9950-337

NEAR TERM HYBRID PASSENGER VEHICLE DEVELOPMENT PROGRAM, PHASE I

CONTRACT NUMBER 955188

FINAL REPORT

APPENDICES C & D

OCTOBER 1979

PREPARED BY

MINICARS, INC.
55 DEPOT ROAD
GOLETA, CALIFORNIA 93017

(NASA-CR-163331) NEAR HYBRID PASSENGER VEHICLE DEVELOPMENT PROGRAM, PHASE 1. APPENDICES C AND D, VOLUME 2 Final Report (Minicars, Inc.) 352 p HC A16/MF A01 CSCL 13F G3/85

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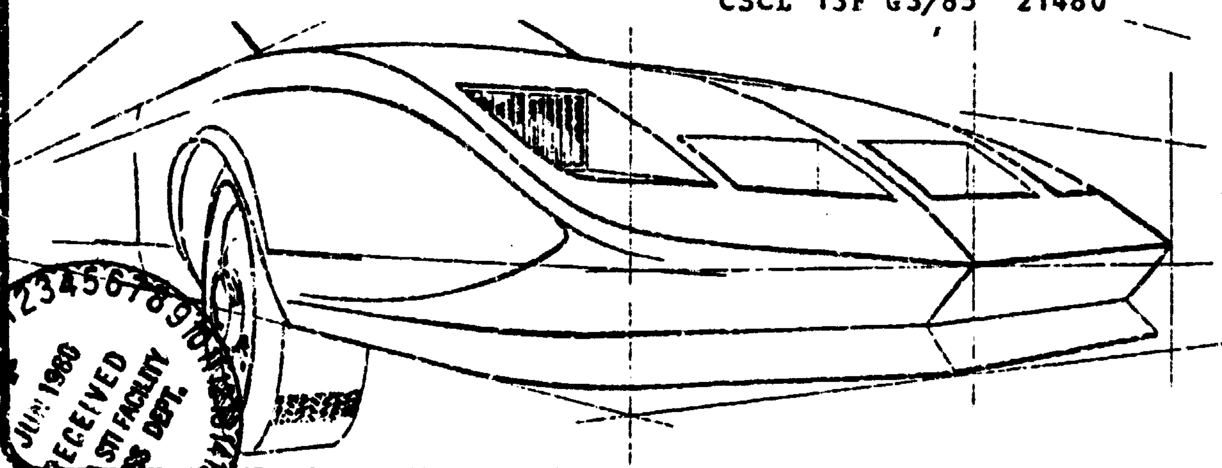

This work was performed for the Jet Propulsion Laboratory, California Institute of Technology, sponsored by the National Aeronautics and Space Administration (NASA) under Contract NAS7-100, pursuant to an interagency agreement between the Department of Energy (DOE) and NASA.

minicars, inc.

55 Depot Road, Goleta, California 93017 (805) 964-6271
TELEX 677605, Telecopier (805) 964-7669

Minicars Report Number
FR-4500-09-79-II

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PRELIMINARY DESIGN DATA PACKAGE

This report contains information prepared by Minicars, Inc. under JPL sub-contract. Its content is not necessarily endorsed by the Jet Propulsion Laboratory, California Institute of Technology, or the National Aeronautics and Space Administration.

27 August 1979

The present document is a revised version of the Minicars, Inc. Preliminary Design Data Package", dated 25 July 1979. The revision consists primarily of an addendum, which is included at the end of the report. Otherwise, only minor changes (mostly typographical) have been made in the test itself. It should be noted that in some cases (notably, the electronic control system hardware design), the preliminary design has been supplemented in Minicars' Technical Proposal for Phase II of the Near Term Hybrid Vehicle Program.

SR-4500-07-79

NEAR TERM HYBRID PASSENGER VEHICLE DEVELOPMENT PROGRAM PHASE I

CONTRACT 955188

PRELIMINARY DESIGN DATA PACKAGE

25 JULY 1979

REVISED 27 AUGUST 1979

SUBMITTED TO:

JET PROPULSION LABORATORY
CALIFORNIA INSTITUTE OF TECHNOLOGY
4800 OAK GROVE DRIVE
PASADENA, CALIFORNIA 91103

SUBMITTED BY:

MINICARS, INCORPORATED
55 DEPOT ROAD
GOLETA, CALIFORNIA 93017

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SECTION 1

INTRODUCTION

The Near Term Hybrid Vehicle (NTHV) Preliminary Design activity has picked up where the trade-off studies left off. The design has been fleshed in to a considerable level of detail, which has been made possible by going beyond the trade-off studies into areas that are not amenable to analytical quantification. In addition, the analytical procedures of the trade-off studies have been applied wherever possible, to measure the effects of design choices on petroleum savings and life cycle cost. Updated results are presented herein.

The NTHV preliminary design reflects a modified Chevrolet Citation, one of the new General Motors X-body compact cars. This vehicle represents the latest Detroit thinking in packaging and weight reduction, and it meets the JPL minimum requirements on vehicle dimensions.

The NTHV is powered by a turbocharged Volkswagen Rabbit diesel engine, which replaces the X body unit, and a 24 kW (peak) compound dc electric motor. The engine and motor are arranged in a parallel configuration, with both operating through a modified General Motors X body automatic transmission. This transmission is coupled to the engine and motor by clutches, and is shifted under the control of an on-board computer. All of the powertrain components are located in the original engine compartment, and the front-wheel drive is retained.

The onboard computer is a distributed-processing system with independent processors, consisting of two microprocessor-based subsystems that communicate with each other through input/output ports. Each system is a Motorola 6801 microprocessor with its own Read Only Memory (ROM), Random Access Memory (RAM), and peripherals. The Motorola 6801 is an eight-bit NMOS device with 65K bytes of address space with an 8 x 8 bit hardware multiply.

The battery pack consists of twelve 6-Volt improved-state-of-the-art lead-acid batteries arranged in two blocks—one ahead of the radiator core support, and the other above or behind the rear axle. The front block of batteries necessitates lengthening the vehicle by 46 cm (18 in), which improves the aerodynamics and, in our opinion, the exterior styling, which is shown in Figure 1-1. The

Figure 1-1. Minicars Near Term Hybrid Vehicle

rear has also been slightly restyled to improve the appearance and aerodynamics (and slightly increase the luggage volume).

This report is organized to present the derivation of the preliminary design, as well as the preliminary design itself. Section 2 presents the starting point for the preliminary design—the results of the trade-off studies in summary form. This is basically a repeat of material included in the Trade-off Studies Report. Section 3 covers various aspects of the overall vehicle design: the selection of the design concept and the base vehicle (the Chevrolet Citation), the battery pack configuration, structural modifications, occupant protection, vehicle dynamics, and aerodynamics. Section 4 presents the powertrain design, including the transmission, coupling devices, engine, motor, accessory drive, and powertrain integration. Section 5 treats the power conditioning unit (the motor controller), and Section 6 discusses the battery type, duty cycle, charger, and thermal requirements. Section 7 describes the control system (electronics): the identification of requirements, software algorithm requirements, processor selection and system design, sensor and actuator characteristics, displays, diagnostics, and other topics. Section 8 covers the environmental system: heating, air conditioning, and compressor drive. Section 9 contains the specifications, weight breakdown, and energy consumption measures. Section 10 identifies advanced technology components, and Section 11 lists the data sources and assumptions used.

It may be useful to key our discussions to the JPL Data Requirement Description No. 3, Preliminary Design Data Package.

Item No. 1, List of all data sources utilized, is found in Section 11.1.

Item No. 2, List of significant assumptions, is found in Section 11.2.

Item No. 3, Brief description of the design process methodology employed, is found throughout Sections 3 through 8.

Item No. 4, Rationale behind major design decisions, is likewise covered in Sections 3 through 8.

Item No. 5, Identification of any advanced technology components, is found in Section 10.

Item No. 6, Description of the preliminary design in detail, is found throughout Sections 3 through 8. Performance projections for individual components are also found in the appropriate sections, and in the Design Trade-off Studies report. Total vehicle performance is found in Section 9.1 The petroleum impact of the design is found in Section 9.3

Item No. 7, Quantification of Energy Consumption Measures, is found in Section 9.3

For Item No. 8, Description of software developed, it should be noted that no software was developed in the Task 3 effort.

SECTION 2

TRADE-OFF STUDY SUMMARY RESULTS

The preliminary design is a direct outgrowth of the design trade-off studies; most of the critical decisions or choices were made in the Task 2 effort, and served as a jumping-off point for further design definition. These decisions have already been extensively documented,[1] and there is little to be gained from repeating that information here. Rather, our intent is to summarize the results of the design trade-off studies, and to use this summary as the catalyst of the preliminary design description that follows in subsequent sections.

2.1 WEIGHT

The most recent new-model introduction in the U.S. is the General Motors line of X-body cars. These front-wheel drive vehicles represent the latest production developments in packaging and weight reduction, and the passenger compartment meets the JPL minimum requirements for a five-passenger car. As a result, it is a good base upon which to design a hybrid vehicle, and, in particular, it is a good base for projecting the NTHV curb weight (that estimate was 1754 kg [3867 pounds].

2.2 POWERTRAIN CONFIGURATION

The heat engine and electric motor are arranged in a parallel configuration because it is the most efficient layout overall and it permits flexibility in the relative power contributions from the two power sources.

2.3 SUBSYSTEM SIZING

The desirable range of battery capacities was narrowed to 10.5 to 14.7 kW-hour on the basis of benefit-cost (life cycle cost change) considerations for various combinations of vehicle subsystems. When accessory loads are added, the choice is biased to the upper end of the range, which corresponds to an 84-Volt battery pack. At this voltage level, the optimal choices are a 24 kW (peak power) electric motor and a 48.5 kW (peak power) heat engine.

2.4 SUBSYSTEM SELECTION

Three dc motors, all with similar performance and cost properties, were considered in the trade-off studies. Of these, the dc compound motor was selected because it requires a less complex control system design. Transmission choices focused on computer-controlled shifting of either a manual or an automatic transmission, on the basis of keeping the driver workload to a level commensurate with today's passenger cars. Performance characteristics, fuel economy, and life cycle costs are insensitive to ten percent variations in either the low gear ratios or the final drive ratio. The computer-controlled manual transmission produced better petroleum savings than the computer-controlled automatic transmission, but there are serious questions concerning the availability and public acceptance of the former compared to the latter.

Battery choices were confined to nickel-iron, nickel-zinc, and lead-acid systems. The nickel-iron battery has the longest cycle life and therefore the lowest life cycle cost. The nickel-zinc battery has the highest specific energy and thus produces the highest petroleum savings. However, near-term availability still favors the improved state-of-the-art lead-acid battery.

2.5 REGENERATIVE BRAKING AND ENVIRONMENTAL CONTROL

The advantages (including cost effectiveness) of regenerative braking have been shown in previous studies and were confirmed in the trade-off studies. To allow for heating the passenger compartment and/or the battery pack, independent of whether the heat engine or the electric motor is running, a petroleum-burning heater was specified. Air conditioning, on the other hand, should be accomplished by a conventional automotive compressor.

2.6 OPERATIONAL STRATEGY

In general, the batteries would be depleted as much as possible during each day, taking into consideration the need for reserve capacity to meet peak power requirements. For example, the initiation of a long trip from a cold start would tend to cause an exception to the general rule, because there is a need for the heat engine to be warm before it can be available to meet sudden power requirements, and because the trip might be too long to be made on battery power alone. In such circumstances, the heat engine, instead of the electric motor, would probably be operated first.

SECTION 3

VEHICLE LAYOUT

Having accomplished the analytical work of the trade-off studies, the next step is to transfer the results to a physical vehicle design and thereby consider many additional factors that are not amenable to resolution by analytical means alone. Some of these factors will undoubtedly influence the trade-off study results, thus initiating the design iteration process.

3.1 DESIGN CONCEPT

In any vehicle development program, there are two basic approaches: design the vehicle from the ground up, or modify an off-the-shelf item (a current production car). The ground-up design offers the potential for a vehicle more specifically tailored to the needs of the program, as well as a more unified (and perhaps better-integrated) product. On the other hand, much attention has to be given to the design and integration of "routine" components, and this causes higher program costs.

In any case, the highest priority goes to powertrain development, and it appears that this effort will suffer little, if any, compromise from a vehicle-modification approach. Moreover, such an approach will distribute program costs more in keeping with the program priorities. It was, therefore, a straight-forward decision to generate the NTHV design by modifying a base vehicle currently in production.

3.2 BASE VEHICLE SELECTION

A wide range of domestic and foreign production cars were examined for suitability as the base vehicle. The main objective was to find a lightweight car that would meet the JPL Minimum Requirements for passenger and luggage volume. The search for low weight essentially meant that the base vehicle would have to have been introduced recently, (because of the recent weight reduction programs for production vehicles) and would be equipped with front wheel drive. The candidate cars included the following:

Table 3-1. Candidate Vehicles; Capacities and Weights

Description	Volume[2] Passenger Compartment (m³)	Volume[2] Luggage (m³)	Curb Weight[3] (kg)
Audi 5000	2.55	0.42	1225
Chevrolet Citation	2.44	0.57	1117
Dodge Omni	2.29	0.48	983
Volkswagen Rabbit	2.26	0.42	833
Volkswagen Dasher	2.38	0.34	981

(Among the General Motors X-body cars, the Citation was selected because its hatchback configuration gives it more luggage volume.) The Audi 5000 was eliminated, due to its higher curb weight, and the Omni and Rabbit were eliminated because they are not five-passenger cars. Restructuring either of them into a five-passenger car would be a major undertaking. This narrowed the choice to the Dasher and the Citation. The Citation is heavier, but has a passenger and cargo volume which is larger than that of the Dasher. Therefore, the Citation was chosen as the base vehicle.

3.3 BATTERY PACK CONFIGURATION

The basic approach to configuring the battery pack was to study the volumes in the vehicle, and to use them in such a way as to avoid major changes in the vehicle's architecture. The major factors used in locating batteries were

1. Commercial battery case sizes that are available
2. Effect on vehicle architecture
3. Ease of service and access
4. Effect on vehicle dynamics
5. Effect on crashworthiness.

The necessary size for individual batteries was taken to be 41 cm high by 20 cm by 30 cm, which allows room for air circulation plus battery terminals, hold-downs, watering hardware, and the battery container cover. The potential locations for the batteries were

1. Across the front of the vehicle, forward of the radiator core support. This would require an extension of the vehicle front surfaces. Four to six batteries could fit, if oriented longitudinally.
2. Across the rear of the vehicle, partially recessed below the luggage compartment floor and partially occupying the spare tire well. Four to six batteries could be oriented longitudinally, but the spare tire would be eliminated.
3. Below and aft of the rear seat, but forward of the rear axle, between the rear suspension trailing arms. Three batteries could be oriented transversely, but would force a relocation of the gas tank, which is well positioned for crashworthiness.
4. Partially recessed below the luggage compartment floor, immediately aft of the rear axle. Three batteries could be oriented transversely end-to-end, or five batteries oriented longitudinally side-by-side. There would be substantial interference with the exhaust system.
5. In a longitudinal compartment above and aft of the rear wheels. Six batteries could be oriented transversely, with a compartment on each side of the car. There would be extreme interference with the exterior surfaces and appearance of the car, not to mention the rear seat room.
6. In a longitudinal compartment in each sill. Six batteries could be located longitudinally end-to-end. But severe interference with foot room, entry and exit would result.
7. In an enlarged center hump. Four batteries could be located longitudinally, end-to-end.
8. Immediately aft of the rear seat. Five batteries could be oriented longitudinally, side-by-side. A reduced luggage volume would result, unless compensated by elimination of the spare tire.

Of these, Location 3 was eliminated on the basis of probable interference with the rear seat. Locations 5 and 6 were eliminated for the reasons noted. From the remaining options, it was clear a number of batteries would go in the rear of the car, where there was room available and a number of options to choose from. The remaining question was whether Location 1, across the front of the vehicle, was superior to Location 7, in the center hump, or vice versa. Since Location 1 involved a reconfiguration of the front surfaces, a Buick Skylark body-in-white that had been procured especially for these studies was reconfigured with an extended nose. It was concluded that the appearance was pleasing and that the vehicle aerodynamics would be improved. (This is discussed below.) The resulting battery location would allow more batteries than with Location 7, and service and access would be superior. Therefore Location 1 was selected.

These considerations resulted in four specific configurations to be evaluated (although variations on these four are possible). These configurations, A through D, are shown in Figures 3-1 through 3-4.

The first three configurations have 12 batteries, and the fourth has 11 (compared to a recommendation of 14 batteries in the design trade-off studies). The reduction reflects the fact that the X-body vehicles are very efficiently packaged, with minimal extra room for batteries. It should be kept in mind, however, that while a 14-battery NTHV would produce better petroleum savings (as shown in the Trade-off Studies), a lower number of batteries would reduce the life cycle cost (and improve the net benefit).

Of the four specific configurations, A was eliminated because its polar moment of inertia was higher than the others. Configuration B was eliminated because it involved severe interference with rear seating. It would have been retained (being the only alternative without a lengthened nose) only if the forward battery location had been found unacceptable.

The remaining choice, between C and D, is largely dependent on the actual height of the batteries selected. Configuration D offers improved safety (from acid release in a rear crash), plus a reduced polar moment of inertia, but battery service and access is not as good. In Configuration C, both the front and rear battery containers could be designed to be removed from either the top or the bottom. However, Configuration C requires a low-profile battery in order to avoid significant reductions of luggage volume, whereas Configuration D could accommodate the tallest batteries with room to spare. All factors considered, Minicars has chosen Configuration C for its preliminary design. Configuration D will be held as a backup, if battery height becomes a problem.

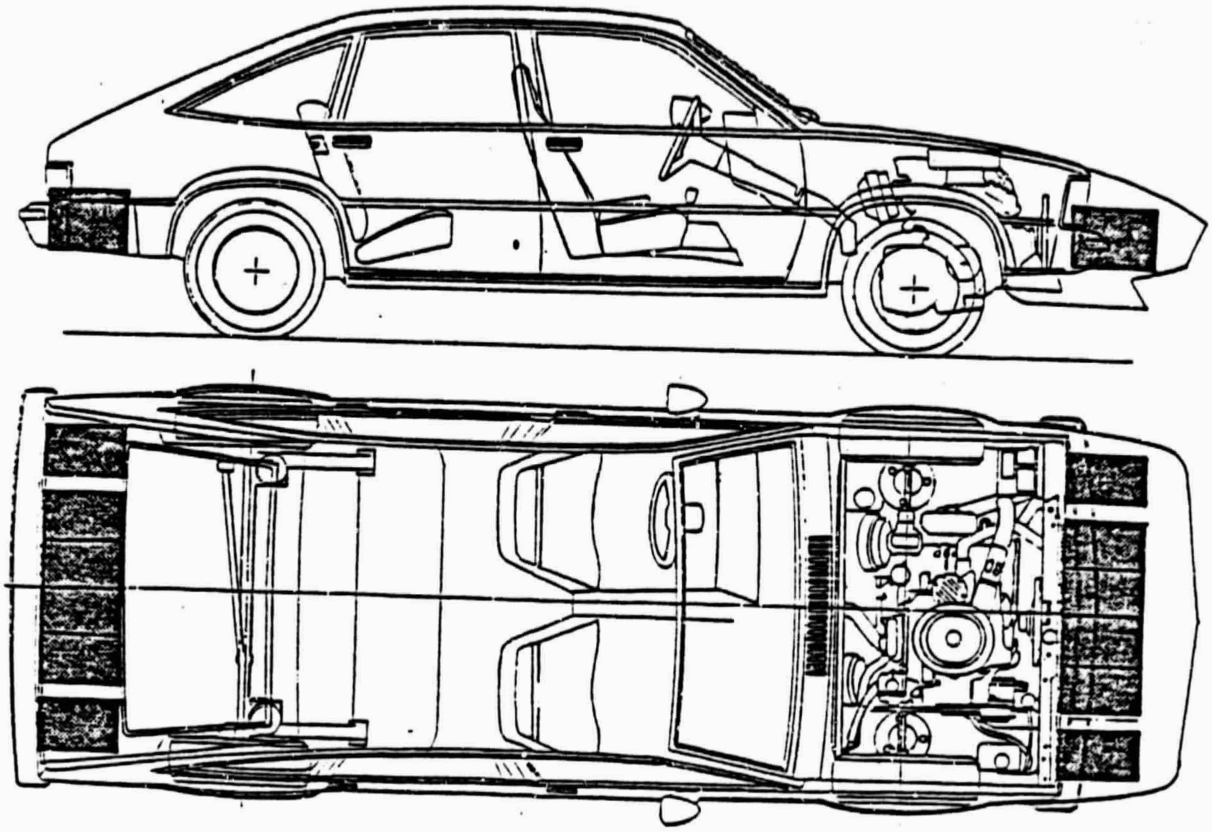

Figure 3-1. Configuration A.

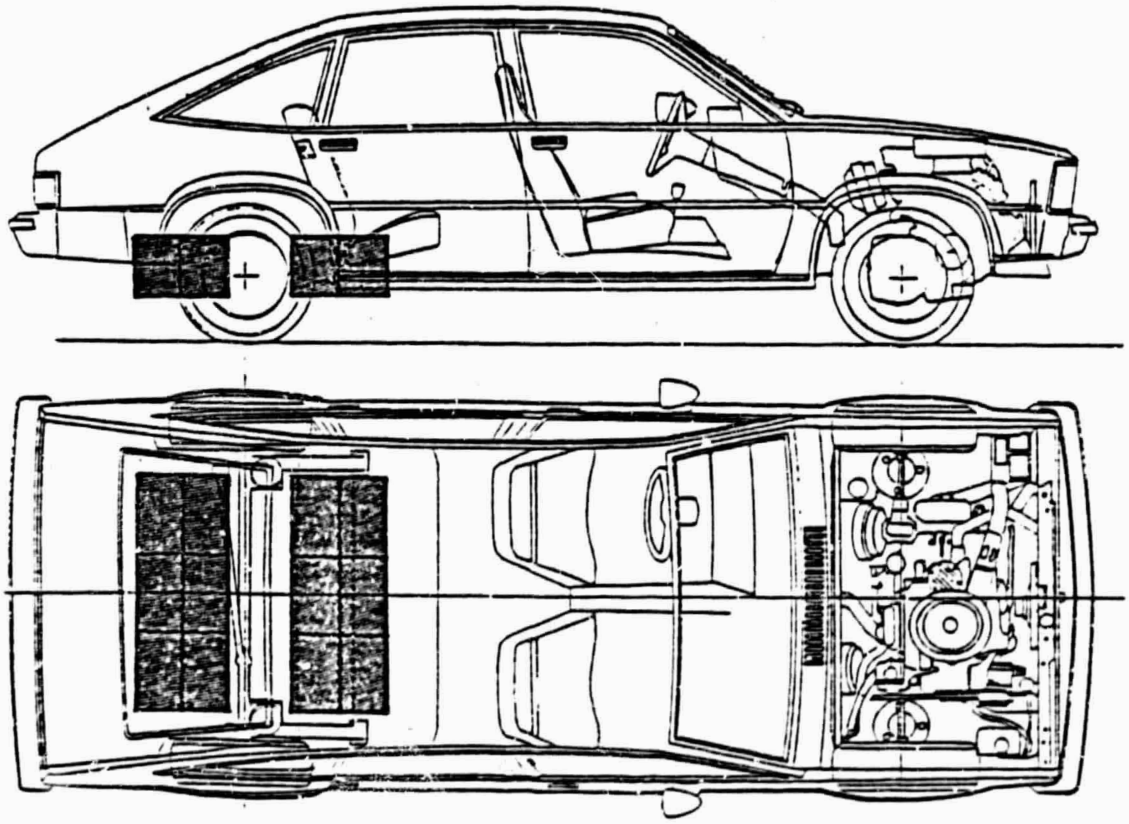

Figure 3-2. Configuration B.

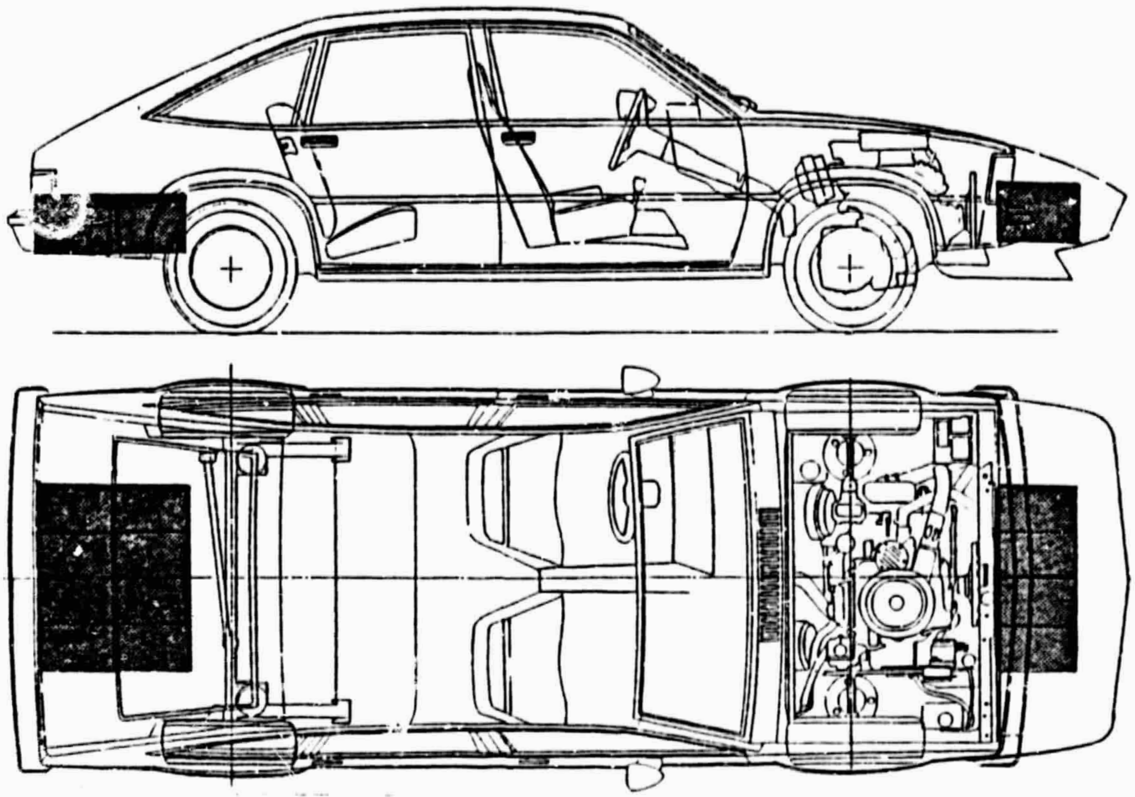

Figure 3-3. Configuration C.

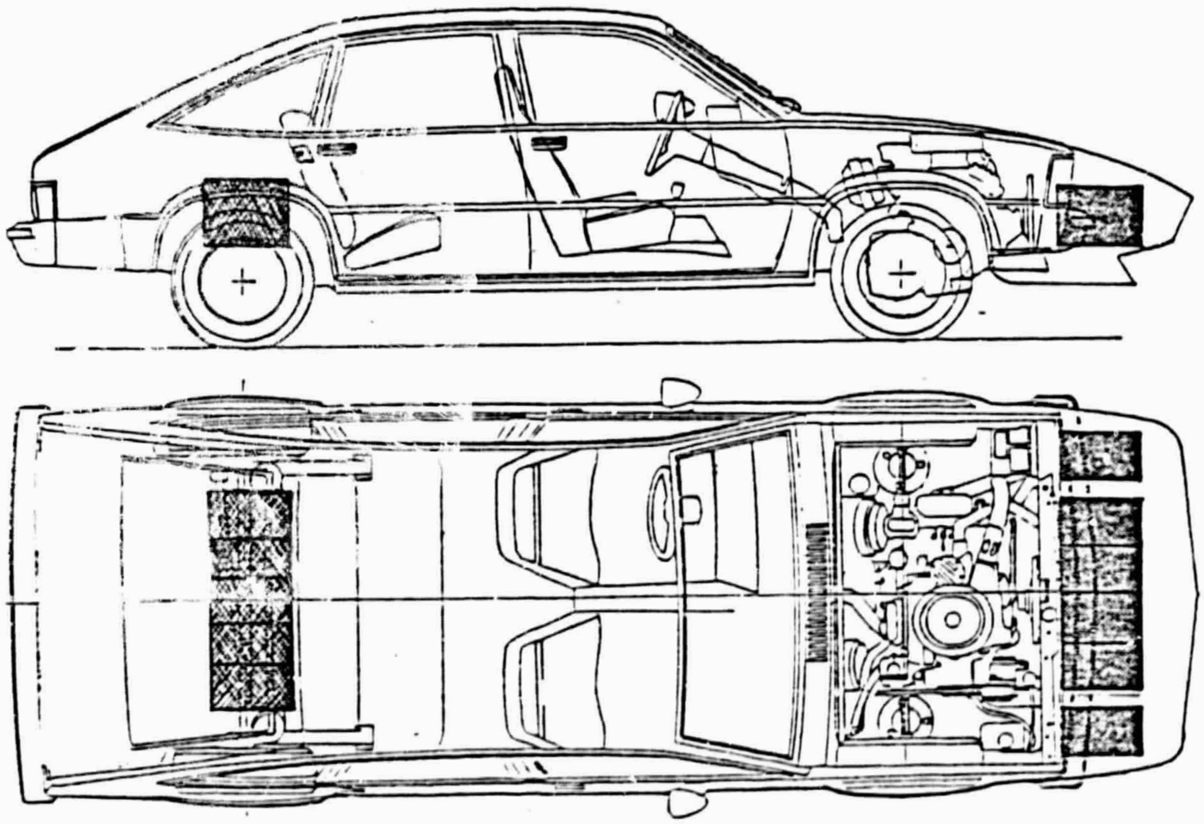

Figure 3-4. Configuration D.

3.4 VEHICLE SIZE AND WEIGHT

Based on available Chevrolet Citation data,[4] the significant NTHV dimensions (including the 46 cm [18 inch] extension of the nose) are as shown in Table 3-2 below.

Table 3-2. Significant NTHV Dimensions

Dimension	cm	in
Length	493	194
Width	173	68
Height	135	53
Ground Clearance	13.6	5.4
Wheelbase	266	105
Track Width, Front/Rear	149/145	59/57
Headroom, Front/Rear	97/95	38/37
Leg Room, Front/Rear	107/88	42/35
Shoulder Room, Front/Rear	143/142	56/56
Hip Room, Front/Rear	140/137	55/54

From the trade-off studies, the NTHV curb weight was established as follows:

Table 3-3. Comparison of Citation and NTHV Curb Weights

Description	kg	lb
Curb Weight of 4-door Citation with Options	1165	2568
Engine Change	- 40	- 87
Transmission Modifications	+ 26	+ 58
Electric Motor	+ 91	+ 200
Chain, Clutch, and Housing	+ 25	+ 55
Controller, On-board Charger, Wiring	+ 23	+ 50
Computer and Power Supply	+ 13	+ 30
Batteries (12 6-Volt Units)	+ 336	+ 740
Structural Modifications	+ 107	+ 236
NTHV Curb Weight	1746	3850

The total vehicle weight increase is 581 kg (1282 pounds); the batteries comprise 19 percent of the curb weight.

3.5 STRUCTURAL MODIFICATIONS

By far the most important factor in determining the appropriate structural modifications is the weight increase relative to the base vehicle. This determination is made according to two major considerations: crashworthiness and durability.

3.5.1 Crashworthiness

It is evident in the design of the X-body cars that General Motors has given very careful attention to the problem of frontal crashworthiness. As with all production vehicles, 30 mph frontal barrier crash tests have been performed by the National Highway Traffic Safety Administration (NHTSA) for compliance with Federal Motor Vehicle Safety Standards (FMVSS) 204, 212, 219, and 301. In addition, the NHTSA has sponsored frontal barrier crash tests of the Citation at 35, 40 and 48 mph (the last test being for comparison with the Minicars Research Safety Vehicle [RSV]). The 48 mph barrier test is well beyond the design capability of the Citation, but in the other tests the vehicle performed quite well. In fact, the 35 mph crash pulse (acceleration vs. time) shown in Figure 3-5 is one of the best we have ever seen.

It is obvious that the addition of 581 kg to the vehicle mass will have a most deleterious effect on crashworthiness unless the structural design is modified appropriately. Our general procedure is to assess the amount of crush space available (allowing for non-crushable elements like the engine), establish a force vs. crush characteristic that will generate controlled collapse of the available space at sufficient force levels to absorb the crash energy, and then provide for the transmittal of those forces through or around the passenger compartment (so as to maintain compartment integrity). These steps are accompanied by careful consideration of the vehicle architecture, computer simulations, static and dynamic crush tests of elements and whole structures, and finally, by vehicle crash tests. All of these steps will have to be applied in the conversion of a Citation into an NTHV.

As discussed further in Section 3.6, our intent is to provide passive protection (which will be required for cars like the NTHV by 1983, according to FMVSS 208) to at least the 30 mph level required by the standard, and perhaps to the level offered by the active three-point belts in the Citation. It is doubtful whether the modified structure will be able to provide as good a crash pulse at 35 mph as the Citation currently has. On the other hand, the crash pulse could be degraded substantially without significant adverse effects on the performance of the air cushion restraints planned for the NTHV. For a number of years Minicars has been developing both structures and restraints that provide passive protection well within the criteria of FMVSS 208 at speeds to 50 mph. This experience indicates that we should be able to

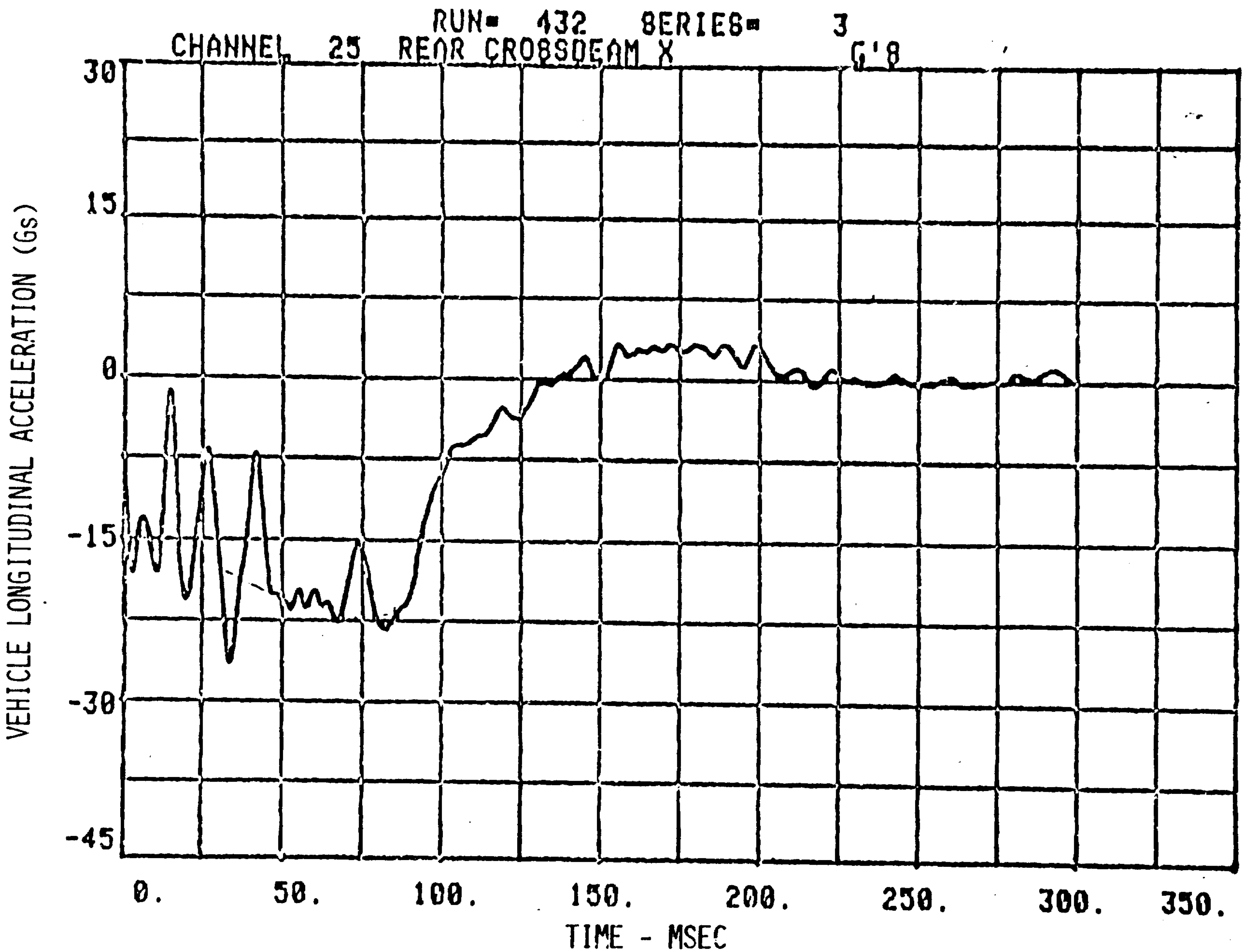

Figure 3-5. Crash Pulse for 1980 Chevrolet Citation:
35 mph Frontal Barrier Impact

match the occupant protection of the Citation (which barely fails to meet the FMVSS 208 criteria at 40 mph). It also suggests that we can estimate quite well what the structural modifications will be. Adjustments of the crush characteristics for any particular structural element can be provided by changing the metal gauge, by converting an open section to a closed section, or by filling closed sections with foam materials of various densities. In either case, there is little or no effect on the vehicle architecture. The preliminary design of the structural modifications to the body-in-white is illustrated in Figures 3-6 and 3-7.

For side impact protection, none of the current FMVSS is applicable per se. FMVSS 214 comes the closest to addressing this problem, but it only requires a specified strength when the door is subjected to a static test with a pole-like indenter. Recognizing this situation, the NHTSA is working to develop a system standard that will involve a side impact by a deformable moving barrier. As a result, we are not planning to address FMVSS 214 explicitly, but rather to design for the crash test condition. (The resulting door will probably meet the FMVSS 214 requirements in any case.) As a result, the preliminary design includes foam-filling the lower doors, which will add to their longitudinal compressive strength in frontal impacts, as well as improve their intrusion resistance in side impacts. The improved intrusion resistance is due to: (a) better interlocking of the door with the shut faces, which keeps the door from being pushed through the opening, (b) the fact that the inner and outer door panels are forced to maintain their separation, and (c) very large increases in the door bending rigidity, due to item (b) and to the foam itself. Loads generated by the doors will be carried into the sills (also foam filled) and the A, B, and C posts; these elements will receive improved lateral support from a lateral reinforcement under the front seats and an additional lateral element between the rear spring pockets. The detail design of the Citation doors will be studied, to ascertain whether the stock door beams should be removed. (Generally, they are present only to satisfy FMVSS 214.) If so, they may be replaced with a lighter weight longitudinal element that is more efficient in compression.

