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Santa Barbara, California

A P P E N D I X   B

DESIGN TRADEOFF STUDIES

AND

SENSITIVITY ANALYSIS

PREPARED FOR:

JET PROPULSION LABORATORIES

CONTRACT NUMBER 955189

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MAY 25, 1979

HYBRID BY SCT

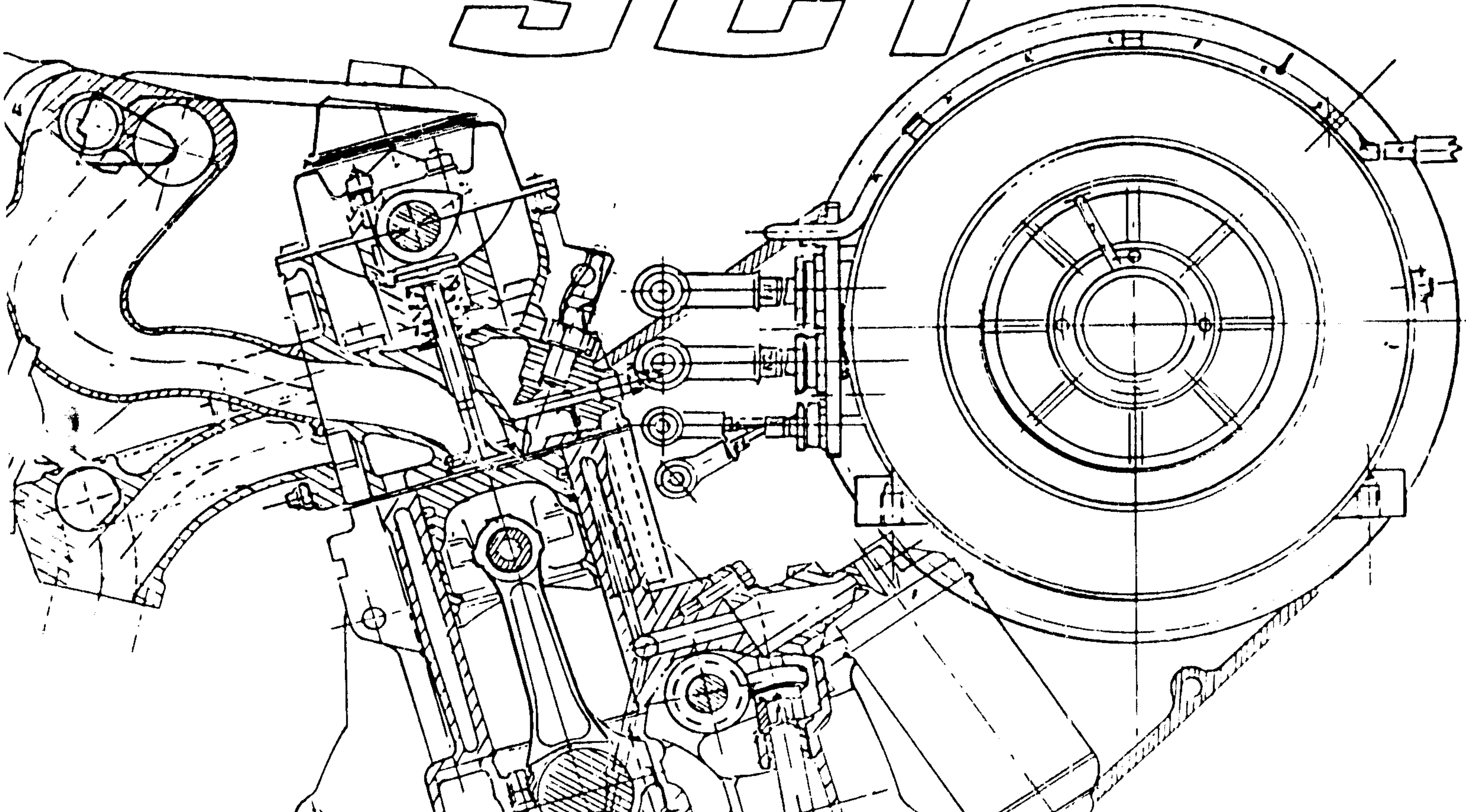

TABLE OF CONTENTS

1. INTRODUCTION

This report presents the results of Task 2 - Design Tradeoff Studies, and Task 4 - Sensitivity Analysis, of Phase I of the Near Term Hybrid Passenger Vehicle Development Program.

The work was performed by South Coast Technology, Inc., with assistance from our subcontractors and consultants, who include:

C. E. Burke Engineering Services - Propulsion system design and cost studies.

EHV Systems, Inc. - Electric propulsion systems.

The Brubaker Group - Material substitution and vehicle packaging.

Wharton EFA, Inc. - Sensitivity studies.

B. T. Andren - Automotive engineering.

S. Renick - Material substitution.

Roy Renner - Flywheels and alternate transmissions

Lonney Pauls - Structural analysis/material substitution.

Assistance was also received from Siemens (electric motors), and from battery manufacturers participating in the ANL ISOA Battery Program.

2. METHODOLOGY

2.1 General Approach

The approach used in the design tradeoff studies task is illustrated schematically in Figure 2-1. As indicated, the work was broken down into two major phases:

- System level tradeoff studies, whose objective was to optimize some basic parameters which have a major influence on cost factors and fuel consumption.
- Subsystem and component level tradeoff studies, whose objective was to develop specific information on subsystem configurations, component design and selection, material selection, vehicle layout, and so forth.

These studies have all been directly oriented to stay within the constraints as specified by JPL in Exhibit I of this Near Term Hybrid Vehicle Program. One of the key constraints deals with technology; and, as defined by JPL, "Components and fabrication techniques must be within state-of-the-art capabilities that can be developed by <u>1980</u> and must be amenable to mass production by the mid-1980's."

Our proposed hybrid vehicle concept fully meets this realistic constraint; although in so doing, we may not be offering the technological spectacle of vehicles that, although producible in prototype form, are not in any respect mass producible by the mid-1980's. In arriving at judgments as to whether a technology meets the criteria of this program, we essentially used a four step procedure:

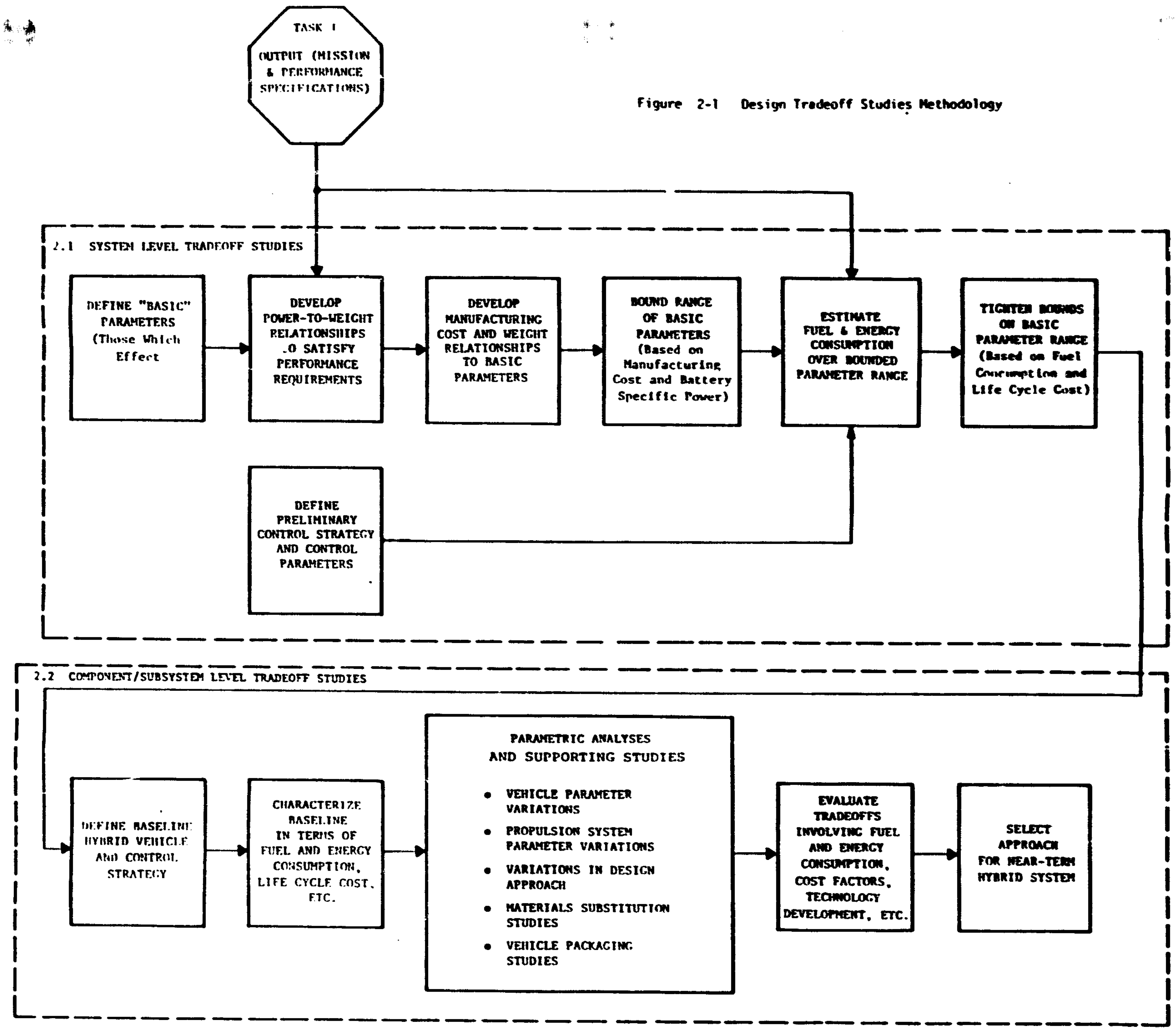

Figure 2-1 Design Tradeoff Studies Methodology

1. Members of South Coast Technology staff working on this program were asked to conceptually construct a 1985 model year hybrid vehicle that they believed represented 1980 technology and 1985 model year mass production capability, based on their experience and their analysis of literature on the subject.

2. Our key body and propulsion system consultants and subcontractors were asked to do the same. Included in our support personnel are experienced automotive planners and engineers.

3. With this input, we developed an approach that we believe represented a 1985 model year hybrid. In our Mission Analysis and Tradeoff Studies reports (Appendix A), we concluded that the most suitable size for a hybrid vehicle is a full-sized, 6-passenger sedan, represented by a Ford LTD; and, thus, our proposed 1985 model year hybrid is going to be derived from a 1985 version of the Ford LTD, which constitutes our reference vehicle. We, thus, developed a proposed hybrid version of the Ford LTD and established a plan as to which parts of the car would change from current production, where material substitutions would be made, and what materials were likely to be used.

4. Our next step was to meet with Ford Research and Engineering staff personnel. Ford has been very cooperative, but it must be clearly stated that our assumptions are not in any way endorsed by Ford. They are, however, considered

to be a realistic way to define a 1985 Ford LTD reference vehicle, and thus, the materials and approach that Ford might choose to use to build a 1985 hybrid vehicle should they opt to do so. In this rapidly changing environment, it is almost impossible to be definitive for the 1985 model year; but certain key directions are clear.

- The basic car will not undergo another major downsizing by 1985.
- Substitution of materials to achieve weight reduction will be on a partial basis. There may be more plastic and aluminum used, but there will not be a composite car or a structural aluminum body.
- For the reference vehicle, fuel economy advances will be achieved by using more fuel efficient components--PROCO engine, automatic transmission with lockup torque convertor and overdrive, etc.
- Hybrid vehicles, like electrics, are more likely to achieve widespread production by retrofitting the hybrid or electric system into an existing car as an option. Thus, we may find Ford offering a PROCO engine Ford LTD with a hybrid option, or VW offering a Rabbit gasoline engine with a diesel and an electric option. Review by some major automotive manufacturers of our Electric by SCT, based on a VW Rabbit retrofit, confirms that this approach is the current thinking of the

manufacturers as opposed to earlier approaches involving an all new car.

In summary, our general approach is realistic, meets the constraints of this Near Term Hybrid Vehicle Program, and results in transferable technology that could be useful to the auto industry and, thus, speed up the introduction of fuel efficient hybrid technology in our nation's fleet of cars. To do otherwise would only create show cars and laboratory devices, which we do not consider to be the purpose of this program.

The tradeoff studies were carried out both for the assumptions specified in the basic work statement and for the variations on these assumptions defined in the work statement for the Sensitivity Analysis Task (Task 4).

2.1.1 System Level Tradeoff Studies

Basic Parameter Definition

The first step in these studies was to define what we have called 'basic' parameters in Figure 2-1. These are the parameters which have a major influence on vehicle manufacturing cost, weight, and fuel and energy consumption. The simplest set of such parameters is the following:

1) Battery type (lead-acid, nickel-zinc, etc.).
2) Battery weight fraction, $\overline{W}_B$, defined as the ratio of battery weight, W_B, to vehicle curb weight, W_V.
3) Heat engine power fraction, $\overline{P}_{HE}$, defined as the ratio of peak heat engine power, P_{HE}, to the maximum vehicle power requirement, P_{TMAX}.

This parameter set intentionally leaves out a great deal of detail; it does not consider variations in the type of heat engine, traction motor, controller, and so forth. Essentially, we made the assumption that such variations would not affect significantly the range of 'basic' parameter values selected as containing an optimum. For example, if the characteristics of a diesel instead of a gasoline engine were used in the various vehicle system models, this would not change the conclusion that the battery weight fraction should fall within a certain narrow range, and the heat engine power fraction within another narrow range, and so forth. This assumption was necessary to permit the universe of possibilities, which would be investigated in more detail in the component/subsystem level studies, to be kept down to a manageable size.

Power-to-Mass Relationships

The next step was to determine the power-to-mass ratio required to achieve the performance requirements defined in Task 1, Mission Analysis and Performance Specifications. Because of the fact that an electric motor has a power curve which is shaped differently than that of an internal combustion engine, the required power-to-mass ratio varies somewhat as a function of the heat engine power fraction (unless a continuously variable transmission is used which keeps both power units at their peak power during a full throttle acceleration). For the sake of simplicity, the following assumptions were made:

1) The heat engine power curve has a shape typified by a four cylinder gasoline engine.

2) The electric motor power curve has a shape typified by a separately excited motor operating under armature control up to a given rpm, followed by field control (i.e., constant power).

3) The transmission characteristics are typified by a four-speed (no torque convertor).

Using these typical characteristics, a series of runs were made with the program VSPDUP, which simulates a full throttle acceleration run. These runs were made for a pure electric ($\overline{P}_{HE} = 0$) and pure IC engine ($\overline{P}_{HE} = 1$) for a range of scale factors on the engine or motor size. It was assumed the electric vehicle weighed 60% more than the IC engine vehicle, and that the frontal areas were the same in both cases. A drag coefficient of .4 and rolling resistance coefficient of .01 were used. The results were plotted, as shown in Figures 2-2 and 2-3, the critical acceleration requirement was identified, and the power-to-mass ratio required to achieve it was determined. Note that for both the pure electric and the IC engine, the critical requirement is to accelerate from 0-90 kph in 15 sec.; if the power-to-mass ratio is adequate for this, the other requirements are satisfied. Note also that separate consideration of the gradeability requirements was not made at this point since they are implied (at least on an instantaneous basis) by the acceleration requirements, as discussed in the report on Task 1 (Section 2.9.3). As indicated in Figure 2-4, if the 0-90 kph time of 15 sec. is met exactly by the electric and IC engine vehicles, the acceleration of the EV to any speed less than 90 kph is better than that of the IC engine vehicle, due to the 'fatter' power curve of the EV.

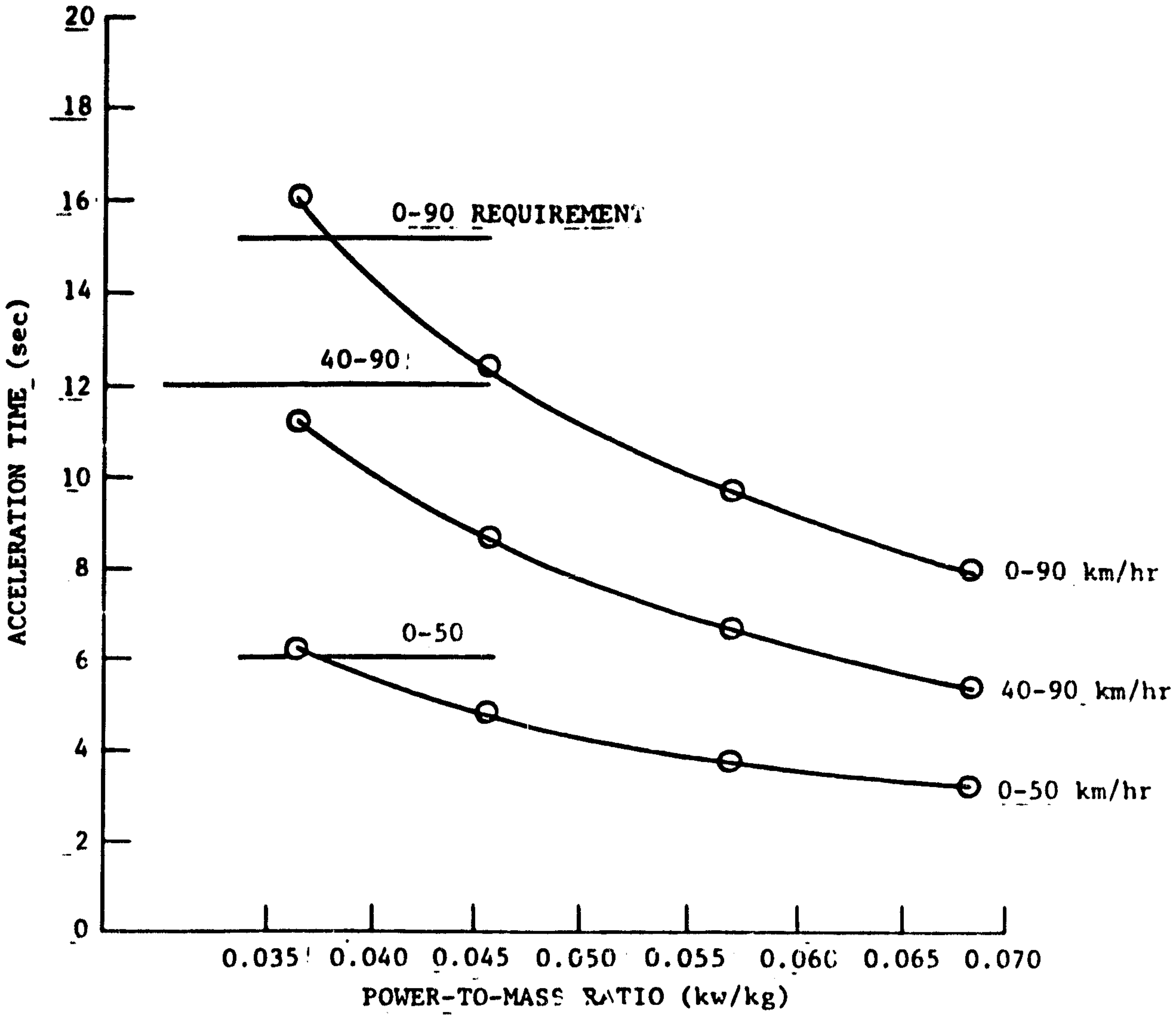

Figure 2-2 Determination of Required Power-to-Mass Ratio for IC-Engine Vehicle ($P_{HE} = 1$)

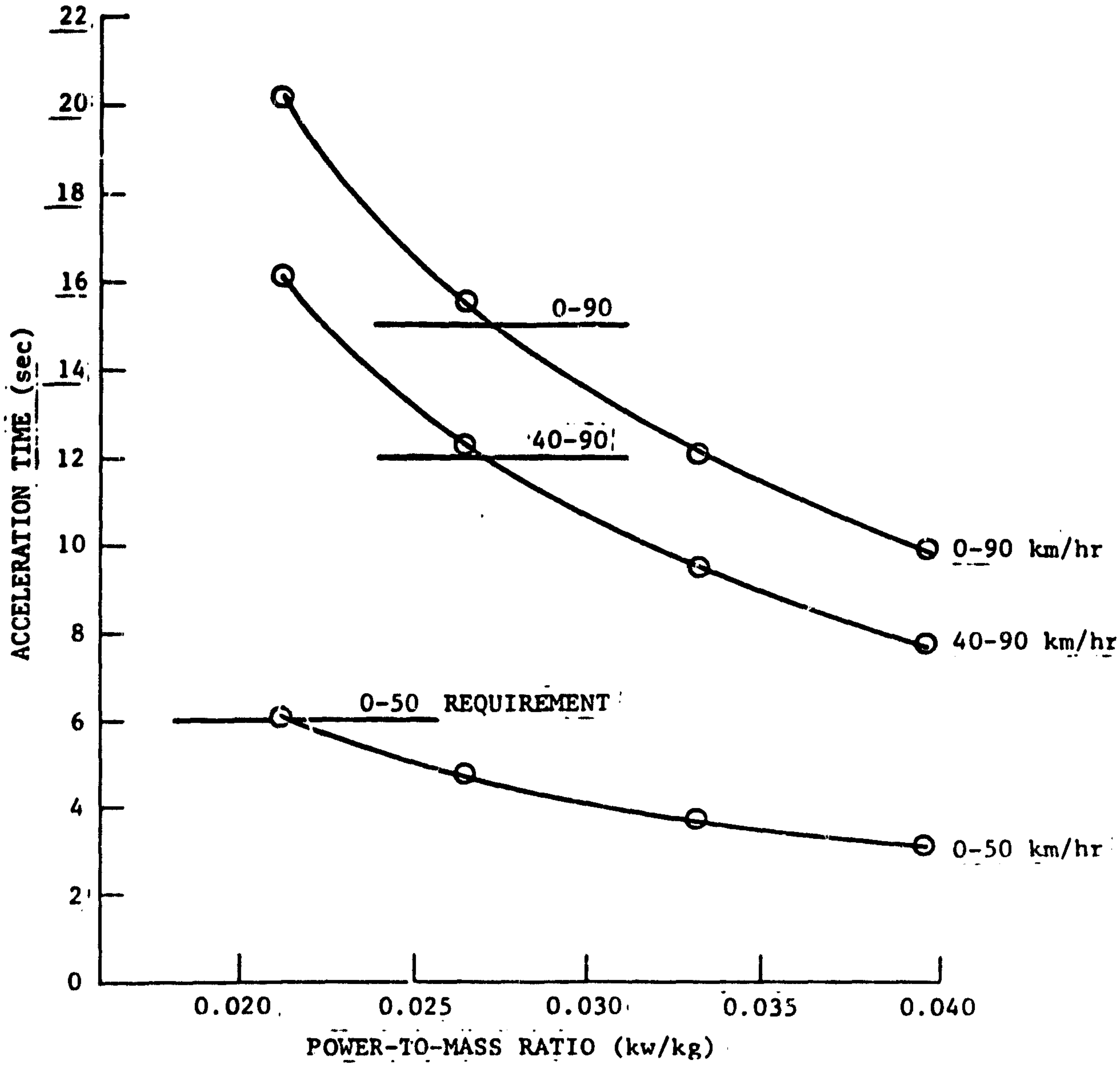

Figure 2-3 Determination of Required Power-to-Mass Ratio for EV (P_{HE} = 0)

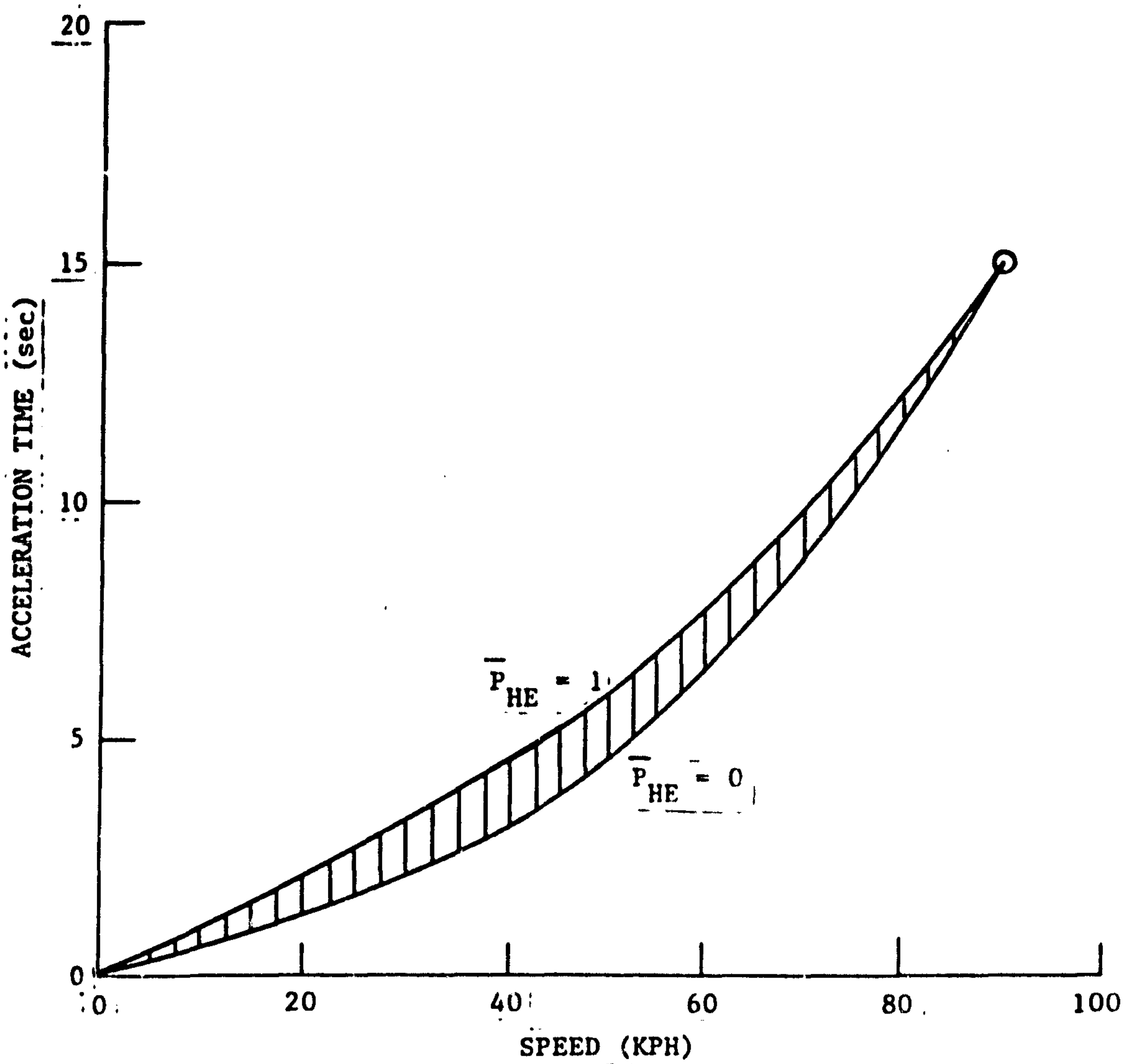

Figure 2-4 Acceleration Characteristics of EV ($\overline{P}_{HE} = 0$) and IC Engine Vehicle ($\overline{P}_{HE} = 1$)

With these results in hand, the assumption was made that the variation in required power-to-mass ratio is linear with $\overline{P}_{HE}$, between the two extreme cases $\overline{P}_{HE} = 0$ and $\overline{P}_{HE} = 1$, as shown in Figure 2-5. The assumption of linearity was made for the following reasons:

- The total variation from $\overline{P}_{HE} = 0$ to $\overline{P}_{HE} = 1$ is not great, and it is clear that the variation must be continuous and monotonic. Consequently, the possible error in this assumption must be small, and certainly acceptable given the relatively gross objectives of the system level studies.
- Linearity of this relationship (along with various mass vs. power relationships for individual components) permits the construction of a vehicle mass model in which a closed form solution of the vehicle curb mass, in terms of the heat engine power fraction and battery weight fraction, is possible. This model and the essential steps in the derivation of this closed form solution are described in Appendix A2 of the Task 1 report.[(1)]

Manufacturing Cost and Weight Relationships

A series of linear cost vs. weight and weight vs. power relationships were developed for use in the WANDC program, which is described in Appendix A2 of the Task 1 report. This program computes the overall vehicle weight, as well as the weights and power ratings of the major propulsion system components, as functions of the three basic parameters: heat engine power fraction ($\overline{P}_{HE}$), battery weight fraction ($\overline{W}_B$), and battery type. The data used in developing the linear relationships used by WANDC came from a variety of sources, as follows:

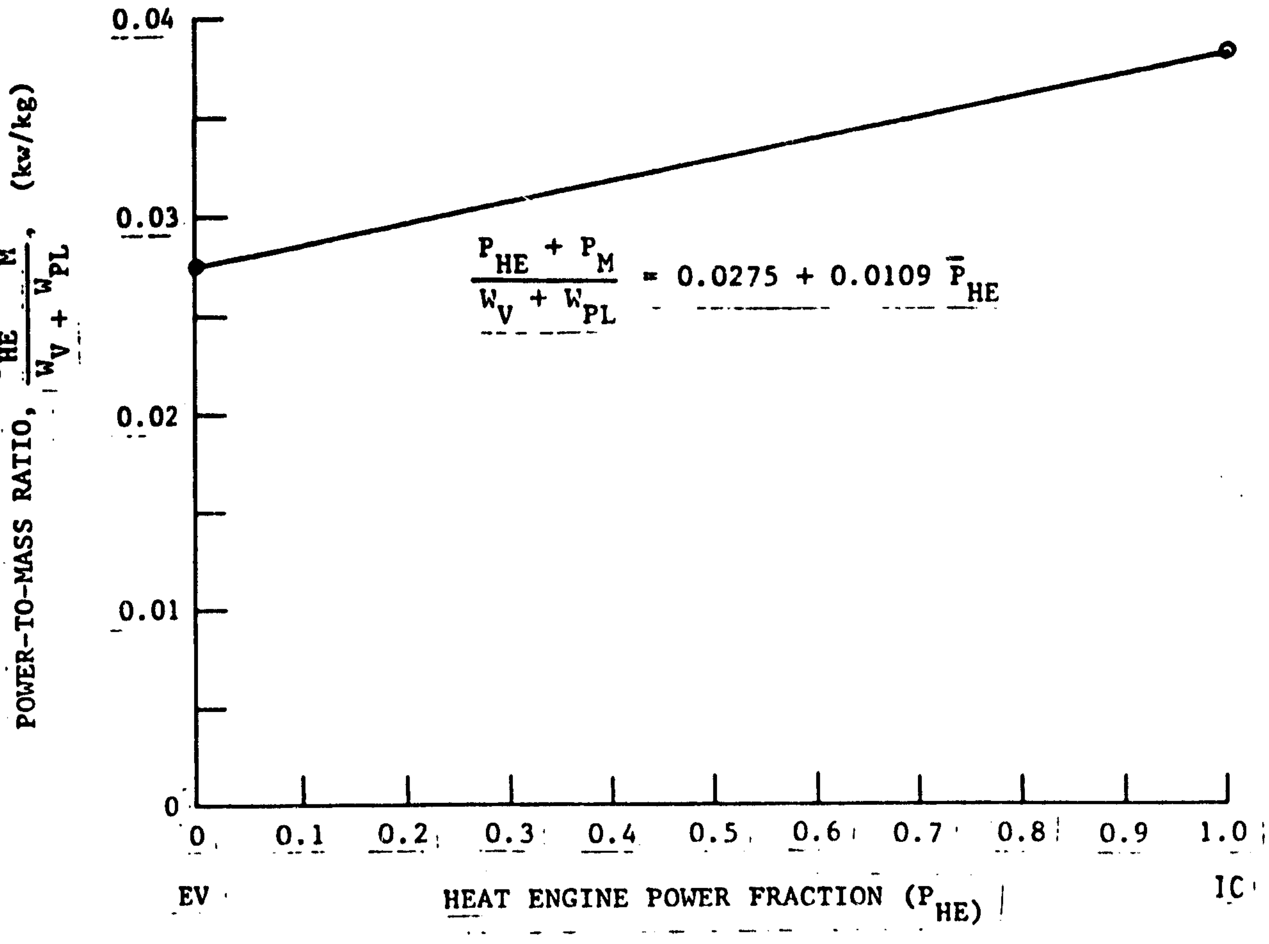

Figure 2-5 Variation of Required Power-to-Mass Ratio with Heat Engine Power Fraction

- Heat engine (weight vs. power and cost vs. power). These data came from an extensive study of weights and costs of various automotive components done by Rath and Strong, Inc.[(2)] Since this study was done, ca. 1975, the cost data were increased by 22.5% to reflect price increases from 1975 to 1978. The relationships used in the WANDC model are shown in Figure 2-6.
- Electric motor and controls (weight vs. power and cost vs. power). Weight vs. power characteristics of the electric motor were estimated based on the weights of the Bosch line of motors which are quite modern designs covering the range from 3 to 35 kw (1 hr. rating). Controls were assumed to add 30% to the motor weight. The resultant relationship is shown in Figure 2-7.

Projection of electric motor and controls prices at production quantities of 100,000 units/year is more difficult because there are no systems in the required size range being built in anywhere near those quantities. For the purposes of the system level tradeoffs, the cost numbers generated by General Electric for the Near Term Electric Vehicle Program[(3)] were used. GE projected an OEM price of $785 (1975 $) for a complete system consisting of a shunt motor (25 hp, one hour rating) and an armature/field controller using high power transistors, in high volume production. This price was adjusted to $980 in 1978 $. For other power levels, the price was assumed proportional to weight;

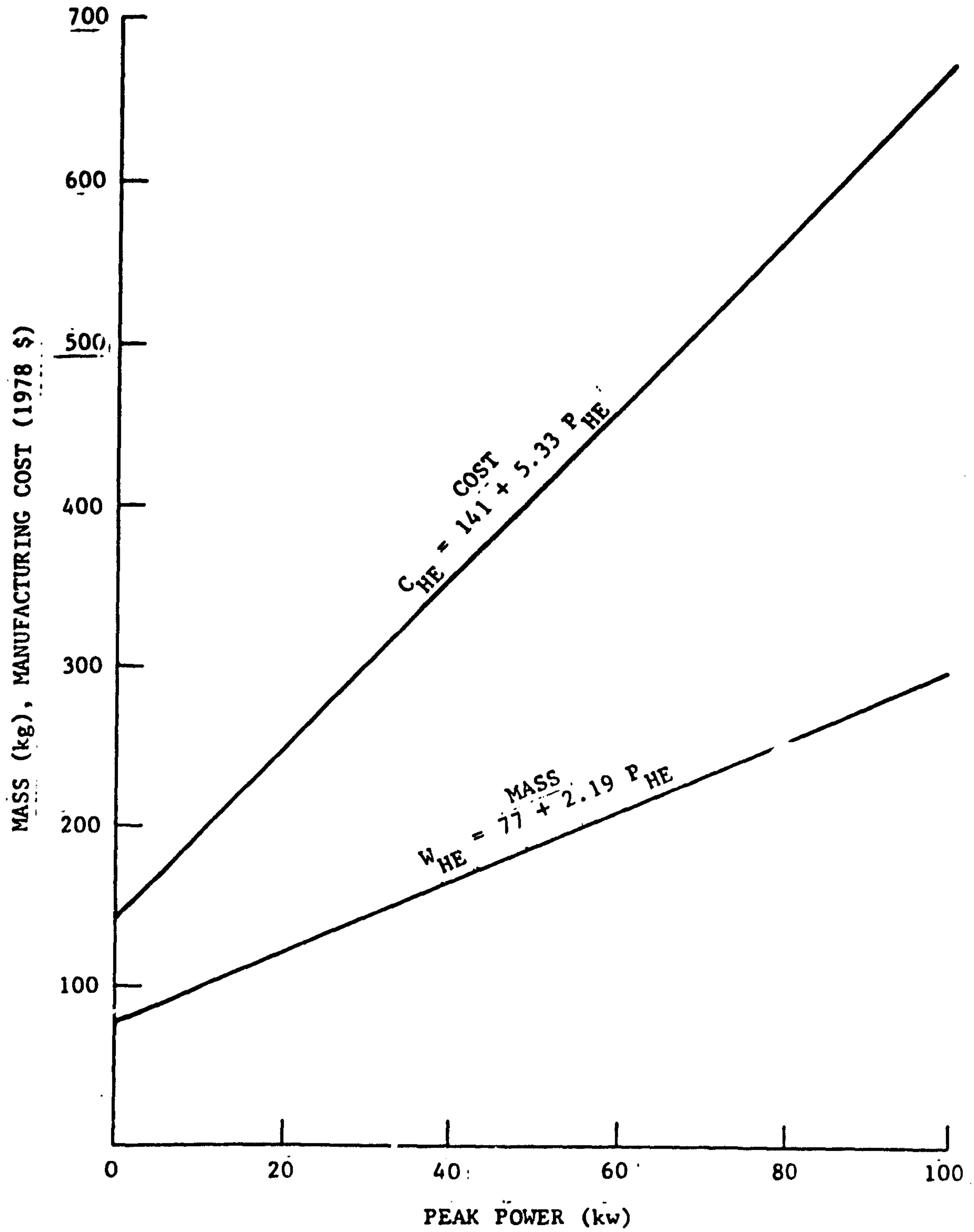

Figure 2-6 Heat Engine Weight and Cost Relationships

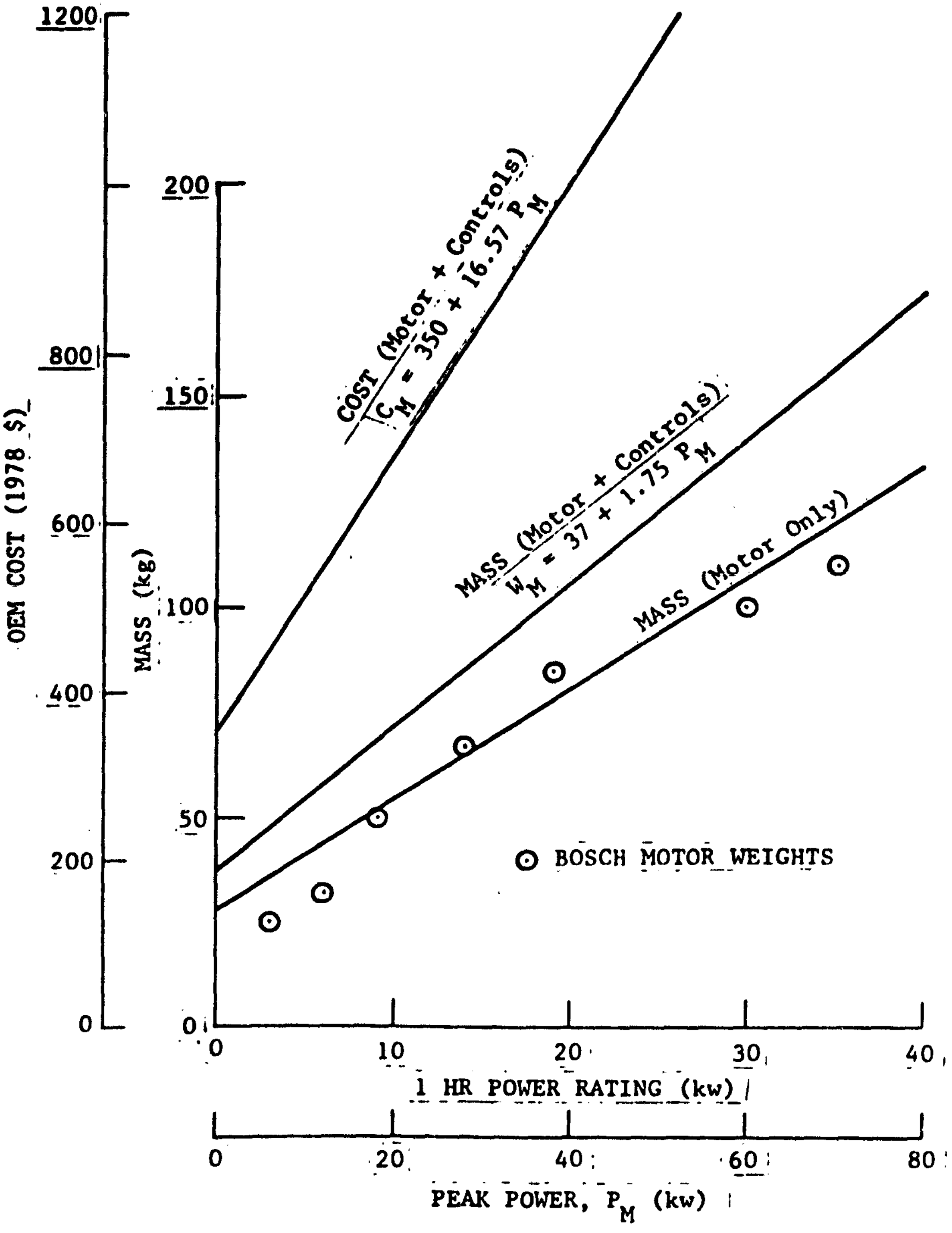

Figure 2-7 Motor/Controls Weight and Cost Relationships

i.e., the cost vs. power relationship is a multiple of the weight vs. power relationship. The cost relationship is also shown in Figure 2-7. In addition, it was assumed that the peak power output of the electric motor is limited to twice the one hour rating.

- Transmission (weight vs. power and cost vs. power). As in the case of the heat engine, the Rath and Strong data used to derive these relationships, under the assumption that whatever transmission was used would be equivalent in cost and weight to a 3-speed automatic. (See Figure 2-8)
- Battery pack (cost vs. weight). Battery costs were based on the goals of the Argonne National Laboratory for improved state-of-the-art (ISOA) batteries. These were derived as follows:

  Lead-acid batteries: \$/kg = .040 kw-hr/kg x \$50/kw-hr = \$2/kg

  Nickel-iron batteries: \$/kg = .050 kw-hr/kg x \$75/kw-hr = \$3.75/kg

  Nickel-zinc batteries: \$/kg = .070 kw-hr/kg x \$75/kw-hr = \$5.25/kg

- Vehicle carriage (cost vs. weight). The portion of the vehicle which remains after the propulsion system (heat engine, traction motor and controls, transaxle, and batteries) is removed, we call the vehicle 'carriage.' We have assumed that this portion of the mass satisfies the following relationship, involving a weight propagation factor θ: for every kilogram of mass added in propulsion system or payload, θ kilograms are added to the vehicle carriage in

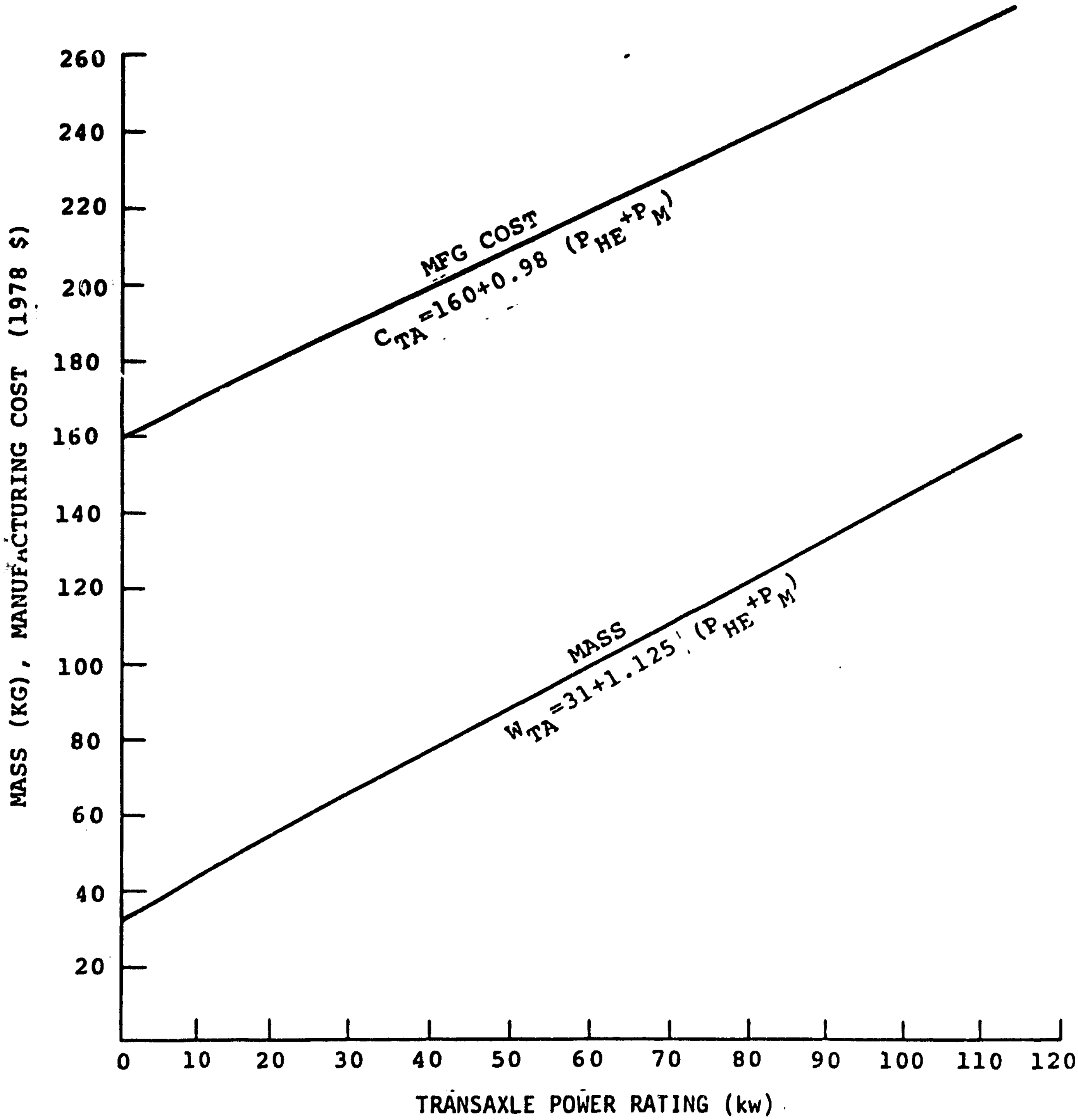

Figure 2-8 Transaxle Weight and Cost Relationships

additional structure, heavier duty suspension and brakes, and so forth. A value of .2 was used for Θ. The cost vs. weight relationship was derived from the Rath and Strong study. Two additional check points were obtained by costing a Fairmont and an LTD; this exercise was performed by C. E. Burke Engineering Services. The costs obtained on the Fairmont and LTD were consistent with the Rath and Strong data.

The cost vs. weight relationship obtained for the vehicle carriage is shown in Figure 2-9. It will be noted that the line shows a negative intercept. This is a result of the fact that the heavier the car, the more it is likely to have luxury appointments and power accessories; hence, the cost tends to rise more steeply than would occur with the assumption of a constant cost per unit weight.

Bounds on Parameter Ranges

In establishing bounds on the ranges of the basic parameters $\overline{P}_{HE}$ and $\overline{W}_B$, three factors were considered:

1) The peak battery output power would have to be limited to a reasonable value. This puts a lower bound on the range of permissible values of $\overline{W}_B$ for a given value of $\overline{P}_{HE}$. Peak battery specific power was limited to 100 w/kg for lead-acid batteries and 150 w/kg for nickel-iron and nickel-zinc batteries.

2) For the purchase price of the hybrid vehicle to be 'comparable' to that of the reference vehicle, the manufacturing cost could not be too great an increment above it.

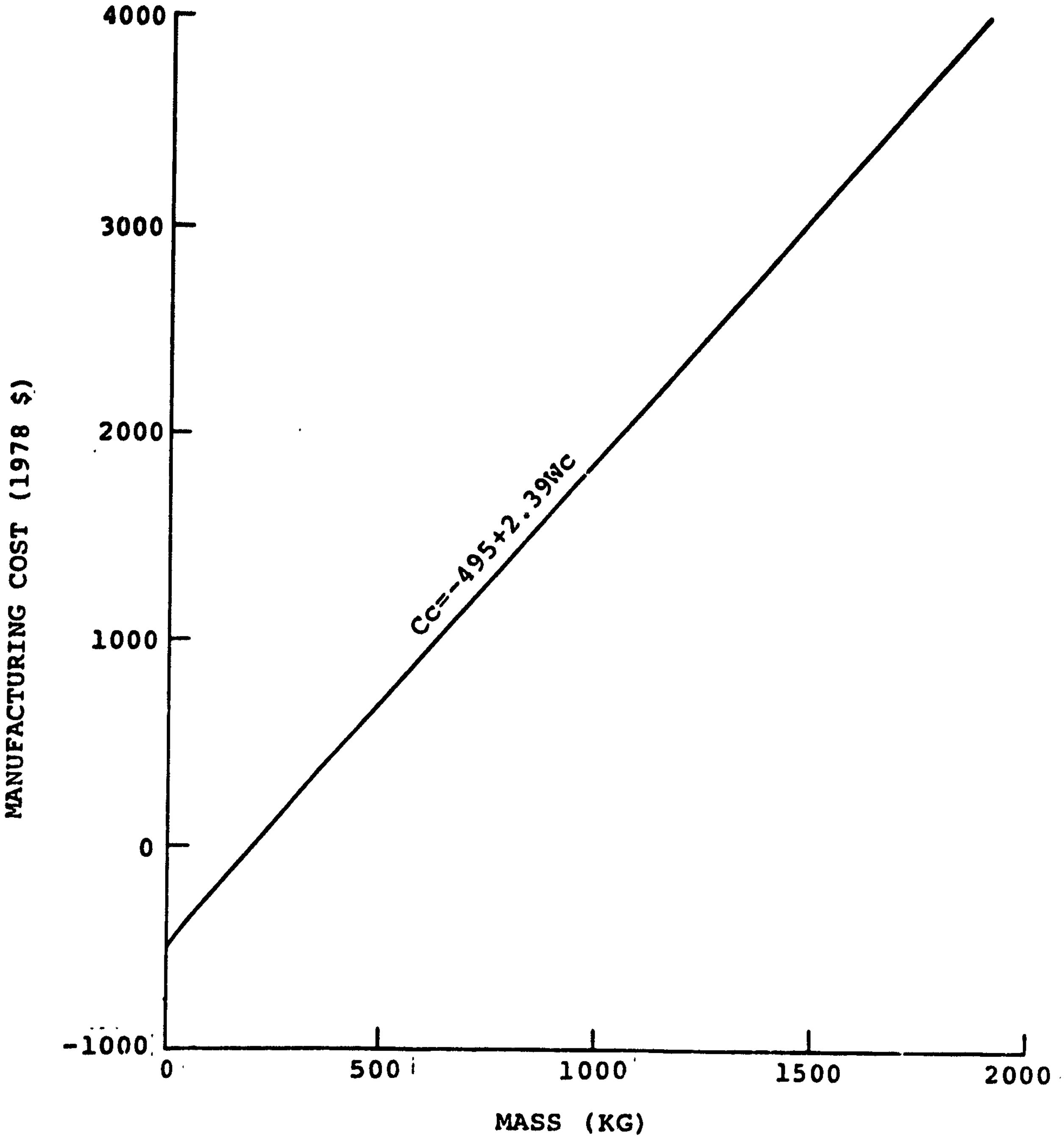

Figure 2-9 Vehicle Carriage Cost vs Weight Relationship

Specifically, if the added cost of the hybrid were handled on a minimum cost-to-the-consumer basis, the purchase price increment associated with the manufacturing cost increment would be a minimum of about 1.25 times the manufacturing cost increment (see discussion in Section 3.6.3). We made the further assumption that 'comparability' means that the retail purchase price of the hybrid should not be more than 25% higher than that of the reference vehicle. This leads to the relationship:

$$C_{R,RV} + 1.25\ (C_{M,H} - C_{M,RV}) \leq 1.25\ C_{R,RV}\ ,$$

where $C_{R,RV}$ = Retail price of reference vehicle

$C_{M,RV}$ = Manufacturing cost of reference vehicle

$C_{M,H}$ = Manufacturing cost of hybrid vehicle.

For $C_{R,RV} = 2 \cdot C_{M,RV}$, this implies

$$C_{M,H} \leq 1.4\ C_{M,RV}\ ;$$

i.e., the manufacturing cost increment of the hybrid over the reference vehicle should not exceed 40%. This limitation puts an upper bound on the range of permissible values of $\overline{W}_B$ for a given value of $\overline{P}_{HE}$.

3) An a priori bound of .8 was placed on the heat engine power fraction, under the assumption that anything over .8 is getting too close to a conventional vehicle.

These three constraints define a triangular region in the $\overline{P}_{HE}$, $\overline{W}_B$ plane; subsequent investigation was limited to this region.

Preliminary Control Strategy and Control Parameters

Before proceeding to the next step, which involved the estimation of fuel and energy consumption and life cycle costs over the range of basic parameters, it was necessary to define some sort of control strategy to use in the computer simulations which would provide the fuel and energy consumption estimates. As a result of some preliminary runs with the HYBRID simulation program, we came very quickly to the conclusion that to minimize fuel consumption, it would be best to shut the heat engine off entirely unless the power demand was too high for the traction motor to handle, or the batteries were at too low a level of discharge. This approach, which involves repeatedly starting the heat engine to supply power when the demand is there, and shutting it down when it is not needed, was initially viewed by us with a great deal of skepticism, in spite of its obvious desirability. However, consultation with our heat engine/transmission subcontractor led us to the conclusion that this type of operation is feasible, although some modifications might eventually be required (like providing an initial warm up period for the heat engine to ensure that operating fluids are up to temperature and parts have lubrication before any full power demands are made). This conclusion was affirmed by information from VW,(4) who have operated an engine successfully in this type of mode for extended mileages.

Consequently, for a preliminary control strategy, we assumed a bimodal strategy with the characteristics defined in Table 2-1. The strategy is defined by two quantities: the maximum battery

TABLE 2-1

Preliminary Control Strategy

MODE	BATTERY DISCHARGE	COMBINED HEAT ENGINE & MOTOR OUTPUT POWER, P	HEAT ENGINE OUTPUT POWER,	MOTOR OUTPUT POWER,
1	$\leq D_{BMAX}$	1.1 $0 \leq P_{SO} \leq P_{EOMIN}$	0	P_{SO}
		1.2 $P_{EOMIN} < P_{SO} \leq P_{EOMIN} + P_{MMAX}$	P_{EOMIN}	$P_{SO} - P_{EOMIN}$
		1.3 $P_{EOMIN} + P_{MMAX} < P_{SO} \leq P_{HEMAX} + P_{MMAX}$	$P_{SO} - P_{MMAX}$	P_{MMAX}
		1.4 $P_{SO} < 0$	0	MAX (P_{SO}, P_{MMIN})
2	$> D_{BMAX}$	2.1 $0 < P_{SO} \leq P_{HEMAX}$	P_{SO}	0
		2.2 $P_{HEMAX} < P_{SO} \leq P_{HEMAX} + P_{MMAX}$	P_{HEMAX}	$P_{SO} - P_{HEMAX}$
		2.3 $P_{SO} \leq 0$	0	MAX (P_{SO}, P_{MMIN})

P_{EOMIN} = Minimum heat engine operating power level (Mode 1)

D_{BMAX} = Battery discharge level (0 = fully charged, 1 = fully discharged)

P_{HEMAX} = Maximum heat engine power output

P_{MMAX} = Maximum traction motor power output discharge level, D_{BMAX} , and a minimum heat engine operating power level, P_{EOMIN} . Until the battery reaches the discharge level D_{BMAX} (Mode 1), the system is operated on stored energy (Cases 1.1, 1.4) unless the system power demand P_{SO} exceeds the heat engine cut-in value P_{EOMIN} . For system demands above P_{EOMIN} , the heat engine is operated at P_{EOMIN} (Case 1.2) unless the system power demand is so great that the motor output exceeds the maximum available, P_{MAX} (Case 1.3). Once the battery reaches the maximum discharge level, the second operating mode takes over. On this mode, the roles of the heat engine and traction motor are essentially reversed; on Mode 1, the heat engine is used for peaking, whereas on Mode 2, the traction motor is used for peaking (and regenerative braking), and the heat engine supplies the average system requirements.

This control strategy is by no means optimum; however, it is plausible; and it accomplishes the two goals of running the heat engine as much as possible near its minimum bsfc and using as much stored energy as possible. Consequently, it is adequate to further localize the range for the two basic parameters.

Estimation of Fuel and Energy Consumption

Fuel and energy consumption were estimated using the program HYBRID, documentation for which is provided in Appendix B1 to this report. This program simulates operation of a hybrid vehicle over a composite driving cycle of the type discussed in the Task 1 report,(1) using a control strategy of the type just discussed. Since the purpose of this program was to help in localizing the range of the basic parameters, rather than optimizing a control strategy or investigating the effects of detailed component changes, the simplest possible representation was used of all components. These representations included the following:

- Heat engine. Represented by a curve of brake specific fuel consumption vs. power output. In effect, this representation assumes a continuously variable transmission which permits the engine to operate at or near its best bsfc for a given power level. The curve was derived from a fuel map of a 140 CID Ford engine[5] and is shown in Figure 2-10.
- Electric motor. Electric motor input is represented by the motor output, divided by a constant efficiency, added to a no load power input (representing field excitation, no load armature current, etc.). An efficiency value of .87 and a no load power input of 1.5 kw was used for these studies.
- Transmission. Transmission efficiency is considered constant. A value of .92 was used.
- Differential. Constant efficiency of .96 was assumed.
- Tires. A constant rolling resistance coefficient of .010 was assumed.*
- Aerodynamic drag. A drag coefficient - frontal area product of .872 m^2 was used, corresponding to a drag coefficient of .4 and a frontal area of 2.18 m^2 (23.5 $ft.^2$).*
- Batteries. Depth of discharge for a given day's travel was calculated based on the nominal battery capacity for ISOA

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* Rationale for rolling resistance and drag figures is given in the report on Task 1(1), pp. 30-32.

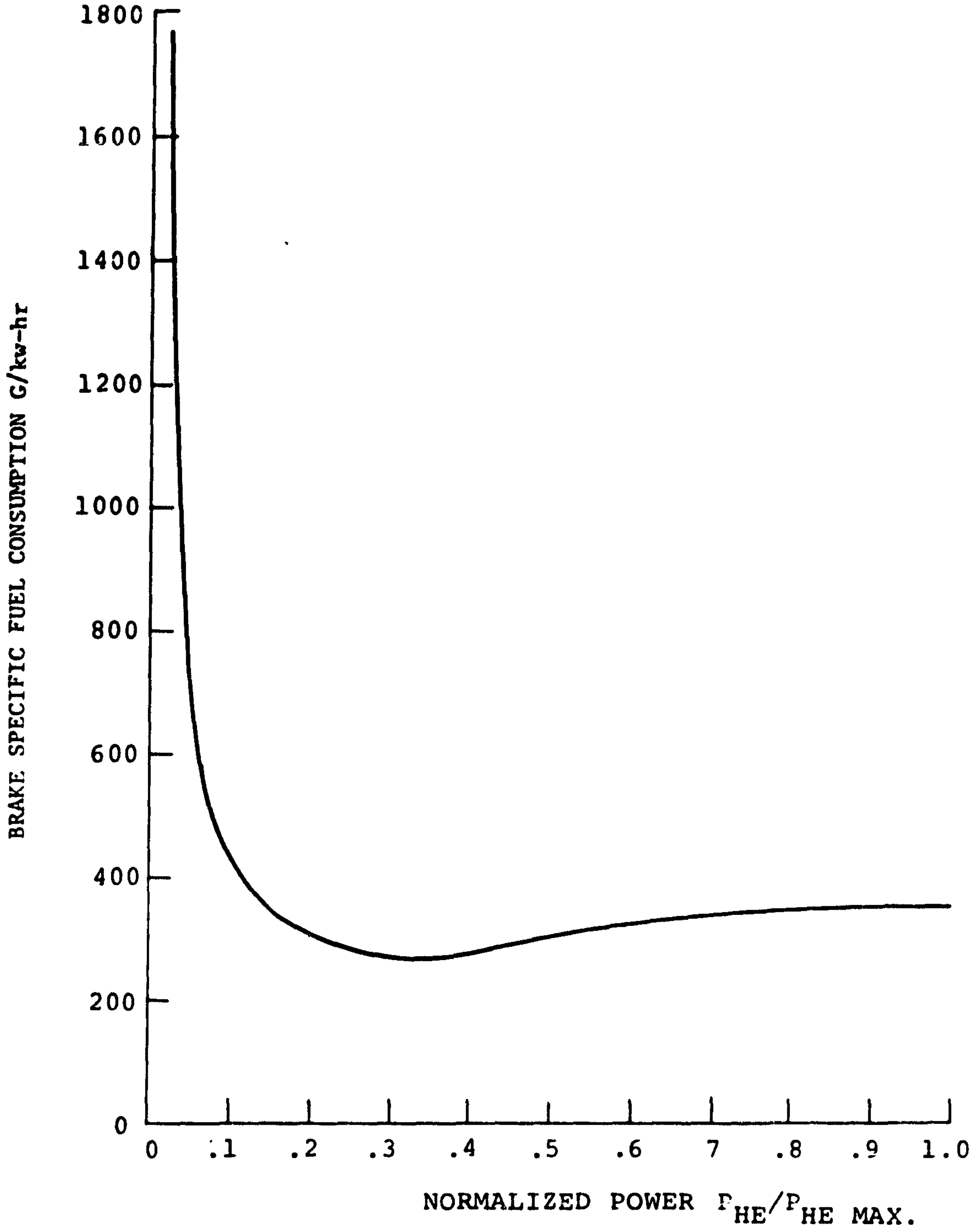


Figure 2-10 Heat Engine BSFC Vs. Normalized Power

batteries (40 w-hr/kg for lead-acid batteries, 70 w-hr/kg for nickel-zinc, 50 w-hr/kg for nickel-iron). An average depth of discharge was calculated over a year's usage, and battery life was estimated based on a curve of battery life vs. depth of discharge. These curves were estimated based on data for existing batteries and Argonne's goals for ISOA batteries at 80% DOD. The curves used for lead-acid, nickel-zinc, and nickel-iron are shown in Figure 2-11. This method tends to overestimate the available battery energy and battery life, because the rates of discharge for the hybrid are generally higher than that on which the ISOA battery capacity is predicated. In addition to the above, a regeneration efficiency of .6 was assumed for Mode 1 operation (high state of charge) and .85 for Mode 2 (low state of charge). Both of these values may be somewhat low. For lead-acid and nickel-zinc batteries, an overall efficiency for battery recharging of .54 was assumed (.6 battery charge cycle efficiency, .9 charger efficiency); for nickel-iron, a slightly higher battery charge cycle efficiency was used (.7), giving an overall efficiency of .63.

