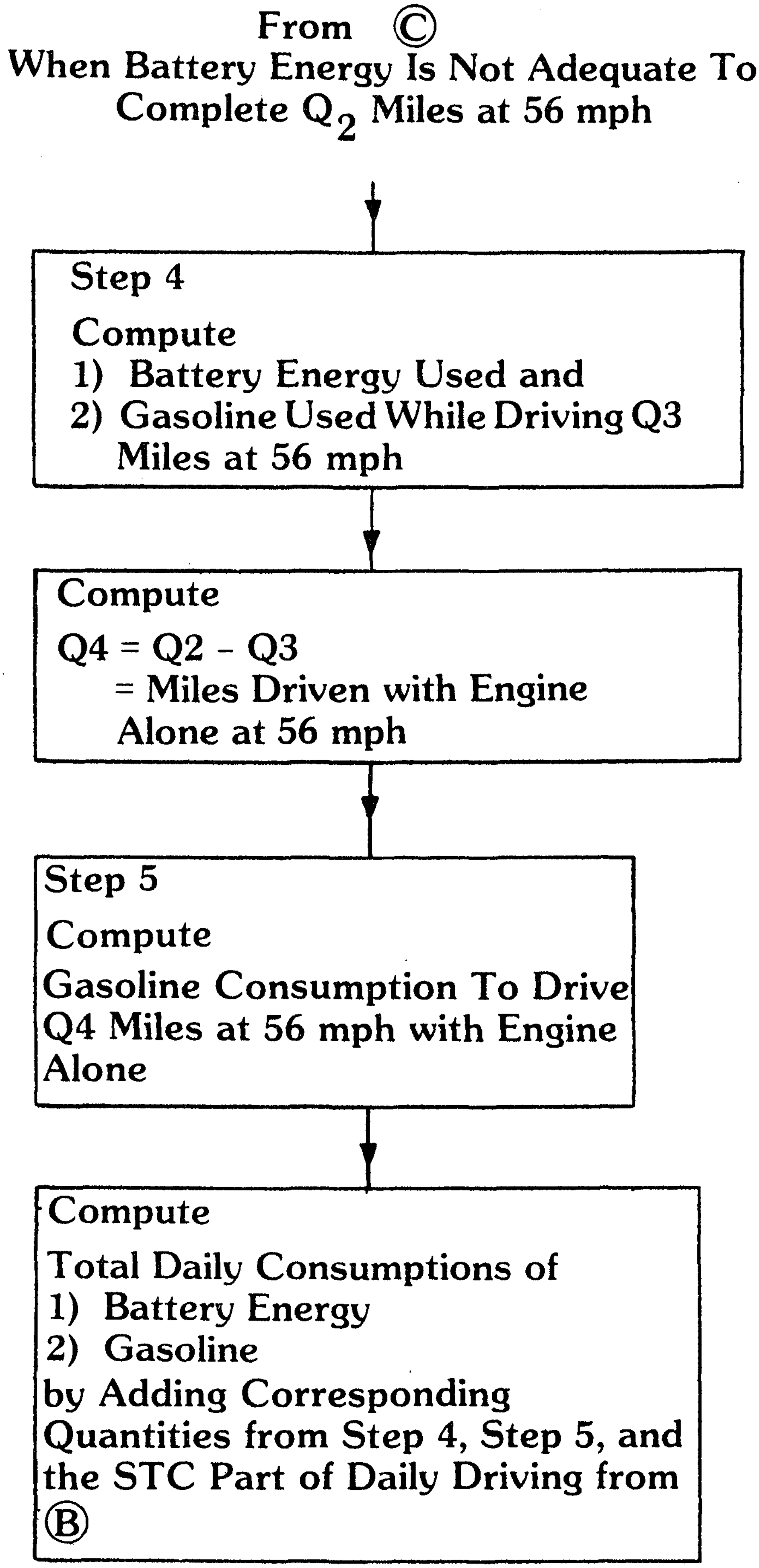


Fig. E-4 Logic Diagram D

Hence

$$\text{Annual gasolone consumption} = \sum_{i=1}^{9} G_i\ D_i$$

$$\text{Annual battery energy consumption} = \sum_{i=1}^{9} E_i\ D_i$$

$$\text{Annual wall plug electricity consumption} = \frac{\text{Annual battery energy consumption}}{0.9 \times 0.85}$$

where

0.9 = battery charger efficiency

$$0.85 = \text{battery charging efficiency} = \frac{\text{useable energy stored}}{\text{output of battery charger}}$$

Calculation of Battery Life

Battery life is specified by the number of deep discharge cycles. Due to the nature of the trip distribution shown in Table 2-2 and the battery weight, the battery may be only partially discharged on certain days. No data are available to show the effect of variable partial discharge on total life cycles. Therefore, an assumption is made that a partial discharge of x% will be considered as x/100 cycle. Thus, for each row in Table 2-2 the extent of discharge is computed using the results of the calculation shown in Figures E-1 through E-4.