Improvements in damageability, with reduced weight, have been demonstrated in the Minicars RSV and LRSV (Large Research Safety Vehicle). The latter is a modified Chevrolet Impala that has demonstrated 64 km/hr passive protection with air cushion restraints, a 390 kg weight reduction (relative to the downsized Impala), and a fuel economy of 11.7 kilometers per liter

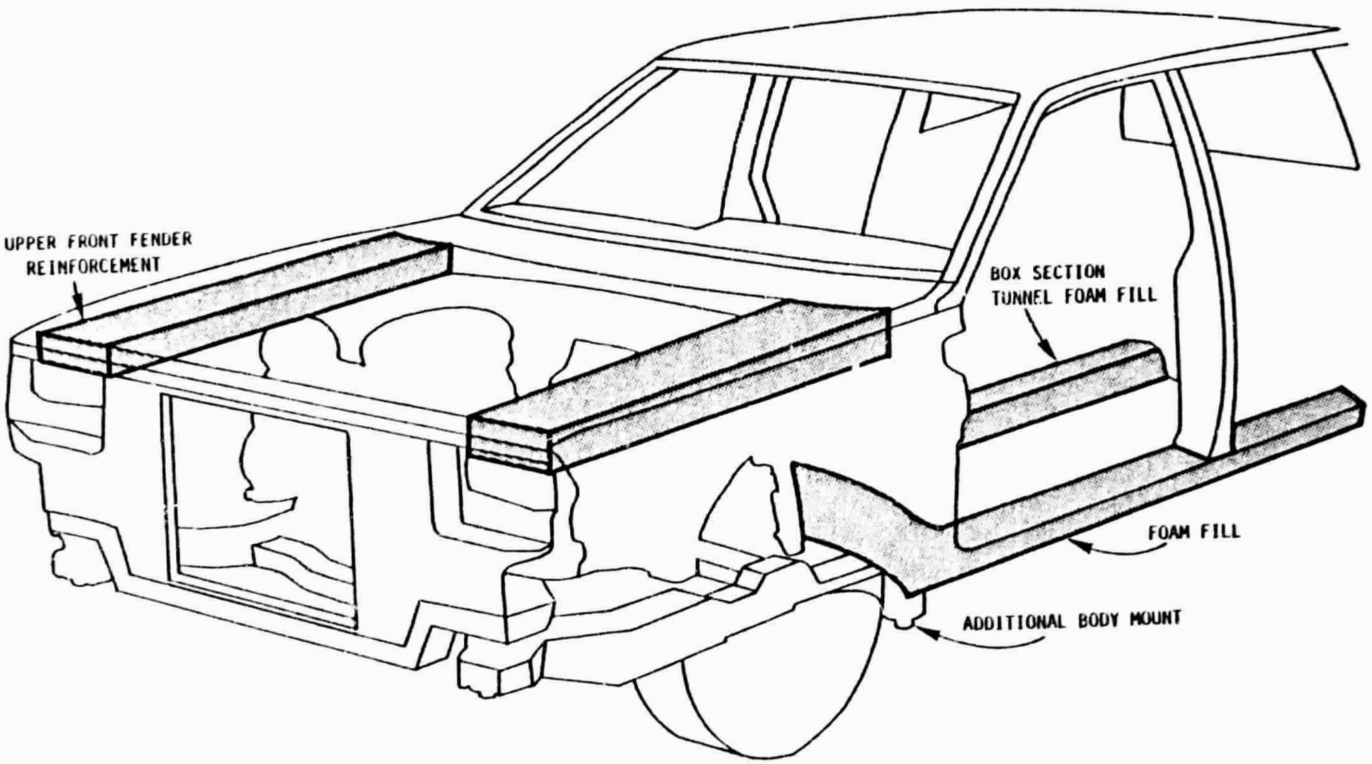

Figure 3-6. Body Structure Reinforcements

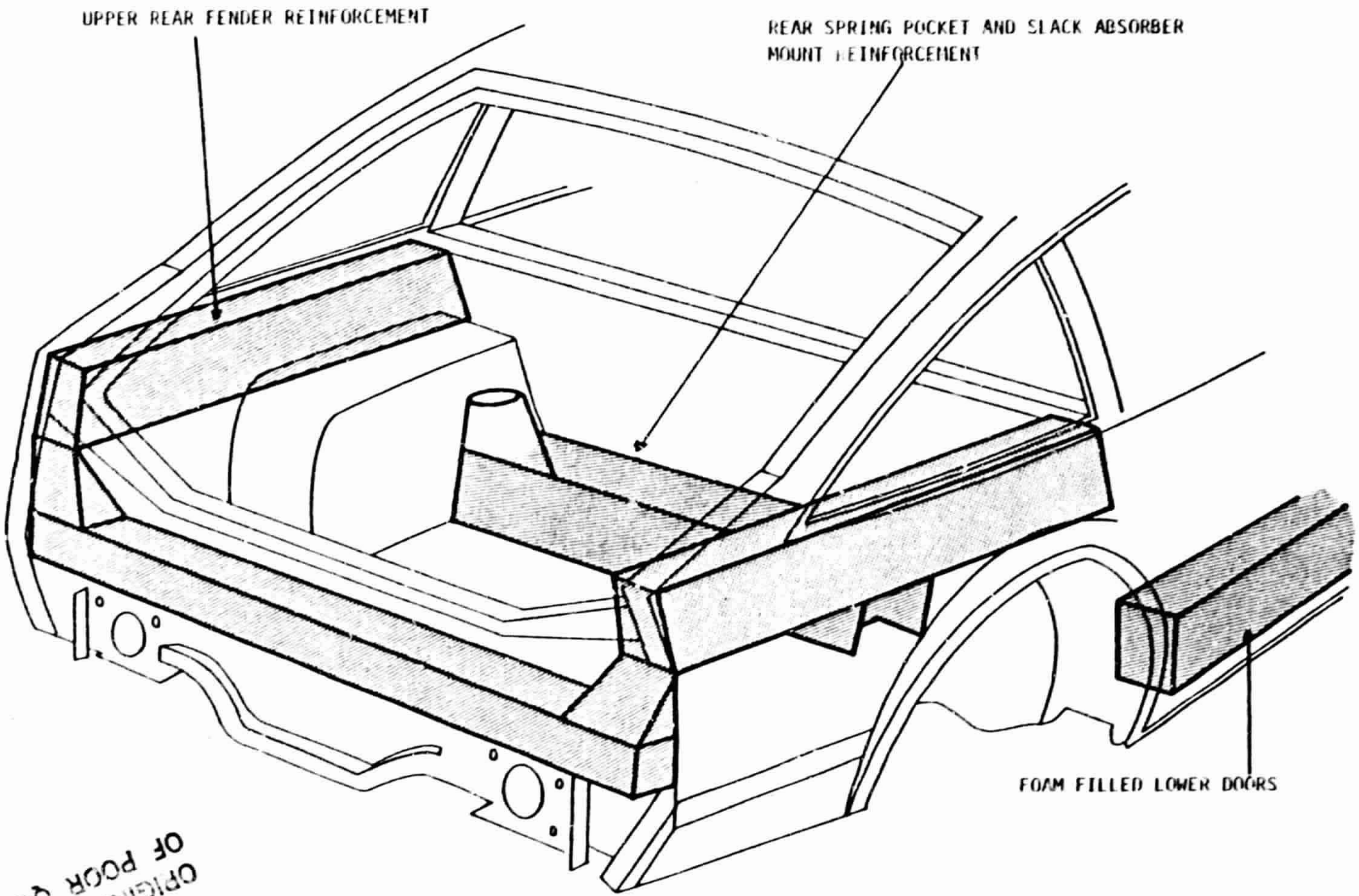

Figure 3-7. Body Structure Reinforcements - Rear

combined. As a result of this experience, we expect the NTHV to meet all the requirements of FMVSS 215, and to prevent all but minor cosmetic damage to the front and rear surfaces in barrier impacts at speeds to 16 km/hr. This will be accomplished by using flexible front and rear fascias in combination with rubberized fabric (or "rubric") elements developed by the Bailey Division of the United Shoe Machinery Corporation.

Because of the increased weight of the NTHV relative to the Citation, there will be higher force requirements for roof crush strength under FMVSS 216. This should entail only minor structural modifications. Satisfaction of FMVSS 301, Fuel System Integrity, may be jeopardized by the aft batteries, at least in Configuration C. This could be handled by structural modifications, but to keep these (and the weight) to a minimum, the preliminary design includes a fuel cell made of the same flexible material as in the RSV. This replaces the Citation gas tank. A planned rear impact test may show that no structural modifications are necessary with the use of such a fuel cell.

As a final note on crashworthiness, an inertia switch will be used to disconnect the electrical systems in the event of a crash.

3.5.2 Durability

The term "durability" refers to the ability of a vehicle to be operated in various driving conditions, with various payloads, over a reasonable lifetime, and with component failures held to a reasonable level. The General Motors X-body cars probably have a design curb weight of about 1270 kg (2800 poinds), which is 476 kg (1050 pounds) less than the NTHV curb weight. The obvious conclusion: some of the stock Citation parts may well be overstressed. To establish appropriate design conditions, we note that the JPL minimum requirement R3 specifies a payload capacity of 520 kg (1147 pounds). This corresponds to two 95th percentile males (98 kg [215 pounds] each) in the front seat, three 50th percentile males (75 kg [165 pounds] each) in the back seat, plus 101 kg (222 pounds) in the luggage compartment. Thus the Gross Vehicle Weight Rating (GVWR) would be 1746 + 520, or 2266 kg (4997 pounds).

The static load distribution of the NTHV for several loading conditions is shown in Table 3-4.

Table 3-4. Static Loads for Selected NTHV Payloads

Payload	Total Weight (kg)	Total Weight (pounds)	Front Load (kg)	Front Load (pounds)	Rear Load (kg)	Rear Load (pounds)
Curb Weight	1746	3850	1036	2283	711	1567
95th Percentile Driver Only 98 kg (215 pounds)	1844	4065	1093	2409	751	1656
2 95th Percentile Front Passengers 195 kg (430 pounds)	1941	4280	1150	2535	792	1745
Full Load 5 Passengers + Luggage 2266 kg (4997 pounds)	2266	4997	1199	2643	1068	2354

These numbers served as the basis of estimating the necessary vehicle changes. Detail design must be based, of course, on an analysis of the dynamic loads transmitted through the suspension to the vehicle structure.

3.5.2.1 Probable Engine/Motor Cradle Reinforcement

Due to the additional vehicle weight and to the additional components (such as the motor) in the engine compartment, we expect that the engine cradle will have to be reinforced. The additional power train weight will also require that the attachment to the body at the rear of the cradle be strengthened, probably by adding a body mount.

3.5.2.2 Front Suspension

The NTHV front suspension is shown in Figures 3-8 and 3-9. The primary components are the McPherson struts, lower A-arms, coil springs, shock absorbers, and anti-roll bar. Due to the higher loads on the front suspension, we expect to substitute stiffer springs and shock absorbers to maintain vehicle ride height and comfort. The lower A-arms will be modified and reinforced to handle the higher front suspension loads, and a new anti-roll bar will be selected to match the NTHV cornering forces. The reinforcements are shown in Figure 3-10.

3.5.2.3 Rear Suspension

The NTHV, like the X-body cars, does not have a drive function for the rear axle. Hence, the simple rear suspension configuration of the baseline vehicle is employed, with modifications, in the NTHV. The modified suspension is shown in Figure 3-11. The rear wheels are connected by a beam with an inverted-U cross section. At each side, a trailing arm and spring pocket are welded to the beam. A transverse track bar (panhard rod), coil springs, and shock absorbers complete the suspension. An anti-roll bar is welded in place inside the U-shaped beam; these together provide the anti-roll stiffness. It is expected that stiffer springs and shock absorbers will have to be substituted to maintain ride height and comfort. An axle bending reinforcement is also anticipated, and a substitute anti-roll bar may be required. Finally, the axial load capacity of the panhard rod and its attachment will be evaluated, and increased if necessary.

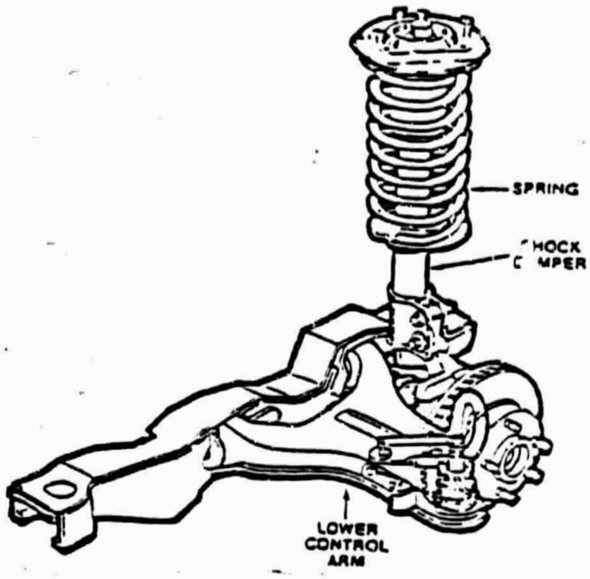

Figure 3-8. Front Suspension Arrangement

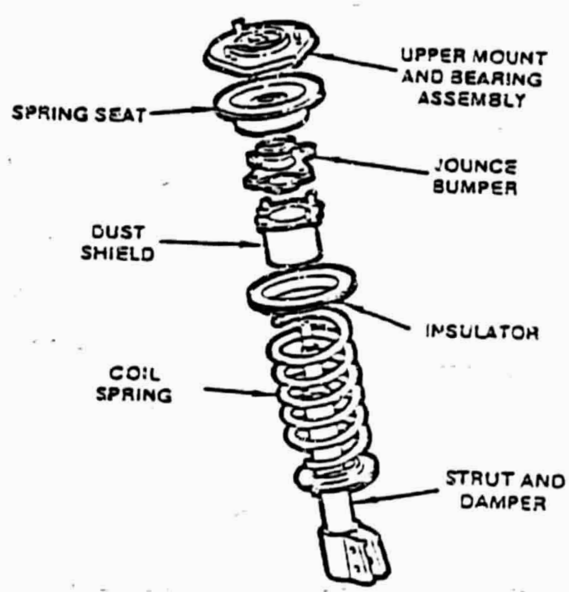

Figure 3-9. Front Suspension Strut Details

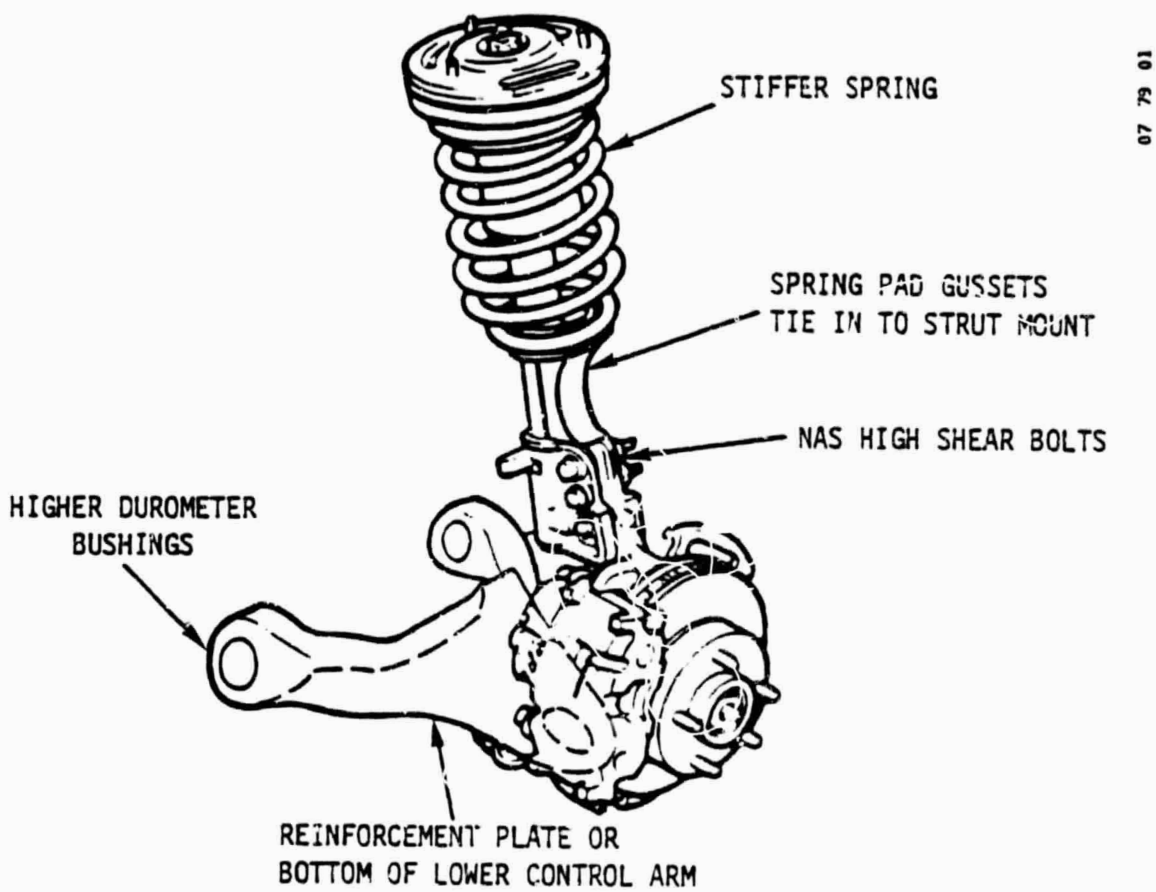

Figure 3-10. Front Suspension Reinforcement

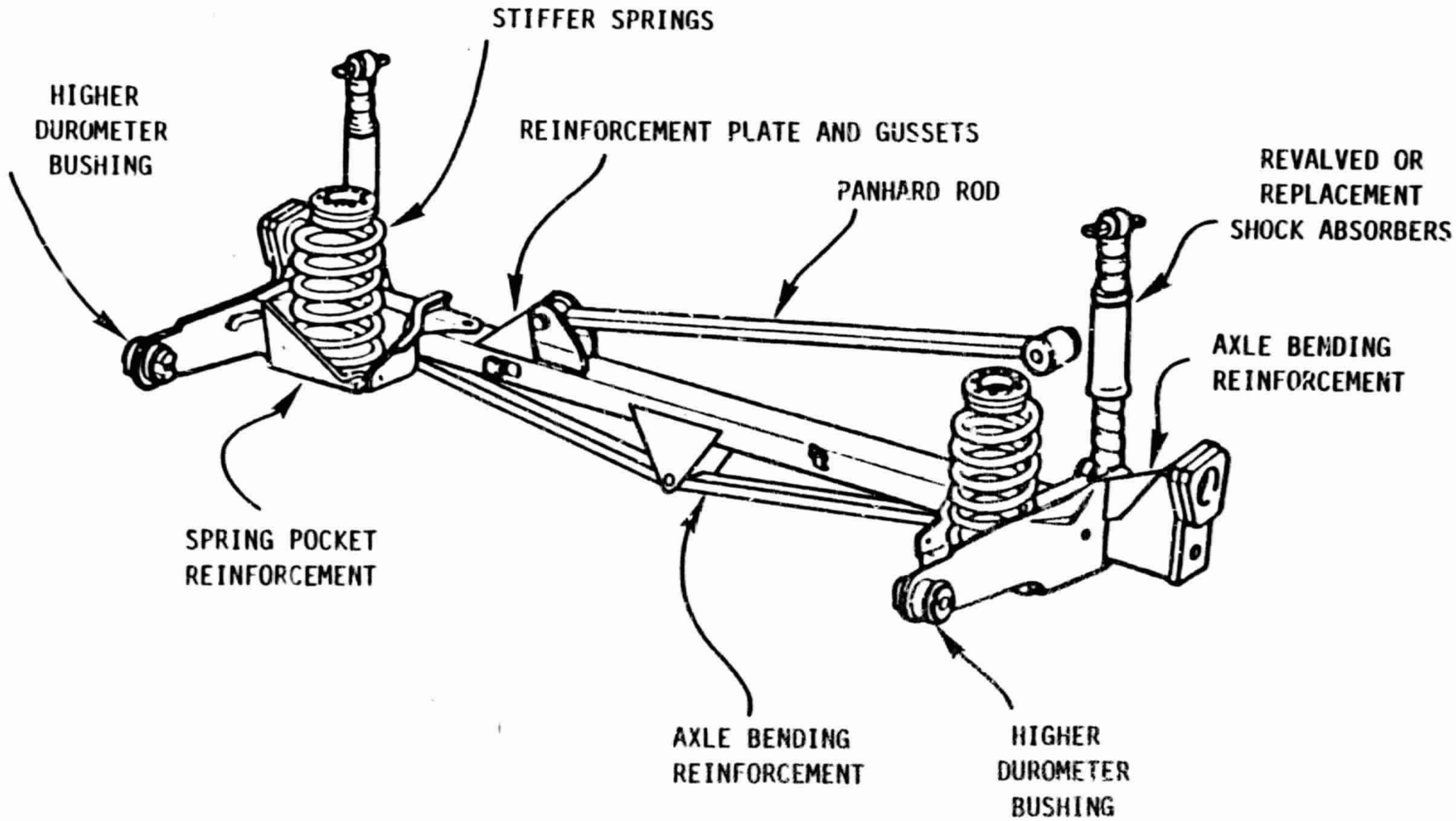

Figure 3-11. Rear Suspension Reinforcements

3.5.2.4 Battery Support Structure Design - Front

The front battery support structure will be designed to distribute the loads imposed by the battery weight. The loads work through three load paths on each side of the front structure: the engine cradle, the mid-rail and the upper fender reinforcement.

3.5.2.5 Battery Support Structure Design - Rear

The rear battery support structure will load primarily into two longitudinal rails beneath the trunk floor. Additional reinforcement will be provided by crossmembers added between these rails at the front and rear of the trunk. The rear crossmember will tie in to the upper rear fender reinforcements, to triangulate the rear structure.

3.6 OCCUPANT CRASH PROTECTION

To be a socially responsible vehicle, the NTHV needs to possess a degree of occupant protection equal to or greater than those vehicles which it is replacing. From the schedule of NTHV production and the NHTSA's near-term rulemaking plan, it is clear that complying with the 1978 safety standards will be insufficient. The NTHV will be replacing vehicles produced in the early and mid-1980's. These vehicles will be built to comply with two safety standards—one dealing primarily with frontal impact protection, the other with side impact protection—that will have significant effects on the compartment configuration.

By 1983, FMVSS 208 will require vehicles of the size of the NTHV to possess so-called "passive" occupant frontal impact protection. This protection is to be confirmed in a 30 mph fixed flat barrier impact with dummies representing 50th percentile males at all designated front seat positions. The automobile industry is currently reacting to FMVSS 208 requirements by developing either air cushion restraint systems or passive belts for their vehicles—the choice of system being dictated by marketing factors, vehicle particulars, and corporate research resources. In the case of the X-body, GM seems to be preparing to introduce passive belts into the vehicle at least in the early years of the standard. It is clear from an inspection of the vehicle compartment layouts, however, that considerable thought has been exercised to ensure that air cushions could be easily adapted to this vehicle.

For side impacts, NHTSA is currently formulating a revision to the present standard (FMVSS 214); the revision will require a significant upgrading of lateral protection requirements. The compliance test will almost certainly involve the impact of a moving, deformable barrier (a bogey representing the most probable or average striking vehicle) against the door of the subject vehicle. A special lateral impact dummy, developed by NHTSA for use in this test, will be located on the near-side front seat. This standard will almost certainly require that future vehicles be equipped with protective padding on the interior of the door.

3.6.1 Selection of Passive Restraints for the NTHV

The selection of the appropriate NTHV passive restraint system (air bags or passive belts) was based on the following considerations:

1. Preference should be given to the most effective restraint system.
2. Consumer acceptance should be seriously weighed.

The selection of the most effective system is highly dependent upon the assumed usage for passive belts. The VWRA data* indicate that, at usage rates of 60 to 80 percent, passive belts are more effective than air cushions. What the actual passive belt usage will be is not clear, but the best estimates are between 40 and 50 percent. At this rate, air cushion systems could quite likely be more effective than passive belts.

Marketing studies of passive restraints are currently being conducted within the automobile industry, but their results will not be available for several months. Based on the data accumulated to date, however, it seems clear to us that air cushion systems offer a distinct marketing advantage. It is important here to realize that usage rates of 40 to 50 percent indicate that a significant percentage of those consumers, who voluntarily have purchased passive belts as optional equipment, have later objected to them sufficiently to not use them (and, indeed, a significant percentage have removed the system from the car). It can be

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*VWRA is the trade name for VW's passive belt system, which is currently being offered as optional equipment on Rabbits. VW is monitoring the effectiveness of this system.

assumed that others object to the system, but not so violently as to disconnect or remove it. On the other hand, the few who have purchased air cushion vehicles have become avid supporters of the system. Disconnecting air cushion systems is virtually unknown. In summary, it appears that the "invisibleness" of the air cushion system outweighs the cost penalty (estimated at about $150 per vehicle). Air cushions appear to be the better choice for the NTHV.

3.6.2 <u>Description of the NTHV Occupant Protection System</u>

The major items of the preliminary design occupant protection system are 1) the driver air cushion system, 2) the passenger air cushion system, and 3) the door interior padding. Figures 3-12 and 3-13 illustrate the existing Chevrolet Citation dashboard, instrument panel and steering wheel layout, and the Citation layout after the integration of the air cushion systems. At the driver station the only visible features of the system are the special Air Cushion Restraint System (ACRS) wheel, with its hub storage volume for the inflator, bag, and cover, and the lower dash area which is subtly altered to function as a lower body (knee) restraint. At the passenger position the torso cushion module pan cover can be seen in the upper dash area, and the lower dash is reworked to form a knee restraint. The details of these systems and the NTHV interior door padding are discussed below.

3.6.2.1 Driver Air Cushion System

The driver air cushion system should be fairly typical of the systems currently being developed within the automobile industry to comply with the passive restraint standard. Much of the same hardware that either has been used in the recent past or will be used in the near future can be employed to configure the system.

The typical driver restraint system is comprised of a knee restraint and an upper body restraint. The knee restraint controls the driver's impact trajectory and absorbs the lower body kinetic energy. The upper body restraint is comprised of a driver cushion module, steering wheel, and energy absorbing column.

The NTHV driver knee restraint can be made similar in construction and characteristics to those knee restraints currently used in production air cushion and passive belt vehicles. Figures 3-14 and 3-15 illustrate, by way of example, the VW passive belt knee

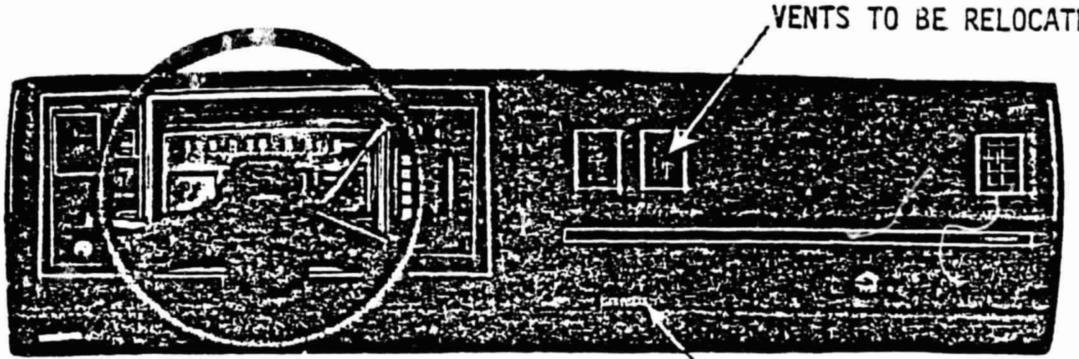


Figure 3-12. Existing Citation Dashboard

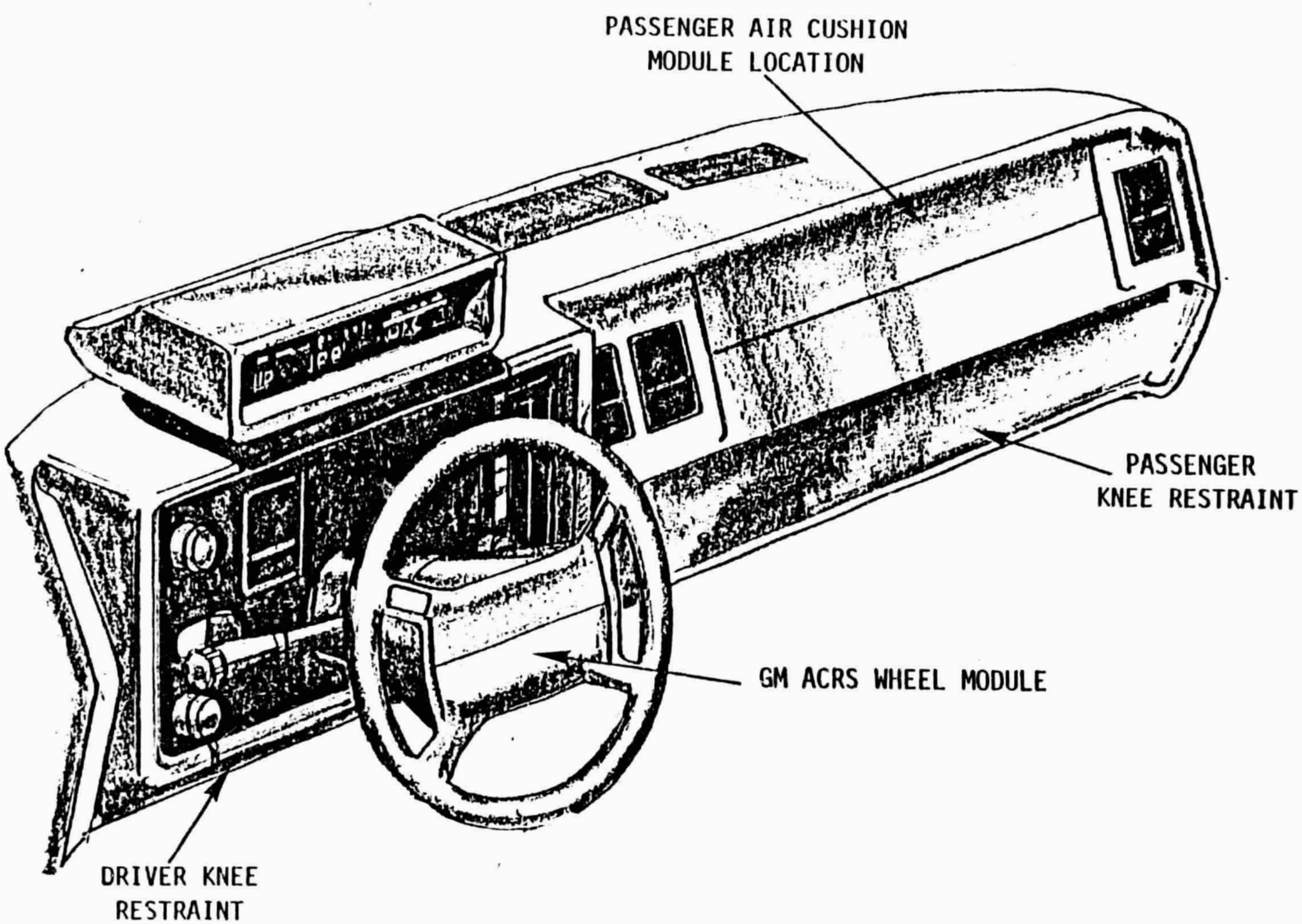


Figure 3-13. ACRS Citation Dashboard

restraint system. It is essentially a lateral, crushable beam that spans from the A-pillar to the tunnel in front of the driver's knees. The beam is constructed of light-gauge sheet metal, spot-welded into a closed section (see Figure 3-15). The outer surface is coated with a decoratively-skinned closed-cell polyurethane foam.

The knee restraint produces proper force-deflection characteristics in three ways. First, the driver's knees depress the foam coating, distributing the forces and producing the desired force onset rate. Second, the sheet metal beam is crushed, further absorbing energy. Third, the brackets supporting the knee restraint beam yield, supplying additional energy absorption and allowing the knees to further penetrate the lower dash area. The NTHV uses a knee restraint system of this basic design, integrated into the X-body dash contouring.

In the driver upper body energy management subsystem the Citation steering wheel is replaced with the GM ACRS wheel and module elements shown in Figure 3-16. These items can be readily integrated into the Citation driver station, and are fully production engineered. The only contemplated changes to these elements are 1) an inflator substitution and 2) the introduction of bag venting to the driver air cushion. These particular modifications have been employed in the past, so, no problems should arise that have not already been addressed and solved. An inflator of essentially the same exterior design and mounting hardware as the GM inflator can be set to perform optimally in the NTHV. The steering column mast will be replaced by a new GM ACRS column mast. The ACRS mast is almost identical to the stock Citation mast — the crucial difference being that the ACRS contains the slip-ring assembly needed to transmit the firing signal to the wheel module (Figure 3-17). In the stock Citation energy-absorbing column, an array of small steel balls set in a jacket between the telescoping tubes provides some energy-absorption by creating a 600-pound column collapse force. Previous studies conducted by Minicars have found that this energy absorption mechanism is not satisfactory for air cushion use.[5] Consequently, the ball jacket would be replaced with an energy-absorbing telescoping steering shaft assembly. This tube/mandrel steering shaft arrangement has been fully engineered and found to provide both superior energy absorption and linkage to the steering gear box. It has been successfully used on a number of operational air cushion vehicles built at Minicars (for example, the Minicars RSV) and thus its adaption to this vehicle would represent essentially no

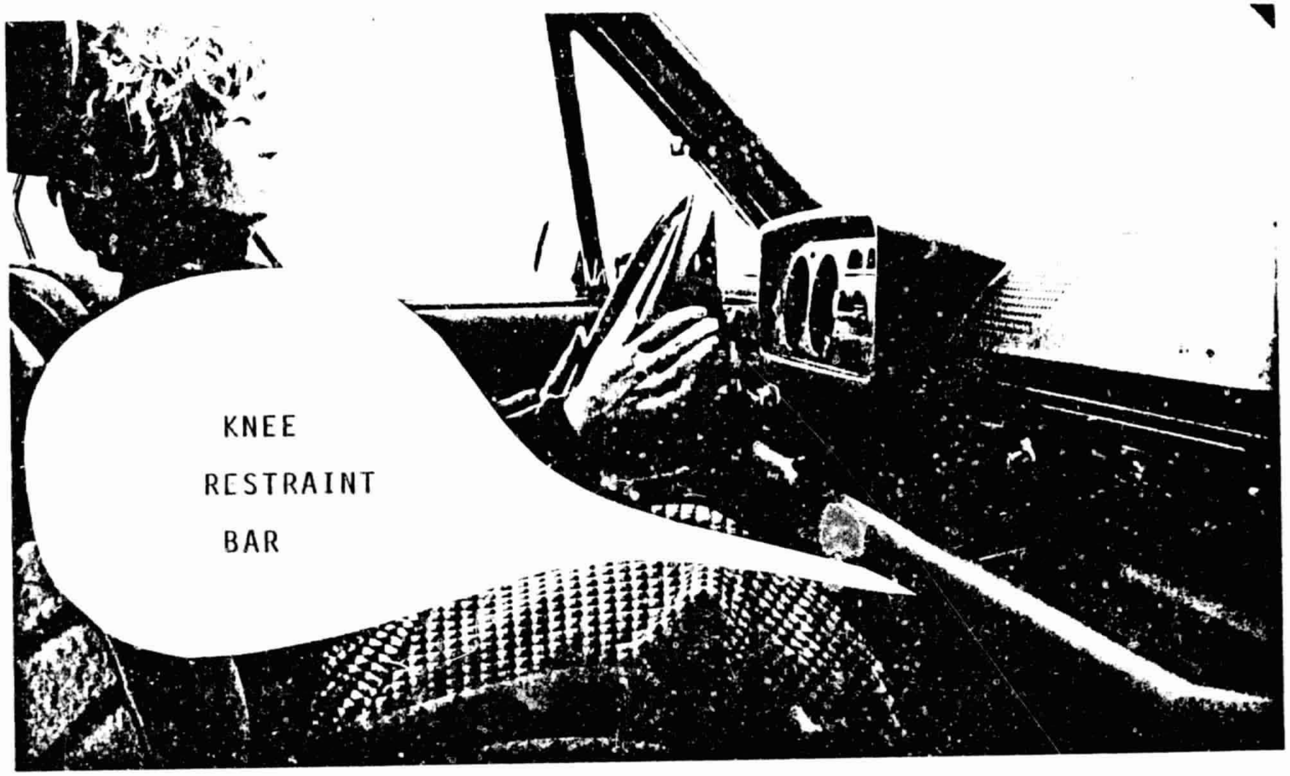


Figure 3-14. VW Passive Belt Knee Restraint Bar

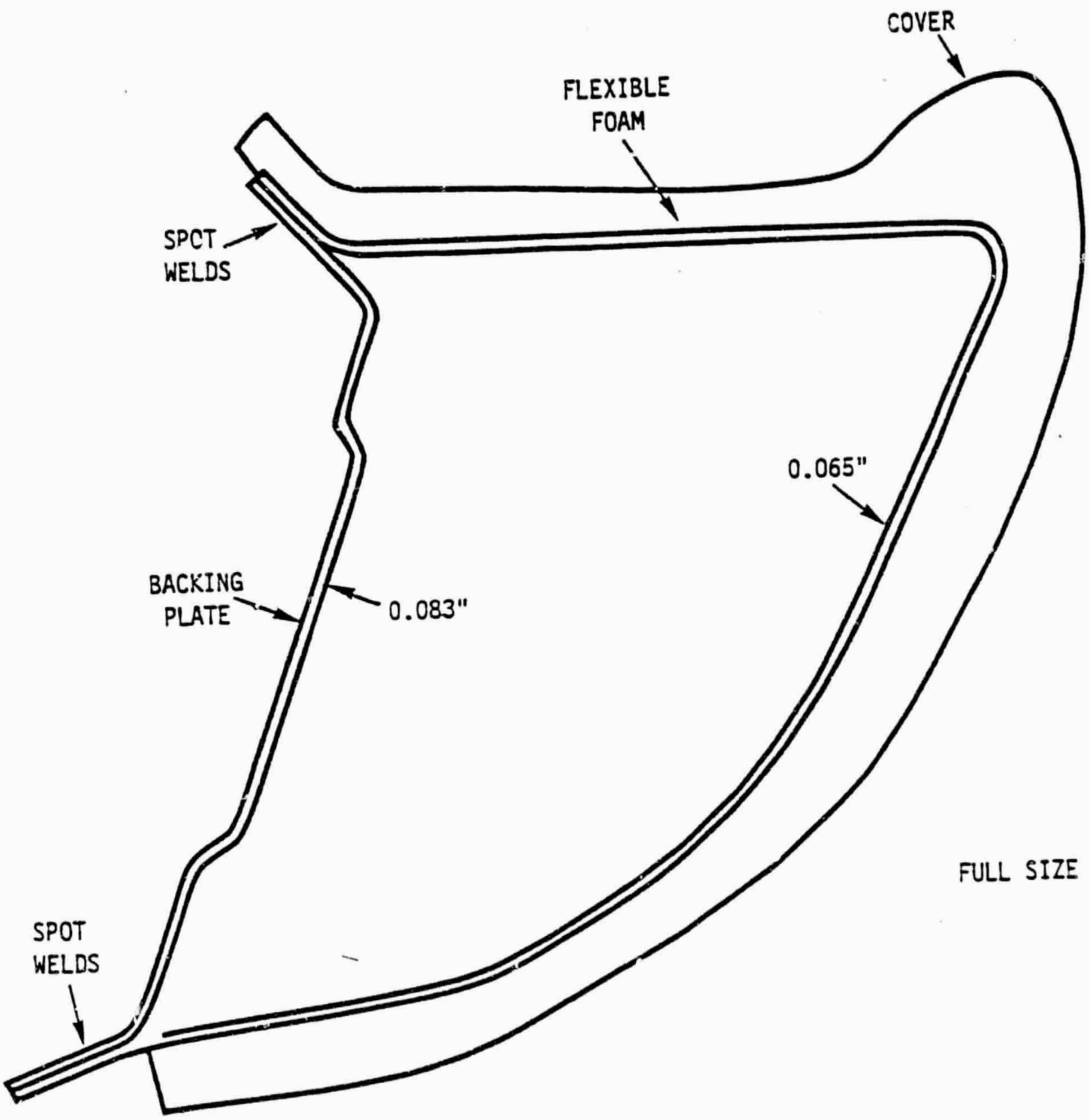
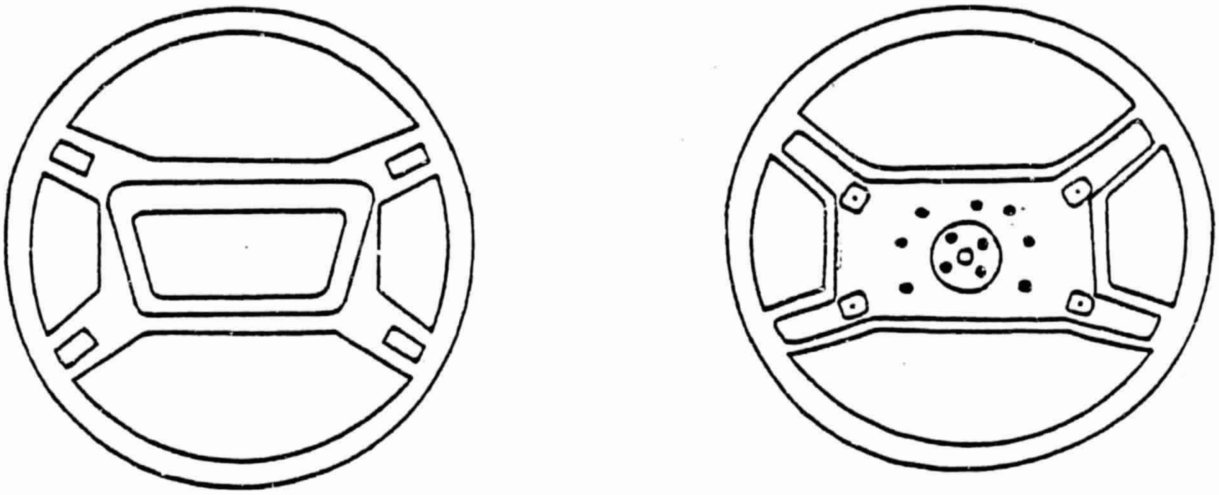
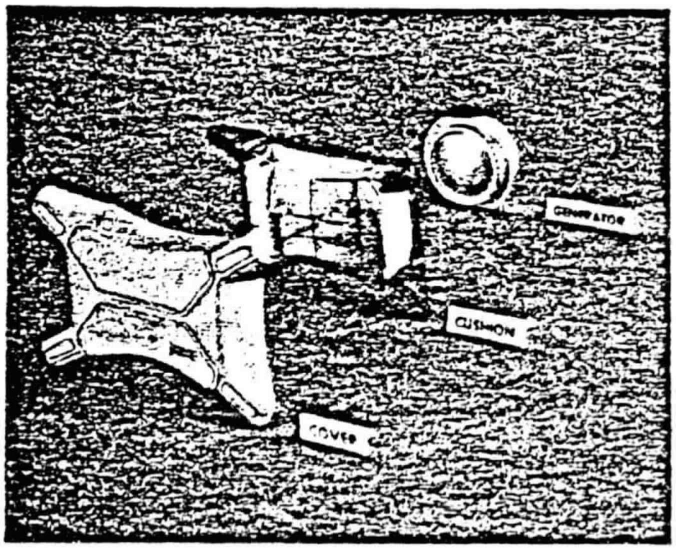


Figure 3-15. VW Passive Belt Knee Restraint Bar Cross Section



(a) GM ACRS Steering Wheel



(b) Driver Module

Figure 3-16. GM ACRS Steering Wheel
and Driver Module

Figure 3-17. GM ACRS Column Mast and Slip Rate Assembly

technical risk, yet proven technical advantages. Figure 3-18 shows the tube/mandrel steering shaft assembly, complete with key/lock mechanism for high torque transmission capability.