The program computes the fuel consumption and battery output energy in gm/km and kw-hr/km, respectively, over each of the component driving cycles (SAEJ227(a)Sched B, FUDC, and FHDC) for both Mode 1 and Mode 2 operation. For each of the 21 composite driving cycles representing different daily travel distances (see Table on p. 75 of the Task 1 report[1]), this information is used to compute

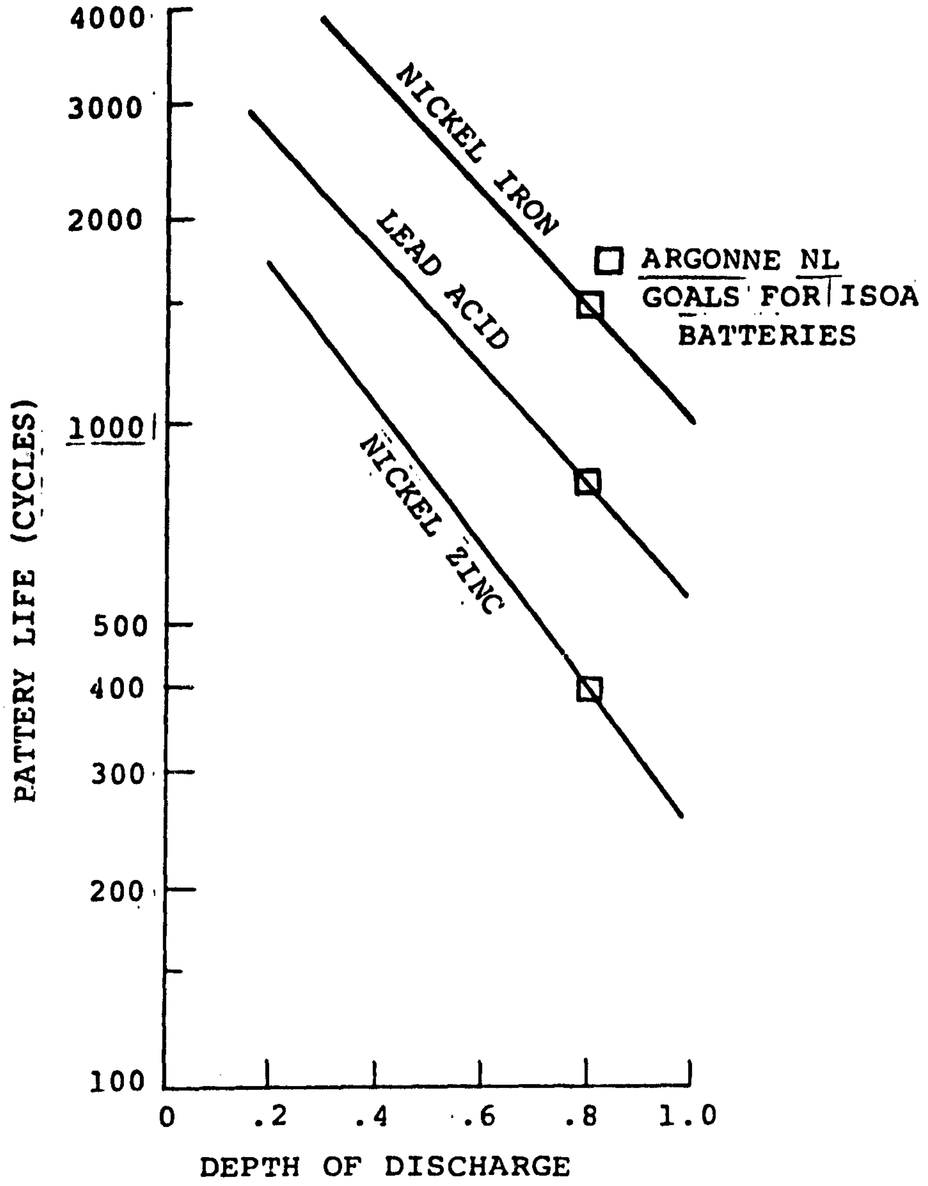


Figure 2-11 Assumed Battery Life Characteristics

the fuel and battery energy consumption on Mode 1, the range on Mode 1, and the fuel consumption on Mode 2 (corrected so that the net battery energy consumption on Mode 2 is zero). Finally, the yearly average fuel and energy consumption are computed based on the distances travelled on Modes 1 and 2 for each of the composite driving cycles, the fuel and energy consumption in Modes 1 and 2 for each cycle, and the distribution of total travel relative to the 21 composite cycles (again, see the above referenced table). The wall plug output is then computed from the battery recharging efficiency.

The program was exercised for the reference vehicle, for which it gave a fuel consumption estimate about 11% lower than the value projected for the 1985 reference vehicle,(1) corresponding to 18 mpg. This optimism is largely due to the assumption about the variation of engine bsfc with engine power (equivalent to the assumption of a CVT). As a result of this, all projections of fuel economy for hybrid vehicles obtained from this program were multiplied by .89 to avoid overestimating the gains from a hybrid propulsion system.

Tightening of Basic Parameter Ranges

In attempting to draw the bounds a little tighter around the acceptable range of the basic parameters $\overline{W}_B$ and $\overline{P}_{HE}$, we took the viewpoint that life cycle cost and fuel consumption are the two principal variables to be considered in doing this. It would be too much to hope for that both these variables would reach minimum values for the same combination of $\overline{W}_B$ and $\overline{P}_{HE}$; and, indeed, this was not the case. As discussed in Section 3.1, low fuel consumption is favored by a high value of $\overline{W}_B$ and low $\overline{P}_{HE}$; low life cycle cost is favored by the reverse situation. In light of this, the approach taken was as follows. For each combination $(\overline{P}_{HE}, \overline{W}_B)$, a number of cases were run with HYBRID, with various values of the control parameters P_{EOMIN} and D_{BMAX}. Life cycle costs were obtained in each case using the program LYFECC (documented in Appendix A3 of the Task 1 report[1]). For each case, the life cycle cost was plotted against the fuel consumption. A series of curves and envelopes of curves was then drawn; and, based on the shape of the overall envelope and the proximity of the individual points to it, a judgment was made as to localizing the range of the parameters $\overline{P}_{HE}$ and $\overline{W}_B$. This will become clearer when the actual data and results are discussed in Section 3. At this point, it suffices to say that, if the overall envelope looks qualitatively like that shown in Figure 2-12, the place to be is somewhere near the knee of the curve, rather than out at the extremes where a small improvement in fuel consumption costs a lot in terms of life cycle cost, or conversely.

2.1.2 Subsystem and Component Level Tradeoff Studies

Construction and Simulation of Baseline Systems

After the selection of a limited range for the basic parameters which define the vehicle weight and major components power ratings, the next step was to construct a baseline hybrid vehicle and propulsion system with parameters within that range. This vehicle would serve as the focal point for making design variations and investigating the tradeoffs involved in such variations. Because of the critical nature of its function as a starting point and as an aid in making

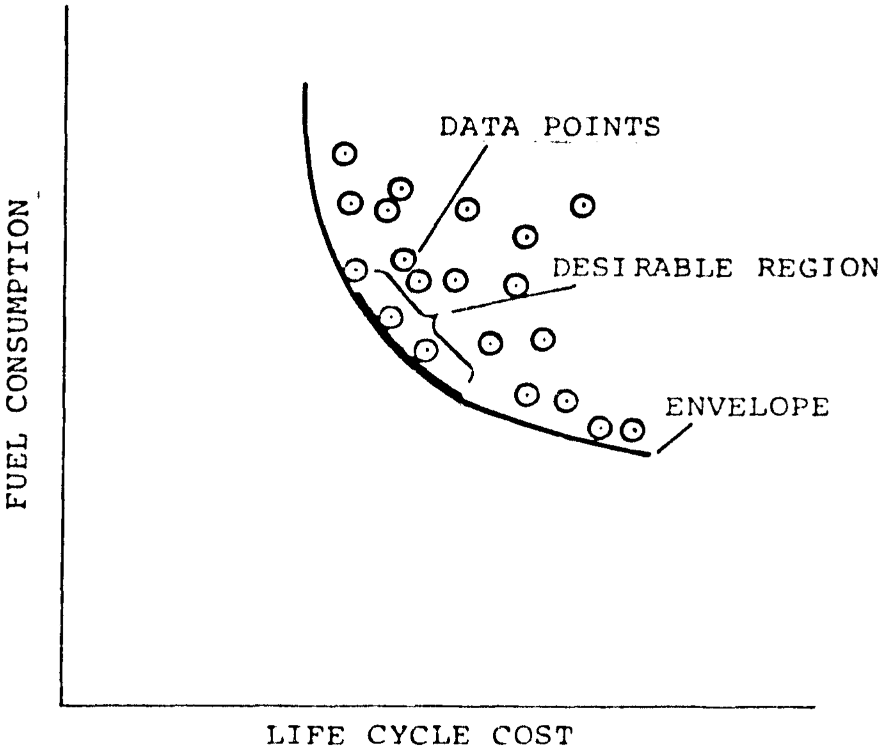


Figure 2-12 Selection of Optimum Region for Basic Parameters

intelligent tradeoff decisions, it was imperative that the baseline system be a reasonably good one to start with. Consequently, considerable effort was expended in selecting the system configuration and in developing a control strategy which would give a good combination of fuel economy and life cycle cost for the system configuration and parameters chosen.

The major tool used in constructing the baseline hybrid was a computer simulation, HYBRID2. This program evolved from HYBRID and differs from it in the following important aspects:

- Treatment of energy flow. The HYBRID program basically dealt in terms of power; i.e., components were modelled as devices with a power input and power output, with factors such as efficiency, fuel consumption, etc., being treated only as a function of power output. HYBRID2, on the other hand, deals with torque and speed separately, rather than just power. This permits more detailed modelling of components such as the heat engine, torque convertor and traction motor.
- Heat engine modelling. HYBRID2 determines the instantaneous heat engine fuel consumption from an engine fuel map which defines the brake mean specific fuel consumption (bsfc) as a function of brake mean effective pressure (bmep) and engine speed. The program also uses a curve of maximum available torque vs. engine speed to determine the available torque at a given car speed, rather than assuming that the peak engine power is available at any car speed, as HYBRID does.

- Traction motor modelling. HYBRID assumes that the peak motor power is available independent of rpm (which is essentially true except when the motor is an armature control, which occurs only at very low vehicle speeds). HYBRID2 includes details on the dropoff in available motor power (both as a motor and as a generator) at both low and high speeds. The modelling of motor input/output characteristics by a combination of a fixed efficiency together with a fixed no load power loss was retained, since it turned out to represent actual motor data quite well (see Figure 3- 22).
- Transmission modelling. The original version of HYBRID2 included the capability of simulating an automatic transmission with torque convertor, with or without lockup. The torque convertor is modelled by curves of speed ratio (output/input) and torque ratio (output/input) vs. an output speed/torque factor (output speed/$\sqrt{\text{output torque}}$). Subsequently, HYBRID2 was expanded to include the capability of simulating a continuously variable transmission, together with a control strategy appropriate to this type of transmission.
- Battery modelling. Although the original version of HYBRID2 and the first runs were made with the same simple battery model as HYBRID, the program was subsequently modified to include the effect of rate of discharge on capacity. For each of the 21 composite driving cycles, the program computes

an average battery specific power for Mode 1 operation, determines the corresponding available specific energy, and determines the range on Mode 1 operation from this and from the battery energy consumption per kilometer on Mode 1.

Selection of the heat engine, traction motor, and transmission for the reference vehicle was made on the basis of using the most advanced technology currently available in production hardware. This meant a conventional, reciprocating gasoline engine, a separately excited DC traction motor with a high limiting speed and power-to-weight ratio, and three-speed automatic transmission with lockup torque convertor. Further discussion of specifics will be found in Section 3 of this report.

Parametric Analyses and Supporting Studies

The purpose of these studies was to generate the data which would provide the basis for making intelligent and realistic trade-offs regarding the selection of design parameters and design of the propulsion system and overall vehicle. They were conducted in a number of different areas, which may be grouped as follows:

1. Determination of the effects of variations in vehicle characteristics from the values used in the baseline vehicle. These characteristics included weight, drag coefficient, and rolling resistance. The intent of these studies was to assess the relative importance of these characteristics in terms of their effects on fuel consumption and to develop data which would provide the basis for estimating how much of a manufacturing cost increase

(associated with any improvement in one of these characteristics) would be justified by an associated improvement in fuel consumption.

2. Determination of the effects of variations in propulsion system characteristics from the values used in the baseline vehicles. These are characteristics over which we have somewhat more control than those in the first group. They include both physical parameters such as the engine size, transmission and rear axle ratios, and control parameters such as the battery discharge limit, and the overall control strategy.

3. Determination of the effects of design approaches which are alternatives to those used for the propulsion system components or subsystems of the baseline system. Such alternative approaches would include the use of a diesel rather than spark ignition gasoline engine, a continuously variable transmission (CVT) rather than an automatic with lockup torque convertor, and so forth.

4. Associated studies not directly concerned with the propulsion system, but which provide supporting rationale for the overall vehicle design. These include material cost and substitution studies and packaging studies.

These studies were generally concerned with quantifiable aspects of the system, such as fuel and energy consumption, manufacturing cost, retail price, life cycle cost, and acceleration performance. These were estimated using the programs HYBRID2, WANDC, and LYFECC, discussed previously. A modified version of VSPDUP, called VSPDUP2, was used for performance estimation. This program includes a detailed torque convertor representation as well as having provisions for a continuously variable transmission simulation.

The analyses of these quantifiable factors were carried out to the level of detail needed to make a good evaluation. If it became clear that there were overriding considerations, other than those mentioned above, which would eliminate an alternative, then detailed quantitative studies were not carried out.

Evaluation of Design Alternatives and Tradeoffs

In addition to sorting through and evaluating the quantitative data on fuel and energy consumption, costs, and performance generated in the studies described previously, other factors were taken into account in evaluating design alternatives and parameter variations. These included emissions, driveability, reliability, and technological requirements. A brief discussion of these additional factors is required at this point.

- Emissions. Although the computer simulation program HYBRID2 could easily be expanded to include steady state engine emission maps, it was concluded that any results obtained in this manner would be next to meaningless because of the unique way in which the engine is operated in the hybrid system, i.e., in an on-off mode. Even with a conventional system, the use of such maps is not particularly useful (with the possible exception of NOx emissions), since a large part of the CO and, particularly, HC, emissions occur during the cold-start period. This, together with the fact that an entirely different engine calibration would probably be required to suit the on-off operation led us to the conclusion that emission control is an area which will have to be treated on an experimental development basis as part of the system development program. An exception to this involves differences in emissions which result from basic differences in engine type. For example, it is safe to predict that a diesel will have more of a problem with particulates than a spark ignited gasoline engine. An emissions question which does not require immediate resolution, but which must eventually be addressed in a development program, involves the test procedure which would be applied to a hybrid vehicle with more than one operating mode. Our assumption would be that EPA would make the minimum possible adjustment to its current test procedure to accommodate the multi-modal operation. This could involve running two tests according to the existing test procedure using the urban driving cycle: one starting with fully charged batteries (Mode 1 operation) and one with batteries discharged at least to the discharge limit D_{BMAX} (Mode 2 operation). The final emission values would then be calculated as weighted averages of the Mode 1 and Mode 2 emissions, with the weights determined by the ratio of the operating range on Mode 1 to the average daily driving distance.

- <u>Driveability</u>. (Throttle response, closed throttle deceleration, etc.) This is obviously one of the major development areas associated with the approach of turning the heat engine on and off on demand. It is likely to require a great deal of development to get sorted out; however, the fuel economy payoff makes the technological development task worth undertaking. This development is going to be difficult with a good gasoline engine; engines with more difficult starting characteristics were thus heavily downrated in our evaluation process. We also included in this category the overall operating 'feel' of the system. A system with operating characteristics which would give the driver the feeling that something is not quite right would be downrated. (Example: an IC engine operating at near full throttle and speed while the vehicle is stationary) Our objective is to provide the driver with a propulsion system that meets the same standards of smoothness and quietness currently attained in full-sized production cars.
- <u>Reliability and failure characteristics</u>. Any design approach involving component or system characteristics which would result in a failure rate significantly higher than a conventional vehicle, or failures which result in hazard to the occupants or other vehicles, or 'fail-hard' failure modes, led to that approach being downrated.
- <u>Technological requirements</u>. By this, we mean the requirements and risk involved in the development of immature

technology to achieve production status by 1985, together with the requirements for implementing the technology in production and the compatibility of those requirements with the manufacturing structure of the automobile industry.

The process of evaluating design alternatives with respect to the above factors and the quantitative ones was as follows: First, a design approach was screened in terms of those factors which did not require detailed computation to evaluate; and, if it was apparent that it had serious shortcomings in one or more areas, it was dropped (for example, if the technology development required to bring it to production status by 1985 appeared to involve a very high risk). It must be remembered that the basic approach we have taken, which involves on-off operation of a heat engine, itself constitutes a significant development task and will require extensive test and development to obtain operating characteristics acceptable to the average American driver, who is used to extremely quiet and smooth operation, particularly in a large vehicle, and who is very well isolated from the inner mechanical workings of his vehicle. In addition, in any electric or hybrid vehicle development program, the development of batteries which meet the vehicle and propulsion system designers' requirements must be regarded as a high risk. As a consequence, our design philosophy was to be quite conservative about introducing additional high risk concepts on top of these. We felt it would be far better to do a thorough job of development on a system with one major development requirement (other than batteries) and achieve 100% of the fuel economy gain possible with the system, than to incorporate additional high risk approaches which may offer a somewhat higher potential and then run the risk of realizing only 50% of the potential fuel economy gain because the total development task is unrealistic for the near term vehicle.

If a design approach survived this preliminary screening process, then it was subjected to detailed analysis using the various computer programs described previously, and an overall evaluation was made relative to the baseline hybrid system.

2.2 Scope

The universe of design alternatives investigated (at various levels of detail) in the design tradeoff studies was limited to the following:

Vehicle size and accommodations: 6 passenger, full-sized car similar to Ford LTD.

Hybrid system configuration: Parallel hybrid only (i.e., both the heat engine and electric motor supply mechanical power to the rest of the drivetrain). Series hybrids were not considered because of the necessity to size the electric motor, controls, and batteries to handle the maximum system power requirement without help from heat engine. To meet the performance requirements, such a system, designed with near term technology, gets to be outlandish in size and manufacturing cost.

Heat engine: Conventional spark ignited gasoline (Otto cycle), stratified charge, and diesel reciprocating engines. Gas turbines, Stirling engines, Rankine cycle engines, and so forth, were excluded as not being capable of reaching production status by the mid-1980's.

Electric motor/controls: DC series, shunt, and permanent magnet and AC induction motors, with appropriate controllers using SCR's or transistors.

Transmission: Three and four speed automatics with lockup torque convertors, various types of continuously variable transmissions, automatically shifted gearboxes.

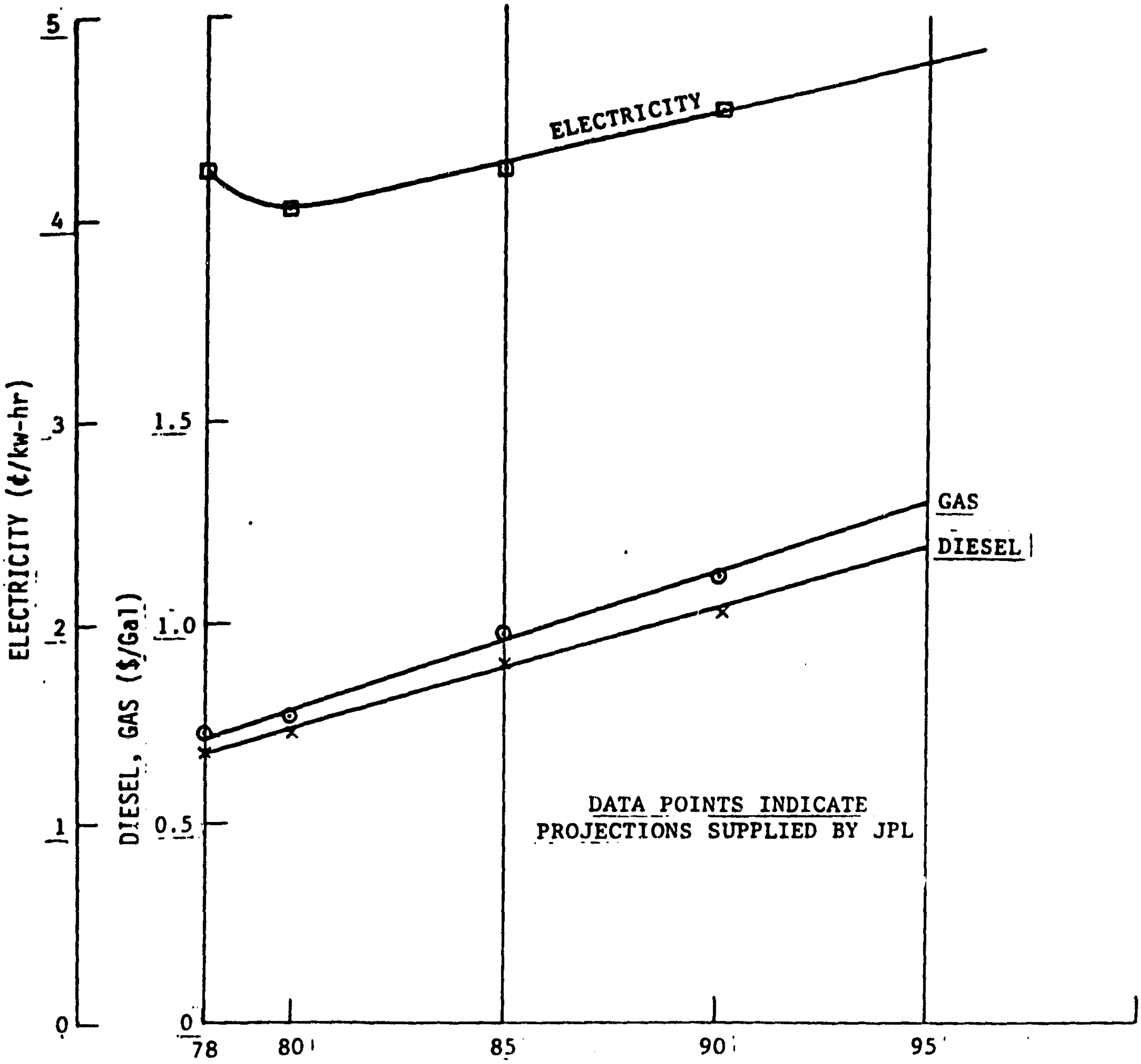
Batteries: lead-acid, nickel-zinc, and nickel-iron battery types. Molten salt and other exotic battery types were excluded on the basis of immature technology.

Drive layouts: Conventional front engine, rear drive only. This was done to obtain compatibility with the reference vehicle; front wheel drive would have been used if the selected reference vehicle had that layout. Arrangements with multiple drive motors, four wheel drive, and so forth, were not considered, again because of incompatibility with the referenc vehicle. We view this point as being quite important. The more components and subsystems which are totally unique to a hybrid vehicle, the less attractive it becomes to a manufacturer as a means to reduce his CAFE, and the less likely is the technology to be transferred to the automobile industry.

Energy buffers: Flywheels only. Hydraulic pumps/motors and accumulators were not considered because of low efficiency, noise problems, and general lack of elegance.

Vehicle body and structure: Separate frame and body only. Again, if the selected reference vehicle had unitized construction, this would also have been used for the hybrid. Detailed consideration was given to the substitution of alternative materials (aluminum and plastic composites) for selected components.

Sensitivity studies: Studies were conducted to determine the impact on the vehicle tradeoff results as they may be affected by high and low volume gasoline/diesel or electricity prices. Values studied were those provided by JPL. In addition, price sensitivity of hybrid vehicles was studied by a subcontractor to show the importance of price and the need to allow flexibility in pricing. Note that the nominal prices for fuel and electricity were assumed to vary, as shown in Figure 2-13; variations were applied to these nominal values.



OVER PERIOD 1985 - 1995

GAS: $\$/Gal = 0.95 + 0.034\ \Delta t\ (\Delta t = t-85)$

DIESEL: $\$/Gal = 0.88 + 0.030\ \Delta t$

Electricity: $\$/kw\text{-}hr = 0.0423 + 0.00049\ \Delta t$

Figure 2-13 Fuel and Electricity Prices

3. DISCUSSION OF RESULTS

3.1 System Level Tradeoff Studies

3.1.1 Ranges of Values for Basic Parameters

Using the weight and manufacturing cost program, WANDC, a series of runs were made for heat engine power fractions ($\overline{P}_{HE}$) ranging from .3 to .8 and battery weight fractions ($\overline{W}_B$) from .1 to .3, for both lead-acid and nickel-zinc batteries. These runs were later expanded to include nickel-iron batteries. Typical run results are shown in Figures 3-1 and 3-2 for a heat engine power fraction of .6. The plots of vehicle and battery weight and peak battery specific power are identical for the two cases; however, the vehicle cost curve rises much more steeply for the nickel-zinc case as a result of the high battery cost. The intersection of the manufacturing cost curve and the peak battery specific power curve with the two constraints of not exceeding 1.4 times the manufacturing cost of the reference vehicle and not exceeding the relevant maximum battery specific power (100 w/kg for lead-acid, 150 w/kg for nickel-zinc) are shown on the plots of Figures 3-1 and 3-2. The first of these constraints is based on a manufacturing cost of $3823 for the reference vehicle, as predicted by the WANDC program.* Note that for $\overline{P}_{HE}$ = .6, the intersection of the region which satisfies both constraints for the nickel-zinc batteries is void; i.e., $\overline{P}_{HE}$ = .6 is not an acceptable parameter value for the case. For the lead-acid

* Note that the manufacturing cost predictions for the reference vehicle and hybrid vehicle both include a cost increase of $325 relative to 1978 production cars due to introduction of a microprocessor, fuel system sensors and controls, lockup torque convertor, and tire improvements. See p. 42 of the Task 1 report (1) for details.

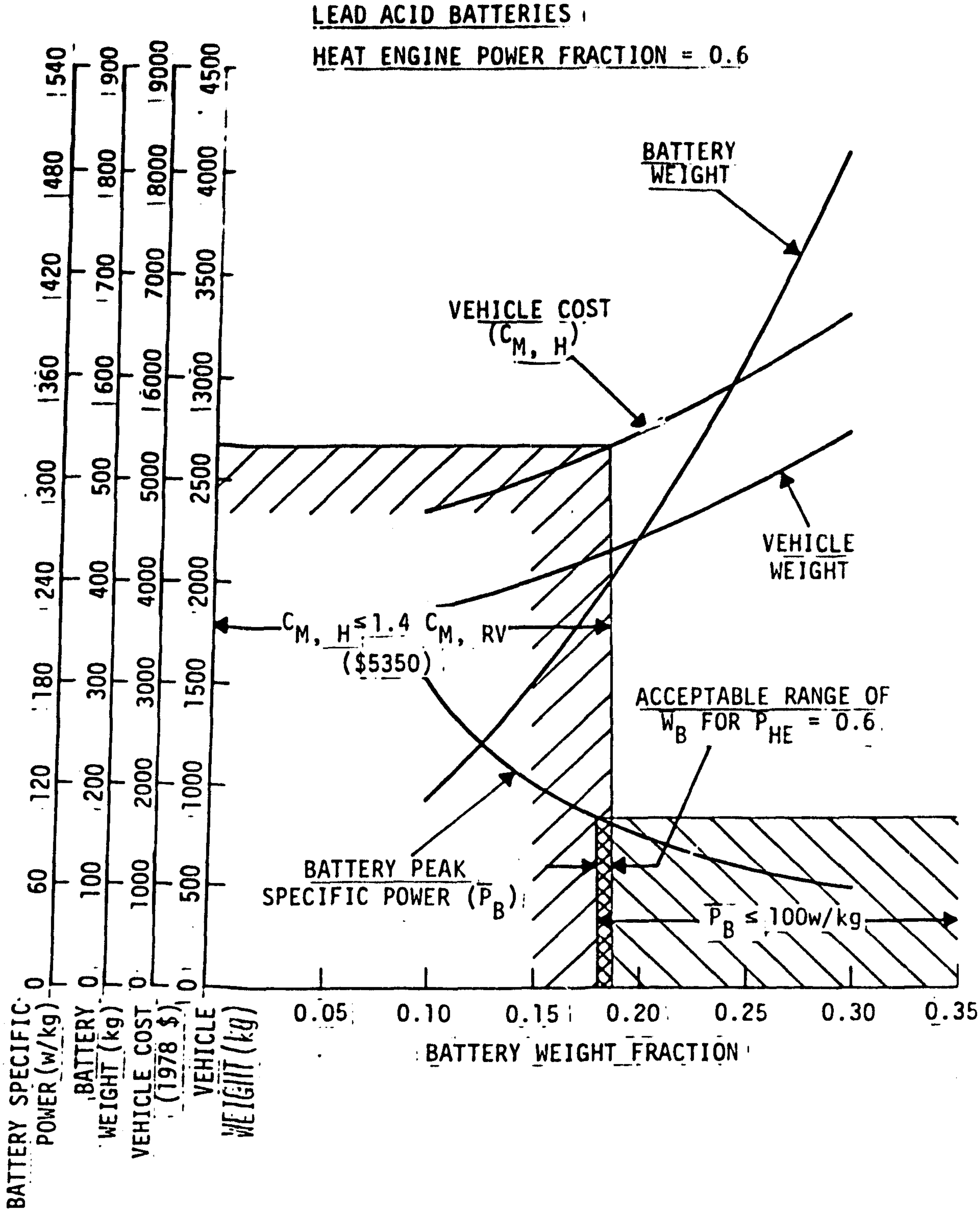


Figure 3-1 Effects of Battery Weight Fraction on Various System Characteristics

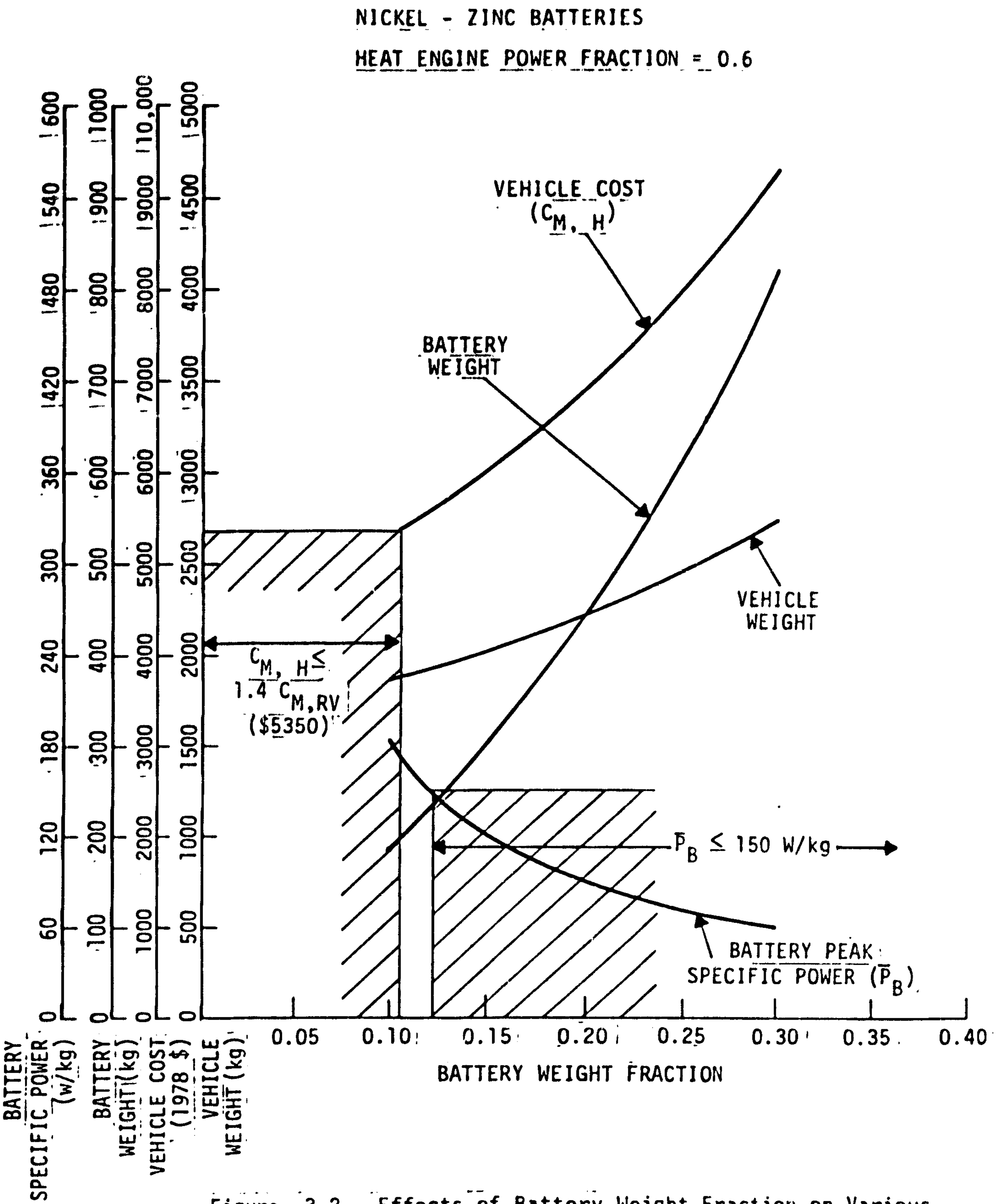


Figure 3-2 Effects of Battery Weight Fraction on Various System Characteristics

case, the intersection is non-void, giving a very narrow acceptable range of battery weight fractions from .180 to .186.

In this manner, the intersections of the relevant curves with the constraint lines were picked off for all values of $\overline{P}_{HE}$ from .3 to .8 and plotted in the ($\overline{P}_{HE}$, $\overline{W}_B$) plane. The results are shown in Figures 3-3 through 3-5. As expected from the standpoint of manufacturing cost limitations, the region of acceptable values of ($\overline{P}_{HE}$, $\overline{W}_B$) is considerably smaller for nickel-zinc and nickel-iron batteries than for lead-acid. At this point, the acceptable region can be reduced still further by considering the following. First of all, the regions above the dashed lines in Figures 3-3 through 3-5 are extremely small due to the fact that the manufacturing cost constraint line is not particularly sensitive to the heat engine power fraction (i.e., additional heat engine capacity can be added at a small penalty in manufacturing cost). Consequently, for the purposes of this analysis, we can restrict the region to the points below the dashed line. Now consider any point in the remaining region which is off the line segment A-B, e.g., point P in Figure 3-3. The vehicle which corresponds to this point can be viewed as a modification to a vehicle P', with the same battery weight fraction, located on the line segment AB. Now, what are the differences between P and P'? P has a larger heat engine; consequently, in the continuous rather than discrete world with which we are dealing at the moment, it has a slightly higher capacity transmission, weighs a little more, hence, costs a little more. Yet, it has the same battery capacity relative to its weight and a lower capacity traction motor; consequently, the

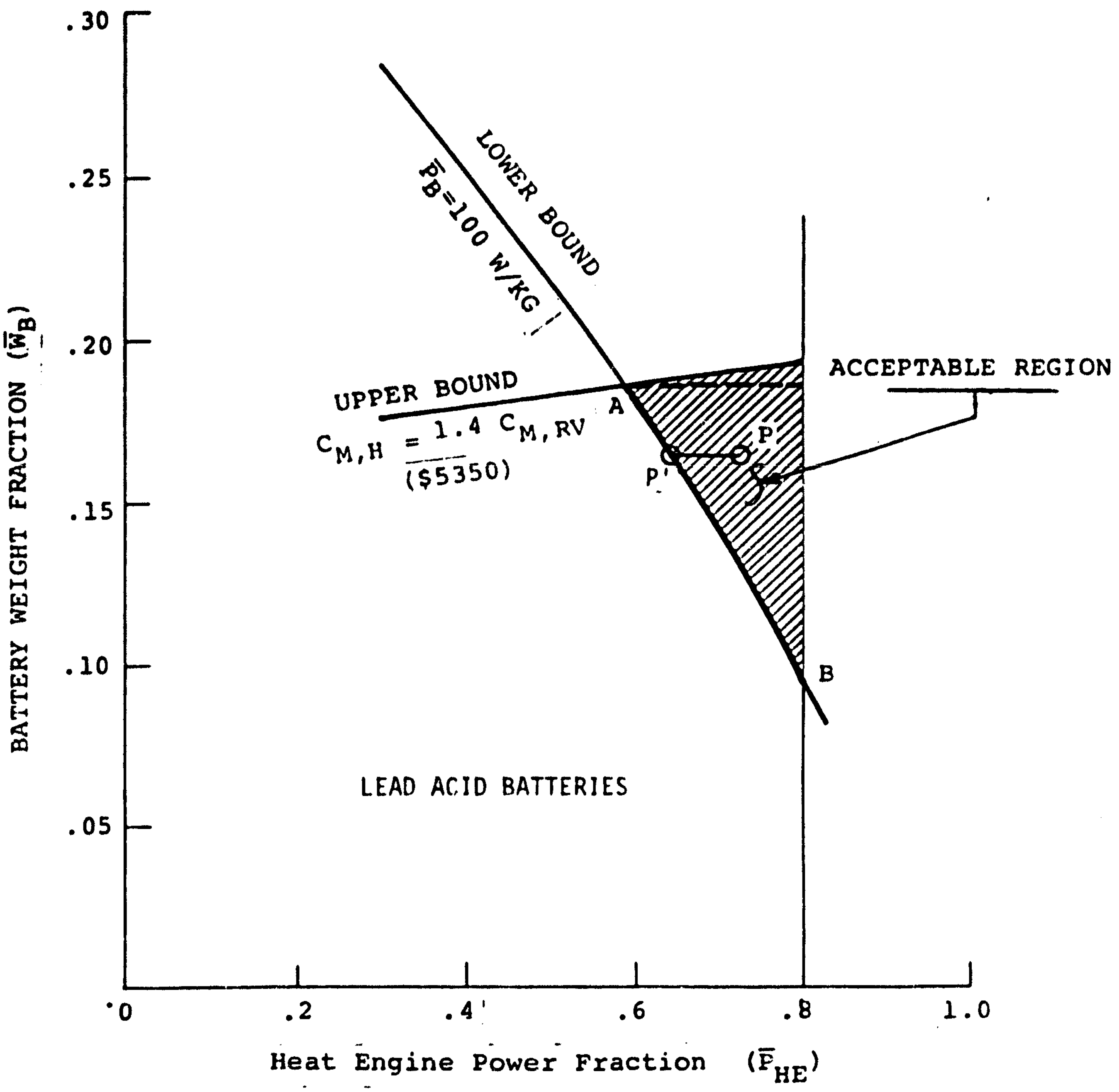


Figure 3-3 Acceptable Range of Basic Parameters

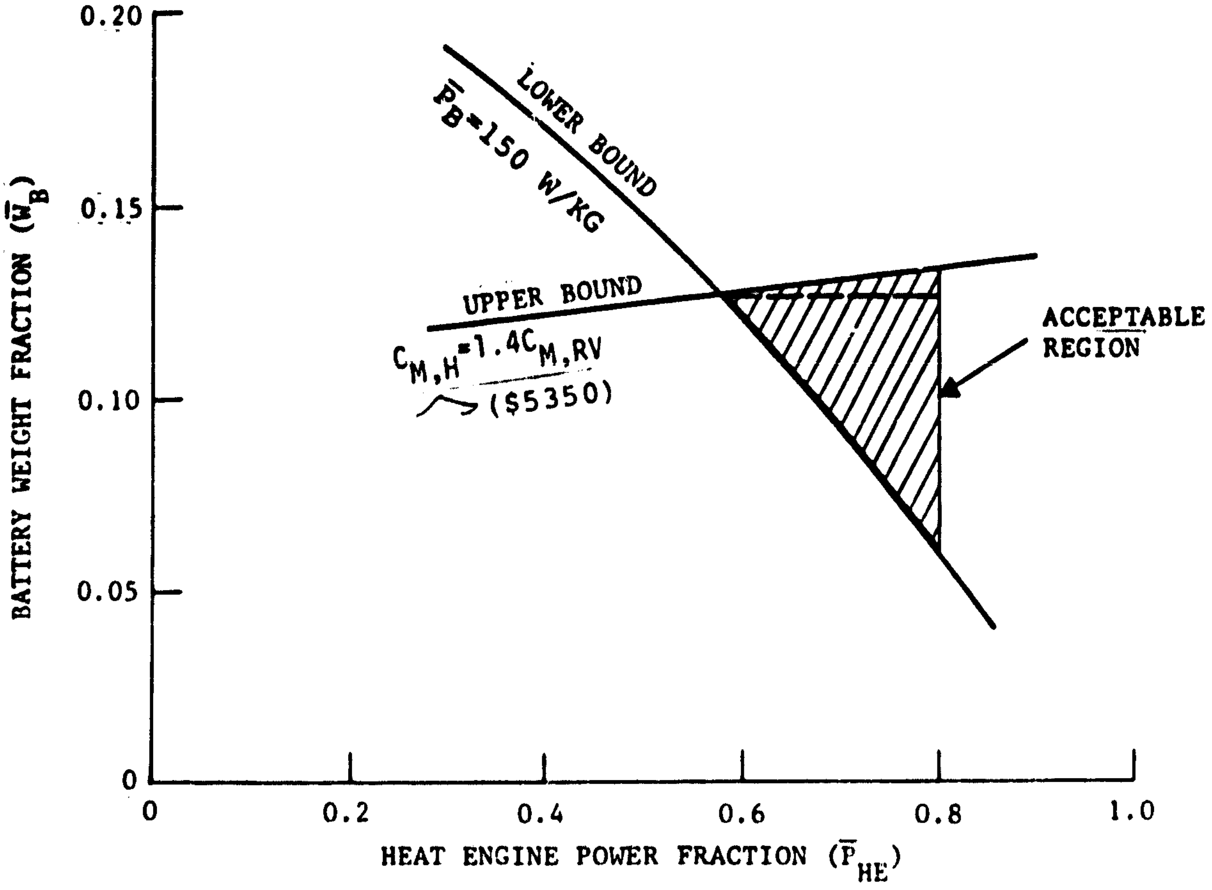


Figure 3-4 Acceptable Range of Basic Parameters (Nickel-Iron Batteries)

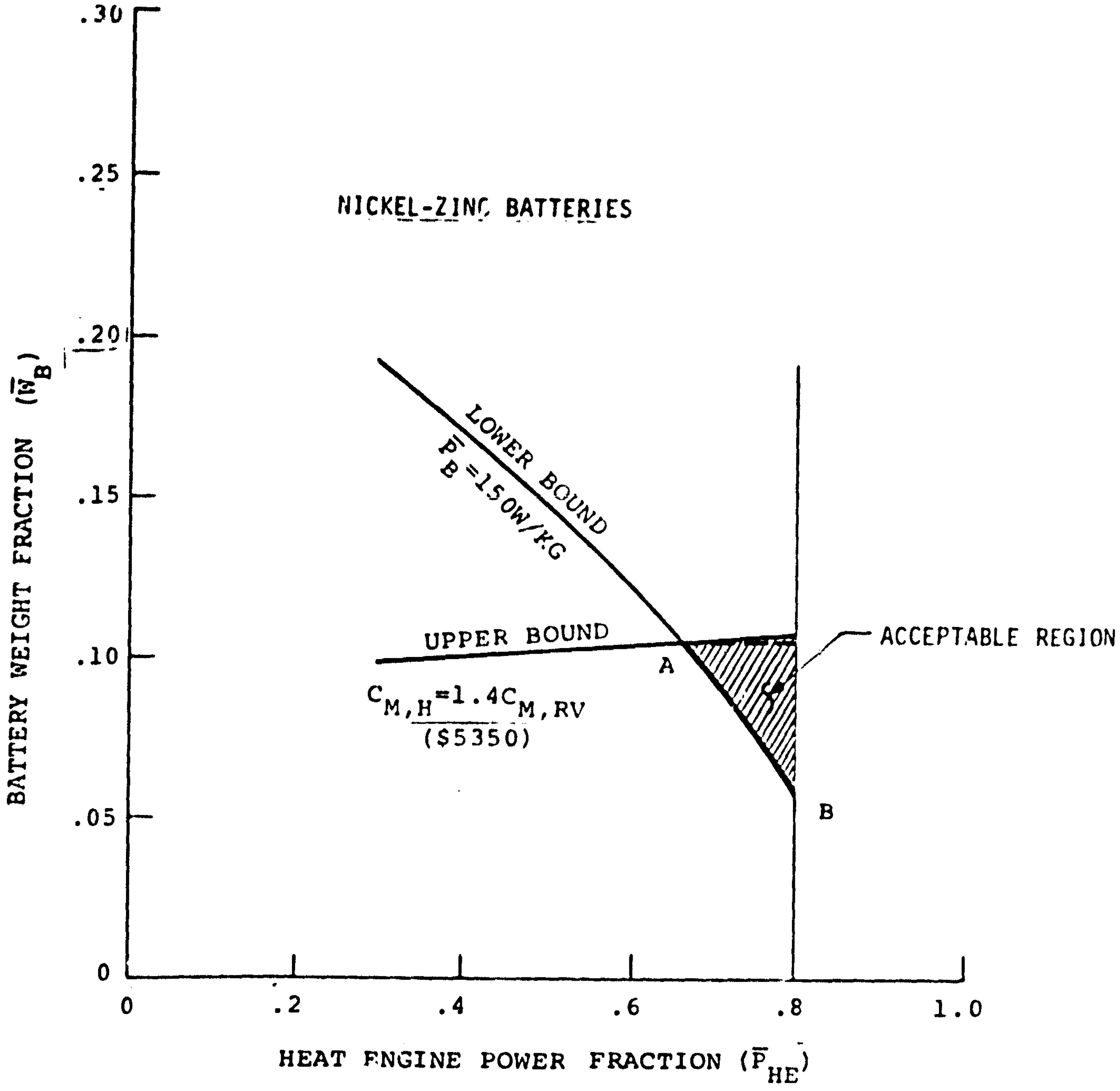


Figure 3-5 Acceptable Range of Basic Parameters

heat engine will have to be worked at least as much as that of P'. However, P' has a smaller, more heavily loaded heat engine, which consequently operates at a lower bsfc; ergo, the fuel economy of P' will be better than that of P. To sum up: P is more expensive, heavier, and consumes more fuel than P'. Consequently, it suffices to consider cases along or near the line segment AB, rather than throughout the whole region. Based on this, the cases shown in Table 3-1 were pursued in more detail using the HYBRID simulation program and LYFECC life cycle cost program.

3.1.2 Fuel and Energy Consumption and Life Cycle Cost Estimates

To estimate fuel and energy consumption, each of the configurations was run on the HYBRID simulation program with various values of the control parameters PEOMIN and DBMAX. In general, values of the heat engine cut-in power PEOMIN from 7 kw up to 20 kw were used except where the traction motor was not capable of delivering 20 kw, and the range of the battery discharge limit was from .4 to .8. The projected in-use fuel economy for these cases ranged from a low of about 12 km/l (28.2 mpg), to a high of 24 km/l (56.4 mpg). Wall plug energy consumption ranged from .1 kw-hr/km up to .26 kw-hr/km. These values will be subsequently discussed and plotted in more detail; however, at this point, it would be well to develop a realistic preliminary understanding of what is involved in trying to keep the life cycle cost of a hybrid vehicle down to a reasonable level.

In Figures 3-6 through 3-8, we have plotted the present value of the fuel consumed over a 10-year vehicle life as a function of

Table 3-1. BASIC PARAMETERS OF SYSTEMS ANALYZED FOR FUEL AND ENERGY CONSUMPTION AND LIFE CYCLE COST

Battery Type	Heat Engine Power Fraction	Battery Weight Fraction	Heat Engine Power (kw)	Motor Power (kw)	Vehicle Curb Wt. (kg)	Battery Weight (kg)	Vehicle* Manufacturing Cost ($1978)
Lead-acid	.6	.18	46.4	30.9	2129	383	5328
	.7	.14	52.7	22.6	2002	280	4965
	.8	.10	58.8	14.7	1889	189	4638
Nickel-zinc	.7	.10	49.5	21.2	1873	187	5293
	.8	.06	55.5	13.9	1774	106	4738
Nickel-iron	.6	.125	42.4	28.2	1934	242	5321
	.7	.10	49.5	21.2	1873	187	5012
	.8	.06	55.5	13.9	1774	106	4579
Reference Vehicle	1.0	0.0	96.6	0.0	1718	0.0	3823

* Includes first set of batteries at $[illegible]/kg for lead-acid, $5.25/kg for nickel-zinc, $3.75/kg for nickel-iron.

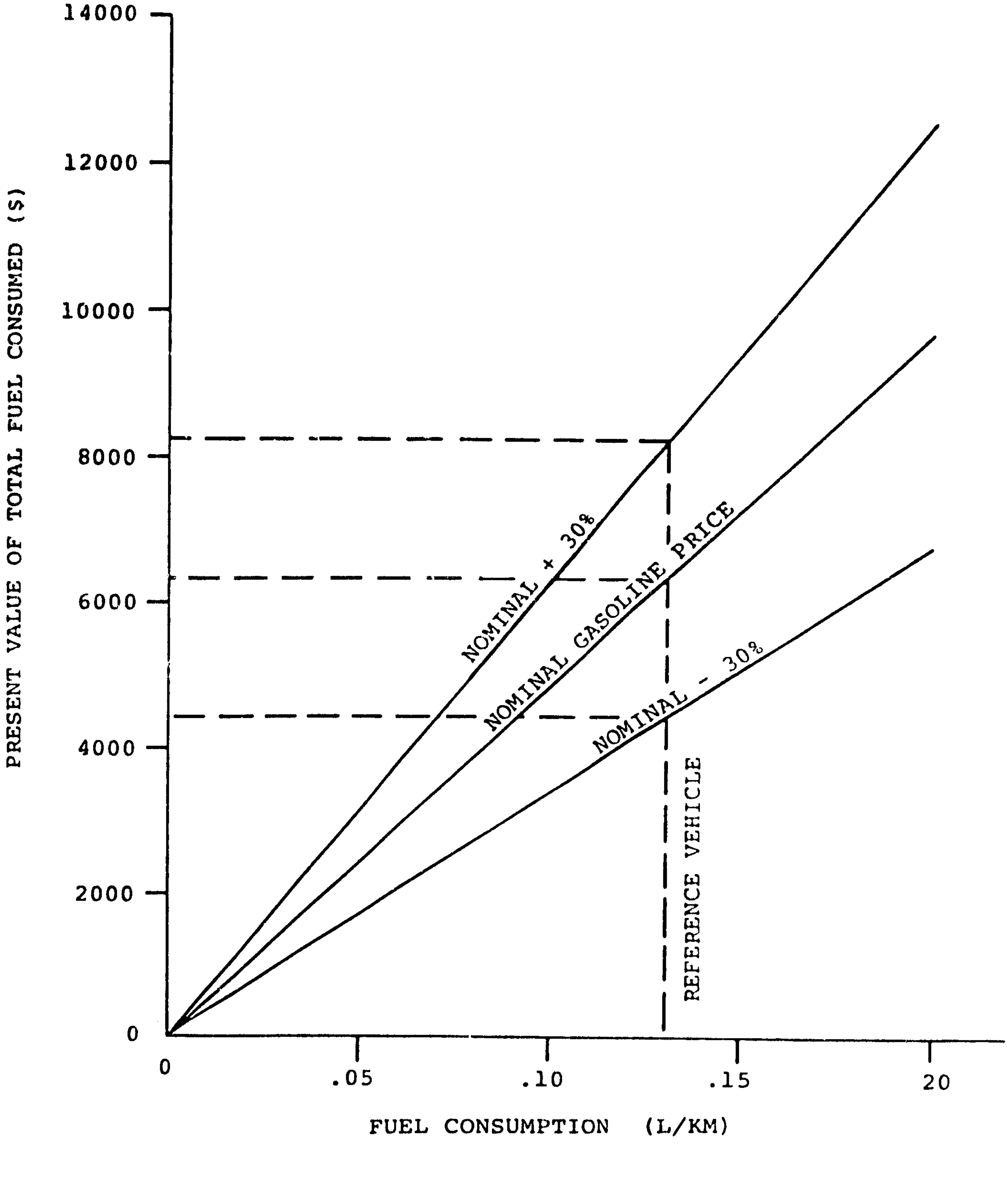


Figure 3-6 Lifetime Fuel Costs

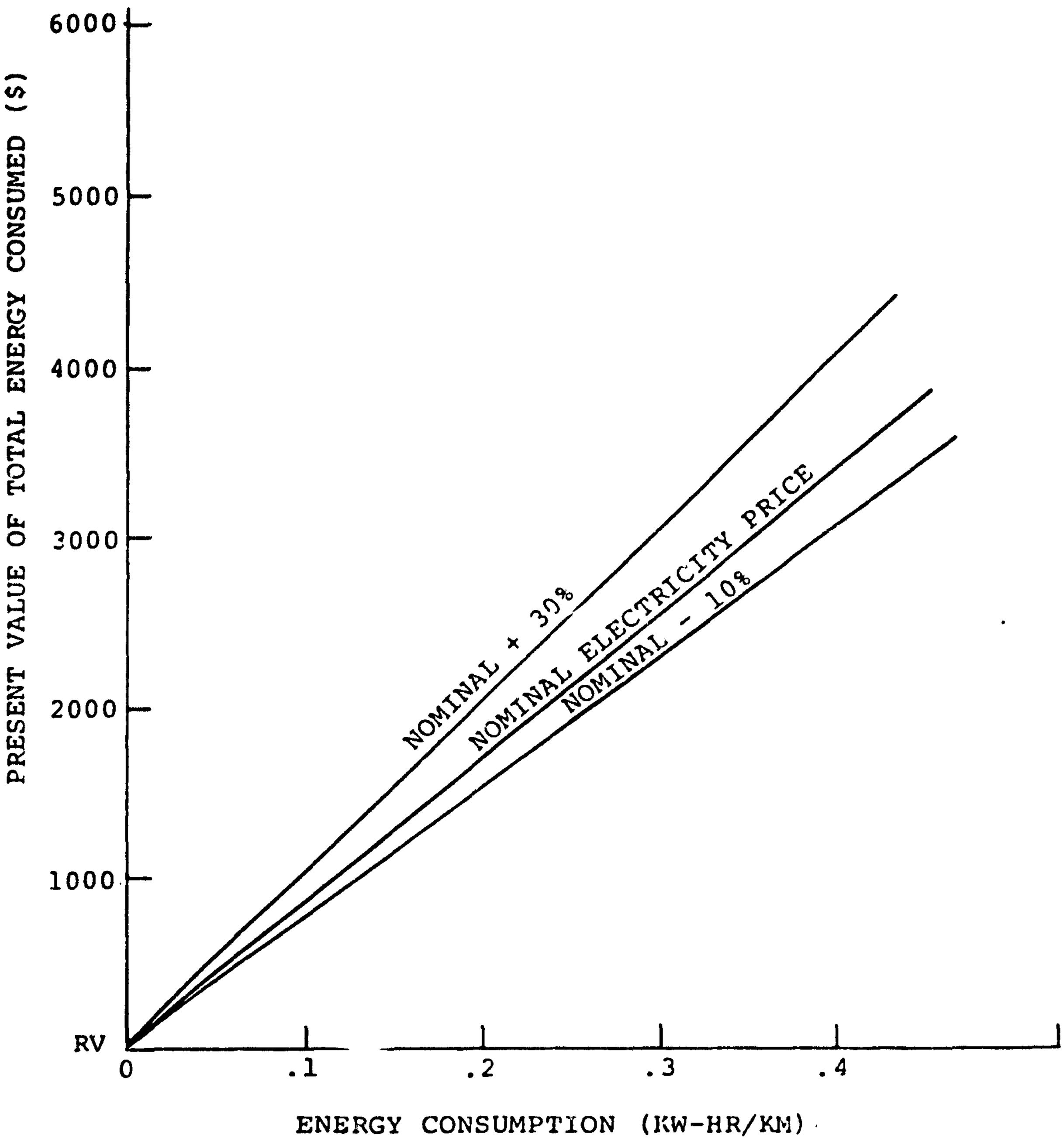


Figure 3-7 Lifetime Energy Costs

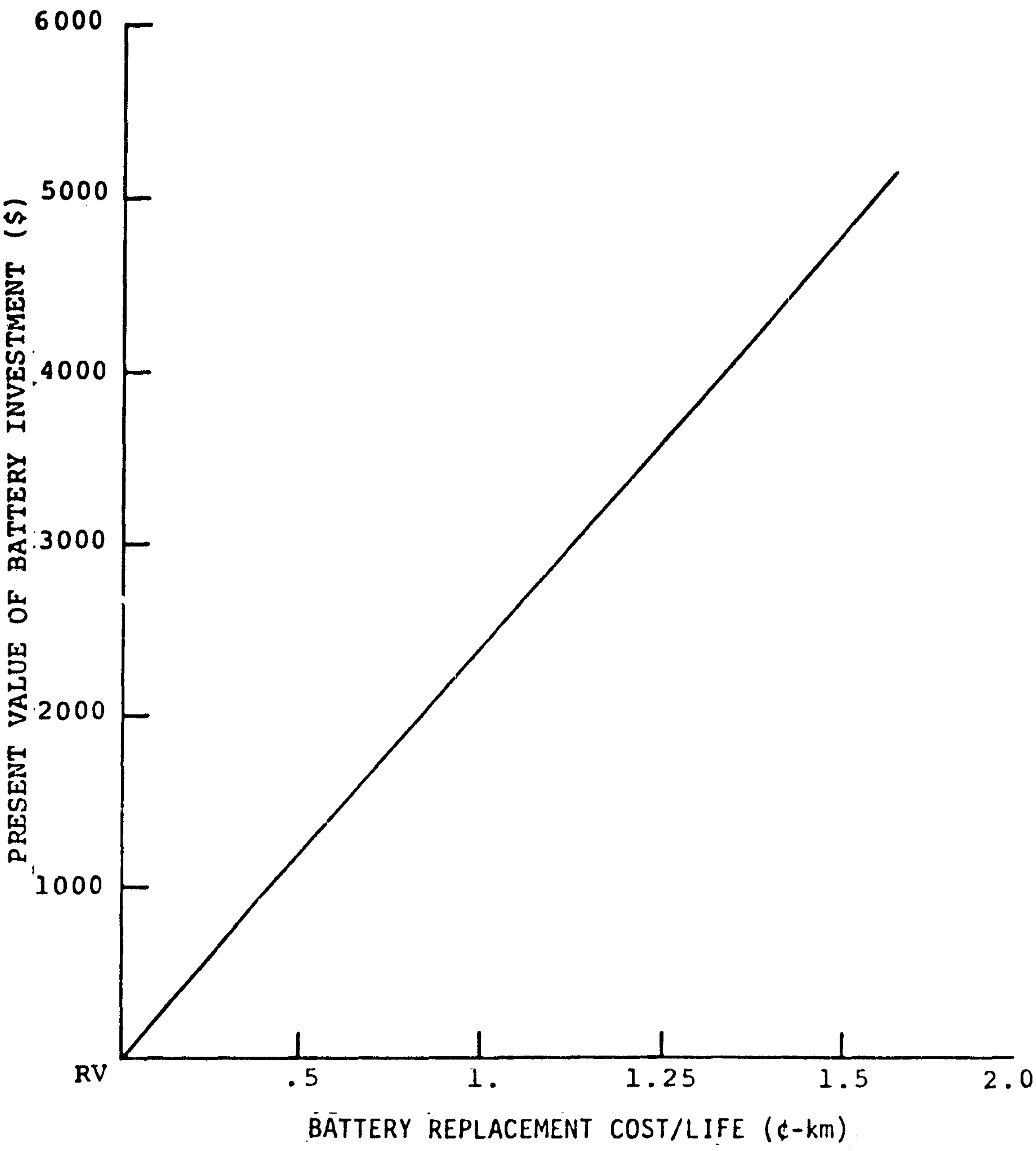


Figure 3-8 Lifetime Battery Costs

the vehicle's fuel consumption, the present value of the wall plug energy consumed as a function of the average energy consumption, and the present value of the total battery investment over the life of the car as a function of the ratio of battery replacement cost to battery life. From Figure 3-6, at the nominal, high, and low values of gasoline prices specified in the work statement, the present value of the gasoline consumed by the reference vehicle is, respectively, \$6300, \$8200, and \$4400. Neglecting for the purposes of this discussion any minor differences in maintenance cost, insurance, and so forth, these numbers give us the bounds within which all the following items must fit for the hybrid vehicle:

- Increment in retail price over the reference vehicle (not including batteries).
- Present value of fuel consumed.
- Present value of energy consumed.
- Present value of total battery investment (including first set).