Let

extent of discharge for each row = $x_{depl\ i}$ for i_{th} row in Table 2-2.

then

$$\text{total number of cycles/year} = \sum_{i=1}^{9} x_{depl\ i}\ D_i$$

The battery life is then computed as:

BATLFE = (800/cycles) years for lead-acid battery

BATLFE = (500/cycles) years for Ni-Zn battery

The numbers 800 and 500 in the above expressions are specified by NASA. See Table 6-3, page 106.

APPENDIX F

COMPUTATION OF LIFE-CYCLE COSTS

The life-cycle costs are computed following the procedure specified in Contract DEN3-92, Exhibit A, Table III. Some of the factors with the greatest impact on the life-cycle cost are repeated below:

- Total vehicle life: 10 years or 160,000 km (100,000 miles)
- Annual mileage: 16,000 km (10,000 miles)
- Discount rate: 2%
- Chassis salvage value: 2% of the purchase price
- Battery salvage value: 10% of the purchase price if fully depleted and 50% of purchase price prorated over remaining life or 10% of purchase price, whichever is greater, for non-depleted battery.

The costs are evaluated only for the components of the propulsion power train and for the utilization of gasoline and electricity. To obtain the present value of total costs, the procedure addresses first costs and annual operational costs in two categories. The following subsections provide the data used in computing various components of the total costs.

First Costs

The acquisition costs for various components, based on data in Reference [4], are as follows:

$$\text{COSTSE} = \$373.3\ (\text{ENGINP}^{0.22})$$
$$\text{COSTMT} = \$\ 13.41\ \text{EMTPWR}$$
$$\text{COSTCT} = \$\ 31.91\ (\text{EMTPWR}^{0.67})$$
$$\text{COSTTR} = \$\ 1.34\ \text{TRPWR}$$
$$\text{COSTDR} = \$\ 3.59\ (\text{TRPWR}^{0.8})$$

where

COSTSE = Stirling engine cost
COSTMT = electric motor cost
COSTCT = electric motor controller cost
COSTTR = transmission cost
COSTDR = drive system cost (differential, propeller shaft, etc.)

ENGINP = engine peak power (kW)

EMTPWR = electric motor peak power (kW)

TRPWR = transmission power rating (kW)

The cost of the hydraulic system is taken to be \$1.76/kg (\$0.80/lb). The cost of the battery, based on data specified by NASA and given in Table 6-3, page 106. is \$50/kW-h for the ISOA lead-acid battery and \$75/kW-h for the Ni-Zn battery. The total cost of propulsion system is given by:

COSTPL = \$1.17 (COSTSE + COSTMT + COSTCT + COSTBT + COSTHY + COSTTR + COSTDR)

where

COSTBT = battery cost

COSTHY = hydraulic system cost

The previous equation considers a 17% markup for the costs of the assembly and dealer.

Operating Costs

Total annual operating cost = annual maintenance costs
+ annual repair costs
+ annual gasoline costs
+ annual wall-plug electricity costs.

The following subsections give the details and associated data for computing each of these separate costs.

Maintenance Costs

Annual maintenance costs are computed from the cost per mile for various components and from the number of miles driven each year. The annual mileage is specified in the trip distribution statistics shown in Table 2-2 on page 13. The maintenance costs per mile for various power train components are shown in Table F-1. This table is extracted from the data given in Reference [4] (Table 4-1, page 4-3). The maintenance and repair costs per mile for the hydraulic system components are assumed to be 1/10 of the total engine maintenance and repair costs.

TABLE F-1

ESTIMATES OF DISTANCE-RELATED COSTS OF VARIOUS POWER TRAIN COMPONENTS

Cost Component	Maintenance Cost per mile (cents/mile) in 1976 Dollars	Repair* Cost per mile (cents/mile) in 1976 Dollars
Engine	$0.18 + 5 \times 10^{-3}$ hp	$0.28 + 8 \times 10^{-3}$ hp
Electric Motor	$0.06 + 2 \times 10^{-3}$ Php	$0.09 + 2 \times 10^{-3}$ Php
Batteries	$4.0 \times 10^{-4}\ W_{BATT}$	
Transmission		$0.05 + 1.3 \times 10^{-3}$ Thp
Hydraulic System	$0.018 + 5 \times 10^{-4}$ Hhp	$0.028 + 8 \times 10^{-4}$ hp

where

hp = engine horsepower

Php = peak horsepower of the electric motor

Hhp = peak horsepower of the hydraulic system

Thp = total horsepower through the transmission

---

*Repair mileage factor is not included.