3.6.2.2 Passenger Restraint System

At the passenger position a so-called "high-mount" air cushion system will be layed out, using a ganged driver inflator gas generation system. The term "high-mount" means that the system uses a mechanical knee restraint, essentially the same as that previously described for the driver, to absorb lower body kinetic energy. A high mount system was chosen for two basic reasons: first, the modifications to the existing right-side dashboard are minimized; second, most air cushion systems being currently developed for vehicles in the Citation's class are of this basic configuration.

Upper body kinetic energy absorption is the sole function of the passenger air cushion module. This module is typical of those currently being developed for FMVSS 208 compliance (Figure 3-19) and consists of a module pan (a sheet metal box about 12 inches by 8 inches by 8 inches deep), a gas generator subsystem, a bag, and a cover. The gas generator subsystem is comprised of two (i.e., "ganged") production-engineered driver inflators bolted to the rear surface of the module pan. The cushion is expected to have a volume of about 180 liters. Minicars will design a decorative cover to integrate with the present Citation dashboard design.

3.6.2.3 Door Interior Padding

Providing lateral impact protection involves properly padding the door interior and, for severe accidents, providing door structure to more positively link the door to the rest of the vehicle compartment. For purposes of complying with the future side impact compliance test, structural reinforcements may not be necessary.

Minicars has had extensive experience in providing operational vehicles with lateral impact crash survivability at energy levels significantly beyond those required here. In both the RSV and LRSV programs, crashworthy door development efforts were required to produce consumer-acceptable designs possessing a high level of survivability. Figures 3-20 and 3-21 are pictures of the RSV and LRSV door interiors, respectively. The extent of the NTHV door

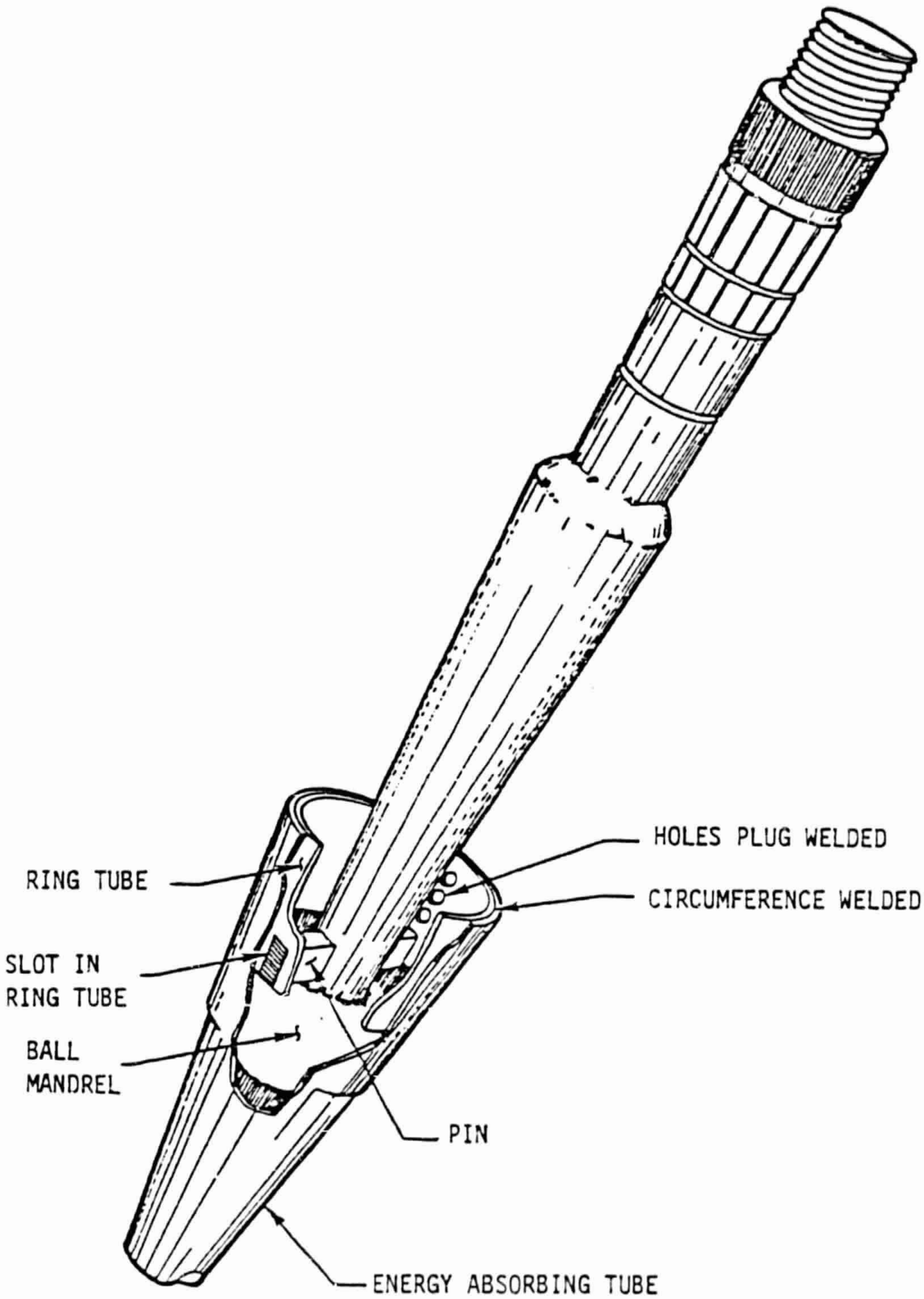


Figure 3-18. Tube/Mandrel Steering Shaft Assembly

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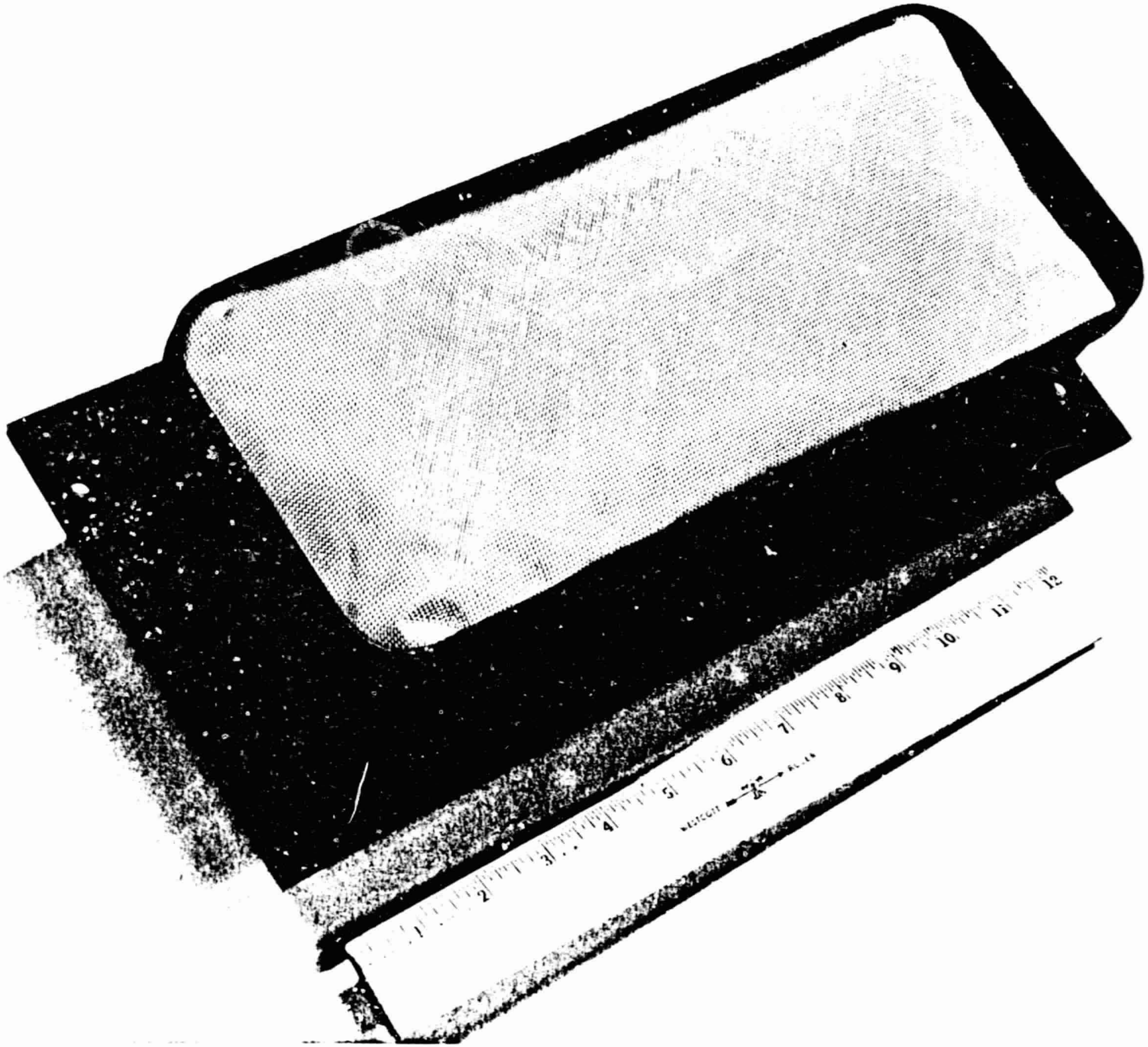
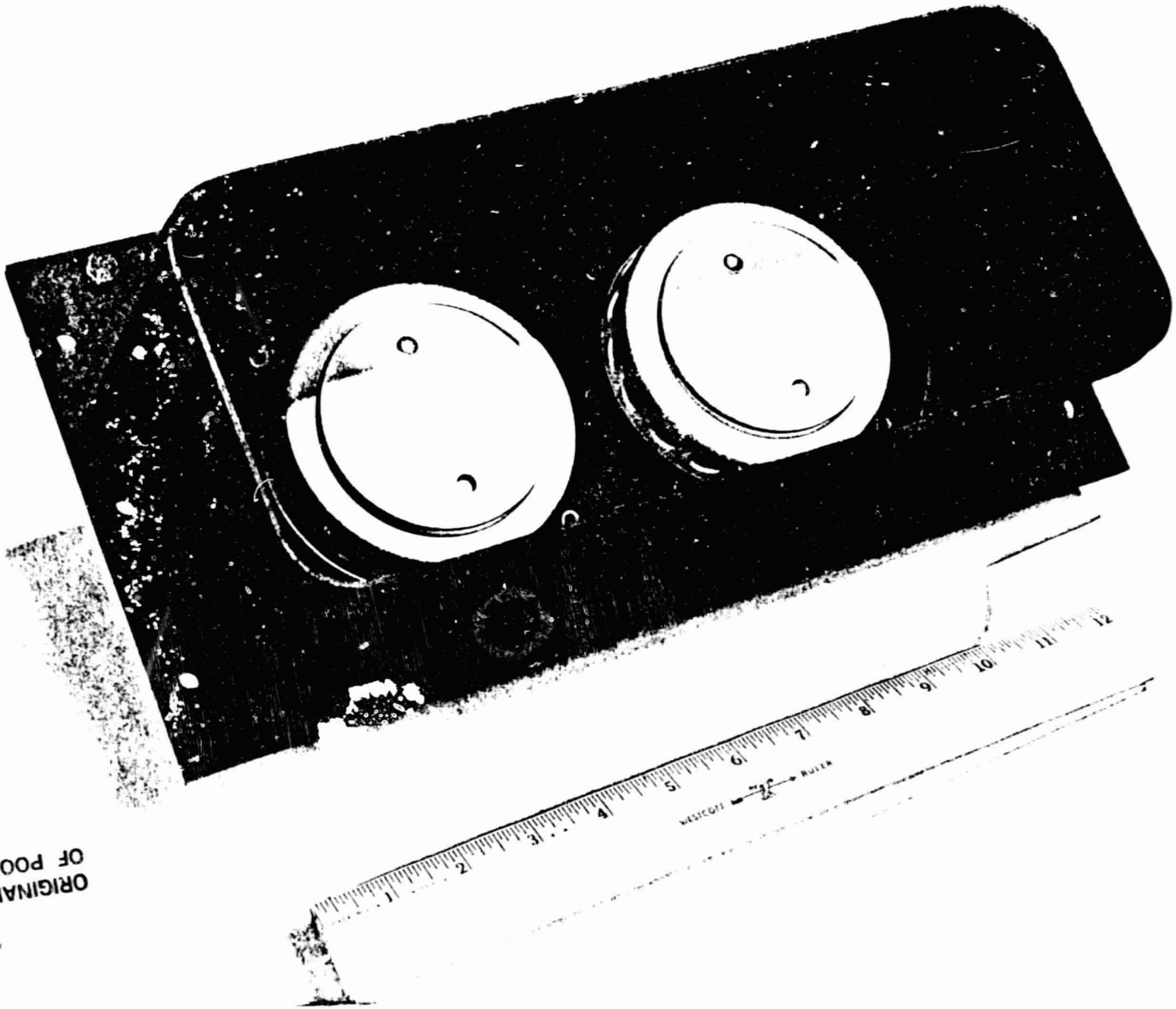


(a) Cover Removed to Show Stowed Air Bag

Figure 3-19. Ganged Driver Inflator Module, Pan Front



(b) Air Bag and Cover Removed

Figure 3-19 (Cont'd)

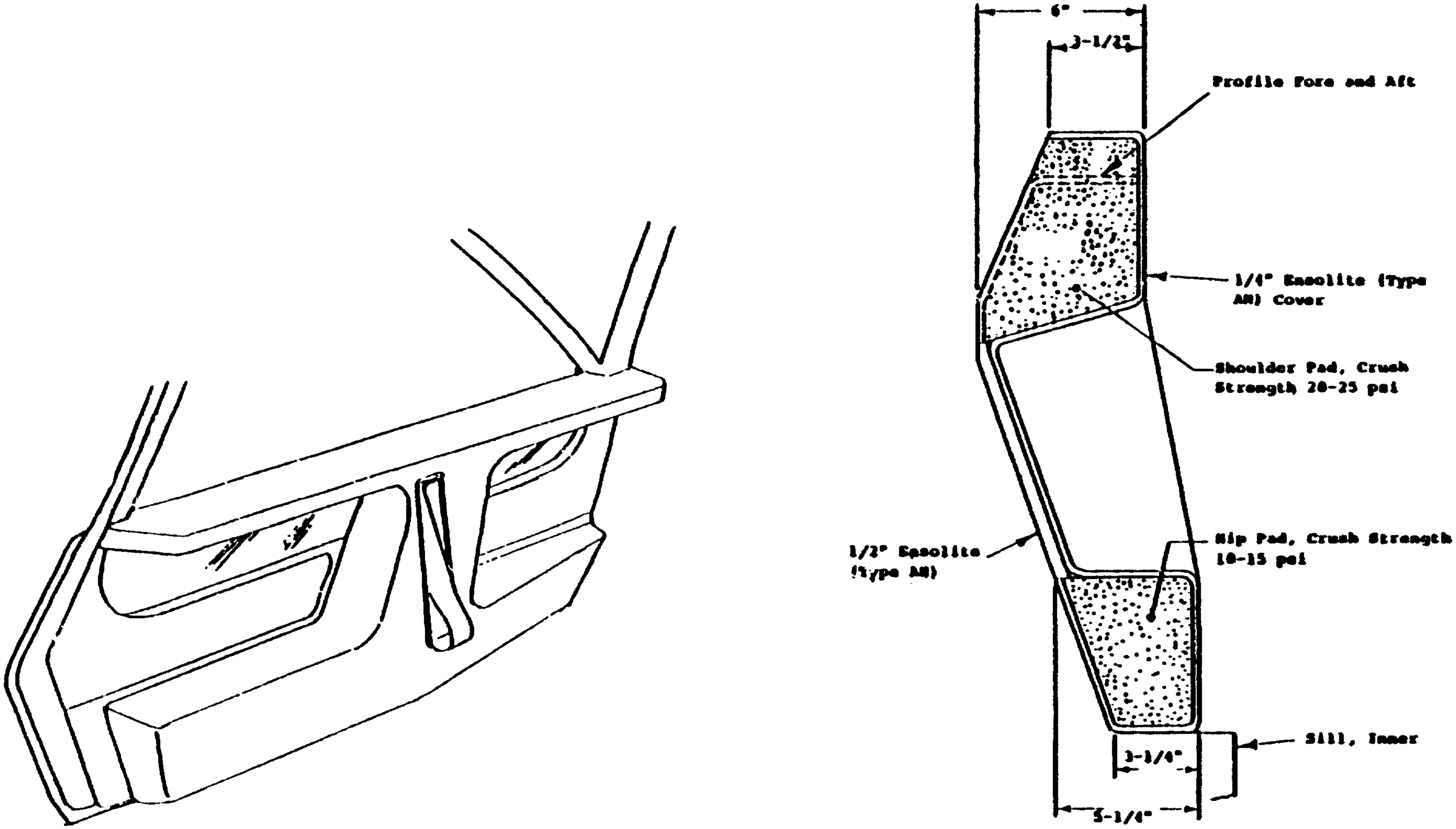


Figure 3-20. RSV Door Interior

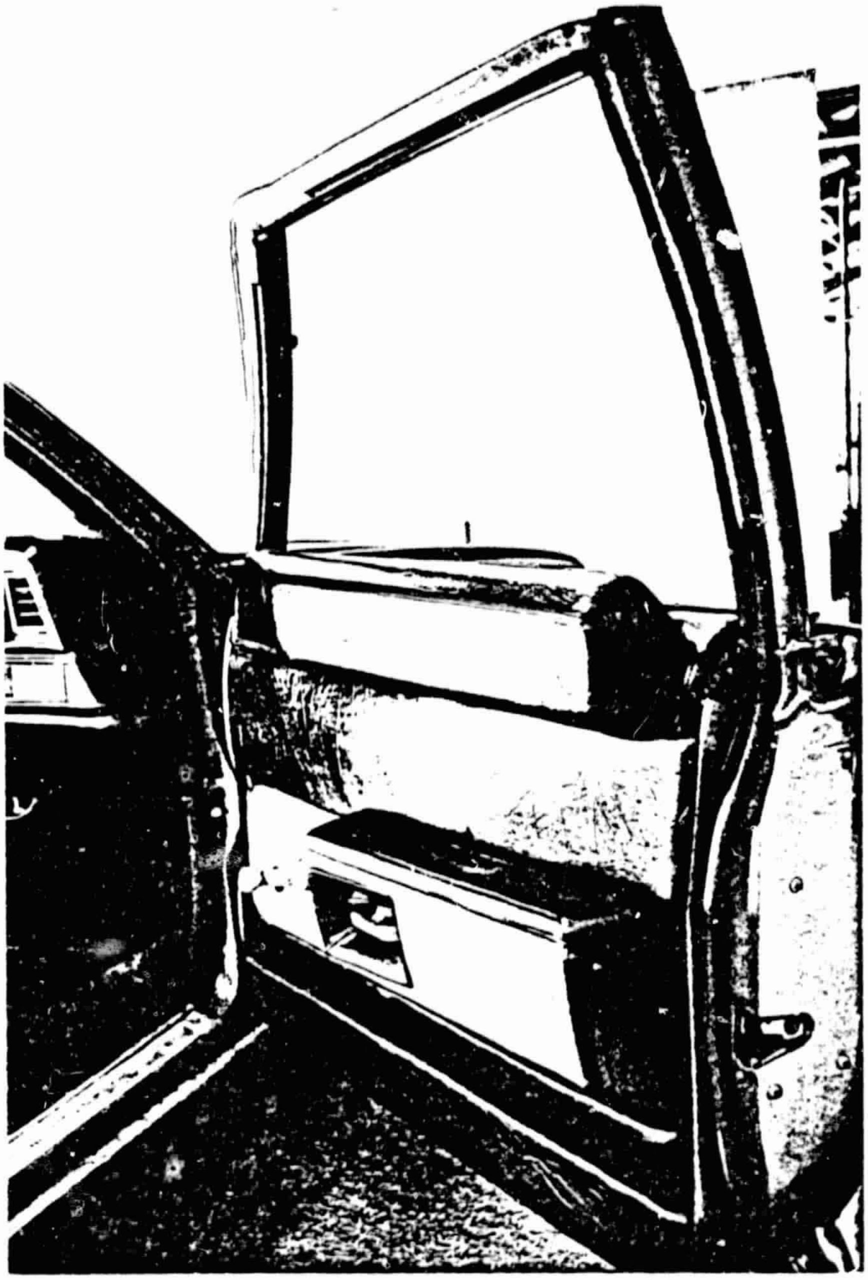


Figure 3-21. LRSV Door Interior

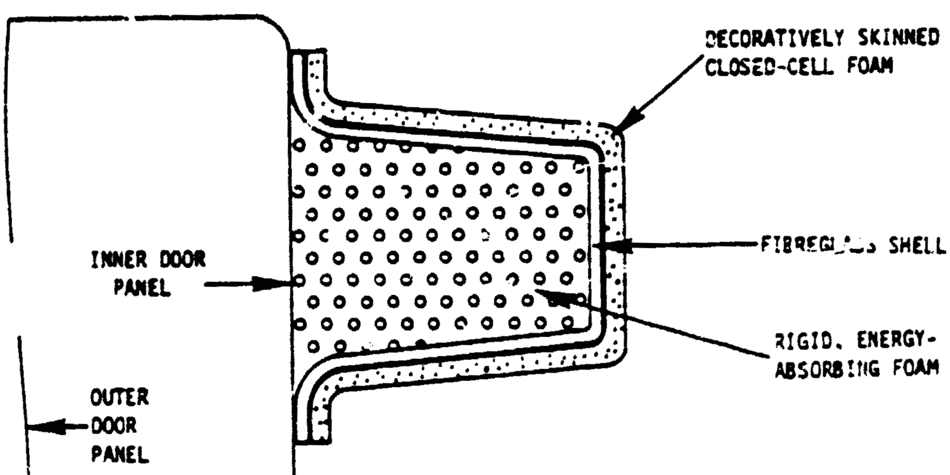
padding is certainly expected to be less than that shown on the RSV - probably on the order of that on the LRSV.

We have come to appreciate and be responsive to the reaction of occupants to encroachments into the living space of the vehicle compartment. One must be very careful to contour door interior padding to maximize the feeling of roominess, and, at the same time, to provide an adequate degree of padding where needed. To meet the anticipated side impact compliance test, we are estimating that about 2-1/2 to 3 inches of padding will be required immediately adjacent to the shoulder and hip areas. By proper design, to minimize the effect of this contour on "elbow room", the negative reaction to such padding can be minimized, if not totally eliminated.

The door contour will have the following features:

1. An adequate degree of padding at the shoulder and hip levels extending longitudinally (fore and aft) to be effective, regardless of seat adjustment position.
2. No padding where it is not required and where it will adversely effect elbow room.
3. A minimum of change to the existing Citation door interior layout.

The same materials that were used in the RSV and LRSV door padding will be used on the NTHV. These vehicles used door interiors configured as shown in the sketch below.



X-SECTIONAL VIEW

The crash energy absorption is accomplished by the rigid polyurethane foam at the core of the pad. The best foam crush strengths for limiting dummy accelerations are in the 15 to 25 psi range. This stiffness is not sufficient to stand up against normal day-to-day abuse. Therefore, the fiberglass shell and closed-cell (recovering) foam are used for this purpose. In a crash, the fiberglass breaks on impact, allowing the foam to perform its intended function. Since the padding contours will most likely be different than those we have developed previously, it is anticipated that density changes of the energy absorbing foam will be necessary, to alter its crush strength.

3.6.3 Expected Safety Performance

The driver and passenger restraints described in Subsections 3.6.2.1 and 3.6.2.2 are designed to meet the FMVSS 208 passive restraint requirements. This standard requires the demonstration of 30 mph barrier impact protection (against both a perpendicular barrier and barriers rotated up to 30 degrees from perpendicular) for 50th percentile male front seat occupants. In addition, the restraints will be developed to protect other occupant sizes at the 30 mph BEV level. At the driver station, protection will be established for the 95th percentile male and the 5th percentile female. At the front passenger position, attention will also be directed to providing protection to the child occupant, particularly when positioned in the proximity of the dash (as would occur in prebraking situations). The out-of-position case is one that is routinely addressed and solved in air cushion development programs.

The door interior padding will be designed to provide lateral impact protection to the 50th percentile dummy (being specially developed for the anticipated compliance test). Although protection for other occupant sizes would be desirable, the lateral response characteristics of the other dummy sizes presently available are not satisfactory.

3.7 VEHICLE DYNAMICS

This section describes the preliminary design of the vehicle subsystems that influence the ride, handling and braking performance of the NTHV. The subsystems addressed are:

- Front and Rear Suspension
- Steering System
- Brake System
- Wheels and Tires.

It is recognized that the primary emphasis of the NTHV program is on the development of the power train. However, the goal of the vehicle dynamics design is to ensure that none of the vehicle dynamic performance attributes that the American public expects will be unduly compromised during the development of the NTHV. To this extent, the preliminary design package described in the following identifies the modifications made to the baseline vehicle systems to achieve the goal.

3.7.1 Front Suspension

As described in Subsection 3.5.2.2, the NTHV employs a modified McPherson strut suspension from the Chevrolet Citation. This suspension is most suitable for the NTHV because it is very compact, maximizing the room in the engine compartment. It also allows easy front end alignment of both camber and toe in. The castor angle is fixed. The location points of the front suspension will be nominally identical to those on the base vehicle. Some design flexibility will be retained for these points, at least until handling tests have definitized their positions. This approach was used to adapt the Fiat X 1/9 suspension to the RSV (a heavier vehicle with different wheelbase and track width), and it should work as well for the NTHV. We expect that variations in vehicle parameters (such as changes in this polar moment of inertia) can be handled by minor suspension adjustments (so as to change the amount of roll steer, for example).

3.7.2 Rear Suspension

The rear suspension is described in Subsection 3.5.2.3. The geometry of the X-body rear suspension will be nominally retained in the NTHV.

3.7.3 Steering System

The base General Motors X-body cars are equipped with rack and pinion steering and have power assist as an option. The compact size and dimensions of the rack and pinion steering arrangement, shown in Figure 3-22, make it very suitable to the front drive arrangement of the NTHV design. The weight analysis indicates

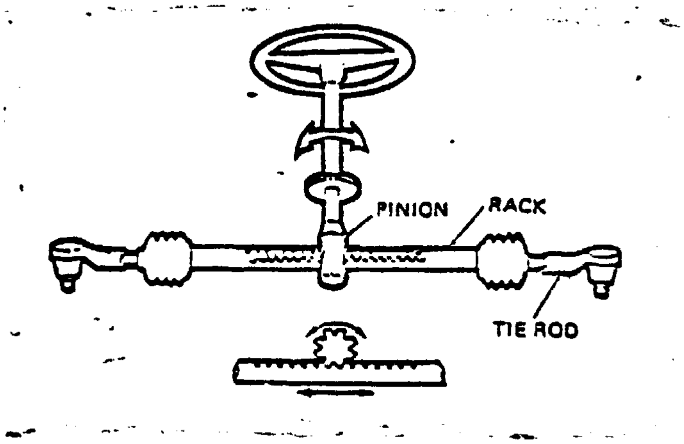


Figure 3-22. NTHV Rack and Pinion Steering Gear System

that the NTHV design will have to use the optional power assist, although vehicle handling tests may indicate otherwise. The addition of the power assist does not change either the basic mechanism or the linkage layout. The retention of the front suspension location points and steering linkage geometry as they are on the baseline vehicles ensures that the NTHV will not have steering geometry errors and other kinematics problems.

3.7.4 Brake System

The NTHV will use General Motors X-body brakes as a foundation brake system. This system is the front disc/rear drum brake system shown in Figure 3-23. The brake system will additionally employ regenerative braking by using the electric motor as a generator. This is discussed in section 3.7.5.

One nicety of the baseline brake system is the incorporation of a 'quick take-up' master cylinder. For years, disc brake pads have been permitted to rest lightly on the disc when the brakes are not applied. This was necessary because the large diameter caliper cylinders require a great deal of fluid to operate. Short pad travel was used to keep this fluid requirement to a minimum. The brake drag, although slight, was constant.

The 'quick take-up' master cylinder operates on a low/high pressure chamber. At the beginning of a brake application, pedal travel forces a large volume of fluid from a low pressure chamber. This fluid is forced into a high pressure chamber and on to the wheel units. The initial surge of the fluid quickly takes up the distance between the brake pads and the discs. A check valve in the low pressure chamber controls the pressure build up and bleeds excess fluid into the reservoir. After the quick take-up phase is completed, further pedal travel applies the brakes normally. Upon release, the fluid flows back into the master cylinder.

Another feature of the selected system is that it incorporates a diagonal split of the two hydraulic circuits. A front wheel and a diagonally opposite rear wheel are connected in a single hydraulic circuit to one chamber of the master cylinder. When a failure occurs in one hydraulic circuit, fifty percent of the braking capacity is still available.

Preliminary analysis of the brake system indicates that the regenerative braking may not always be available. Hence, the brake system preliminary design is based on the premise that the

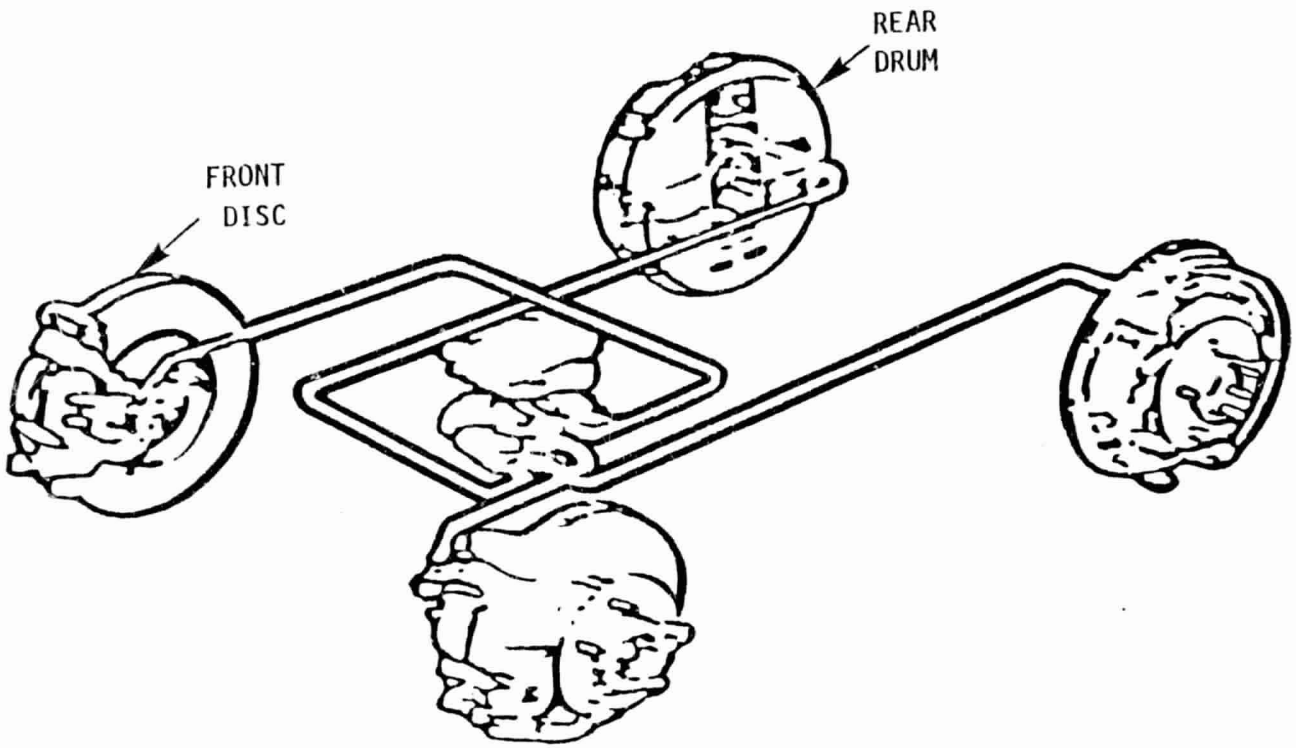


Figure 3-23. NTHV Diagonal Split Brake System

hydraulic brake system can provide the full braking capacity of the vehicle at any time.

The preliminary design indications on the brake system are as follows:

- The brakes will have power assist to keep brake pedal application to a low level
- The front disc brakes will be upgraded to accommodate the additional brake torque requirements of the NTHV, resulting from its greater weight
- The rear drum brakes will be upgraded for the same reason. Disc brakes are a candidate for the rear brake upgrading.

3.7.5 Regenerative Braking

Regenerative braking of some design has been used on several of the newly developed electric and electric hybrid vehicles.[6,7] The primary objective of regenerative braking is to use the braking energy to recharge the batteries, thus extending the range of the vehicle on one battery charge. The regenerative brake system designs described in Reference 6 included seven designs that used electric motors as generators and one design that incorporated use of a hydraulic regeneration unit. The trade off studies confirmed the merits of regenerative braking for the NTHV, and regenerative braking is thus part of the preliminary design. However, we are aware of the limitations and possible control difficulties of incorporating regenerative braking into the vehicle's hydraulic brake system. The more important of these limitations are discussed below.

1. The electric motor will not run below its base speed. Therefore, the transmission will have to automatically downshift in order to keep the motor speed up to where regenerative braking is available. The ability of the transmission to perform power shifts (discussed in Section 4.1.2) will be necessary to avoid nulls in the regenerative braking effort; nevertheless, the changes in motor speed will tend to cause changes in the regenerative braking effort. These changes could be annoying to the driver, or even unsafe (e.g., wheel lockup could occur), so the control system will probably have to provide smoothing of the regenerative braking effort.
2. The batteries have limited ability to accept high charging currents, particularly at or near full charge. The control

system will monitor state of charge and limit regenerative braking accordingly, in which cases the braking effort will have to be diverted to the foundation brake system. This must be done without significant changes in the brake system gain - at least as perceived by the driver.

3. When the electric motor is not being used, there will be a slightly longer delay time between brake pedal application and generation of brake torque from regenerative braking. This is because the motor will have to come up to speed. Again, the foundation brake system will have to make up the difference.
4. Regenerative braking will be available only at the drive (i.e., front) wheels.

Three strategies have been considered for integrating regenerative braking with the foundation brake system. They are:

- Regenerative braking can be activated when the driver takes his/her foot off the accelerator pedal, thus simulating heat engine motoring torque.
- Regenerative braking can be activated by the first portion of the brake pedal travel.
- Regenerative braking can be activated by a combination of the above two.

The last alternative is favored at this stage.

3.7.6 Tires and Wheels

The tire selection takes into account the following factors:

- low rolling resistance
- load carrying capacity
- cornering and ride comfort.

On this basis, the preliminary design uses P205/75 R 14 tires and 14-inch wheels.

3.8 AERODYNAMICS

An improvement in the aerodynamics of the base vehicle through drag reduction will make a significant contribution to energy savings. Generally, a drag reduction of 10 percent can add as much as 2 miles per gallon to the performance of a conventional automobile. The published coefficient of drag (C_D) for the

Chevrolet X body (4-door fastback sedan) is 0.42. It is estimated that, with modifications, the C_D can be reduced significantly without major alteration of the cross-sectional area.

The major improvements in drag reduction will be found in changing the nose shape (required for the front battery compartment) and flow characteristics, and the addition of a front air dam and appropriate rear spoiler, with some small changes in the rear deck and bumper shapes. Further drag reduction can be achieved with fairings or by re-shaping the lower surfaces of the windshield (see Figure 3-24). The details of these improvements will be worked out in wind tunnel testing during Phase II.

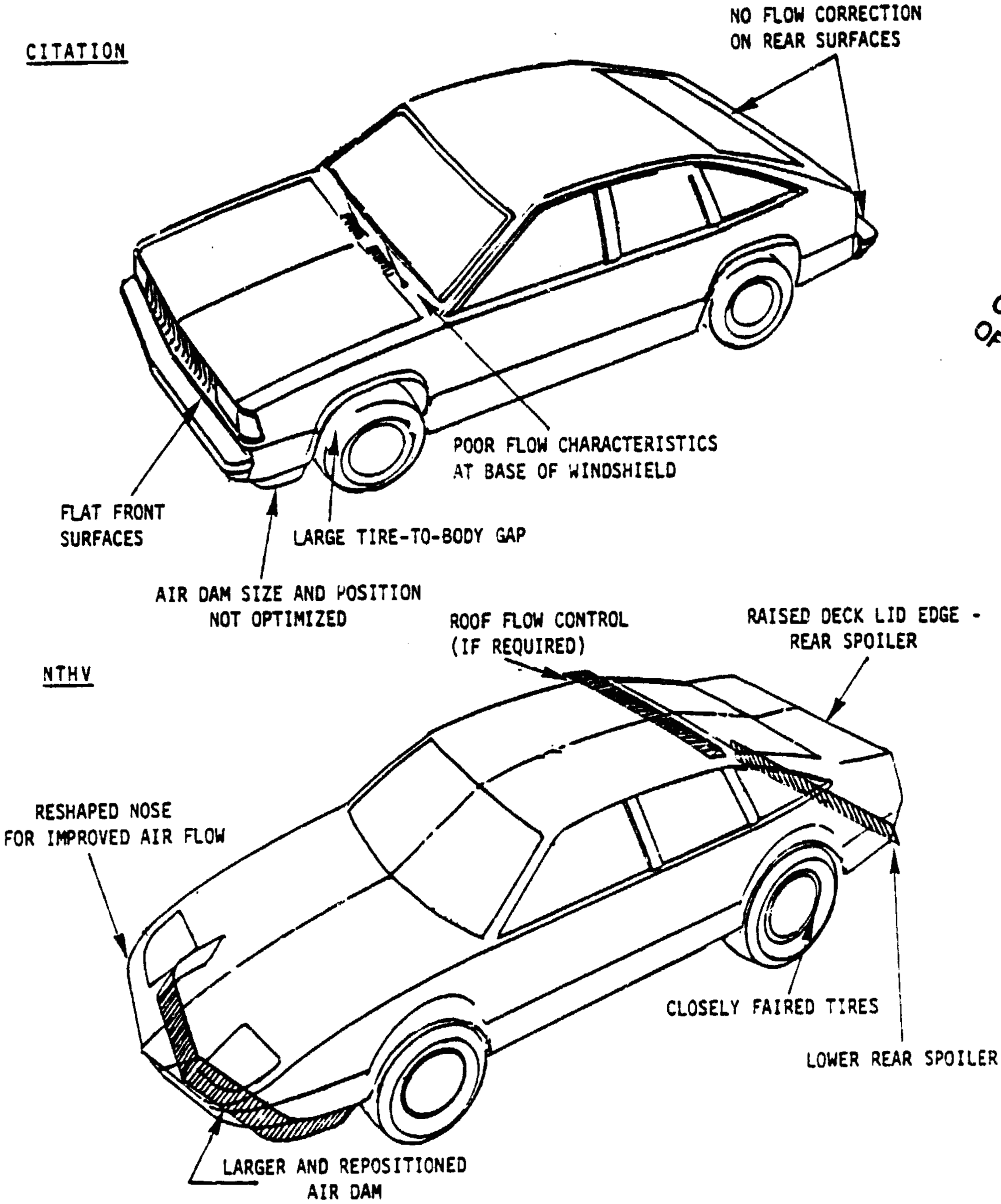


Figure 3-24. Comparison of Citation and NTHV Aerodynamics

SECTION 4

DRIVETRAIN DESIGN

4.1 TRANSMISSION CONCEPT SELECTION

The NTHV transmission must perform a number of functions. It has to connect either the electric motor, or the diesel engine, or both, to the drive wheels. It has to provide a means for starting the vehicle from rest with either the motor (which with field control alone cannot run below its base speed) or the engine (which cannot run at a speed lower than its idle speed). It has to have sufficient gear ratios to effectively transfer the power at various vehicle speeds. It has to select the proper gear ratios and the proper power source for optimum performance. And it must perform all of these functions both efficiently and reliably.