Based on the previously mentioned ranges for fuel and energy consumption for a hybrid vehicle, let us take representative values of 20 km/l (47 mpg) and .2 kw-hr/km. This gives present values of \$2400 and \$1700 for the fuel and electricity, respectively, consumed during the life of the car, at nominal prices. In other words, at nominal fuel and electricity prices, the retail price increment (less batteries) plus the present value of the total battery investment must not exceed \$6300 - (2400 + 1700) = \$2200, if the life cycle cost of the hybrid is not to exceed that of the reference vehicle.

With gasoline prices 30% above nominal, the situation is little more favorable to the hybrid; then, we have a net \$8200 - (3100 + 1700) = \$3400 to play with for the retail price increment and total battery investment.

Returning to the nominal price case, the projected manufacturing costs, exclusive of the battery pack, ranged from \$4180 to \$4560, vs. the reference vehicle's \$3820 (numbers from Table 3-1, with the battery OEM cost subtracted from the vehicle manufacturing cost). Thus, the manufacturing cost increment for the vehicle itself ranges from \$360 to \$740 for the cases considered; let us use an average of \$550 as being representative. If we use the work statement guidelines of retail price = 2 x manufacturing cost, the corresponding retail price increment is \$1100, leaving us, in the nominal gasoline price case, with a total allowable battery investment on the order of \$1100. Looking at Figure 3-8, this corresponds to a ratio of battery replacement cost to battery life of about 0.5¢/km. Referring back to Table 3-1, take as being representative a lead-acid battery pack weighing, say, 300 kg with an OEM cost of \$600 (\$2/kg). If we continue to assume a factor of 2 to obtain battery retail price, we are up to \$1200 to replace the battery pack and, based on .5¢/km, a life of 240,000 km would be required from the battery pack, which is obviously a little too much to hope for. On the other hand, if we assume that both the manufacturing cost increment of the hybrid and the battery OEM cost was passed on at a minimum level of 1.25 x (manufacturing or OEM) cost, the numbers become as follows:

Representative retail price increment = 1.25 x 550 = \$668

Allowable battery investment = 2200 - 688 = 1512

Battery replacement cost/life = .63¢/km

Retail price of 300 kg battery pack = \$750

Required battery life = 119,000 km

This battery life is probably still somewhat outside the realm of reality, but at least it is getting closer. It must also be emphasized that the foregoing is based on assumptions extremely favorable to the hybrid; i.e., a 10-year life, with the assumptions provided by JPL relating to annual vehicle travel, which result in a high total mileage of about 220,000 km, or 137,000 miles. This, in turn, means that fuel costs are more heavily weighted relative to the initial investment than they would be for the average driver who keeps his car on the order of five years and who makes the initial purchasing decision that gets the car into the fleet.

The point of the foregoing discussion is not to degrade JPL's requirement to achieve a life cycle cost for the hybrid which is no higher than that of the reference vehicle; this is obviously a desirable goal. However, given the realities of near term battery technology and the battery life characteristics likely to be provided, its achievement is unlikely under the pricing assumptions specified in the Assumptions and Guidelines provided by JPL[6], unless the battery/electric portion of the drivetrain becomes the equivalent of a vermiform appendix. It is primarily for this reason that we adopted the approach, described in Section 2 of this report, for localizing the range of the basic parameters $\overline{P}_{HE}$ and $\overline{W}_B$, rather than simply picking values which give a life cycle cost equal to that of the reference vehicle. As we have pointed out in the Task 1 report,(1) the question of retail pricing of a hybrid (or any car which has substantially better fuel economy than a conventional car of the same inertia weight) is much more complex than simply applying a factor of 2 to the manufacturing cost. Application of such a simplistic formula, together with rigid adherence to the requirement to obtain a life cycle cost equal to the reference vehicle, does not take into account two realities:

- A car with extremely high fuel economy, such as a hybrid, has value to a manufacturer in terms of its ability to help him meet CAFE requirements and still have a reasonable and highly profitable product mix. Consequently, there is incentive to keep the retail price increment on such a vehicle to the minimum value which will cover the manufacturing cost increment associated with the improvement in fuel economy. The reference gasoline or diesel powered car is already at the upper spectrum of profitability.
- The perceived value of a highly fuel efficient vehicle to a consumer is not based on a computation of life cycle costs (as attested to by the number of individuals buying Oldsmobile and, in a different class, Mercedes, diesels. Few of these people will keep these cars long enough to realize any net cost benefits.).

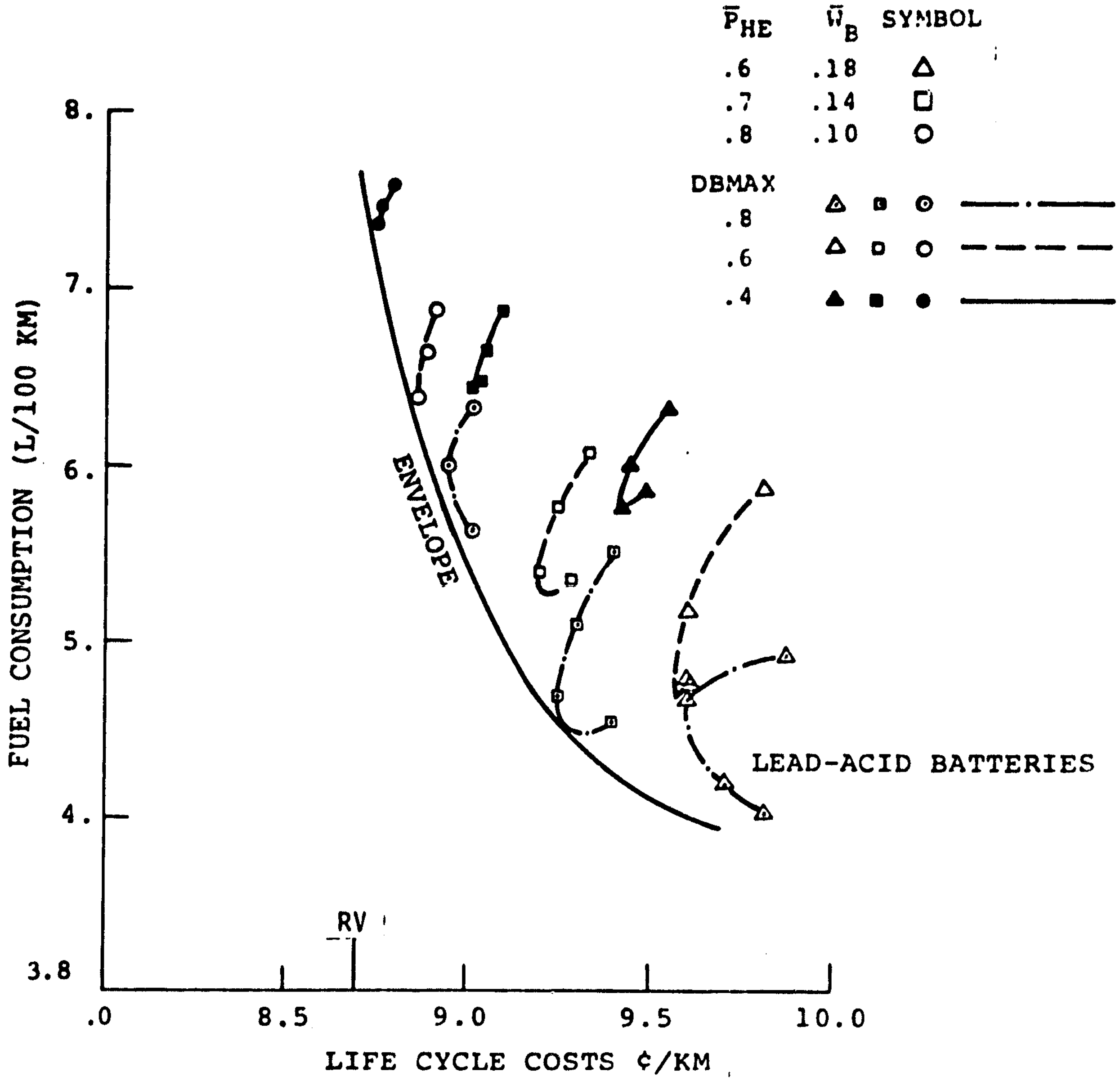
A more complete discussion of hybrid pricing strategy, and the sensitivity of the market to price, will be found in Section 3.6.3;

however, for the purposes of these system level tradeoffs, life cycle costs were computed on two bases, which should provide upper and lower boundaries for the real situation. The first (nominal) case corresponds to the assumptions provided by JPL; i.e., retail price = 2 x manufacturing cost in all cases, and retail price of replacement batteries = 2 x OEM cost. The second case corresponds to the manufacturer adding the minimum possible retail price increment to cover the added manufacturing costs of the hybrid over the reference vehicle. If the hybrid system components are bought on an OEM basis and do not require substantial capital investment by the vehicle manufacturer, the minimum retail price increment corresponds to about 1.25 times the manufacturing cost increment.[6] Consequently, life cycle costs for the second case were computed on the following basis:

Retail price (hybrid) = 2 x manufacturing cost (reference vehicle) + 1.25 x Δ manufacturing cost (hybrid over reference vehicle)

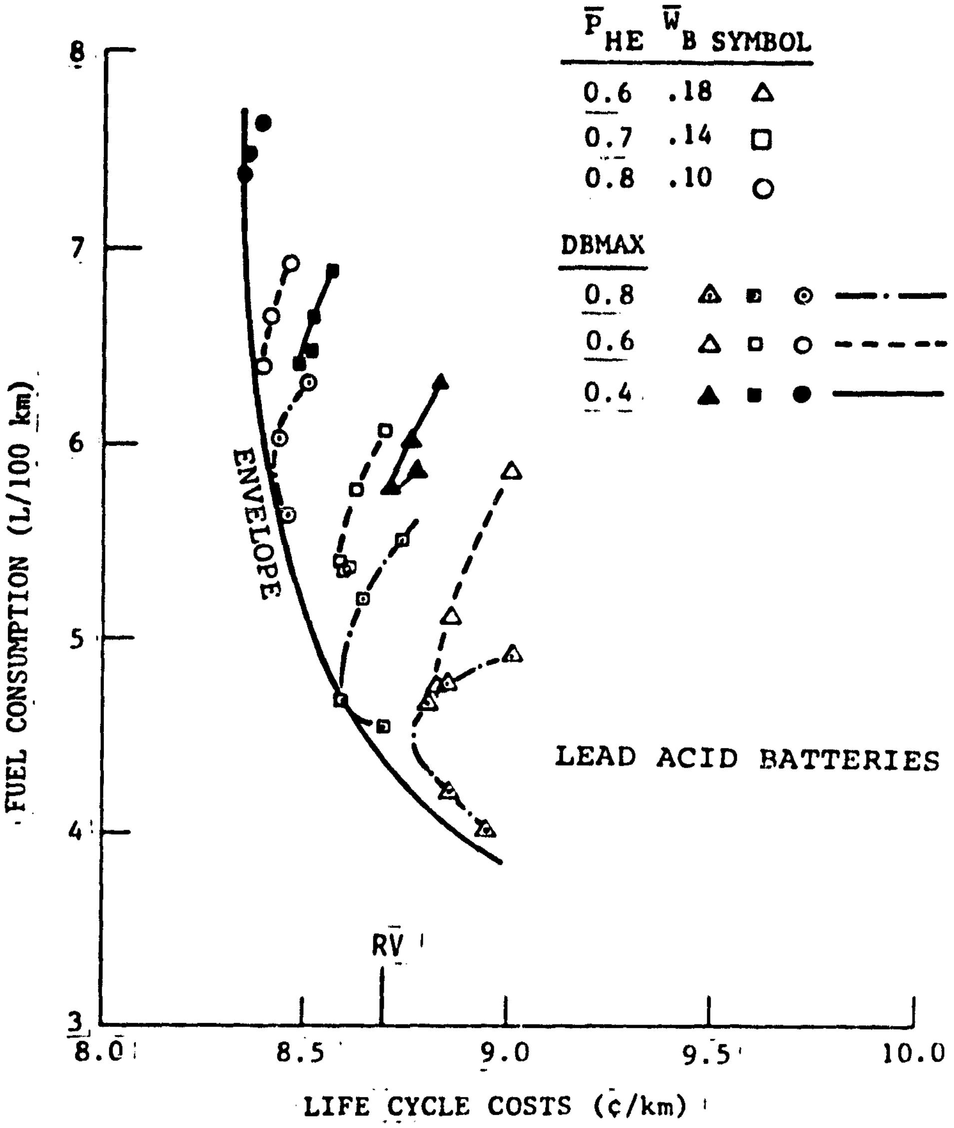
Battery replacement cost = 1.25 x battery OEM cost

Plots of life cycle costs vs. fuel consumption are given in Figure 3-9 for the nominal cost case and in Figure 3-10 for the minimum cost increment case, for lead-acid batteries. The corresponding plots for nickel-zinc batteries and nickel-iron batteries are given in Figures 3-11 through 3-14. The individual curves plotted in these figures show the variation of fuel consumption and life cycle cost as the control parameter P_{EOMIN} is varied, for a fixed combination of basic parameters ($\overline{P}_{HE}$, $\overline{W}_B$) and a fixed battery discharge limit (D_{BMAX}). Note that lower life cycle costs are



NOMINAL COST CASE-RETAIL PRICE-2xMFG COST
BATTERY REPLACEMENT PRICE-2xOEM COST

Figure 3-9 Hybrid Vehicle Life Cycle Cost Vs. Fuel Consumption



MINIMUM COST CASE - RETAIL PRICE = 2 x MFG COST (Ref. Vehicle) +1.25 x Δ MFG COST (Hybrid - Ref Vehicle)

BATTERY REPLACEMENT PRICE = 1.25 x OEM COST

Figure 3-10 Hybrid Vehicle Life Cycle Cost vs Fuel Consumption

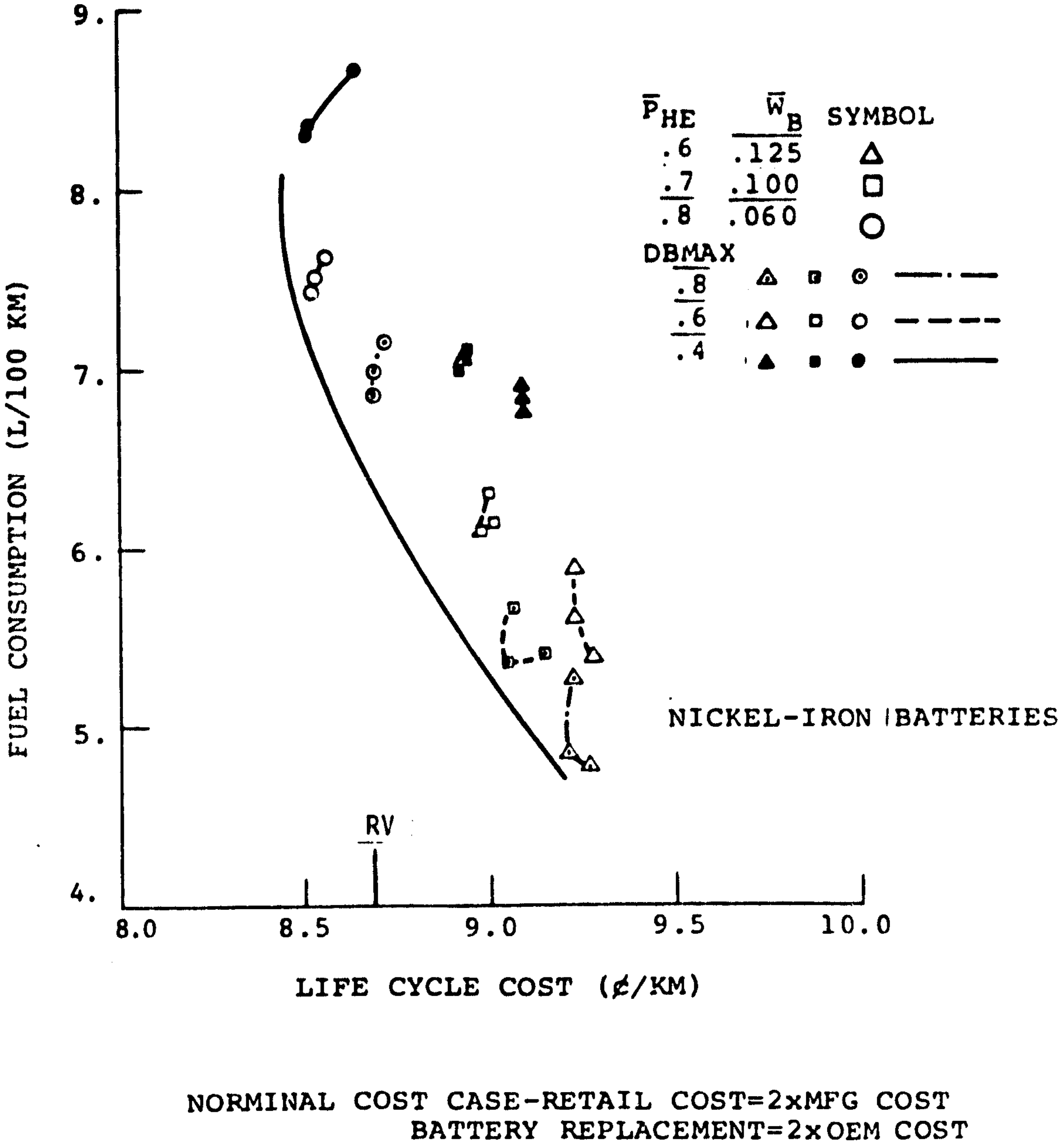
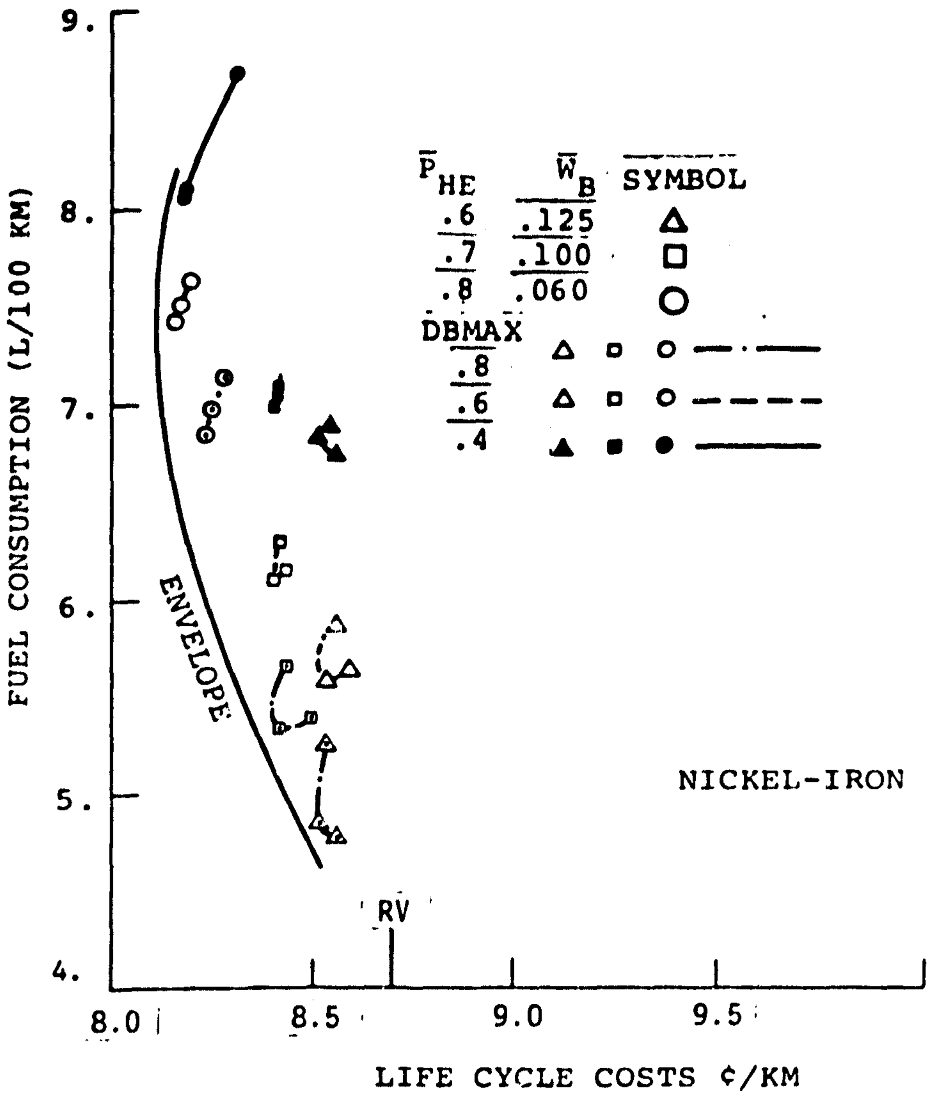
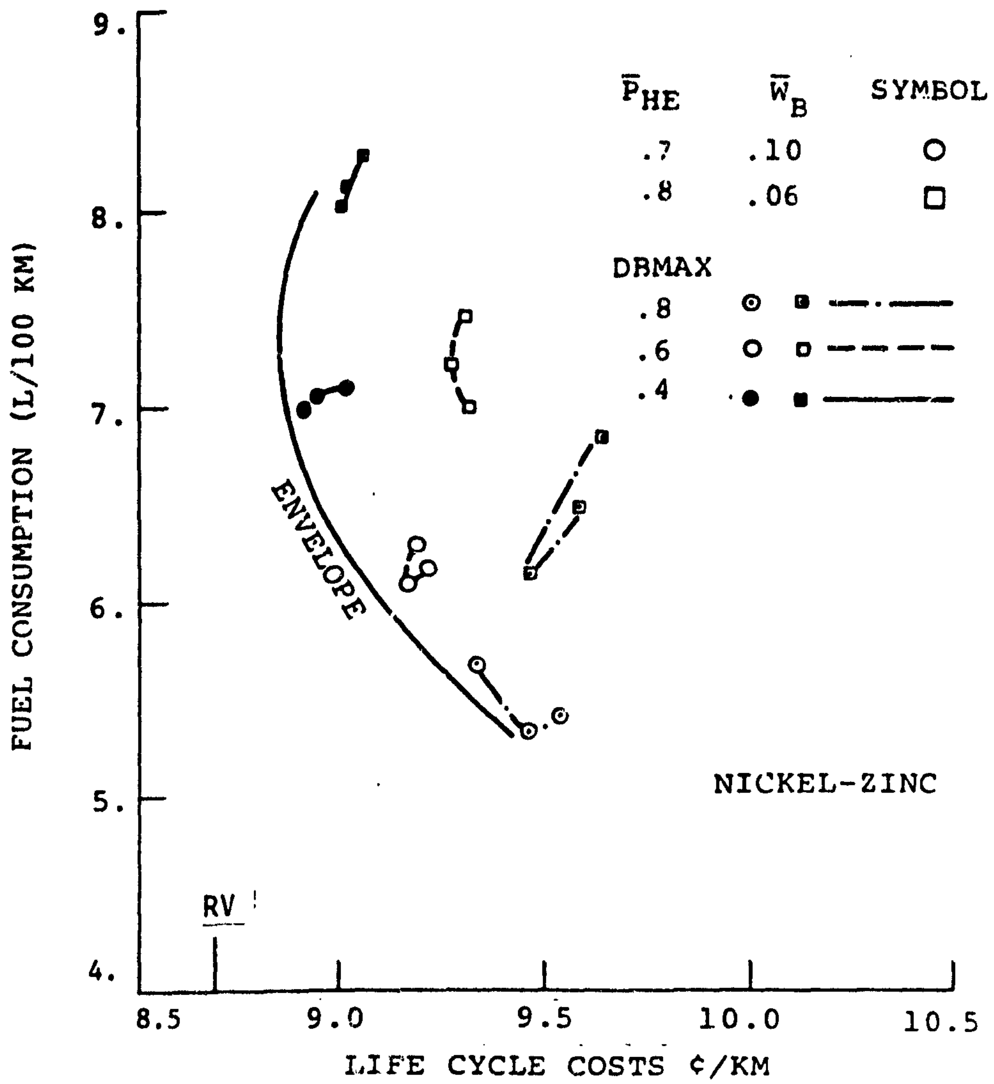


Figure 3-11 Hybrid Vehicle Life Cycle Cost Vs. Fuel Consumption



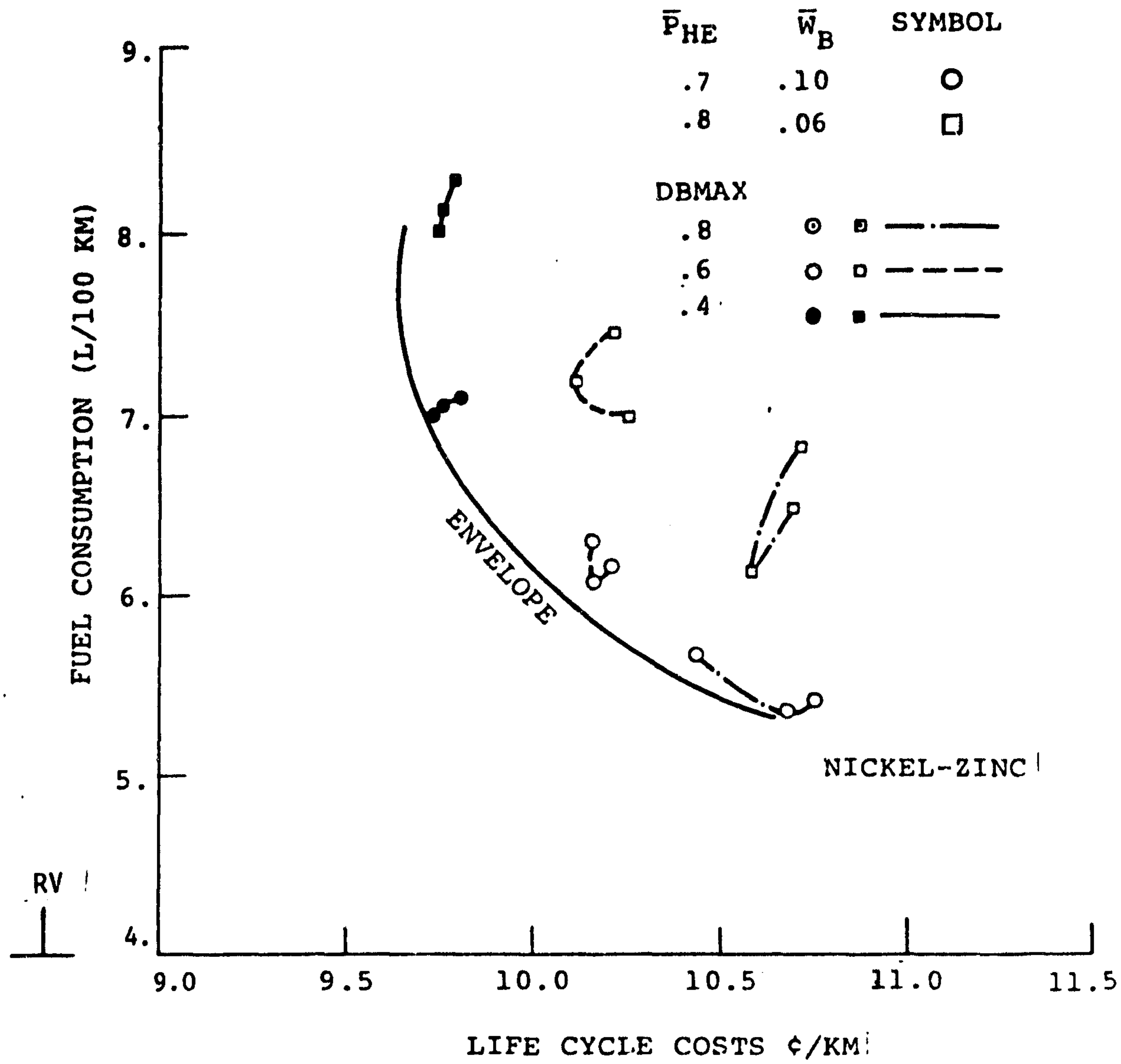
MINIMUM COST CASE-RETAIL PRICE-2xMFG COST (REF.VEHICLE)
+1.25xΔCOST(HYBRID-REF.VEHICLE)
BATTERY REPLACEMENT=1.25xOEM COST

Figure 3-12 Hybrid Vehicle Life Cycle Cost Vs. Fuel Consumption



MINIMUM COST CASE - RETAIL PRICE=2xMFG COST (REF. VEHICLE)
+1.25xΔCOST (HYBRID-REF. VEHICLE)
BATTERY REPLACEMENT COST=1.25xOEM COST

Figure 3-13 Hybrid Vehicle Life Cycle Cost Vs. Fuel Consumption



NOMINAL COST CASE - RETAIL PRICE=2xMFG COST
BATTERY REPLACEMENT=2xOEM COST

Figure 3-14 Hybrid Vehicle Life Cycle Cost Vs. Fuel Consumption

favored by using a larger heat engine power fraction and smaller battery weight, and by not discharging the battery pack too deeply. Low fuel consumption, on the other hand, is favored by the reverse - smaller heat engine, larger battery, deeper discharge.

The approximate envelopes plotted in Figures 3-9 to 3-14 represent the locus of points corresponding to the best attainable combinations of fuel consumption and life cycle cost; in other words, points to the left of these envelopes are unrealizable under the constraints and assumptions on which the fuel consumption and life cycle cost analyses are based. In Figures 3-9 and 3-10 (lead-acid cases), it is evident that the envelope has a 'knee' to the right of which life cycle cost goes up more rapidly than the reduction in fuel consumption, and to the left of which fuel consumption goes up rapidly without much reduction in life cycle cost. This is perhaps most evident in the minimum cost increment case, Figure 3-10. For lead-acid batteries, the cases which are grouped in the vicinity of this knee are those for heat engine power fractions of .6 and .7, with battery discharge limits between .6 and .8.

In studying these curves, it should be noted that the projected life cycle cost for the reference vehicle is 8.7¢/km at nominal gasoline prices. This is to the left of the knee of the curve for the nominal price cost, and just about at the knee for the minimum cost case. In view of the preliminary nature of these studies, and high priority placed in the Near Term Hybrid Vehicle Program on minimizing fuel consumption, we selected a heat engine power fraction of about .65 as a starting point for the subsequent detailed tradeoffs, rather than the right hand end of the .6 to .7 interval. The associated battery weight fraction should be about .17.

The curves for nickel-iron (Figures 3-11, 3-12) show a little different behavior than for lead-acid. For one thing, the goal of attaining a life cycle cost competitive with the reference vehicle appears to be more nearly attainable. The second area of difference is that the fuel consumption rises much more steeply with respect to life cycle cost as the heat engine power fraction is increased, than is the case with lead-acid; in fact, it appears that the envelope might even have a minimum with respect to life cycle cost. Because of the steepness of the curve, forcing the heat engine fraction to high values does not buy too much in terms of lower life cycle costs. Consequently, we would choose a heat power fraction only slightly higher than that used with lead-acid batteries, primarily to keep the retail price increment down to something reasonable.

Nickel-zinc batteries (Figures 3-13, 3-14) show life cycle costs considerably higher than the others, even at high values of heat engine power fraction. Although the battery weight fractions are low enough to keep the manufacturing cost within the constraints described earlier, the fundamental problem with life cycle costs is the frequent replacement of the battery pack (half the life of lead-acid, and less than a third that of nickel-iron).

Based on the above, the three battery types were ranked as follows:

1. Nickel-iron

2. Lead-acid

3. Nickel-zinc

Because of the preliminary and rough nature of the system level studies, we did not feel that this was yet the time to totally exclude any battery type, and all three types were carried forward into the next level of tradeoff studies. Although the nickel-iron systems appear to have advantages in terms of lower life cycle cost, we elected to use lead-acid batteries for the construction of a hypothetical baseline system due to the fact that the technology is more developed and the batteries better characterized. The other two batteries were later investigated in terms of their relation to this baseline system, as will be discussed in Section 3.4.

3.1.3 Sensitivity Considerations

In this section, we shall discuss the impact of variations in the assumptions regarding travel distribution, number of vehicles, and fuel and electricity prices, which underly the foregoing analysis. The variational boundaries were as specified by JPL in the Task 4 (Sensitivity Studies) work statement. Before proceeding to discussing the sensitivity of the system level tradeoffs to these variations, however, we shall first define their effect on the results of Task 1.

Sensitivity of Task 1 Results

S1. Sensitivity boundaries of +7% and -7% applied to the number of passenger cars in 1985 produced the following results on fuel consumption. (Refer to Mission Analysis and Performance Specification Studies Report pages 45, 46, and 76 for fuel consumption using nominal number of passenger cars). The results shown should be compared directly to Tables 2-22 and 2-23 of the Task 1 report; they represent a simple increase or decrease in the amount of fuel used in each mission classification. We see no reason to assume that a change in the total number of vehicles would result in a change in the distribution of the mission classifications.

TABLE 2.22 - Distribution of Fuel Consumed by Reference Vehicles in 1985 Fleet (+7%)

Mission		Fuel Consumption (Gal. x 10^{-6})		
Usage	Vehicle Size	Cars at Single Family Units	Multi-Family Units	TOTAL
Secondary	Tight	4740	1038	5,778
	Roomy	5436	1188	6,624
Only	Tight	6698	3713	10,411
	Roomy	7009	3820	10,829
Primary	Tight	8646	2161	10,807
	Roomy	9223	2297	11,470

TABLE 2.22 - Distribution of Fuel Consumed by Reference Vehicles in 1985 Fleet (-7%)

Mission		Fuel Consumption (Gal. x 10^{-6})		
Usage	Vehicle Size	Cars at Single Family Units	Multi-Family Units	TOTAL
Secondary	Tight	4120	902	5,022
	Roomy	4724	1032	5,756
Only	Tight	5822	3227	9,049
	Roomy	6092	3320	9,412
Primary	Tight	7514	1879	9,393
	Roomy	8017	1953	9,970

TABLE 2.23 - Distribution of Fuel Consumed by Reference Vehicles in 1985 Fleet with Off-Street Parking (+7%)

Mission		Fuel Consumption (Gal. x 10^{-6})		
Usage	Vehicle Size	Cars at Single Family Units	Multi-Family Units	TOTAL
Secondary	Tight	<3702	963	<4665
	Roomy	>4248	1070	>5318
Only	Tight	<5083	3467	<8550
	Roomy	>5339	3585	>8924
Primary	Tight	<6741	1990	<8731
	Roomy	>7212	2012	>9224
		Total 'Tight'		21,946
		Total 'Roomy'		23,466

TABLE 2.23 - Distribution of Fuel Consumed by Reference Vehicles in 1985 Fleet with Off-Street Parking (-7%)

Mission		Fuel Consumption (Gal. x 10^{-6})		
Usage	Vehicle Size	Cars at Single Family Units	Multi-Family Units	TOTAL
Secondary	Tight	3218	837	4055
	Roomy	3692	930	4622
Only	Tight	4418	3013	7431
	Roomy	4641	3116	7757
Primary	Tight	5859	1730	7589
	Roomy	6268	1748	8016
		Total 'Tight'		19,075
		Total 'Roomy'		20,0395

M8 - Estimated Fuel Consumption of Mission Performed Entirely by Reference Vehicles: (+7%)

28890×10^6 gal. (total)

23433×10^6 gal. (vehicles potentially replaceable by hybrids)

M8 - Estimated Fuel Consumption of Mission Performed Entirely by Reference Vehicles: (-7%)

25110×10^6 gal. (total)

20367×10^6 gal. (vehicles potentially replaceable by hybrids)

S2. Sensitivity of mission analysis to +7% and -7% change in average annual vehicle kilometers traveled per car (1985) are given in Table M1. Sensitivity of life cycle costs of reference vehicle to indicated

TABLE H1

DAILY TRAVEL (KM)	-7% FRACTION OF TOTAL DRIVING Only	+7% FRACTION OF TOTAL DRIVING Only
0-20	.0851	.0673
20-30	.0656	.0524
30-40	.0733	.0660
40-50	.0864	.0675
50-60	.0728	.0688
60-70	.0688	.0650
70-80	.0579	.0592
80-90	.0562	.0523
90-100	.0608	.0530
100-120	.0971	.0847
120-140	.0622	.0825
140-160	.0463	.0606
160-180	.0187	.0458
180-200	.0461	.0329
200-220	.0232	.0242
220-240	.0152	.0221
240-260	.0662	.0192
260-280		.0260
280-300		.0502
300-320		

changes in average annual vehicle kilometers traveled are:

	Relative change in discounted life cycle cost
10 year life cycle +7%	-3.6%
10 year life cycle -7%	+4.8%
7 year life cycle +7%	-3.0%
7 year life cycle -7%	+4.1%

The change in travel distribution associated with the variation in annual kilomage is not enough to effect the fuel economy to any extent which would be significant within the accuracy of this study. Consequently, on terms of the effect on the fuel consumption for the various mission classifications, it would be the same as for the same variation in the total number of vehicles in the fleet; i.e., a 7% increase in annual vehicle kilomage has the same effect as a 7% increase in the vehicle fleet.

S3. Changes in life cycle costs of reference vehicle caused by +30% and -30% changes in gasoline and diesel prices over a 10 year life cycle are:

	Discounted life cycle costs (per km)
Nominal prices +30%	+9%
Nominal prices -30%	-9%

None of these variations has any effect on the selection of the reference vehicle or the development of specifications for the hybrid vehicle.

Sensitivity of System Level Tradeoffs

It can be concluded at this point that the sensitivity variables of annual vehicle kilomage and number of vehicles in the fleet have no relevance to the design tradeoffs, at least within the ±7% boundary values specified by JPL. As must be clear by now, the basic design decisions (selection of a heat engine power fraction and battery weight fraction) are by no means precise. The best the system-level studies could do was to provide an indication of what would be an appropriate range of design parameter values to pick, within fairly narrow limits. Selecting specific values to design around was then a matter of exercising reasonable engineering judgement, and it is impossible to quantify the effect of a ±7% change in the first two sensitivity variables on such a judgement. Consequently, in this and subsequent discussions regarding sensitivity studies, we shall consider only the sensitivity variables of fuel and electricity prices.

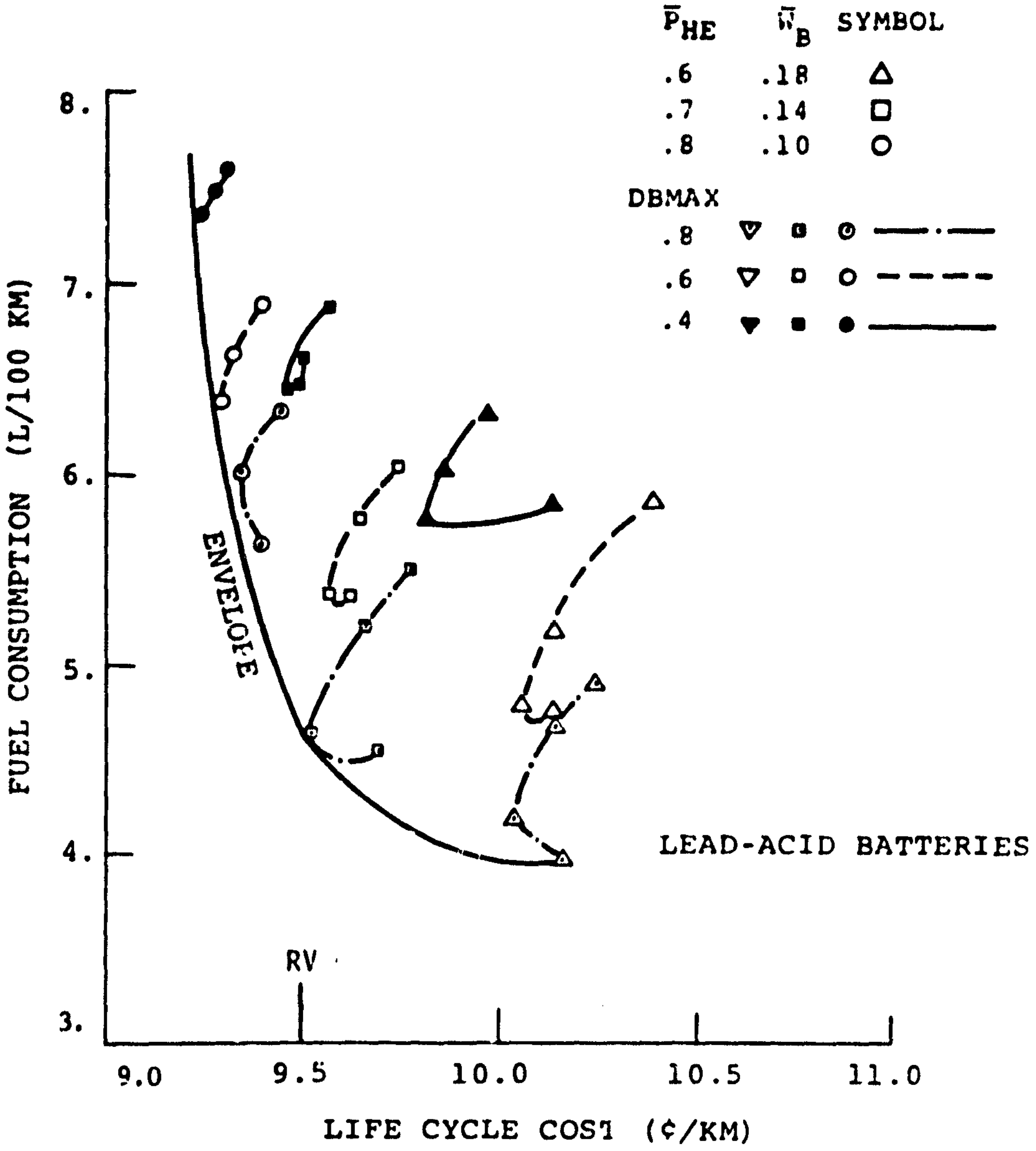
To determine whether variations in these prices would have any effect on the basic design decisions, the same plots of fuel consumption vs. life cycle costs were made for the sensitivity boundary values of ±30% on fuel prices and +30%, -10% on electricity prices. This was done for the case of lead-acid batteries, at both the nominal price level (retail price =2x manufacturing cost) and the minimum price level. The results are shown in Figures 3-15 through 3-18. In examining these curves, keep in mind that the reference vehicle life cycle costs are:

- Nominal fuel - 8.7¢/km
- Fuel +30% - 9.5¢/km
- Fuel -30% - 7.8¢/km

The reference vehicle life cycle cost is indicated by a tick mark on the life cycle cost axis.

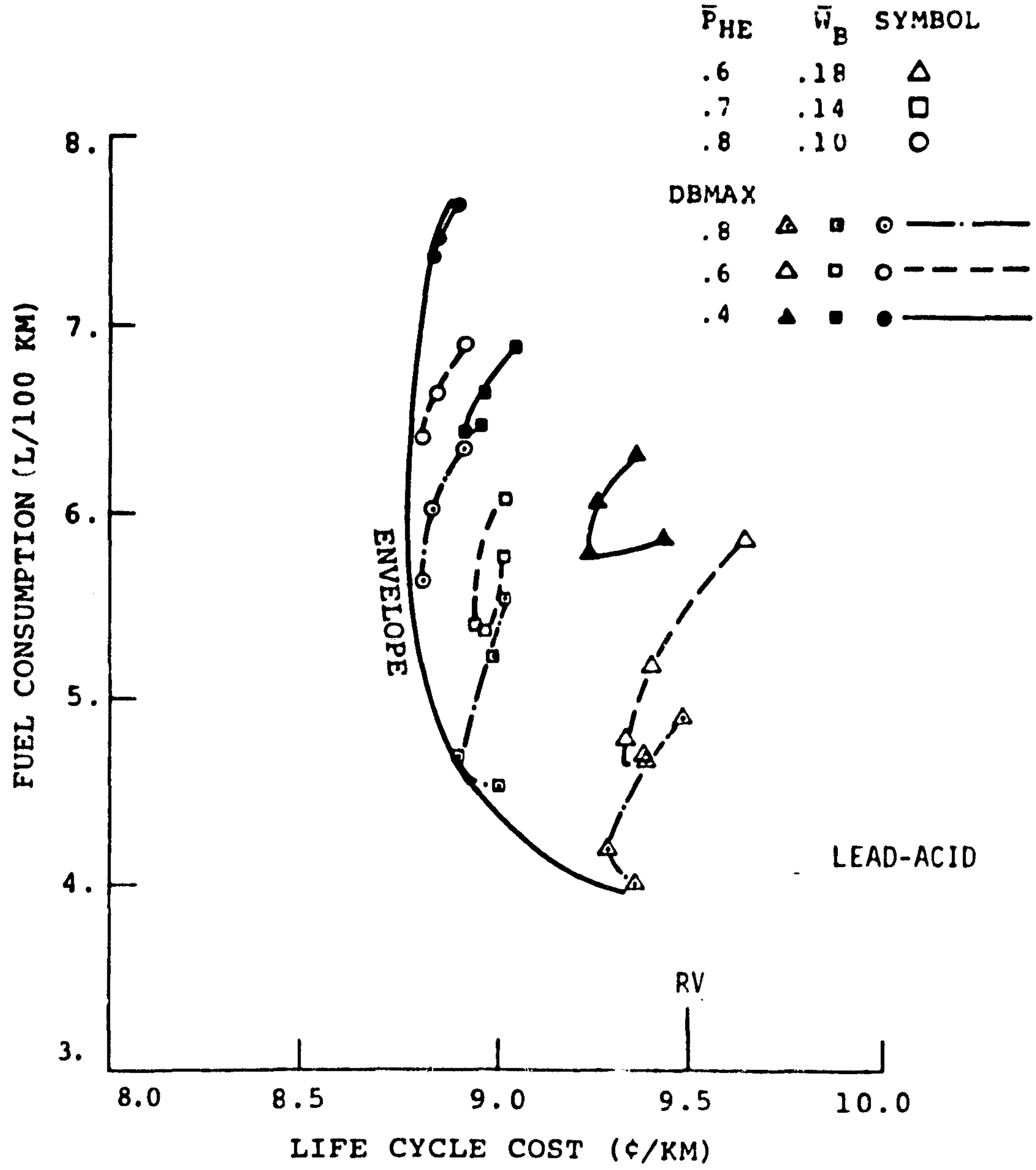
Increasing fuel prices 30% (Figures 3-15 a, b) tends to make the knee of the fuel consumption vs. life cycle cost curve more distinct (compared with Figures 3-9 and 3-10). The minimum cost case also shows more of a tendency to achieve an actual minimum of life cycle cost, similar to the nickel-iron nominal fuel cost, minimum price case. Decreasing gasoline prices, on the other hand, 'softens' the envelope. With fuel at +30%, the decision to pick a heat engine power fraction of .65 appears to be quite reasonable, making the system competitive with the reference vehicle even in the nominal price case. For the minimum price case, an even lower heat engine power fraction might be justified. With fuel at -30% (Figures 3-16 a, b), .65 does not look so good in terms of achieving a life cycle cost which is reasonably close to the minimum attainable. Consequently, gasoline prices at this level would tend to drive us toward a heat engine power fraction of at least .7. However, in view of the current gas situation, the probability of the fuel pricing assumptions being high by this amount seems highly unlikely.

High electricity prices (Figures 3-17 a, b) have a similar effect as low gasoline prices in terms of 'softening' the curves. However, because the lifetime electricity costs for the hybrid are less than its fuel costs, and because raising them does not influence the life cycle cost of the reference vehicle, the overall effect is less and does not remove the hybrid out of competition with the reference vehicle in terms of life cycle cost, at least for the minimum cost case. Such an increase in electricity price would tend, though,to push the heat engine power fraction toward .7.



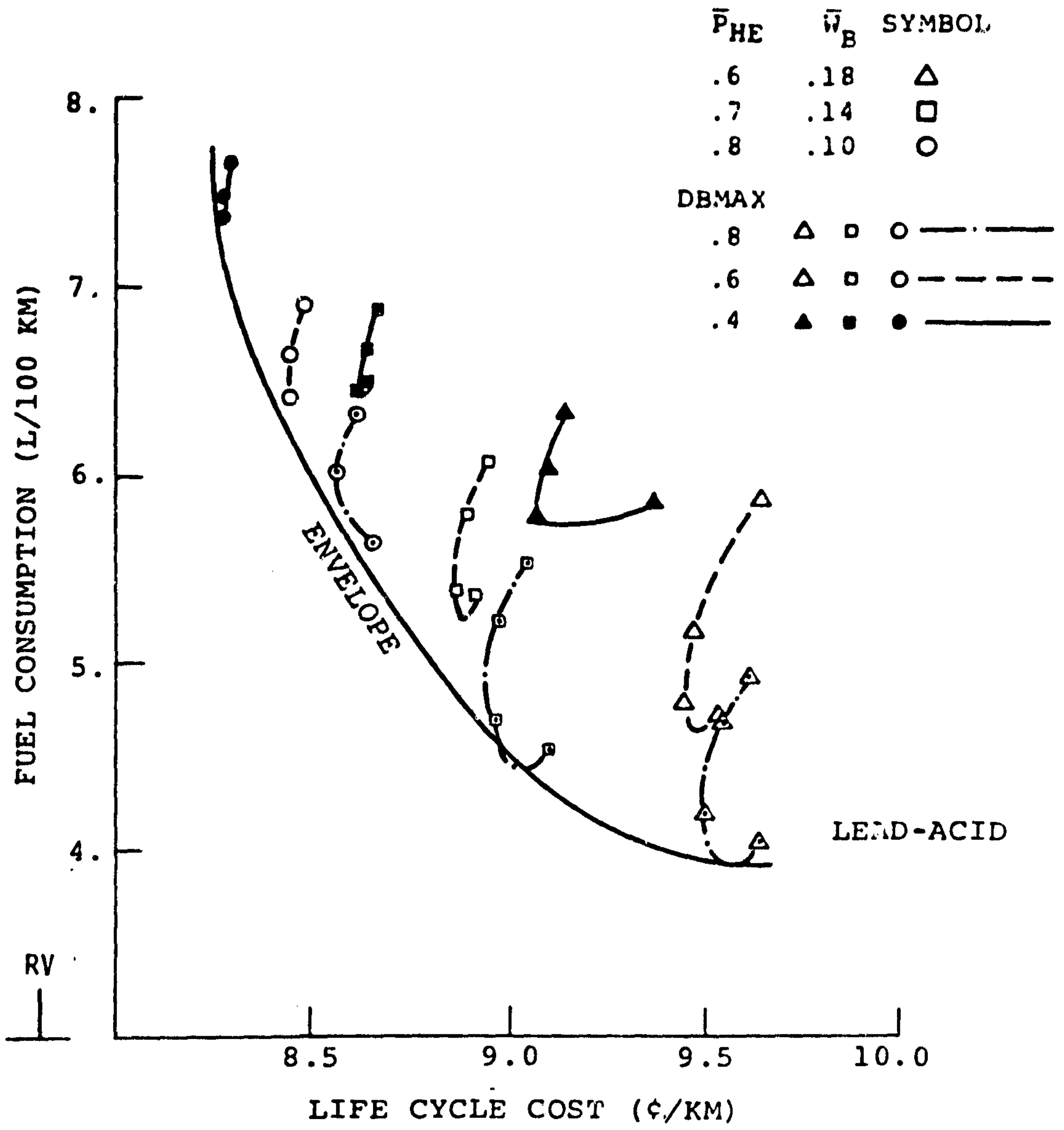
NOMINAL COST CASE-RETAIL PRICE-2xMFG COST
BATTERY REPLACEMENT-2xOEM COST
30% INCREASE IN GASOLINE PRICES

Figure 3-15a Hybrid Vehicle Life Cycle Cost Vs. Fuel Consumption



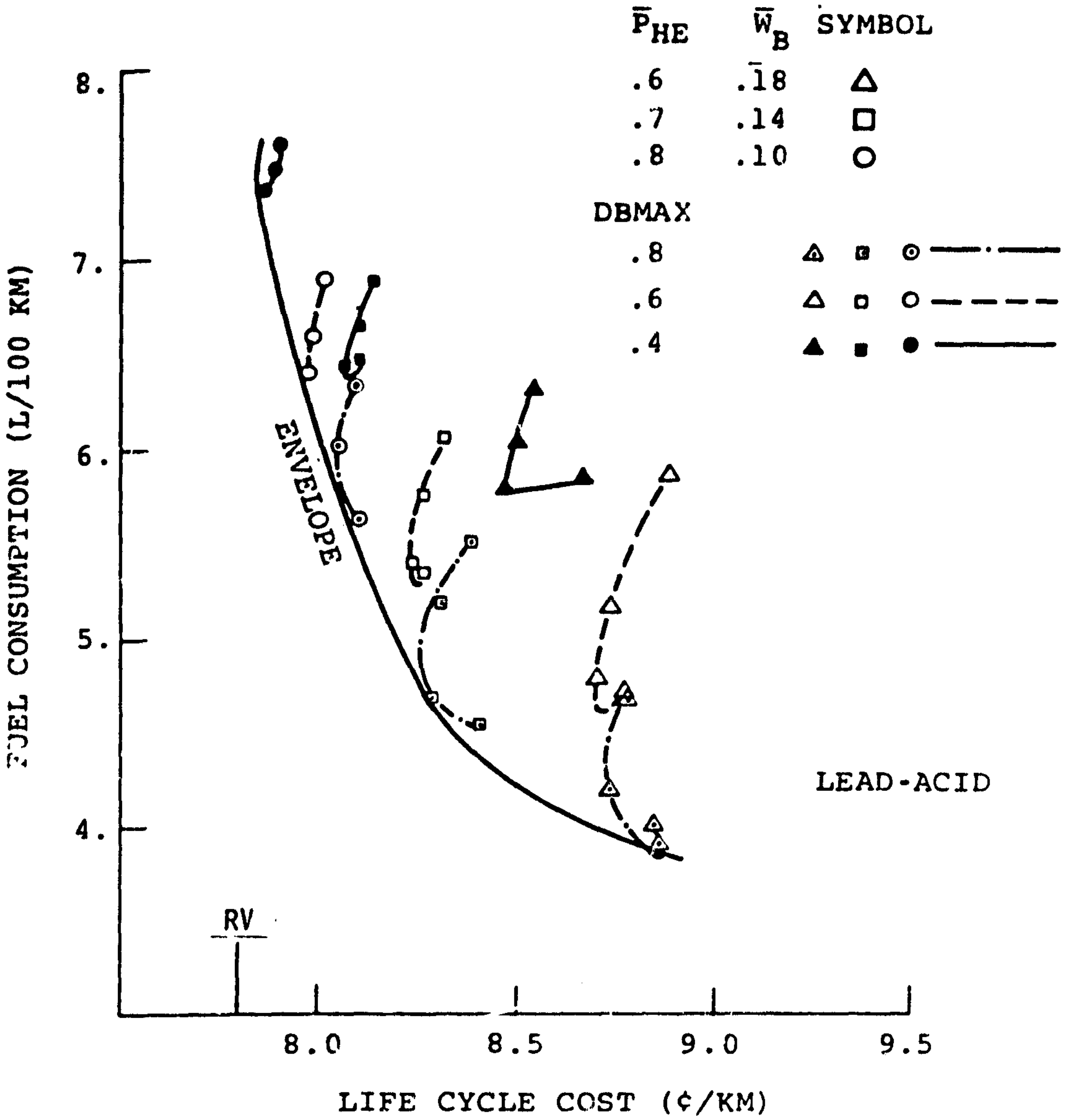
MINIMUM COST CASE-RETAIL PRICE=2xMFG (REF.VEHICLE)
+1.25xΔCOST(HYBRID-REF.VEHICLE)
BATTERY REPLACEMENT=1.25xOEM COST
30% INCREASE IN GASOLINE PRICES

Figure 3-15b Hybrid Vehicle Life Cycle Cost Vs. Fuel Consumption



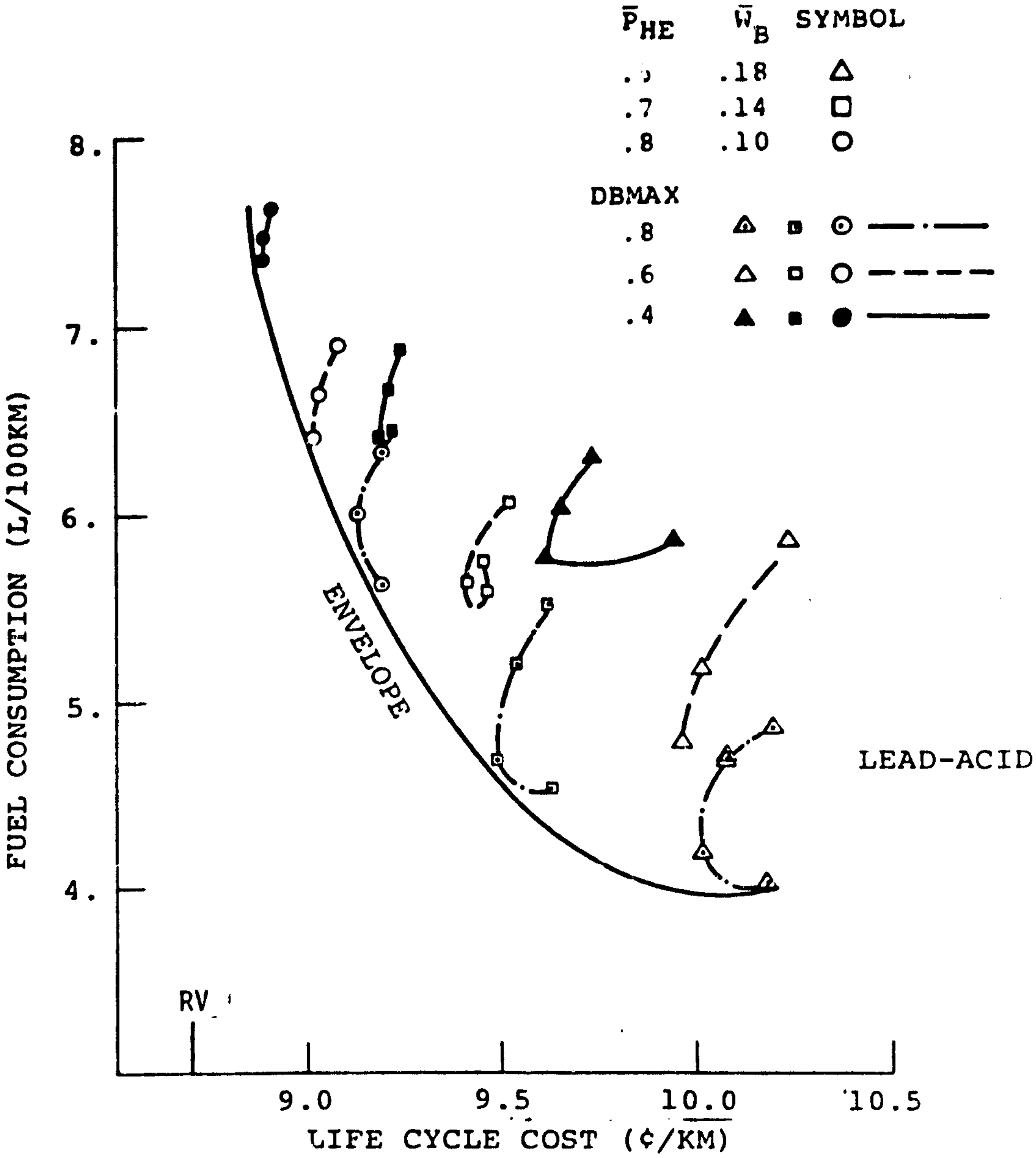
NOMINAL COST CASE-RETAIL PRICE=2xMFG COST
BATTERY REPLACEMENT=2xOEM COST
30% DECREASE IN GASOLINE PRICES