The cited reference table does not give different maintenance costs for different engine types. An assumption could be made that the engine maintenance costs given in that table refer to internal combustion engines. The routine maintenance costs for the Stirling engine are expected to be smaller than for internal combustion engines. Furthermore, in the hybrid mode, the Stirling engine will be used less frequently over a given distance travelled annually, resulting in a further reduction of the engine component's maintenance costs. However, no allowance was made in this study to reflect these reductions, as no authenticated data base was available to quantify the extent of these reductions at the time the study was performed. Thus, annual maintenance cost is given by:

$$MCOST = (ENGINM + EMTRM + BATTM + HYDRM)m_i$$

where

m_i = number of miles driven in year i (specified by the trip distribution statistics of Table 2-2 to be 16,000 km [10,000 miles] for all the years of operation)

ENGINM* = engine maintenance cost per mile

ENTRM = electric motor maintenance cost per mile

BATTM = batteries' maintenance cost per mile

HYDRM = hydrualic system maintenance cost per mile

The maintenance costs per mile are assumed to be constant throughout the life of the vehicle.

Repair Costs

The annual repair costs are considered to vary depending on the number of miles driven and on the total number of miles driven over the lifetime of the vehicle. Generally, annual repair costs are insignificant during the first few miles when the vehicle is new, rise to a certain peak value, and then, during the last year, fall off again. This variation in repair costs during the vehicle lifetime is computed by multiplying the Repair Mileage

---

*These four quantities are derived from Table F-1.

Factor for a given year by the annual basic repair costs which are determined solely from the cost per mile and the number of miles driven annually.

The curves for determining the Repair Mileage Factor for each year of operation are given in Figure F-1 reproduced from Reference [4] (Figure 4-2). The repair costs per mile for various power train components are given in Table F-1. Thus, annual repair costs in year i of operation,

$$RCOST_i = RMF_i \times (RCOST_b)$$

where

RMF_i = Repair Mileage Factor for year i determined from Figure F-1

$RCOST_b$ = $(ENGINR + EMTRR + TRANSR + HYDRR)m_i$

where

ENGINR* = engine repair cost per mile

EMTRR = electric motor repair cost per mile

TRANSR = transmission repair cost per mile

HYDRR = hydraulic system repair cost per mile

Fuel Costs

The annual fuel cost is given by:

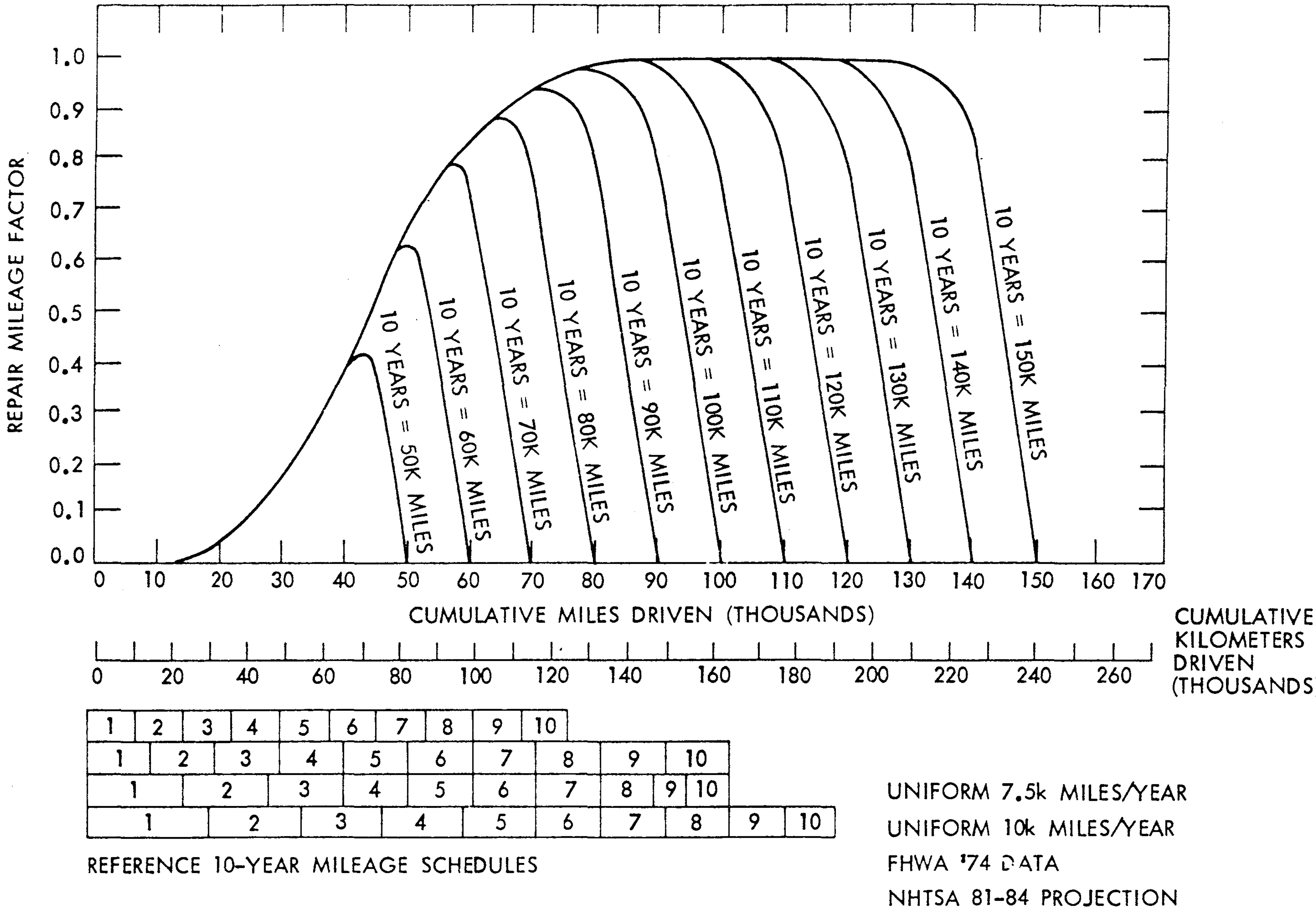
$$GASOLN = GALONS \times COSTGS$$

where

GALONS = number of gallons consumed annually (determined by the procedure outlined in Appendix E)

COSTGS = price of gasoline/gallon in 1976 dollars [in this study four different values are assumed: $0.40/liter ($1.50/gal); $0.66/liter ($2.50/gal); $0.92/liter ($3.50/gal); and $1.19/liter ($4.50/gal).

*These four quantities are derived from Table F-1.



Source: Reference [4], Figure 4-2.

Fig. F-1 Repair Mileage Factor

Electricity Cost

Annual electricity cost is given by:

ELECTR = EE3 x COSTEL

where

EE3 = kW-h consumed from wall plug every year (determined by the procedure given in Appendix E).

COSTEL = $0.06/kW-h in 1976 dollars.

Battery Replacement Cost

A procedure to compute the battery life was outlined in Appendix E, under Calculation of Battery. At each replacement, the cost of purchasing a new battery set, less the salvage value of the depleted battery, is considered to occur in the year of the replacement. This value is entered in Contract DEN3-92, Exhibit A, Worksheet 2, "Life-Cycle Cost Worksheet", Row 5 - Battery Replacement, in the column corresponding to the year of replacement.

Battery Salvage Value

In general, the total vehicle life of 10 years is not an integer multiple of battery life. Thus, the last set of batteries may have significant life left at the end of the tenth year when the vehicle is to be sold for its salvage value. At that time, the salvage value of the battery is the larger of the following two quantities:

- 50% of the purchase price prorated over the remaining life of the battery
- 10% of the purchase price.

The salvage value is then entered in the previously mentioned Worksheet 2, Row 7 - Battery Salvage, in the column corresponding to the tenth year.

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16. Abstract

This report evaluates a series hybrid system, utilizing a free-piston Stirling engine with a linear alternator, and a parallel hybrid system, incorporating a kinematic Stirling engine, for various specified reference missions/vehicles ranging from a small 2-passenger commuter vehicle to a van.

Parametric studies for each configuration, detail trade-off studies to determine engine, battery and system definition, short-term energy storage evaluation, and detail life-cycle cost studies were performed.

The selection of a parallel Stirling engine/electric, hybrid propulsion system can significantly reduce petroleum consumption by 70% over present conventional vehicles.

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