4.1.1 Automatic vs. Manual Control

For a hybrid vehicle, the terms "manual" and "automatic" need additional definition when applied to the transmission. As the term is used here, the "manual" transmission hybrid is a vehicle whose transmission ratios and power source are selected by the driver. The transmission would most likely have a clutch or clutches to disconnect the engine and/or motor from the driveline while the vehicle is stopped. An "automatic" transmission is one in which the transmission control system performs most or all of these functions. The gear ratios are selected "automatically," as is the power source, and the starting of the vehicle is controlled by the transmission, with the only input required by the driver being the control of the accelerator pedal after the transmission is put into Drive. Advantages of a manual transmission are simplicity, efficiency, light weight, and full controllability by the driver. On the other hand, an automatic transmission offers greater ease of vehicle operation, its operation is more repeatable, and it always performs as programmed.

The hybrid vehicle is of necessity complex and quite different from the cars most drivers are used to. As a matter of philosophy, it is not the object of this program to develop a vehicle that must be driven by an engineer or specially trained individual in order to achieve smooth operation or optimal efficiency. Rather, our objective is to demonstrate the utility of hybrid vehicles in normal use, by drivers who have no particular technical expertise.

This is the overriding factor in the selection of transmission type.

The NTHV has to operate with no more input or thought on the part of the driver than he or she would expend in a non-hybrid. The average driver expects to get into the car, turn the key to start, put the transmission into drive, and move away. After that, he or she expects to run the car with only three controls: the accelerator pedal, the brake pedal, and the steering wheel. To be an acceptable replacement for the reference vehicle, the NTHV must be capable of being driven the same way. It is essential that the driver not be required to make any decisions for the transmission other than the direction of travel. As a result, the NTHV must have an automatic transmission.

4.1.2 Power vs. Non-Power Shift

Transmissions, whether automatic or manual, may be divided into two types — power shift and non-power shift. The difference between the two is in the interruption (or lack of it) in the power flow to the driving wheels during gear changes. In the U.S. market, manual transmissions are generally non-power shifted and automatic transmissions are power shifted.

A power shift transmission is onein which the power flow to the drive wheels is maintained even while the transmission is shifting between gears. This is usually accomplished with a planetary transmission, in which the ratios are changed by the application and release of brakes and clutches. The release of one friction element and the application of another can be performed smoothly, without an interruption of power transmission. On the other hand, there are such things as power shift non-planetary, non-automatic transmissions. One is the Hondamatic, available in all Honda automobiles. This is a spur gear transmission, similar to a Honda manual transmission, but using clutches to connect the two forward gears to the driveline. The change from one of these clutches to the other allows for a power shift, but is controlled by the driver, and is not an automatic shift.

In a non-power shift transmission the engine/motor power must be disconnected from the wheels, the gear ratio changed, and the power re-connected. A normal manual transmission is a typical example. When it is time to shift gears, the throttle is closed, the clutch is disengaged, the gears are changed, the clutch is re-engaged and the throttle is opened again. By the very nature of the unit, a synchromesh transmission can not have a new gear engaged, unless the drive is disconnected from the transmission.

A non-power shift automatic transmission is also possible. Minicars, Inc. is currently developing such a unit for the High-Technology version of the Research Safety Vehicle (RSV). This transmission is an automatically shifted version of the Honda five speed manual that is used in the standard RSV. An on-board microcomputer controls the clutch, the transmission and the engine throttle. The operating force for all of these functions is provided by compressed air. The throttle, clutch and transmission are operated as if a skilled driver were at the controls.

This transmission was constructed in an effort to obtain the efficiency and fuel economy of a manual transmission, but the driving ease of an automatic. Currently, no test data are available on the transmission, since it is still in the mechanical development stage, but the computer simulations predict a 7 percent improvement in urban fuel economy, due entirely to the ability of the computer to select the proper gear ratio for best economy, which the average driver would not do. Although this transmission is still under development, it is far enough along to show that it is a potential alternative to the conventional automatic transmission, one that is expected to offer lighter weight and better efficiency than a conventional automatic transmission.

There are, therefore, two viable alternatives for an automatic transmission in the NTHV — a power shifted unit such as a conventional U.S.-built automatic, or a non-power shifted transmission, such as the computer controlled manual being developed for the High-Technology RSV. The computer controlled manual has the potential advantages of lighter weight and greater efficiency, and the disadvantage of being newly developed technology. The more conventional power shift unit has the advantages of using a well developed technology and the smoothness and consistency of a power shift transmission.

One very important factor in favor of the power shift transmission is its predictability and familiarity. Since the NTHV, as proposed, will be an expensive vehicle in a fairly expensive segment of the automotive market, it will tend to appeal to drivers who are used to driving at least semi-luxury vehicles, especially to the smoothness and predictability of the typical automatic transmission. The interruption of the driving power during the shifts of the non-power shifted, computer controlled manual transmission could cause considerable concern to the non-technically oriented driver of such a vehicle. The shifts will be, in the eyes of the driver, essentially random, unexpected, and, at times, very inconvenient. A potentially dangerous situation could result while passing another vehicle on a two lane road with on-coming traffic. If the transmission shifted, the engine throttle would close and the clutch would disengage for a minimum of 0.5 seconds. This would slow the entire passing manuever; more importantly, it would cause an apparent loss of power during a critical situation.

With a power shift transmission the shift logic might start the shifting sequence at the same point during the passing manuever, but there would be no interruption of the power flow to the wheels. The driver might not even be aware that the transmission had shifted. This difference could be enough to make the whole NTHV unacceptable to a large portion of the public. As a result, we feel that it is essential that the NTHV have a power shift transmission, despite its potential lower efficiency, in order to provide a fully acceptable vehicle for a non-technically oriented driver.

4.1.3 Continuously Variable Transmission

The continuously variable transmission (CVT) is another possibility for the NTHV. The CVT has the advantage of providing optimum gear ratios for all driving conditions, which theoretically should optimize energy usage. Unfortunately, even though a great deal of research and development has been conducted on CVTs, none of these units has yet reached the level of near term availability with reliability and efficiency.

4.1.4 Transmission Efficiency[8]

A power shift automatic transmission is usually associated with greater power losses. Figure 4-1 shows the losses of typical automatic and manual transmissions. The losses in the manual transmission are due almost totally to friction and oil churning. The losses in the automatic transmission are caused by the torque converter, the transmission fluid pump, band and clutch drag, and friction. All of these items can be considerably reduced over the values shown in Figure 4-1. As will be described in more detail later, the torque converter losses can be eliminated, for all conditions except start-up, by the use of a lock-up clutch. The pump losses can be minimized by the use of a variable displacement

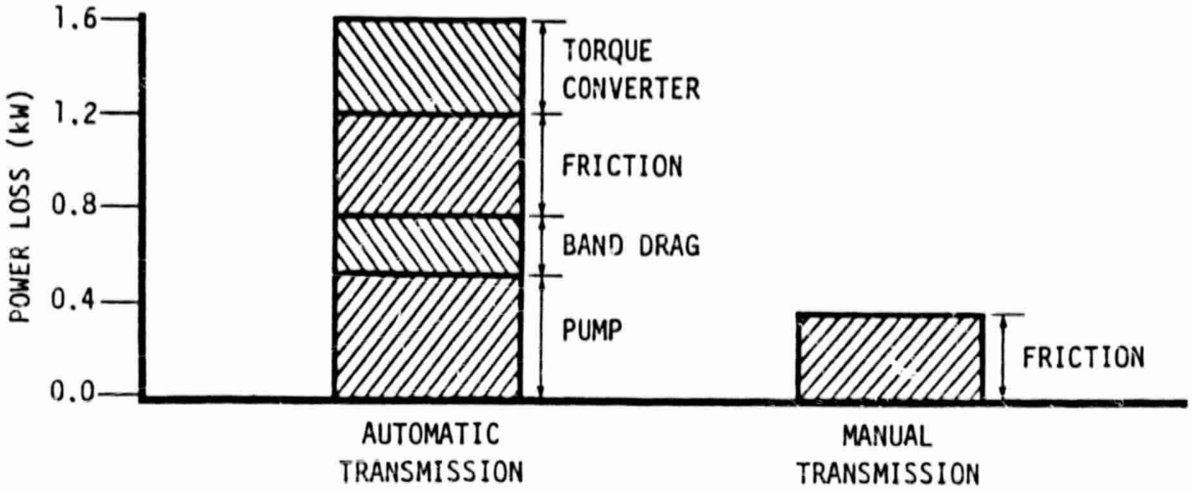


Figure 4-1. Transmission Power Loss – Typical Automatic and Manual Transmission

or variable speed pump with an accumulator. The clutch and band drag can be reduced by careful attention to detail and greater free play in the clutches when they are not applied. The transmission friction can also be minimized by careful attention to detail and fit. Figure 4-2 shows an estimate of the level to which the automatic transmission losses can be reduced, together with the losses associated with the computer controlled manual transmission (which requires a pump). The power shift transmission still has greater losses than the non-power shift transmission – but we regard those losses as a price to be paid for its greater acceptability.

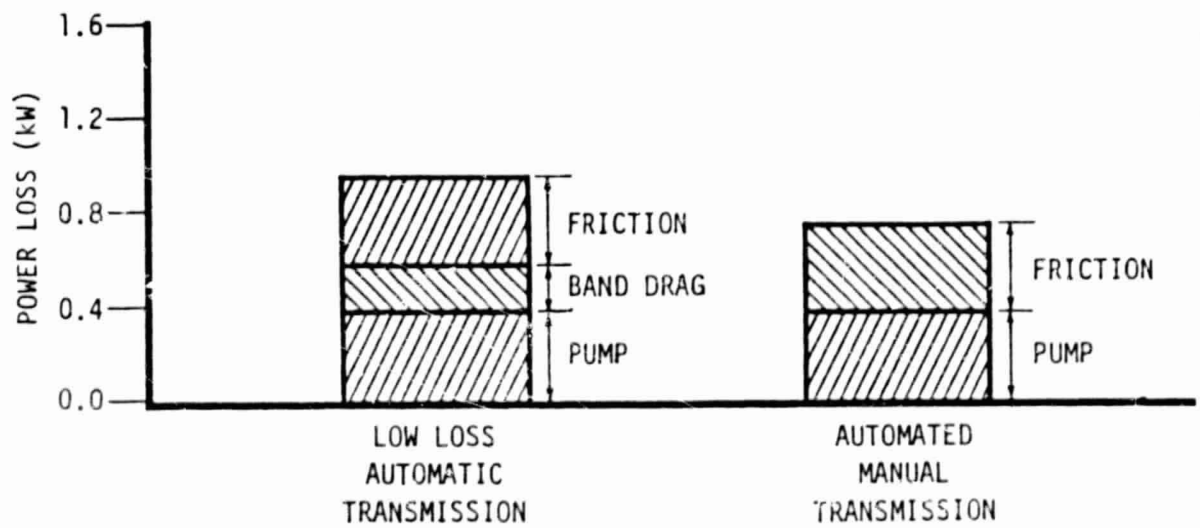


Figure 4-2. Transmission Power Loss – Low Loss Automatic and Automated Manual Transmissions

4.1.5 Gear Ratio and Final Drive Ratio Selection

The power shift automatic transmission can have any number and spacing of gear ratios, although currently four gears is the maximum used for passenger car automatic transmissions. Minicars trade-off studies showed that overall fuel and energy efficiency are relatively insensitive both to the spacing of the transmission ratios and to the exact final drive ratio. Vehicle acceleration, on the other hand, varies with gear ratio, and the overall ratio in first gear is the most important factor for standing start acceleration. Since fuel and energy consumption is of principal importance, the selection of the transmission and final drive ratios can be determined primarily according to other factors in the selection process, such as the transmission adaptability to the NTHV.

4.2 TRANSMISSION DESIGN

4.2.1 Development Approach

The transmission must accept inputs from the engine and the motor, either separately or together, and provide the proper gear ratio and final drive ratio for best efficiency under any sort of driving condition. It can be either one transmission with separate inputs for the engine and the motor, or two separate transmissions, one for each power plant, whose outputs are combined at the final drive. Obviously a single transmission would be lighter and simpler and should be used unless there is an overriding need for the two separate transmissions. No such overriding need has been found, since the same three gear ratios can be used effectively by both the engine and the motor. Therefore, the NTHV preliminary design reflects one basic transmission with two separate inputs.

The transmission for the NTHV can be either a new unit designed for this particular vehicle, or it can be a modified version of an existing production transmission. In either case it must be a transaxle unit, a combined transmission and final drive assembly. It must be designed for use on a transverse engine layout to fit into the X-body engine compartment.

If a new transaxle unit is designed specifically for the NTHV, it will have the advantage of having exactly the gear ratios and final drive ratios that will best match the mission for which the vehicle is designed. The transmission housing can be designed to locate all of the components in the best location for optimum packaging. The disadvantages of designing a transmission specifically for the NTHV are time and cost. The modern automatic transmission is a very complex, highly developed unit that requires a long extremely expensive development program. The design and development of a new automatic transmission in Detroit is usually a several year, multi-million dollar program — even excluding the transmission's production design and tooling. It is possible to make a transmission that will work in 9 to 12 months for less than a million dollars, but it is highly unlikely that this transmission would have the durability and smoothness of operation that the American public has come to expect.

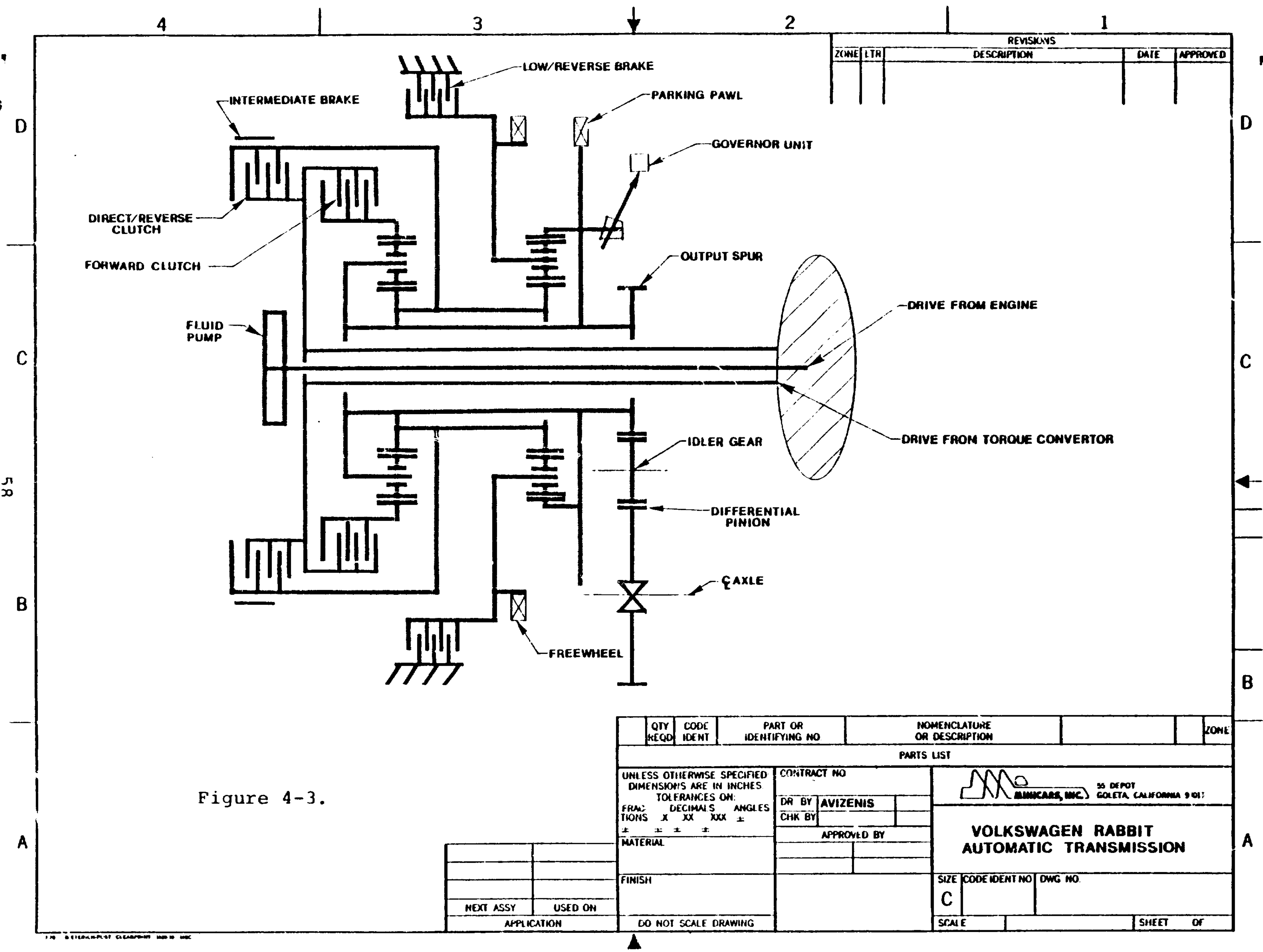
Since the basic approach of this program is to develop an easily built vehicle with a minimum of new technology, it appears highly desirable to use a modified version of a production automatic transmission, as long as the overall performance of the vehicle is not compromised. As will be shown below, this can be done; a modified production transmission can be used for the NTHV.

4.2.2 Production Transmission Alternatives[9,10,11]

There are, currently, three production transmissions that are suitable for the NTHV. These are: the Volkswagen automatic transmission used on the VW Rabbit and the Fiat Strada, the Chrysler A-404 used on the Omni and Horizon, and the Turbo-Hydramatic 125 used on the GM X-body cars. The other transverse-engine automatic transmissions made in Europe have various shortcomings in adaptability or reliability.

All three of the candidate transmissions could be modified for use on the NTHV. All are three-speed planetary transmissions which use torque converters and are representative of the latest practice in automatic transmission design. Schematics of the three are shown in Figures 4-3, 4-4, and 4-5.

The Volkswagen and Chrysler transmissions are similar in basic design, in that the torque converter in each is on the end of the crankshaft, and the three-speed transmission comes directly after the torque converter. The transmission output power is transferred by a set of helical gears to an intermediate shaft or gear and then by a helical final drive gear set to the differential. The output of the Volkswagen transmission is concentric with its input shaft, and the transfer gears are located between the torque



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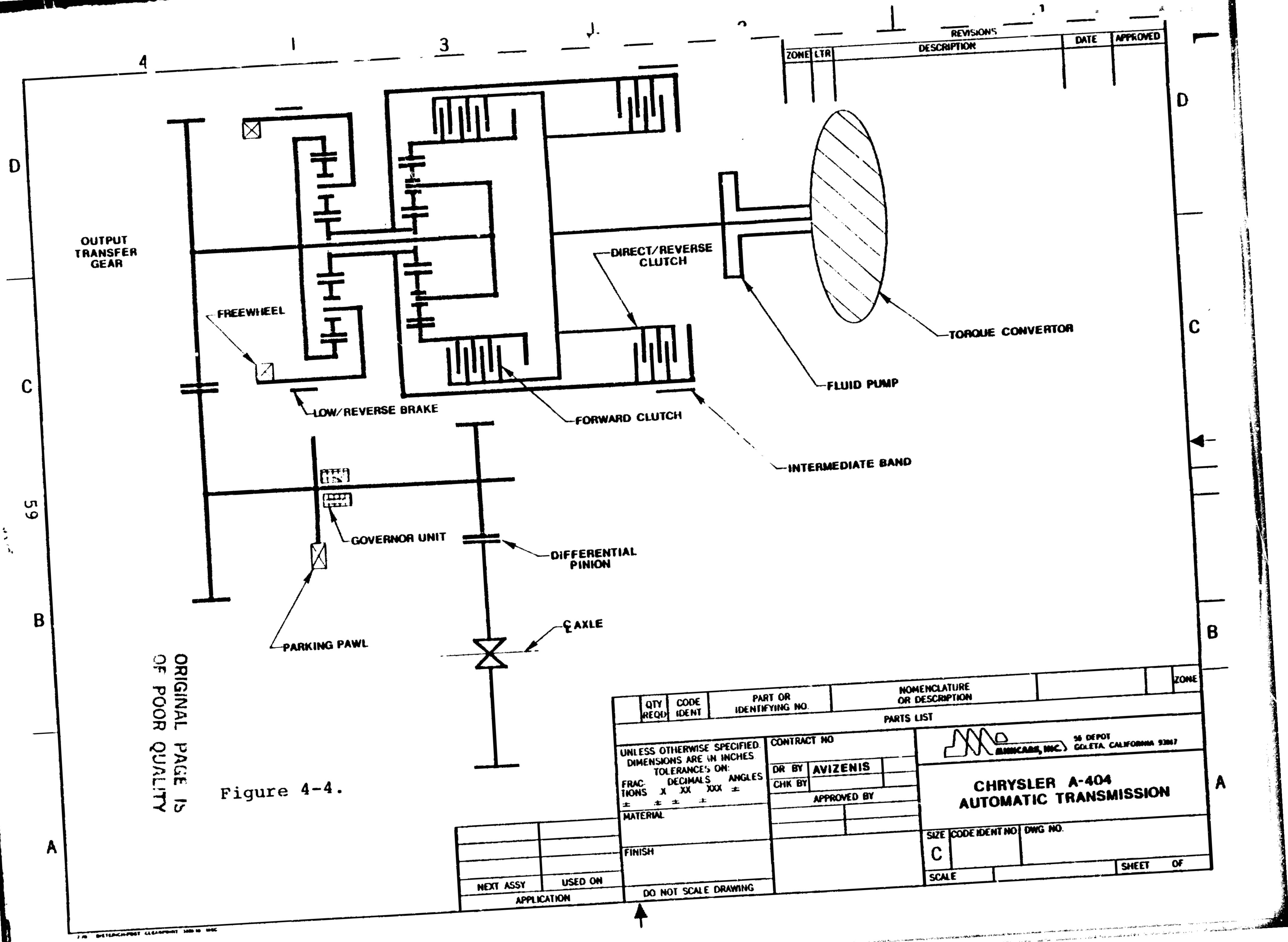

Figure 4-4.

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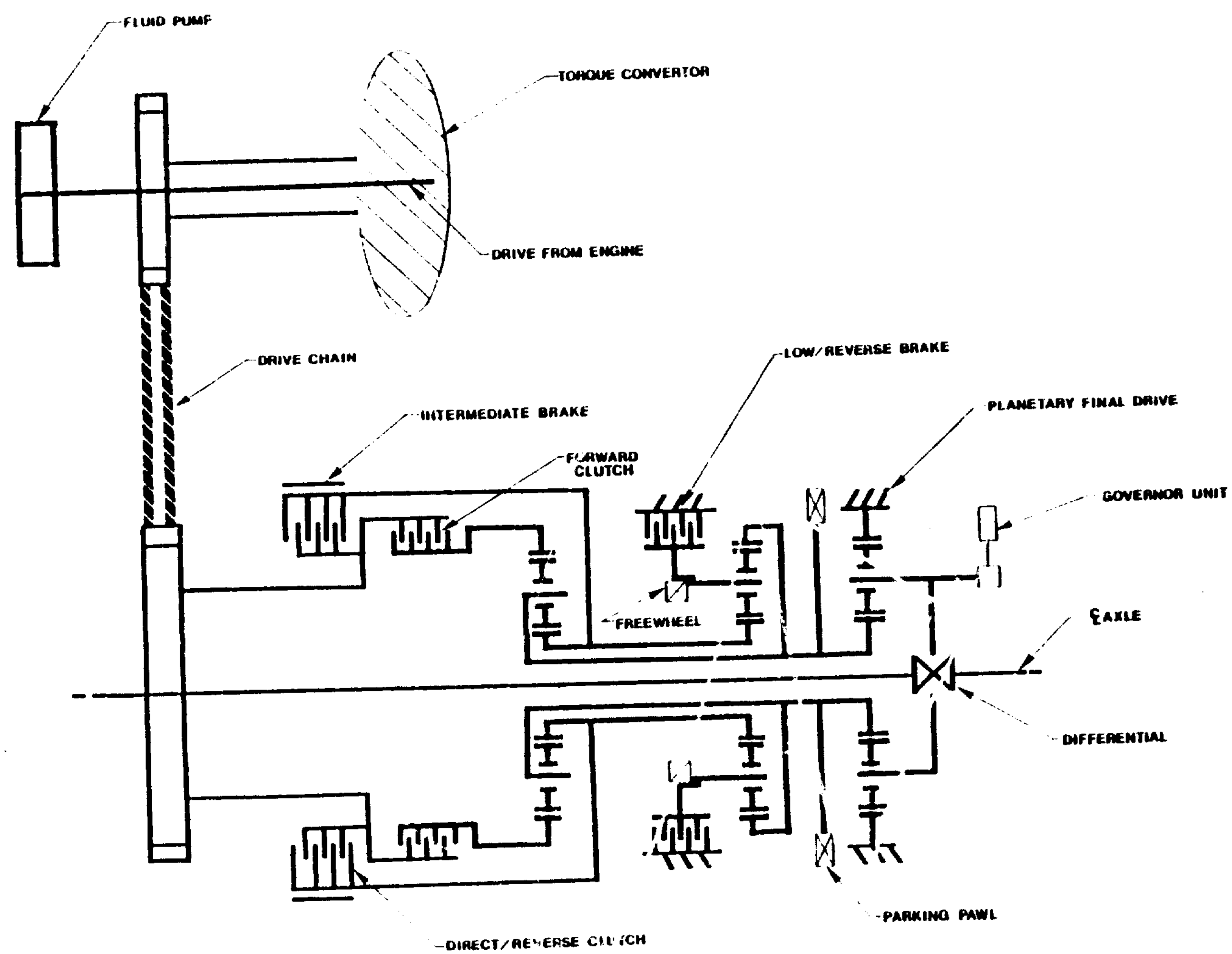

Figure 4-5.

GENERAL MOTORS TH-125
AUTOMATIC TRANSMISSION

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converter and the transmission gears. This design was chosen because the same transmission section is used for Volkswagens and Audis (such as the Volkswagen Dasher and the Audi 5000) that have their engines located fore and aft. In these cars the hypoid final drive gears come between the torque converter and the transmission gears. The pinion gear of the hypoid final drive is mounted where the transfer gears are on the Rabbit transmission.

In the Chrysler transmission the output is on the other side of the transmission from the input. The transfer gears are located on the very end of the transmission housing, and the intermediate shaft returns the power (toward the center of the car) to the final drive gears. With this layout Chrysler can retain the option of using the same basic transmission in front engine, rear wheel drive cars.

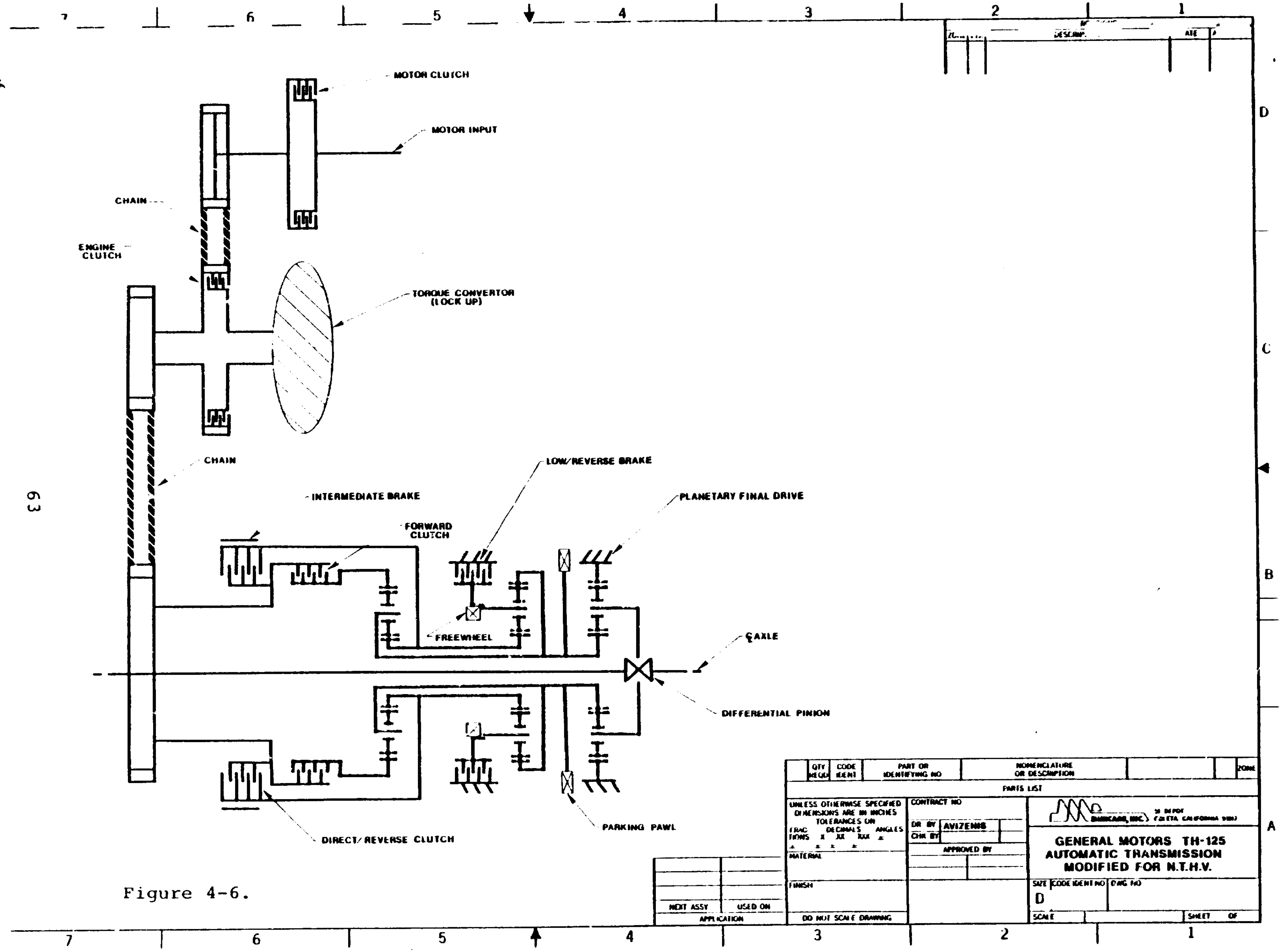
The layout of the General Motors transmission is quite different from the others. The torque converter is still on the end of the crankshaft, but the converter output is connected to a chain drive that transfers the power to the transmission itself (which is behind and parallel to the engine). Since there is no reversal of rotation in the chain drive, as there is in the helical gears used by Chrysler and Volkswagen, General Motors would have to use an intermediate gear if they had a helical gear final drive. To eliminate this extra gear, the final drive is a planetary reduction gear driving the differential. The output shaft to the left wheel runs through the center of the transmission before coming to the left universal joint. This universal is beyond the chain sprocket which drives the transmission. General Motors has chosen this type of transmission for three reasons: to use their previous experience (on the Toronado) with chain drives between the torque converter and the transmission, to obtain a shorter overall engine-transmission length (the 2.5 liter four-cylinder engine is longer than most transversely mounted engines), and to allow the use of the basic Turbo-Hydramatic 200 gear train that is designed for front engine, rear-wheel drive cars.

All three of these transmissions could be adapted for use in the NTHV, and none has a marked superiority over the other. The Volkswagen unit has the fewest advantages for the NTHV, both because of the complexity of its multiple concentric shafts and the somewhat greater difficulty of adapting it for input from an electric motor. The General Motors and Chrysler units are essentially equal in adaptability. They are, therefore, our primary candidates for the NTHV.

Figure 4-6 shows the method of adapting the General Motors Turbo-Hydramatic 125 to the NTHV. The transmission housing is modified to put the input from the motor between the torque converter and the chain drive that transfers the power from the torque converter to the transmission. The power from the motor goes first through the slipping clutch mounted on the motor output, then through a chain drive, and finally to the torque converter output. The rest of the mechanical portion of the transmission remains the same. The drive from the motor is controlled by its clutch. The drive from the engine is controlled by either a friction or an overrunning clutch between the torque converter and the motor input. The overrunning clutch would disconnect the engine when it was running slower than the motor input, or was turned off, and would automatically let the engine take up the drive when it came up to speed. It would also automatically disconnect the engine during overrun conditions, so that the motor could provide regenerative braking. Either the friction or the overrunning clutch would serve this purpose. The only potential problem with the overrunning clutch is that torsional vibrations from the diesel engine could carry through to the clutch. These vibrations could cause the unit to lock and unlock at the torsional frequency, which would drastically reduce the life of the clutch.

Figure 4-7 shows the method of adapting the Chrysler A-404 transmission to the NTHV. In this case the chain drive from the motor and its slipping clutch is located on the end of the transmission (beyond the transfer gears for the intermediate shaft). The power from the motor passes through a shaft in the center of the transmission output shaft, and is connected to the transmission input shaft (converter output). So, despite the totally different physical layout, the motor would connect to the transmission in the same place in the power flow. In this transmission the power from the motor would be controlled by a slipping clutch and the power from the engine by an overrunning clutch. There is not sufficient room for a clutch to disconnect the engine in the Chrysler transmission (as there is in the General Motors transmission), so the torsional vibrations of the diesel engine would have to be controlled by a vibration damper on the engine and damper springs on the lock-up clutch in the converter.

The amount of physical modification to the transmission cases and shafts would be about equal for the two transmissions. On the General Motors transmission the modifications would be entirely within the torque converter housing - the transmission would be adapted to take a different torque converter and the motor drive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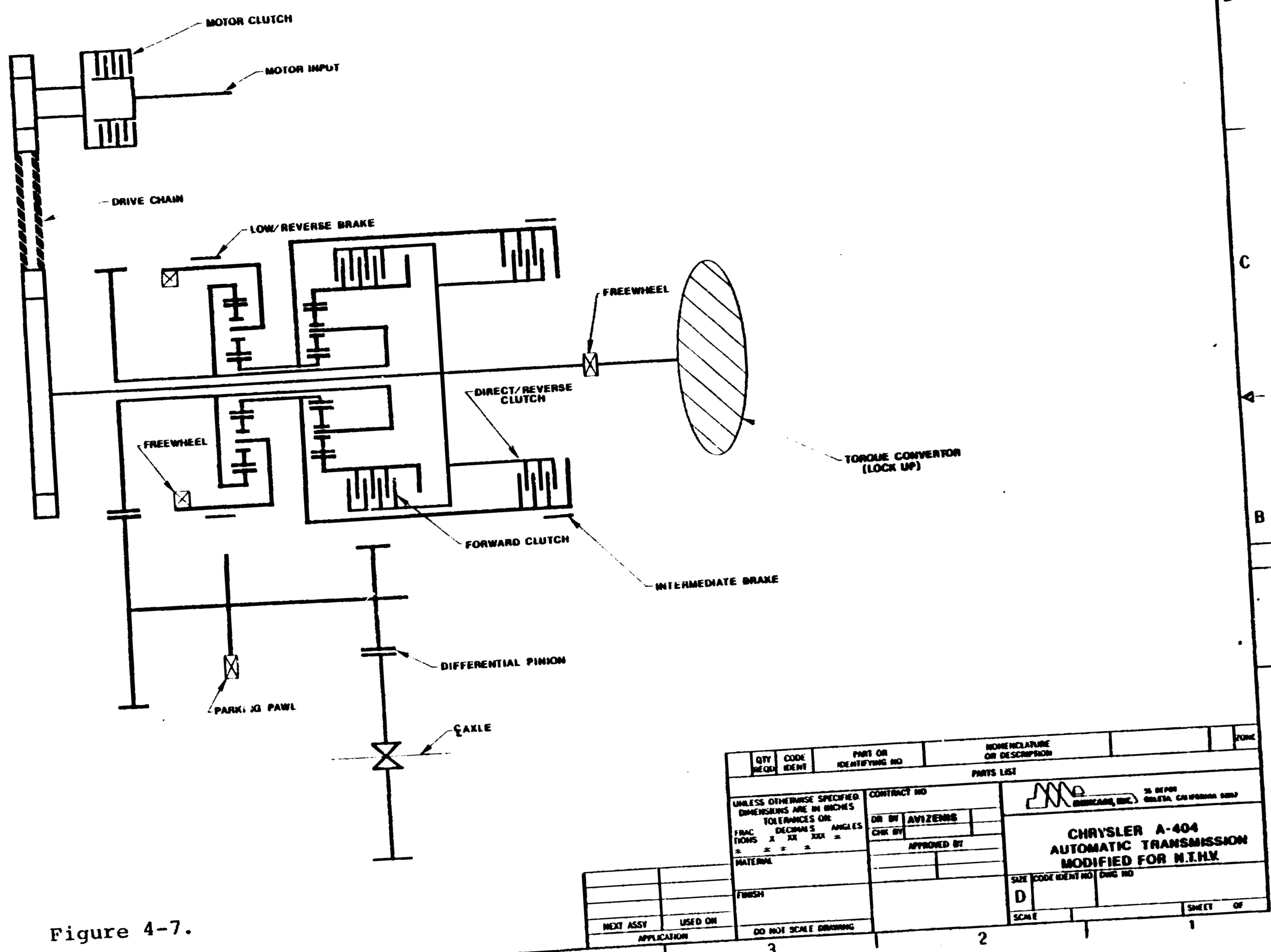
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Figure 4-7.

sprocket and engine clutch would be installed. The transmission case would have to be extended to allow for the additional length of the motor drive input. The case would also have to be modified to bolt to the Volkswagen engine and the housing for the chain and the motor slipping clutch would have to be added.

The modifications to the Chrysler transmission would involve the addition of the motor chain drive sprocket and housing and the alteration of the transmission shafts to allow the motor drive to reach the transmission input. The complete chain case and slipping clutch housing would bolt to the main case, to replace the normal cover of the transfer gears. The Chrysler transmission would bolt directly to the Volkswagen diesel engine (since the Omni/Horizon engine is a Volkswagen engine).

Neither of the two proposed transmissions has a major advantage over the other. There is a small benefit to the General Motors unit because its wider gear ratio spacing would be of some advantage in fuel economy when the diesel is running. Further, this transmission would have the room to use a friction rather than an overrunning clutch to connect the engine to the transmission, thereby avoiding one possible area of additional development.

4.2.3 Transmission Oil Pump

For either transmission, the standard oil pump will have to be replaced. The pump drive in the Turbo-Hydramatic 125 is ahead of the torque converter, so that it can be driven only by the engine. In the NTHV the pump will also have to be driven by the motor (ahead of the motor clutch). To drive the pump from both of these locations would require a complex drive and another chain just for the pump. Fortunately, there are two alternatives to dual drive.

The first alternative is to drive the transmission oil pump from the accessory drive system (described below). The accessory drive would be powered by either the engine or the motor, whichever is running. This would provide a drive to the pump whenever the transmission needs oil pressure.

The other alternative is to drive the pump from a separate, variable speed 12-volt electric motor. Despite the inefficiency of having an additional drive, the electric pump is to be preferred, because it has a lower power loss than a mechanical pump.

On most automatic transmissions the pump is a constant displacement pump and is driven at engine speed. The pump, therefore, has to have adequate capacity at idle and at stall speed, to provide all of the necessary oil flow and pressure. At higher engine speeds the pressure control valve bypasses the excess oil; but the power to drive the pump increases with engine speed. The General Motors Turbo-Hydramatic 125 is unique among current production automatic transmissions in that it uses a variable displacement oil pump; the pump eccentricity is varied (by the pressure regulator) to control the oil flow. At higher speeds this pump has much lower power requirements than the conventional pump, but its power consumption still increases with speed.