Figure 3-16a Hybrid Vehicle Life Cycle Cost Vs. Fuel Consumption



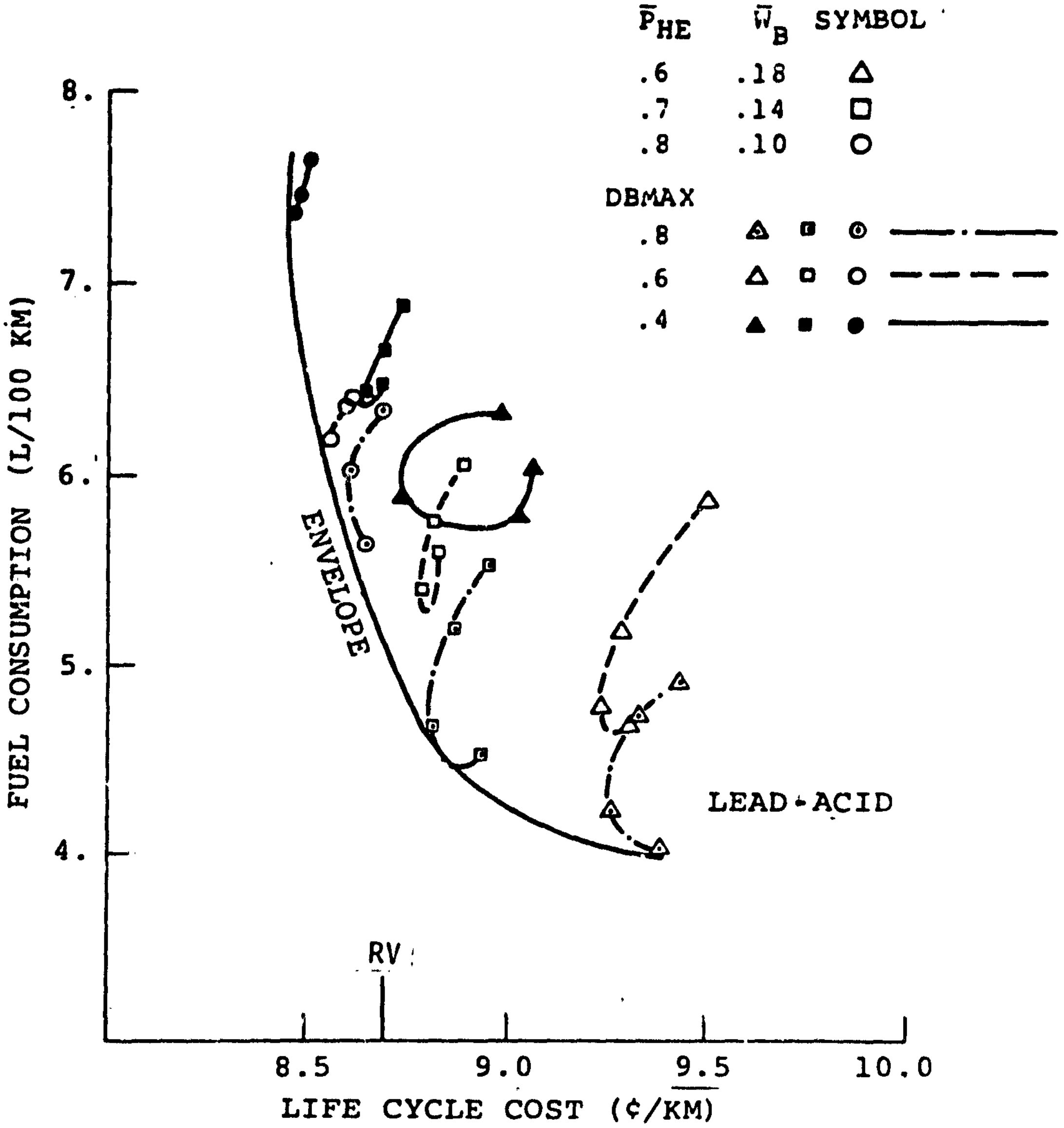
MINIMUM COST CASE-RETAIL PRICE= 2 xMFG COST (REF.VEHICLE)
+1.25xΔCOST(HYBRID-REF.VEHICLE)
BATTERY REPLACEMENT=1.25xOEM COST
30% DECREASE IN GASOLINE PRICES

Figure 3-16b Hybrid Vehicle Life Cycle Cost Vs. Fuel Consumption



NOMINAL COST CASE-RETAIL PRICE= 2xMFG COST
BATTERY REPLACEMENT=2xOEM COST
30% INCREASE IN ELECTRICITY PRICES

Figure 3-17a Hybrid Vehicle Life Cycle Cost Vs. Fuel Consumption

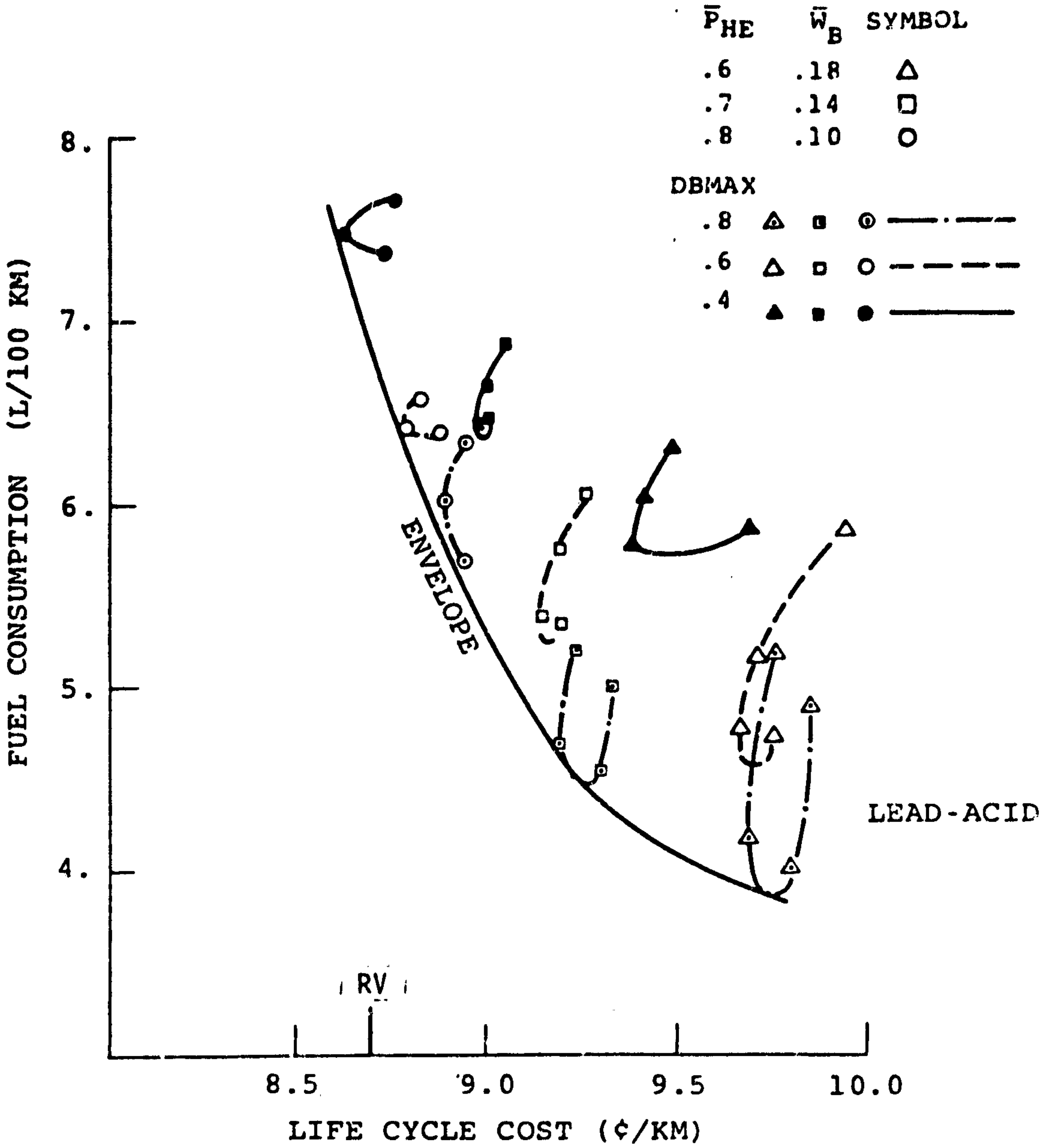


MINIMUM COST CASE-RETAIL PRICE=2xMFG(REF.VEHICLE)
+1 25xΔCOST(HYBRID-REF.VEHICLE)
BATTERY REPLACEMENT=1.25xOEM COST
30% INCREASE IN ELECTRICITY PRICES

Figure 3-17b Hybrid Vehicle Life Cycle Cost Vs. Fuel Consumption

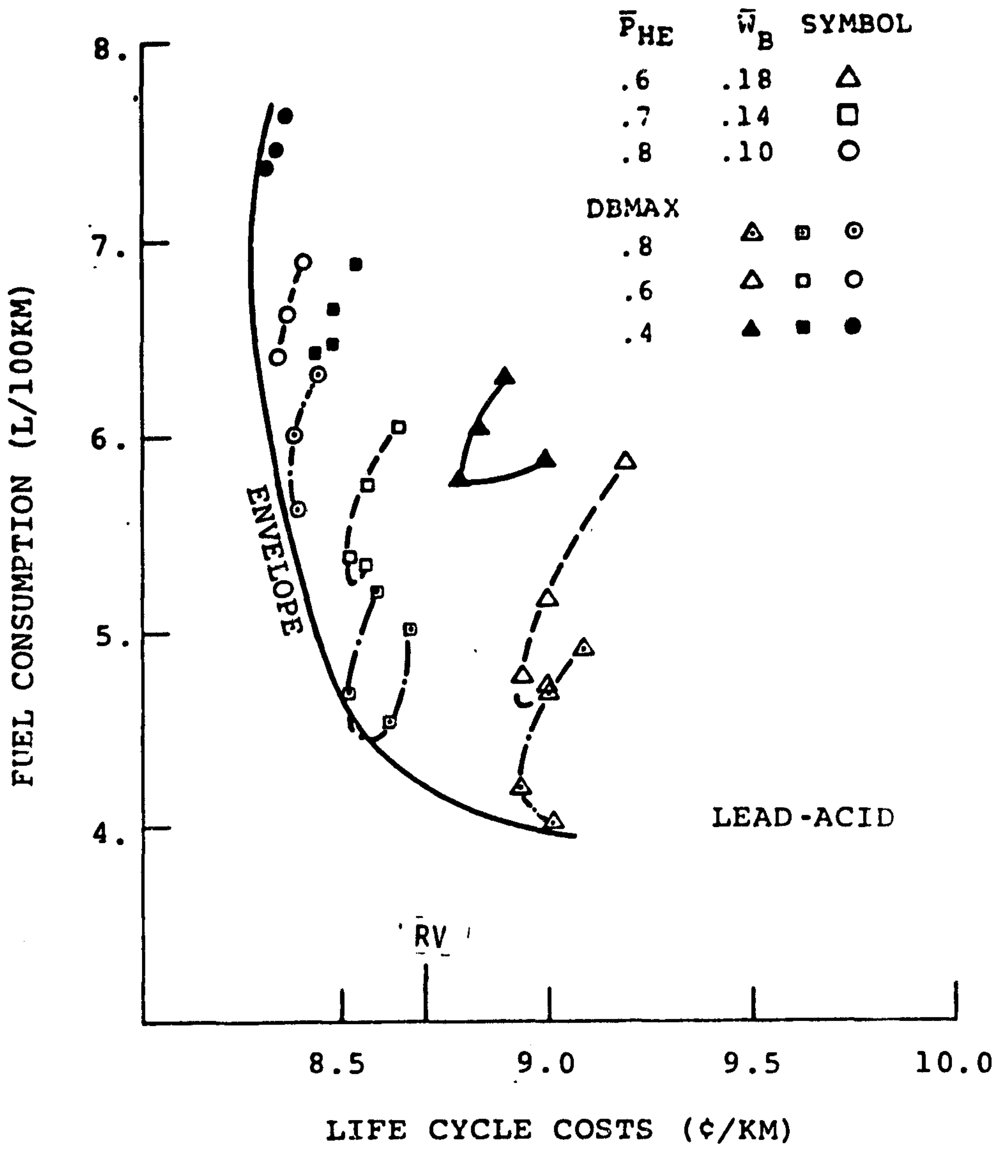
The effects of a 10% decrease in electricity prices are shown in Figures 3-18 a, b. The effects relative to the nominal case are negligible.

To summarize, the hybrid basic design parameters are most sensitive to gasoline prices. A 30% increase in gasoline prices appears to make the hybrid very competitive with the reference vehicle in terms of life cycle cost, at the selected heat engine power fraction of about .65. A 30% decrease in gasoline prices would make it non-competitive unless the heat engine power fraction reached the .7-.8 range, and even then, only in the minimum cost case. Changes in electricity prices are less significant and would not affect the selection of the basic design parameters, although they would affect, to a certain extent, the competitive positions of the hybrid with respect to life cycle costs.



NOMINAL COST CASE-RETAIL COST=2xMFG COST
BATTERY REPLACEMENT=2xOEM COST
10% DECREASE IN ELECTRICITY PRICES

Figure 3-18a Hybrid Vehicle Life Vs. Fuel Consumption



MINIMUM COST CASE-RETAIL PRICE= 2xMFG(REF.VEHICLE)
+1.25xΔCOST(HYBRID-REF.VEHICLE)
BATTERY REPLACEMENT=1.25xOEM COST
10% DECREASE IN ELECTRICITY PRICES

Figure 3-18b Hybrid Vehicle Life Cost Vs. Fuel Consumption

3.2 Baseline Hybrid Vehicle

Based on the results of the system level studies, a baseline hybrid vehicle was constructed with the following basic parameters:

Heat engine peak power = 53 kw (VW Rabbit gasoline)

Traction motor peak power = 30 kw (Siemens 1GV1, separately excited)

Heat engine power fraction = .64

Vehicle curb weight = 2080 kg

Battery type and weight = lead-acid, 355 kg

Battery weight fraction = .17

The heat engine and traction motor are currently available hardware, and they were chosen to give a power-to-test weight ratio slightly in excess of that predicted by the relationship used in the system level studies. For a heat engine power fraction of .64, that relationship predicts a power-to-weight ratio of .0345 kw/kg to give a 0-90 kph time of 15 sec; the power-to-weight ratio chosen for the baseline vehicle is .0374 kw/kg. This was done to ensure that the minimum performance requirement would be met at all battery states of charge down to the discharge limit.

3.2.1 Preliminary Tradeoffs

Before finalizing the configuration of the baseline hybrid vehicle, some preliminary tradeoffs were made with respect to the system mechanical configuration and the type of armature current control. This was followed by some preliminary control strategy optimization.

Mechanical Configuration

Two mechanical configurations were considered for the baseline vehicle propulsion system. These are shown in Figure 3-19. On the face of it, system B has the advantages of not requiring the electric motor to drive through the torque convertor, and of having higher overall efficiency due to the smaller torque convertor. It does, however, have the following disadvantages:

a) When the heat engine is to be started up to satisfy a power demand, the power to start it must come through the torque convertor if, as seems reasonable, the starting impetus is to come from the electric motor and the vehicle inertia. This will probably introduce considerable delay in starting the engine, compared to system A, in which the heat engine is directly coupled to the electric motor once the clutch is engaged.

b) In system B, the electric motor must be stationary when the car is at rest; if the heat engine is also to be shut off at this time (which is one of the prerequisites for the type of control strategy we are using), then the automatic transmission must have a separate oil pump driven by an auxiliary motor to keep operating pressure available in the transmission. (Unless operating pressure is available when the car is at rest, the transmission would be in neutral when the motor is accelerated, and while the vehicle is moving away from a stop, the low gear clutch of the transmission would be slipping, just like the clutch associated with a manual

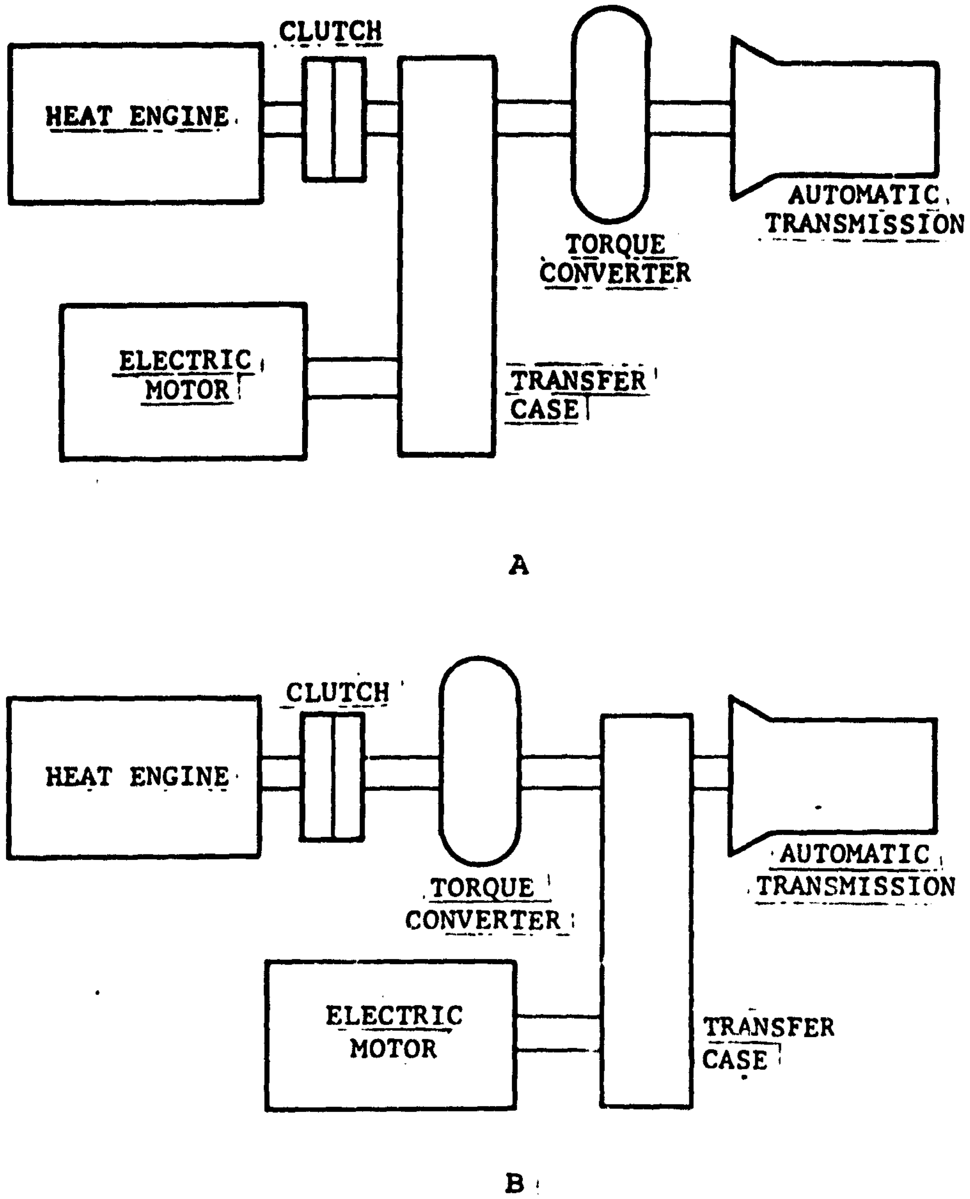


Figure 3-19 Alternative Mechanical Configurations for Baseline Propulsion System

shift transmission. The clutches in an automatic are not designed for such service.)

c) System B is mechanically more complex, due to the necessity for separating the torque convertor from the transmission and for a separate oil pump.

Because of the control problems and complexity associated with system B, a study was undertaken to quantify the fuel consumption and energy consumption differences between the two systems to ascertain whether the advantages of system B would be worth the cost.

The characteristics used for the heat engine and electric motor in this study are shown in Figures 3-20 through 3-22. An actual engine map of the VW Rabbit gasoline engine was not available, and a composite map was constructed based on data from several contemporary engines and corrected to fit the known max bmep and max power points of the Rabbit engine. The resultant fuel map and max bmep line is shown in Figure 3-20. Figure 3-21 shows the maximum (driving) and minimum (braking) torque for the separately excited motor. For this study, the characteristics with the full power controller were used (dashed line). Figure 3-22 shows the input power to the motor as a function of shaft torque and speed. It will be noted that the Siemens data for this motor falls very close to the simplified representation used in the computer model. Battery specific power vs. specific energy characteristics assumed for the ISOA lead-acid battery pack are shown in Figure 3-23; refer to Figure 2-11 for the assumed life characteristics.

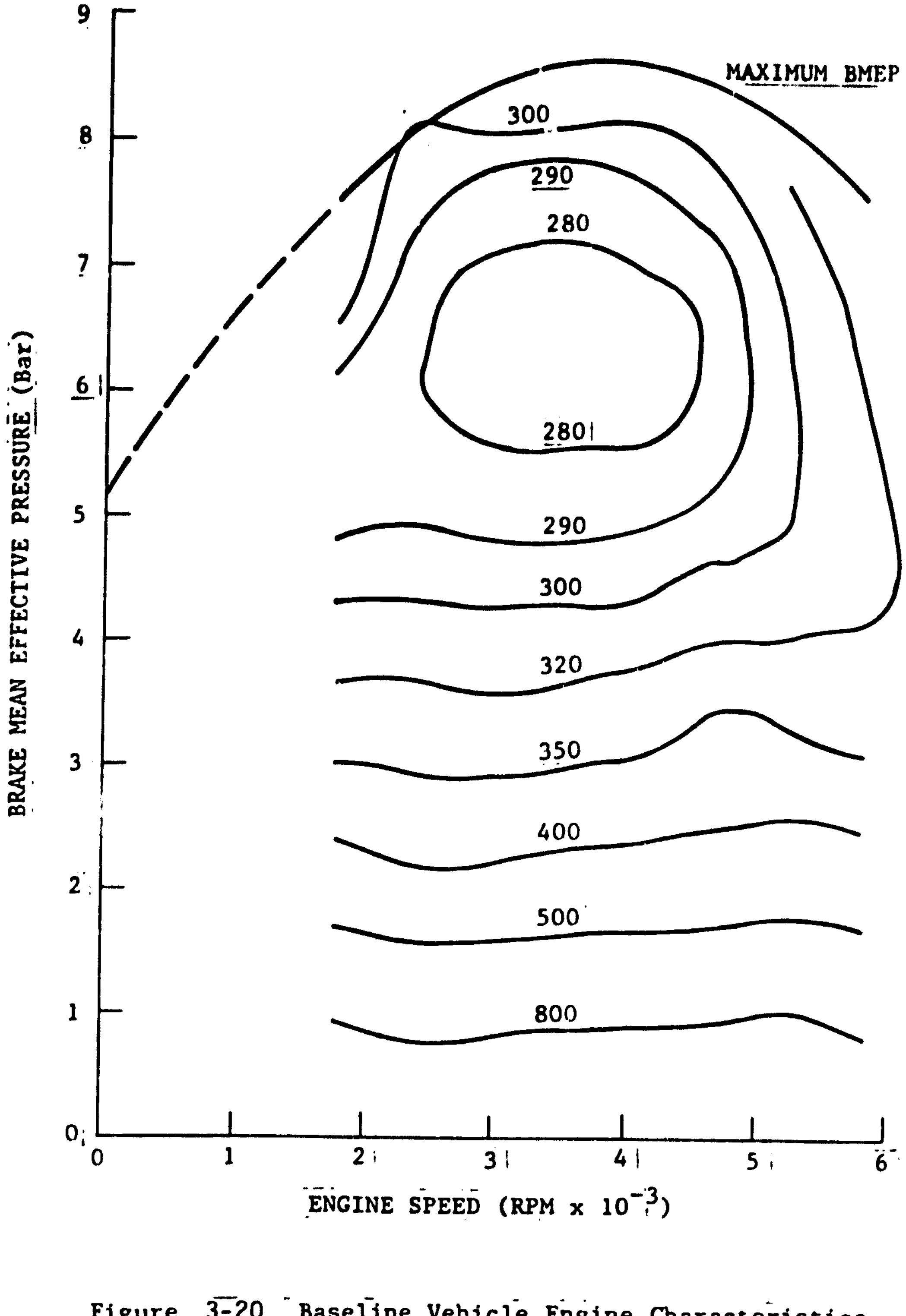


Figure 3-20 Baseline Vehicle Engine Characteristics

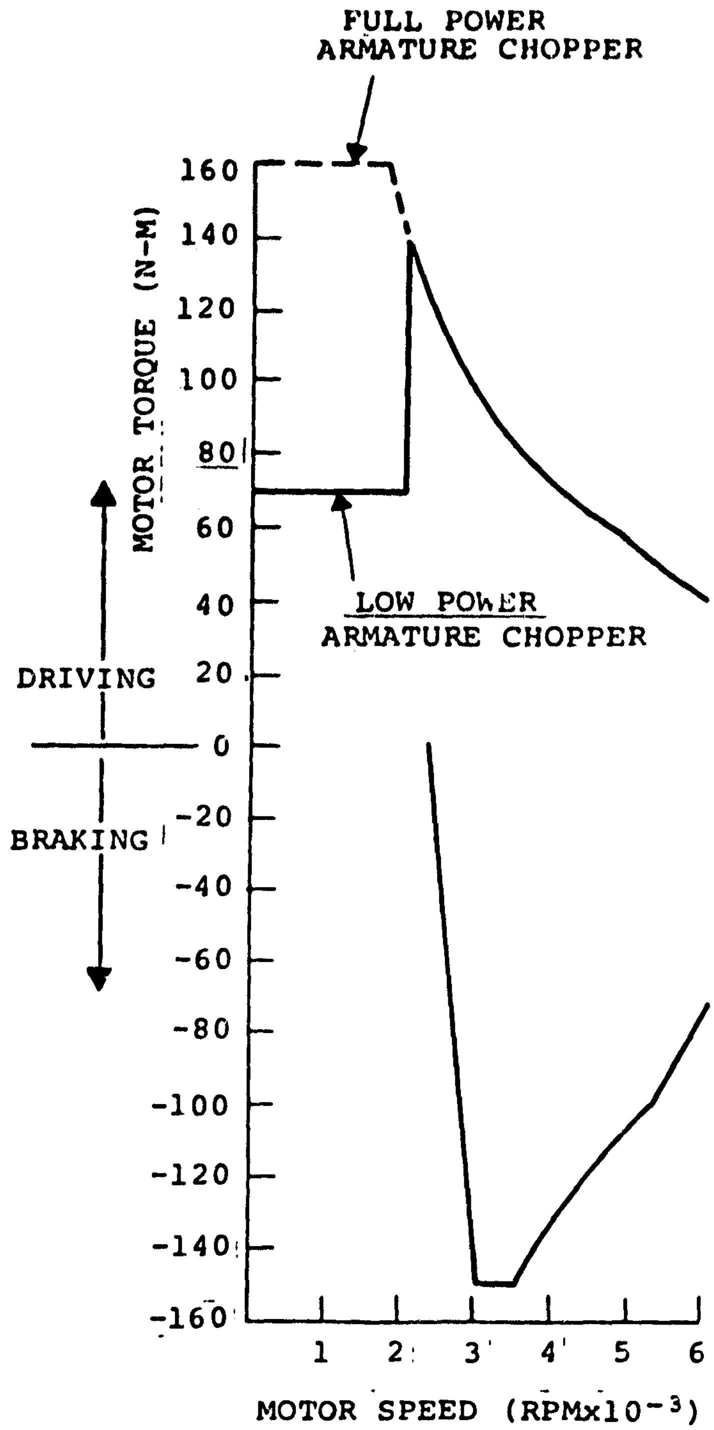


Figure 3-21 Baseline Vehicle Motor Characteristics

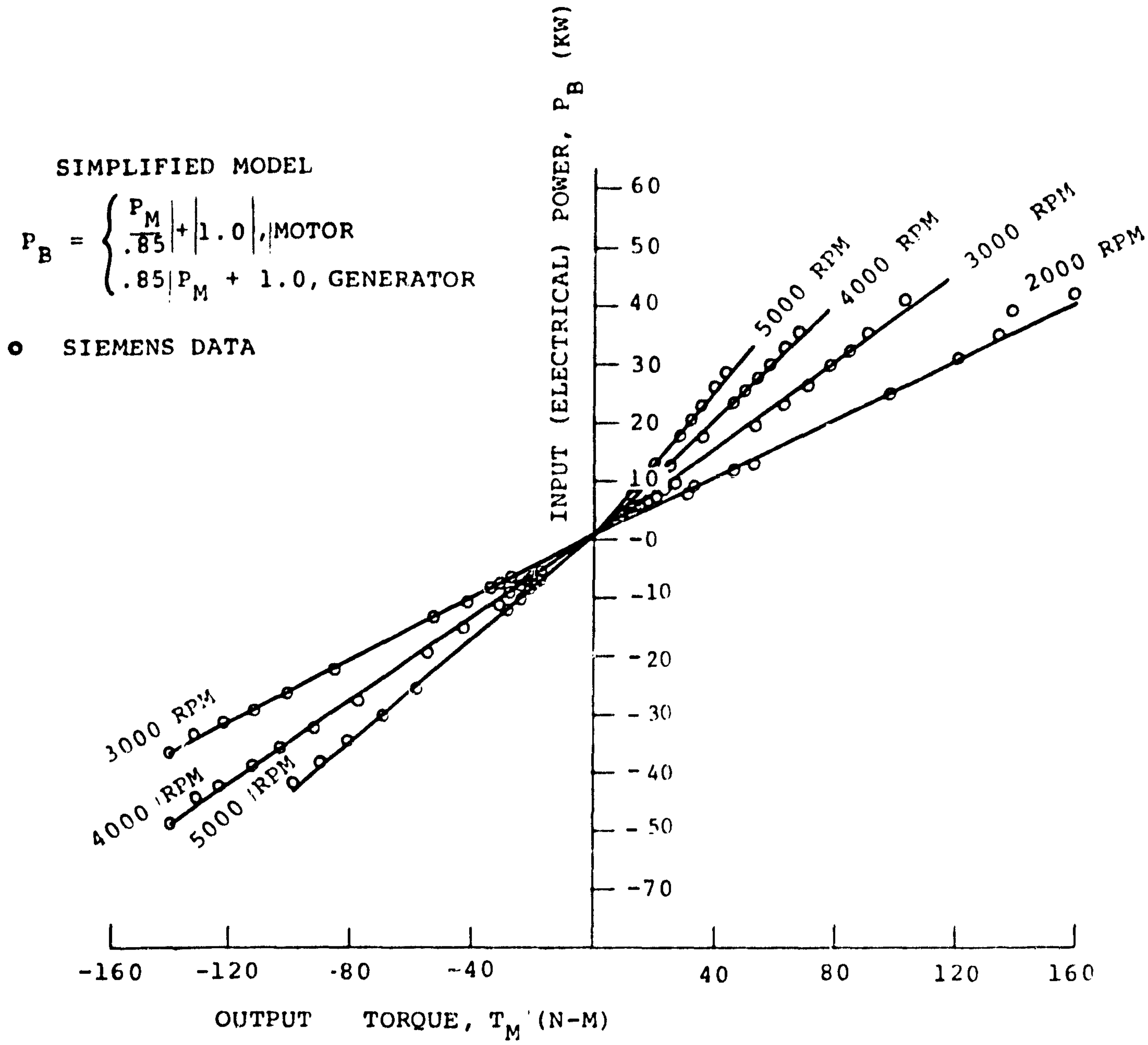


Figure 3-22 Baseline Vehicle Motor Characteristics

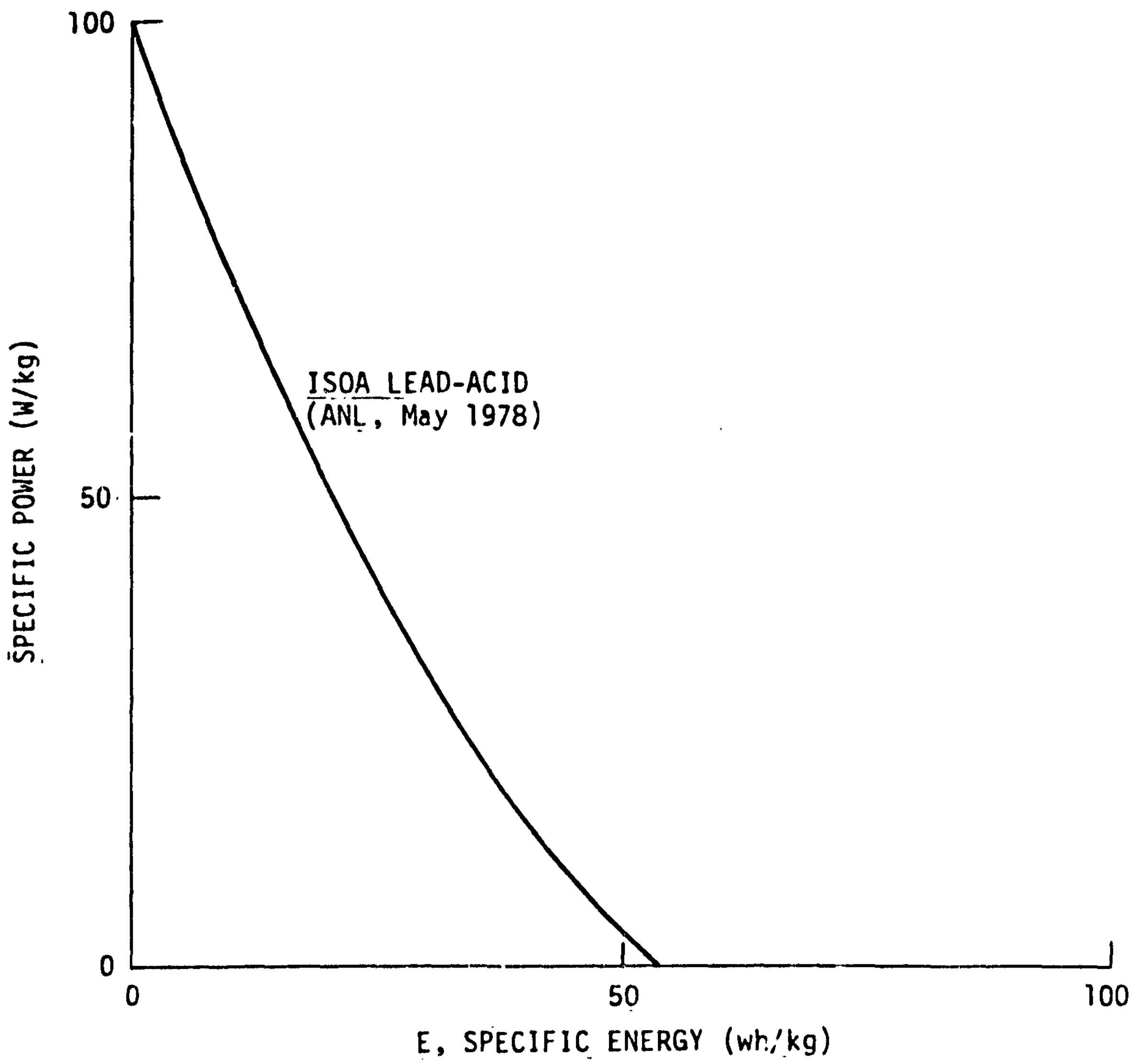


Figure 3-23 Improved State-of-the-Art Lead-Acid Battery Characteristics

For both configurations A and B, a three-speed transmission was used, with ratios of 2.45 in 1st, 1.45 in 2nd, and 1.0 in third; final drive ratio was 4.1:1. For configuration 'A', a three element torque convertor with a stall torque ratio of 2.1:1 and a diameter of 276 mm (10.87 in.) was used, giving a stall speed of 1800 rpm at full throttle with both the heat engine and electric motor operating. The transmission/torque convertor combination is essentially identical to the Chrysler Torqueflite used with the 3.7 l (225 in.^{3}) six. A plot of road load power vs. available power is shown in Figure 3-24 for configuration A with the full power controller.

For configuration B, the 276 mm torque convertor was replaced by a 242 mm (9.5 in.) unit with a stall torque ratio of 2.4, giving a stall speed of 2100 rpm at full engine throttle. This unit is similar to the torque convertor used on the Rabbit automatic transmission. Its characteristics, along with those for the system A torque convertor, are shown in Figure 3-25.

The results of the study may be summarized as follows:

Prior to making adjustments to the fuel consumption to obtain a net zero battery output on the Mode 2 operation, the two systems showed almost identical fuel consumption on the two modes. However, due to the presence of torque convertor losses when the vehicle is at rest and less efficient regenerative braking, system A showed slightly higher battery energy consumption. For example, on the urban cycle, the figures were as follows:

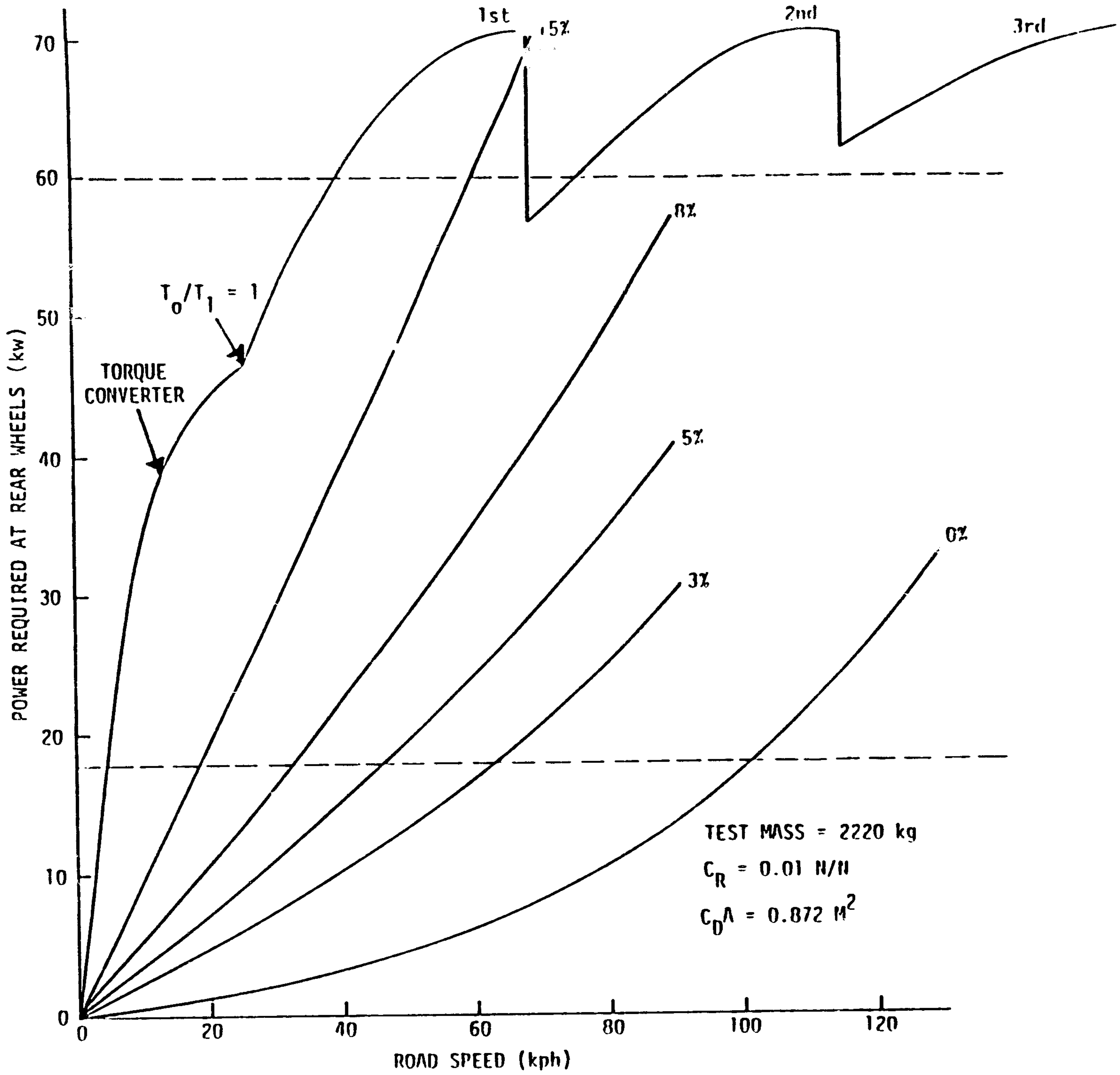


Figure 3-24 Power Requirements for Baseline Hybrid

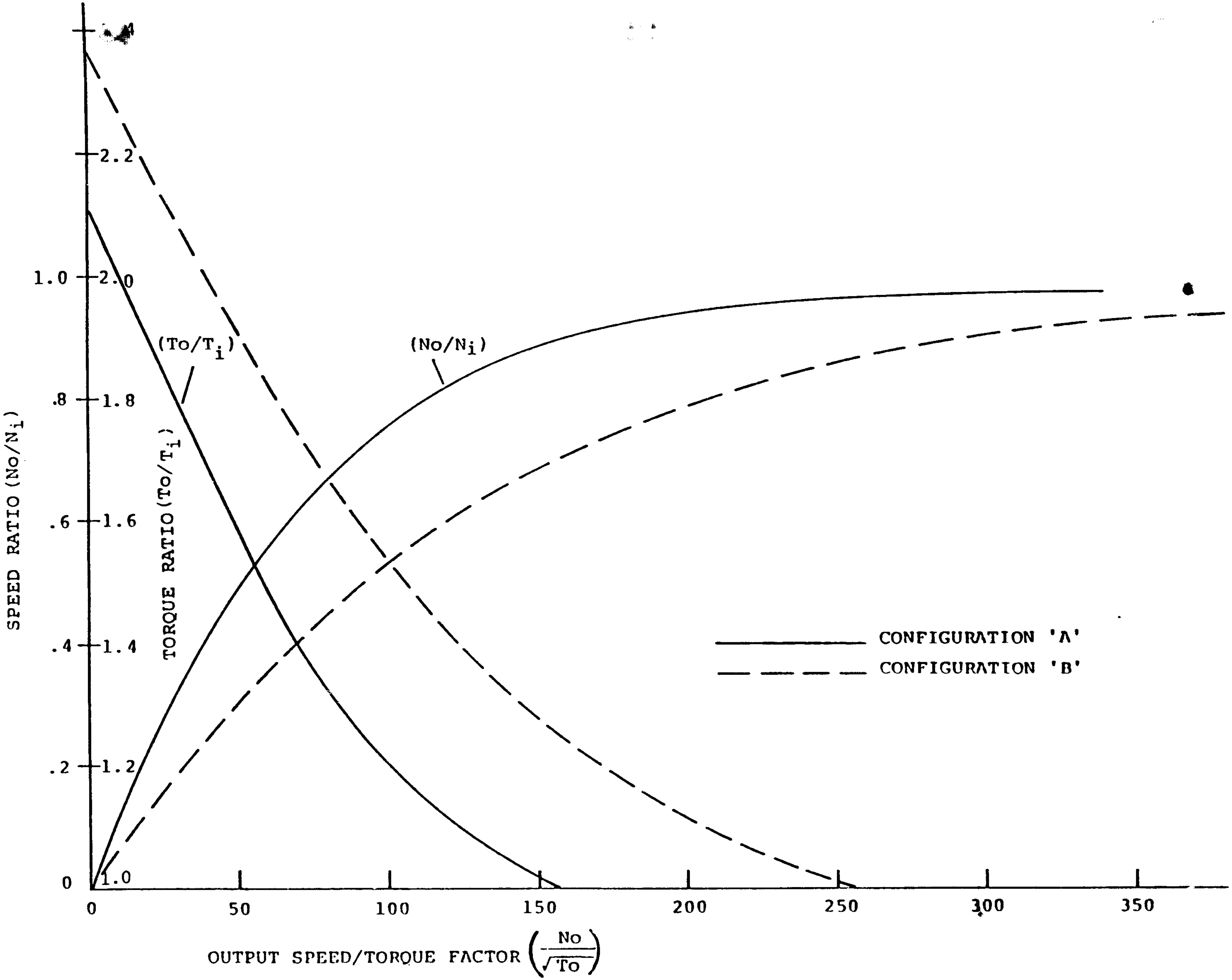


Figure 3-25 Torque Convertor Characteristics for Hybrid (Baseline)

		A	B
Battery energy consumption (kw-hr/km):			
	Mode 1	.182	.176
	Mode 2	.016	.010
Fuel consumption (g/km):			
	Mode 1	27.0	27.4
	Mode 2	85.1	86.1

After corrections were made to bring the net battery output on Mode 2 to zero, and the yearly average fuel economy and wall plug energy consumption computed, the results were as follows: System B provided slightly under 5% better fuel economy with identical wall plug output. Keeping in mind that 5% more fuel economy on a vehicle which is getting on the order of 40 mpg does not represent much fuel, we came to the conclusion that the additional complexity and control problems associated with B were not worth the cost; and we opted for configuration A for the baseline vehicle.

<u>Armature Control Methods</u>

Due to the wide speed range over which motor speed can be adjusted by field control, the armature chopper turns out to be functional only when the vehicle is starting from rest, and then only over a narrow speed range. In fact, with a torque convertor stall speed of about 1800 rpm, about the only thing that the armature chopper does in a full throttle acceleration is to bring the motor up to that stall speed before the vehicle has picked up any speed. In light of this, and in view of the fact that the armature chopper is a high cost item, we looked at the effect of reducing the power rating, and hence cost, of this portion of the controller. In particular, the effect of reducing the maximum current (and torque) to the motor in the region below base speed by more than half was investigated. The effects of this on the motor torque curve are shown in Figure 3-21.

Using the full throttle acceleration program, VSPDUP2, the effects of using the low power controller on acceleration times were analyzed. The result was an increase in all standing start acceleration times (0-50 kph and 0-90 kph) by .4 sec.; all were still within specification. The 40-90 kph time and the time required for a high speed pass maneuver were unaffected, of course, because the motor operates under field control only in these speed ranges. Likewise, gradeability at anything other than zero speed was unaffected. The maximum climbable gradient (at zero speed) was reduced from 100% to 49%, still more than adequate.

As a result, we came to the conclusion that a low power armature chopper would be more suitable for the baseline vehicle than a full power chopper. A further discussion of the tradeoffs involved between the two chopper types will be found in Section 3.5.4.

Control Strategy

The control strategy used for the runs described above was similar to that described in Section 2 for the system level studies using HYBRID; however, instead of cutting the heat engine in when the system output reached a minimum power level during Mode 1 operation, the cut in point was determined by a minimum torque level, T_{EOMIN}. A value for T_{EOMIN} of 45 n-m was found to be best; this corresponds to a bmep of 3.9 bar. Referring back to the fuel map of Figure 3-16, operating the heat engine only above 3.9 bar on Mode 1 means that the bsfc during this mode is always less than about 320 g/kw-hr, or within 15% of the best bsfc. This strategy had one disadvantage: it required the electric motor to operate at power levels well above its nominal rating when operating on Mode 1 at high motor speeds (above 3000 rpm). This was not desirable, first from the standpoint of motor durability (particularly brush life), but more importantly, from the standpoint of the batteries. Consequently, a revised control strategy was constructed in which the heat engine cut-in point occurred when the system demand exceeded a certain torque level T_{EOMIN}, as long as the speed was such that the corresponding power did not exceed a maximum level P_{EOMIN}. If the power determined by T_{EOMIN} and the motor speed exceeded P_{EOMIN}, then the cut-in point was determined by P_{EOMIN}. With this strategy, using a cut-in torque of 45 n-m and a maximum motor power of 20 kw (only slightly above the motor's nominal rating of about 16 kw), the fuel economy was only 2.6% less than that obtained for the system in which only torque was used to determine the cut-in point; wall plug energy consumption was essentially identical. This basic strategy, which tries to keep the heat engine operating above a minimum torque level, but also avoids excessive power demands on the electric motor and battery, was consequently adopted for the baseline vehicle.

3.2.2 Characterization of the Baseline Vehicle

After having performed the small amount of preliminary optimization just described, the baseline vehicle was characterized in terms of fuel and energy consumption, performance, and cost, relative to the reference vehicle.

Fuel and Energy Consumption

The program HYBRID2 was used to estimate the fuel and energy consumption of the baseline hybrid and the reference vehicle. For the reference vehicle, the simulation program came up with an estimate of 8.66 km/l (20.4 mpg) average fuel economy over a year's use, for the composite driving cycle described in Section 3 of the Task 1 report.(1) The projected in-use mileage of the reference vehicle was 7.65 km/l (18 mpg); so a correction factor of 18/20.4 was applied to all subsequent fuel economy calculations.

The results, in terms of yearly averages, are summarized in Table 3-2. The results indicate that a hybrid vehicle with a suitable control strategy could provide about two times the fuel economy of a conventional vehicle which employs comparable engine and vehicle technology. It is also of interest to note that the total energy requirement (crude oil thermal equivalent) of the hybrid is similar to the reference vehicle; however, the petroleum based energy consumption is only about half, under the assumption that 15% of the electrical energy generation comes from petroleum.

For the individual driving cycles which comprise the yearly composite driving cycle, the breakdown of energy expenditures is as shown in Table 3-3. The numbers given are for one pass through the driving cycle, and they are given both in absolute terms (in megajoules) and as percentages of the total system (heat engine + traction motor) output.

The following points with respect to the numbers in Table 3-3 are noteworthy.

Table 3-2. FUEL AND ENERGY CONSUMPTION FOR BASELINE HYBRID AND REFERENCE CONVENTIONAL VEHICLE

	Baseline Hybrid	Reference Vehicle
1. Average Fuel Economy (km/l)	16.7	7.65
2. Average Wall Plug Energy Consumption (kw-hr/km)	.212	-
3. Average Total Energy Consumption (1) (kw-hr/km)	1.324	1.371
4. Average Petroleum Energy (2) Consumption (kw-hr/km)	0.732	1.371

(1) Computed as the energy equivalent of the total crude oil required at the refinery input, under the assumption that all the input energy comes from crude oil, and under the following assumptions:

Refinery/distribution efficiency = .93 (fuel oil)
.84 (gasoline)

Electrical generation efficiency = .36

Electrical distribution efficiency = .91

(2) Same as (1), except the assumption is made that only 15% of the electrical energy generation comes from petroleum.

Table 3-3. ENERGY EXPENDITURES ON COMPONENT DRIVING CYCLES

	SAEJ227(a)B		FUDC		FHDC	
Energy Expenditure	MJ	%	MJ	%	MJ	%
Rolling Resistance	.0739	38.1	2.604	36.9	3.594	37.3
Aerodynamic	.0119	6.1	1.401	19.9	4.563	47.4
Differential	.0071	3.7	.389	5.5	.418	4.3
Transmission	.0153	7.9	.797	11.3	.746	7.8
Torque Convertor	.0160	8.2	.361	5.1	.021	.2
Brakes	.0700	36.0	1.503	21.3	.291	3.0
System Output	.1942	100.0	7.053	100.0	9.633	100.0
Heat Engine Output on Mode 1	.0079	4.1	1.933	27.4	2.532	26.3
Motor Shaft Output on Mode 1 (driving)	.1923	99.0	6.824	96.8	7.818	81.2
Motor Shaft Output on Mode 1 (braking)	-.0060	-3.1	-1.704	-24.2	-.717	-7.5
Net Battery Output on Mode 1	.2945	-	8.385	-	9.536	-
Average Battery Output Power on Mode 1 (kw)	4.09	-	6.11	-	12.47	-

- Rolling resistance tends to be an almost constant fraction of the total energy expenditure, regardless of driving cycle. The apportioning of the rest of the expenditure varies widely however, as would be expected.
- Regenerative braking has the largest effect on the urban cycle, with the motor and the brakes absorbing comparable amounts of energy. Although the 227(a)B involves a lot of stop and go driving, the speed range (0-32 kph) is too low to provide effective regenerative braking with the transmission shift logic used.
- Due to the assumption of lockup of the torque convertor on the top two gears, the torque convertor losses are relatively lower on the urban cycle than on the 227(a)B cycle, and much lower on the highway cycle (which uses 3rd gear almost 100% of the time).
- The specific output power corresponding to the average battery output power is 11.5 w/kg on the 227(a)B cycle and 17.2 w/kg on the urban cycle. These values are reasonably consistent with the ISOA goals for lead-acid batteries. However, the value of 35 w/kg on the highway cycle is very high for lead-acid batteries (nearly twice the ISOA goal of 20 w/kg sustaining specific power). It is also fairly high for nickel-iron and nickel-zinc batteries (ISOA goal of 30-40 w/kg).

Consequently, we came to the conclusion that, in the subsequent optimization of the control strategy, speed dependence in addition to power and torque dependence should be considered. This optimization is discussed in Section 3.4.

The approximate distributions of battery output power, as obtained for the baseline vehicle, on the two most significant driving cycles (urban and highway) are shown in Figures 3-26 and 3-27.

Performance

The acceleration curve obtained for the baseline hybrid vehicle is shown in Figure 3-28. It slightly exceeds the minimum performance specifications of 0-50 kph in 6 sec., 0-90 in 15 sec., and 40-90 in 12 sec. This allows a little margin for the slightly lower motor performance at the battery discharge limit of .6. Maximum gradeability as a function of speed is shown in Figure 3-29 for the baseline hybrid. The gradeability requirements given in Section 5 of the Task 1 report[1] must be satisfied at all battery states of charge down to the discharge limit. The implications of these requirements for the baseline hybrid are summarized in Table 3-4. The most severe of these requirements, as far as the battery is concerned, is that of maintaining 85 kph (53 mph) on an 8% grade for 5 km. It implies that the battery must be able to supply 1.47 kw-hr at a 25 kw rate, starting with a state of charge corresponding to the battery discharge limit. For the baseline lead-acid battery, the specific power at the 25 kw rate is 70 w/kg and, from Figure 3-19, the available specific energy at this rate is 12 w-hr/kg, or a total of 4.26 kw-hr (to 100% DOD). Thus, if the battery discharge limit corresponds to a 60% DOD, the battery would reach a depth of discharge of .6 + 1.47/4.26, or .945. This is very close to complete discharge;

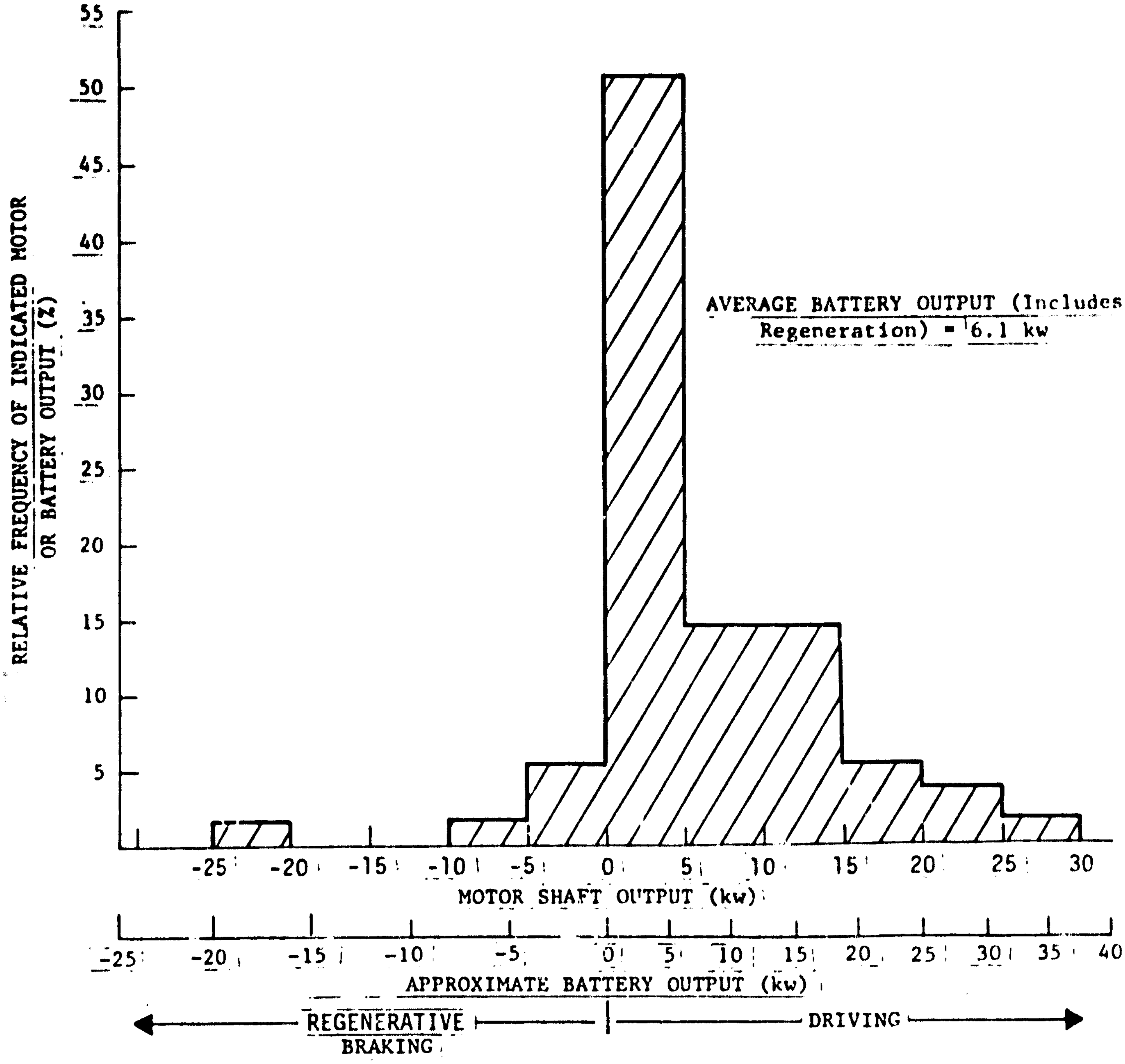


Figure 3-26 Distribution of Motor and Battery Power Output
Urban Cycle, Control Mode 1

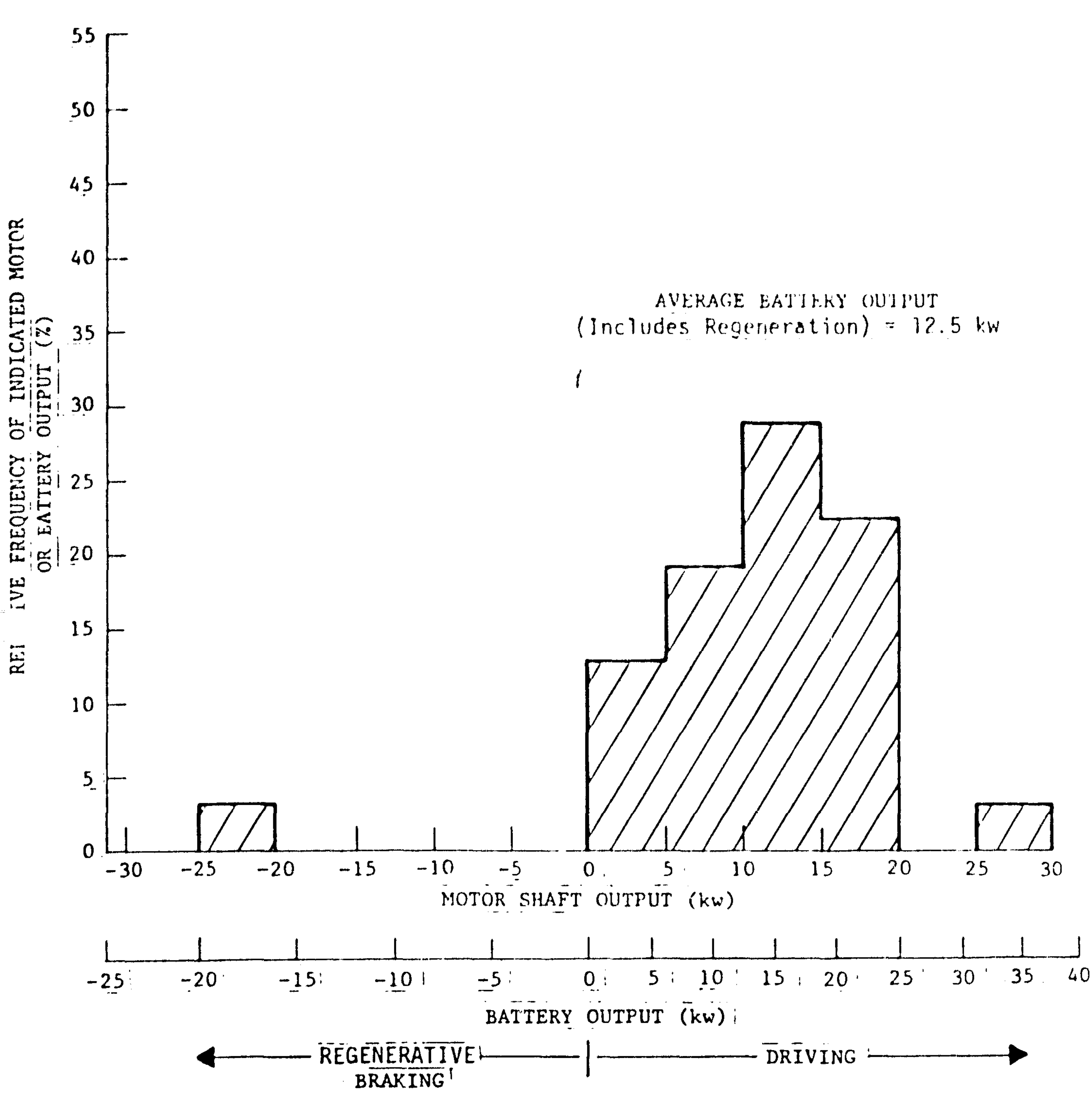


Figure 3-27 Distribution of Motor and Battery Power Output Highway Cycle, Control Mode 1

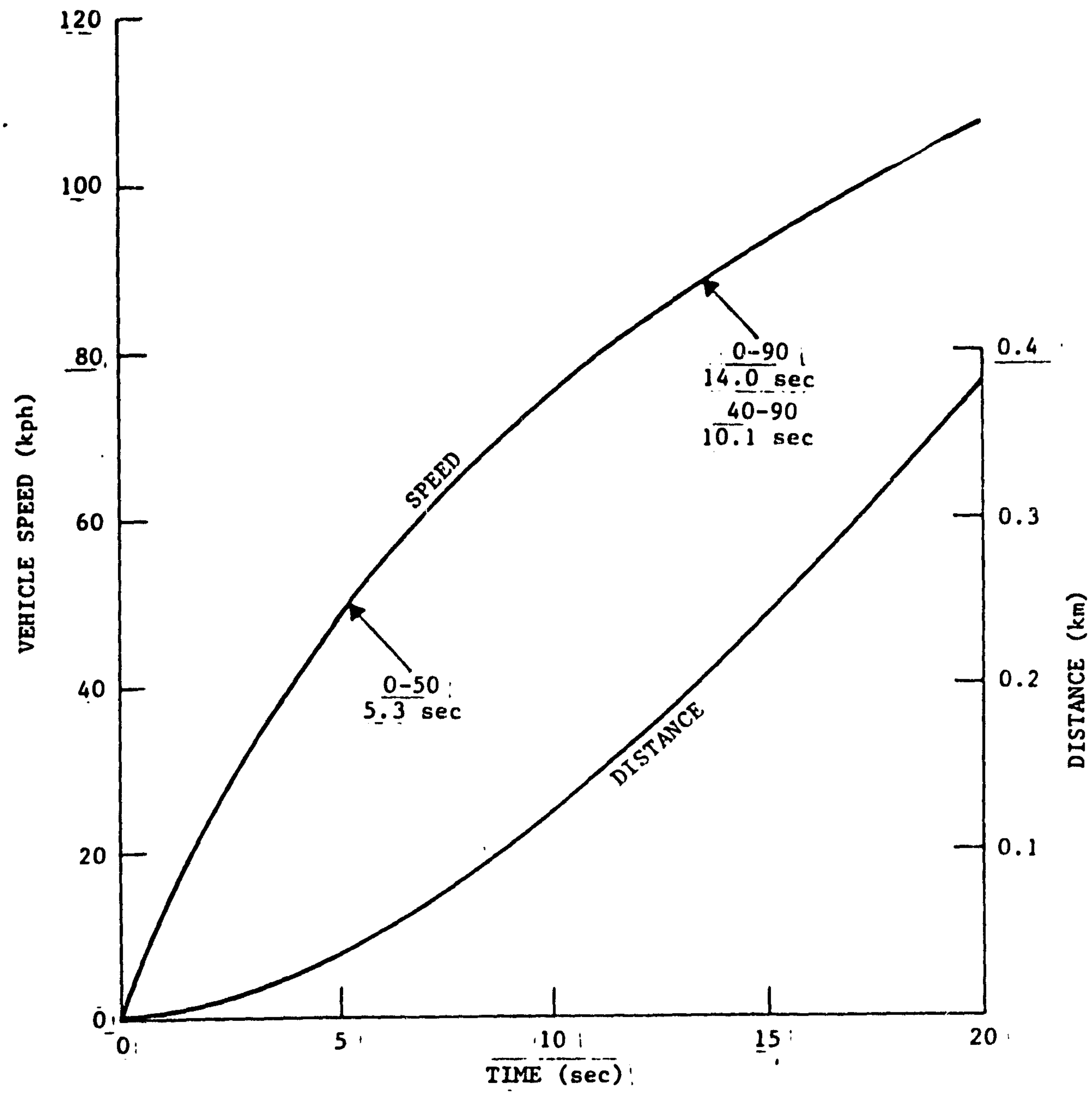


Figure 3-28 Baseline Hybrid Acceleration Characteristics

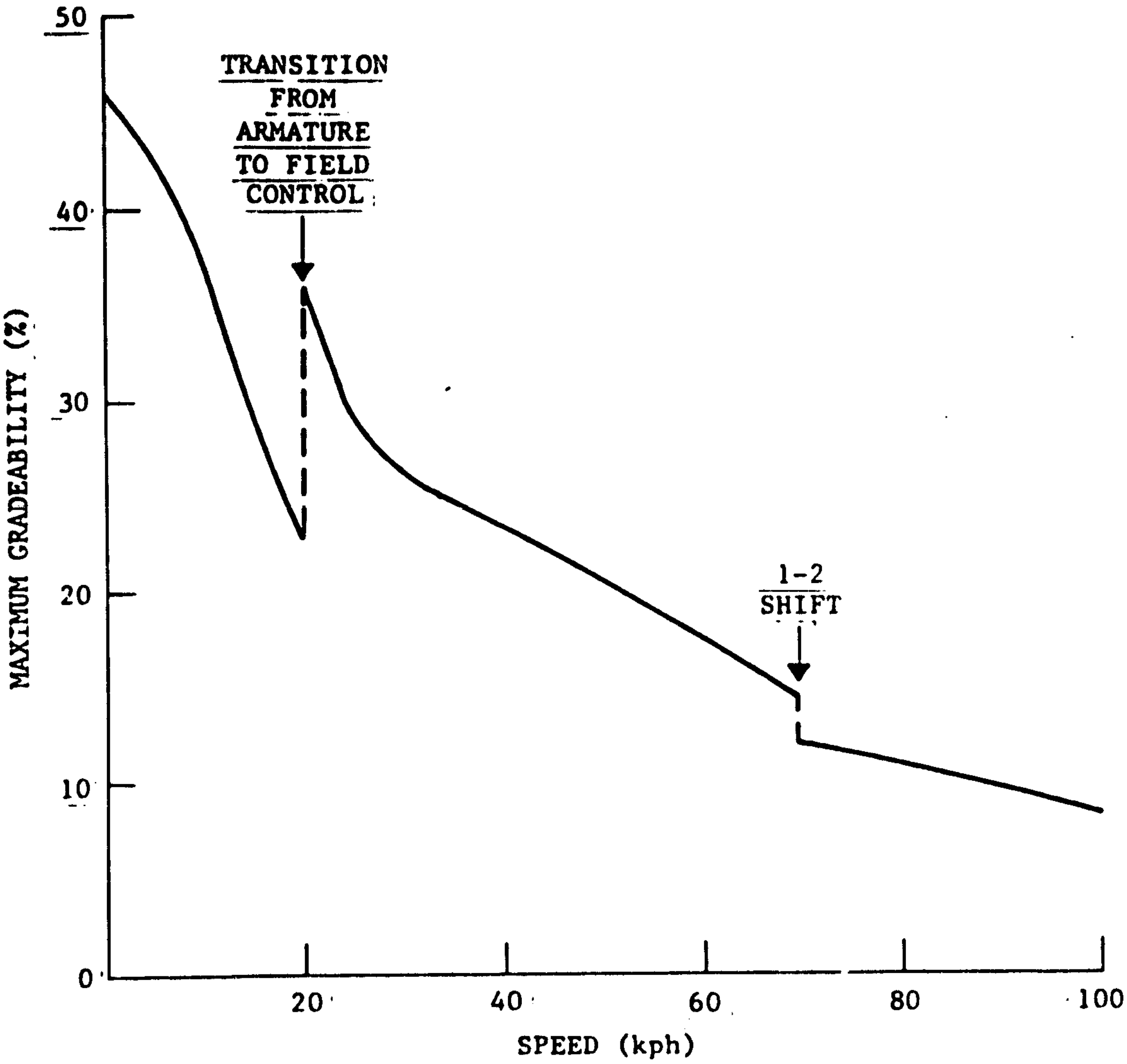


Figure 3-29 Baseline Hybrid Gradeability Characteristics

Table 3-4. IMPLICATIONS OF GRADEABILITY REQUIREMENTS FOR BASELINE HYBRID

Requirement	Required Power Engine + Motor (kw)	Max Engine Power Available, Gear	Power Req. from Motor	Power Req. from Battery	Energy Req. from Battery (kw-hr
3% @ 90 kph Indef.	37.2	47.9, 2nd	-	-	-
5% @ 90 kph 20 km	50.6	47.9, 2nd	2.7	3.4	.76
8% @ 85 kph 5 km	65.7	45.8, 2nd	19.9	25.0	1.47
8% @ 65 kph Indef.	47.5	53.1, 1st	-	-	-
15% @ 50 kph 2 km	60.9	45.5, 1st	15.4	19.2	0.77

however, given the severity of the requirement and the fact that encountering a real world situation represented by this requirement would be a rare occurrence, this depth of discharge is probably acceptable. It is clear, however, that for a lead-acid battery pack of the weight assumed for the baseline hybrid, a discharge limit more than .6 would not be acceptable.