The electrically driven pump would be a constant displacement, but variable speed, pump. The motor speed would be varied to control the oil pressure, and the pump flow would be matched to the requirement of the transmission by varying the pump speed to maintain the desired line pressure. An accumulator would be placed in the output line of the pump to take care of sudden large oil flow requirements (for instance, during a shift).

4.2.4 Transmission Control System [12, 13, 14, 15]

4.2.4.1 Standard Hydraulic Control System

Figure 4-8 shows the hydraulic control system of the Turbo-Hydramatic 125. This is typical of the hydraulic system on most automatic transmissions. The transmission oil is used for a variety of different purposes. It lubricates the transmission, provides the fluid for the hydrodynamic torque converter, operates the hydraulic analog control system, and provides the pressure (controlled by the analog system) to actually shift gears. The control system does all of these jobs well, but there are limitations in its degree of control. If a greater degree of control could be exercised over the shift points of the transmission, gains in fuel economy could be achieved. When the second power source of the hybrid vehicle is added to the transmission, a totally hydraulic system cannot provide the control necessary.

4.2.4.2 Control Requirements for the NTHV

The addition of the electric motor and the other changes in the transmission system for the NTHV add several new functions to the

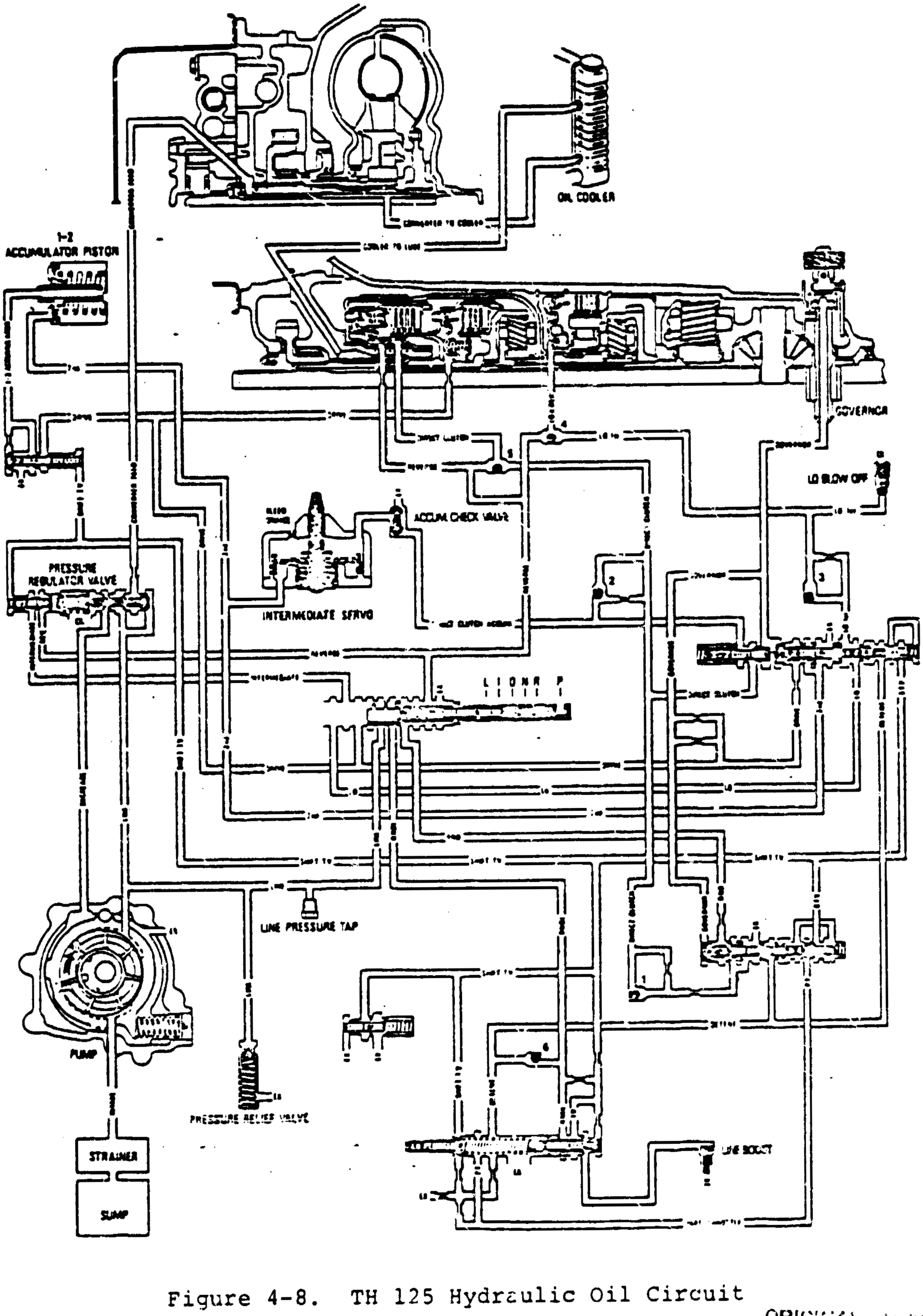


Figure 4-8. TH 125 Hydraulic Oil Circuit

hydraulic control system. At the very least, it must control the application of two clutches and two brakes. Used in different combinations, these four units provide reverse, neutral, and three forward gears. In the NTHV we will add three additional devices described later: the torque converter lock-up clutch, the motor slipping clutch and the engine engaging clutch. The hydraulic system will also provide lubrication for the chain that connects the motor to the transmission and the elements of the motor clutch. And it can be used in normal operation to control the diesel engine fuel flow, because hydraulic pressure is the most available force to move the control lever on the fuel injection pump.

4.2.4.3 Electronic Transmission Control System

As we mentioned above, the normal hydraulic analog control system is limited in sophistication, and, with the addition of the hybrid functions (particularly the selection of the proper power source), the hydraulic system would be completely overpowered. To be able to control both the transmission and the selection of the power plant requires the capabilities of a very sophisticated computer-based system. The system itself will be described in a later section of this report, but the means by which the computer commands are transformed into transmission functions will be discussed here.

The actual operation of the transmission will remain hydraulic, since a complete design change would be required to apply the various transmission clutches and brakes electrically. But then a question arises about the level at which to interface the electric and hydraulic systems. This question has two possible answers: the hydraulic control level and the shift valve level.

To explain the difference between these two levels requires a brief description of the method of operation of a production hydraulic transmission control system. The only inputs to the hydraulic control system shown in Figure 4-8 are the vehicle speed and the accelerator pedal position. These inputs are entered into the transmission control system as the governor pressure, which is related to vehicle speed, and the throttle pressure, which is related to accelerator pedal position. (There is an additional control signal when the kickdown detent is passed; that is, when the accelerator pedal is almost wide open.) These two control pressures act on opposite ends of the shift valves; shift valves have varying diameters and biasing springs, so that the governor pressure loading on the valve will exceed the throttle pressure loading at the speed at which the shift should occur. In essence, the shift valves are the control servos of the shift control system.

Computer signals can control either the inputs to the shift valves, or the direct inputs to the clutches and brakes. If the latter, then the controlling computer will have to modulate the rate of application and release of the individual clutches and brakes. Many of the shifts require the release of one brake or clutch with the application of another. The synchronization of these two elements is critical to a smooth shift. If the first is released too quickly, the motor or engine will run faster than necessary. If the first is released too slowly, there will be an overlap - both elements will be engaged at the same time and the vehicle will bog down. The rates of release and application are not constant, but depend on speed and engine or motor torque. The computer control would have to be quite complete, for it would have to modulate the rate of clutch and brake application and release, and probably do so in conjunction with feedback signals of speeds and/ or torques. On the other hand, if the control inputs are at the shift valve input level, then the computer control signals will only need to control the shift valves on essentially an on-off basis.

While the computer control system is completely capable of controlling the entire shifting sequence, such a development would be of little benefit to the NTHV. It would take a great deal of computer simulation and test bed development for the computer control system to reach the level of shift control that General Motors has already designed into the hydraulic control system. And, since the potential gains in efficiency with computer controlled clutches and brakes is very small, it would not appear to be worth the considerable effort. Rather, it is much better to control the shifts at the shift valve input level and to use the General Motors hydraulic control system to time the shift events.

In an effort to use as much of the original hydraulic system as possible, the standard control system will be used for the Park, Reverse, Neutral, and probably for the Intermediate and Low positions. The computer control will be used only to control the shifts when the transmission is put in the Drive range. The inputs will control the shift valves and the kickdown control for downshifting. The computer will also control the line pressure of the transmission, which normally is a function of the accelerator pedal position. The line pressure is a factor in the control of the shifts and in minimizing the power consumed in driving the transmission pump.

This system could be developed with a minimum of transmission modification, so that the major transmission effort of this program could be devoted to improving the driving efficiency of the NTHV.

4.2.5 Proposed Shifting Characteristics

Once the basic transmission and its control system have been defined, it is necessary to address the shift points. Figure 4-9 shows the shifting characteristic of a typical automatic transmission, plotted in terms of percent throttle and vehicle speed. When this data is converted into terms of driving force at the wheels and vehicle speed, the result is as shown in Figure 4-10. This shift schedule, while adequate for a single power source, is not completely adequate for the NTHV.

Figures 4-11 and 4-12 show the optimum shift points for both up- and down-shifts for the NTHV in the diesel only and electric only modes. In each case the petroleum and electrical energy consumptions were compared in each gear for the various levels of driving force, and the optimum shift points were located where the petroleum or electricity consumption in each gear was lower than in any of the other potentially available gears. These data not only produce different curves for the diesel and electric power sources, they also are totally different from the conventional shift characteristics shown in Figure 4-10. The variation in shift points provides further justification for computer control of the shift valves.

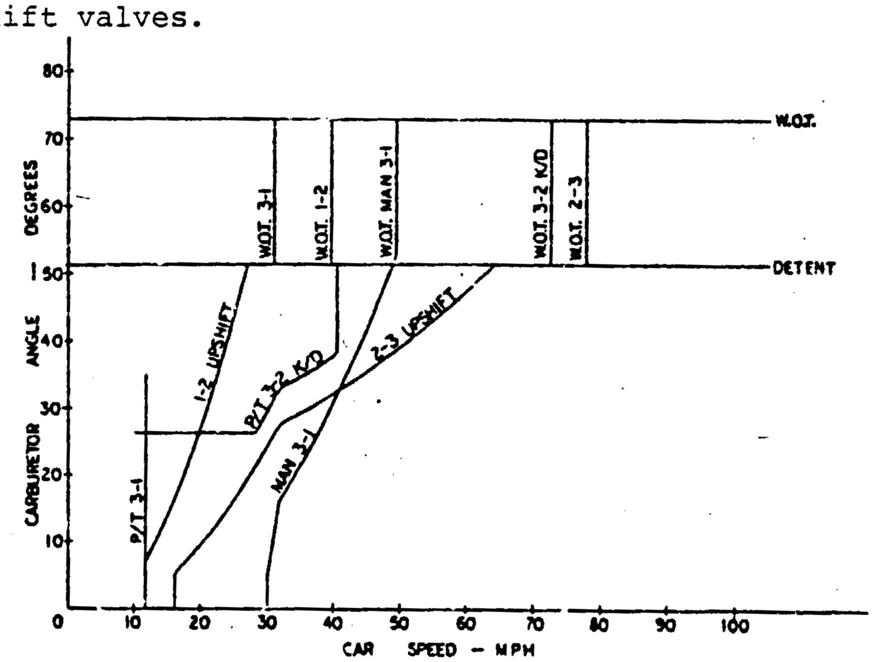


Figure 4-9. Typical Shift Pattern for an Automatic Transmission

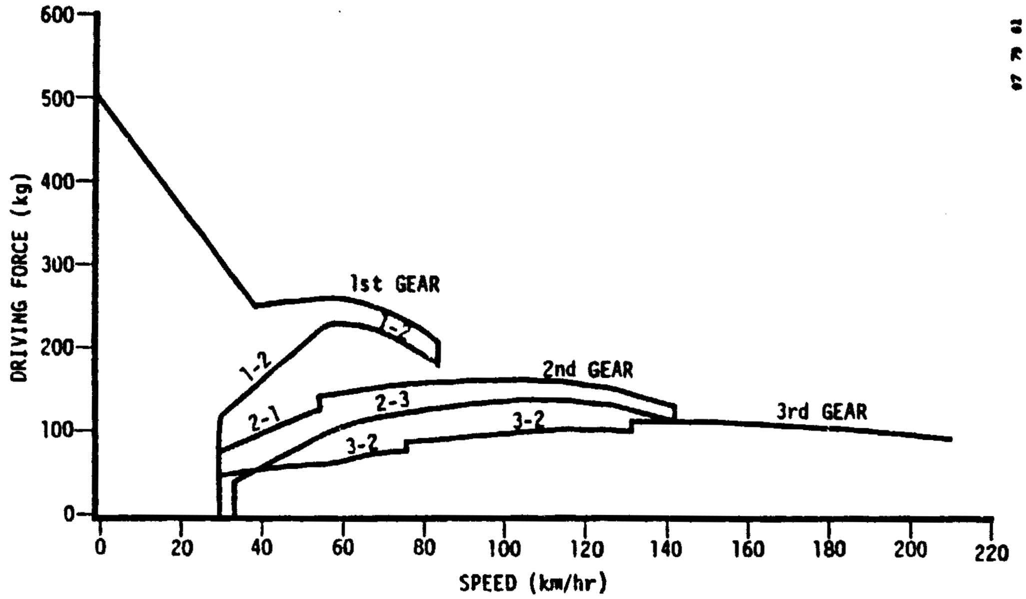


Figure 4-10. Shift Pattern--Typical Automatic Transmission

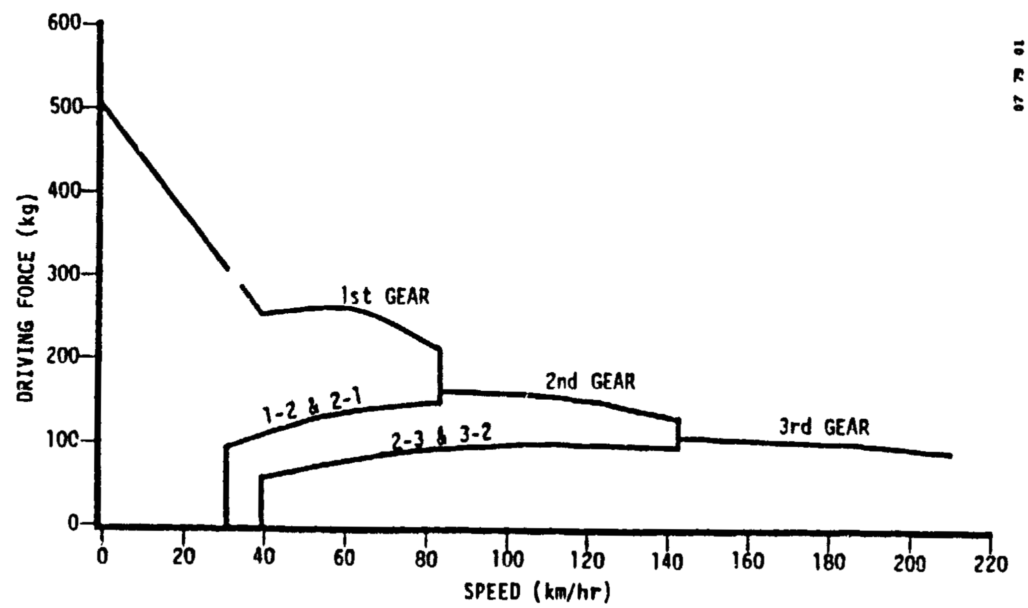


Figure 4-11. Shift Pattern for Best Economy--Diesel Engine Only

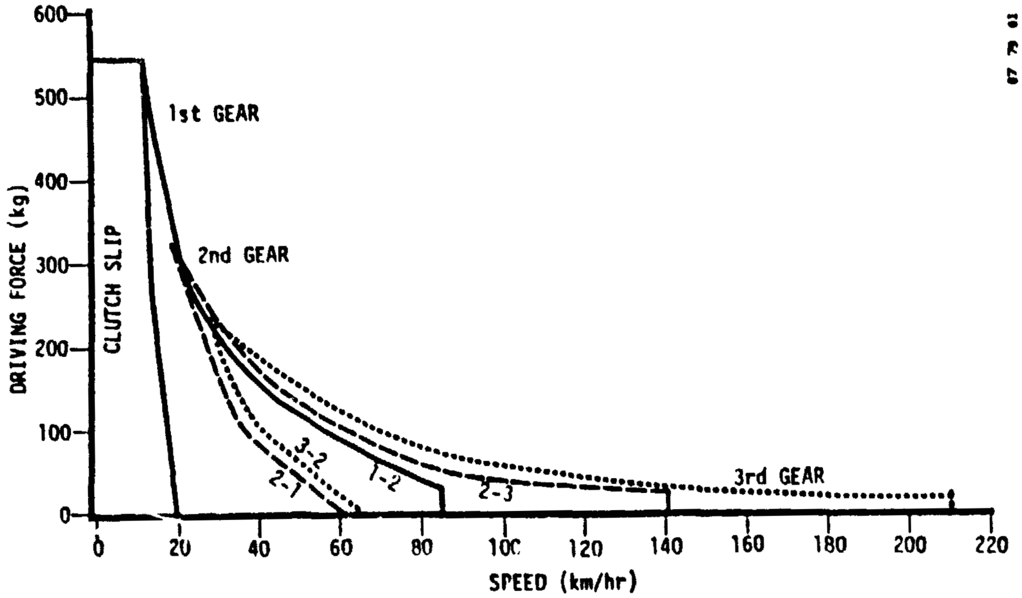


Figure 4-12. Shift Pattern for Best Efficiency—Electric Motor Only

4.3 COUPLING DEVICES

Neither the diesel engine nor the field controlled electric motor is capable of starting a vehicle from rest without some intermediary coupling device. Neither can start from zero speed under load. The diesel engine must be brought up to its idle speed (750-1000 rpm) before any load can be applied. The shunt or compound electric motor with field control alone must be brought to its base speed (1500-2500 rpm) under no load before it is capable of accepting a load. As long as a transmission with finite gear ratios is used, there must be some device that allows either the motor or the engine to run at or above its minimum speed while the vehicle speed is brought up to match the engine/motor speed in the lowest gear. The two most common methods of performing this function are a clutch on a manual transmission and a torque converter on an automatic. Due to their different power output characteristics, the engine and motor put very different requirements on this coupling device, as is discussed below.

4.3.1 Engine to Transmission Coupling[8,16,17,18,19]

In a power shift planetary transmission, a hydrodynamic torque converter is the most common method of coupling the engine to the transmission. A clutch is not normally used for this application, but could be, with either driver or transmission control.

Figure 4-13 plots vehicle speed against the force at the driving wheels produced by a typical engine (at wide open throttle) and

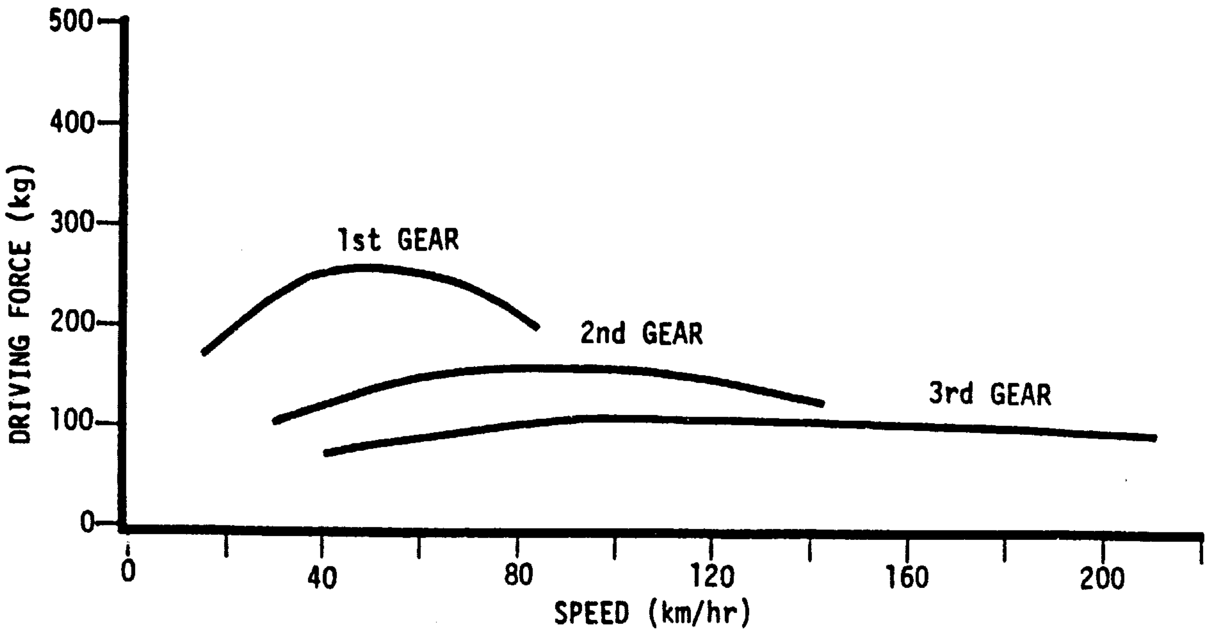


Figure 4-13. Driving Force vs. Vehicle Speed for a Typical Engine and Transmission

three speed transmission. The vehicle speed can only go down to that comparable to the idle speed of the engine in first gear. If the vehicle is slowed below this point, the engine will stall.

The simplest device to allow the vehicle to run at lower speeds is a clutch. Figure 4-14 shows two lines of clutch operation. For line A, the engine speed is kept at its minimum, and the clutch is slipped until the vehicle speed matches the engine speed, Line B shows the available driving force when the engine speed is held at the speed of maximum engine torque; the driving force is higher, and, therefore, the acceleration level is higher. These curves show the conditions for the engine running at wide open throttle, but essentially the same sort of reqults occur at smaller throttle openings. A clutch, being a two element device, must have output torque equal to input torque. Only speed can be varied. The speed ratio of the clutch $\left(\frac{\text{output speed}}{\text{input speed}}\right)$ can vary from 0, when the clutch is completely disengaged with full slip, to 1.0, when the clutch is fully engaged with no slip. The efficiency of the clutch is equal to the speed ratio.

As Figure 4-14 shows, the method of getting the best low speed performance from a vehicle with a clutch is to keep the engine running

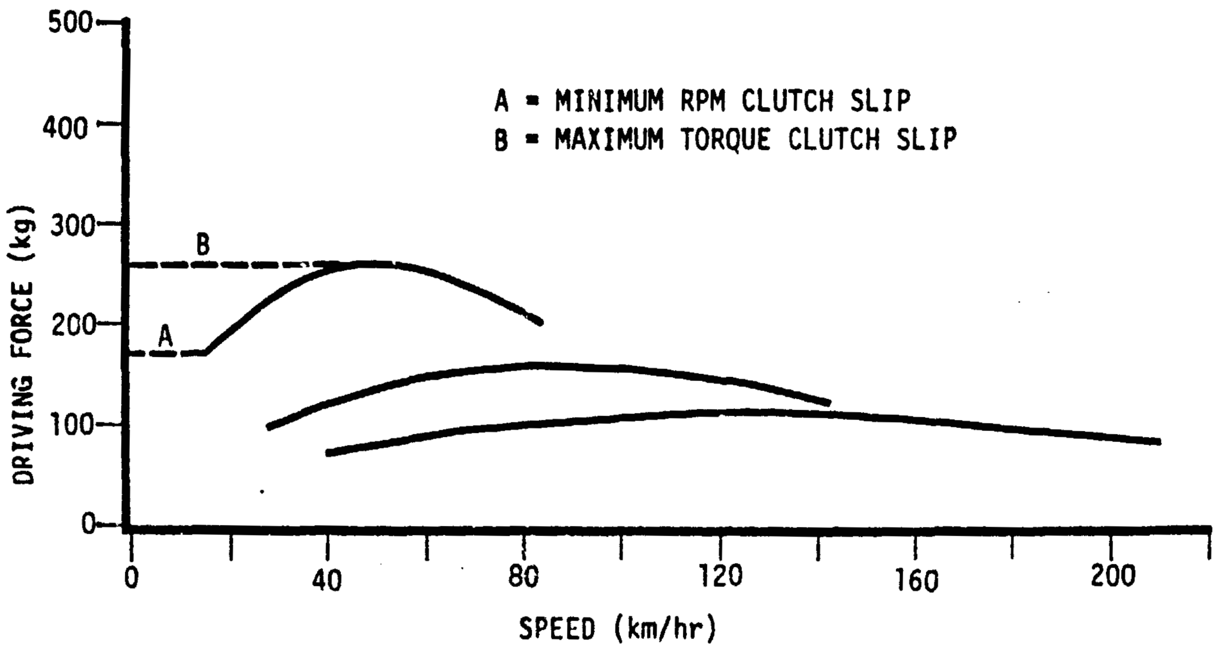


Figure 4-14. Clutch Engagement at Minimum RPM and Maximum Torque

at its maximum torque point and to control the clutch so that the speed is held at that point until the clutch is fully engaged. For more relaxed starts, and at less than wide open throttle, the clutch can be engaged at a lower speed. At very small throttle openings, the clutch can be engaged at the idle speed of the engine. In the typical manual transmission car, the driver modulates the clutch and accelerator pedals to obtain the desired starting acceleration rate. This function can also be performed by means of computer control.

The usual alternative to a clutch as a coupling device between an internal combustion engine and a transmission is a torque converter. A torque converter is a hydrodynamic device that not only couples the engine to the transmission, but also multiplies the torque produced by the engine while the converter is slipping. Figure 4-15 shows the plot of the driving force versus vehicle speed for the same engine and gear ratios as were used for Figures 4-13 and 4-14, but this time with a torque converter between the engine and transmission. The dotted lines represent the driving force from Figure 4-13 (without the torque converter). Figure 4-15 shows that at low speeds in each of the three gears, the torque converter, because of its torque multiplication capabilities, gives a much higher level of driving force.

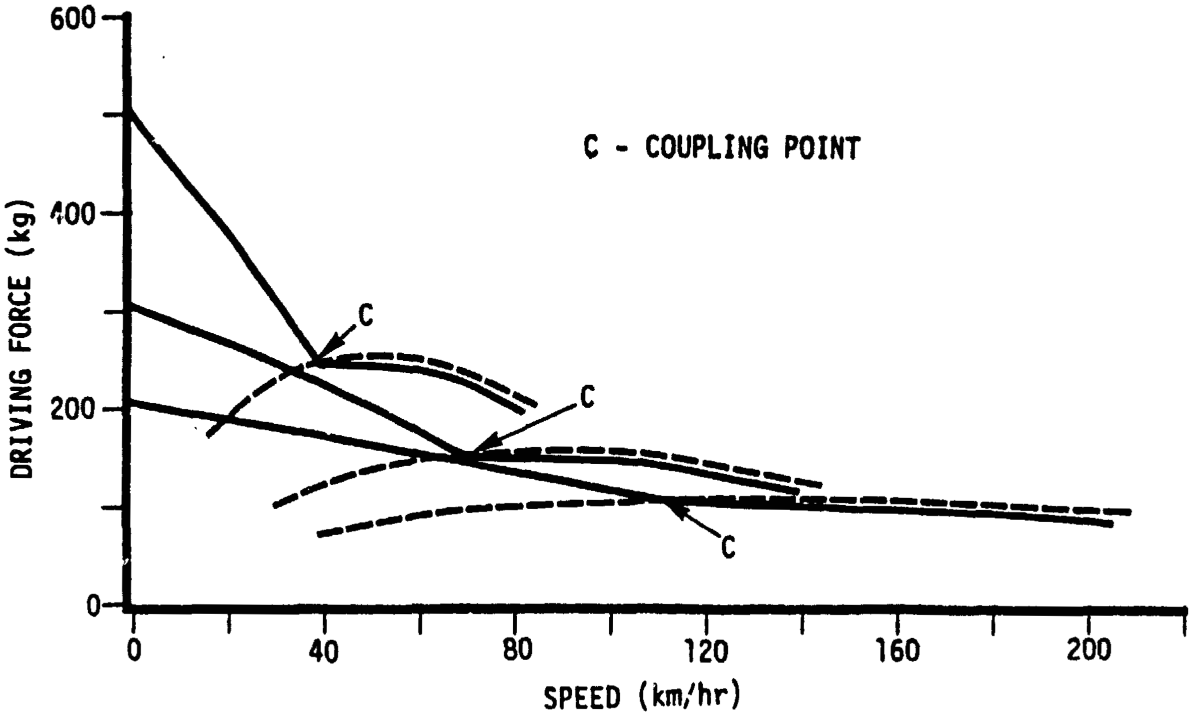


Figure 4-15. Wide Open Throttle Driving Force vs. Speed

At the point marked "C" in each gear, the converter reaches its coupling point, the point at which the torque stops multiplying. Above the coupling point, the torque converter will have some slip (being a hydrodynamic device), and in this portion of the range the driving force will be less than that with the engine driving through a clutch. This loss can be corrected with a torque converter lock-up clutch (which will be described below).

Figure 4-16 is a cross section drawing of a three element hydrodynamic torque converter, the type normally used in passenger cars. The input from the engine drives the impeller, or input member. The impeller acts as a centrifugal pump and imparts a high velocity to the oil that fills the torque converter. This oil then impinges on the turbine, the output member of the torque converter. The high velocity oil gives a driving force to the turbine. The oil leaving the turbine comes to the stator, or reaction member, which is held stationary by an over-running clutch. This clutch allows the stator to rotate forward, but not backward. The stator reverses the direction of the oil and feeds it back into the input of the impeller. Figure 4-17 shows the flow of the oil through the torque converter. Figure 4-17a shows the flow at low speeds, and reveals that the stator, in reversing the flow between

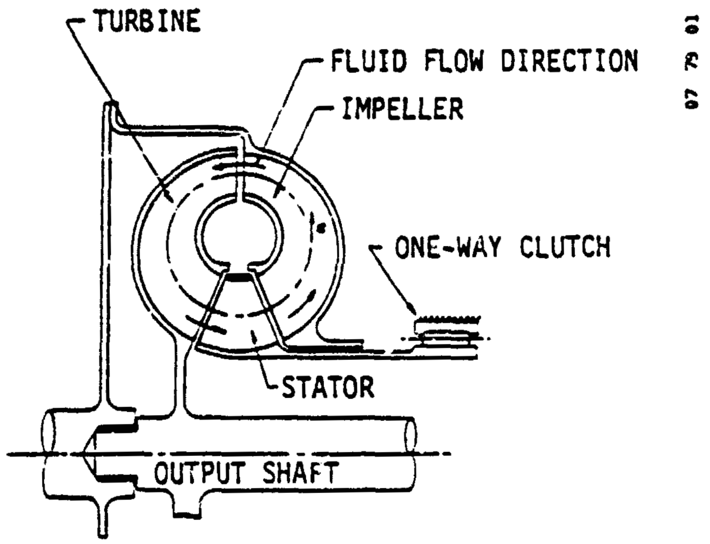


Figure 4-16. Three Element Hydrodynamic Torque Converter

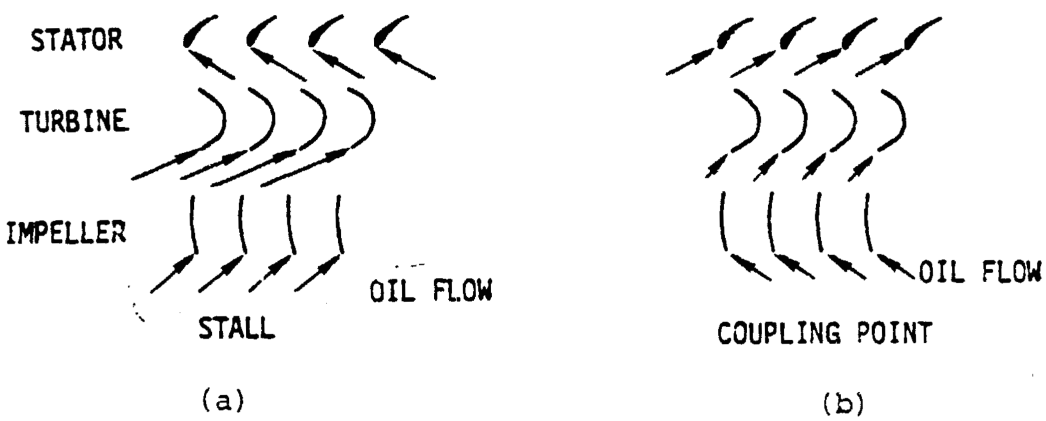


Figure 4-17. Torque Converter Oil Flow

the turbine and the impeller, takes a significant reaction torque. Since output torque must equal input torque plus reaction torque, the output torque is much greater than the input torque—there is torque multiplication. Figure 4-17b shows the condition at higher speeds at which the turbine speed approaches the speed of the impeller. The output vector of the oil from the turbine is now changing direction, and is coming around to the back side of the stator. Since the stator is held by an overrunning clutch, when the oil starts to push on the rear of the stator, the stator will freewheel. Under these conditions there is no reaction torque, so torque in equals torque out. The point at which the stator starts to freewheel is the coupling point.

A torque converter, or any rotating hydrodynamic device, follows the basic equation of $T = CN^2 D^5$, where T = input torque, C is a constant, N is the input speed and D is the diameter of the tip of the impeller. For a given converter, the diameter will be fixed, so the equation may be written as $K = \frac{N}{\sqrt{T}}$. K, in this equation is equal to $\frac{1}{\sqrt{CD^5}}$ and is referred to as the capacity factor, or the "K Factor," of the torque converter. The K factor defines the operating characteristics of a given torque converter. For a given speed ratio $\left(\frac{\text{output speed}}{\text{input speed}}\right)$ of a given torque converter, the converter has a single K factor, and, to operate at the specified speed ratio, the input speed and torque must satisfy the equation $K = \frac{N}{\sqrt{T}}$. Figure 4-18 shows the K factor, torque ratio, and efficiency curves of a typical torque converter.

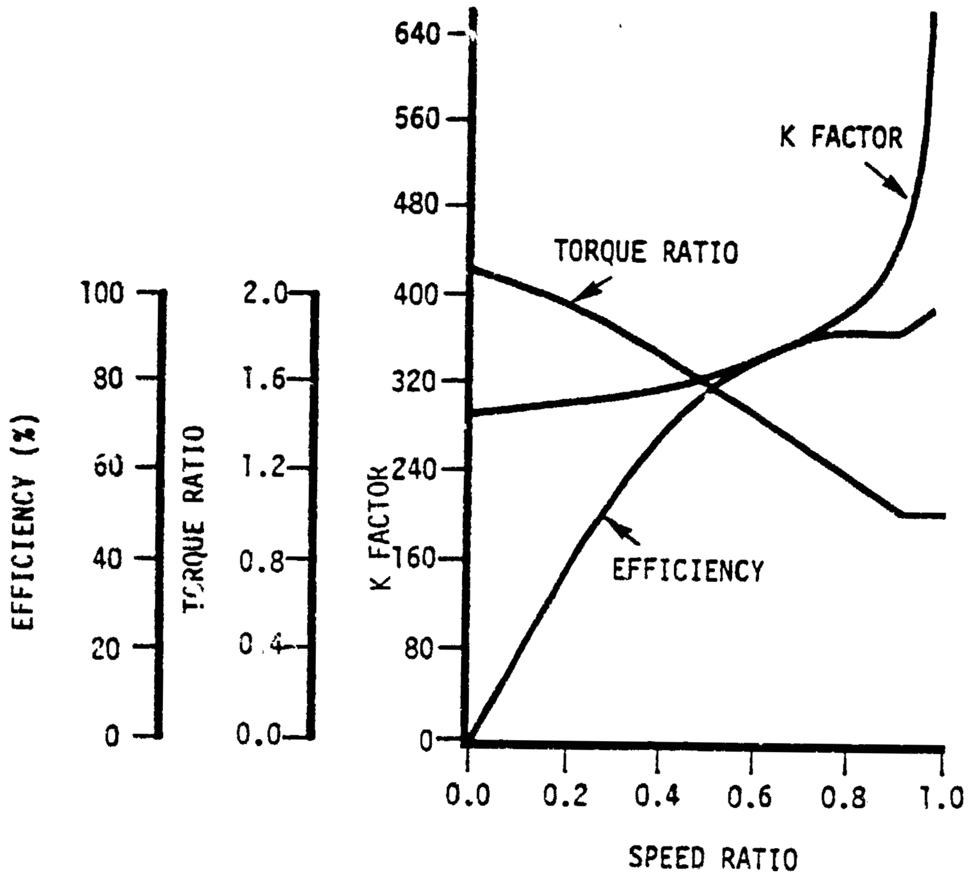


Figure 4-18. Torque Converter Performance Curve

This plot shows the K factor as a function only of speed ratio. For any given speed ratio the converter input torque and speed must follow the K factor equation. The torque ratio $\left(\frac{\text{output torque}}{\text{input torque}}\right)$ is a function only of the speed ratio. The converter efficiency $\left(\frac{\text{output power}}{\text{input power}}\right)$ is the product of the speed ratio and the torque ratio. The inflection point in the torque ratio curve, at the torque ratio of one, is the coupling point of the torque converter. At lower speed ratios the stator is stationary, and the converter is multiplying torque. At higher speed ratios the stator is free-wheeling and there is no torque multiplication. (The torque ratio above the coupling point is usually about 0.99, since the free-wheeling stator has a small amount of drag and prevents a torque ratio of one.)

Figure 4-19 is a plot of the input torque and speeds for different

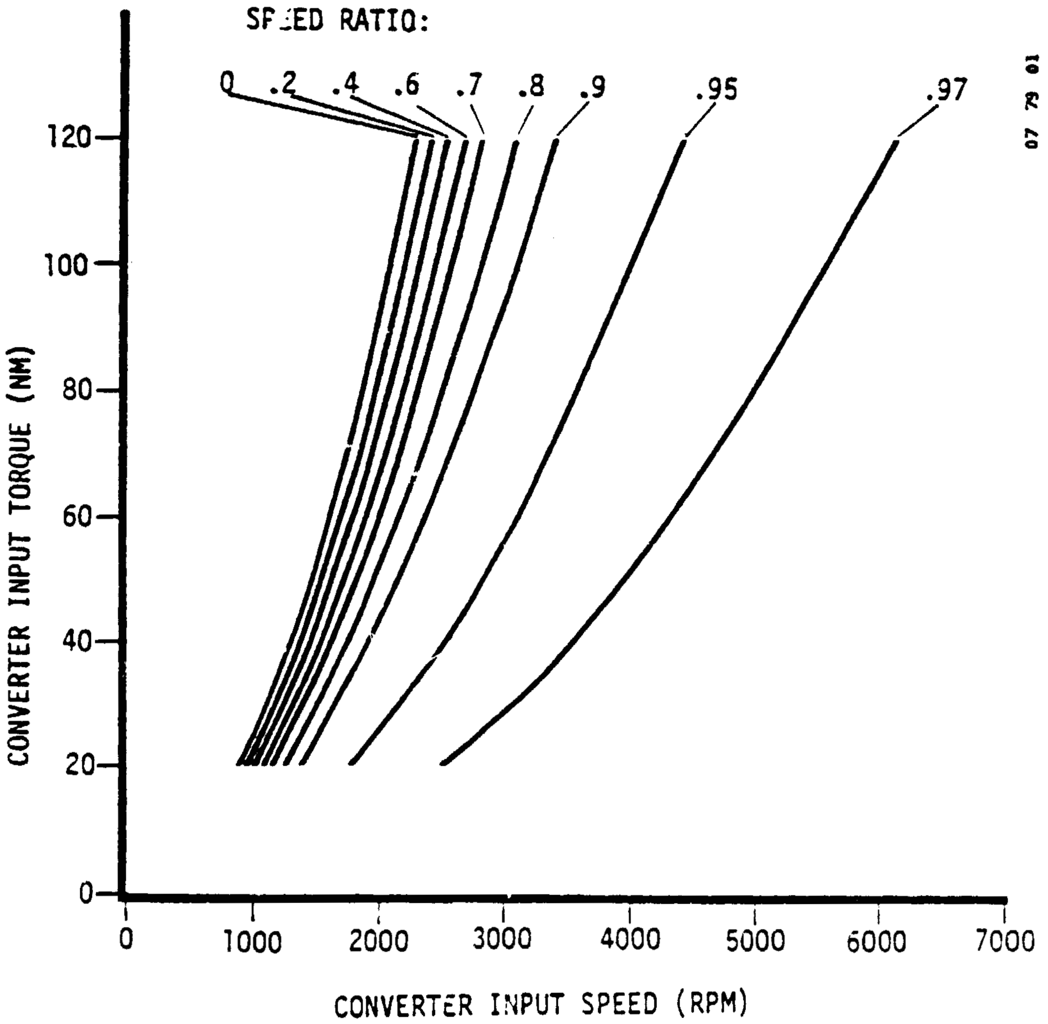


Figure 4-19. Torque Converter Input Speed and Torque For Various Speed Ratios

speed ratios, as called out by the K factor curve in Figure 4-18. For any combination of speed and torque falling on the zero speed ratio line, the converter will give a speed ratio of zero. For points falling on any other speed ratio line, the speed ratio must be that indicated. Since, in almost all cases, the input conditions to the torque converter are the same as the output of the engine, the two conditions can be combined. Figure 4-20 shows the output torque of the engine plotted on the torque converter input characteristic curves. The intersection of the engine torque curve with a line of constant speed ratio is the operating condition of this engine at wide open throttle with this torque converter. The intersection of the torque curve with the zero speed ratio curve is the operating condition of the engine/converter system at zero speed ratio, with zero output speed. The engine cannot run at a steady state condition at a lower speed than this speed, which is referred to as the stall speed.