The rest of the gradeability requirements are not as much of a problem. The only thing we are slightly uncomfortable with is the requirement to shift down to second gear to satisfy the 90 kph gradeability requirements; it would be somewhat difficult to keep the engine from being obtrusive if it is necessary to shift down often during highway cruising to as high an engine speed as 4660 rpm, which is the engine speed at 90 kph in second gear. A further discussion of this aspect will be found in Section 3.4 and 3.5.5.

Cost Factors

A concept design package was prepared for the baseline hybrid and used to prepare rough order of magnitude cost estimates of the total vehicle system. In most areas, data exists to develop realistic costs at an annual volume of 100,000 units. In other instances, costs are based on judgment values only. These data will all be refined in the Preliminary Design task in order to be certain that we accurately reflect costs that are achievable.

A summary of the current order of magnitude costs for the baseline hybrid follows:

	Costs (over)/under Reference Vehicle
Four cylinder engine vs. V-8	$ 250
Parallel system hardware costs	(140)
Added clutch hsg & clutch pkg.	(32)
Suspension & tire upgrading	(9)
Battery packaging & cooling	(37)
Engine exhaust & emission control	150
Engine cooling system	(20)
Motor cooling system (blower motor)	(14)
Accessory drive	(15)
Motor	(800)
Controller/charger & actuators	(300)
Batteries and cables	(713)
Instrumentation	(120)
TOTAL HYBRID (OVER) REFERENCE	$ (1800)

These cost data compare the reference vehicle to the hybrid baseline system. Factors working favorably (+) or unfavorably (-) to the comparison would be the use of a PROCO engine in the reference vehicle$^{(+)}$, or the use of nickel-iron batteries$^{(-)}$; use of nickel-iron batteries would result in higher initial cost but lower operating costs as discussed in Section 3.1.2.

These costs would result in retail price increments ranging from $2250 to $3600, for a pass through at minimum increase up to a 2 x manufacturing cost assumption,as discussed in Section 3.8. At the lower pricing assumption, hybrid sales volumes would be significant, but would be severely limited at the higher 2 x manufacturing cost level pricing assumption.

We believe the costs shown above are conservative, they are extremely useful to us at this time to focus our attention on those high cost items that can effectively be cost reduced through system configuration and/or design changes.

Using the estimated retail price of $7636 for the reference vehicle and the range of retail price increments quoted above, the retail price range of the baseline hybrid would be from $9886 to $11,236. With these figures, and the fuel and energy consumption values given in Table 3-2, the LYFECC program was used to estimate life cycle costs. These ranged from 10.0¢/km for the case in which both the manufacturing cost increment and the replacement battery OEM costs were passed to the consumer at a minimum factor of 1.25, to 11.0¢/km for the maximum price case (factor of 2 on both manufacturing cost and battery costs). The reference vehicle life cycle cost was estimated at 8.7¢/km; i.e., the life cycle costs of the baseline hybrid range from 15% to 26% higher than the reference vehicle, depending on pricing strategy. The increment would, of course, be higher if the life cycle costs were computed over lesser mileages.

The retail and life cycle costs computed for the baseline vehicle are higher than the best values estimated during the system level tradeoff studies. This is primarily a result of the more detailed estimation of the baseline vehicle manufacturing cost than that provided by the WANDC program, and more realistic estimation of the capacity of the battery pack at discharge rates representative of those that occur in the hybrid vehicle.

3.2.3 Sensitivity Analysis for Baseline Hybrid

Table 3-5 shows the variation in the life cycle cost of the hybrid vehicle which occurs at the sensitivity boundary values of ± 30% on fuel prices and + 30%, - 10% on electricity prices. As discussed in connection with the system level tradeoffs, the effect of electricity price variations on the competitiveness of the hybrid vis-a-vis the baseline is much less than the effect of variations in fuel prices. At the + 30% fuel price level, the life cycle cost of the baseline is within about 8% of that of the reference vehicle for the minimum; at the - 30% level, the difference amounts to 23%.

Table 3-5. LIFE CYCLE COST SENSITIVITY

	Reference Vehicle	HYBRID (Baseline)	
		Cost Case 1 (Low)	Cost Case 2 (Nominal)
Nominal	8.7	10.0	11.0
Fuel +30%	9.5	10.3	11.4
Fuel -30%	7.8	9.6	10.6
Electricity +30%	8.7	10.2	11.2
Electricity -10%	8.7	9.9	10.9

3.3 Affects of Vehicle Parameter Variations from Baseline

In Figures 3-30 through 3-32, the effects of ± 10% changes in the following vehicle and propulsion system parameters are plotted.

- Rolling resistance (CTIRE1)
- Product of drag coefficient and frontal area (CDA)
- Vehicle test mass (VMASS)

The dependent variables plotted are:

- Fuel economy (FE)
- Energy consumption (e)
- Time to accelerate to 90 kph (t)

Rolling Resistance

The influence coefficient of rolling resistance on fuel economy is about -.5; i.e., a 10% increase in rolling resistance results in about a 5% decrease in fuel economy, and inversely. The influence on wall plug energy consumption is almost negligible; the reason for this is that, during most of the year's driving, the battery is discharged to the discharge limit. Consequently, on those days the energy consumption is essentially fixed. It is only on the days on which the battery discharge limit is not reached that the rolling resistance has an effect on energy consumption. The effect of rolling resistance on the 0-90 kph time is also small since most of the energy expended in this time goes into vehicle kinetic energy.

The baseline value of rolling resistance ($CTIRE1_0$) was 0.010; this includes not only tire rolling resistance (which is the major component), but also bearing losses and brake and seal drag. Analysis of coast-down test data obtained recently by JPL on SCT's electric

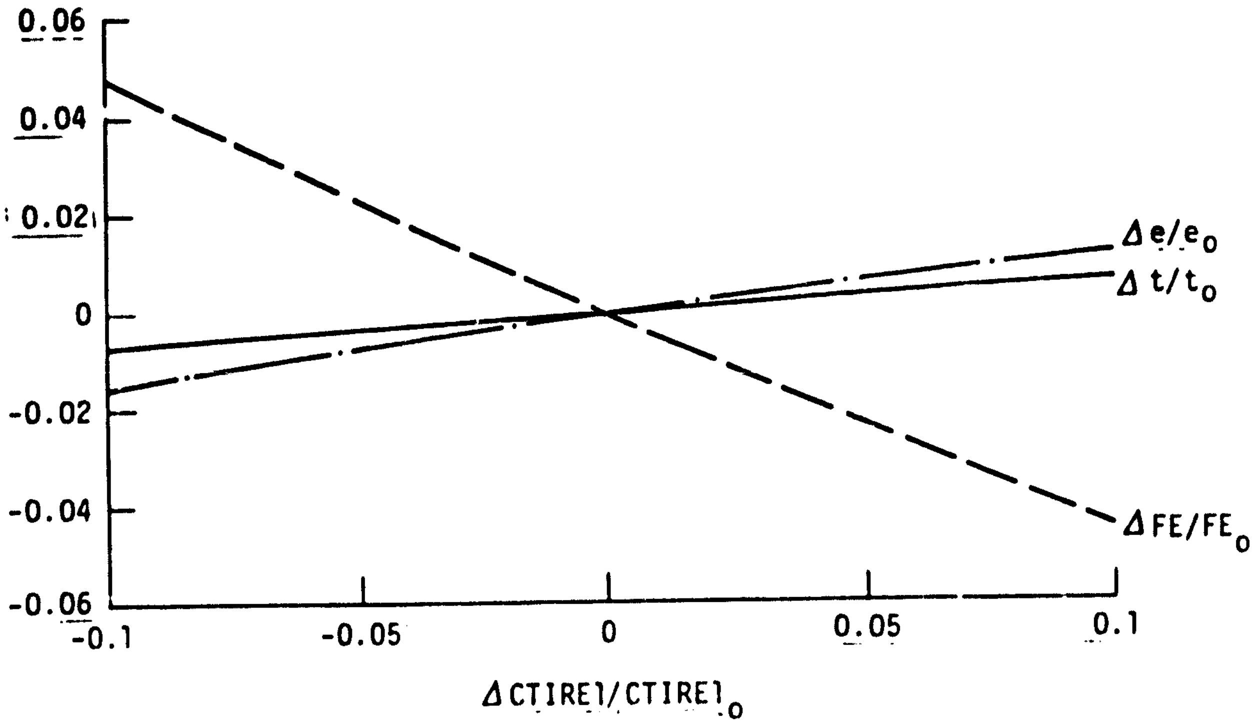


Figure 3-30 Effects of Change in Rolling Resistance

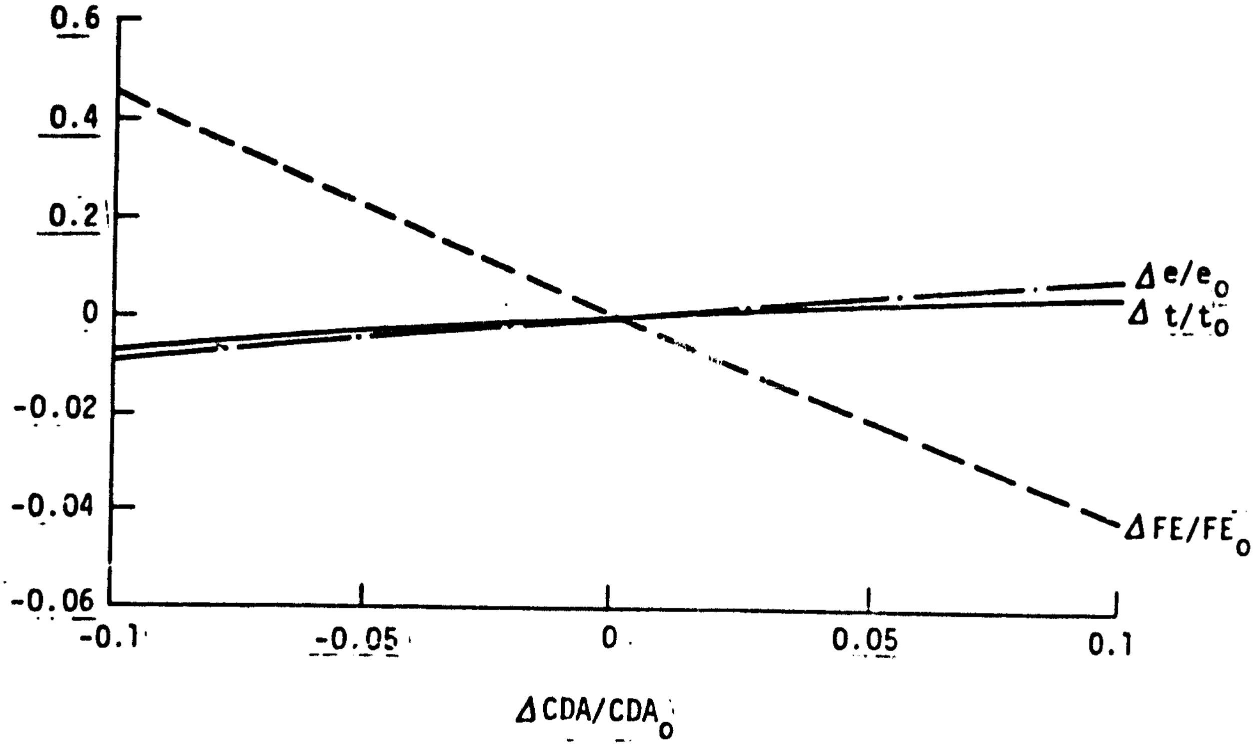


Figure 3-31 Effects of Change in Aerodynamic Drag

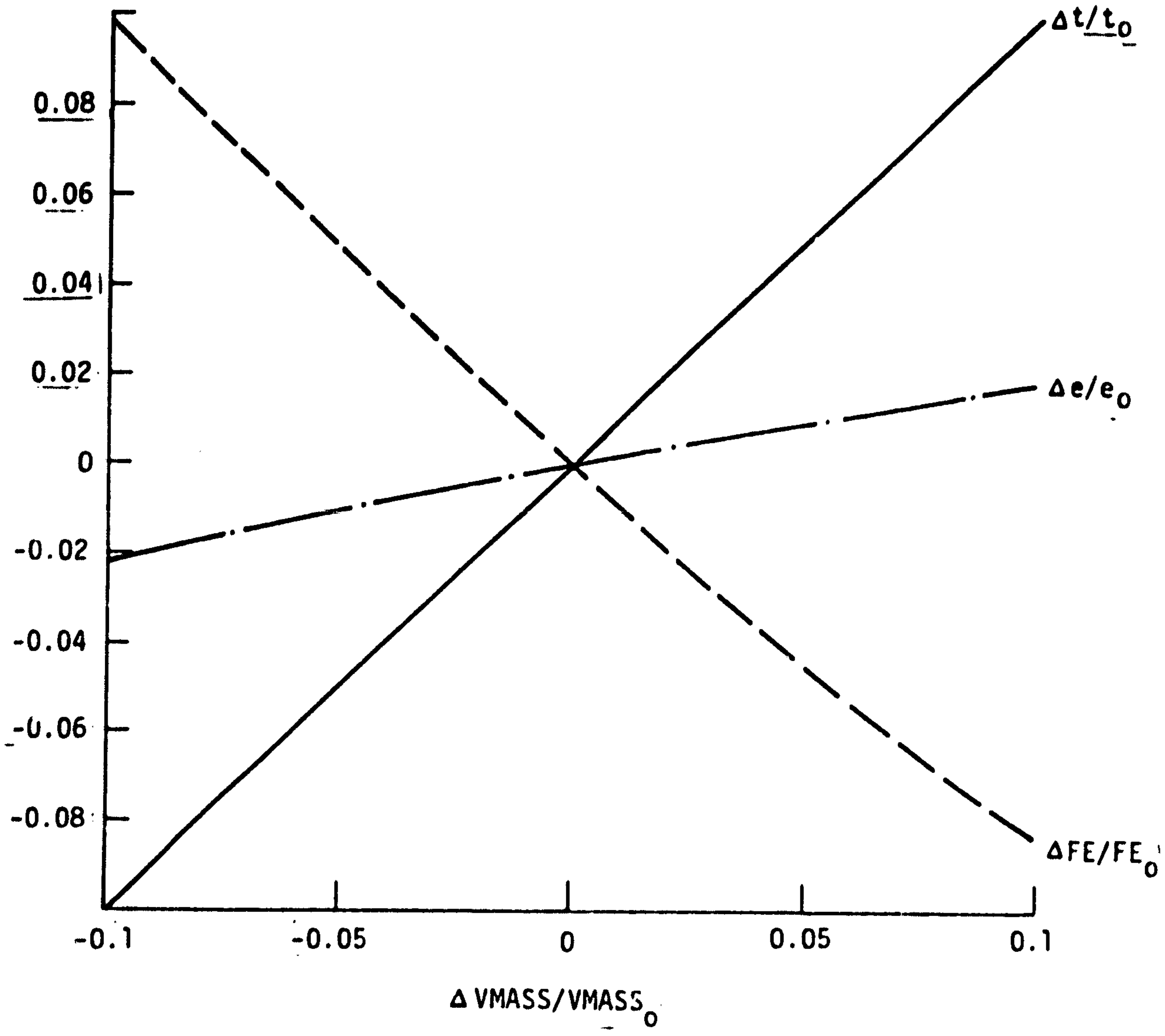


Figure 3-32 Effects of Change in Vehicle Mass

conversion of a VW Rabbit indicates that a rolling resistance coefficient in the .011-.012 range can be attained by current production steel belted radial tires, operating at about 36 psi inflation pressure range and within the tires' rated load range. This is somewhat lower than the value we had expected; however, it is consistent with our own measurements of motor input power in constant speed tests, as indicated in Figure 3-33. On this basis, together with information that improvements on the order of 10% for currently experimental tires with respect to existing production steel belted radials have been obtained,[7] we concluded that the baseline value of .010 is entirely realistic for 1985 production tires.

Improvements beyond the .01 value would have to be justified in terms of cost. At nominal fuel prices, the present value of the fuel consumed over the life of the baseline hybrid (under the assumptions described in (1)) is about $3000. Consequently, the reduction in fuel expenditure associated with each 10% reduction in rolling resistance coefficient is about .05 (3000), or $150. For the size tires used on the hybrid, the total retail investment on tires over the life of the car will be about $1000 (assuming about 70,000 km life from a set of steel belted radials); so a 10% improvement in rolling resistance should not result in an increase in tire cost/mile of more than about 15%. As an example, a tire whose rolling resistance is 10% less but whose life is 20% less would not be an economical proposition.

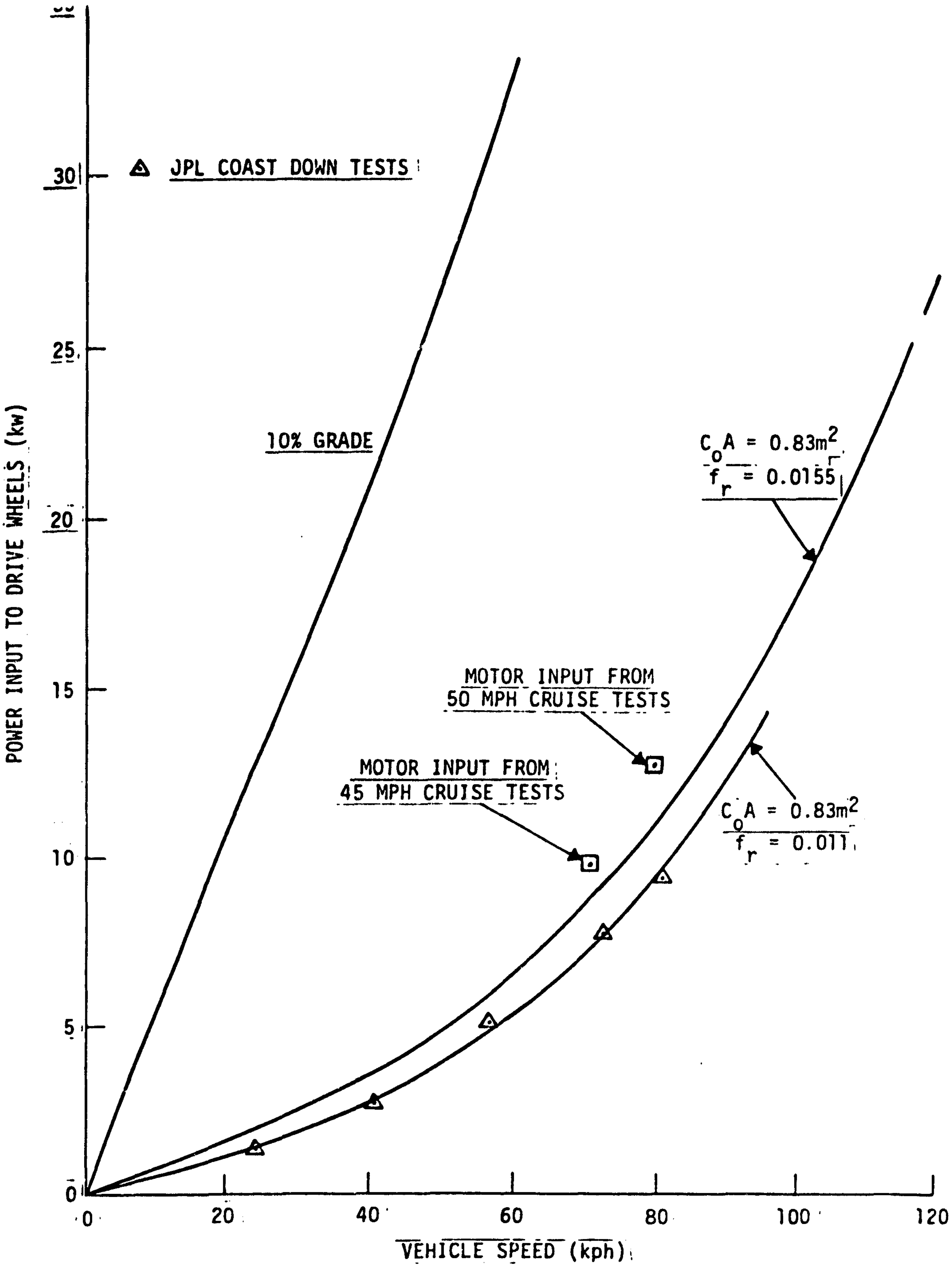


Figure 3-33 Electric Rabbit Load Power Requ... ments

Drag Coefficient x Frontal Area

The influence of the C_DA product on fuel consumption is about -.4, not a great deal less than that of rolling resistance. The reason for this is that a large part of the fuel consumption of the hybrid occurs on days with a lot of travel (since on the low travel days, it makes heavy use of stored energy). On these long travel days, there is a lot of highway travel; and under these conditions, aerodynamic drag represents a significant energy expenditure (see Table 3-3). As a result of this, the C_DA product is more important, in relative terms, for the hybrid than for a conventional vehicle. In absolute terms, it is not. In other words, a given reduction in C_DA will result in a larger total fuel saving for a conventional vehicle; however, this represents a smaller fraction of total fuel consumed than for the hybrid.

The baseline value of C_DA was .872 m^2, corresponding to a drag coefficient of .4 and a frontal area of 2.18 m^2 (23.5 $ft.^2$). We see no evidence that C_D values much lower than .4 are likely to be achieved in the 1985 time frame on full-sized sedans (except in subscale wind tunnel tests). Much depends on the front body contours, and these depend to a great extent on the engine/motor/controls package. If this permits lowering of the hood line, then we might see numbers in the .35-.37 range, i.e., about 10% lower than the assumed value. In terms of dollars, such a reduction would be worth about $120 in fuel savings over the life of the vehicle.

Vehicle Test Mass

The influence coefficients of vehicle test mass on fuel economy and 0-90 time are, respectively, about -.9 and 1.0. For reasons we have already discussed, the influence on energy consumption is much smaller (about .2). The weight influence on fuel economy for the hybrid is similar to that for a conventional car; however, due to its much lower fuel consumption to start with, it means much less in absolute terms for the hybrid. Again considering nominal fuel prices, a 10% decrease in vehicle mass means about a 10% reduction in fuel consumed, with a present value of about $300. On a strictly economic basis, this means that the retail price of the car should not increase by more than $1.35 per kilogram of weight saving, or about 60¢/lb. This comes down to a manufacturing cost increment of 68¢/kg (30¢/lb) under the JPL assumptions regarding the relation between manufacturing and retail price, or 1.08 per kilogram (49¢/lb) if the increment is passed on to the consumer at the bare minimum required to cover costs. It must also be remembered that the life cycle cost estimation procedure being used weights fuel consumption heavily due to the high mileage assumed. For the first owner of the car, who is unlikely to put as much mileage on the car as we have assumed, the fuel savings are less and the price per kilogram which is justifiable on the basis of those fuel savings is lower than the numbers quoted above.

From the manufacturer's standpoint, weight savings are of significance only if they permit him to lower a car's inertia weight and if the fuel economy the car starts with is low enough so that the change in inertia weight class and resultant fuel economy increment is significant in improving the manufacturer's CAFE. (The difference between the effects of making changes in high mileage and low mileage cars on CAFE was discussed in the Task 1 report,[1] pp. 48-49.) Although the hybrid is in a high inertia weight classification, it is a 35-40 mpg vehicle; and consequently, improving its mileage further does not mean a whole lot to the manufacturer's CAFE. In this respect, the hybrid is equivalent to a subcompact car in terms of its effect on his CAFE; and the way to use such cars to improve CAFE is to sell them at acceptable prices rather than attempt to extract the ultimate fuel economy through the use of high cost techniques that must also be passed on to the ultimate consumer. The hybrid will have a substantial price increment over a conventional car which will tend to restrict its market share; a manufacturer would obviously try to keep this increment to a minimum to avoid restricting that market any more than is absolutely necessary.

It comes down to a question of where the manufacturer (and, eventually, the consumer) puts his money. If he elects to stay with a conventional vehicle design, then weight reduction becomes extremely important in reducing his CAFE, and spending money on exotic materials may become worthwhile for him. On the other hand, if he elects to introduce a hybrid, that step alone can get him where he needs to be in terms of fuel economy; increasing his (and the consumer's) expenditure beyond that step does not make a whole lot of sense.

On the basis of these considerations, we came to the conclusion that, at the most, the same weight reduction techniques used in 1985 production conventional cars would be used in the hybrid. For this reason, we rejected such concepts as an all-composite car because they are simply not going to happen in the production world of 1985, and approached the weight reduction problem from the standpoint of investigating the economics of making material substitutions to a conventional car. A complete discussion of this area will be found in Section 3.6.2.

Sensitivity Analysis

Variations in fuel prices affect somewhat the cost tradeoffs discussed on the previous portions of Section 3.3, whereas, the effects of variations in electricity prices are almost nil. These fuel price effects are as follows:

- Rolling Resistance: At +30% fuel prices, a somewhat higher investment is justified in lower rolling resistance tires. In this case, the present value of the fuel consumed over the life of the hybrid is about \$3800, and the reduction in fuel expenditure associated with a 10% reduction in rolling resistance is about \$190; i.e., another \$40 can be invested in tires. A corresponding increase in tire cost/mile of up to 19% can be justified by a rolling resistance reduction of 10%. If fuel prices decrease by 30%, however, a cost increase of only about 10% is justifiable by a 10% rolling resistance reduction.

- Aerodynamic Drag: At the high and low values of fuel prices, the fuel savings brought about by a 10% reduction in drag would be worth about \$160 and \$80, respectively.
- Vehicle Mass: At the high fuel price level, the price increase per kilogram of weight saved which can be justified by the savings in fuel costs rises to about \$1.80/kg. At the lower level, it decreases to about \$1/kg.

3.4 Affects of Propulsion System Parameter Variations from Baseline

Propulsion system parameters which were investigated were the following:

- Heat engine power rating (SKALE)
- Final drive ratio (DRATIO)
- Battery type
- System voltage

In addition, several variations in control strategy were also investigated. It will be noted that variations in motor power rating and battery weight were not investigated except insofar as changes in these parameters were appropriate when considering batteries other than lead-acid. The reason for this is that the heat engine power fraction and battery weight fraction for each of the three battery types combined were localized fairly well in the system level studies for a performance level corresponding to the JPL minimum acceleration requirements. The system level studies also made it very clear that to minimize life cycle costs, the electric portion of the hybrid system should be minimized. Higher levels of performance can be achieved much more cheaply by increasing the heat engine power or changing gearing than by increasing the electric motor rating and corresponding battery size, at a very modest penalty in fuel consumption. Consequently, variations in motor rating and battery weight for a given battery type were not considered.

Heat Engine Power Rating

The effects of changing the heat engine power rating, within ± 10% limits, from the baseline of 53 kw, are summarized in Figure 3-34. The influence coefficient on fuel economy is about -.3, on acceleration time about -.8, and on wall plug energy consumption, negligible. Thus, to reduce the 0-90 kph acceleration time by 14% from 14 sec. to 12 sec. (which is more in line with current norms), would require about an 18% increase in heat engine displacement from 1460 cc to 1720 cc, with a concomitant decrease in fuel economy of about 5.4%. As pointed out in the Task 1 report,[1] the JPL minimum acceleration performance standards imply gradeability which is adequate from a safety standpoint; however, this acceleration performance is certainly in the bottom 5th percentile of current production vehicles, and we feel that it would be desirable to improve it somewhat. In order to make a decision in this area, however, it is necessary to include a discussion of gearing.

Final Drive Ratio

The effects of varying final drive ratio from the baseline value of 4.1 are summarized in Figure 3-35. The influence coefficients

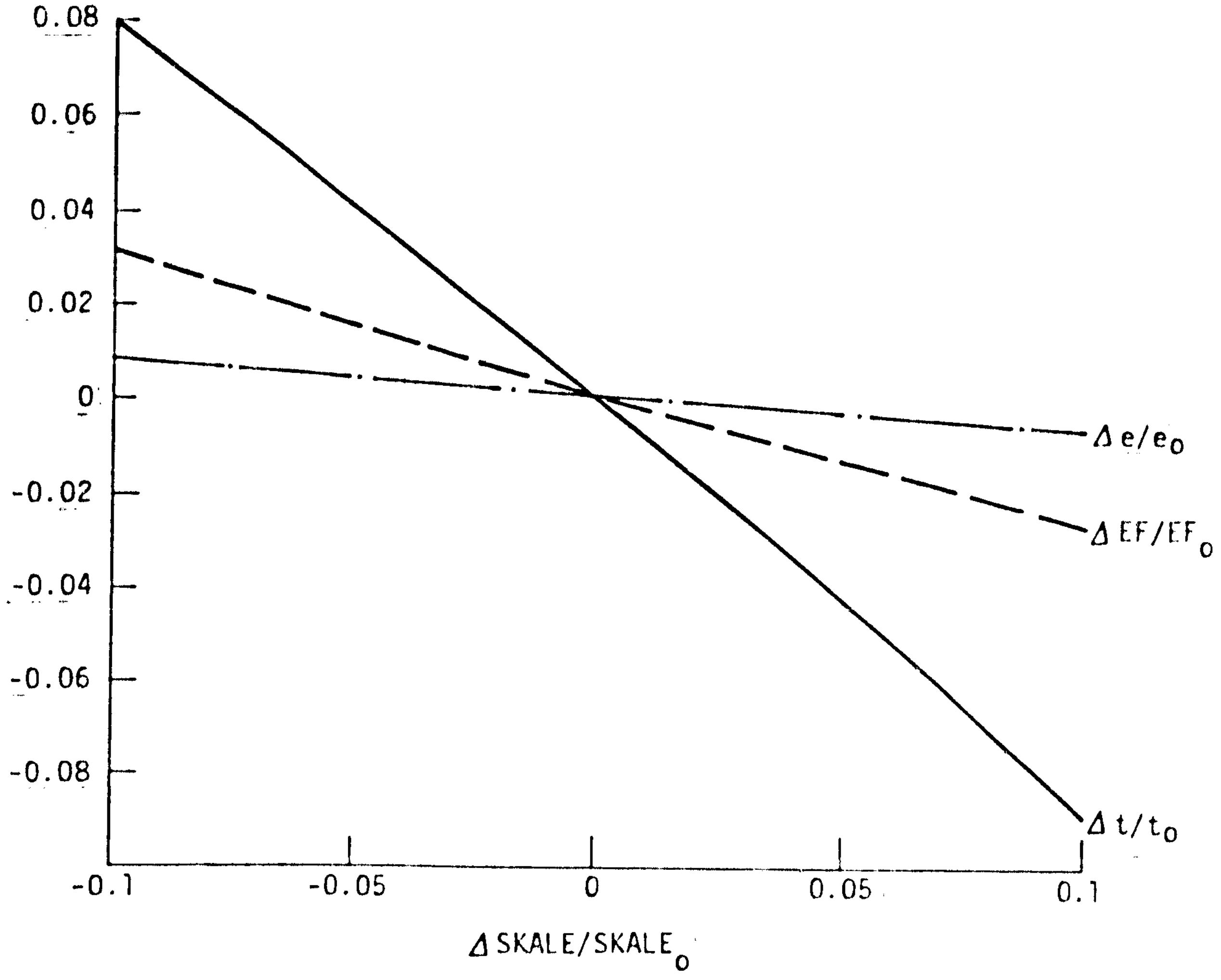


Figure 3-34 Effects of Change in Heat Engine Power Rating

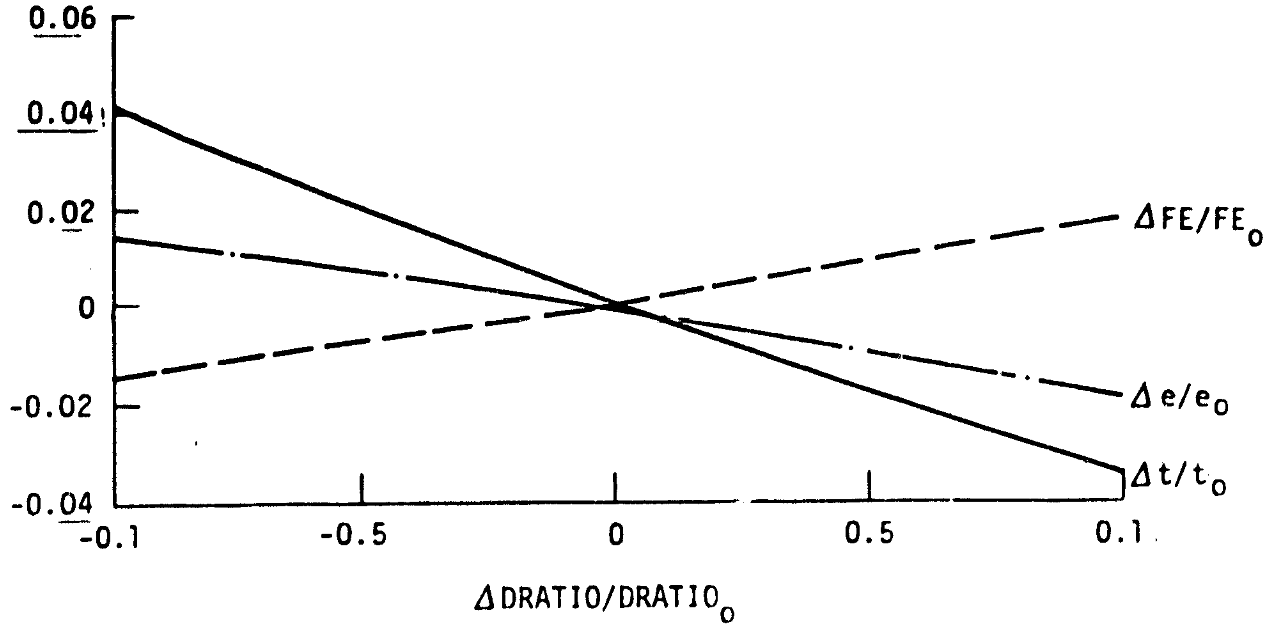


Figure 3-35 Effects of Change in Final Drive Ratio

are about .17 on fuel economy, -.16 on energy consumption, and -.32 on 0-90 kph time. The fact that fuel economy increases and energy consumption decreases with an <u>increase</u> in final drive ratio is surprising; however, this indicates that the baseline definitely has too 'tall' gearing, and as a result is spending too much time in the stop and go cycles in 1st gear in which the transmission efficiency is lower and the torque convertor is not locked up. This effect apparently outweighs the improved high gear efficiency which results from the higher engine loading and lower bsfc with the tall gearing. Because of the small engine size, the engine is loaded heavily in highway cruising; and minor reductions in final drive ratio do not affect the bsfc in high gear enough to offset the penalties associated with the additional time spent in first gear.

Consequently, we came to the conclusion that the final drive ratio should be increased over that of the baseline. In doing this, there are two possibilities: reduce the heat engine size at the same time to hold the performance constant, or hold the heat engine output constant and take the additional performance which the shorter gearing provides. Figure 3-36 shows a plot of what happens when the first alternative is chosen. Plotted are the relative changes in fuel economy and energy consumption against the relative change in heat engine power rating. The final drive ratio increments which are required to keep the same 0-90 kph time are also shown on the horizontal scale.

These curves indicate that a point of diminishing returns is reached at a final drive ratio of about 6:1 and a heat engine power

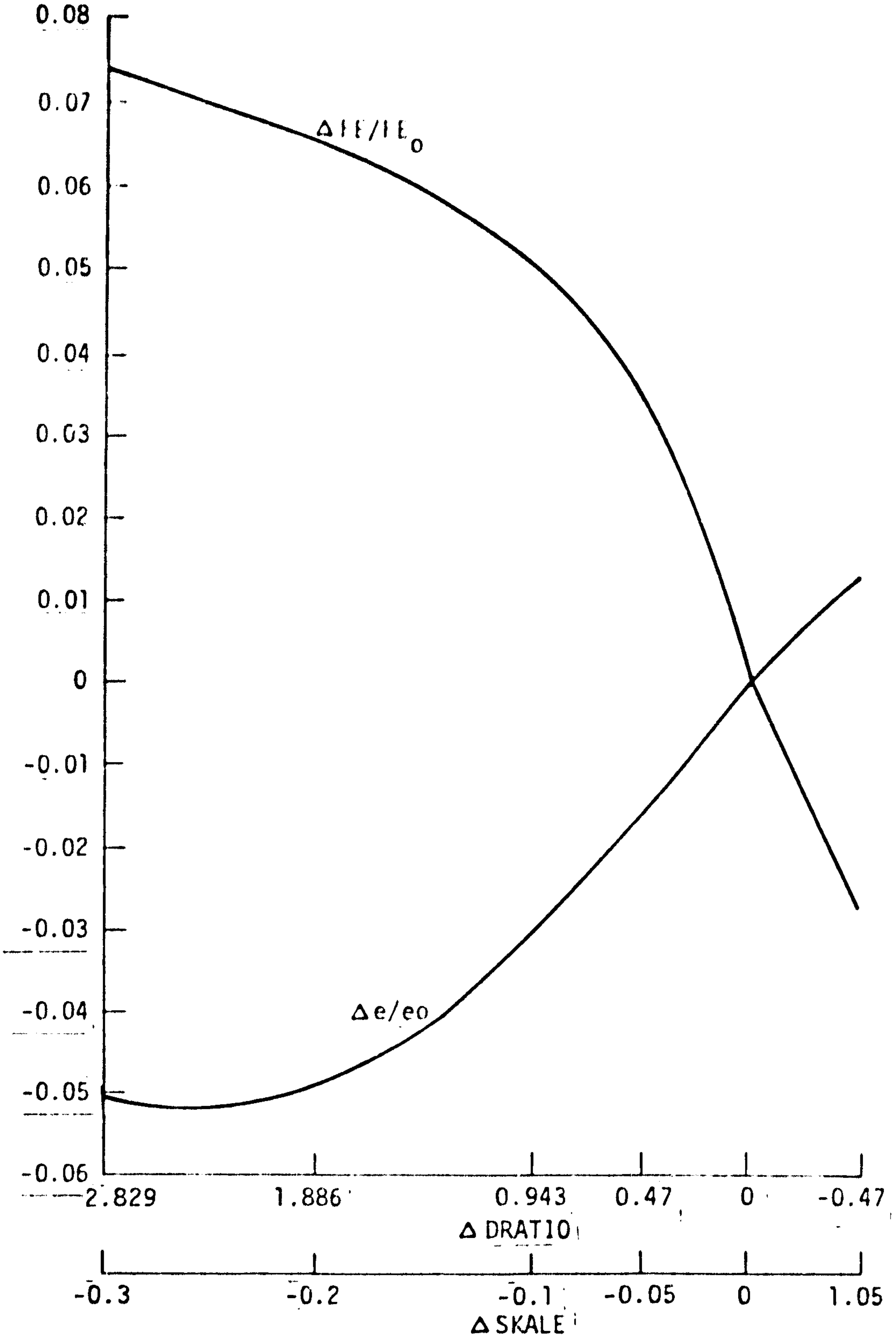


Figure 3-36 Effects of Combined Change in Engine Size and Final Drive Ratio (Constant Performance)

output about 80% of the baseline, or 42.4 kw. The difficulty with this is a very high engine rpm at normal road speeds. The baseline gearing provides nearly the same rpm at a given road speed as in the VW Rabbit. Engine speeds under cruising conditions much higher than this would be, we believe, unacceptable to the buyer of a full-size American car since such a buyer is used to a total lack of mechanical 'busyness' at normal cruising speeds. A better approach would be to increase the numerical axle ratio and provide an overdrive fourth gear to maintain the appropriate unflurried feel. Such an approach would also be beneficial from the standpoint of gradeability at highway cruising speed. If the final drive ratio was changed from 4.1 to 5.125 without changing the heat engine output, the following would happen (refer back to Table 3-4 for the baseline hybrid gradeability characteristics):

- The requirement to climb a 3% grade at 90 kph for an indefinite period could be met in 3rd gear instead of 2nd, at a more comfortable engine speed of about 4000 rpm instead of 4700 rpm.
- The requirement to climb a 5% grade at 90 kph for 20 km could also be met in 3rd gear. The required battery output would be 11.3 kw, which could be sustained for .27 hr. or 24 km while the battery depth of discharge increased from the nominal discharge limit of .6 to .9.
- The requirement to climb an 8% grade at 85 kph for 5 km would still require 2nd gear; however, the required battery

output power would drop to 17.6 kw, which could be sustained for .12 hr or 10.2 km in going from .6 to .9 DOD.

- Climbing an 8% grade at 65 kph would require 2nd gear instead of first, since the engine speed in first gear would be too high. For this case, about 4.9 kw would be required from the battery, which could be sustained for .9 hr or 58.5 km in going from .6 to .9 DOD. This corresponds to an elevation change of 4680 m (15,000 ft.), which is well in excess of the elevation change over any reasonable distance for any highway in the country. Ergo, the requirement to be able to climb an 8% grade at 65 mph for an indefinite distance would still be met.
- The requirement to climb a 15% grade at 50 kph for 2 km would still use first gear; however, the battery output required would drop to 11.3 kw, which could be sustained for .27 hr (13.5 km), well in excess of the requirement.

If the engine power rating is dropped by about 10% to keep a constant 0-90 kph time, however, much of the gradeability improvement goes away. For example, in the 90 kph/5% requirement, in 3rd gear the battery output required would go from 11.3 kw up to 16.9 kw, which could only be sustained for .176 hr even if the battery was allowed to go to 1.0 DOD from its starting point of .6. This does not meet the requirement; so 2nd gear would be required, as in the case of the baseline, except now an engine speed of about 5800 rpm would be needed.

Because of this, and because we feel that the performance of the baseline hybrid is a bit too marginal for the class of vehicle we are considering, we feel that a better approach would be to go to a higher rear end ratio without downsizing the heat engi .e, and add an overdrive ratio to the transmission. This would provide a slight fuel economy improvement, better acceleration performance, and a much better combination of gradeability and lack of fuss at highway speeds. The ratio of 5.125, which was just discussed, in combination with an overdrive ratio in the .7-.8 range looks like a good compromise. Further discussion of the alternative of a 4-speed overdrive transmission will be found in Section 3.5.5.

Control Strategy Variations

As discussed earlier, the control strategy utilized for the baseline hybrid made decisions regarding the operation of the heat engine and electric motor based on two variables - system power demand and input speed to the torque convertor (or, equivalently, power and torque). This resulted in high continuous battery output in Mode 1 in highway driving. To cut this output back to a more reasonable value, a modified control strategy was tried in which the heat engine was called on to handle the entire system demand if the vehicle speed was above a certain value. The value used was 20 mps (72 kph, or 45 mph). This change resulted in an increase in fuel economy from 16.75 km/l (39.4 mpg) to 17.4 km/l (41 mpg), or about a 4% improvement. Average battery output on the highway driving cycle was reduced to 5.51 kw, a substantial improvement and much more in accord with the sustaining power capability of ISOA batteries.

Yearly wall plug energy output was virtually unchanged (increase from .212 to .214 kw-hr/km).

Up until this point, the transmission shift logic used was similar to that of a conventional transmission: a decision to upshift or downshift is made on the basis of transmission input (torque convertor output) speed and throttle opening. However, no distinction was made in determining the shift points, between heat engine on and heat engine off conditions, or between Mode 1 and Mode 2 operation. This resulted on the following: With the part throttle upshift points set low enough to provide good fuel economy on Mode 2, the closed throttle downshift points had to be so low that regenerative braking was not too effective. Since the heat engine is always off when decelerating at closed throttle, it makes sense to set the shift logic under these conditions solely on the basis of the motor characteristics to provide effective regenerative braking. The shift logic with the engine operational (accelerating and cruising) could still be based on keeping the engine bsfc as low as possible.

With the incorporation of this change (along with the previous change to include vehicle speed sensitivity), fuel economy took another 6% step upward to 18.5 km/l (43.4 mpg). Wall plug energy consumption dropped slightly to .205 kw-hr/km.

Since neither of these modifications to the control strategy involve significant costs, they were included for subsequent investigations. Thus, subsequent comparisons will involve a somewhat higher level of fuel economy as a baseline than that shown in Table 3-2.

The control strategy described above is not necessarily optimal; the optimization process will continue during the Preliminary Design Task. However, it does bring us to the conclusion that the strategy must be sensitive to the power demand and both system output (torque convertor input) speed and vehicle speed. Moreover, the transmission shift logic must differentiate between engine on and engine off conditions.

Variations in Battery Type

In defining the cases to consider in assessing the effects of substituting nickel-iron or nickel-zinc batteries for the baseline lead-acid batteries, we considered the directions indicated by the system level tradeoff studies; i.e., the heat engine power fraction should be larger for nickel-iron batteries than for lead-acid, and highest of all for nickel-zinc; and the reverse relationships should hold for battery weight fraction. The cases considered, including the baseline, are shown in Table 3-6 . The same 53 kw heat engine was used for all three cases; thus, the increased heat engine power fraction resulted from the decreased motor power needed to maintain the same acceleration requirement with a reduced vehicle weight. The reduction in vehicle weight takes into account the reduction in battery weight, along with a 20% weight propagation factor.

These cases are not to be considered as optimum for the battery types in question; they simply provide a point from which to draw some preliminary conclusions and to indicate directions for change.

A battery discharge limit of .6 was used in all cases; based on the results of the system level tradeoffs, this appears to provide

Table 3-6 . PARAMETERS FOR ALTERNATIVE BATTERIES

Parameter	Lead-Acid (Baseline)	Nickel-Iron	Nickel-Zinc
Battery weight (kg)	355	270	210
Nominal battery capacity (kw-hr, 3 hr rate)	14.2	13.5	14.7
Maximum motor power (kw)	30	26.8	24.5
Vehicle Curb weight (kg)	2080	1978	1906
Heat Engine Power Fraction	.639	.664	.684
Battery Weight Fraction	.171	.140	.110
Battery OEM Cost	$710	$1012	$1102

a reasonable compromise between battery life and fuel consumption, and also provides a reasonable reserve of energy to draw on to meet gradeability requirements and perform successive high speed pass maneuvers.

The specific energy vs. specific power characteristics assumed for the nickel-iron and nickel-zinc batteries are shown in Figure 3-37 along with the baseline lead-acid. Assumed life characteristics are shown in Figure 2-11.

Results are summarized in Table 3-7, along with those of the baseline case. Fuel consumption for the nickel-iron and lead-acid cases are the same; energy consumption for the nickel-iron battery is slightly lower. This results from three factors:

- Overall energy consumption of the vehicle with nickel-iron batteries is slightly lower due to the lower curb weight.
- The nickel-iron battery has slightly lower total energy capacity (nominal 13.5 kw-hr vs. 14.2 kw-hr for the lead-acid battery); since a discharge limit of .6 is used in all cases, it means that generally less energy is extracted from the nickel-iron battery in a day's driving.
- In accordance with ANL goals for the three battery types, a higher energy efficiency (%) was assumed for the nickel-iron battery than for the other two types.

The nickel-iron case showed almost the same energy consumption as the baseline, with about 7% better fuel economy. In this case, because the nickel-zinc battery has slightly higher capacity than the lead-acid, most of the reduction in total energy consumption

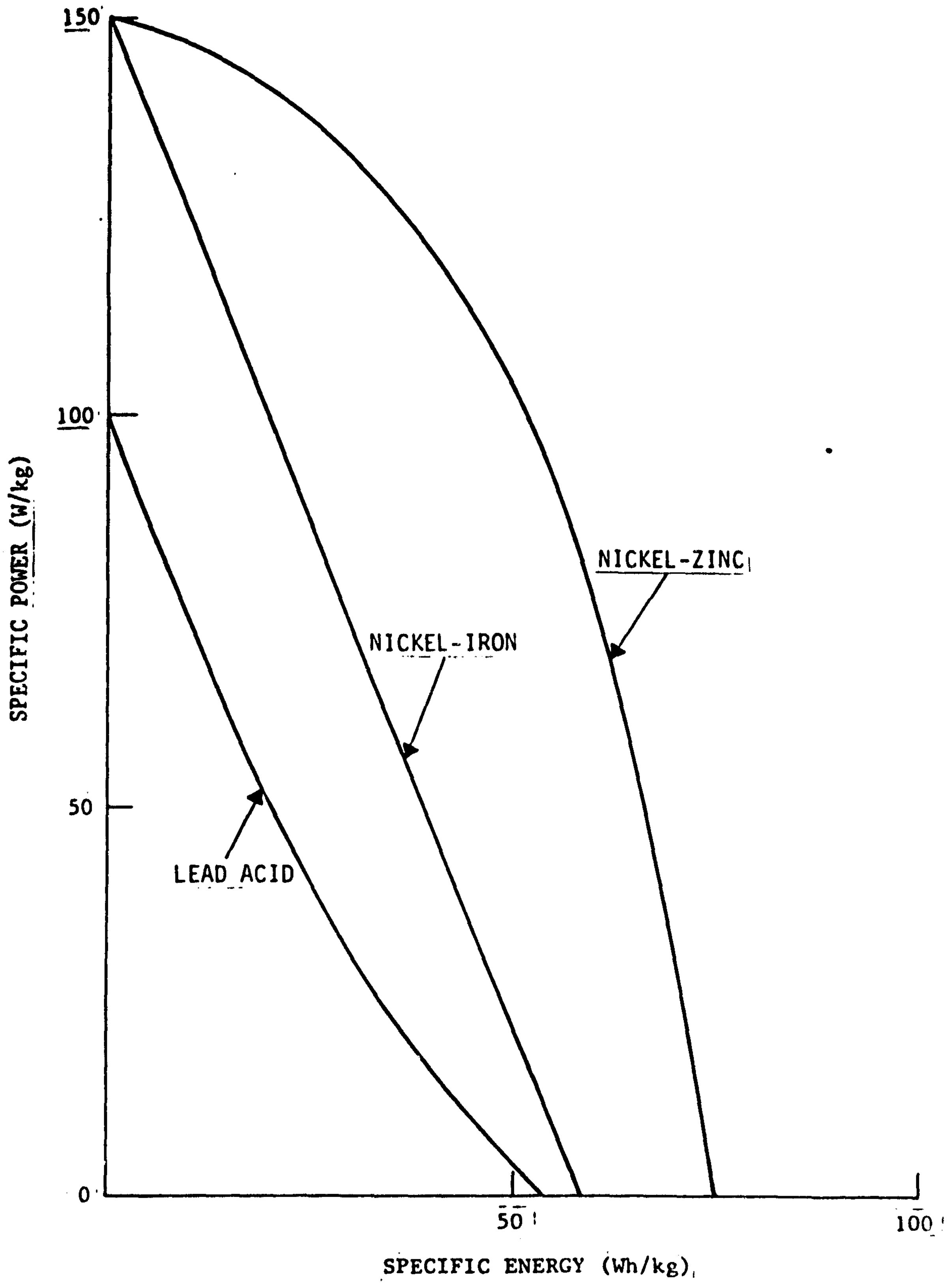


Figure 3-37 ISOA Battery Characteristics

Table 3-7. COST FACTOR FOR ALTERNATIVE BATTERY TYPES

	Lead-Acid (Baseline)	Nickel-Iron	Nickel-Zinc
Fuel Economy (km/l)	18.5	18.5	19.8
Energy Consumption (kw-hr/km)	.205	.167	.201
Projected Battery Life (km)	67000	125000	36000
Battery Cost/Life (¢/km)	1.06	0.81	3.06
Δ Manufacturing Cost* (over baseline)	-	+$230	+$193
Δ Life Cycle Cost* (¢/km) (over baseline)			
- Retail = 2 x OEM	-	-0.4	+2.9
- Retail = 1.25 x OEM	-	-0.5	+1.7

* Includes adjustment in manufacturing cost for lower vehicle weight, etc.

caused by the lower vehicle weight gets reflected in the fuel consumption, rather than energy consumption.

The battery life figures shown are somewhat conservative (assuming the ANL goals are achieved), because they were computed under the assumption that the batteries are discharged to the discharge limit of .6 every day the vehicle is driven; in actuality, there are days during the year when the battery DOD is less than .6. However, the number of days when the DOD is substantially less than .6 is small, so the estimate based on a uniform .6 DOD is probably not bad. A factor which would tend to compensate for this conservation is the cycling above and below the battery discharge limit which occurs on Mode 2 as a result of accelerations and, particularly, climbing grades.

All the battery models used and the estimates which result from these are based on the ANL goals for ISOA batteries, and on the assumption that these goals are all equally probable of attainment. If we put blinders on to any considerations other than ISOA battery goals, then the conclusion to be drawn from Table 3-7 is clear; and it is the same conclusion drawn from the life cycle cost studies done during the system level tradeoff studies: economically, the nickel-iron battery has an advantage over lead-acid; and nickel-zinc is a rather poor third. For the nickel-zinc battery to be competitive, the weight of batteries in the hybrid would have to be reduced from 210 kg to about 70 kg. At this weight, the electric motor would have to come down to about 10 kw peak output, the heat engine would grow substantially, and fuel economy would drop substantially as a result of the increase in heat engine size and of the reduction in potential utilization of wall plug electricity due to the reduced battery capacity. It is clear from the results shown in Table 3-7 that there would have to be gross disparities between the probabilities of achieving the ANL goals for the different battery types in order for nickel-zinc to be competitive in terms of cost.

To come to a final decision regarding the type of battery to be used in the near term hybrid, it is necessary to take off the blinders with respect to the ANL goals and make a critical appraisal of the battery development situation relative to the attainment of these goals, and in light of the timing constraints of the Near Term Hybrid Vehicle Program. This appraisal is underway and will be completed during the preliminary design phase. At this point, our ranking of the three battery types would be as follows:

1. Nickel-iron
2. Lead-acid
3. Nickel-zinc

The preference for nickel-iron over lead-acid is based not only on the potential for achieving a lower life cycle cost, but also on vehicle packaging considerations. A nickel-iron battery pack of the weight indicated in Table 3-6 could be packaged more neatly and compactly than the more bulky lead-acid battery. The better specific energy vs. specific power characteristics of the nickel-iron battery at high power levels (Fig.3-37) also provide more margin in meeting gradeability requirements (see Section 3.2.2).

During the preliminary design phase, the ranking described above will be reviewed based on our continuing appraisal of the state-of-the-art and a final decision will be made as to battery type, and a battery manufacturer will be selected to work with for the duration of this phase and subsequent phases.

System Voltage

The studies described in the previous sections did not explicitly consider system voltage. The motor used for the baseline vehicle represents the best of currently available technology in the range of power ratings required for the hybrid, and it has a design center of 130V (nominal 144V battery pack). In using this motor, we have essentially assumed that the ISOA goals could be met with a battery pack designed for this nominal voltage. We shall now examine the validity of this assumption and the tradeoffs involved with respect to system voltage.