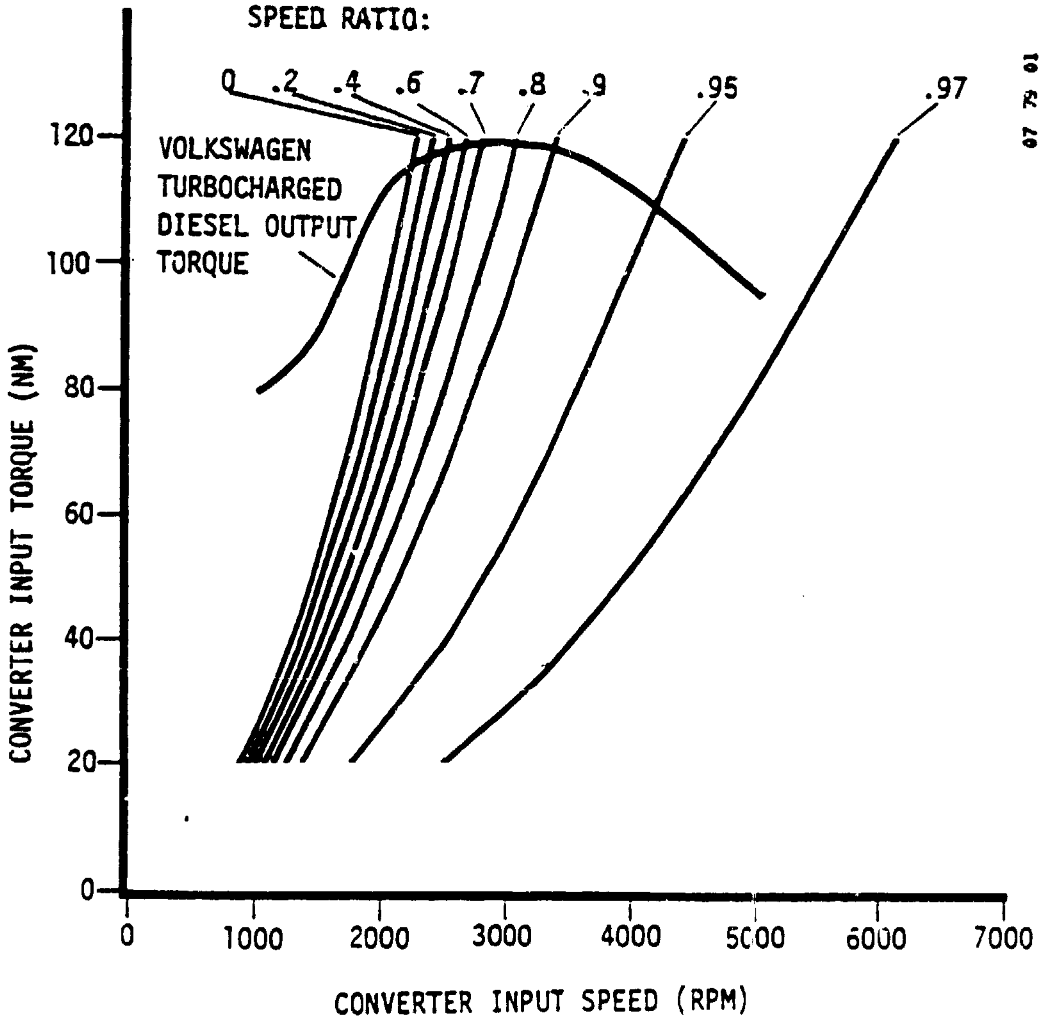


Figure 4-20. Input Speed vs. Torque for the Engine and Torque Converter Combination

Since the intersection points of the constant speed ratio lines and the wide open throttle torque curve represent operating points of the engine/converter system, and since the converter torque ratio is a function only of the speed ratio, it is easy to take these input conditions and calculate the converter output speed and torque. This output condition is plotted in Figure 4-21, which shows the output speed and torque. Since the output speed is zero at zero speed ratio, the torque converter allows the vehicle to start under load while the engine speed remains above idle speed. As the vehicle accelerates, the speed ratio increases, and, above the coupling point, the converter output speed is quite close to the engine speed.

Figures 4-20 and 4-21 show the operating conditions for the torque converter at wide open throttle. Normally however, the vehicle is not driven at wide open throttle for any large portion of the time. When the vehicle is driven at smaller throttle openings, the engine torque for a given engine speed is less, and the operating speeds of the torque converter for a given speed ratio are

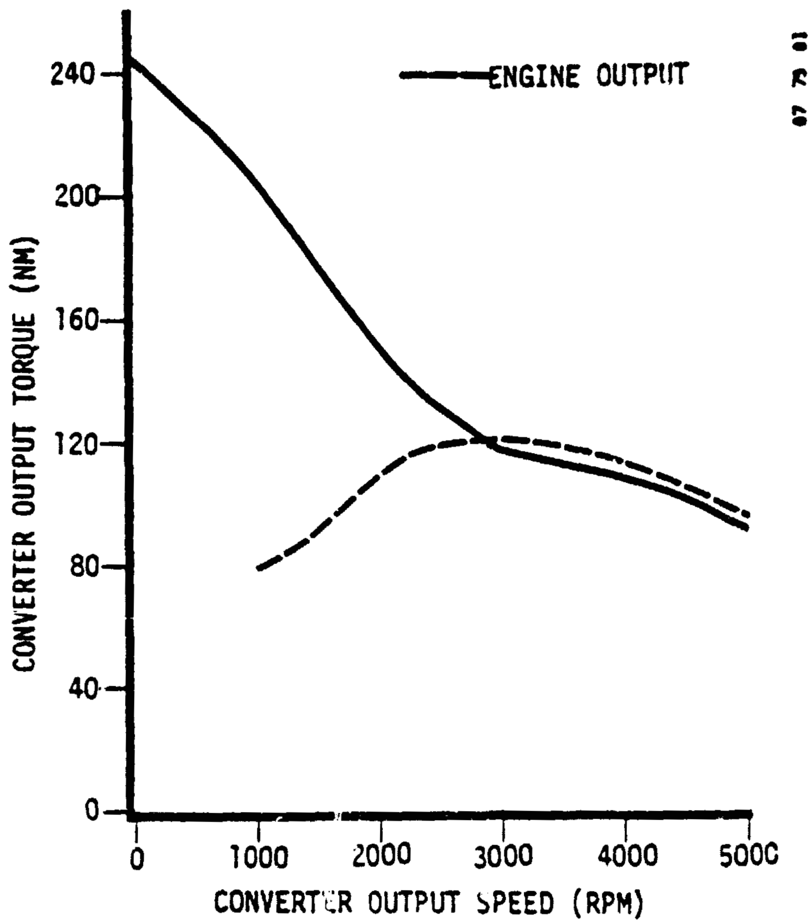


Figure 4-21. Torque Converter Output Torque vs Output Speed

lower. Thus the hydrodynamic torque converter gives the internal combustion engine good start-up characteristics at all throttle openings. The excellent driveability and control which it provides have resulted in the almost universal use of the torque converter in automatic transmissions all over the world.

4.3.2 Matching the Torque Converter to the Engine[8,16,20]

A torque converter characteristic must be selected to go with the Volkswagen turbocharged diesel engine chosen for the NTHV. The K factor of the converter at stall (zero speed ratio) should give a stall speed that falls at a lower engine speed than the engine torque peak. A higher stall speed above the torque peak will give an overall reduction in maximum vehicle acceleration. The minimum acceptable stall speed has usually been found to be in the 1400-1600 rpm range. Any lower stall speeds than this will tend to give a vehicle a very dead feel in low speed acceleration. Between these two limits there are no hard and fast rules for the selection of torque converter characteristics. As the stall speed increases in this range, the acceleration capability of the vehicle from a standing start will improve. This low speed acceleration is extremely important for obtaining good acceleration times (whether to a certain speed or over a given distance). However, the higher the stall speed, generally the greater the slip of the converter under road load conditions. This greater slip causes poorer fuel economy, since more converter slip means lower converter efficiency. With a normal torque converter the selection of the desired stall speed for a vehicle is a trade-off between the desired acceleration level and fuel economy.

An alternative to this tradeoff is beginning to reappear on U.S.-built passenger cars, after an absence of over 20 years. The alternative is a lock-up clutch, built into the torque converter. This clutch, controlled by the transmission shift control mechanism, can be locked at higher speeds, totally eliminating any slip from the torque converter. Chrysler has used lock-up torque converters on some of their cars since 1978, and Ford is expected to introduce this feature on some 1980 models. With a lock-up torque converter the converter stall speed can be set so that acceleration is optimized, and the lock-up feature will eliminate the loss of fuel economy due to converter slip at higher speeds and at lighter loads.

The selection of a particular torque converter for use with the turbocharged VW engine is practically limited to production torque converters that will fit onto that engine, or can be adapted to that engine, and to torque converters with lock-up clutches or with the potential for installation of lock-up clutches. While a new torque converter could be designed and developed for this particular application, this would be an extraordinarily expensive and time consuming project. Probably a number of design iterations would be required to develop the desired torque converter, since converter design is still part art and part science. This expense and time would only be justified if there were no production torque converter available that could be modified to the desired characteristics.

Fortunately, there is an available torque converter that is very well suited to this engine—the Chrysler A-397 torque converter used on the Omni and Horizon. Since this converter is designed for the gasoline VW 1.7 liter engine used in these cars, it also matches the turbocharged 1.5 liter VW diesel, which has approximately the same power output. Figure 4-22 shows the characteristic curve of this converter, and Figure 4-23 the output torque of the converter/engine combination. The stall speed of this converter will be approximately 2400 rpm. The dotted line in Figure 4-23 shows the gain in output from a lock-up clutch that is applied at the coupling point.

Currently, this converter is not being produced with a lock-up clutch, but it has been designed to accommodate one. For our use, we can modify the converter to accept a lock-up clutch which is a smaller version of the one used in the larger Chrysler converters. Figure 4-24 is a drawing of the current production Chrysler lock-up torque converter.

4.3.3 Coupling the Electric Motor to the Transmission[21]

A field controlled electric motor has the same sort of start-up problems as does an internal combustion engine. The motor can not run under load below a certain speed, the base speed, so some type of slipping device must be provided to bring the vehicle speed up to that matching the base speed of the motor. Figure 4-25 shows the electric motor output torque plotted against speed, for the motor proposed for the NTHV. The portion of the curve labeled "full field" is the base speed of the motor. The motor cannot be run below this speed under load. If an armature controller were used, it would be possible to control the motor under

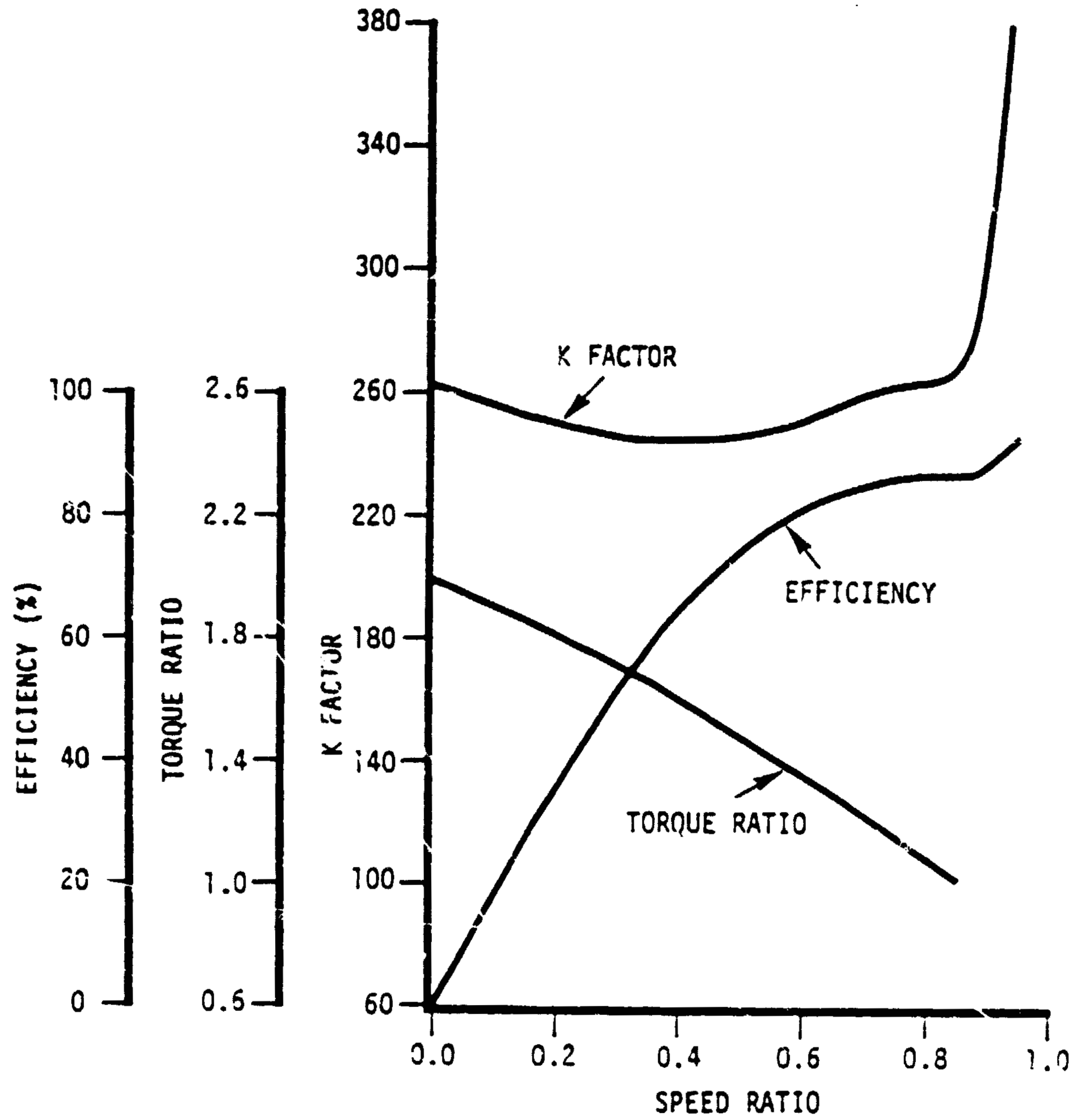


Figure 4-22. Torque Converter Performance, Chrysler A 397

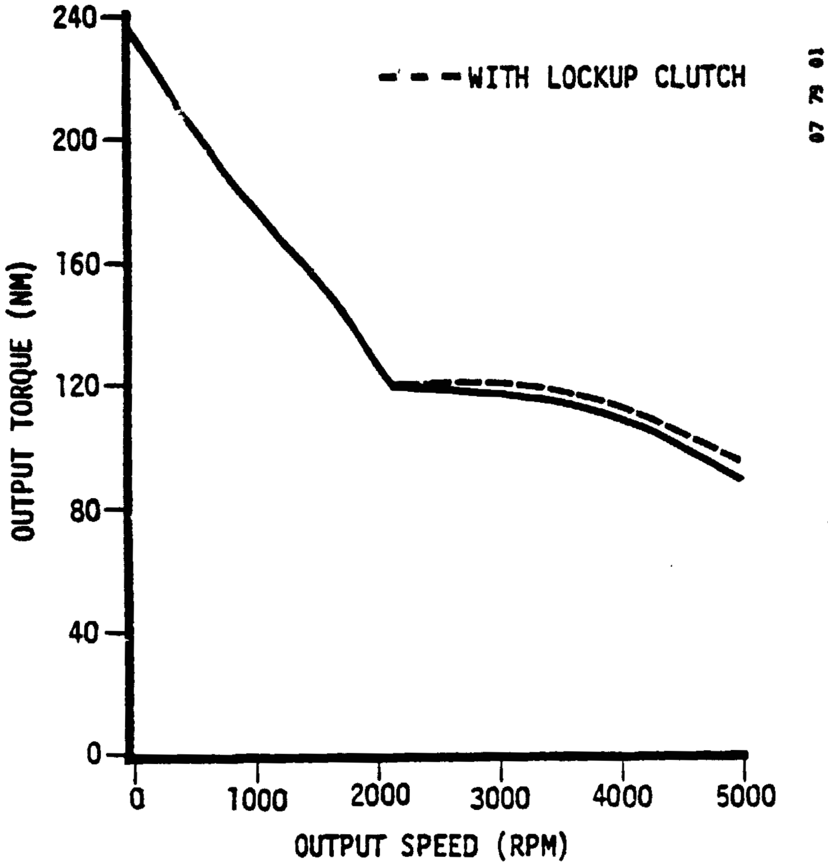


Figure 4-23. Torque Converter Output Torque vs. Output Speed, Turbocharged VW Diesel/Chrysler A-398 Converter

C-2

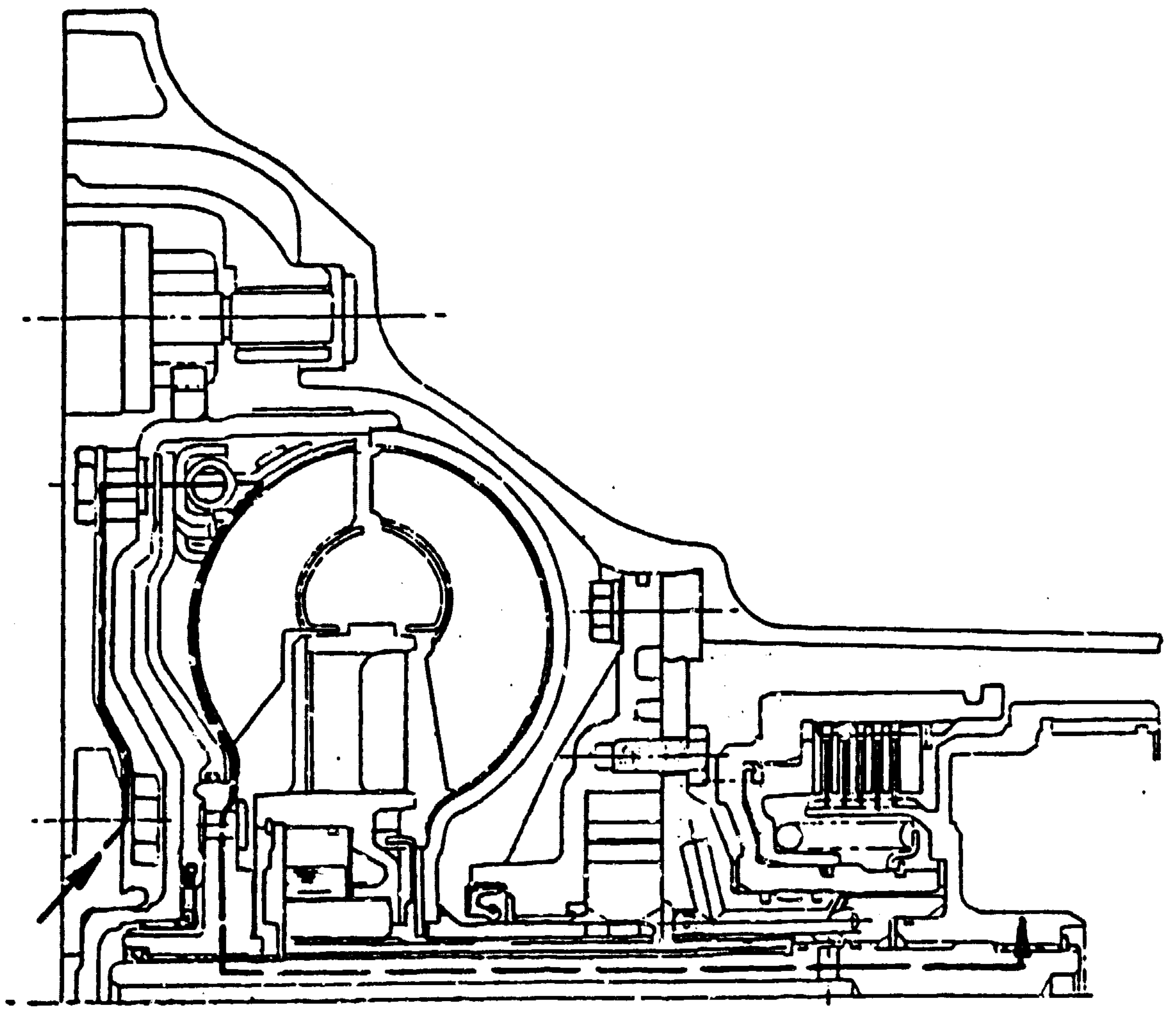


Figure 4-24. Cross Section of Lock-up Torque Converter

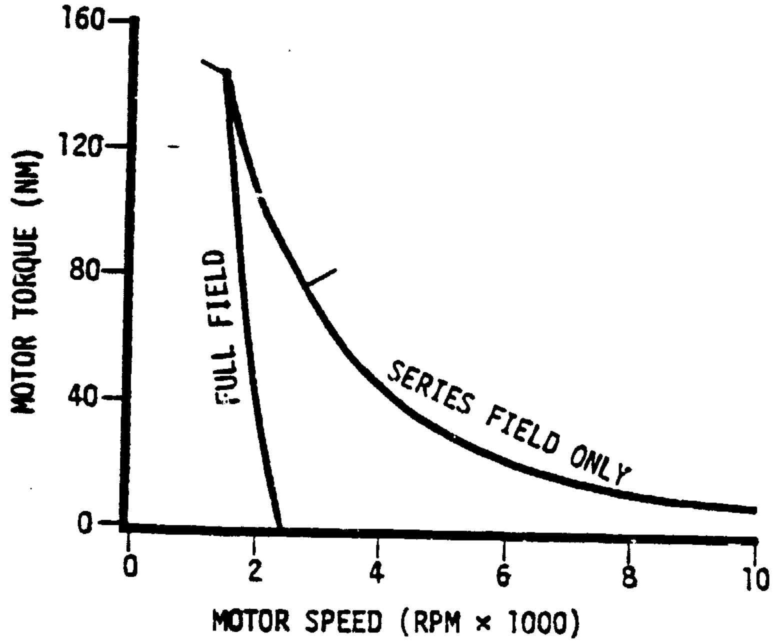


Figure 4-25. Motor Output

load down to zero speed. However, the use of an armature chopper is redundant for a hybrid vehicle which already has a transmission. The transmission can perform the same starting functions as the armature chopper, at much less cost and weight (assuming that the transmission is already in place).

Since a torque converter will be used with the diesel engine, it would be desirable to employ the same torque converter for the electric motor. Unfortunately, this approach is not as easy as it would appear. Figure 4-26 shows the output torque of the electric motor with zero speed ratio K-factor lines for torque converters with three different stall speeds. The converter represented by line A would certainly yield the best acceleration, since it provides a stall speed, and hence a maximum torque ratio, at the maximum torque output of the motor. However, at zero speed ratio this converter would only be capable of running under equilibrium conditions at maximum torque. The maximum torque point is the only

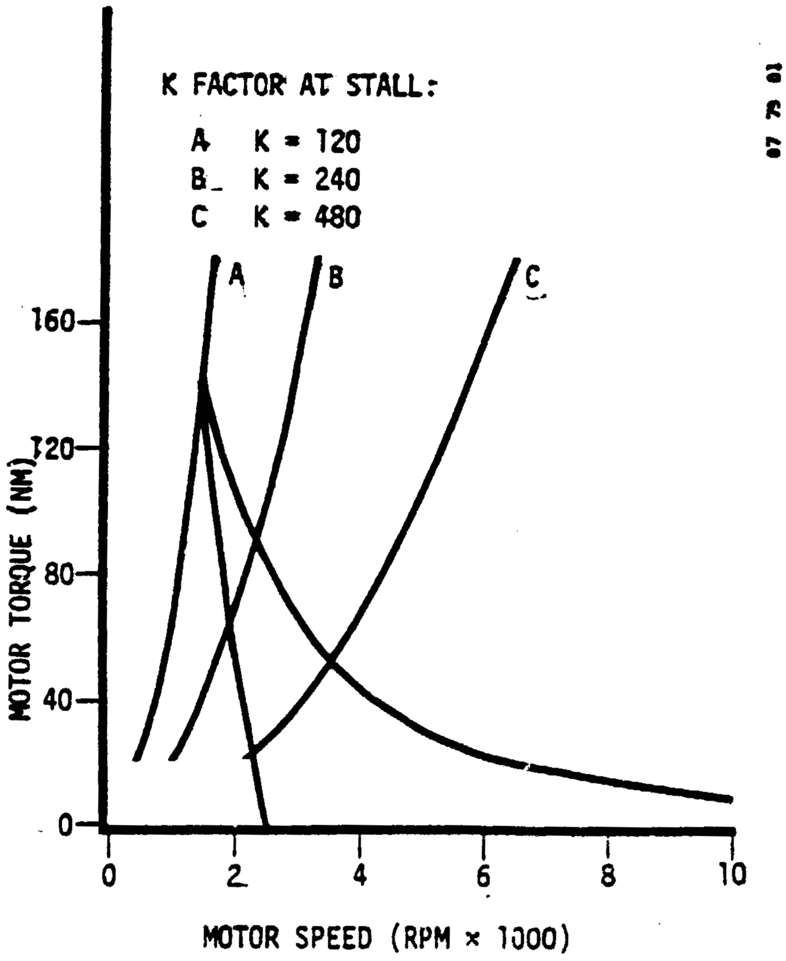


Figure 4-26. Torque Converters for Field Control Electric Motor

intersection between the base speed curve of the motor and the zero speed ratio curve of the torque converter. At a lower torque than the maximum, the motor could not be brought up to base speed in gear, since the base speed is higher than the zero speed ratio speed for that torque. In other words, if the transmission were shifted into gear and the motor torque was less than peak torque, the converter would try to pull the motor down to a speed well below the base speed.

The zero speed ratio lines of two other possible torque converters, labeled "B" and "C", are shown in Figure 4-26. These lines intersect the torque curve in two places, one at base speed and one at a higher speed. The converter can run under steady state conditions anywhere between the two lines. Figures 4-27, 4-28 and 4-29 show the converter output torque range that would be possible from the three converters proposed in Figure 4-26. The shaded portions of these plots are the ranges in which the motor/converter combination can operate. The upper edge of the area is equivalent to the wide open throttle limit that would be expected from an internal combustion engine and torque converter, such as shown in Figure 4-21. The lower edge of the area is the minimum output level at which the motor/converter combination can be run. This has no direct equivalent in an internal combusion engine. The motor/converter combination is not capable of providing an output torque/speed combination that falls to the left and below the shaded portion. The lower limit of the shaded portion is the lowest level of converter output that is possible from these specific components.

Converter A provides the best output torque, and therefore the best acceleration potential, but its minimum possible converter output is almost the same as its maximum output at low vehicle speeds. This is obviously not a practical type of control for the NTHV, or for any electric vehicle with field control only. Figures 4-28 and 4-29 show the maximum and minimum output torques of the converters marked "B" and "C" in Figure 4-26. While these converters allow a wider latitude of range of motor output, the minimum converter output is still at a substantial level, and the maximum torque is less than that of the converter in Figure 4-27.

Figures 4-27, 4-28 and 4-29 show why a torque converter is not a suitable coupling device between an electric motor with field control alone and a transmission. Depending on the converter selected, the motor/converter combination will only be capable of running at high torque outputs, or the maximum torque and power output of

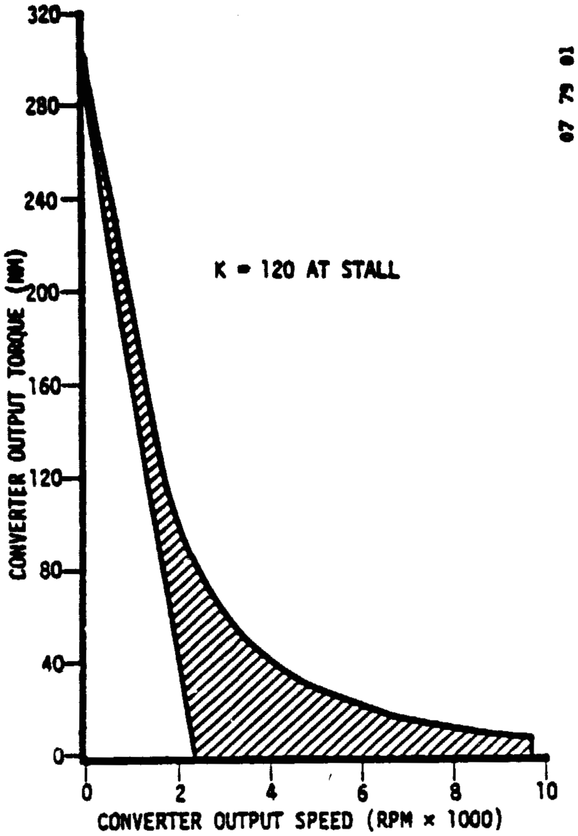


Figure 4-27. Range of Converter Output Field Control Motor with Torque Converter A

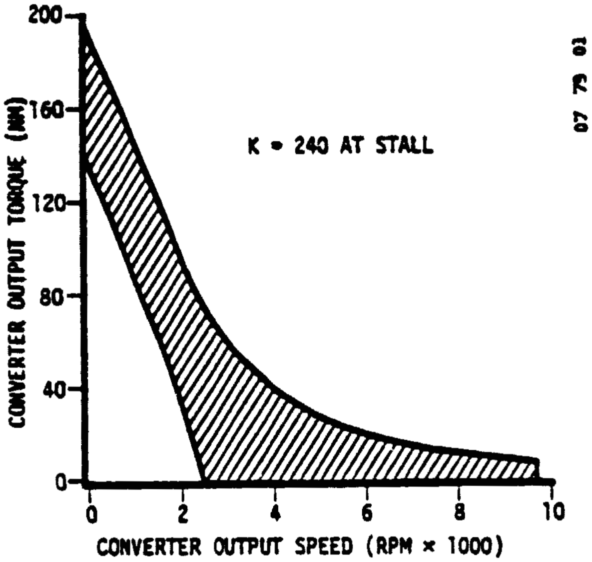


Figure 4-28. Range of Converter Output Field Control Motor with Torque Converter B

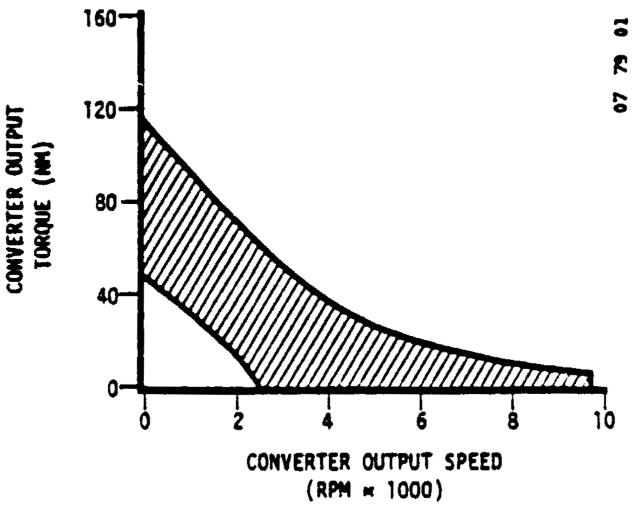


Figure 4-29. Range of Converter Output Field Control Motor with Torque Converter C

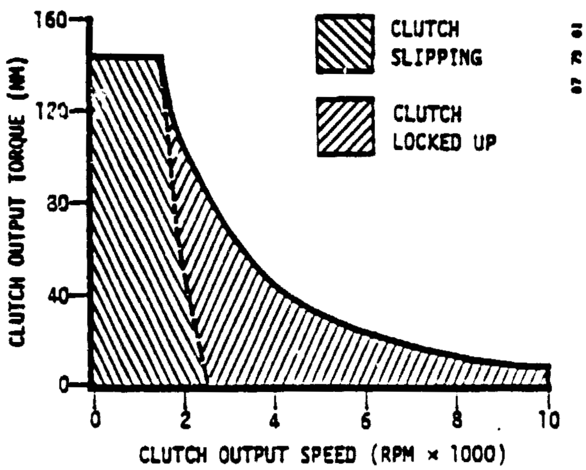


Figure 4-30. Range of Clutch Output Field Control Motor with Clutch

the motor will not be available for use. There are two devices that can be used to couple the field control electric motor to the transmission - a slipping clutch and a variable fill fluid coupling.

4.3.3.1 Slipping Clutch [15]

A clutch is in essence a device that transmits torque at speed ratios between 0 and 1.0, with the output torque always equal to the input torque. As such, a clutch is a suitable device to use to connect a field controlled electric motor to a transmission. Figure 4-30 shows the output torque of the electric motor, which is also the input torque to the clutch. By varying the clutch capacity, any desired level of input torque to the transmission can be maintained. Since clutch output torque is equal to clutch input torque, Figure 4-30 is also a plot of clutch output torque. Thus, it is comparable to Figures 4-27, 4-28 and 4-29, which show torque converter output torque. Since the clutch does not multiply torque like the torque converter, at low speeds the output torque with the torque converter (matched to the maximum torque of the motor, Figure 4-27) is higher than the output torque with the clutch. However, since the condition of the converter is one that does not allow transmitting torque at lower output levels, the higher output is not really usable. Over the entire speed range of the motor the other two torque converters provide generally equal or less output torque than the clutch. The clutch is a more suitable device than the torque converter for connecting the field controlled electric motor to the transmission.

The capacity of a clutch is defined as:

$$C = F \cdot R \cdot \mu \cdot N \quad ,$$

where

C = Clutch torque capacity
F = Normal force acting on cluth faces
R = Effective radius of clutch lining
μ = Coefficient of friction
N = Number of friction faces in clutch

Since the radius and number of friction faces in a specific clutch are constant, the clutch's torque capacity is a function of the coefficient of friction of the forces and the normal force applied to them. If the coefficient of friction of the lining is constant, then the clutch capacity varies directly with the normal force.

For our purposes there are basically two kinds of friction clutches: clutches whose normal force is provided by springs which, when compressed, release the clutch, and clutches which are held open by springs and are applied by an external force. The typical manual transmission clutch is an example of the first type, while the clutches used in an automatic transmission are examples of the second.

The spring applied clutch needs a mechanical or hydraulic release system. A releasing force must be supplied any time the motor is disengaged, and, to prevent accidental engagement for the motor during start up procedures, it would be necessary to provide another disengaging device (such as a neutral gear) to be sure that the motor was not engaged before sufficient hydraulic pressure had been generated in the control system to disengage the clutch. Also, a spring applied clutch will have some hysteresis losses between the clutch release pressure vs. clutch capacity curve and the clutch application curve. This can be reduced to a small difference only by installing an exceptionally low friction linkage or by making the clutch release hydraulic piston act directly on the springs and pressure plate. Both of these tend to make the design more complex.

The spring released clutch uses an external force to engage the clutch. This clutch is not particularly suitable for direct control from a pedal, since the driver would have to push on the clutch pedal to apply the clutch. But it is very suitable for hydraulic application from a pressure supply, since the pressure piston can be built directly into the clutch and can act directly on the clutch pressure plate. This almost eliminates any hysterisis differences between the application and release pressures. The spring release clutch also has the advantage of being disengaged when the entire system is turned off, and will not engage until high pressure oil is routed to the pressure piston.

The spring release, hydraulically applied clutch is definitely the best clutch for coupling the field controlled motor to the transmission. The application pressure can be modulated by the control system to provide exactly the clutch capacity needed for any given purpose.

In order to obtain sufficient capacity in a small diameter, the clutch will have to be a multi-plate design. To minimize clutch drag when the clutch is disengaged, the clutch should have springs between the different plates to keep the friction surfaces separated.

Since the clutch capacity is a function of the coefficient of friction of the material used in the clutch, and since this coefficient will frequently vary with temperature, it is essential to keep this type of clutch cooled when it is slipping. This is best accomplished by flooding the clutch plates with large quantities of oil while the clutch is actually slipping.

4.3.3.2 Variable Fill Fluid Coupling

A fluid coupling is a hydrodynamic device very similar to a torque converter, but with only an input member (the impeller) and an output member (the turbine). Since there is no reaction member, the fluid coupling can not multiply, so its input and output torques are equal (the same as a clutch). Figure 4-31 shows a typical fluid coupling. The fluid coupling obeys the same basic hydraulic laws as a torque converter, and can be described by the same curves. Figure 4-32 shows the characteristic curves for a typical fluid coupling. The input K factor curve is very similar to a torque converter, and represents the same $K=(N/\sqrt{T})$ input condition that is constant for a given speed ratio. The torque ratio of the fluid coupling is 1.0, by definition, and, because the efficiency equals the torque ratio times the speed ratio, the efficiency is equal to the speed ratio. The torque ratio and efficiency curves are the same for any coupling, so they are frequently not plotted for a fluid coupling. Since the fluid coupling has the same input speed vs. torque characteristics as a torque converter, on the surface it has no advantage for coupling a field controlled electric motor to the transmission. What does make it practical is varying the amount of oil in the coupling.

A fluid coupling is normally considered to be completely filled with oil, and its characteristics are measured that way. However, if the coupling is filled with less than the the full amount of oil, its characteristics will change. Centrifugal force will cause the oil to be thrown to the outside diameter of the coupling, and the inside diameter of the oil will increase. This will yield a smaller cross section for oil flow between the two elements, and will essentially raise the K factor curve. Figure 4-33 shows the effects of various fill levels on the zero speed ratio line for coupling input. By varying the amount of the coupling fill, any desired level of coupling input can be achieved at stall. It is, in essence, a variable stall speed device. Since the variable fill coupling can match the input torque exactly, the coupling output curve will be essentially the same as that for the slipping clutch shown in Figure 4-30.