In general, increasing battery voltage while keeping the same physical constraints on the battery means a smaller and less efficient cell design; cell connectors and partitions become a larger percentage of the total battery mass and specific energy drops. Now, in the case of lead-acid batteries, the volume and weight assumed for the baseline hybrid correspond approximately to 12 modules of the same size as golf cart batteries, which is the module size for which the ISOA lead-acid battery development is being carried out. Thus, the battery weight and volume would correspond to a 72V system, rather than 144V. In an attempt to ascertain the voltage tradeoffs involved, both the motor manufacturer Siemens and battery manufacturers were asked to estimate the differences in their products at voltage levels of 72 and 144V.

With respect to the motor, the response was that decreasing the nominal battery voltage from 144 to 72V would do the following:

- Increase the motor OEM price by $200.
- Decrease the typical operating efficiency range to 78-82%. This corresponds to a reduction in average efficiency of about 4%.
- Increase motor weight by 6-7 kg.

The motor OEM price increase was computed on the basis of about 10,000 units per year; increasing this to 100,000 units per year would reduce the cost increase to about $160, assuming the usual logarithmic relationship between production volume and cost.

The battery manufacturers who responded were somewhat less definitive in terms of the magnitude of the effects of increasing system voltage from 72 to 144V. From the responses, we came to the conclusion that the specific energy would drop 10 to 20% at the higher voltage. Cost per kg would not change significantly; so, if the same weight and package size were maintained for each battery type, available energy would drop 10-20%, and the battery cost would not change significantly.

To assess the effects of these changes, the following series of runs with HYBRID2 were made to assess the influence of changing motor efficiency (which is the major operational effect on the motor of changing system voltage) and changing battery specific energy (the major operational effect on the batteries of changing system voltage). The results were as follows: For lead-acid and nickel-iron, reducing the battery specific energy 10% increased the fuel consumption by about 4.5% and decreased the energy consumption by a like amount. For nickel-zinc batteries, the corresponding numbers were about 3.5%. The difference between these cases apparently results from the different shapes of the specific power vs. specific energy curves for the different battery types.

If we assume the worst, i.e., a 20% decrease in battery specific energy associated with the higher voltage, we come up with about a 9% decrease in fuel economy and a 9% decrease in energy consumption. Referring back to Figures 3-6 and 3-7 , for the baseline case, this amounts to an increase in the present value of fuel consumed of about $250 and a decrease in energy value of about $160, for a net increase of $90.

On the other hand, if we reduce the battery voltage and take the slightly less efficient and more costly motor, the fuel consumption increases relative to the baseline by about 5% and energy consumption by 2%. The corresponding present values of fuel and energy consumed are about $130 and $30.

The conclusion reached is the following: If we stay with the approximate voltage implicit in the motor selection for the baseline system, and adopt a more realistic estimate of what we are likely to get in terms of battery specific energy at this voltage, we come up with a total cost penalty of about $90. If we lower the motor voltage to an appropriate value to get the ISOA battery specific energy, the cost penalty is about $320, without considering any added cost due to increased cable sizes, contactor sizes, and so forth.

The cost tradeoffs for the hybrid thus appear to favor the sacrifice of specific energy to obtain a higher voltage system, if one considers only the motor and battery. As far as the controller is concerned, it would be beneficial to keep the nominal system operating voltage down to about 120V to avoid having to go to more expensive, triple-diffused transistors in order to obtain a peak voltage rating which would be required at a system voltage in excess of 120V. (See discussion in Section 3.5.4) A 120V nominal system voltage would involve only a slight degradation in motor characteristics, and a slight improvement in battery characteristics when compared to a 144V system. Thus, the final adjustment of nominal system voltage can be made on the basis of controller economics; and on this basis, a 120V system was chosen.

Sensitivity Analysis

Variations in fuel and electricity will affect the tradeoffs discussed in the previous portions of Section 4.4, as follows:

- Heat Engine Power Rating
- Final Drive Ratio
- Control Strategy Variations

There is no significant impact on the discussion or the conclusions drawn in these areas.

- Variations in Battery Type

In the case of the nickel-iron batteries, the comparison in life cycle costs shown in Table 3-7 with respect to the baseline system is not affected by fuel prices, since the fuel economies for the baseline system and the nickel-iron system are identical. At +30% electricity price, the difference in the present values of the electricity consumed by the two systems increases from \$250 to \$300. The corresponding increment in the life cycle cost difference between the two systems is less than .02¢/mile; i.e., there is no significant change. Likewise, there is no significant change at -10% electricity prices. For nickel-zinc, the situation is similar: none of the data shown in Table 3-7 is significantly affected by fuel or electricity price variations.

- System Voltage

The tradeoffs in this area are affected as follows:

	+30% Fuel	-30% Fuel	+30% Electricity	-10% Electricity
(1) High voltage (20% decrease in battery energy)				
Δ Fuel Cost	325	175	250	250
Δ Energy Cost	-160	-160	-210	-140
Total	\$165	+ 5	+ 40	\$110
(2) Low voltage (4% decrease in motor efficiency)				
Δ Motor Cost	160	160	160	160
Δ Fuel Cost	170	90	130	130
Δ Energy Cost	30	30	40	25
Total	\$360	\$280	\$330	\$315

In short, the tradeoffs are still in favor of going the high voltage route.

3.5 Alternative Design Approaches

3.5.1 Use of Flywheels as Energy Buffers

In order to limit the instantaneous power output required from either or both of the heat engine and traction motor, a flywheel could be used to release energy during acceleration and store it during deceleration. There are theoretical advantages in doing this.

- The ability to store energy during deceleration is not limited by the power capacity of the electric motor/generator, or by the battery's ability to accept charge at a high rate (which is a function of its state of charge).
- The output of the battery can be load levelled so that it is nearly a constant current discharge. This is favorable in terms of maximizing the available energy from the battery at a given average discharge rate.

The disadvantages of using a flywheel as an energy buffer are of a practical nature. They include:

- High overall system complexity, in terms of both mechanical layout and controls.

- Some form of continuously variable transmission is required between the flywheel and the rest of the drivetrain for speed matching.
- Composite flywheels appear to be the only type which have a chance of providing acceptable energy density, and the status of technology of these devices appears to be highly tenuous relative to a 1985 production target.

Because of the potential advantages of an energy buffered system, we conducted a critical survey of the state-of-the-art flywheel technology to assess its applicability to the near term hybrid vehicle. An overview of the results of this survey follows.

Energy Storage

While flywheel concepts have advanced enormously in the last few years, the net available specific energy of a flywheel system is still less than that of a fresh lead-acid storage battery. At present, we expect 10 wh/lb from a storage battery, vs. perhaps 5 wh/lb or so for state-of-the-art flywheel systems. This applies to either isotropic or composite flywheels. In the future, composite flywheel systems may be improved to the range 10-25 net wh/lb, but by then there are likely to be storage batteries of 40 wh/lb or so. As a first approximation, then, we can say that flywheels do not have any apparent advantage over batteries from the viewpoint of storage capacity.

Power

Flywheels do have a large advantage over batteries when it comes to power handling ability. Even a relatively small flywheel could, in principle, deliver or accept hundreds of kilowatts over at least a short time. Moreover, high power does not diminish the available energy in flywheels; in fact, the quicker the drain, the less the rundown loss. By contrast, the energy capacity of batteries is significantly reduced at high drain rates. Thus, a flywheel used in conjunction with a battery may be a very complementary arrangement; peak loads can be handled with the flywheel. The technical advantage to the driver of a car with a flywheel-battery combination would be, in terms of performance and under some driving conditions, a range extension or possibly an overall energy saving. The latter two possibilities would come about through the gain in battery discharge efficiency with load levelling.

Service Life

All known types of storage batteries have rather limited lives. A battery may have a useful life of a few hundred recharge cycles, while a flywheel should be good for a million cycles or more. The aging properties of composite flywheels have not yet been experimentally verified, however.

Losses

Self-discharge rates for flywheels are still not very precisely known over a range of designs and conditions, but these losses may be higher than for batteries by a factor of 100 or more. Favored applications for flywheels will, therefore, be for uses characterized by prompt discharges and with a minimum of long idle periods. True, the flywheels energy can be discharged and sent to the battery; but this involves losses also.

Using or replacing flywheel energy will usually require an electric or mechanical transmission (or equivalent), also involving losses. A conspicuous need exists for a good, efficient continuously variable transmission, with a range of ratios on the order of 16:1 to permit the application of a flywheel in a vehicle with as wide a speed range as the hybrid. Thus far, no one has produced one which demonstrates all the required characteristics.

Cost and Effectiveness

A number of calculations have been done which show a range improvement or an energy saving when a flywheel is used in conjunction with a battery. We should know within a year or so (when the Garrett electric car is tested) whether this promise can be verified. A probable outcome is that the energy benefits are realizable for some driving schedules but not for others. A corollary would be a net benefit for some vehicle owners but not for others.

Because near term flywheel systems (at least) will have a relatively high 'technological density,' using expensive and high grade materials and components, the most difficult hurdle will be the economic one. Will the average auto buyer feel justified in the extra cost when compared with added benefits? The difference a flywheel would make in a hybrid vehicle, as far as the driver can readily perceive, is an available temporary boost in performance; and it is legitimate to ask, whether or not the difference in acceleration capability will perturb the thing being measured, namely, the energy consumption?

A study by Rockwell forecasted the manufacturing cost of Kevlar composite rotors in the range \$100-\$400 per kwh of rating, if production rates were 10^5 units/year. Housing, transmission, controls, and auxiliaries would presumably be extra. The same study showed \$75 per kwh for a projected, advanced battery.

Another study by the Lawrence Livermore Laboratory was in reasonable agreement with Rockwell, in finding the total cost per mile of flywheel equipped autos to be higher than conventional autos. This implies that energy cost savings would not be great enough to offset added purchase cost per mile.

Both studies examined heat engine/flywheel hybrids and heat engine/flywheel/battery hybrids, and found the former to be more nearly cost competitive than the latter.

Safety

With proper design and careful engineering, safe flywheel systems can be produced in the future for automobiles. Much more testing and evaluation will be needed (particularly with composite rotors) before the required engineering experience will have been accumulated, and this will take some years of time. Safe prototypes can be built now, but only for use in a laboratory environment or for supervised and carefully monitored operation. Production designs of the future would have to be safe under all conceivable conditions of climate, use, abuse, and vehicle mishap.

R & D Activities

The work of over 20 agencies and organizations involved with flywheel R & D was reviewed in preparing this summary. Both metal and fiber composite versions are being developed and demonstrated. Rockwell International and the AiResearch Manufacturing Co. (Garrett Corp.) are among the leaders in these two flywheel types, respectively. AiResearch, at Torrance, Calif., is now completing a prototype electric car for the U. S. DOE; and this car features a flywheel as a power augmenting device. Chassis dynamometer tests are underway now, and road test results should be available before the end of 1979. Such tests may verify the hoped-for energy saving benefits, and if so, under what driving conditions.

Appropriately, the Garrett car will receive much attention, being about the only testable machine of its type in this country. Yet, the flywheel industry is still in the stage of concept development; and many years of testing of experience remain before a manufacturer could seriously consider mass production.

The Process of Commercialization

The process of commercialization of new technology always takes many years. After reviewing the state-of-the-art, we have come to the conclusion that an adequate characterization of flywheel technology will take another five years. In other words, 1984 is about the earliest year that potential manufacturers could seriously consider a decision of whether or not to enter the production engineering stage. If the answer is affirmative, then pilot production (1000-10,000 units) might begin in 1988. This could be followed by limited production (20,000-50,000 units) in 1989 and full production in 1990 (at least 100,000 units/year each manufacturer).

Conclusion

Quantity production of flywheels, as major elements in an electric or hybrid automobile drivetrain, is not foreseen prior to around 1990. (Quantity production is defined as at least 10^5 units per year.) Given this long a lead time, prototypes in 1980 could not be very representative of future production designs, and a demonstration of such would not be instrumental in bringing about quantity production by 1985.

On the other hand, it might be ventured that without the testing of a number of well engineered models during the early 1980's, the earliest quantity production could well be delayed to beyond 1990. The conclusion is this: Present technology will support the construction of educational machines of great value, but such models should not be regarded as prototypes for mass production in 1985. As a consequence, a system using a flywheel as an energy buffer would not be a viable alternative for the near term hybrid.

3.5.2 Alternatives to Naturally Aspirated Gasoline Engines

Diesel

The diesel engine offers higher fuel economy than an Otto cycle engine. In passenger car use, prechamber diesels are the norm for reasons of smoothness, flexibility and low emissions. Most of the fuel economy benefits of such engines result from much lower fuel consumption under light load; the minimum bsfc under heavy load may not be more than 10% better than an Otto cycle engine. Consequently, the fuel economy advantage of a prechamber diesel over a good gasoline engine largely disappears when the engine is operated like it is in the hybrid; i.e., only under relatively high load.

Against the minor fuel economy improvement attainable by using a diesel in the hybrid must be weighed the following:

- The displacement of a naturally aspirated diesel would have to be about 40% larger than a conventional gasoline engine for the same output. Costs will consequently be higher, thereby compounding a problem which the hybrid already has. Because of the very good fuel economy of the gasoline engine hybrid and the small improvement obtained with the diesel, it would be very difficult to justify the added cost of the diesel on the basis of reduced fuel costs.
- Cold start characteristics are much worse than a conventional engine, and even when the engine is warmed up, the problem of starting up and delivering power almost instantaneously may be worse with a diesel than with a conventional engine.
- There are a great many unknowns regarding the diesel in the emissions area, relating primarily to NOx and particulate emissions. It is not clear at this point what standards will ultimately be applied to the diesel with respect to these emissions, and whether or by what means practical control to these standards will be obtained.

As a result of these considerations, we came to the conclusion that utilization of a diesel in the near term hybrid would not be desirable because of the added cost and development problems associated with only a small improvement in fuel economy (on the order of 10%).

Stratified Charge

Stratified charge engines fall into both open chamber and prechamber categories. None of the former are in production, although Ford's PROCO is close; however, the first production versions of this engine will be V-8 for larger cars (following the pattern of GM in introducing diesels). Like the diesel, the open chamber stratified charge engine obtains most of its fuel economy advantage in a passenger car from low fuel consumption at light load, although some benefit is obtained throughout the load range from the ability to operate at a higher compression ratio than a conventional engine. Consequently, its fuel economy advantage over a conventional engine, in the hybrid application, will be small. Also, like the diesel, these engines have a lower specific output and cost more to manufacture than a conventional engine; however, the penalties in these areas are not as severe as with a diesel.

Essentially, then, the situation with regard to the open chamber stratified charge engine is the following: If a manufacturer had available a complete line of such engines ranging from four-cylinder on up, and decided to offer a hybrid option in his large cars, he might elect to use his small stratified charge engine in the hybrid. If such an engine were not already in use in a small car line, however, he would be unlikely to develop one specifically for a hybrid application in preference to a conventional spark ignited engine.

For this reason, and due to the fact that Ford's current emphasis is on large PROCO engines and small four cylinder production

PROCO's are not in the offing, a PROCO or other open chamber stratified charge engine would not be an attractive alternative for the near term hybrid vehicle.

The prechamber stratified charge engine, as exemplified by the production Honda CVCC engine, has one major advantage, and that is having sufficiently low uncontrolled emissions to avoid having to rely on a catalytic convertor for emissions control, thereby, being able to use leaded fuels. Whether this advantage will continue to exist for tighter emissions standards is open to question. The engine has no advantage over a conventional engine in terms of fuel economy (in fact, appears to have narrower speed range over which it has low bsfc), and has lower specific output. Consequently, we saw no reason for choosing it over a conventional engine.

Turbocharging

Turbocharging offers the advantage of raising the maximum bmep of an engine without significantly affecting the bsfc at lower values of bmep. In short, on a fuel map, it spreads out the islands of low bsfc vertically. Consequently, for a given power rating, using a small turbocharged engine provides better fuel economy than a large naturally aspirated engine, in a conventional vehicle. The amount of improvement to be gained in a hybrid application, however, is less since, even with a naturally aspirated engine, the hybrid spends most of its time operating close to the minimum bsfc region.

Apart from the minor fuel economy benefit, there would be a problem of scale in attempting to use a turbocharged engine of the same peak output as that of the baseline (53 kw). Such an engine would probably be only 1000 cc or so in displacement, and there are virtually no modern engines to work with in this size range except for motorcycle engines, which lack the emissions control technology and low production cost associated with passenger car engines, as well as their durability. Under most driving conditions, motorcycle engines are even more lightly loaded relative to their peak output than passenger car engines, yet life between overhauls tends to be on the order of 50,000 miles, rather than 100,000, with a few exceptions.

As a consequence, we would regard turbocharging as an alternative (to increased engine size) method of obtaining higher performance than that provided by the baseline hybrid, at little or no fuel economy penalty, using the same engine size as in the baseline. Using it to downsize the baseline engine and keep the same performance level would not be particularly useful.

3.5.3 Alternatives to Separately Excited DC Motor

Three methods of motor control were considered in addition to the DC motor with separately excited field. These were

- Three phase AC motor/invertor
- DC (series field)
- DC (permanent magnet field)

A discussion of the advantages and disadvantages of each of these techniques, along with the baseline separately excited DC motor, follows.

Three Phase AC Motor/Invertor

This system offers many advantages in terms of the motor design. An AC motor is generally smaller, lighter, simpler, and therefore, cheaper than an equivalent DC motor. The inherent advantage in the AC motor is the elimination of the brush and commutator assembly. This is a limited life component which is relatively expensive. An AC motor uses a time varying input voltage to drive the rotor, while the DC motor relies on its commutator to produce a time varying voltage. Also, because of the much more common usage of the AC motor, the AC motor has been refined more extensively than its DC counterpart.

However, the AC motor requires a source of AC power. In a vehicle, this must be derived from the main battery, which is obviously DC. This invertor must convert DC battery power into AC power of the proper voltage, frequency, and waveform. This invertor must be capable of delivering full motor power (30-40 kw peak input).

Such an invertor would have to operate in a switching mode to have reasonable cost, size, and efficiency. In order to synthesize an AC voltage, several modulation techniques could be used, two of which are shown in Figure 3-38 . Pulse Position Modulation might be applicable if the drive circuit could be tailored to deliver a particular width pulse very efficiently. Usually, though, the large number of driver transitions per AC cycle would lower system efficiency by increasing dissipation. Pulse width modulation, on the other hand, has a constant number of transitions per AC cycle. This is generally a more efficient technique for AC synthesis. Pulse width modulation requires, however, a longer integration time constant

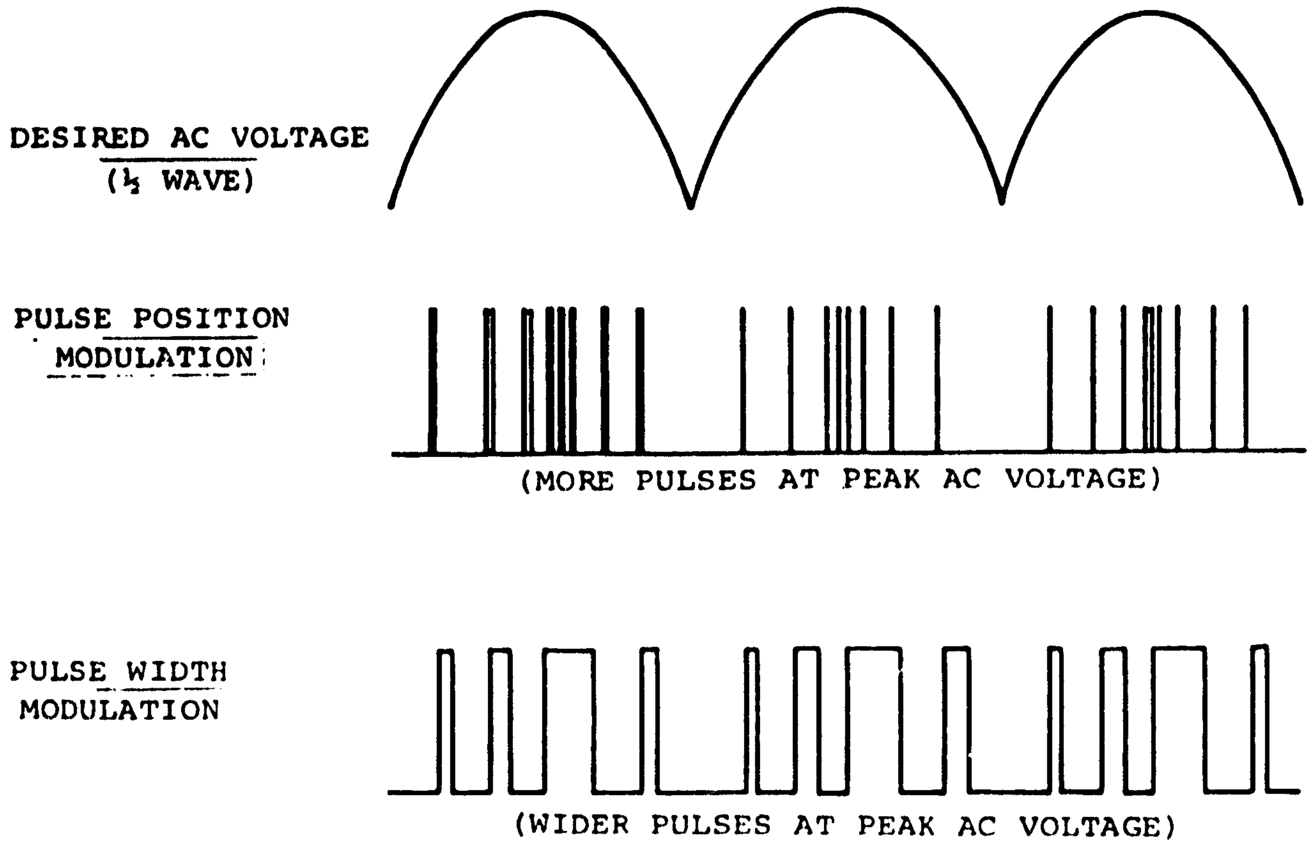


Figure 3-38 Synthesis of an AC Waveform

to achieve acceptable results. This is usually not a problem, however, because an AC motor is a highly inductive load with a relatively long time constant.

Unfortunately, although such invertors can, and have been built, they are relatively large and very expensive. A state-of-the-art 20 kw invertor using SCR's or power transistors could be 90 to 95% efficient if operated from a 120V system. This would still result in 1000 to 2000 watts of power to be dissipated. Even with forced air cooling, a large volume and weight would be associated with cooling alone. A state-of-the-art heat dissipation system consisting of a blower and ducted aluminum extrusions would weigh about 25 lbs and occupy about one cubic foot, exclusive of the actual invertor components such as drive circuits, switching elements, inductors, and capacitors.

In the near term, the switching elements would be extremely expensive. Although electric and hybrid vehicles may ultimately utilize AC drives, their implementation in production will have to await the development of much lower cost production methods for high power switching devices. We do not see this happening in time for this motor and control technology to be employed in a 1985 production vehicle. (See Section 3.5.4 for additional discussion on switching devices.)

DC Traction (series field)

The DC traction motor has been used in vehicular drive systems for nearly a century. Its speed/torque curve closely matches the requirements of many types of vehicles. A series motor develops maximum torque at stall, and as load torque decreases, speed increases. This is precisely what is required to accelerate a vehicle from a standstill. For this reason, these motors have found widespread use in trains, material handling equipment, golf carts, etc. In these applications, the required acceleration is relatively constant and well defined, so that the motor's speed-torque curve is well defined and can remain fixed.

For an on-the-road vehicle, however, the load demands can vary so much that a particular motor cannot handle these variations efficiently. By using an armature chopper, the controllability of the motor is increased, but efficiency suffers. Most of the inefficiency in a motor is a result of I^2R losses; that is, the motor losses increase in proportion to motor current squared. For a given power level, running a motor at reduced voltage (and, therefore, higher current) results in much higher losses. In addition to the losses in the motor, the armature chopper itself has losses. These losses result from voltage drop across the switching element in addition to switching losses during turn-on and turn-off.

Another way of altering the characteristics of a series motor is by switching parts of the series field in and out of the circuit. This form of field weakening is only practical if a few steps are needed. Series field switching requires large contactors capable of switching full motor current.

DC/Permanent Magnet

Although permanent magnet designs are usually considered practical in small motors only, integral horsepower motors have been built. Permanent magnet motors are inherently very efficient because field excitation is supplied by a magnet without consuming any power. For instance, a 20 kw motor typically requires about one kw of field power which does no useful mechanical work. In addition, the temperature rise caused by the dissipation of field power causes the armature to heat up, limiting the amount of power the armature can dissipate, and, therefore, limiting power output.

Another effect of dissipative field power is to raise the armature temperature. Thus, when operating a given armature at a given load, a hotter armature will have a somewhat higher resistance. As a result, the motor will have higher I^2R losses and overall armature circuit efficiency will be lower by as much as a few percent.

As a further illustration of how field dissipation affects motor performance, the ratings of General Electric's electric vehicle drive motors can be studied. GE offers a wide selection of motors, and this data can, therefore, be considered typical. Table 3-8 shows a comparison of the horsepower ratings of the entire GE line for conditions of blower ventilation (independent of motor speed) and 120 volts input. The differences in rated horsepower under full field and weak field conditions are negligible in the smaller motors, but reach 50% in the largest frame sizes.

Although inherently efficient at a given armature voltage and a corresponding narrow speed range, the permanent magnet motor suffers from the inefficiencies of the chopper design when both load and speed are varied over a wide range. Lowering motor speed is accomplished

Table 3-8 . MOTOR RATINGS AT TWO FIELD CONDITIONS (BLOWER VENTILATION)

	Full Field (Rated Field Current)	Weak Field (50% Rated Field Current)
BT2378	36	48
BT2376	36	48
BT2368	23.5	28
BT2366	24	27
BT2364	24	25
BT2348	17.5	28
BT2346	18	17.5

All Ratings in Horsepower

by reducing average armature voltage, and so chopper complexities and chopper losses limit the usefulness of the permanent magnet motor.

<u>DC - Separately Excited Field (Baseline)</u>

The separately excited DC motor is somewhat of a compromise in the manner in which the field excitation is supplied. The series DC motor has the field constantly in the armature circuit, and hence, field losses are proportional to the square of armature current. This results in the highest losses occurring at maximum output power. The permanent magnet motor has losses only in its armature chopper circuits, but these losses still increase with armature current.

A separately excited motor has maximum losses at its base speed where field excitation is maximum. However, field losses are independent of output power. Instead, field losses vary inversely with motor speed. Field current is reduced to increase motor speed, resulting in higher efficiency at higher speeds (neglecting rotational losses). In addition, the relationship of motor speed to field current is highly nonlinear, resulting in a large reduction in field current to produce a proportionately smaller increase in motor speed. Figure 3-39 , for example, shows the variation in base speed with field voltage of the Siemens 1GV1 motor as used in the SCT electric conversion of the VW Rabbit, at a motor voltage of 100V. In fact, in Figure 3-39 , a typical operating point of 3500 rpm is achieved with 2.4 amps, which is only 24% of maximum field power. Figure 3-40 shows efficiency plots of various versions of a GE BT2376

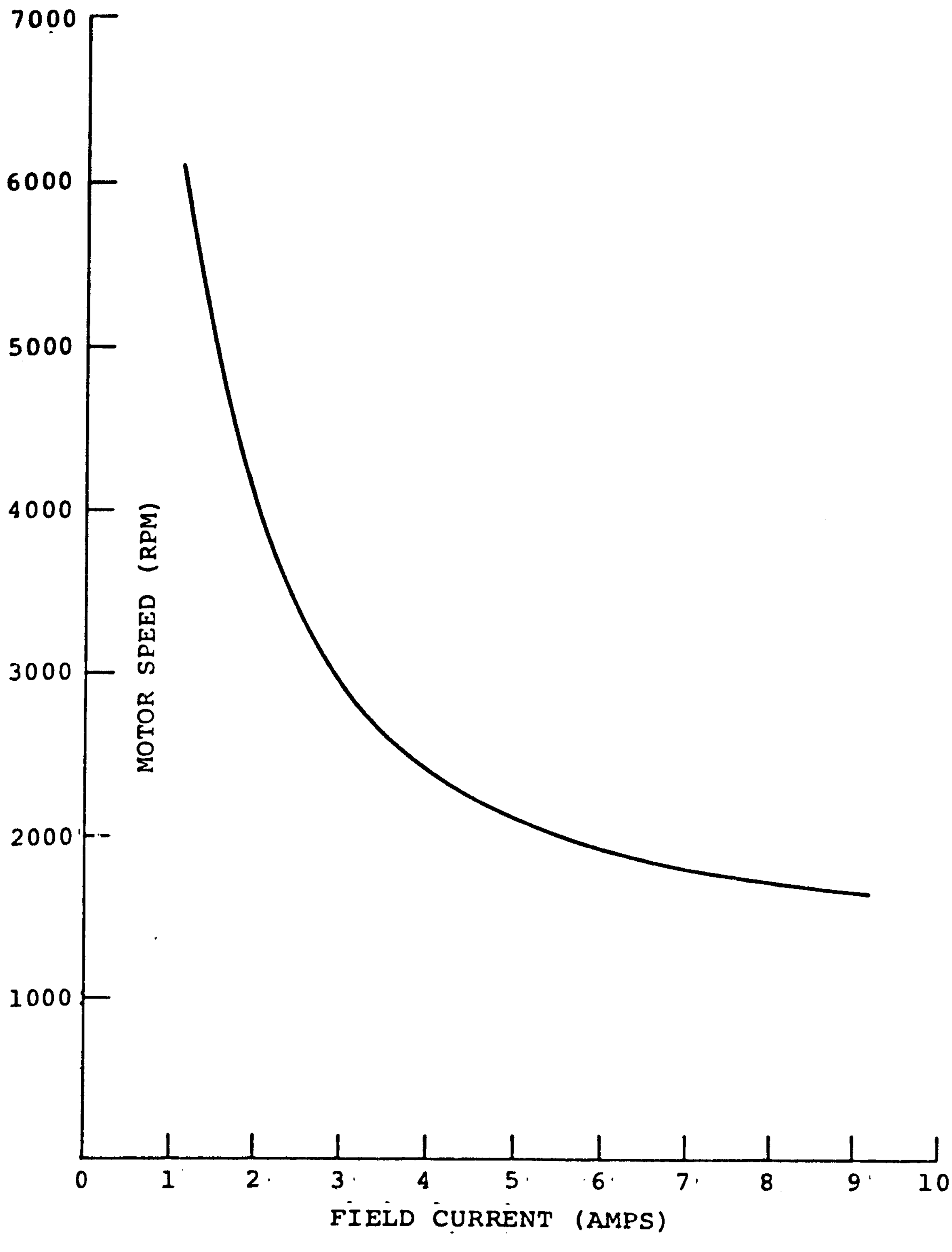


Figure 3-39 No Load Speed vs Field Current Siemens IGVI Motor
Armature Voltage = 100V

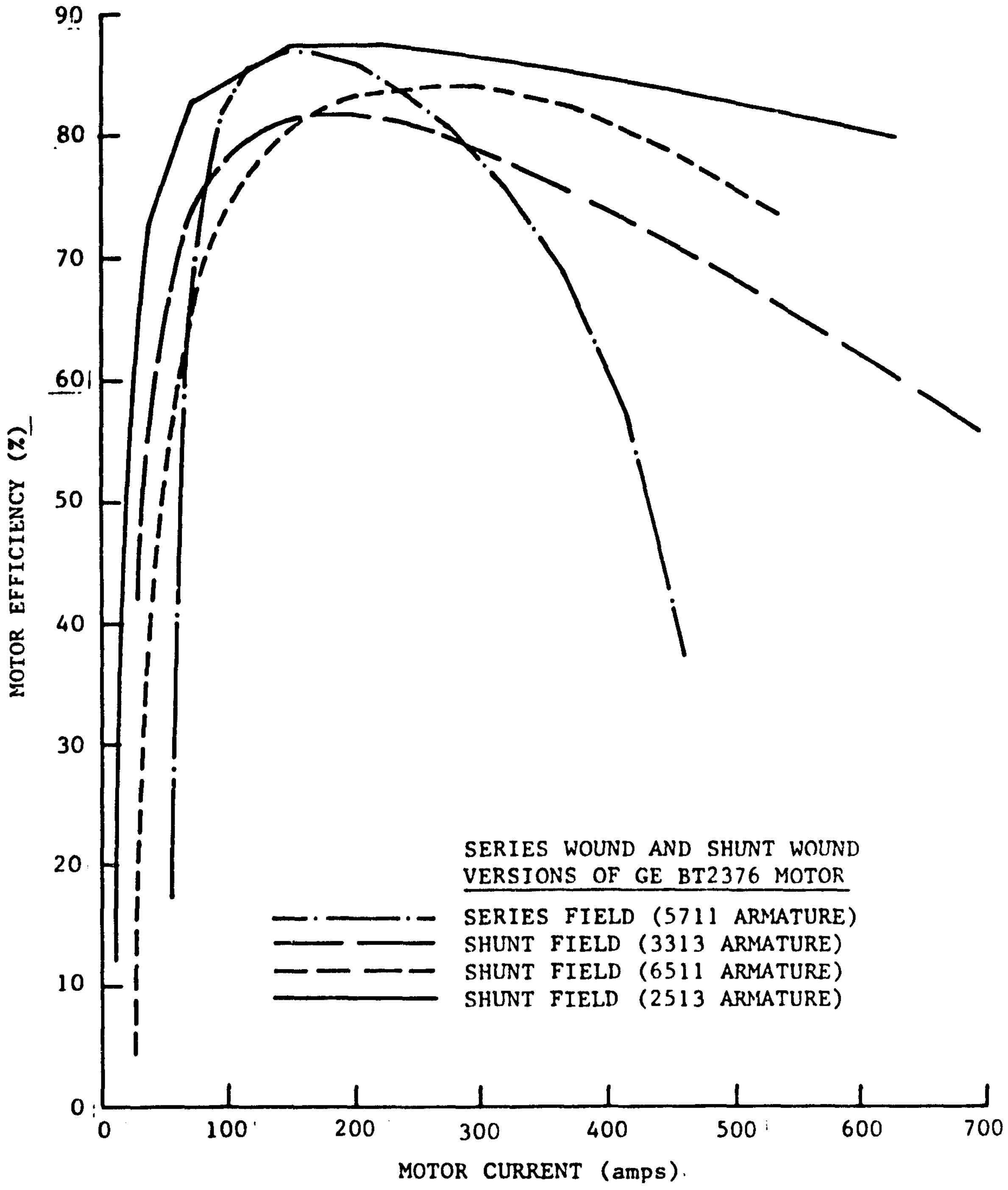


Figure 3-40. Typical Motor Efficiencies

motor. Note how the efficiency of the series version is substantially lower than the various shunt versions at high armature current.

Another significant advantage of the separately excited DC motor is the 'free' use of regenerative braking. Since the armature is always connected directly to the battery, current flow between the armature and battery depends on the 'operating point' of the motor. Increasing field current at a given operating point (above base speed) reduces motor speed. As the vehicle slows, the stored energy of the moving vehicle is converted back to electrical power. The related losses are both mechanical (drive wheels to motor shaft) and electrical (motor efficiency/charge acceptance of batteries) in nature. With a proper control system, the driver is not conscious of the fact that the motor is changing to a generator mode.

A separately excited motor, used in conjunction with a transmission, can be controlled over nearly the entire vehicle speed range by field weakening; i.e., the motor torque and speed can be controlled by a device which needs to handle directly only about 5% of the motor rated power. Consequently, losses associated with the controller are negligible compared to the motor output. Also, there are no induced losses due to motor armature current ripple resulting from a chopped motor voltage, and the battery sees a constant current discharge, which is better from a battery efficiency standpoint than the chopped wave form which results when an armature chopper is used.

As a result of these considerations, we came to the conclusion that the DC separately excited motor is the most suitable of the alternatives investigated for the near term hybrid vehicle.

3.5.4 Motor Control

As discussed in the last section, controlling field current provides an effective and efficient method of controlling motor torque and speed over most of the vehicle speed range. However, below base speed (speed at max field), the motor is no longer controllable by this means.

There are several alternative methods of torque and speed control at vehicle speeds at which the motor would normally be below base speed. These include the following:

- Allow the motor to idle at base speed and slip the clutch. This is the method used on SCT's electric conversion of the VW Rabbit; and it works quite well, with clutch slippage required only up to about 8 mph. However, it requires the use of a manual transmission. Since the vast majority of car buyers want an automatic transmission in a car of the hybrid's size, this is not a viable alternative.
- Allow the motor to idle and use torque convertor slippage to make up the difference between vehicle speed and motor speed; use of the service brake would be required to hold the vehicle at rest or to modulate its speed below motor base speed. This is not a viable alternative either because of extremely high torque convertor losses when the vehicle is at rest.
- Resistor control of motor voltage and speed below base speed. This involves dissipating a significant amount of energy in heat, particularly if the electric motor is to be

kept idling at a low speed to maintain transmission oil pressure. This we considered to be a 'last ditch' alternative.

- Battery switching control of motor voltage and speed below base speed. The discrete voltage and torque steps which result from this technique give rise to a 'jerky' startup; we do not regard this being acceptable driveability in a vehicle of the hybrid's class.
- Use of an armature current chopper rated to handle maximum motor current (full power chopper).
- Use of an armature current chopper rated to handle a fraction of maximum motor current (low power chopper).

Of these alternatives, the last two are the only ones which are likely to be acceptable from the standpoints of driveability and efficiency. As discussed in Section 3-2 on the baseline hybrid, using a low power chopper instead of a full power chopper results in a very small penalty in acceleration time from a standstill; and, of course, there is no difference in performance in the normal driving speed range.

The major advantages to a low power (current limited) chopper are cost, size, and weight. For instance, a full power (300 amp) chopper operating from a 120 volt battery pack is somewhat large for a transistor design (see discussion on switching elements). An SCR chopper of that size would be large, expensive, and relatively noisy.

A 100 amp chopper could be economically built using power transistors. Such a chopper could operate at higher frequencies without the commutation problems of SCR choppers. Higher frequency operation also reduces noise and eliminates the need for extra series inductors.

The high frequency characteristics of available power transistors permits closed loop current-sensing type circuitry to be used. This enables the chopper to operate very close to its maximum current limit without exceeding it. Typical time constants of armature inductances are on the order of 100 uS. Since many transistor switching times are 50 to 100 times faster, the motor armature can be switched directly with no additional series inductance. The elimination of a series inductor reduces weight and improves efficiency.

The advantages of the low power chopper appear to outweigh the slight performance loss of about 1/2 second on the standing-start acceleration times, and we have chosen to pursue this approach.

Figure 3-41 shows a representative block diagram of a low power chopper capable of 150 amp peak (125 amp average) operation. The power transistors shown are gain rated at 50 amps, continuous duty to 100 amps, have 350 volt breakdown voltages along with submicrosecond switching times. Single piece prices are currently about $145.

Solid State Switching Elements

There are two basic types of solid state switching elements suitable for a DC power chopper: the SCR and the transistor.

The SCR (Silicon Controlled Rectifier) is a device which looks effectively like an open circuit until its gate is triggered. Once triggered, the SCR remains in a conducting state (short circuit)

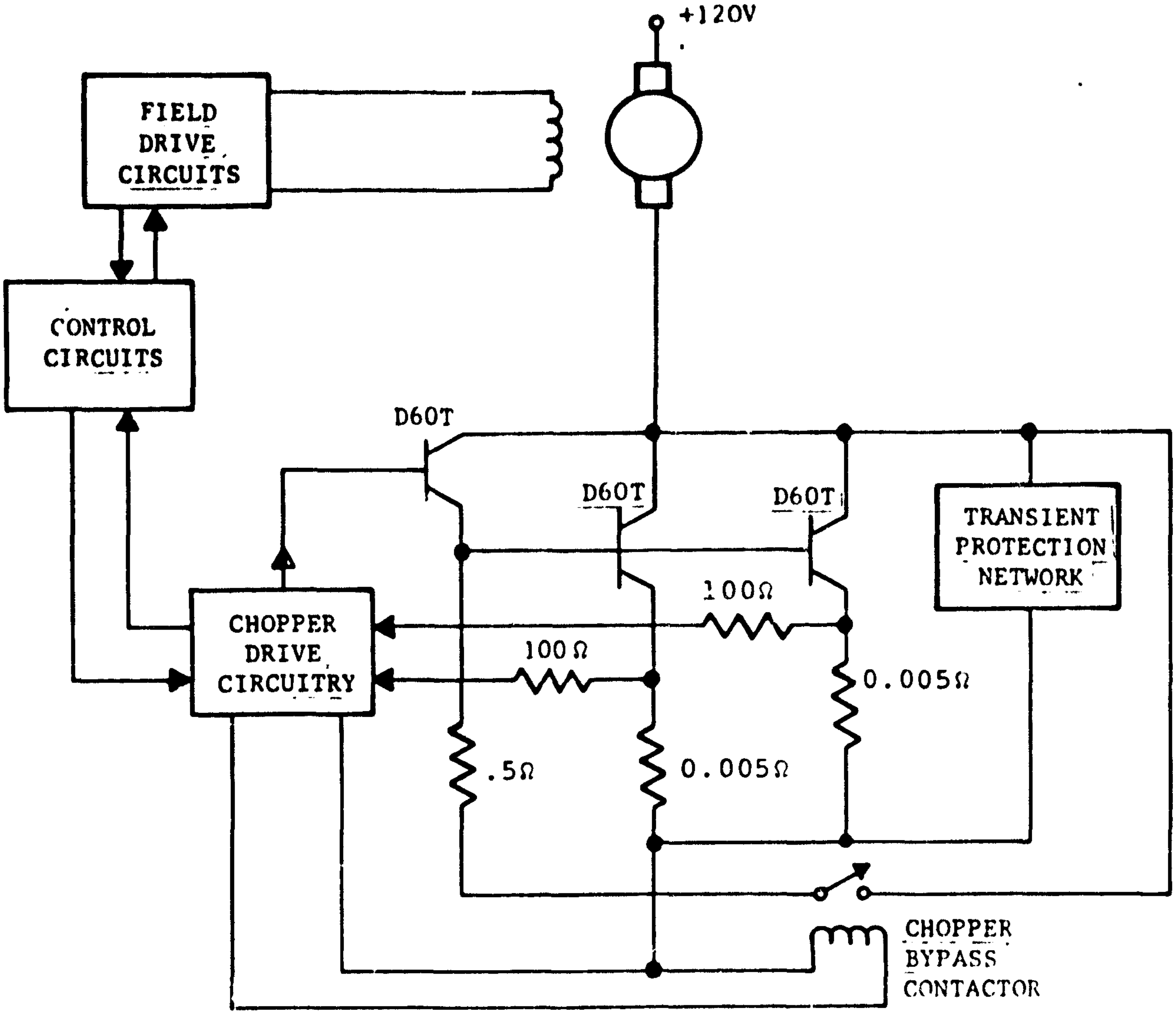


Figure 3-41 Low Power Armature Chopper Used in Conjunction with Field Control

until the load is removed. In an AC circuit, SCR turn-off occurs naturally at the end of an AC cycle. By varying the trigger point (in time), the 'on' time (duty cycle) is varied from minimum to maximum. The type of operation from an AC input is called phase control. There are many relatively low cost SCR's that have been developed specifically for this application. Unfortunately, since the standard AC line frequency is only 60 Hz, these SCR's have a very slow speed requirement; and any chopper using them must operate at relatively low switching speeds (1 KHz). High speed SCR's have been developed for certain applications; but high power, high speed SCR's are generally not available.

As mentioned before, an SCR, once triggered, remains in conduction until the load is removed. In a DC system, an SCR, once triggered, will remain on. Turning off an SCR under DC conditions is referred to as commutation. Commutation can be accomplished by another SCR in series with a capacitor. Essentially, the second SCR shorts the load momentarily, allowing the first SCR to turn off. Such circuits are very difficult to design properly because they are very dependent on the characteristics of the load, the SCR's, and the capacitors. Since the load will change greatly, and SCR and capacitor characteristics will vary with time and temperature, an SCR chopper must be designed to operate under all variations of temperature, motor load, and battery voltage. In addition, several key SCR limitations must be observed (turn-off time, dv/dt and di/dt limitations, in addition to false triggering).

Power transistors have been traditionally relegated to relatively low voltage, low current applications. Only recently have high voltage, high current transistors become available. Because they have only recently been developed, costs tend to be higher. Performance benefits can often outweigh cost considerations, and the circuit simplification and reduction in associated high power components usually favors power transistors, assuming the devices are available.

A transistor switch is turned on and off by its base drive. Unlike an SCR, removing base drive turns off the device, independent of the load. Since the current gain of most high power transistors is about 10 at their operating point, base drive requirements can be quite high, much higher than the triggering current of SCR's.

A simple way around this problem is the Darlington configuration shown in Figure 3-41. This configuration minimizes base drive losses, and results in overall current gain on the order of 100. Since the base drive voltage is only a few volts, the drive power requirements to this configuration are quite low. Figure 3-41 also illustrates a parallel transistor output stage.

Transistor paralleling is a way to obtain high current ratings from smaller devices. Although very often the cost of several smaller devices is less than the cost of a single large device of equivalent rating, the added circuitry required for balanced current sharing and the degradation in switching performance that this produces usually pushes the designer to a single transistor circuit, or to one which uses only a few in parallel. Large numbers of paralleled transistors also suffer in terms of switching times. If the transistors do not turn on and off at the same time, one transistor will be subjected to a much greater load during switching, which will cause localized heating and possible device failure.

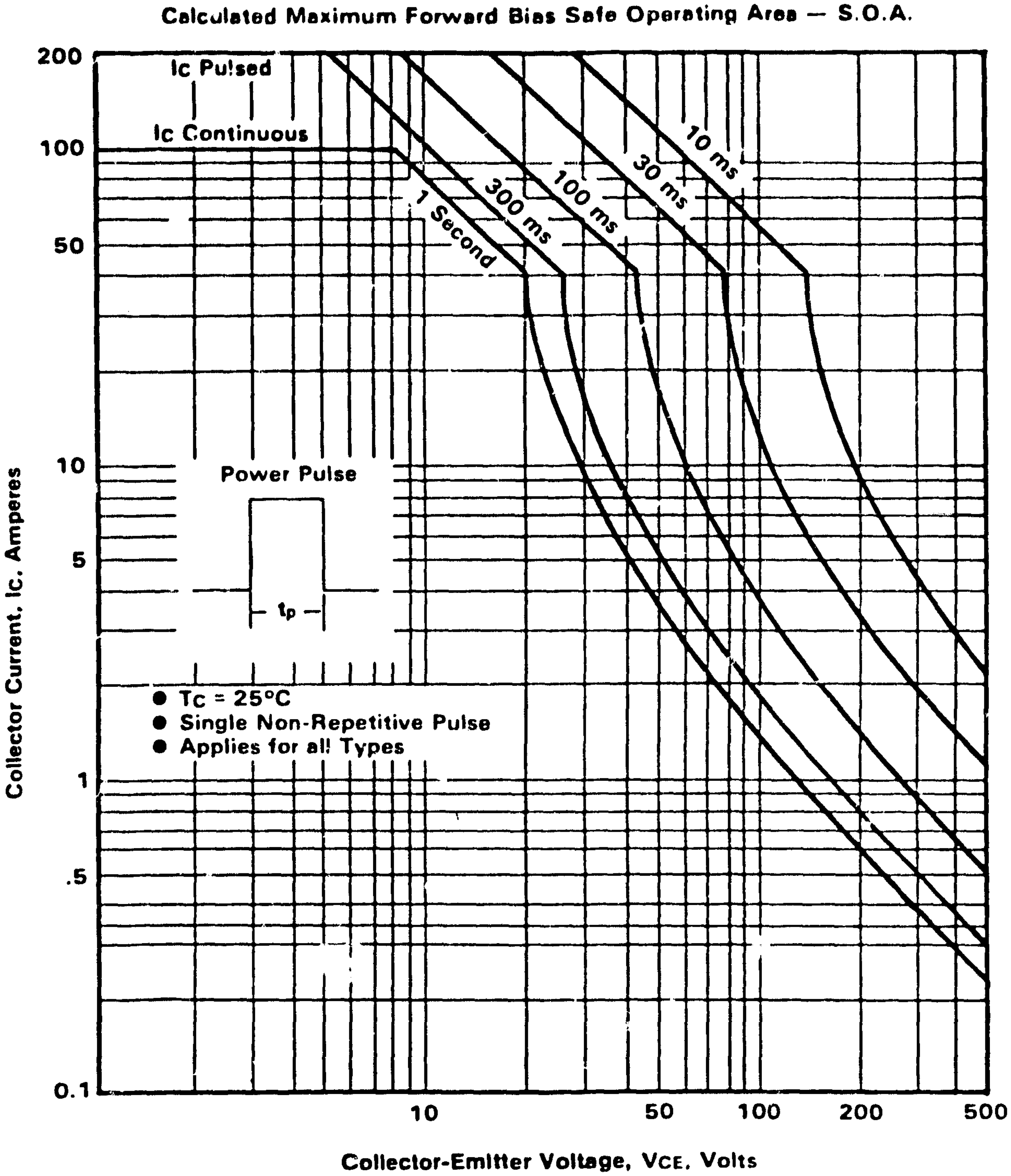
Effects of System Voltage on Switching Transistor Selection

For a given power requirement, the product of voltage and current will remain constant. As a result, once the required power has been established, an operating point (in terms of voltage and current) must be chosen. In addition to considering the relative efficiencies and costs of the motor and batteries, the costs and efficiencies of the switching transistors as a function of system voltage must also be considered.

Since transistors are basically current operated devices, their power handling capability is a function of their current rating, while operating in a switching mode at their highest rated voltage. If transistor availability was the driving force in a design, the system voltage would be established just below the voltage rating of the transistor having the lowest cost per watt of capacity.

However, realistic designs have many other parameters to consider. One of these is switching speed. Since the transistor will be operating with a pulse waveform, the device's performance with a particular waveform must be considered.

As an illustration of how important the waveform can be, refer to Figures 3-42 and 3-43. Each transistor has a safe operating area diagram. This diagram shows the permissible combinations of voltage, current, and time. Typically, a particular transistor operating



(WESTINGHOUSE D60T)

Figure 3-42 Typical Safe Operating Area Curves

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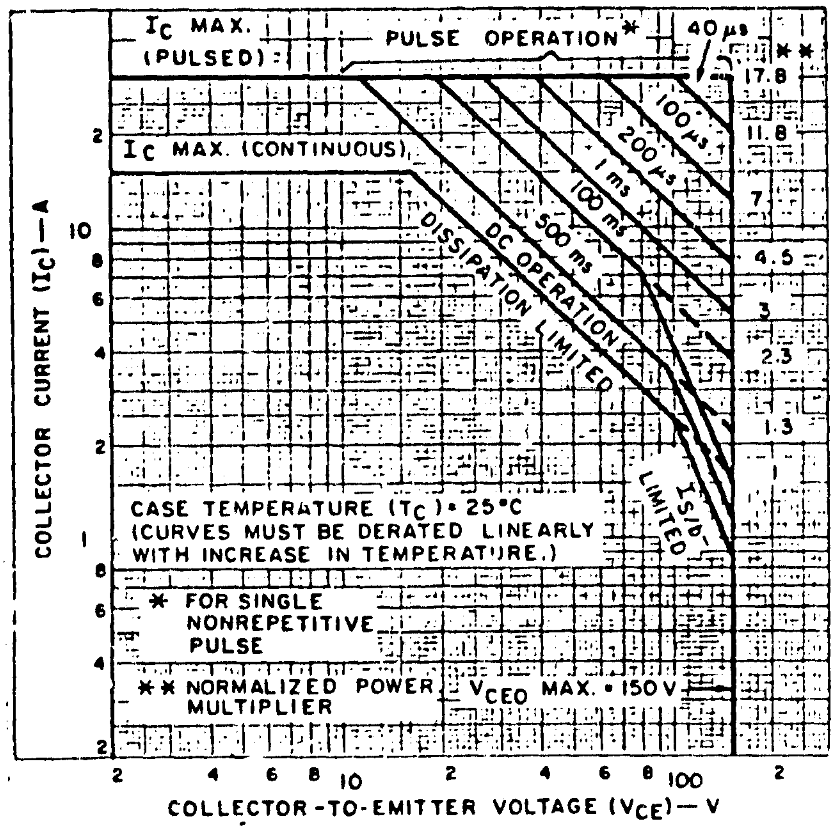


Figure 3-43 Typical Safe Operating Area Curves

(RCA 2N6259)

point is considerably below the 'maximum' ratings. Figure 3-44 shows the data from several of the curves plotted at a constant voltage of 100 volts. It is interesting to note that transistors having widely differing costs and ratings very often cross over unexpectedly. As an example, compare the 300 ms current ratings of a D60T and a 2N6259. The D60T is a 200 amp, 400V, $200 transistor, while a 2N6259 is a 15 amp, 180 volt, $6 transistor. At this point, the D60T can handle only 2 amps, but the 2N6259 can handle over 3 amps.

Fortunately, most designs result in switching operation which minimizes the time spent in the linear (non-saturated) state. Switching speed is still important, because tremendous amounts of power can be dissipated by the transistor during turn-on and turn-off.

Because of the complexities of transistor selection, it is very difficult to generalize about the effects of system voltage on transistor cost. There is one fairly well defined breakpoint, however. Below about 180 volts, single diffused transistors can be used. This technology is well established and relatively low cost. There are many transistors of this type which have current ratings of 50 and 60 amps. Above 180 volts, triple diffused transistors must be used. This is a newer, more expensive technology. It enables transistor voltage ratings to be much higher (typically 300 volts), but the devices' current handling capability is usually low. Transistors with maximum voltage ratings above 180 volts and maximum current above 25 amps are very rare and very expensive.

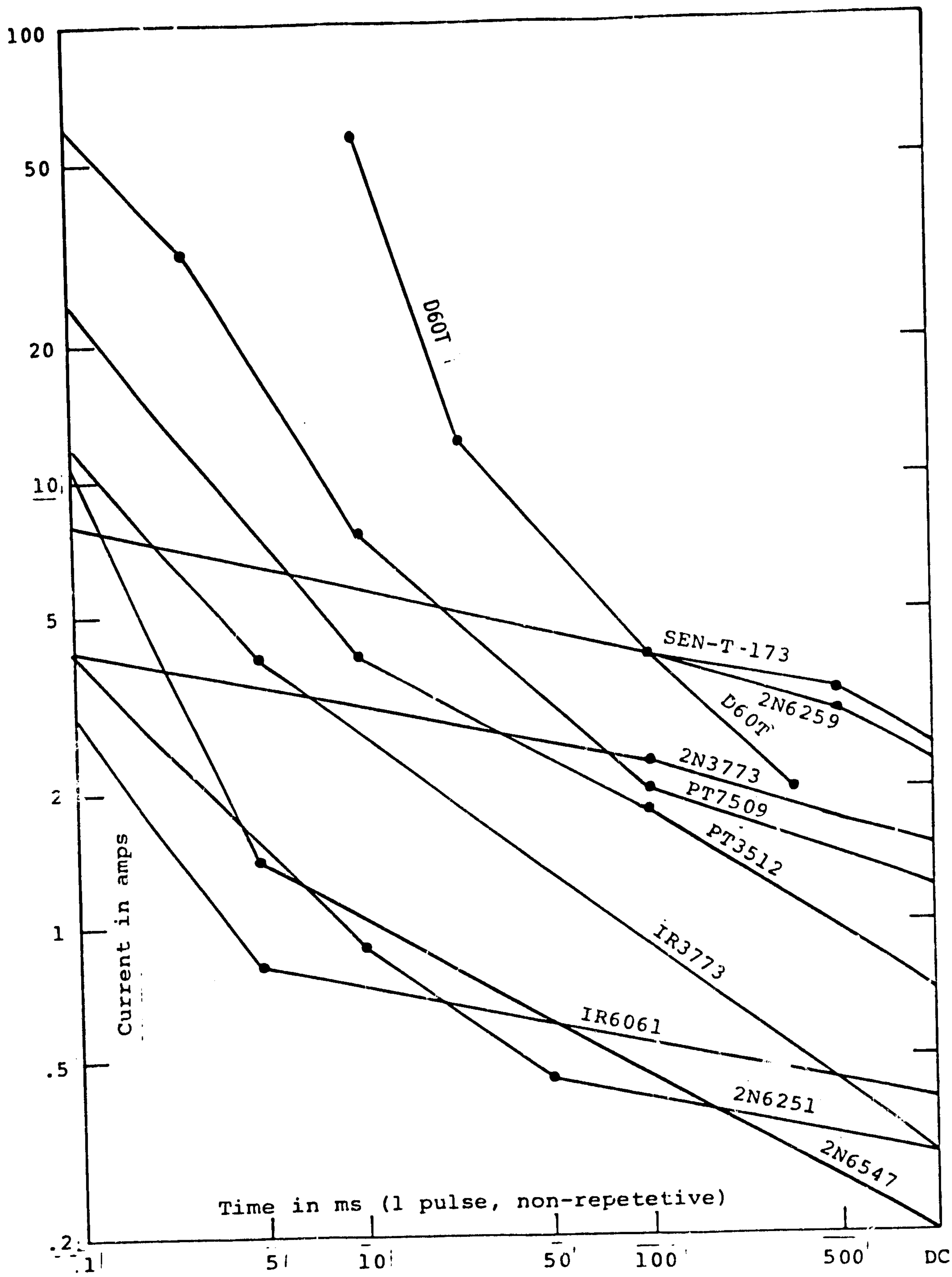


Figure 3-44 Current vs Time @ Vce = 100 volts

(points taken from published safe operating area curves)

It would seem that the only significant point in voltage selection is the 180 volt point for using single diffused transistors. Aside from that point, there are enough transistor choices and enough system variables that one must only assume that the proper transistors would be available and reasonably cost effective for a particular application.

The 180 volt transistor rating does not allow a system voltage of 180 volts, however. The 180 volt rating is a maximum voltage rating with the base-emitter junction reverse biased by about 4 volts. This is usually difficult to do at the levels of base drive normally required. With the base held at the emitter voltage, the collector-emitter breakdown voltage is usually 160 volts. This is the maximum voltage the transistor can withstand while turned off. However, even though the large motor inductance is clamped by a diode, practical chopper circuits have enough stray inductance and capacitance to cause small voltage spikes of about 15% of the open circuit voltage. This reduces the maximum open circuit voltage to 139 volts. Allowing another 10% for charging voltage rise during regenerative braking leaves a system voltage of 126 volts.

For a typical chopper design, transistor costs will approximately double as this voltage threshold is crossed, although total chopper cost may only increase about 25%.

System Control

Based on the complexity of the control strategy required for the hybrid, it is clear that overall system control must be handled by a microprocessor. The requirements for the microprocessor will be defined more fully during the Preliminary Design task after affitional optimization of the control strategy is done. Reserve capacity will be provided in the microprocessor to handle heat engine spark and fuel control, if that should prove necessary. Whether it will be or not is something that will have to be determined experimentally.

3.5.5 Transmission Alternatives

The alternatives to the three-speed automatic transmission with lockup torque convertor chosen for the baseline hybrid vehicle are as follows:

- Four speed (overdrive) automatic with lockup torque convertor.
- Four speed automatic with fluid coupling.
- Automatically shifted transmission with automatic clutch.
- Continuously variable transmission.

Of these, the second and third were quickly eliminated. The four speed automatic with fluid coupling (i.e., the old GM four speed hydramatic) is a good transmission with slightly higher overall efficiency than an automatic which uses a conventional torque convertor. However, the advent of lockup torque convertors removes this efficiency advantage, and the torque multiplication provided by the torque convertor at low speeds is very useful. (For example, it makes possible the use of a low power armature chopper without significant performance penalty and without needing a very wide range of transmission ratios.)

Automatically shifted transmissions with a fully automatic clutch (i.e., manual transmissions which are shifted and declutched without driver intervention) have had a checkered history. Those attempts which have reached production have generally been only semi-automatic; usually, eliminating the clutch only. A fully automatic gearbox type transmission was under development recently at Fiat; however, it seems to have passed into limbo. The major problem with this type of transmission appears to be driveability; without the shock absorbing characteristics of a torque convertor or the sensitivity of a human controller, it is extremely difficult to get smooth shifts over a wide range of throttle settings and vehicle speeds.

Four Speed Automatic with Overdrive

A four speed automatic with overdrive requires a little more consideration, as we have already indicated in Section 3.4. Such a transmission improves fuel economy in a conventional car by reducing the engine speed and increasing the engine load (torque) under cruise conditions. This moves the engine operating point into a lower bsfc region, and with the engine power remaining the same, the fuel consumption drops. With the hybrid, on the other hand, the engine is already fairly heavily loaded under cruise conditions due to its small size relative to the vehicle size it is driving. Consequently, the relative improvement that would be expected by the introduction of an overdrive fourth gear (keeping the final drive ratio the same) would be considerably less than would be obtained in a conventional car. Such a transmission would be better utilized in the hybrid by increasing the final drive ratio somewhat, thereby not dropping the engine speed as much and providing a better performance. The beneficial effects of such a change on gradeability at cruising speeds have already been discussed. A simulation of such a situation was run using the same transmission ratios as the 3-speed baseline, with the addition of a .75:1 overdrive high gear and an increase in the axle ratio from 4.1 to 5.12. Fuel economy with this configuration improved by 2.8%, wall plug energy consumption decreased by 3.4%, and 0-90 kph time improved from 14 sec. to 13.1 sec. Further improvement was obtained by an adjustment in torque convertor diameter, which provided an additional .8% improvement in fuel consumption to 19 km/l (44.7 mpg), with no change in energy consumption at .199 kw-hr/km, and a further decrease in 0-90 kph time to 12.8 sec.

The four speed configurations were assumed to have lockup on the top three gears. A simulation was also run of the last configuration described above, with lockup only on the top two gears. This gave 18.8 km/l fuel economy and a .200 kw-hr/km energy consumption, from which we concluded that the decision to lock up second gear or not would have to be based more on driveability than energy efficiency.

As a result of these considerations and those discussed in Section 3-4 , we came to the conclusion that the four speed automatic transmission offers advantages in overall performance (including gradeability, low noise, and smoothness) which would make it highly probable that a manufacturer would use one in a hybrid, particularly if he had one in his parts bin. It is known that transmissions of this type are under development for production within the next two years by major manufacturers (e.g., Ford); consequently, replacement of the three speed assumed for the baseline hybrid by a four speed would appear to be warranted.

Continuously Variable Transmission

A survey was conducted of the state of the art in continuously variable transmissions in an attempt to determine whether any CVT's are suitable for use in a passenger vehicle of the hybrid's size and have the potential of becoming production hardware by 1985. CVT's of the hydrostatic, electric, traction drive, variable ratio belt drive, hydromechanical, and electro-mechanical were looked at.

The conclusion reached on the basis of the results of this survey was that the unit which shows the most near term promise is the metallic belt drive being developed by Van Doorne's Transmissie B.V. in Holland and Borg Warner in the U. S. This is well along in development. Units are quite compact, and there does not appear to be any fundamental limitation which would prevent scaling up the existing designs (primarily for small cars) to the power requirements of the hybrid.

As a belt transmission, it is unique in transmitting torque by forces on the 'compression' rather than tension side of the belt. The belt consists of a set of endless maraging-steel bands, which support and guide a set of wedge shaped steel elements. These wedge shaped elements ride on the pulley surfaces, and transmit torque from one pulley to the other by thrust forces between the elements. Tensioning of the bands must be greater than the thrust forces between the elements; this tensioning, together with the positioning of the pulleys to vary the transmission ratio, is accomplished hydraulically. A separate clutch is required for startup since slippage of the belt relative to the pulleys is not permissible. In a conventional vehicle, this clutch would normally be of the centrifugal type.

Advantages of this type of transmission relative to a conventional automatic are the elimination of torque convertor losses and the possibility for obtaining optimum loading of the heat engine at any power demand.

Considering these advantages of this transmission, together with its advanced state of development, we concluded that a more detailed study was warranted to quantify its fuel economy advantages.

Control Strategy for CVT

The elements of the control strategy used in conjunction with the CVT can be summarized as follows:

On Mode 1: As in the case of the baseline hybrid, the heat engine cuts in only if the power demand exceeds a nominal power level, which is selected to avoid operation of the electric motor and batteries at excessively high power levels, to the extent possible. When the heat engine does cut in, the transmission ratio and electric motor power level are adjusted to keep the heat engine operating at its best bsfc. This is shown as point P_1 in Figure 3-45 which is an engine map on which bsfc is plotted in terms of power (rather than bmep) and speed. If this is not possible (for example, if the total power demand is too high or too low), the heat engine is kept as close as possible to this point, and preferably along the

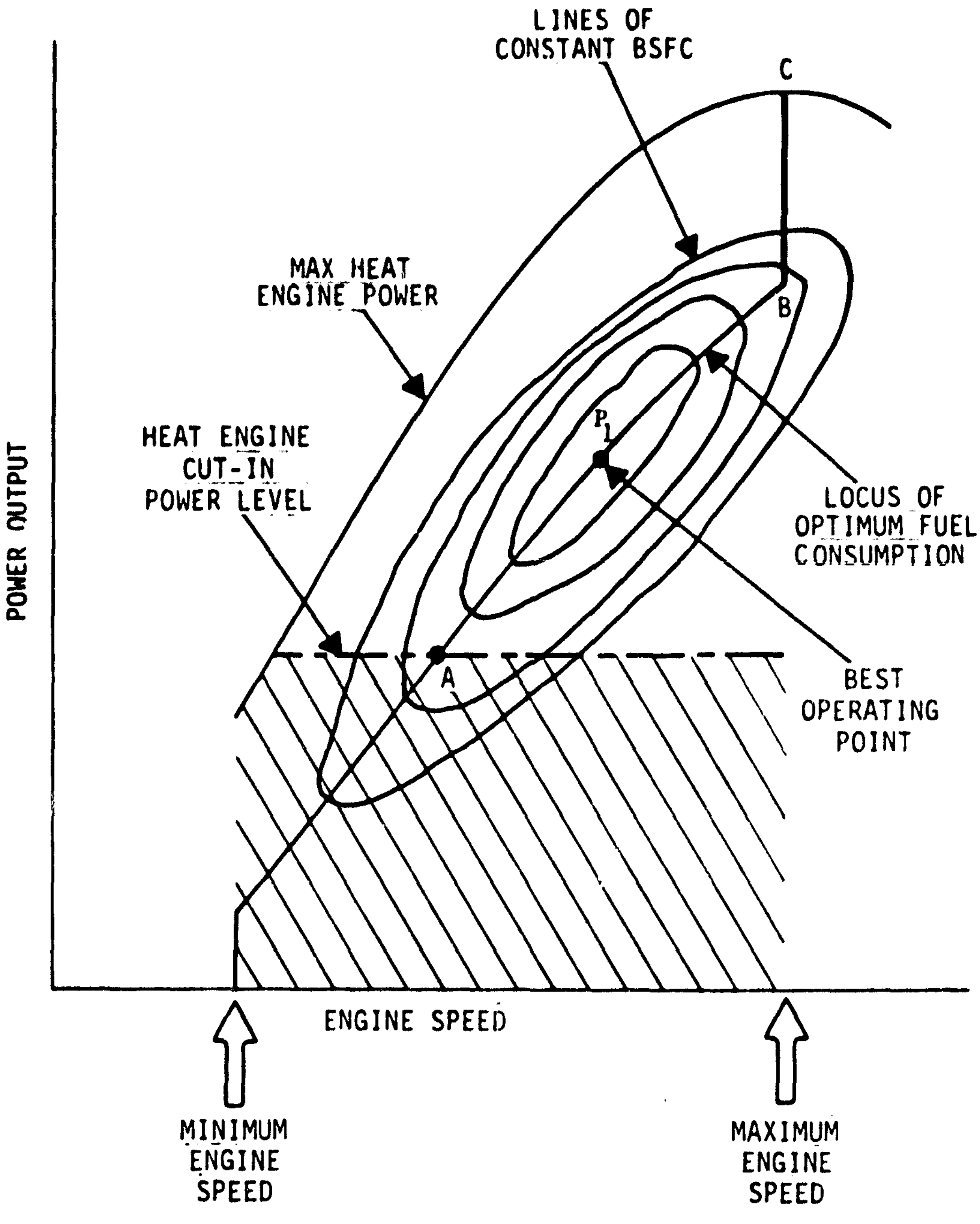


Figure 3-45 Operational Region for Heat Engine with CVT

curve AB, which presents the locus of points of the lowest achievable bsfc for a given heat engine power output.

On Mode 2: The transmission ratio is adjusted to keep the heat engine on the curve AB, if possible; the electric motor is used only if the power demand exceeds the maximum capability of the heat engine within the range of engine speeds available at the given vehicle speed.

Simulation Results

Simulation of a hybrid vehicle with a CVT gave a fuel economy of 18.1 km/l (42.6 mpg), an energy consumption of .161 kw-hr/km, and a 0-90 kph time of 12.8 sec. Compared to the best four speed automatic case simulated, this represents a decrease in fuel economy of about 5%, a reduction in energy consumption of 19%, and no change in 0-90 kph acceleration time. Presumably, with adjustments in the control strategy, the fuel economy and energy consumption increments could be traded off to obtain about a 14% increase in fuel economy at no change in energy consumption relative to the best four speed simulation. This is considerably less than the improvement that could be expected from the use of a CVT in a conventional vehicle, which would be on the order of 20-25%. The reason for this is that the primary advantage of the CVT is in its ability to keep the engine operating close to its minimum bsfc region; and in the hybrid, the engine is already operating fairly close to this region most of the time.

Another point which should be noted is that the control strategy used in the CVT simulation incorporated information on the best operating point of the engine and on the speed vs. power line which provides the lowest bsfc at a given power. The transmission ratio was controlled to keep the engine in the closest possible proximity to these optimum values. The most sophisticated strategy used with the four speed automatics, however, was that described in Section 3.4 under Control Strategy Variations. With this shift logic, there was no reference to an optimum operating line for the engine. When the heat engine was operating, the shift points were set as a linear function of the ratio of power demand to peak available total power (i.e., essentially accelerator pedal position), with zero power demand corresponding to a 3000 rpm upshift and maximum power demand corresponding to a 6000 rpm upshift. When the heat engine was not operating, the shift point schedule was modified to keep the electric motor speed up in a region (around 3500 rpm) which would provide effective regenerative braking. This strategy could be considerably improved by making it a discrete approximation of the strategy used with the CVT, i.e., incorporating knowledge of the optimum heat engine operating conditions for a given power demand and shifting accordingly.

Because the region of low bsfc is rather broad for the baseline gasoline engine, covering a range from roughly 2500 to 4500 rpm, such a strategy would be able to keep the heat engine operating at almost an average bsfc which is not significantly different from that attainable with a CVT. Consequently, we concluded that the improvement in fuel consumption resulting from the use of a CVT would result primarily from the improved efficiency relative to a four speed automatic, given control strategies of equal sophistication for the two transmission types.

The difference in efficiency between these two types of transmission resides primarily in the torque convertor of the four speed automatic. Both require oil pumps to supply pressure for actuating clutches and bands (automatic) or the variable ratio pulleys (CVT). If anything, we would expect the overall efficiency of an automatic (sans torque convertor) to be slightly higher, since one gear is direct drive. From Table 3-3 , torque convertor losses on the three component driving cycles range from over 8% of the total system output on the 227a(B) cycle down to .2% on the highway cycle. Using the number obtained on the urban cycle (5%) as representative, we conclude that the overall efficiency of the CVT is probably about 5% higher than that of an automatic with lockup torque convertor. Therefore, it can be expected that fuel economy of a hybrid with CVT would not be more than 10%, and probably would be more like 5% , better than a hybrid with a four speed automatic with a fully optimized control strategy and shift logic.

One more point should be mentioned before concluding this discussion, and that involves the relationship between transmission characteristics and the engine startup transient as felt by the driver. The lower the gear (i.e., the higher the overall gear ratio), the larger the acceleration and velocity change suffered by the vehicle during the engine startup transient. A torque convertor is extremely useful in reducing the magnitude of this transient, as seen by the vehicle. For example, in a couple of simulations of an engine start transient at 20 kph in first gear, the peak vehicle accelerations were .01G with a torque convertor in the system, and .08G without. The presence of an active torque convertor, then, in first, or first and second, gears, may be important from a driveability standpoint. A CVT of the Van Doorne type has no slippage or shock absorbing capability over any portion of its speed range.

We concluded from this that a conventional car is a much better place to put a CVT than a hybrid, in terms of the potential gains in fuel economy. Again putting outselves in the position of a manufacturer, if a CVT in the right power range were already developed and available for a conventional vehicle and did not cost more to produce than a more conventional automatic, it would be logical to use it in a hybrid vehicle. However, it would probably not be worth the investment to develop one specifically for a hybrid. For the Near Term Hybrid Vehicle Program, the Van Doorne CVT is an interesting possibility with unknowns attached to it in the areas of manufacturing cost and durability, and unessential to the basic objective of achieving a very large increase in fuel economy using near term technology. We, consequently, elected to stay with a four speed automatic with lockup torque convertor.

3.6 Supporting Studies and Analyses

3.6.1 Vehicle Layout/Packaging

From the start of our tradeoff studies, we had concluded that the South Coast Technology hybrid would be a derivative model of the 1985 Ford LTD reference vehicle. Our analysis and judgment led us to conclude that there would be little change between the current LTD and its 1985 counterpart, a judgment shared by those with whom we spoke that were employed by the vehicle manufacturer.

Our packaging task, thus, becomes a practical, matter-of-fact approach using actual 1979 Ford LTD dimensions and layout as the base we must work within. We have done our basic packaging work concurrent with the tradeoff studies and are, thus, in a firm position to state that no significant problems exist in developing the hybrid within these spatial constraints. Propulsion system hardware and controls all fit within the existing engine compartment, and there are many alternative battery layouts that offer acceptable weight distribution, safety, and accessibility.

Our propulsion system layout is shown on Figure 3-46 -47. The system provides for the packaging of the VW Rabbit gasoline engine, the Siemens electric motor, the clutch and transfer case assembly in a space under the hood of the LTD left vacant by the removal of the standard V-8 engine. This engine position maintains the existing automatic transmission position and does not interfere with the firewall or any vehicle system such as brakes and windshield wiper system.

During the Preliminary Design Task, these conclusions will be confirmed using design aides and layout drawings. As appropriate,

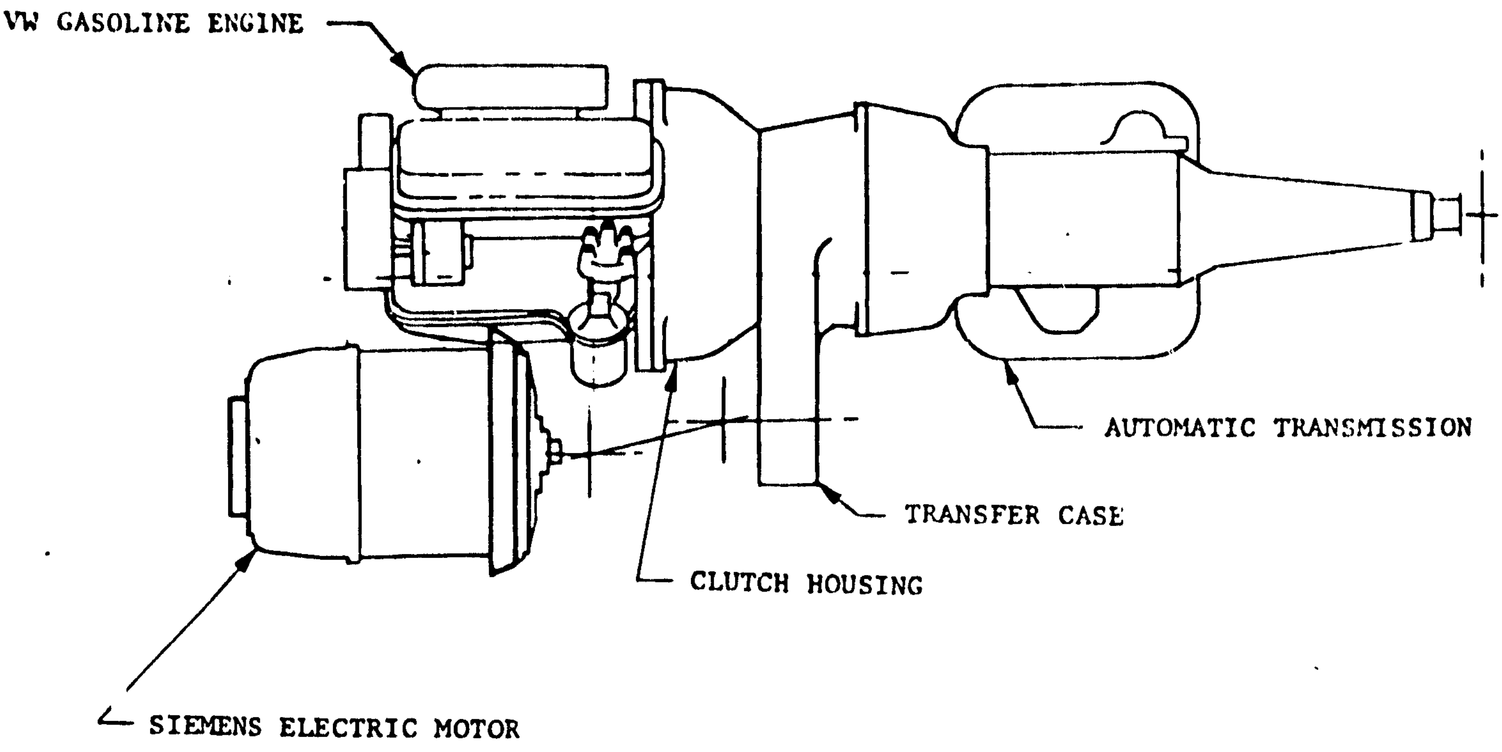
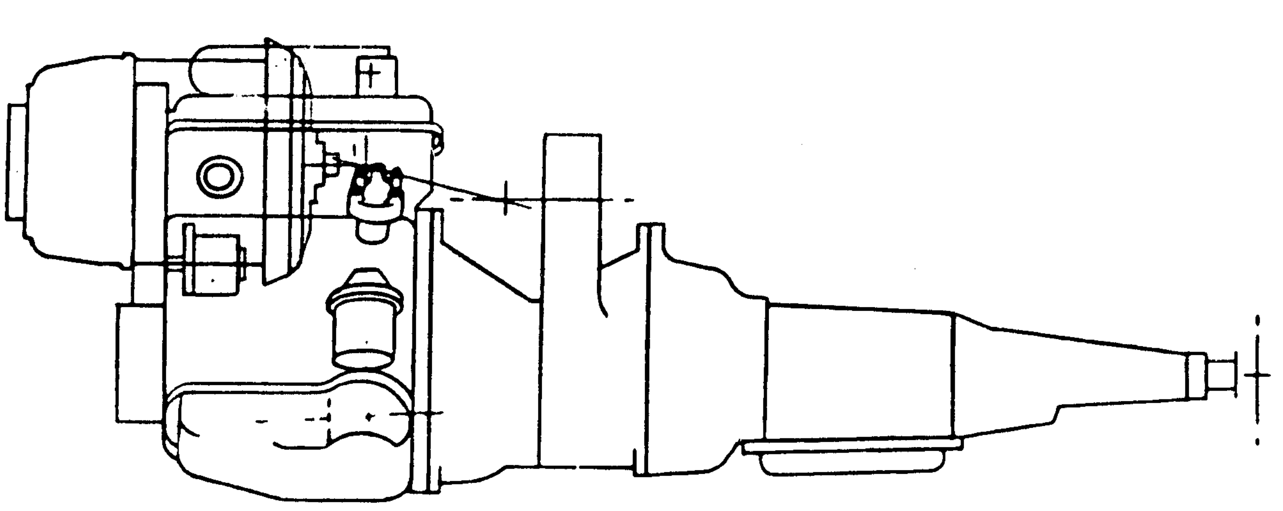


FIGURE 3- 46. HYBRID PROPULSION SYSTEM POWER COMPONENTS



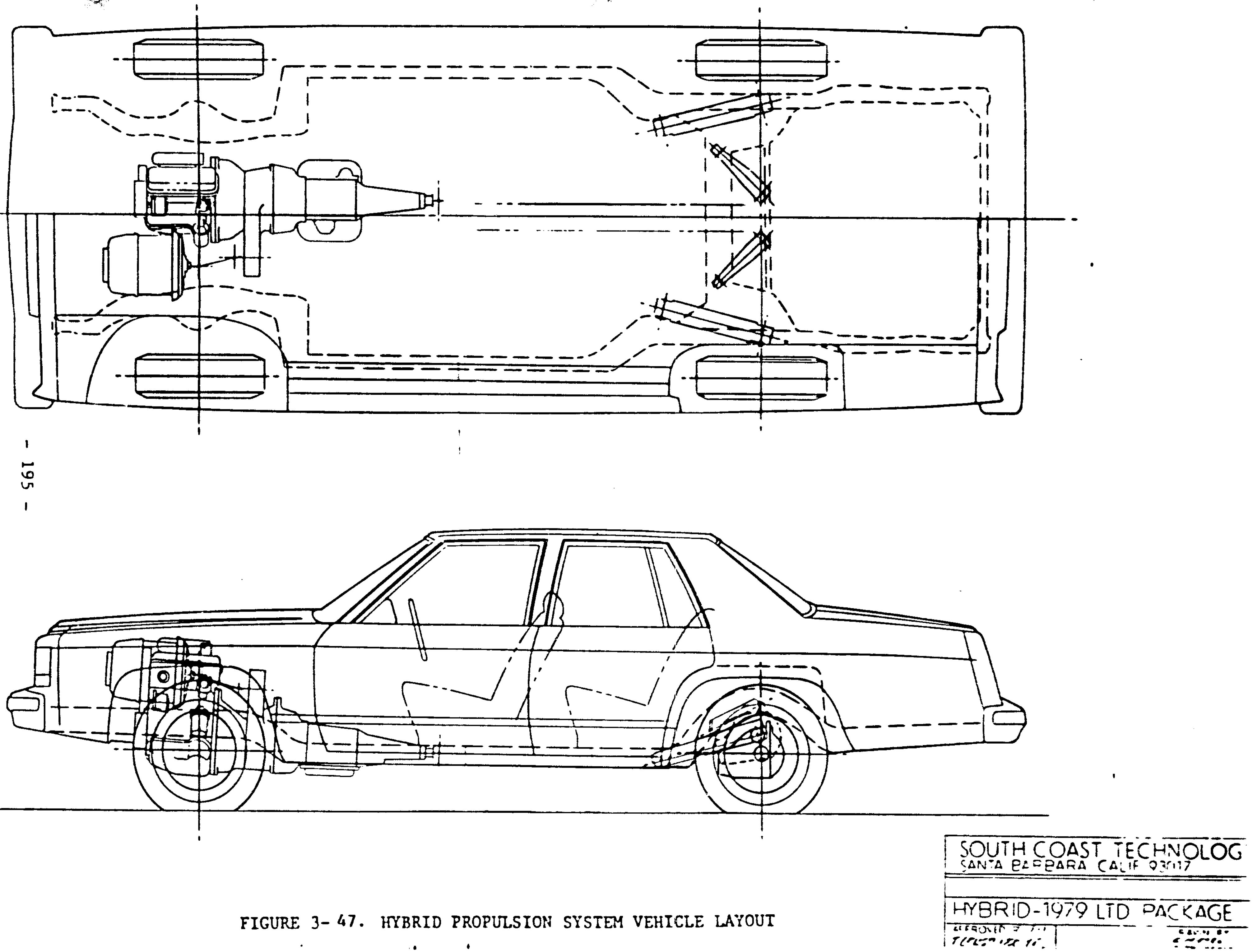


FIGURE 3- 47. HYBRID PROPULSION SYSTEM VEHICLE LAYOUT

modifications will be made to optimize the propulsion system packaging and to avoid costly or time consuming changes to the Ford LTD.

Battery compartment packaging studies were done under the assumption that lead-acid batteries would be used, with a volume equivalent to 12 golf cart modules. This represents a worst case in terms of packaging; the more highly ranked nickel-iron battery would have fewer packaging problems and less impact on weight distribution and handling.

Battery compartment packaging alternatives developed in these studies were reviewed in considerable detail to find satisfactory layouts and to then evaluate each layout against a set of criteria, Table 3-9 , which were then subjectively weighted in accordance with the factors shown on Table 3-10 .

Thirteen alternate battery compartment layouts were studied. The battery positions are shown schematically in Figures 3-48 - 3-60. Each of these 13 alternates indicate that the required hybrid system battery pack can be integrated into the Ford LTD without any major changes to the vehicle and to its primary passenger carrying and cargo utility. Further, it shows that there are many acceptable battery layouts with respect to vehicle handling characteristics. The scores for each of the 13 (Table 3-11) vary from a low of 294 to a high of 358, a relatively narrow spread. The calculations and analysis used in establishing the basis for scores on weight distribution and vehicle handling characteristics are found in Appendix D.

TABLE 3 9

BATTERY LOCATION MERIT FACTOR ANALYSIS - WEIGHTING

FACTOR:	Relative Value X	Weighting =	Merit Rating
	(1 - 10)	(6 - 10)	(6 - 100)
	10 Excellent	10 Very Important	
	9 = Very Good	9 = Important	
	8 = Good	8 = Highly Desirable	
	7 = Fairly Good	7 = Desirable	
	6 = Average	6 = Necessary	
	5 = Moderate		
	4 = Moderately Poor		
	3 = Poor		
	2 = Very Poor		
	1 = Unacceptable		

TABLE 3-10

BATTERY PACKAGING EVALUATION CRITERIA

Safety -

Passenger Envelope

Battery Mass Management

Crash Energy Management

Weight -

Distribution

YAW Moment

Feasibility/Cost

Cost of Implementation

Likelyhood of Achieving Cost Objectives

Accessibility -

Servicing and Maintenance

Ventilation

Space Utilization -

Interior

Cargo (Payload Capability)

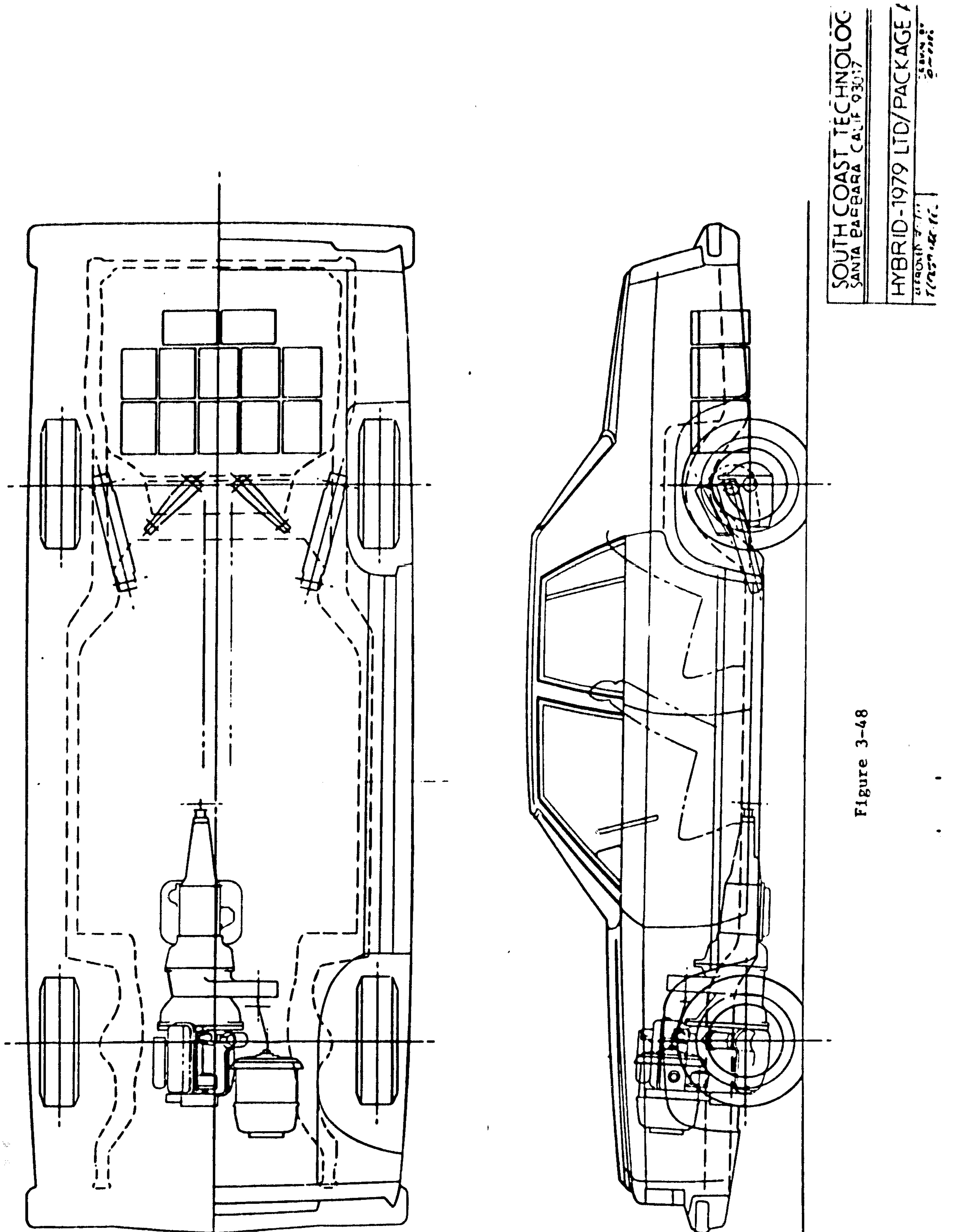
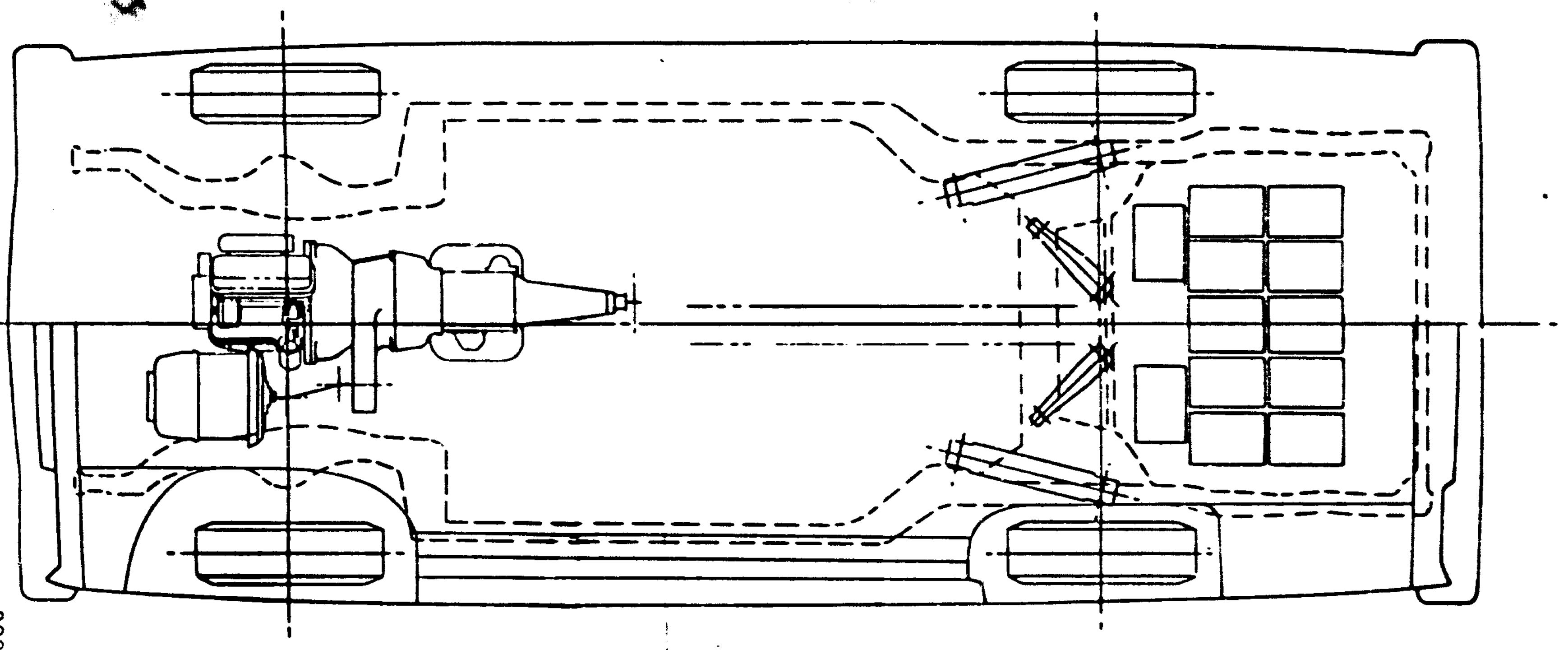
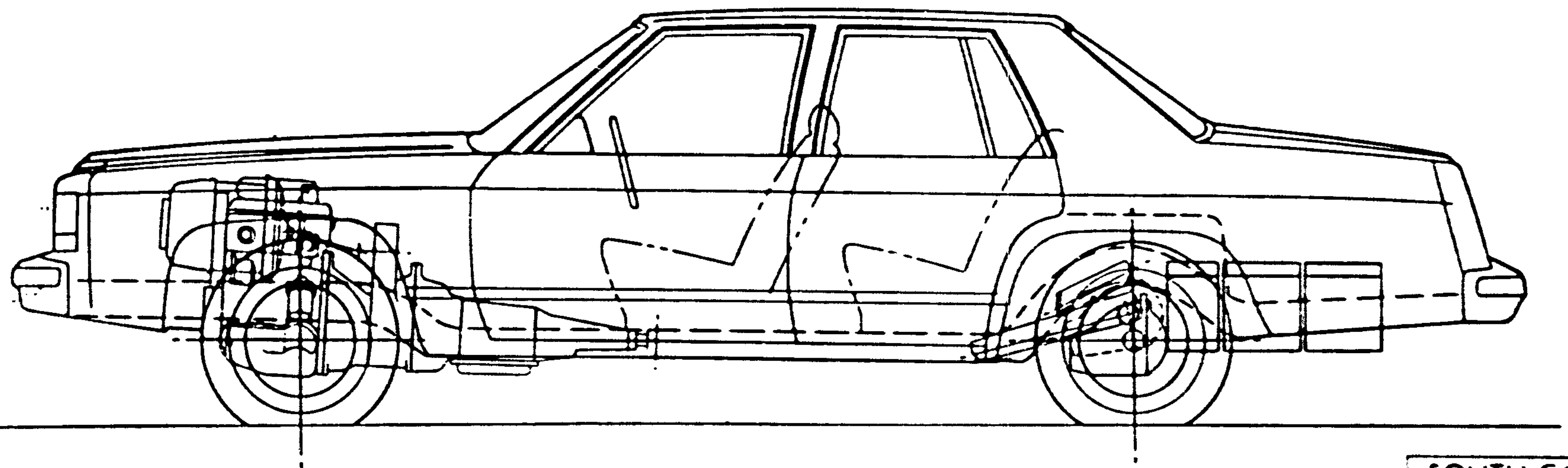

SOUTH COAST TECHNOLOG
SANTA BARBARA CALIF 93017
HYBRID-1979 LTD/PACKAGE

Figure 3-48





SOUTH COAST TECHNOLOG
SANTA BARBARA CALIF 93017

HYBRID-1979 LTD/PACKAGE

Figure 3-49

SOUTH COAST TECHNOLOG
SANTA BARBARA CALIF

HYBRID-1979 LTD/PACKAGE

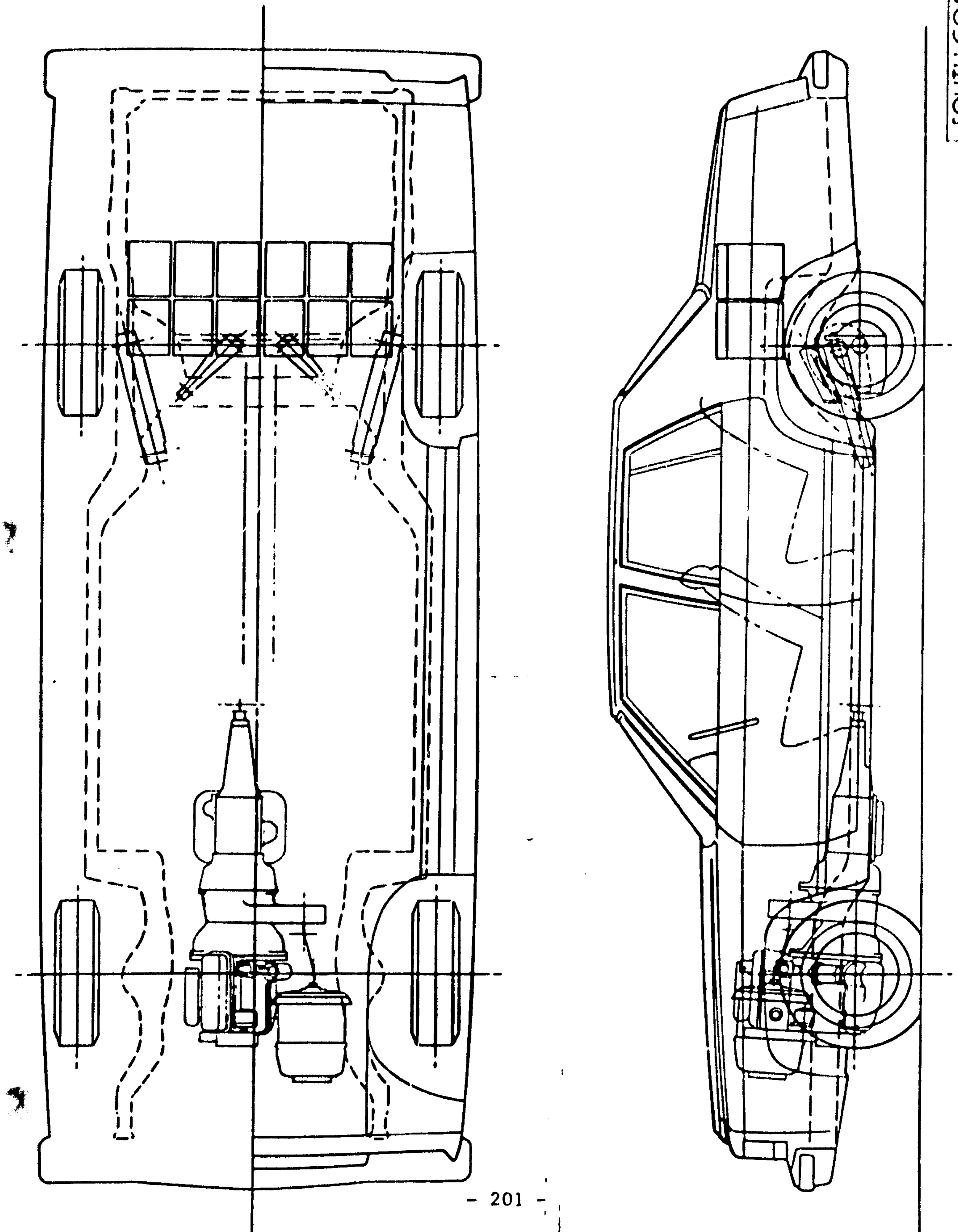


Figure 3-50

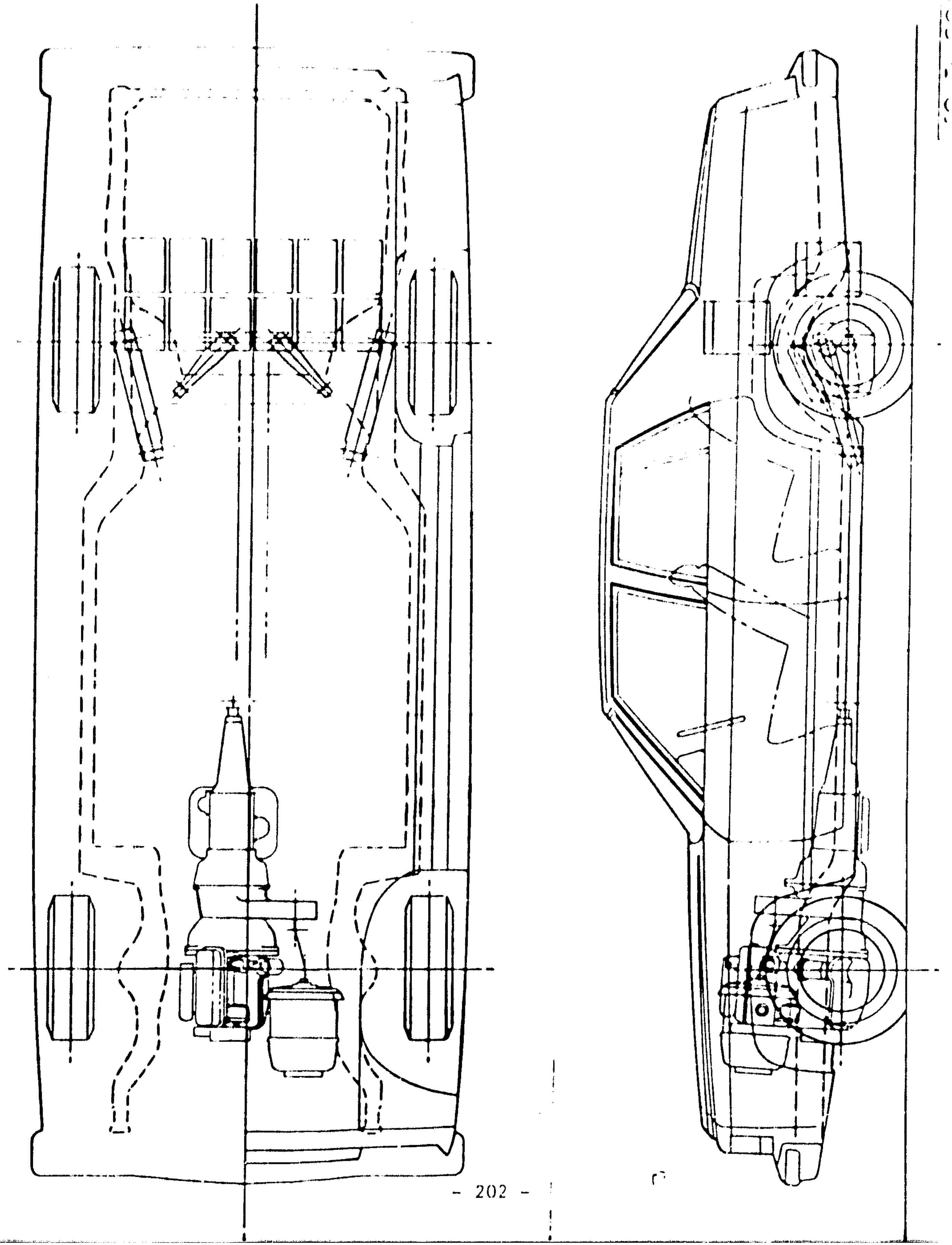


Figure 3-51

SOUTH COAST TECHNOLOGY

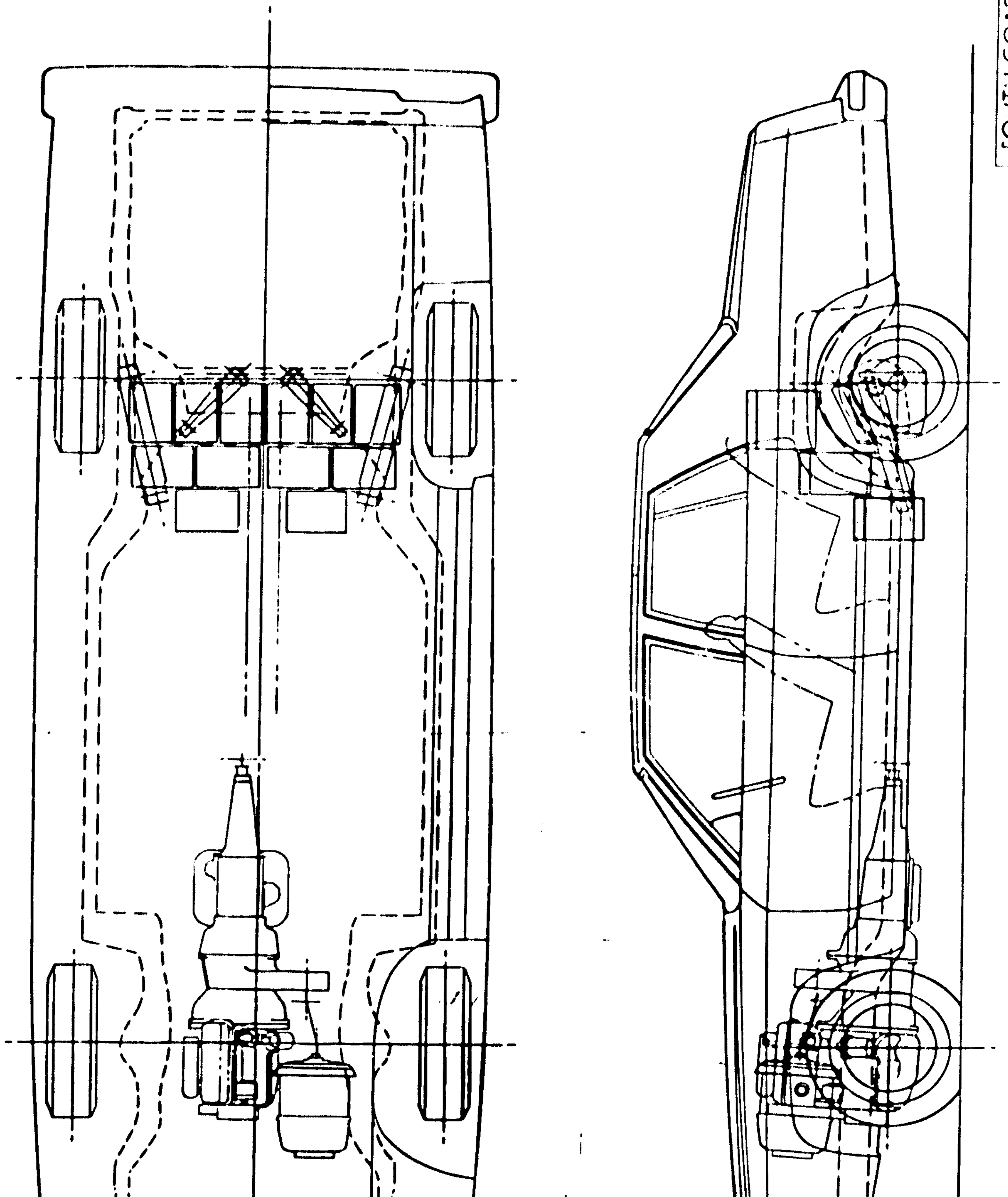


Figure 3-52

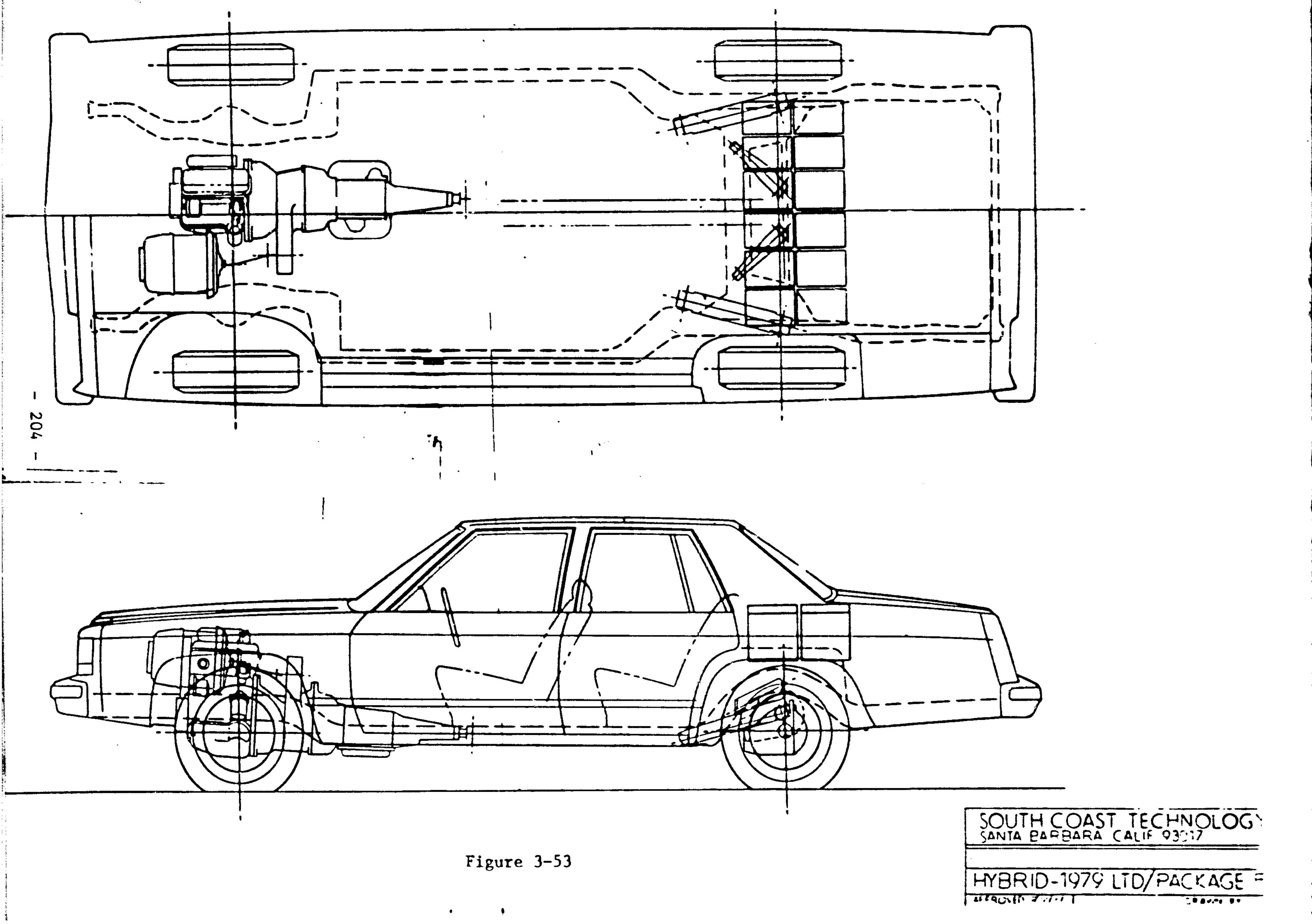

SOUTH COAST TECHNOLOGY
SANTA BARBARA CALIF
HYBRID-1979 LTD/PACKAGE

Figure 3-53

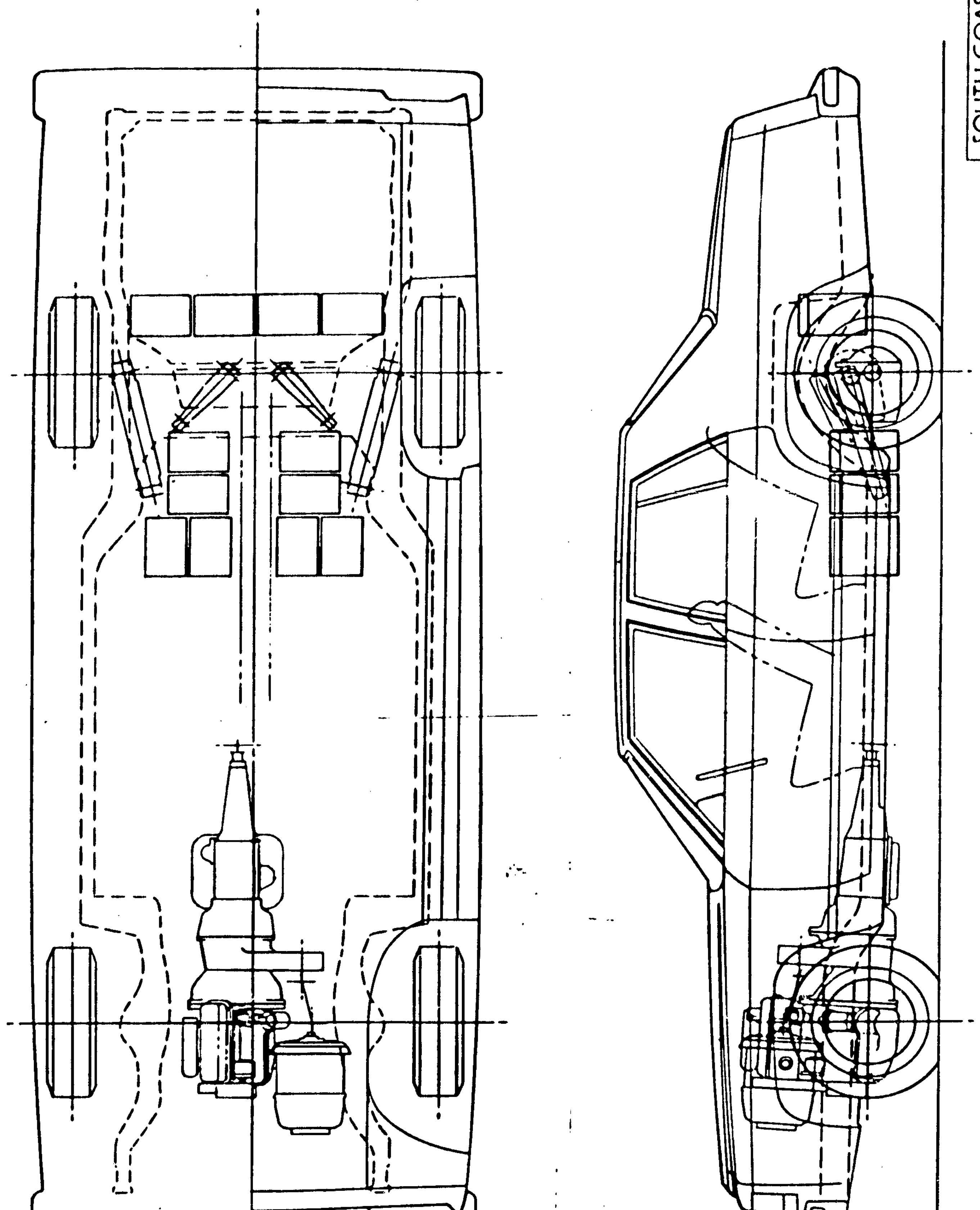
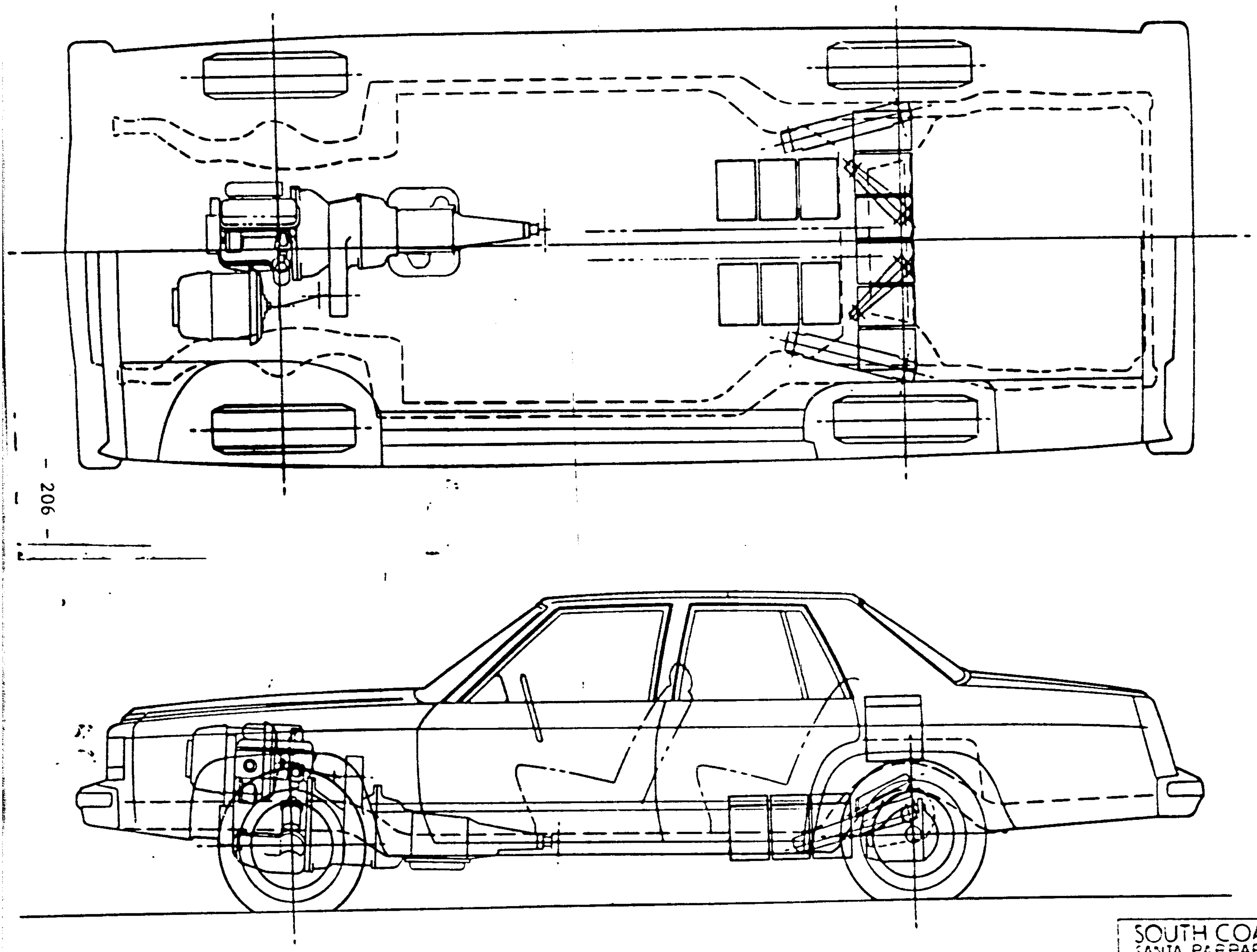


Figure 3-54



gure 3-55

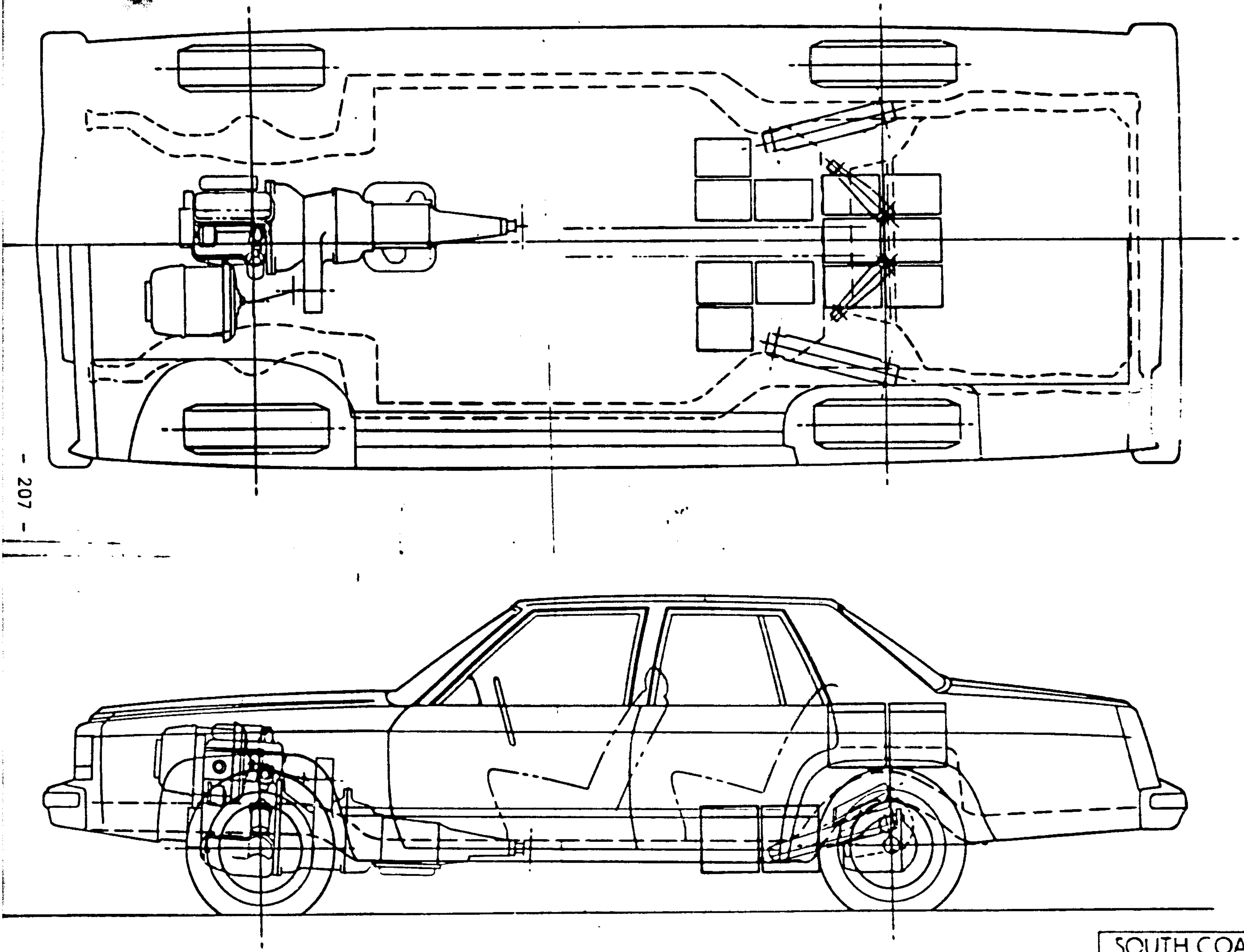
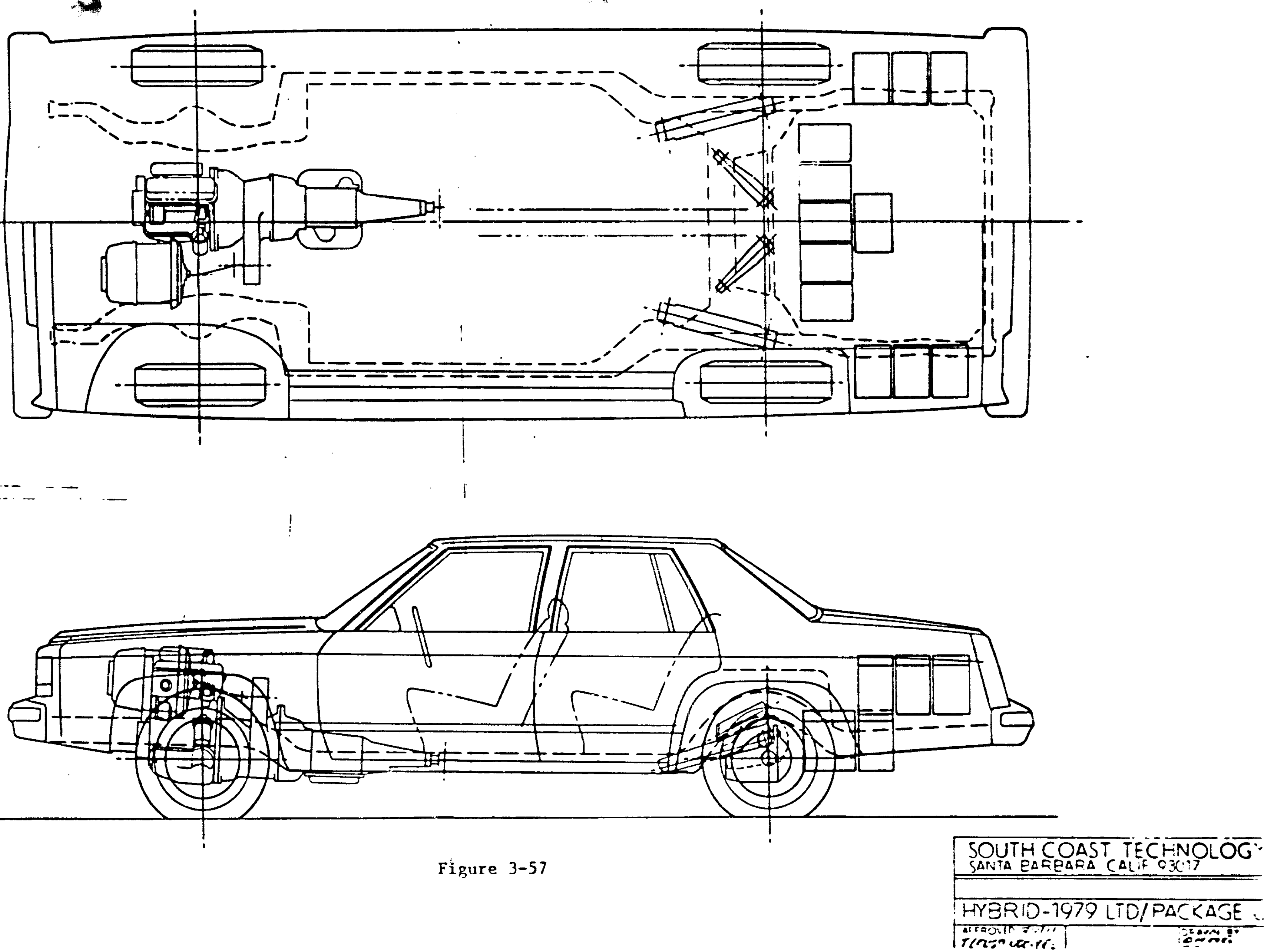