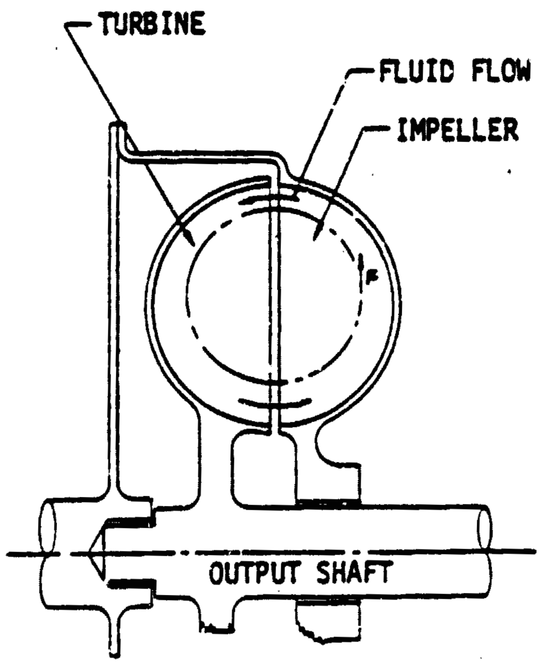


Figure 4-31. Fluid Coupling

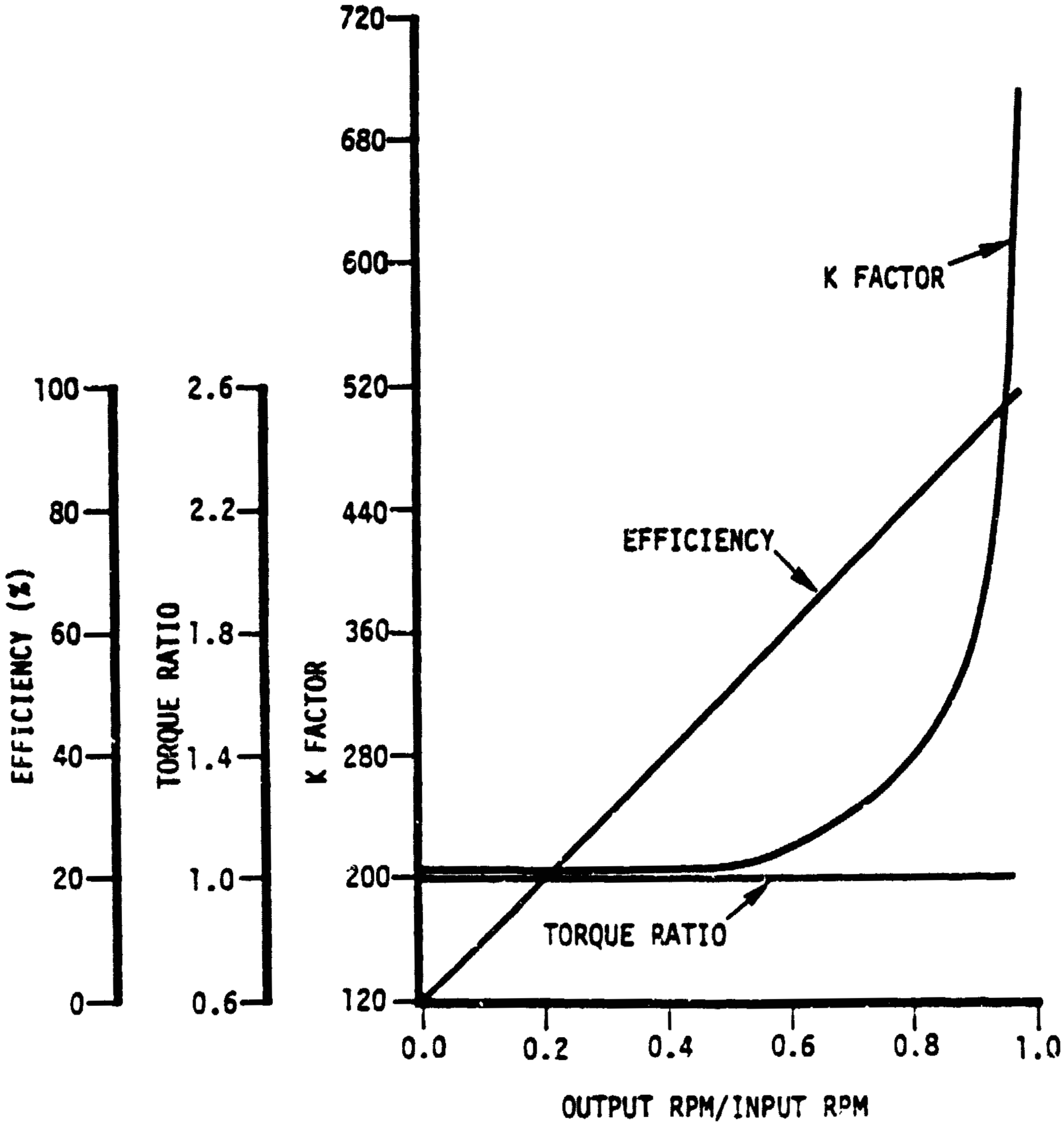


Figure 4-32. Fluid Coupling Performance Curve

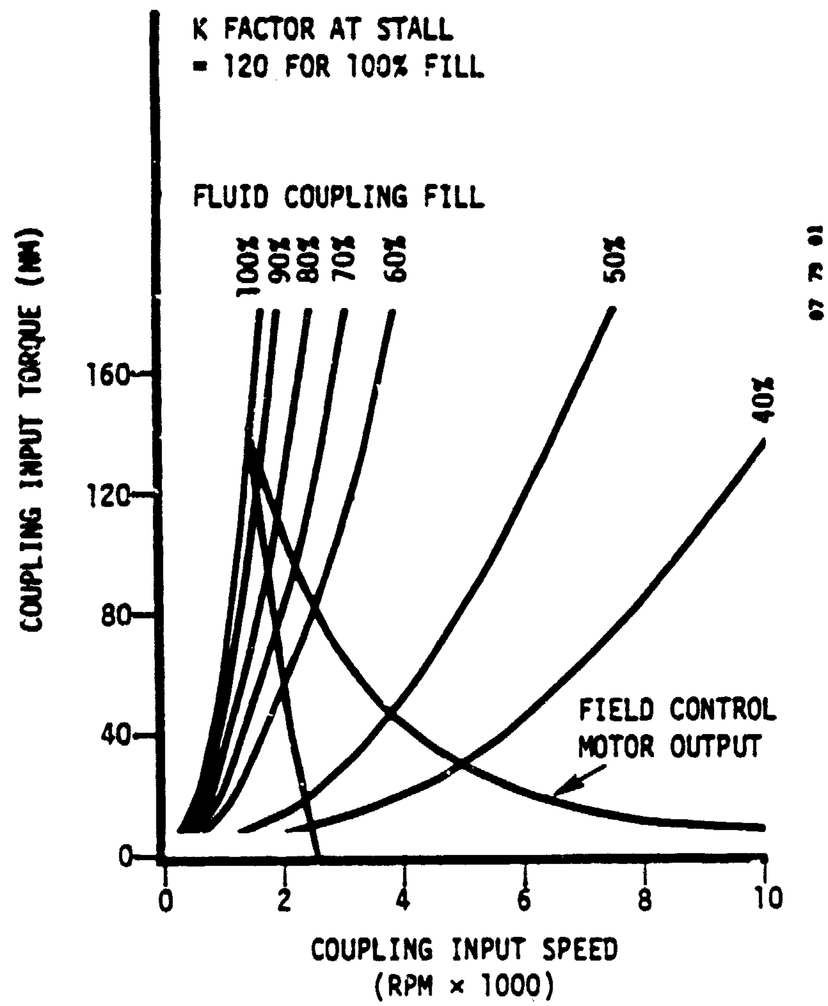


Figure 4-33. Effect of Fluid Coupling Fill On Zero Speed Ratio Inputs

The obvious question is: if the variable fill principle will work on a fluid coupling, why not use it on a torque converter and have the torque multiplication as well? The torque converter has an inner or torus ring (Figure 4-34), which is necessary to assure good oil flow over the three elements of the converter. This torus ring is also used in some fluid couplings, but cannot be used in a variable fill coupling. In a torque converter, if the oil fill were reduced from 100 percent to about 75 percent, the torus ring would severely interfere with the internal flow of the oil, and would give very unpredictable variations in K factor. With less than 100 percent fill, the stator would be less and less in the oil path; thus its ability to multiply torque would be reduced.

The variable fill fluid coupling is not a new concept, although varying the fill to obtain a desired set of characteristics may be. Fluid couplings are frequently used to couple induction motors to loads to reduce the starting load on the motor, and it is a common practice to vary the amount of oil that is put in the coupling to match the desired starting characteristics of the motor for the particular load. In this case, the variation in fill is performed by physically changing the amount of oil in the coupling. A different use of the same principle was made by General Motors on the Controlled Coupling Hydramatic transmission introduced in 1956. In this transmission (Figure 4-35) two fluid couplings were used: a large coupling (that was always full) for starting, and a small coupling to smooth the gearshifts. The small coupling was empty in first and third gears and was full in second and fourth gears. The coupling was filled during the 1-2 shift and the 3-4 shift to give very smooth shift. This eliminated the rough shifts that had been a problem with earlier Hydramatic transmissions. In this case the small coupling was filled over a 1 to 2 second period, and was emptied over a shorter period. The partial fill conditions were transient, rather than steady state conditions, but the variation of the K factor during the fill process was the key to smooth shifting.

For the NTHV, the variable fill coupling would be a combination of these two procedures. The variations of the fill of the coupling would be controlled in a manner similar to that used by General Motors, but the coupling would be able to run under either steady state conditions or transient conditions. The amount of oil both entering and leaving the coupling would be controlled by the powerplant control system, so that any desired slip characteristic could be achieved. The oil entering the coupling would come from the

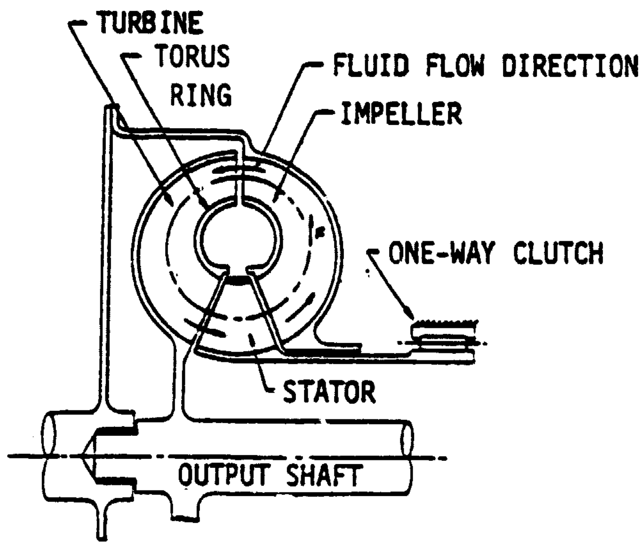


Figure 4-34. Torque Converter

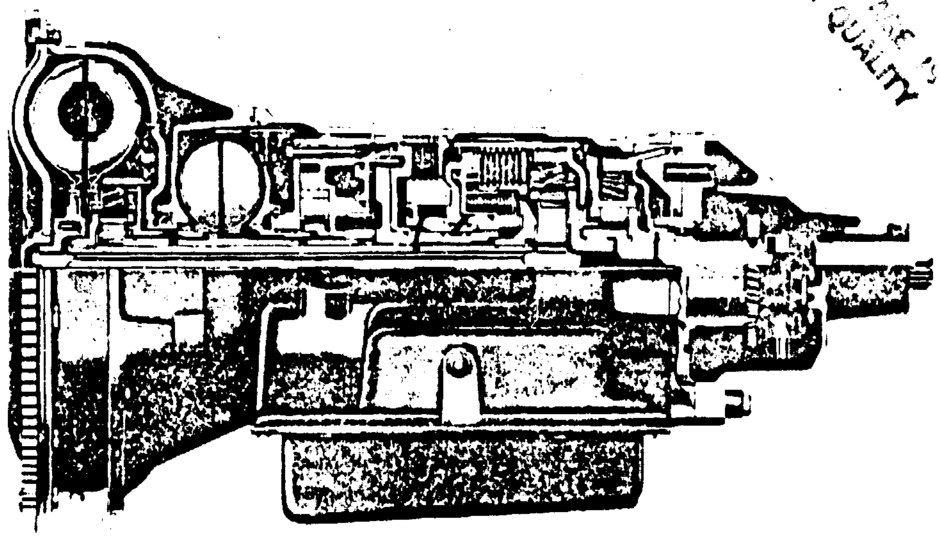


Figure 4-35. 1956 Twin Coupling Hydramatic

high pressure oil supply of the transmission into the center of the coupling through a controllable valve. The excess oil would be exhausted by means of one or more controllable valves on the outside diameter of the coupling. Continual oil flow would be required through the coupling when it is under load, to remove the heat generated by the coupling slip.

Once the coupling has brought the vehicle speed up to match the motor speed, its only function is to connect the motor to the transmission as efficiently as possible. The coupling will have some slip at all times (like a torque converter), although a coupling can be designed for lower slip than a converter. So, for the motor it would probably be desirable to build a lock-up clutch into the variable fill fluid coupling.

4.3.3.3 Variable Fill Coupling vs. Slipping Clutch

The slipping clutch and the variable fill fluid coupling have the same output torque vs. speed characteristics (Figure 4-30), so there is little choice between them with respect to vehicle performance. Their differences are in the areas of size, weight, reliability, consistency, and ease of control.

The variable fill coupling must be larger than the slipping clutch. With the proposed motor, the coupling would require an outer diameter of approximately 20 cm to have a stall speed matching the torque peak of the motor. The diameter of the clutch is less critical, since a clutch of any diameter could be used, with the number of plates varying to provide the required capacity. For this application a clutch having a 12 to 15 cm outer diameter with four driven plates would have more than sufficient capacity for the electric motor, and would have sufficient friction area to be able to dissipate the heat during clutch slip.

The weight of the two different units would be in the same general range as the dimensions; the variable fill coupling would be heavier due mainly to its greater size.

We expect the reliability of the variable fill coupling to be very high. Being an oil filled device, it should have no wear problems, except for the lock-up clutch. It should have the same reliability as a torque converter in a production automatic transmission, and these units rarely fail from converter related problems. Converter failures are usually caused by the loss of oil flow due to some other failure in the transmission.

The reliability of the slipping clutch would be lower, since it is a friction device. However, the reliability should still be quite high, as long as the clutch friction area is sufficient to prevent localized heating during clutch slip. This clutch should at least have the 100,000-150,000 km life typical of multiplate clutches in automatic transmissions.

The consistency of the variable fill coupling will probably be better than that of the clutch. The coupling performance is dependent only on the amount and density of the oil in the coupling. The density will change with temperature, but as long as the flow of oil is sufficient, this should not be an important factor. The clutch performance depends on the clutch application force and on the coefficient of friction of the clutch lining. The clutch application pressure can be controlled accurately, but the coefficient of friction of the clutch lining will vary with temperature, rubbing velocity, and the age of the material. The consistency of the clutch can be kept at an acceptable level with adequate inputs to the control system. The clutch application pressure will have to be varied, for a given torque capacity, with the variations in the coefficient of friction of the clutch lining. But if clutch slip is checked regularly during the start-up process, the clutch application pressure can be controlled to give the desired clutch performance.

Both the variable fill coupling and the slipping clutch have potential problems in ease of control. The fill required for the coupling will be known for a desired torque capacity, but it will not be a simple matter to control this fill. The actual amount of oil in the coupling does not lend itself to easy measurement, so the performance of the coupling will have to be measured to see if the fill is correct. The coupling input and output speeds will need to be measured and the motor torque calculated from motor speed, armature current and field current. Once these are known, it will be possible to calculate whether the coupling is operating at the correct fill level or not and to vary the flow into and out of the coupling to move toward the desired performance. The oil flow through the converter must, however, be maintained at a sufficient level to assure that the oil is not overheated by coupling the slip. With a computer control system and the correct software, this control problem will be soluble.

The slipping clutch will have a control problem similar to that of the variable fill coupling. The clutch will have variations in torque capacity for changes in temperature and in friction material conditions. The clutch control problem differs from that of the coupling only in that there is one control function, clutch application pressure, rather than two, flow in and out of the coupling. Both of the control systems can be developed to a satisfactory level with detailed computer simulations and test bed development.

There is no overriding reason to choose either the slipping clutch or the variable fill fluid coupling for coupling the field control motor to the transmission. However, the slipping clutch would have to be preferred, since it is smaller and lighter, and there is more known about clutch control (which would give a higher degree of assurance that the system would work as required). Thus the slipping clutch is the selected device for the preliminary design, although both devices will be carried forward in the development program until bench tests indicate a clear preference.

4.4 ACCESSORY DRIVE

Several different accessories will be driven by the motor or engine in the NTHV, and the method of driving these accessories must be determined.

4.4.1 Required Accessories

The NTHV requires a power steering pump, engine water pump, alternator, brake vacuum pump and air conditioning compressor. All of these accessories will be necessary at one time or another, when either the motor or the engine are running.

The power steering pump is needed to provide the oil pressure for the power steering. With the weight of the vehicle, the weight bias with front wheel drive, and the public expectations for this class of car in the U.S. market, power steering will be expected by the driver of this vehicle. The power steering pump will be required whenever the vehicle is moving, under either electric or diesel power.

The engine water pump performs more functions in the NTHV than just circulating the engine coolant through the engine and the radiator. The water pump is the basis of the vehicle and battery heating systems, and, as such, must be run even when the vehicle is being driven by the electric motor and the diesel engine is turned off. Under these conditions the heat source for the vehicle and battery heating system is a combustion heater, using diesel fuel. This heater transfers the heat to the coolant, which is in turn circulated through the vehicle and battery heater systems. These systems are described in more detail in Section 8.

The alternator is needed to provide power for the 12 volt systems of the vehicle and to keep the separate 12 volt battery for starting and lighting fully charged. The alternator is required when the diesel engine is running, as is the case with all ICE powered vehicles, to provide the necessary power. When the vehicle is running under electric power, the accessories and lights can be powered from a dc-dc converter that reduces the 72 volt traction battery voltage to 12 volts, or they can be powered by an alternator which returns the energy to electricity. The converter requires an additional electrical system, and cannot supplant the alternator, for the alternator will always be required with the diesel engine. The 12 volt battery will be charged when the traction batteries are charged, and so will be fully charged in the first part of the daily usage of the vehicle. This is the portion of the day when the electric drive system will be used the most, so the actual draw from the alternator, when driven by the motor, should be low. Overall, the expected loss in efficiency of driving the alternator from the electric motor is very small.

The brake vacuum pump is needed for the standard vacuum brake booster system of the X-body vehicle. Since neither the diesel engine nor the electric motor produce vacuum, it is necessary to produce the vacuum with a pump. This is standard practice for current diesel engine passenger cars. Volkswagen uses a rotary vacuum pump on the Dasher, and Oldsmobile uses a diaphragm vacuum pump on their diesel. Either type of pump could be used on the NTHV, but, whichever is used, it must be driven by both the diesel engine or the electric motor, whichever is powering the vehicle.

A vehicle of this size and price class will be expected to have an air conditioning system, as was the case with 80 percent of all American built cars sold in 1978. The normal method of cooling a vehicle is by means of an engine powered air conditioning compressor. There are alternate methods of cooling that do not require a compressor, but none of these systems are sufficiently developed, and consequently would not fit the near term requirements of this program.

The compressor could be driven by either the engine, or the motor, whichever is running at the time. The power to drive the air conditioning compressor is too high for motor supply to be practical, since the compressor takes from 20 to 40 percent of the total available motor power output. As a result, it will be necessary for the compressor to be powered only by the diesel engine, and for the engine to be running, at least at idle, when the compressor is required.

4.4.2 Method of Driving Accessories

Since the accessories, with the exception of the air conditioning compressor, must be run whenever the vehicle is moving, they must be capable of being driven when either the engine or the motor is powering the vehicle, and they must be run at their necessary minimum speed when the vehicle is moving at very low speeds. This means that we need a drive that will use either engine or motor power.

The method of driving the accessories used in the preliminary design is shown in Figure 4-36. There will be an accessory drive shaft on the extension of the motor centerline. All of the accessories, except the air conditioner, will be driven from this drive shaft. The shaft will be powered directly by the motor, or by a belt from the engine. The drive from the engine will have a 2:1 ratio to match the 2:1 ratio in motor to engine speeds. Both the drive from the motor and the accessory shaft pulley from the engine will be powered through overrunning (freewheel) clutches, so that the shaft will be driven by whichever powerplant is running.

Belts from this accessory drive shaft will drive the water pump, the alternator, the power steering pump and the vacuum pump. The air conditioning compressor will be driven from the engine crankshaft by a separate belt.

A potential variation on this design would be to use a variable speed belt drive from the engine to the accessory drive shaft, so that the accessory speeds would be kept more nearly constant when the vehicle was being driven by the engine. A variable speed drive from the motor would add more complication and is of somewhat less importance, since the variation of motor speed when the vehicle is being driven in the electric mode will normally be less than 2:1, compared to the 3:1 to 5:1 range experienced with the internal combustion engine.

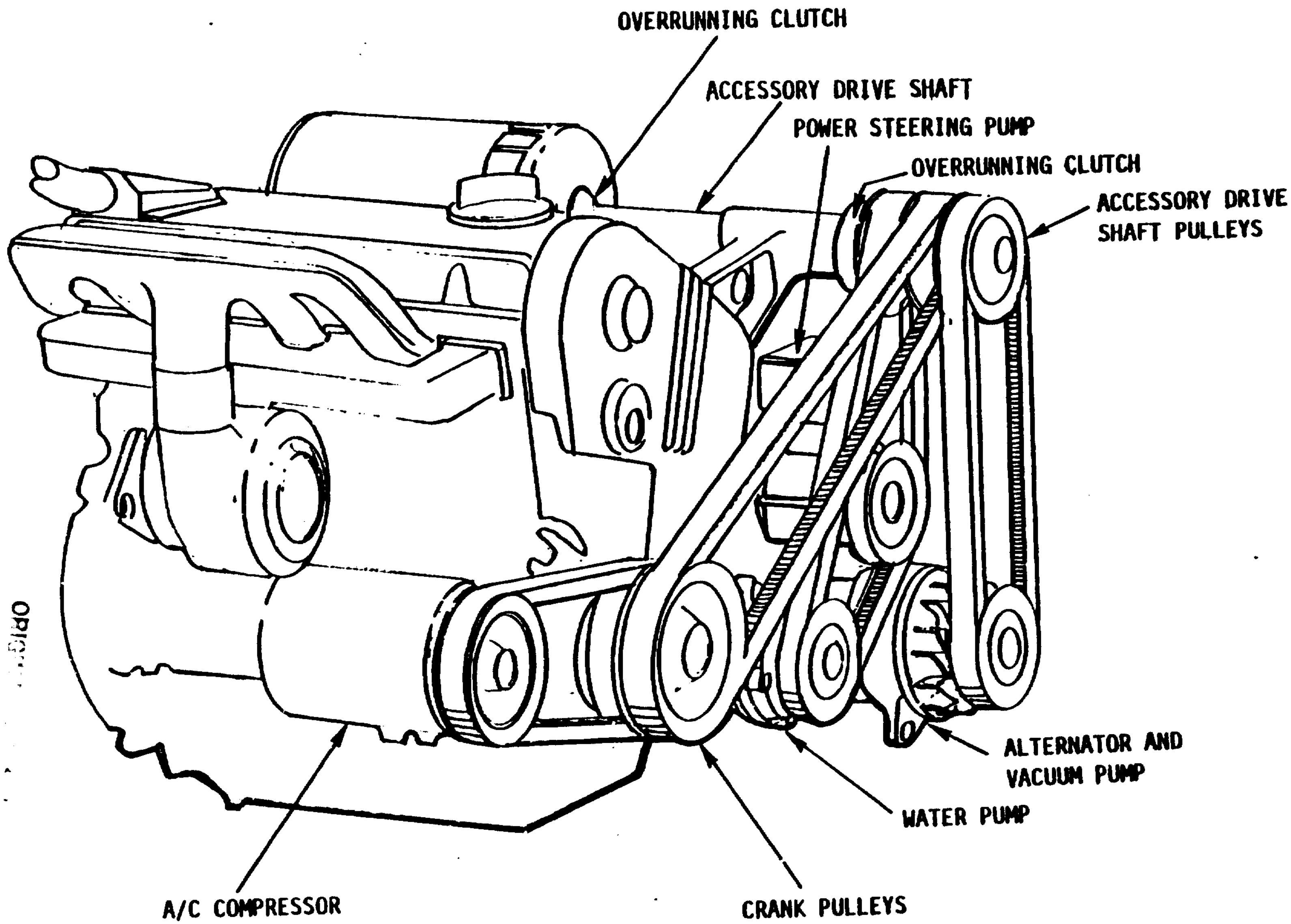


Figure 4-36. Accessory Drive Concept

4.5 POWERTRAIN INTEGRATION

After the selection of the components of the powertrain, it is necessary to package these components into a compact unit that will fit into the vehicle, and that will have the most efficient layout for vehicle operation and servicing.

Figures 4-37, 4-38 and 4-39 are three views of the preliminary design drivetrain package, as it will be installed in the NTHV. The package consists of the turbocharged Volkswagen Rabbit diesel engine, a 24 kW compound motor, and a modified Turbo-Hydramatic 125 transmission. The transmission is mounted in the standard position for a General Motors X-body car. This simplifies mountings and allows the use of standard X-body half-shafts and universal joints. The torque converter portion of the transmission case is modified to allow room for the drive from the motor and the clutch that connects the drive to the engine. This requires the torque converter to be located about 10 cm closer to the right side of the car than would be the case in a standard X-body, since the Volkswagen diesel is much shorter than the four cylinder 2.5 liter General Motors engine. There is still room for the engine to fit within the standard X-body frame rails.

The motor is connected to the transmission by a chain drive, with a 2:1 drive ratio to allow for the higher speed potential of the motor compared to the engine. The motor is actually mounted ahead of the engine and is well above the crankshaft centerline. In this position the motor is located in a free area of the engine compartment, but still remains close to the engine for ease in tying the motor and engine together in the powertrain.

Other possible arrangements of the different components could be considered, but the basic simplicity of mounting the transmission in its original position in the car, and mounting the engine and torque converter in approximately their original positions, is a major factor in selecting this layout. Only the motor location remains free under these restraints, and the position forward and above the engine is an obvious choice.

4.6 DIESEL ENGINE[23]

The selection of the turbocharged Volkswagen 1.5 liter diesel engine for the NTHV is explained in the Design Trade-off Studies Report.[1] One area not covered in that report was the emissions level of the NTHV.

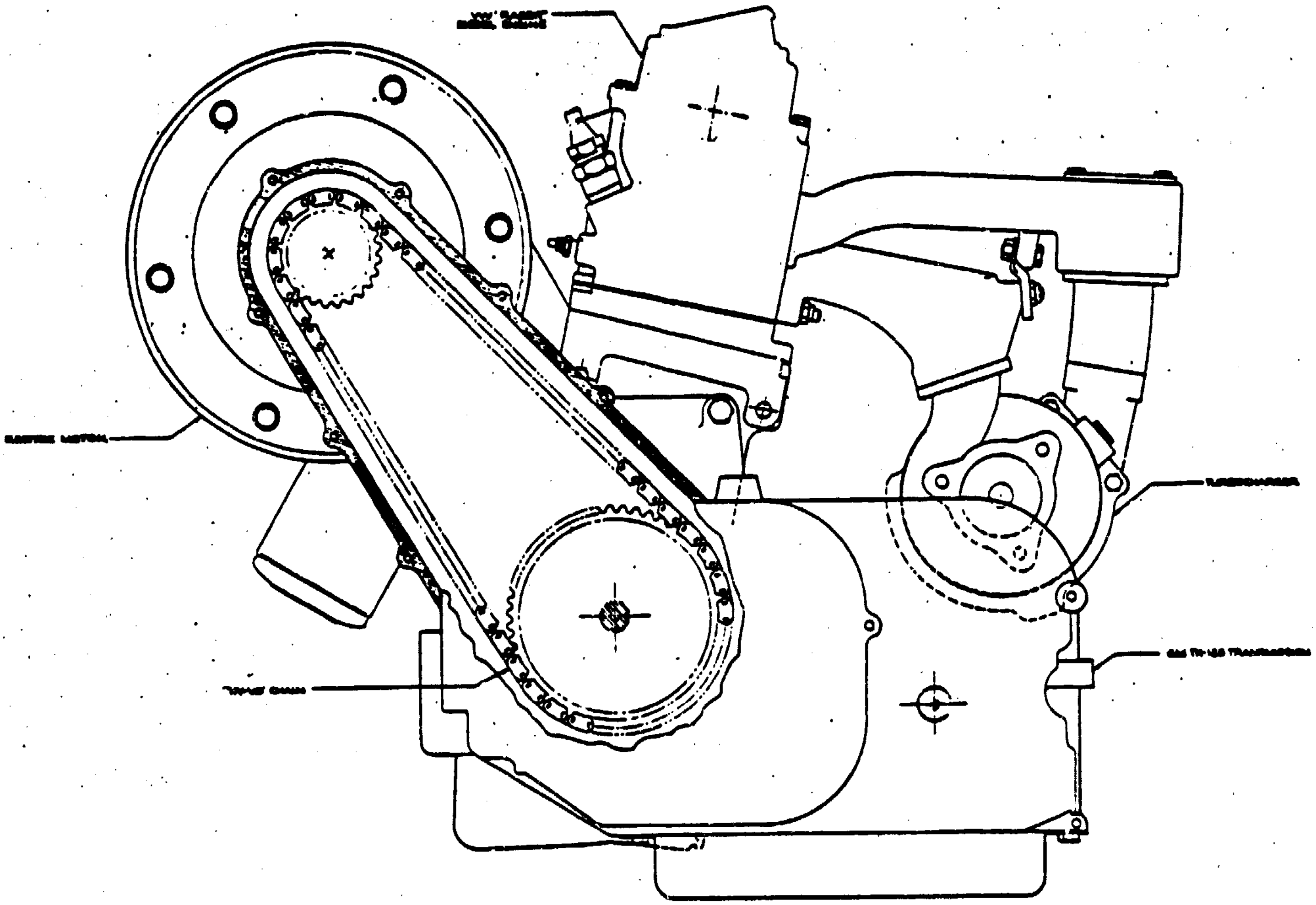


Figure 4-37. Drivetrain Package

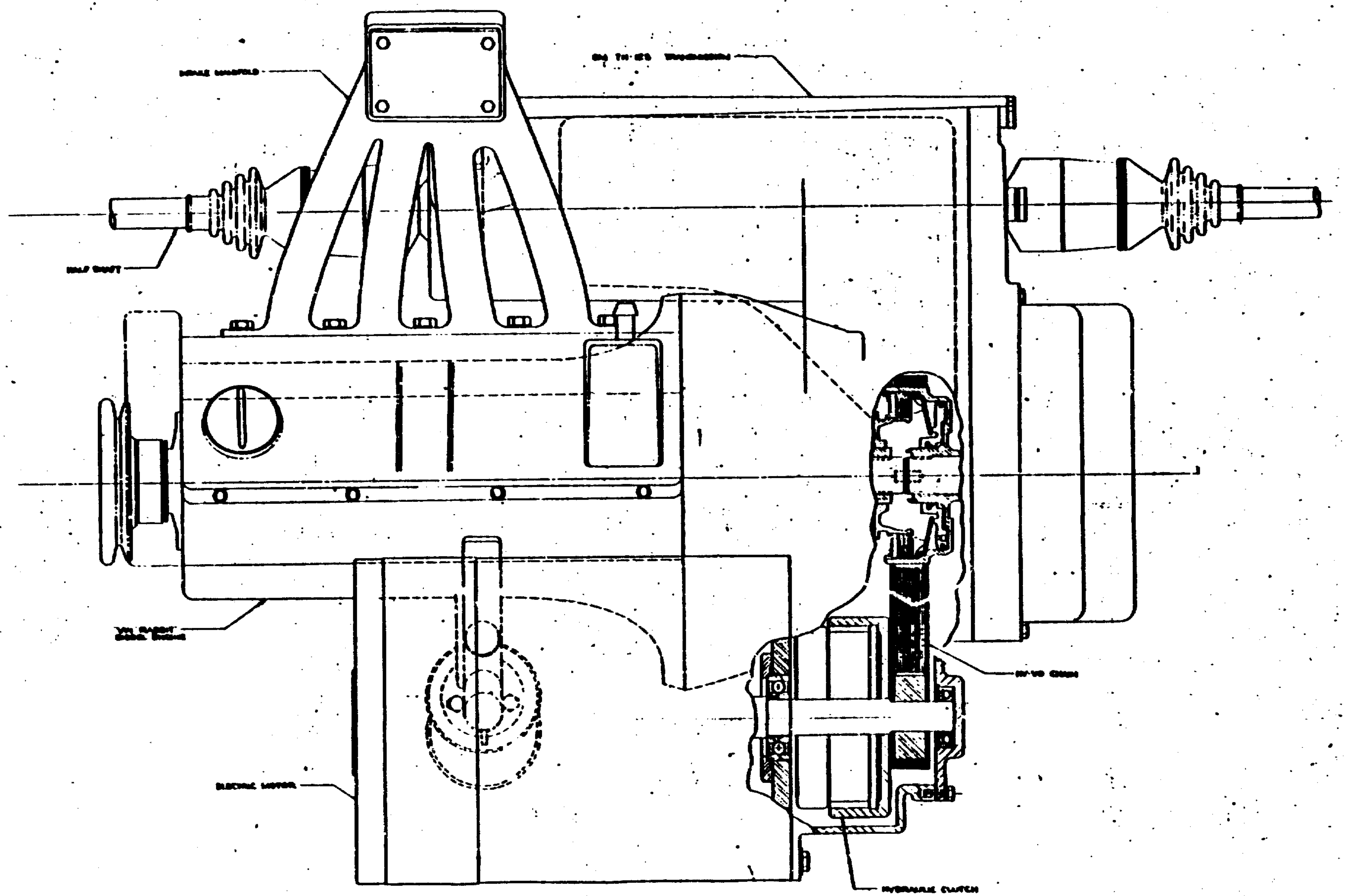


Figure 4-38. Drivetrain Package

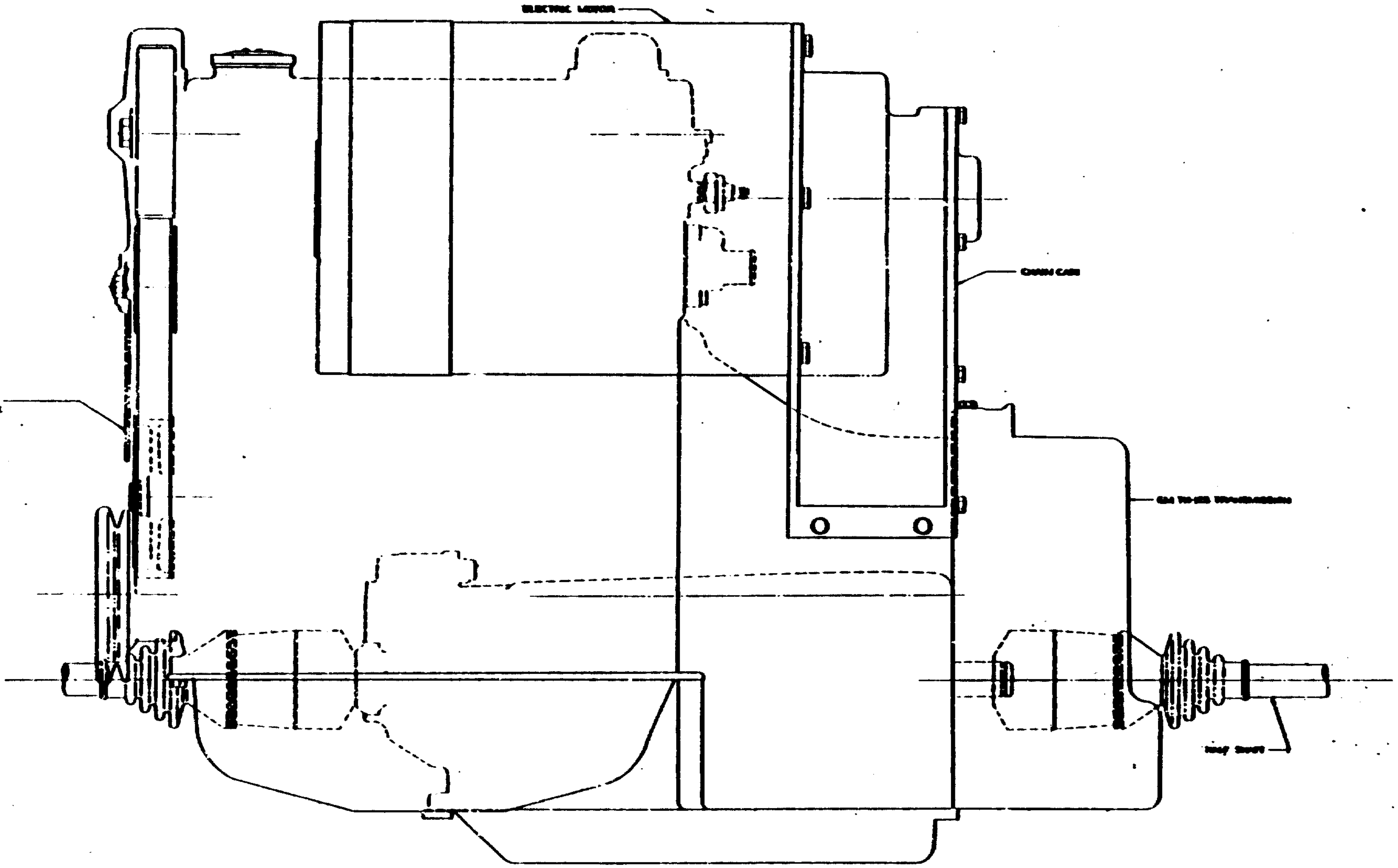


Figure 4-39. Drivetrain Package

NTHV Emission Controls

A hybrid vehicle, designed for petroleum saving rather than for emission reduction, is not covered specifically by the EPA test procedures. The vehicle would yield a variety of different emission data when put through the EPA tests, depending on the state of charge of the batteries. Assuming that the operational strategy calls for the use of electric power with topping by the diesel engine when the batteries are fully charged, the measured emission levels will be very low when the batteries are fully charged. But when the batteries are discharged to their minimum level, the emissions will rise to those of the diesel alone.

Since there is no specified method for selecting the battery state of charge for the emissions test procedure, we propose that the vehicle be tested both with batteries fully charged and with batteries discharged, and that the two sets of emission numbers be combined in the ratio of electric to diesel operation assumed for the vehicle.

EPA tests of the turbocharged Volkswagen diesel engine with a manual transmission give the following results for various inertia weights:

Inertia Weight	HC	CO	NOx
1023 kg (2250 lb)	0.25 gr/km	0.62 gr/km	0.71 gr/km
1250 kg (2750 lb)	0.21 gr/km	0.59 gr/km	0.86 gr/km
1364 kg (3000 lb)	0.23 gr/km	0.59 gr/km	0.80 gr/km

By extrapolating these results and carefully inspecting the emission maps of this engine, we have estimated the emissions at the expected inertia weight class of 1818 kg (4000 lb) to be:

Inertia Weight	HC	CO	NOx
1818 kg (4000 lb)	0.25 gr/km	0.62 gr/km	1.06 gr/km
1981 EPA standards	0.26 gr/km	2.11 gr/km	0.62 gr/km

These data show the well known problem of diesel engines in heavier vehicles meeting the upcoming NOx standards.

The NTHV was found in the Mission Analysis and Performance Specification Studies to have an expected operation over Mission A of

47 percent in the electric mode and 53 percent in the Diesel mode. Assuming that on electric power there is no emission output (which is only slightly off the probable situation), then the overall emission level of the vehicle would be 53 percent of the emissions measured for diesel power. This would mean a weighted emission level of

HC	CO	NOx
0.13 gr/km	0.33 gr/km	0.56 gr/km

Using the same weighting method, the particulant level of the diesel engine would be well below the 1981 standard of 0.37 gr/km.

4.7 ELECTRIC MOTOR

4.7.1 Motor Design

Figure 4-40 shows the performance of the motor selected for the preliminary design of the NTHV. Its performance is shown at a constant 70 volts. This motor is a dc compound motor which, with field control alone, is designed to produce 24 kW output with a 72 volt battery pack. The motor has a base speed in the 1500 to 2500 rpm range, to minimize the amount of clutch slip required to start the vehicle. Its maximum speed is 10,000 rpm, to allow for maximum regenerative braking at high engine speeds and to allow a 2:1 drive ratio between the motor and transmission without overspeeding the motor at maximum engine speed. The 2:1 ratio allows the motor base speed to be equivalent to a speed of 750 to 1250 rpm at the transmission, the same range as an internal combustion engine.

Figure 4-41 is the power output of the motor when connected to a 72 volt lead-acid battery. This data is corrected for battery voltage drop with increasing current and for battery state of charge. The data is plotted for batteries with 0, 40 and 80 percent discharge. (Eighty percent is the maximum allowed for the NTHV batteries.)

Figure 4-42 is a cross-section drawing of the motor used in the preliminary design. The motor has a conventional dc motor construction and is expected to weigh 91 kg.