Figure 3-56

SOUTH COAST TECHNOLOGY
SANTA BARBARA CALIF 93017

HYBRID-1979 LTD/PACKAGE I



SOUTH COAST TECHNOLOGY
SANTA BARBARA CALIF 93017
HYBRID-1979 LTD/PACKAGE

Figure 3-57

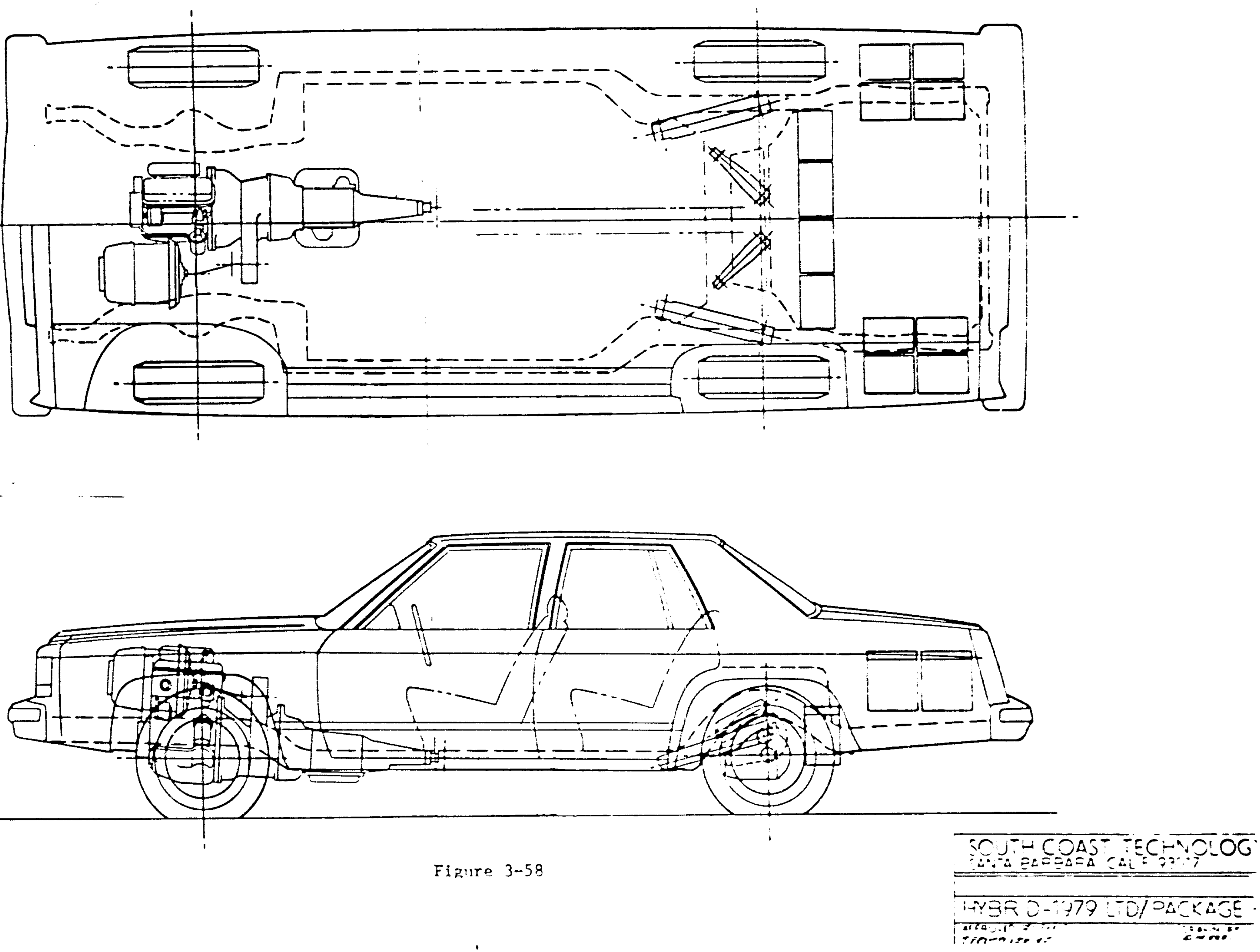

SOUTH COAST TECHNOLOGY
SANTA BARBARA CALIF 93107
HYBRID-1979 LTD/PACKAGE

Figure 3-58

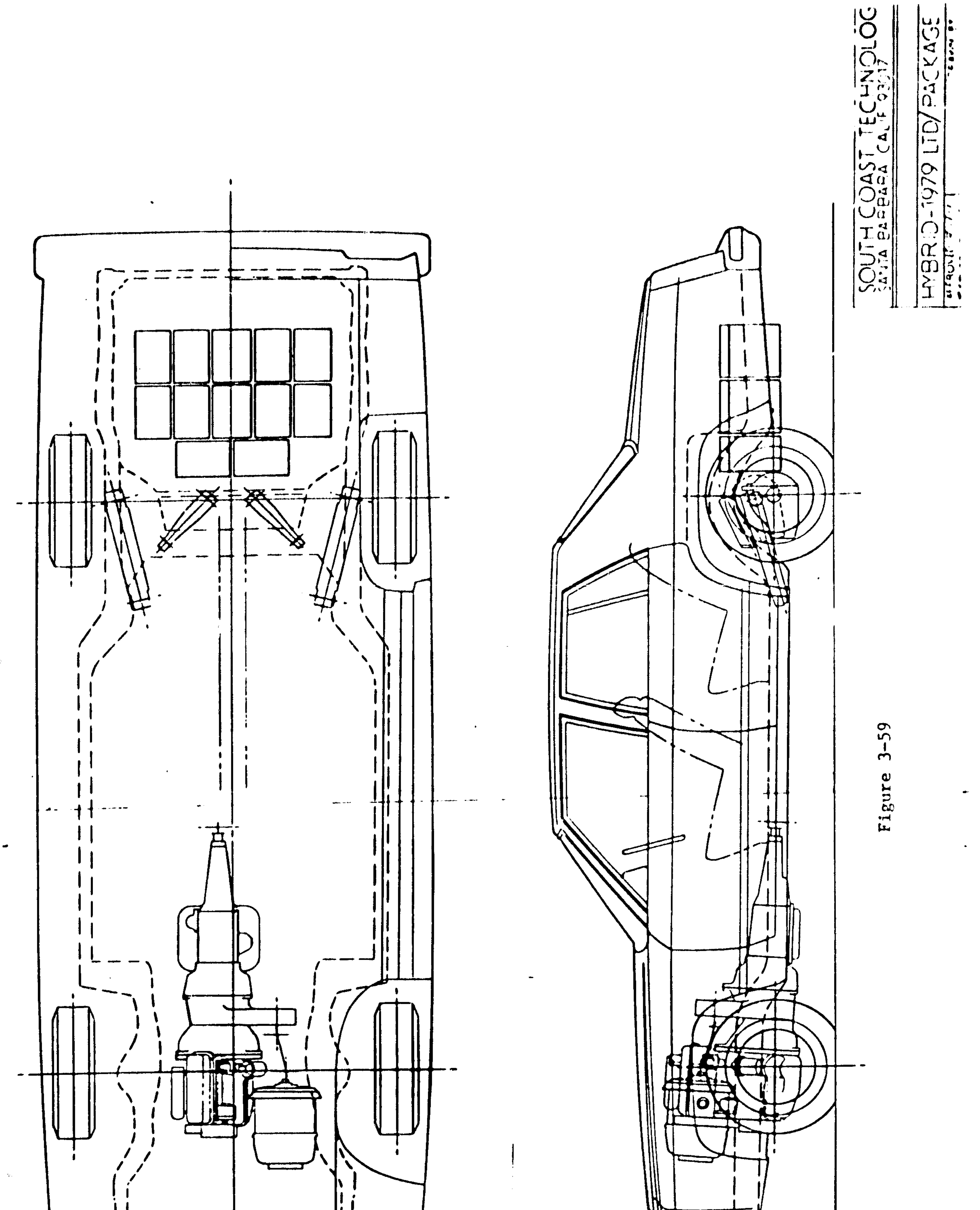

SOUTH COAST TECHNOLOG
SANTA BARBARA CALIF
HYBRID-1979 LTD/PACKAGE

Figure 3-59

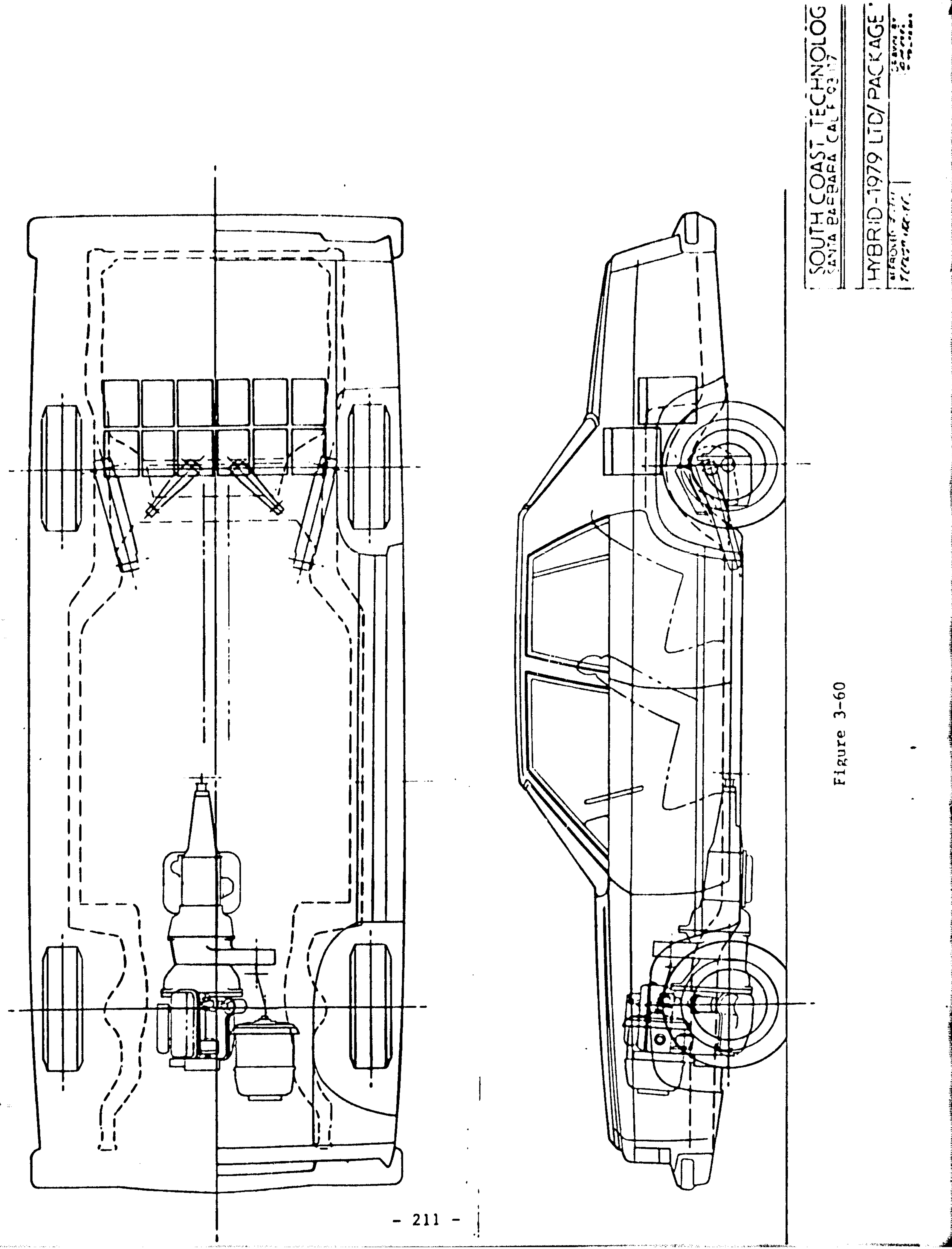

SOUTH COAST TECHNOLOG
SANTA BARBARA CAL
HYBRID-1979 LTD/PACKAGE

Figure 3-60

Table 3-11

VEHICLE PACKAGING FACTORS

		1 - 10		6 - 10	
		Value	X	Weighting	Merit Rating
A.	Safety	8		10	80
	Weight	7		8	56
	Feasibility	9		6	54
	Accessibility	9		9	81
	Space Utilization	8		8	64
					335
B.	Safety	7		10	70
	Weight	6		8	48
	Feasibility	8		6	48
	Accessibility	9		9	81
	Space Utilization	6		8	48
					295
C.	Safety	7		10	70
	Weight	7		8	56
	Feasibility	8		6	48
	Accessibility	8		9	72
	Space Utilization	8		8	64
					310
D.	Safety	7		10	70
	Weight	5		8	40
	Feasibility	8		6	48
	Accessibility	8		9	72
	Space Utilization	8		8	64
					294

Table 3-11

VEHICLE PACKAGING FACTORS

		1 - 10		6 - 10	
		Value	X	Weighting	Merit Rating
E.	Safety	8		10	80
	Weight	9		8	72
	Feasibility	9		6	54
	Accessibility	8		9	72
	Space Utilization	10		8	80
					358
F.	Safety	7		10	70
	Weight	7		8	56
	Feasibility	8		6	48
	Accessibility	8		9	72
	Space Utilization	8		8	64
					310
G.	Safety	9		10	90
	Weight	9		8	72
	Feasibility	8		6	48
	Accessibility	6		9	54
	Space Utilization	10		8	80
					344
H.	Safety	9		10	90
	Weight	9		8	72
	Feasibility	8		6	48
	Accessibility	6		9	54
	Space Utilization	9		8	72
					336

Table 3-11

VEHICLE PACKAGING FACTORS

		1 - 10		6 - 10	
		Value	X	Weighting	Merit Rating
I.	Safety	8		10	80
	Weight	8		8	64
	Feasibility	7		6	42
	Accessibility	5		9	45
	Space Utilization	9		8	72
					303
J.	Safety	7		10	70
	Weight	7		8	56
	Feasibility	9		6	54
	Accessibility	9		9	81
	Space Utilization	9		8	72
					333
K.	Safety	7		10	70
	Weight	6		8	48
	Feasibility	9		6	54
	Accessibility	9		9	81
	Space Utilization	8		8	64
					317
L.	Safety	7		10	70
	Weight	6		8	48
	Feasibility	9		6	54
	Accessibility	9		9	81
	Space Utilization	7		8	56
					309

Table 3-11

VEHICLE PACKAGING FACTOR

		1 - 10		6 - 10	
		Value	X	Weighting	Merit Rating
M.	Safety	8		10	80
	Weight	9		8	72
	Feasibility	9		6	54
	Accessibility	6		9	54
	Space Utilization	9		8	72
					332

On the basis of our preliminary battery packaging, we plan to explore alternatives F and G in greater detail during the preliminary design in order to determine whether to use a straightforward (alternative F) vs. one that offers weight distribution advantages at the expense of added design complexity (alternative G).

3.6.2 Material Substitution/Weight Reduction

Defining the material substitution and resultant weight reduction potential of a 1985 reference vehicle is a task we approached from a variety of viewpoints.

As a starting point, one must decide if any radical approach will find its way into a relatively high volume passenger car, regardless of the beneficial effect it would have on weight reduction and, thus, fuel economy. The 1985 model year is near at hand to the auto industry that must make its long lead decisions 5-7 years in advance. This led us to conclude that there would not be a high volume aluminum, or plastic composite car in 1985. Our review of literature, discussions with auto industry suppliers, and with an auto industry manufacturer confirms our assumption. Aluminum or composite cars may be introduced by 1985 but only in very limited production volumes to prove out technology which might be used in the 1990's on high volume production cars.

A second building block to our material substitution plan is determined by the actions Ford Motor Company is planning to take with the LTD. The downsized LTD was introduced in 1979 and is a very weight effective solution to a large car. It was our premise that no major changes would take place between now and 1985 except for a facelift in the early 80's, a material substitution program to reduce weight, and/or change to a more fuel efficient PROCO or diesel powerplant. An open issue is whether Ford would stick with a front engine, rear wheel drive layout, or would change to front wheel drive. Our premise of no change was reviewed with Ford, who find it to be a reasonable basis for planning. Thus, planning will be based on a carryover driveline arrangement through 1985.

The last key assumption made was that the hybrid would, whenever common to the reference vehicle, be interchangeable with or equivalent to it. The unique nature of the hybrid propulsion system will bring about dramatic fuel economy improvements and will, thus, not necessitate further unique component usage that, at best, could only marginally further improve fuel economy. To a marked degree, one could consider a hybrid propulsion system as an alternative to further downsizing or costly material substitution. Based on these assumptions, we then developed a material substitution plan for the hybrid vehicle which includes the following items:

		Alternate Materials				
Component	Material	HSLA		ALU		Plastics
Frame	Steel	X	or	X		
Bumpers	Steel	X	or	X		
Hood Outer	Steel			X	or	X
Hood Inner	Steel			X	or	X
Deck Outer	Steel			X	or	X
Deck Inner	Steel			X	or	X
Door Outers	Steel			X	or	X
Door Inners	Steel			X	or	X
Fenders	Steel			X	or	X
Wheels	Steel			X	or	X
Power Strng. Pump Hsng.	C.I.			X		
Axle Housing	C.I.	X				
Radiator	CU			X		

The above listing does not include many chassis items, as such changes for this hybrid vehicle program would be beyond our means to forecast, plan, or implement. When Ford makes a weight reduction change, we will use the new lighter parts. To assist in our vehicle planning, Ford will keep us advised of their weight reduction plans in the form of a total weight savings distributed between front and rear weight.

To arrive at a weight reduction potential for the changes outlined above, we first determined the material description and weight for the reference (1979 LTD) vehicle. A methodology for determining the weight of the equivalent part in aluminum or plastic was developed. This methodology is attached to this report as Appendix D . Using this methodology, we were able to complete columns 1-5 of Table 3-12 , Material Substitution/Weight Reduction Analysis. Columns 6-8, the estimated cost in each material, were developed for us by our subcontractor personnel who are experienced automotive cost estimators. These weight and cost factors result in the use per pound calculations shown in columns 9 and 10.

These data would support selected use of aluminum panels in large cars, a practice that should be introduced in a high price, luxury car being introduced this fall. Aluminum panels in this application were selected to achieve a desired inertia weight class and, thus, achieve better fuel economy ratings. We were surprised by the high costs for plastic components and intend to delve further into this matter during the Preliminary Design task with selected manufacturers of the plastic materials.

The total weight for the selected items, less the frame, amounts to 490 pounds in steel vs. 276 pounds in aluminum, a savings of

Table 3-12

MATERIAL SUBSTITUTION - WEIGHT REDUCTION ANALYSIS

PRELIMINARY ANALYSIS

COMPONENT	CURRENT MATERIAL	WEIGHT	ALT MATERIAL WEIGHT ALU	ALT MATERIAL WEIGHT PLASTIC	$COST/CAR STEEL	$COST/CAR ALU	$COST/CAR PLASTIC	COST PER LB. SAVED ALU	COST PER LB. SAVED PLASTIC	REMARKS
FRAME	STEEL	300	150	-	$84.00	$108.00	-	.16		REQUIRES FURTHER FEASABILITY STUDY
BUMPERS FRT & REAR										
OUTER	STEEL	59	28	53	16.52	19.94	63.72	.11	7.87	
INNER	STEEL	61	29	-	17.08	20.66				PLASTIC NOT FEASIBLE FOR INNER
HOOD OUTER	STEEL	29	18	26	8.12	12.75	31.32	.42	7.73	
HOOD INNER	STEEL	17	9	15	4.76	6.12	18.36	.17	6.80	
DECK OUTER	STEEL	26	16	23	7.28	11.45	28.08	.42	6.93	
DECK INNER	STEEL	17	9	15	4.76	6.12	18.36	.17	6.80	
DOORS OUTER FRT & REAR	STEEL	40	24	35	11.20	17.57	43.20	.40	8.00	
DOORS INNER FRT & REAR	STEEL	71	43	64	19.62	31.18	76.68	.41	8.15	
FENDERS OUTER	STEEL	60	30	29	16.80	21.60	35.28	.16	.64	
WHEELS	STEEL	110	70	-	30.80	60.90	-	.75		COST DATA NOT AVAILABLE FOR PLASTIC WHEELS
TOTAL ABOVE		790	476		220.94	316.29				COST AND WEIGHT SHOWN FOR STAMPED ALU WHEELS
TOTAL ABOVE LESS FRAME		490	276		136.94	208.29		.33		

214 pounds that could be achieved at a cost of 33 cents per pound. The frame has been excluded from this computation as the enthusiastic advertising of the virtues of aluminum frames is not supported when one attempts to find a realistic means to design and build prototype frames.

Other weight savings and related costs are excluded from our computation at this time, as they require Ford Motor Company to make the weight savings changes to the LTD. We will quantify that weight savings during the Preliminary Design phase of this program. An important aspect of the material substitution changes we have defined and those Ford will implement in the LTD chassis is that they can be made using essentially a carryover car. A new unique body for 1985 high volume production could not facilitate achieving greater weight reductions. The implications of this will be discussed further in Section 5 of this report.

3.6.3 Market Assessment/Price Sensitivity

To obtain an independent expert assessment of the hybrid vehicle market potential and to obtain an analytical judgment with respect to car prices and the impact of alternate gasoline/diesel fuel prices and electricity costs, we employed the services of Wharton Econometric Forecasting Associates, Inc.

Their specific assignments were:

1. Forecast sales volume of mid-size and full-size/luxury hybrids.

2. Use two sets of price assumptions for this purpose--a low price that was intended to represent a minimum cost pass-through and a high price that represents a retail price approximately 2 x manufacturing costs. (It should be noted that these figures are only estimates and do not represent the cost and pricing of the SCT hybrid vehicle.) Determine the sales impact of the vehicle at both prices and, thus, the price sensitivity.

   The price assumptions provided to Wharton were specified as "Midsize" Ford Fairmont Low Price Increment - $2000, high price increment - $4000. The corresponding figures for the Ford LTD were low price increment - $1750, high price increment - $3500.

3. Review the impact of alternative high and low gasoline/diesel and electricity prices,and determine sensitivity of each. Data as provided by JPL for sensitivity studies.

4. Provide backup and methodology used to arrive at forecasts.

Excluded from this study was the review of sensitivity boundary values concerning the number of passenger cars in 1985 and the average annual miles travelled. This was done for two reasons: It concentrated their effort on the key variables, and the Wharton model has values for these two factors that are close to the JPL nominal and within the sensitivity boundaries:

	JPL Value	Wharton Forecast
No. of Passenger Cars (1985)		
Nominal	113,224,000	112,130,000
- 7%	105,298,000	
+ 7%	121,150,000	
Average Annual Vehicle Miles Travelled	11,852	12,120
- 7%	11,022	
+ 7%	12,682	

Further, variations in the number of cars and miles travelled would only have a minor impact on hybrid vehicle demand in comparison to other factors such as vehicle price, gasoline, or diesel fuel price, and fuel availability.

Of interest, the Wharton model has a "Severe Regulation" scenario which would, in fact, optimize market conditions for the sale of fuel efficient hybrid vehicles. This scenario would come about as a result of stringent CAFE and emission requirements after 1985. One could hypothesize that the current petroleum shortfall, the high rate of inflation fueled by petroleum imports, combined with viable technology to improve fuel economy and CAFE such as hybrids, could lead to this scenario. Sensitivity factors leading to this situation are the total number of passenger cars in operation and the annual miles travelled. At the upper boundary limits, these factors will necessitate severe regulation.

An explanation about the delta price assumptions for a mid-size (Ford Fairmont) and full-size (Ford LTD) hybrid. The price data provided to Wharton shows that the + cost for the full-size car is less. This is due to two reasons: larger, more costly engine is deleted (a V-8 vs. a smaller I-4), and the LTD is more capable of carrying the added weight of the battery pack and related hardware without need for as much reinforcement. This latter factor could make it extremely problematic to convert a mid-size (Ford Fairmont or GM X body) to a hybrid without major structural and suspension modifications.

Results of the Wharton EFA, Inc., study are included in their entirety as Appendix B3 to this report.

Their key findings and our comments follow:

1. The added price of hybrids is very important: A price differential in the 25% to 40% range yields a market share of 25%, with volume of between 3 and 4 million units annually (by 1990), a 45%-80% price differential produces only a 5% share, with volume less than 1 million units.

   Comments: The cost/price relationship of hybrid vehicles must be carefully evaluated to avoid pricing the vehicle out of the market or establishing a design cost budget that is inadequate to develop a fuel efficient, reliable vehicle. Using a full-size/luxury car as a base for developing our hybrid, the profitability of our reference vehicle is adequate to permit a minimum cost price pass through in order to achieve hybrid volume sales that a manufacturer must achieve to improve his CAFE average while selling the larger cars that consumers still desire to purchase.

2. Maximum hybrid sales would occur if manufacturers had to replace all mid-size and larger vehicles with hybrids due to stringent CAFE and emission requirements after 1985; this could yield a 45% market share, with sales of 5-7 million, although domestic (produced units) would be lower. Comments: With the results of downsizing and introduction of improved technology (GM X body and Ford Fairmont), the fuel economy of the mid-sized cars of the 80's is becoming very respectable; and drastic action, such as hybrid option, may not be needed. PROCO and diesel engines with four speed overdrive automatic transmissions may prove to be quite adequate. It is in the larger cars with their relatively poor fuel economy that presents a CAFE challenge to the auto industry that may restrict hybrids to such vehicles in the near term.

Our discussions with research and engineering personnel of a major U. S. manufacturer support our position. However, nothing can be firm in new model planning under the current situation.

3. The real price of gasoline is critical: Each 1% change produces almost a 1% change in hybrid sales; for real electricity prices, the effect is almost exactly half as important.

Comments: Although Wharton EFA did not believe the high price of gasoline assumed in the upper bound of sensitivity (their report was issued in March, 1979), it now appears that an even higher price should be assumed for the upper limit.

From a marketing point of view, we accept the Wharton projection. From the point of view of the impact of gasoline and electric prices on our product (see Section 3.4), there is little room for varying the heat engine fraction in order to shift more of the burden to wall plug electricity. This issue will receive attention during the preliminary design task and will remain open for review during the Phase II effort.

4. The most effective way to maximize hybrid sales is with models in each market segment: Even though large cars benefit the most, the size of the mid-size/intermediate segments established in the U. S. market makes this a significant potential source of hybrid sales.

Comments: See comments under (2) above. Maximizing hybrid sales should not be an overriding consideration. Maximizing CAFE using a variety of propulsion system technologies should be the goal. Small battery electrics, fuel efficient mid-size PROCO, and diesel powered cars, etc., with hybrids used in larger, less fuel efficient vehicles may be a very fuel efficient scenario.

5. The long term petroleum fuel savings are very substantial and very sensitive to the hybrid's sales volume as well as to gasoline prices: Our baseline hybrid forecast suggests annual fuel savings of over 11 billion gallons by 1995, a 14% reduction.

   Comments: Supports the priority being given to this program by JPL and the active interest of the auto industry. These savings would even be greater using the upper boundary values for number of passenger cars in 1985 and the average annual miles travelled.

The Wharton study focuses attention on the need to evaluate the delta cost to delta price relationships in automotive pricing. A simplistic formula cannot apply in establishing the cost/price relationship of our proposed hybrid vehicle.

Factors that must be considered are as follows:

1. The proposed hybrid is positioned in the full-size/luxury market segment which has unit profits at the extreme upper end of automotive products.

   Manufacturers would, thus, be more than willing to retain these profit margins rather than to seek even higher profits.

2. The alternatives available to manufacturers to enable them to retain highly profitable larger cars are limited.

Although the market for large cars will undoubtedly remain large due to Americans past history of driving large cars, there is and will continue to be a shift into smaller cars.

The current gasoline crunch has brought an immediate shift toward smaller cars; and a continuing scenario involving high gasoline prices, shortages, and general inflation will perpetuate that movement. In a profit oriented industry, this shift in car size presents a major threat to profits. Thus, manufacturers could be expected to pass on the delta cost of a hybrid at a nominal markup.

3. Auto industry pricing is not done on a cost basis, but rather on a cost, image, and competitive price basis. Markups to dealer cost and to retail, thus, depend on a wide variety of factors.

As a recent study[8] shows, the retail price of an average 1978 model year passenger automobile is 131 percent of its manufacturers cost.

A study done to support fuel economy rulemaking[7] examined the delta price for making changes to a car. Their formula, which follows, would support an assumption of a nominal cost pass through for a hybrid vehicle which would be dependent on the variable cost, investment, and rate of return:

$$\text{Delta Price} = (1 + 25\%)(GR \times CI + \text{Delta VC})$$

$$GR = NR/(1 - TR)$$

where

GR - is the implied gross rate of return on required investment

CI - is the capital investment per produced unit (for example, 600 million dollars, divided by 400,000 units per year of a converted facility would require 150 dollars per car)

VC - is the manufacturers variable cost per produced unit

NR - is the desired net rate of return on capital investment

and

TR - is the applicable tax rate of the manufacturer

Data from a more recent cost study done by DeLorean Engineering Associates shows the delta cost/price relationship of selected safety systems at a price-to-cost relationship of 140%. This example involves high tooling cost with a one year amortization.

It is interesting to note that these studies have been done by different support contractors who base their data on prior auto industry experience. The studies cited cover Chrysler and GM practice. Our own experience at Ford and AMC would indicate that these studies are a reasonable representation of the facts.

4. DESCRIPTION OF OPTIMIZED HYBRID VEHICLE DESIGN

As a result of the Design Tradeoff Studies, we are now in a position to provide a preliminary description of our optimized vehicle conceptual design. As design is an iterative process, it should be anticipated that some of these preliminary descriptions may be modified during the preliminary design task.

General Vehicle Description

The hybrid by SCT is a full-size, six passenger sedan built on a Ford LTD chassis, frame and suspension components. It has a conventional front engine-rear drive layout.

The car has an OAL of 5.31 m, OAH of 1.39 m, and is 1.97 m wide--all dimensions that are identical to the Ford LTD.

All interior dimensions are identical to the Ford LTD, and there will be some reduction in usable trunk space depending on the battery packaging solution that is selected. All the battery packaging alternatives preserve a major part of the large LTD trunk.

Curb mass is estimated at 1980 kg (nickel-iron batteries), or 2080 kg (lead-acid batteries).

Propulsion System

The propulsion system is a parallel hybrid. The heat engine and motor are coupled together by a chain coupling and drive, and a four speed automatic transmission with a lockup torque convertor. Torque convertor lockup will be provided on at least the top two gears. Provision is made to decouple the engine from the rest of the drivetrain and shut it down under the following conditions:

- Deceleration

- Idling

- System power demand below a threshold level (with battery state of charge above a discharge limit)

A summary of the system design features follows:

Heat engine: VW Rabbit 1460 cc, 53 kw fuel-injected gasoline engine with design modifications to permit operation in an on-off mode.

Electric motor: Siemens 1GV1 separately excited.

Motor controller: Combination armature chopper and field chopper. Armature chopper is of the transistor type, with the output current limited to a value in the 125-140 amp region. Field chopper is also of the transistor type, incorporating control circuitry, as used in the SCT electric conversion of a VW Rabbit, to limit motor maximum current to a pre-selected value. This value will correspond to a peak motor output of no more than 30 kw.

Battery pack: Nominal 120 V. Based on life cycle cost and packaging considerations, nickel-iron is the preferred battery type, with a battery mass of 270 kg. Lead-acid may be substituted for this during the Preliminary Design Task if we conclude that nickel-iron technology is not compatible with the time constraints of the Near Term Hybrid Vehicle Program. The lead-acid battery pack would weigh about 355 kg. Nickel-zinc is not viewed as a viable alternative because of high life cycle cost. In either case, a special design will be required for the hybrid vehicle; the ISOA (golf cart) module size/voltage combination will not be suitable.

System controls: Microprocessor based control system incorporating a bimodal control strategy. On the first mode, energy is withdrawn from the battery pack until the battery discharge limit is reached, corresponding to about 60% depth of discharge; on this mode, the heat engine is used in a peaking capacity and to meet steady-state cruise requirements. On the second mode, the heat engine supplies the average energy requirements and maintains the average battery state of charge at the battery discharge limit; on this mode, the electric motor is used in a peaking capacity, to provide regenerative braking, and to supply accessory loads at idle. The control strategy is sensitive to the system power demand (acceleration pedal position), battery state of charge, and vehicle speed; it will control the heat engine, electric motor, and transmission to accomplish the following:

(1) Keep the sustained power output of the electric motor down to a value consistent with its nominal rating and the sustaining power capability of the battery pack.

(2) Keep the heat engine, when it is operating, as close as possible to its best (lowest bsfc) operating point.

(3) Keep the electric motor speed high enough, during deceleration, to provide regenerative braking capability down to as low a vehicle speed as is possible.

Body and Structure

The hybrid will utilize a separate frame and body to be compatible with the projected 1985 LTD. These will be of generally conventional steel construction, with the substitution of aluminum or plastic composite components in areas where their replacement can be shown to be cost effective.

Projected Performance, Fuel Economy, and Energy Consumption

The acceleration characteristics projected for the optimized hybrid vehicle are shown in Figure 4-1, and the maximum instantaneous gradeability as a function of speed is plotted in Figure 4-2. Gradeability over extended distances, for the nickel-iron batteries, is indicated in Table 4-1. Yearly average fuel economy and wall plug energy consumption are estimated at 16.5 km/l (39 mpg) and .15 kw-hr/km, again, for nickel-iron batteries. These numbers are predicated on a 20% degradation in available specific energy, relative to the ISOA goals, because of the smaller cell size required for the hybrid. With optimization of the control strategy, we would expect a shift upward of about 10% in fuel economy and a corresponding upward shift in energy consumption; however, these numbers provide an adequate measure at this stage of the hybrid's efficiency. The distributions of battery output power on the urban and highway cycles are shown in Figures 4-3 and 4-4.

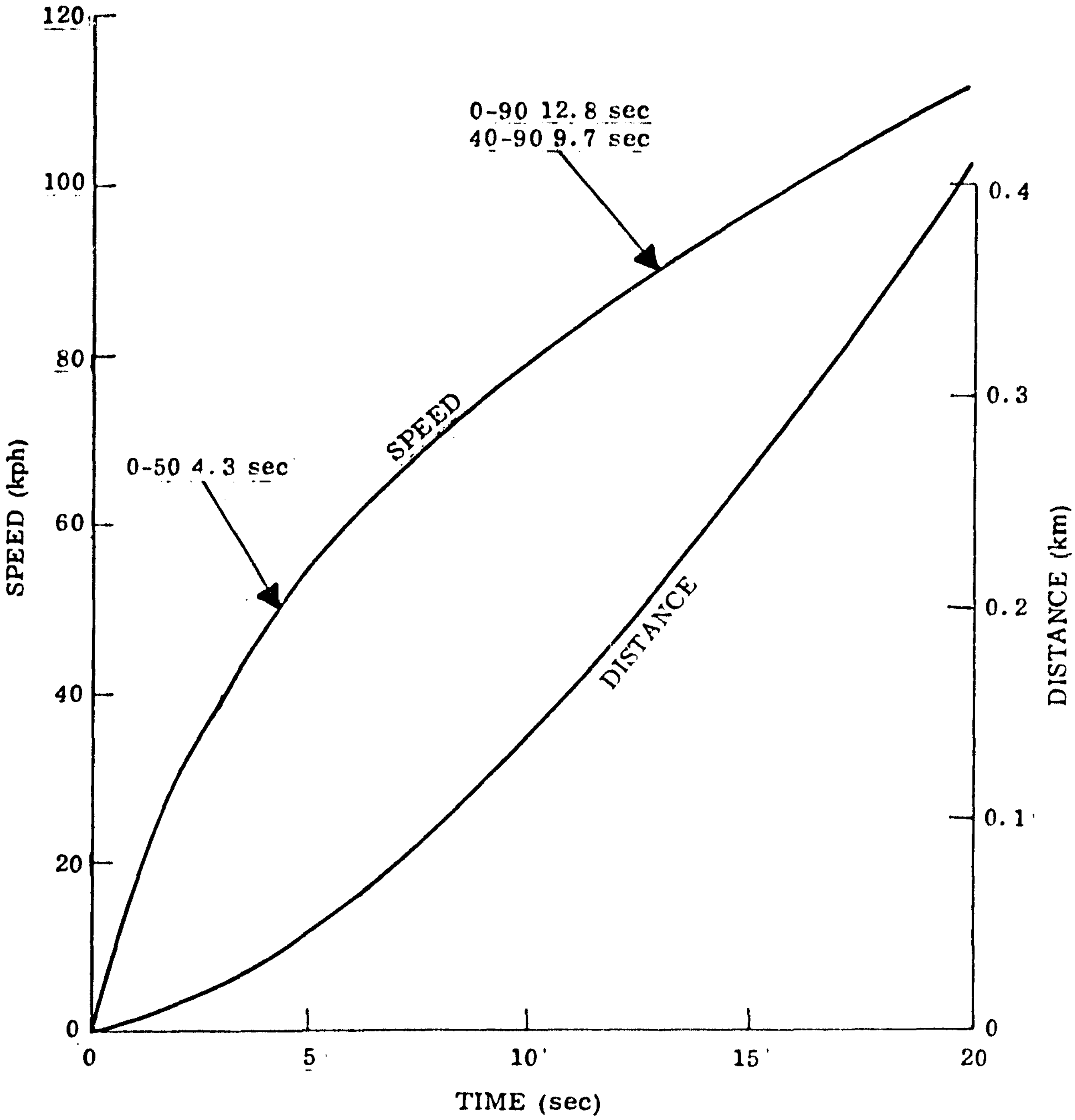


Figure 4-1 Optimized Hybrid Acceleration Characteristics

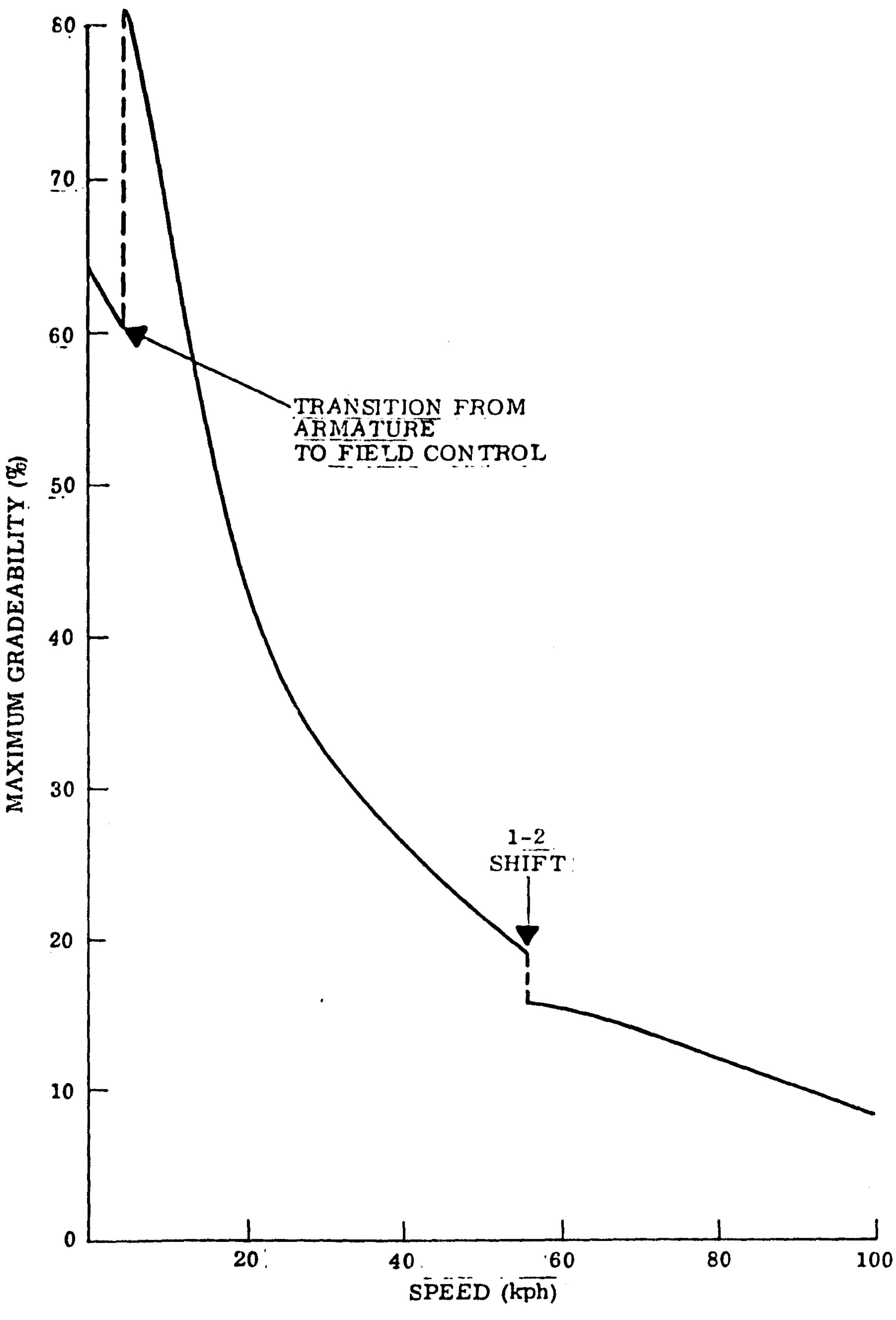


Figure 4-2 Optimized Hybrid Gradeability Characteristics

Table 4-1. GRADEABILITY OF OPTIMIZED HYBRID VEHICLE WITH NICKEL-IRON BATTERIES

		Distance (km)*	
Grade	Speed	Specification	Estimated
3	90	Indef.	Indef.
5	90	20	29
8	85	5	15
8	65	Indef.	102 (≈ Indef)
15	50	2	20

* Assumes battery is allowed to drop from 60% DOD to 90% DOD. Battery specific energy assumed to be 80% of that projected for ISOA batteries.

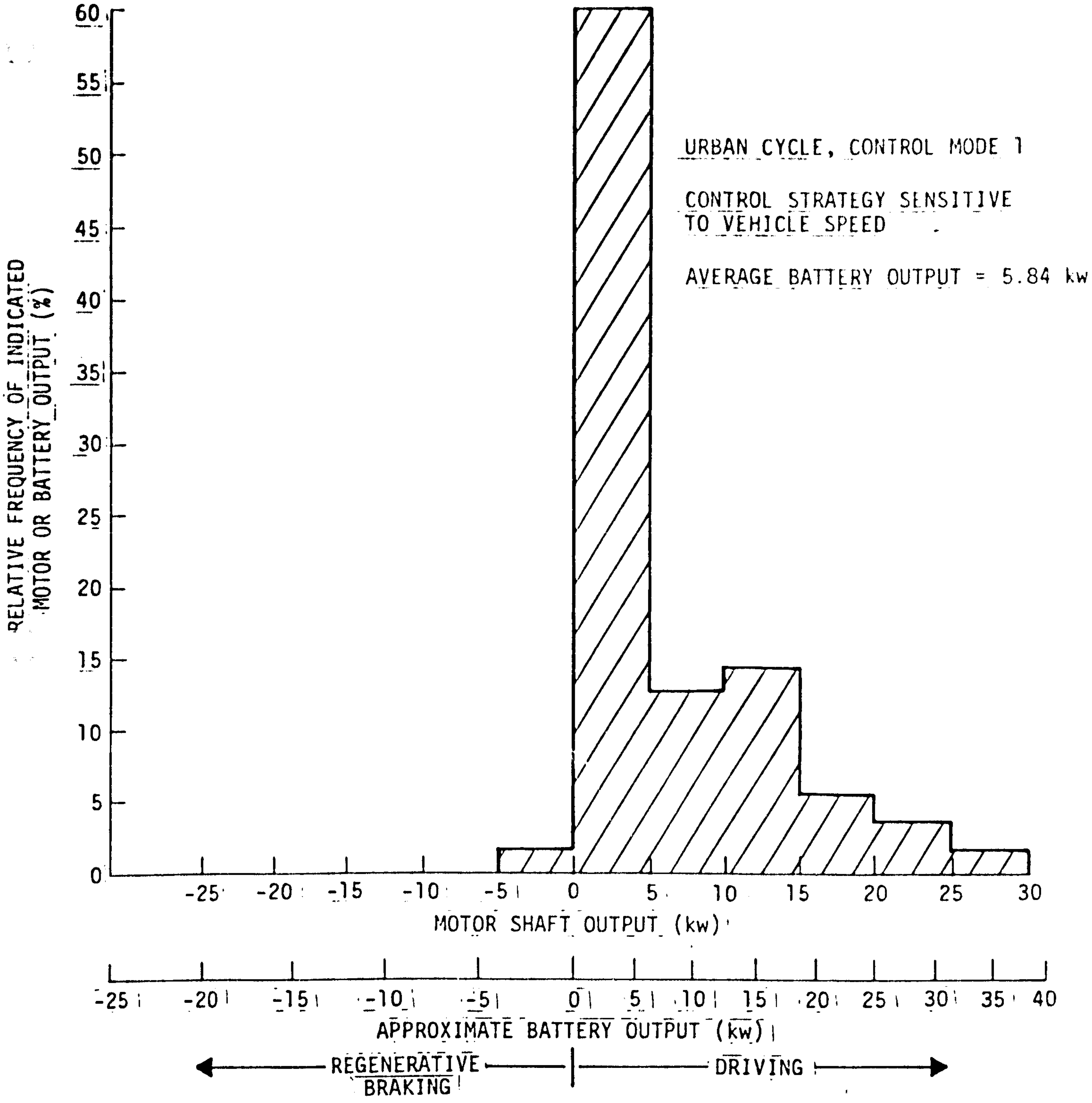


Figure 4-3 Distribution of Motor and Battery Output

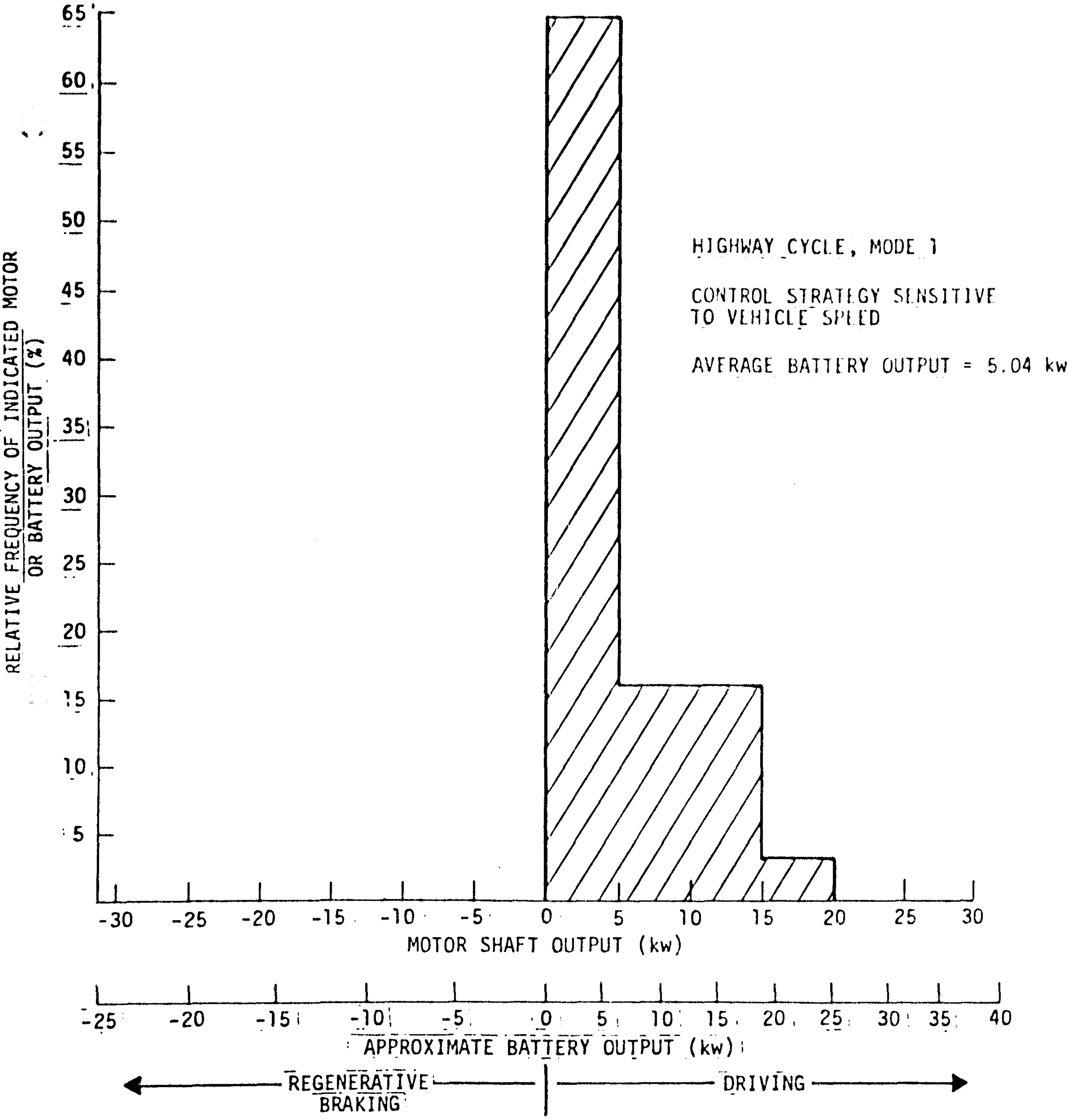


Figure 4-4 Distribution of Motor and Battery Output

Projected Retail and Life Cycle Costs

Retail cost with a nickel-iron battery pack is projected to range from $10,250 to $11,800, with the actual value dependent on pricing strategy. The reference vehicle projected retail cost is estimated at $7,650. Life cycle cost estimates range from 9.3¢/km to 10.3¢/km, again, depending on pricing strategy for the hybrid manufacturing cost increment and replacement battery OEM cost. The lower of these values is fairly close to the life cycle cost estimate for the reference vehicle, which is 8.7¢/km.

Sensitivity Analysis

Sensitivity of the optimized hybrid vehicle life cycle cost to variations in fuel and electricity prices is summarized in Table 4-2. Life cycle costs are essentially identical only for the minimum retail cost case, at +30% fuel prices. These results are somewhat more pessimistic than those obtained in the system level studies and earlier phases of component/subsystem tradeoff studies; however, they reflect a more detailed and realistic estimate of the incremental cost of the hybrid, and of the battery energy and life characteristics likely to be obtained under the hybrid's operating conditions.

Table 4-2. LIFE CYCLE COST SENSITIVITY

(¢/KM)

	Reference Vehicle	HYBRID (Optimized Vehicle)	
		Cost Case 1 (Low)	Cost Case 2 (Nominal)
Nominal	8.7	9.3	10.3
Fuel +30%	9.5	9.7	10.6
Fuel -30%	7.8	8.9	9.9
Electricity +30%	8.7	9.5	10.4
Electricity -10%	8.7	9.2	10.2

5. CONCLUSIONS AND RECOMMENDATIONS

The conclusions drawn from the Design Tradeoff Studies may be summarized as follows:

Propulsion System

The mode of operation which offers the greatest potential fuel savings involves running the heat engine only when it is needed. This requires it to be started and brought up to full power almost instantaneously in order to meet the driver's power demands. This type of operation appears to be feasible and the technology required to accomplish it can be developed within the time constraints of the Near Term Hybrid Vehicle program.

It is not possible to, simultaneously, maximize fuel economy and achieve a life cycle cost which is comparable to that of a conventional vehicle performing the same mission. Maximum fuel economy occurs for a configuration which is too close to a pure electric vehicle to be both cost effective and meet the performance requirements of the hybrid. It is, however, possible to achieve fuel economy on the order of 2 to 3 times that of a conventional vehicle, with a comparable life cycle cost.

To actually achieve a life cycle cost which is no higher than that of a conventional vehicle, the fuel savings of the hybrid must be accumulated over a long vehicle life (at least 10 years, at the nominal annual mileage projections made by JPL), and at fuel costs which are at the upper limit of the sensitivity boundaries (30% above nominal projections). In addition, the manufacturing cost increment over a conventional vehicle, and the replacement battery OEM cost would have to be passed on to the consumer at a level which is considerably less than the factor of 2 specified by JPL.

As in the case of an electric vehicle, the two most significant factors in keeping the life cycle cost down to a reasonable value are the retail price (hence, manufacturing cost) increment and the ratio of battery replacement cost to battery life. In the hybrid vehicle, both these factors can be reduced by reducing the power rating of the electric drive portion of the system relative to the system power requirements. Even when a bias in favor of better fuel economy is applied (at some sacrifice in life cycle cost), we come to the conclusion that the peak rating of the electric drive portion of the system should be no more than 35% of the system requirement for lead-acid batteries, and less for nickel-iron and nickel-zinc types. Moreover, the peak power rating of the electric motor should correspond to working the battery near the upper limits of its peak power capacity. High energy density appears to be somewhat less important for the hybrid than for a pure electric vehicle, and the economic tradeoff appears to favor higher voltages (around 120V) even if these entail some loss in energy density. This, in turn, requires smaller cell sizes than are under development for the ANL ISOA (improved state-of-the-art) battery program, since the hybrid battery pack is smaller, and implies a unique battery design for the hybrid.

The type of battery which appears to be most suitable for the hybrid, from the point of view of minimizing life cycle costs, is nickel-iron, with lead-acid a reasonably close second. Although nickel-zinc is highly desirable because of its high power and energy density, its short life and high cost puts it well behind the other two from the standpoint of economics. These conclusions assume that the ANL goals for ISOA batteries are all equally probable of attainment within the time frame of the ISOA battery program. A critical review of the state of the art and, hence, of this assumption, is underway; and the conclusions in this area are subject to modification during the Preliminary Design Task.

The characteristics of the hybrid propulsion system, with respect to the effects of various parameters on its fuel and energy efficiency, give rise to a conclusion which appears rather startling on first glance, but inevitable upon further reflection. That is, the hybrid is much less sensitive than a conventional vehicle is, in terms of the reduction in total fuel consumption and resultant decreases in operating expense, to reductions in vehicle weight, tire rolling resistance, etc., and also to propulsion system and drivetrain improvements which are designed to improve the bsfc of the engine under low road load conditions (for example, use of diesel or stratified charge engines, continuously variable transmissions, etc.). Consequently, once the step to the incorporation of a hybrid system is made, this implies that the most appropriate policy toward additional radical modifications should be one of conservatism and justification on purely economic grounds, rather than technological glitter.

Vehicle Considerations

The vehicle packaging studies indicate that the packaging of a hybrid propulsion system in a vehicle such as the Ford LTD can be done with a minimum of sacrifice of luggage capacity. This situation is quite unlike that of a high performance pure electric vehicle which uses near term technology, and supports our belief that a hybrid vehicle, if produced by a major manufacturer, would come into being as a modification or option on an existing line of conventional vehicles, not as a unique car line.

Impact of Tradeoff Study Results on Phase II Planning

As originally conceived by JPL, the end product of these Phase I studies would enable JPL to select one or more contractors to proceed into a Phase II Final Design and Integrated Test Vehicle Fabrication.

The scope of the Phase II effort is broad as it includes not only a hybrid propulsion system that will meet all its objectives, but also, a new vehicle design that would offer significant advantages over existing production vehicles in terms of weight, aerodynamic shape, and rolling resistance.

As our program unfolds, it becomes increasingly apparent to us that it would be possible to achieve the JPL hybrid vehicle objectives without the need to undertake a costly body development program. Modifications to package the propulsion system and battery pack can be easily accommodated within the confines of a modified carryover body such as the Ford LTD. We are certain that auto industry planners would adopt the same approach.

We, thus, recommend to JPL that an alternate Phase II program be structured that will conserve available hybrid vehicle R & D funds by changing the direction from an all new hybrid vehicle to a hybrid propulsion system offered in a modified production car. A discussion of the basis for the recommendation and its advantages follow:

1. Cost of a new body development is extremely high and could use close to half the available Phase II R & D funds allocated for the hybrid program.

2. If a new body were to be developed, only a few areas could be developed to be better than a modified production body. These areas and the achievable results with a modified body are discussed below:

   - Aerodynamic drag. Our package studies indicate that packaging the proposed propulsion system components in the LTD reference vehicle offers little or no flexibility with respect to front end shape and hood height. (See Section 3.6.1) This would limit the range of improvement possible with a new body. Planned weight reduction bumpers and minor sheet metal changes can improve front end drag of the carryover car.

   - Vehicle weight. Our assumption for this 1985 model year vehicle rule out drastic body and structural changes. Without the need to redesign an all aluminum or composite body, one can achieve material substitution changes, as discussed in Section 3.6.2, with either a new body or with a carryover modified car. Plastic or aluminum panels can be made without the need to undertake the det. led engineering job of making new parts. Plastic parts can, in fact, be molded from production panels; and aluminum can be formed using production parts as die models or hammerform checking fixtures. Note that rolling resistance changes due to tires, lubricants, etc., are independent of body shape.

3. We believe that the most important part of this program is to develop technology which is transferable to the automobile industry. It seems to us that the way to do this is not to spend half or more of the effort and funds on developing new body designs, but to concentrate the bulk of the effort on the development of propulsion systems. It is in this area that this program might have something to offer the automobile industry in terms of transferable technology; we seriously doubt whether we, or our fellow Phase I contractors, are going to teach it a whole lot about body design. If the Phase II program were structured around the modification of production cars, rather than building ground up vehicles, it would permit multiple Phase II awards, with more propulsion system concepts being represented. The program as a whole would have a better chance of producing transferable technology.

We would urge that JPL redirect the Phase II effort prior to issuing the request for a proposal covering Phase II. We would suggest a redirect to specify the use of a modified production vehicle.

6. REFERENCES

1) "Mission Analysis and Performance Specification Studies Report," Appendix A (Task 1 Report for Phase I of the Near Term Hybrid Vehicle Program), JPL Contract No. 955189, South Coast Technology, January, 1979.

2) R. G. Fitzgibbons & L. H. Lindgren, Estimated Weights and Manufacturing Costs of Automobiles, Contract No. DOT-TSC-1067, Task 1 Report, Rath & Strong, Inc., 1975.

3) Near Term Electric Vehicle Program, Phase I, Final Report, General Electric Company, August, 1977.

4) Personal discussion between Dr. Seiffert, Director of VW Research and H. Siegel regarding on-off engine mode experience of VW, February, 1979.

5) Fuel Consumption, Emissions, and Power Characteristics of the 1975 Ford 140 CID Automotive Engine - Experimental Data, U. S. Department of Transportation, Office of the Assistant Secretary for Systems Development and Technology, Office of Systems Engineering, November, 1976.

6) Near Term Hybrid Passenger Vehicle Development Program - Phase I Assumptions and Guidelines as provided by JPL.

7) Rulemaking support paper concerning the 1981-84 Passenger Auto Average Fuel Economy Standards. U. S. Department of Transportation, July, 1977.

8) Letter and attachment to R. Schwarz from R. F. Weber, Advanced Tire Engineer, April, 1979.