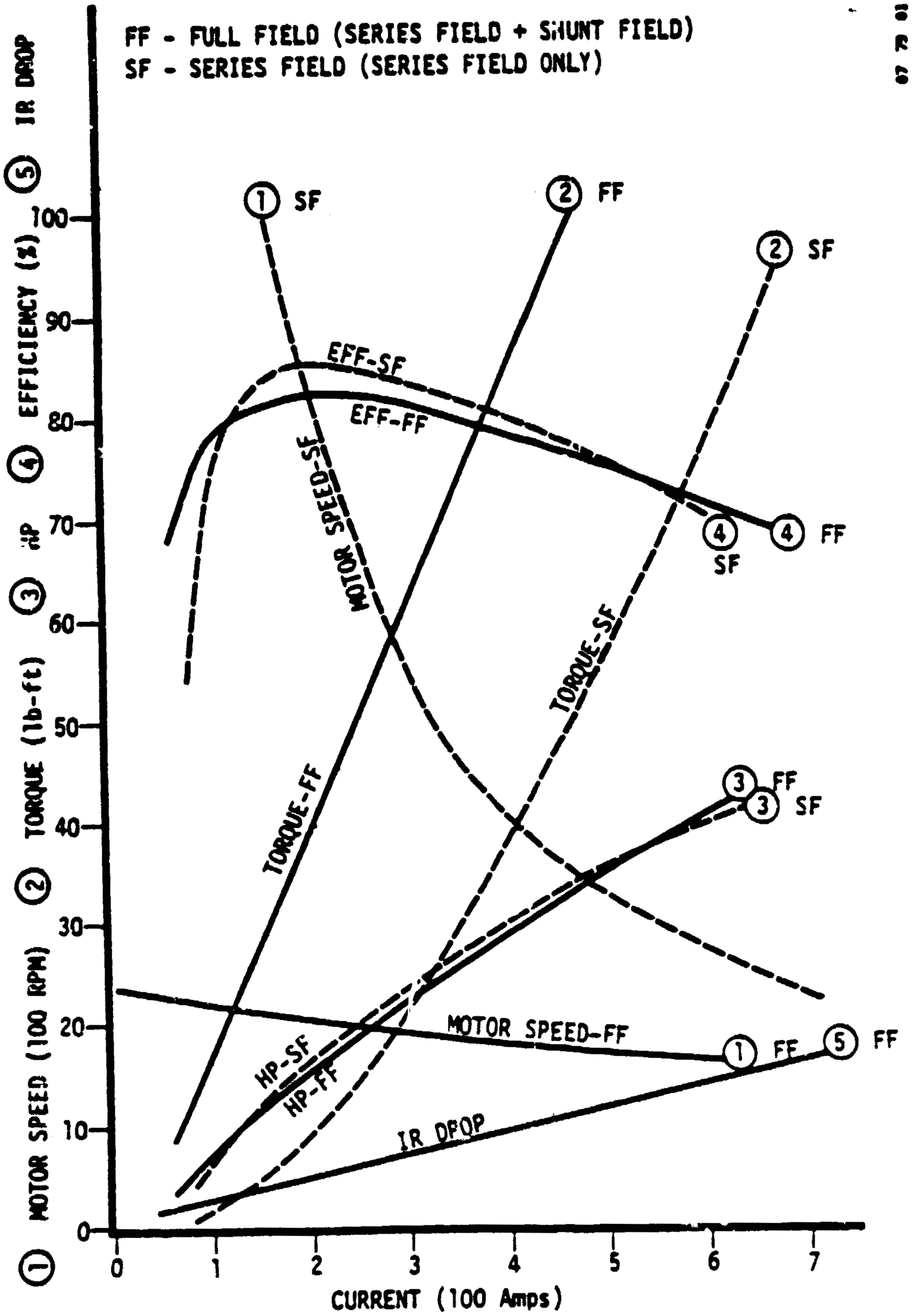


Figure 4-40. Motor Output at 70 Volts

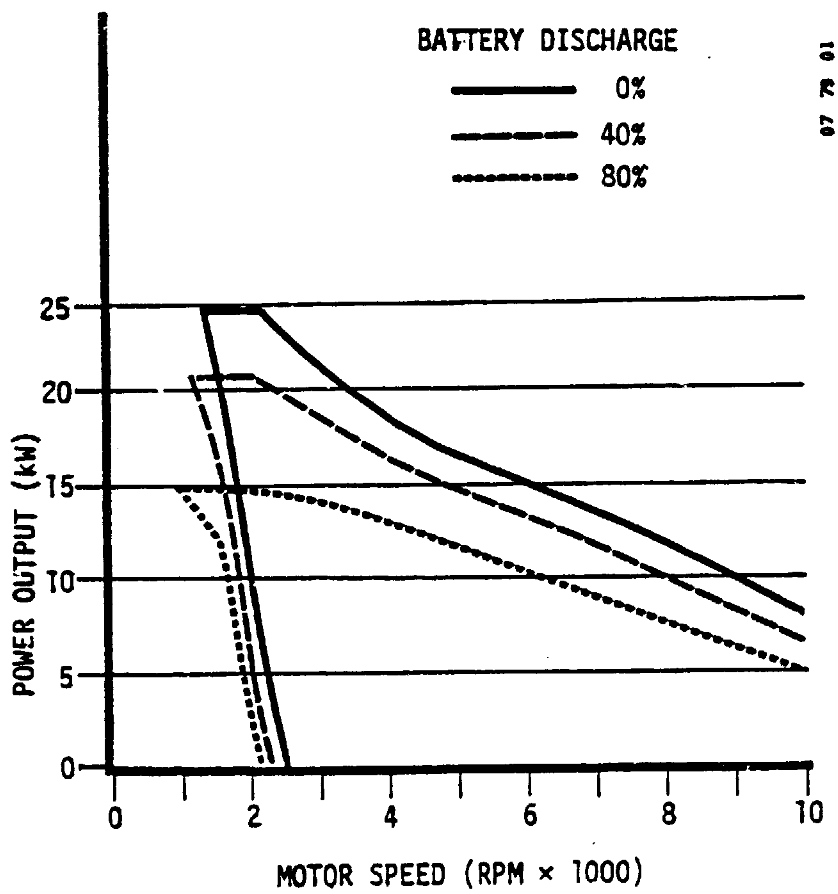


Figure 4-41. Motor Power Output with 72 Volt Battery Pack, 24 KW DC Compound Motor with Field Control

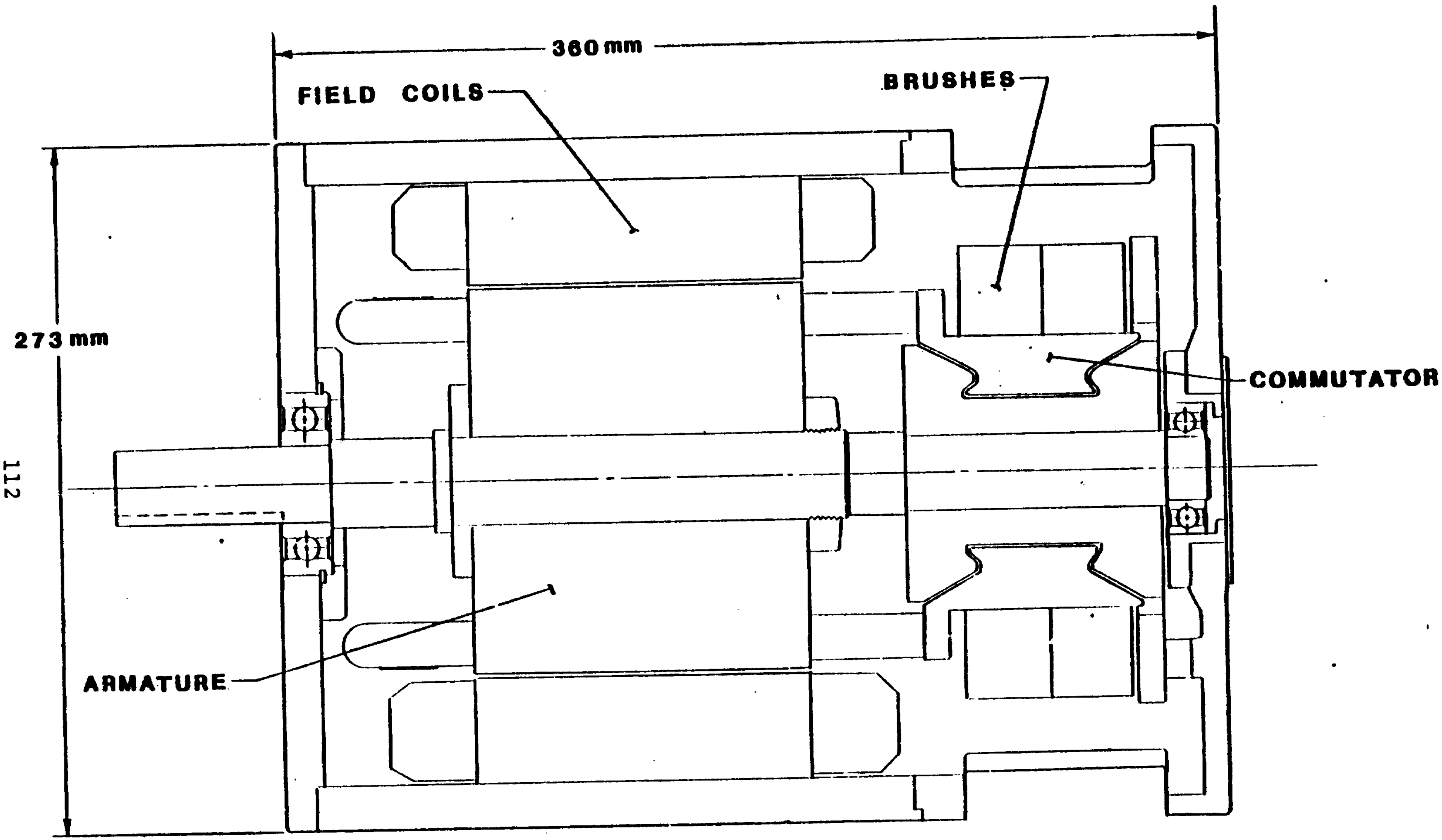


Figure 4-42. Proposed Electric Motor

SECTION 5

POWER CONDITIONING UNIT

5.1 INTRODUCTION

The power conditioning unit includes the motor field controller, battery charger, armature control contractors and starting resistor. Separate controllers for the motor field and battery chargers were considered in the NTHV Design Trade-Off Studies Report of May 1979.[1] Subsequent investigations showed the feasibility of controlling the field current and charging current with the same power electronics circuitry.

5.2 FIELD CONTROLLER REQUIREMENTS

The motor field controller requirements are summarized below.

Input Voltage. The field controller will be designed to operate over an input voltage range of 50 to 95 Vdc. This covers the conditions of maximum discharge to maximum charge.

Load Impedance. The field controller load impedance is the motor field winding which can be described as an equivalent series resistance and inductance.

The load resistance has a nominal value of 1.65 Ohms at a winding temperature of 20°C and varies from 1.40 Ohms at -20°C to 2.50 ohms at 150°C.

The incremental load inductance has a nominal value of 0.10 H at low current and follows the saturation curve shown in Figure 5-1.

Output Current. The maximum output current under any condition of battery voltage or field winding temperature is 20 Adc.

Output Power. The maximum continuous power output varies with field winding temperature from 550 W at -20°C to 1000 W at 150°C.

Output Voltage. The maximum continuous output voltage varies with winding temperature from 28 Vdc at -20°C to 50 Vdc at 150°C.

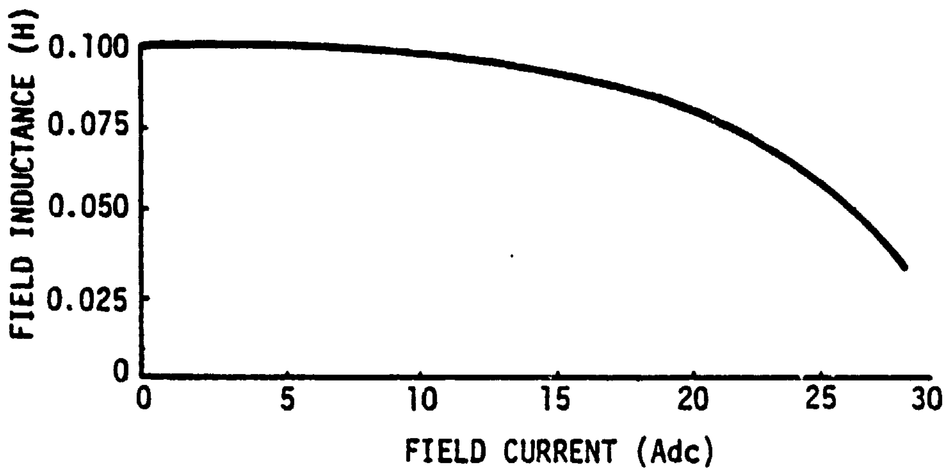


Figure 5-1. Field Saturation

Field Forcing. Higher than rated voltage is applied to the field winding for a period of less than 0.5 second to provide rapid motor deceleration when changing gears. In the field forcing mode, full battery voltage (less any switching element voltage drop) is applied to the field winding.

Electromagnetic Interference (EMI). EMI from the field controller must be limited to a level which does not interfere with the vehicle electronic systems.

Efficiency. The efficiency of the field controller must be greater than 90 percent at maximum output current and greater than 80 percent at 50 percent output current.

5.3 FIELD CONTROLLER DESIGN APPROACHES

The field controller can be implemented with a force-commutated SCR chopper or a self-commutated transistor chopper. The 20A switching requirement could be readily handled with a single small main SCR. However, the additional commutating SCR, diode, inductor and capacitor which are required makes the SCR chopper approach unattractive at this power level, when compared to the simplicity of the transistor chopper.

Bipolar transistors are available which can handle the current and voltage requirements of the field controller with a single transistor. In the past several years, the metal oxide semiconductor field effect transistor, or MOSFET, has been developed as a power switch having inherent advantages over the bipolar transistor. These advantages include: low drive power, higher switching speed, ease of current sharing in paralleled devices, and freedom from secondary breakdown. Accordingly, the MOSFET is selected for the field controller switching element, even though, at the present state-of-the-art, several devices in parallel are required to handle the field controller current requirement with a reasonable device temperature rise.

5.4 BATTERY CHARGER REQUIREMENTS

The on-board charger requirements listed in the Vehicle Minimum Requirements of the RFP for this program are

Input: 120 Vac 60 Hz single phase
Output: 15 Adc and 30 Adc maximum

5.5 CHARGER DESIGN APPROACHES

The primary considerations for the selection of a design approach for the on-board charger are: weight, cost, charger efficiency, input line power factor and line voltage waveform distortion. The efficiency of the battery charging process (expressed in terms of energy available from the battery, relative to energy input to the battery during charging) is not a selection factor, because any of the charger approaches considered can, through proper design of the low level control circuitry, achieve the acceptable maximum charging efficiency of 70 percent. This circuitry does not significantly impact the overall cost or size of the charger.

There are two feasible approaches to implementing the charger power conversion circuitry: 60 Hz SCR phase controlled rectification of the ac line voltage and high frequency chopper control of the battery current.

If the propulsion battery is not connected to the vehicle chassis, transformer isolation is not required. Note that isolation of the battery from the chassis is desirable in any event, to allow at least one fault (inadvertent connection) from some point in the cell array to the chassis without causing current flow. A non-grounded battery should thus be less likely to produce damaging currents in a crash during which a fault is likely to occur.

In a transformerless charger, the SCR and transistor approaches give approximately equal weight and efficiency. The transistor controller will provide a higher line power factor and give less line voltage distortion, but will be a more significant source of EMI than the SCR rectifier. It is assumed that this EMI can be suppressed with proper wiring techniques and a reasonable amount of filter components. Hence the transistor controller is the preferred approach for the charger.

As seen from the above discussion, transistor chopper control is preferable for both the motor field controller and the on-board charger. Also the current requirements are comparable (20A vs. 30A). Therefore, it is reasonable to combine the field control and charger functions. This yields an estimated 50 percent reduction in equipment cost, weight and volume.

5.8 DETAILED MOTOR CONTROL SYSTEM OPERATION

A block diagram of the motor power conditioning system is shown in Figure 5-2. This system consists of the motor starting contactors, resistor and fuses, the field/charger power electronics assembly, the car control computer, the charger ground fault interrupter and circuit breaker, and K3, the field control/battery charger mode change relay. This relay is shown in the de-energized state in which the MOSFET switch in the power electronics assembly is connected in series with the field winding and the propulsion battery. When 115V power is applied by plugging the line cord into the vehicle receptacle, K3 is energized, placing the MOSFET switch in series with the line voltage rectifier and the propulsion battery.

5.7 POWER TRANSISTOR SELECTION

The state-of-the-art in MOSFET transistors is represented by the International Rectifier line of HEXFET^{tm} devices. Available ratings include the IRF150 with V_{DS} = 100V and $R_{D(ON)}$ = 0.055 Ohm and the IRF350 with V_{DS} = 400V and $R_{D(ON)}$ = 0.30 Ohm. The battery charging mode imposes a voltage rating requirement of V_{DS} = 200V, assuming that a silicon avalanche transient suppressor having a breakdown voltage of 180V is placed across the power transistor. It is expected that a transistor in the IRF series will be available in the near term with a 200V rating. The $R_{D(ON)}$ is approximated proportional to voltage rating. Interpolating from the resistance values for the 100V and 400V devices gives an estimated $R_{D(ON)}$ = 0.14 Ohm for the projected 200V transistor. Additional

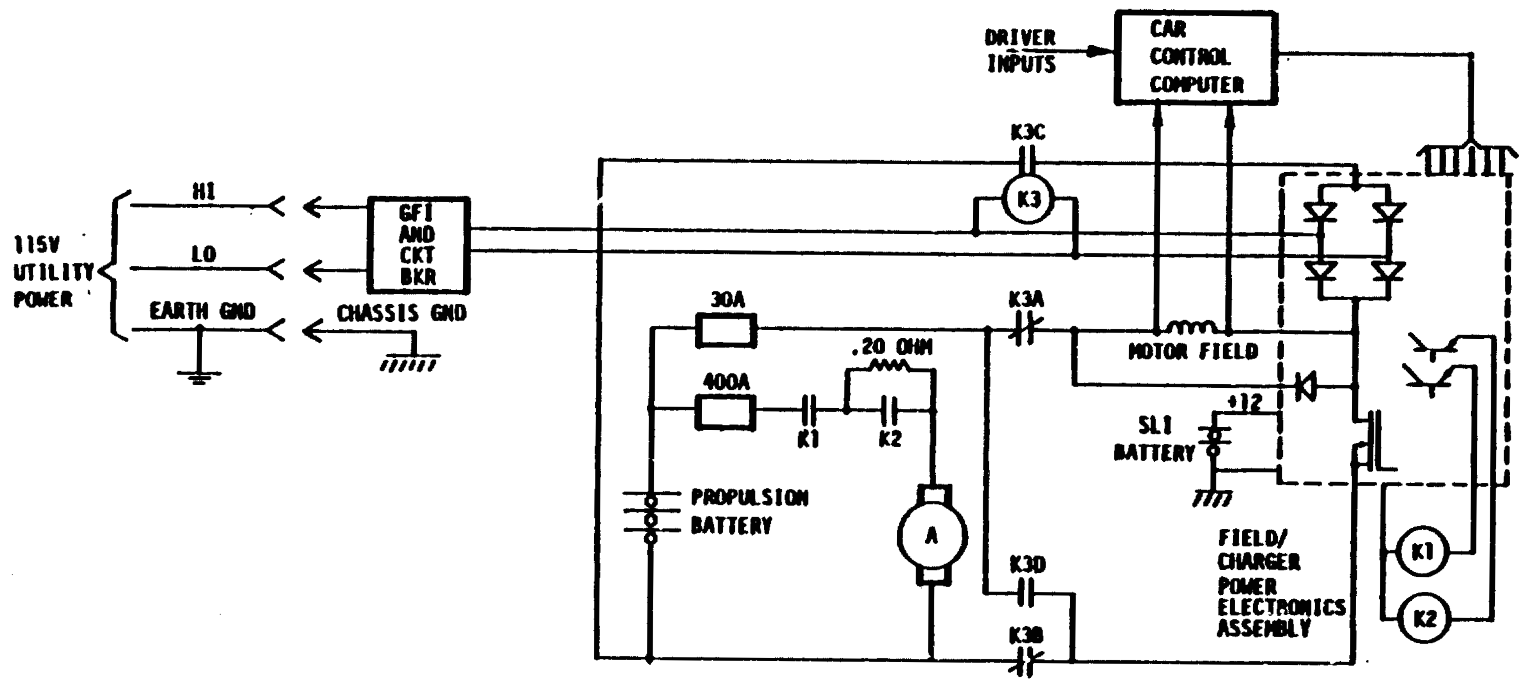


Figure 5-2. Power Conditioning System

calculations which account for unequal current sharing, the increase in resistance with temperature, the allowable junction temperature, the thermal resistances from junction-to-case, case-to-heat sink and heat sink-to-ambient air dictate a requirement for three parallel transistors to handle the 30 Adc maximum battery charging current.

5.8 DETAILED CIRCUIT DESCRIPTION

A circuit diagram of the field/charger power electronics assembly is shown in Figure 5-3. Paralleled MOSFET transistors Q1-Q3 provide the power switching function which varies the average voltage applied to the field winding when the external mode select relay is in the de-energized state, and which varies the average rectified voltage applied to the battery when the relay is energized.

The current command signal is received from the car control computer in 8-bit serial data format. Isolation of the computer ground from the floating ground of the power electronics assembly is provided by photocoupler PC1. Data is converted to 8-bit parallel format by means of shift register SRI. The digital-to-analog converter (DAC) then provides the analog current command signal to the current regulation loop.

5.8.1 Current Regulator Operation

A simplified version of the current regulator loop is shown in Figure 5-4. The operating principle of the circuit involves turning on the transistor switch at instants set by a clock pulse and turning off the switch when the current rises to slightly more than the commanded current.

The current regulator response to a step command is shown in Figure 5-5. Upon initiation of the step command, the current error signal becomes large, exceeding the instantaneous value of the ramp generator voltage. When this occurs, the output of the threshhold comparator goes high. This enables the latch to be set by the next clock pulse. The high latch output is amplified and applied to the paralleled gates of the three MOSFET transistors, Q1-Q3. The transistors then change from the high to the low resistance state (approximately 0.05 Ohm for the three paralleled transistors).

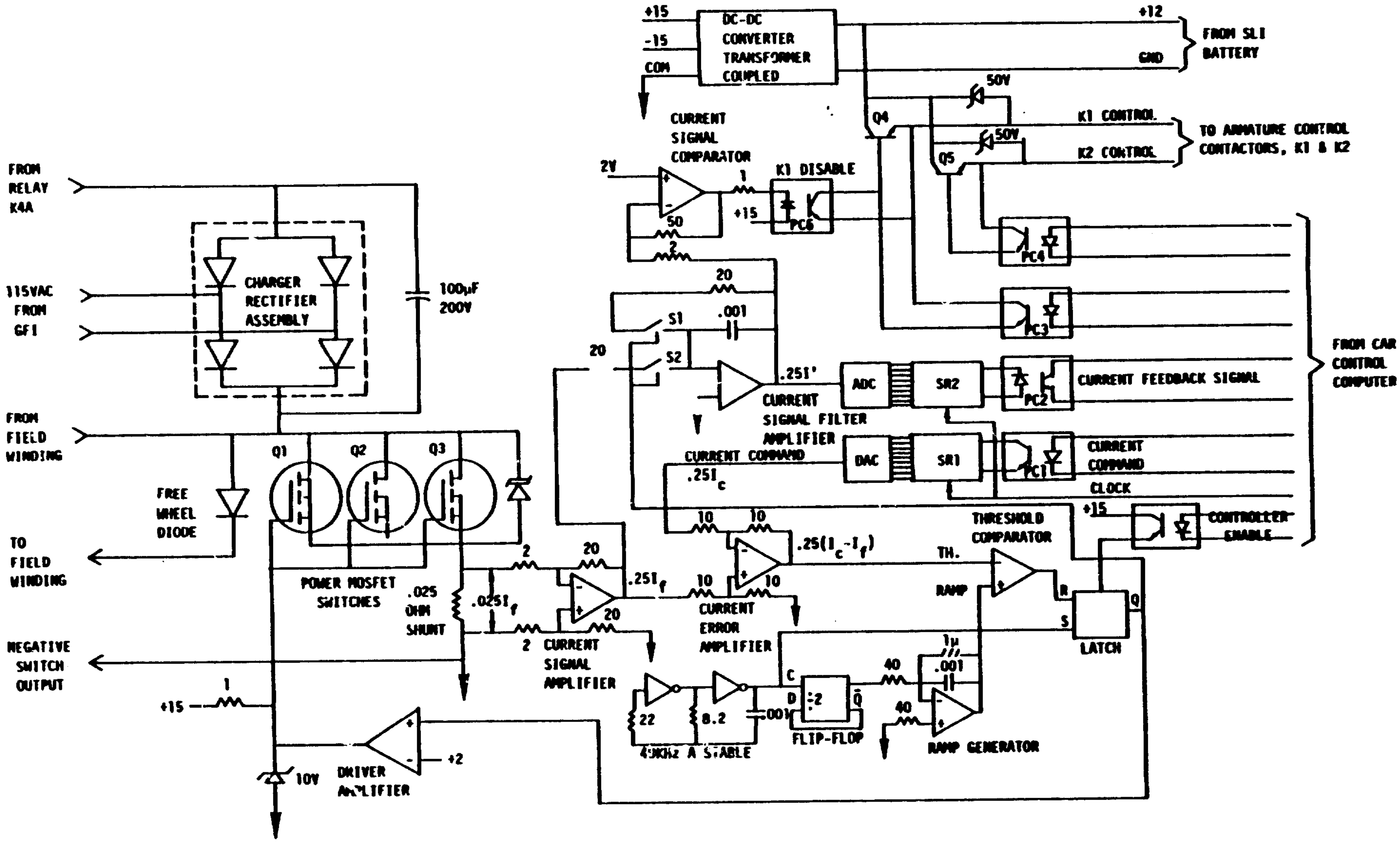


Figure 5-3. Field/Charger Power Electronics Assembly

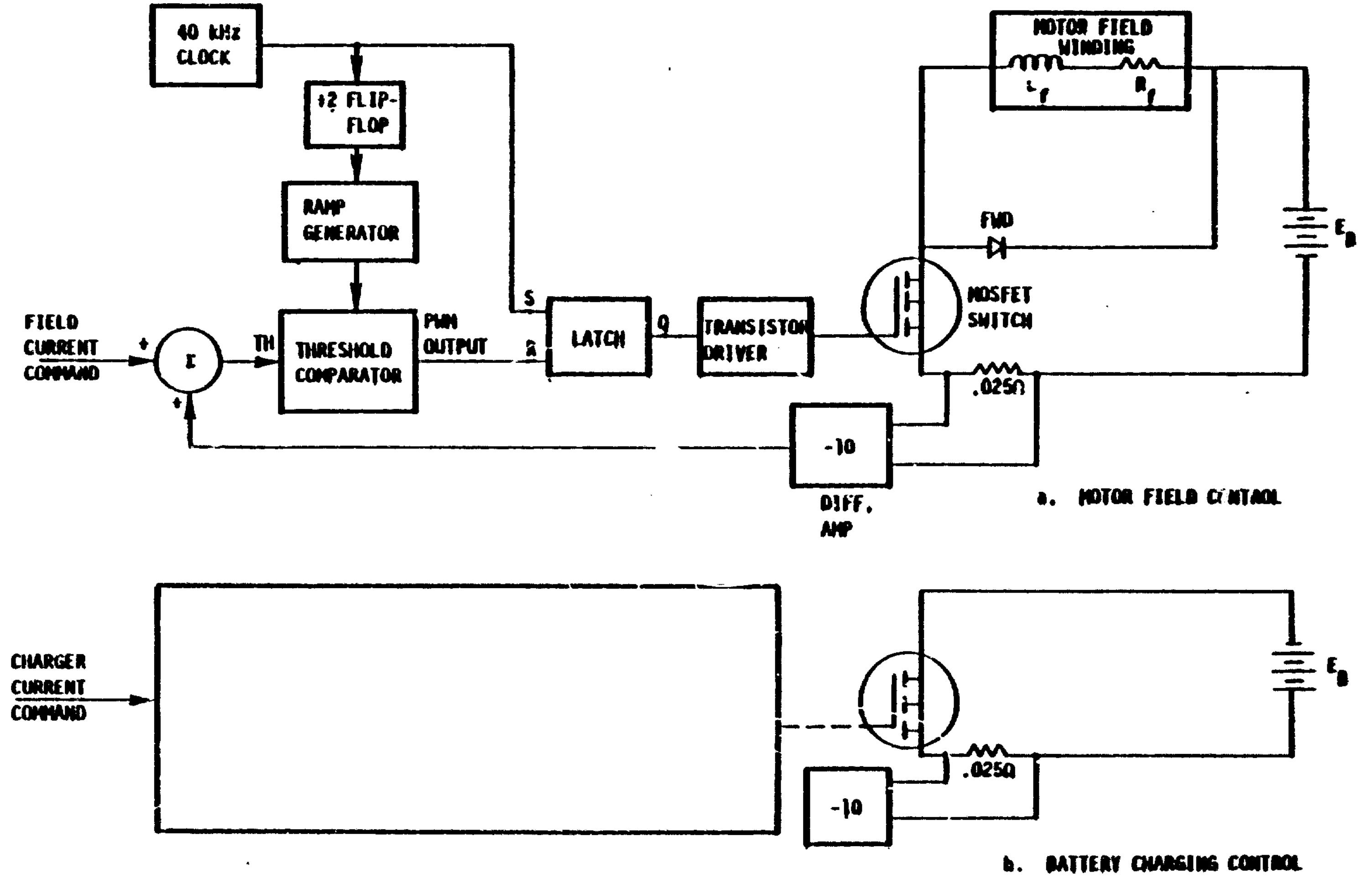


Figure 5-4. Current Control Loop

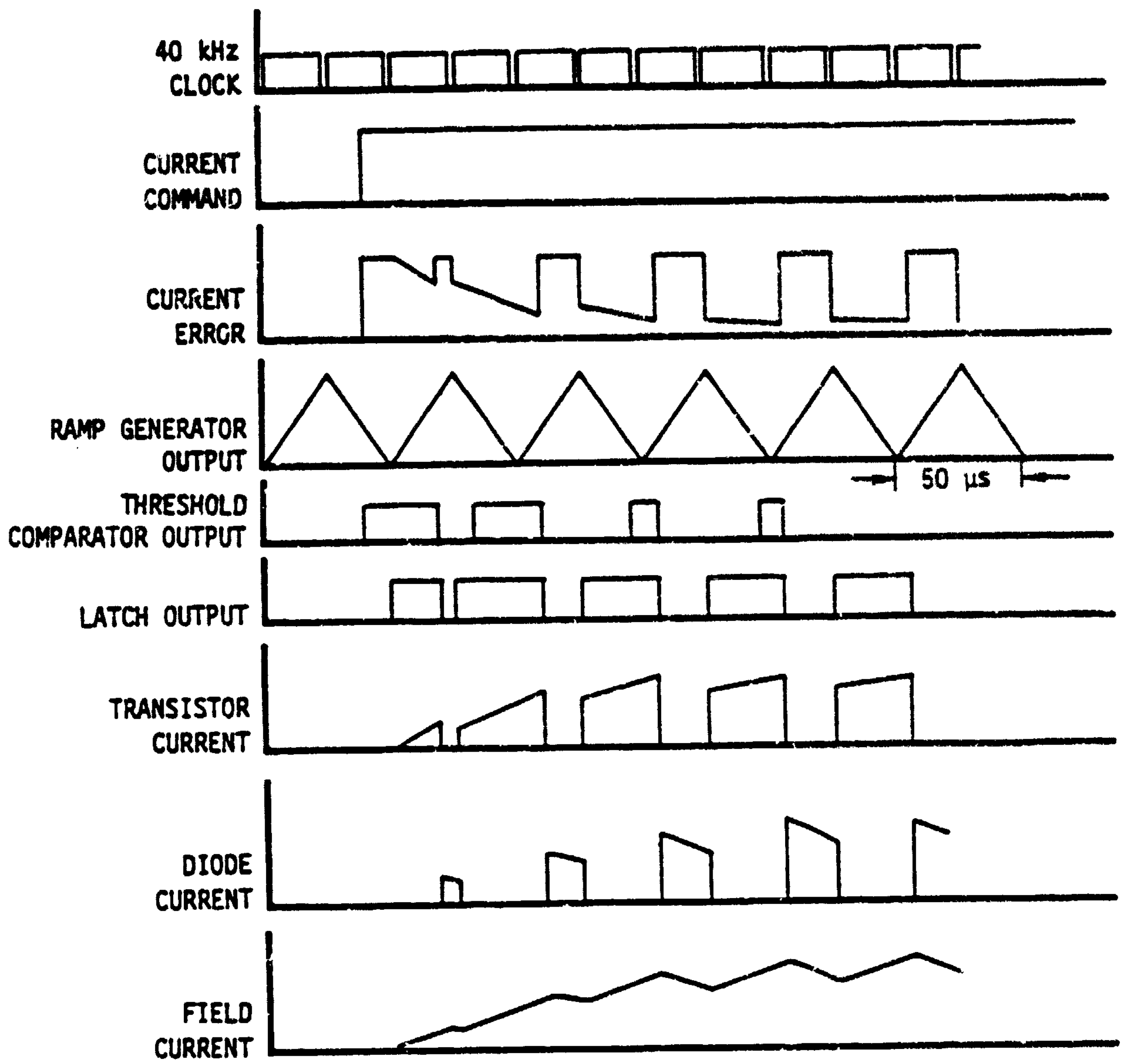


Figure 5-5. Current Control Loop Response Waveforms

5.8.2 Field Control Circuit Waveforms

Current response waveforms for the field control circuit, Figure 5-4a, are shown in Figure 5-5. The field current response to the initial turn-on of the power transistors is

$$i_{f1} = \frac{E_f}{R_f} \left(1-e^{-tR_f/L_f}\right) \quad .$$

When the decreasing current error signal intersects the increasing ramp voltage, the threshhold comparator output goes low, resetting the latch and turning off the power transistors. Field current then transfers to the free-wheel diode, decreasing as

$$i_{f2} = I_o e^{-tR_f/L_f} \quad ,$$

where I_O is the final value of i_{f1}. When the next clock pulse in the sequence goes low, the latch is again set and the power transistors are turned on. Current then transfers from the free-wheel diode to the transistors, again building exponentially as shown. Operation continues until the field current reaches its steady-state level.

5.8.3 Controller Feedback Signal

The car control computer requires a signal representing the actual field or charger current. Because of the discontinuous nature of the signal from the current shunt, a signal representing average current is not available. The current signal filter amplifier, shown in Figure 5-3, serves as a sample-hold amplifier to form a replica of the current signal. During the period that the power transistor is on, the latch output closes CMOS switches S1 and S2, making the filter amplifier output proportional to the current signal input. During the transistor offtime, the control signal to S1 and S2 is zero. This removes the summing junction input and feedback currents, making the feedback capacitor and amplifier output hold the signal level existing at the instant the switches were opened. When the latch again changes state, the power transistors conduct the S1 and S2 close to allow the amplifier output to track the actual current signal.

The current signal replica is processed by an analog-to-digital converter (ADC), the parallel-to-serial shift register converter SR2, and photo-coupler PC2 to produce the isolated current feedback signal for the car control computer.

5.8.4 Failsafe Field Control

Failsafe operation of the field controller is required to prevent motor overspeed from loss of field current. Such a loss of field current could result from an incorrect field current command from the computer or a component failure in the power electronics assembly. The approach to failsafe operation is described as follows. The current signal filter amplifier output, 0.25 I in Figure 5-3, is processed with a current signal comparator. The comparator output energizes the photo-coupled transistor, PC6, which bypasses the base current of the main contactor drive transistor, Q4. Q4 then turns off, removing current from the contactor coil, opening the contactor and disconnecting the armature from the battery.

The failure mode of the MOSFET power transistors is to become a virtual short circuit. This will cause the motor to run at slightly less than the normal base speed, because the rated motor field voltage is somewhat less than the battery voltage to permit field forcing. The computer monitors the field temperature by taking the ratio of the field voltage and current. If temperature becomes excessive because of a controller transistor failure or other reason, the motor armature is disconnected from the battery and the field controller is de-energized.

5.8.5 Battery Charger Operation

Current control loop operation in the charger mode produces a set of waveforms which differs from the above in that the current out of the rectifier into the battery becomes zero when the power transistors turn off. This imposes a 20 kHz pulse current load on the line voltage rectifier which, if unattenuated, would create severe conducted EMI in the utility power lines. This EMI is partially suppressed at the source by a 100 μF 200V capacitor, rated for approximately 20 ARMS, which is placed across the bridge output. Further investigation is needed to determine the amount of additional EMI suppression required.

Another significant difference between the charger and field current control loop waveforms is due to the 120 Hz ripple voltage component in the line voltage rectifier output. The control loop will respond to this in attempting to maintain charging current at the constant commanded value by varying the pulse width of the charging current. This could be avoided by decreasing the control loop bandwidth when in the charging mode. Further study is needed to determine if this added complexity is required.

5.9 POWER ELECTRONICS ASSEMBLY PACKAGING

The size of the power electronics assembly is determined by the finned heat sink extrusion required to cool the three MOSFET power switching transistors. The extrusion is cooled by natural convection. It should be noted that the maximum heat sink temperature rise will occur during the charging mode. Therefore, no advantage is gained by placing the electronics assembly in the path of cooling air set in motion by either vehicle movement or the IC engine cooling fan.

An analysis of the power transistor heat transfer led to the selection of 230 mm long thermalloy type 6192 extrusion for the preliminary design. This extrusion is 143 mm wide with twelve 25 mm high fins. The extrusion forms one side of the sealed power electronics assembly package having overall dimensions of 235 mm x 150 mm x 150 mm.

Three connector receptacles are provided. The wiring harness from the field winding, mode control relay, and the ground fault interruptor is terminated in a 7 pin size 25 MS type connector. The cable from the 12 volt battery and the armature control relays is terminated in a 6 pin size 20 MS type connector. The power electronics assembly is linked to the car control computer through a 20 conductor jacketed ribbon cable terminated in a D-connector with backshell strain relief.

SECTION 6

BATTERY SUBSYSTEM

6.1 SELECTION OF BATTERY TYPE

Studies of various battery types disclosed that the improved state-of-the-art lead-acid battery is the most likely to meet the requirements of hybrid vehicle energy storage, and to be available in the near term. The nickel-iron system has some drawbacks, along with its desirable features. It is both mechanically and electrically rugged and will probably achieve a 2000 cycle life quite readily (which was shown in Reference 1 to contribute substantially to low life cycle costs). Because the iron electrode is not soluble in the electrolyte (in contrast to the zinc electrode in a nickel-zinc battery), a much simpler and less costly separator system can be used. Improvements in capacity per unit weight can be expected, and improved iron electrodes are being developed. However, the battery gasses severely on charging — especially when cold and in the latter part of charging — due to the relatively low hydrogen overvoltage on iron. A cure for this problem would improve the charge efficiency relative to the lead-acid and nickle-zinc systems, and would introduce the possibility of sealing the battery for zero or near-zero maintenance for water addition. However, its limitations do not seem soluble in the near term; commercial U.S. production ceased in 1974.

Although the nickel-zinc system has impressive energy density (and hence petroleum savings), it suffers from short cycle life and the likelihood that it will not be able to achieve an 800 cycle life (at 80 percent depth of discharge) in the near term. A sophisticated and costly separator system is needed to minimize electrode slumping and shape change, not to mention internal shorting by the growth of zinc trees through the separator. Moreover, an overcharge on a nickel-zinc system, if sealed, is not tolerable. Even if not sealed, over a modest number of cycles there is a problem of exhausting the reserve capacity (of ZnO) in the zinc electrode because of the necessity of charging sufficiently to ensure the full charge of the nickel electrode. The consumption of the reserve capacity of ZnO quite markedly increases the risk of shorting. However, over the next several years it is likely that there will be improvements in the charge acceptance of the nickel electrode at its upper levels of state of charge. There will also be continued modest improvements in separators for the nickel-zinc (and likewise for the silver-zinc) batteries.

Although this is a brief recitation of some of the problems facing only the nickel-zinc and nickel-iron systems, it is felt that many of the problems related to the lead-acid system are likely to be reduced in the near term. The large production by many manufacturers of lead batteries enhances the opportunities to generate improvements, especially since company self-interest is involved.

6.2 DUTY CYCLES AND VEHICLE REQUIREMENTS

Daily driving patterns are by and large characterized by combinations of the Federal Urban and Highway Driving Cycles (FUDC and FHDC, respectively). For the FUDC, an examination of the results of CARSIM computer runs reveals the following requirements on battery current:

FUDC:

a.	Number of current peaks of 450 ±10A	14
b.	Approximate duration of peak pulses	≈1-2 sec
c.	Range of intervals between peaks	3-372 sec
d.	Approximate A-hr (from 0 to 450A, with pulse durations of item b above)	1.9 A-hr
e.	Number of current peaks of 400 $^{+40}_{-10}$ A	19
f.	Approximate duration of peak pulses (range)	≈1-5 sec
g.	Range of intervals between peaks	3-349 sec
h.	Approximate A-hr (from 0 to 400A, plus the A-hr from 400 to 450A for the 14 peaks to 450A)	2.8 A-hr
i.	Number of current peaks of 300 $^{+140}_{-10}$ A	40
j.	Approximate duration of peak pulses (range)	≈1-10 sec
k.	Range of intervals between peaks	3-147 sec
l.	Approximate A-hr (from 0 to 300A, plus the A-hr from 300 to 450A for the 14 peaks to 450A and the 19 peaks of 400A)	8.4 A-hr

Fortunately, no peaks exceeding approximately nine seconds were identified at or above the 300A level, although the nine-second duration around 300A is significant, since it represents about twice the battery capacity.

In the FHDC, CARSIM computer runs generated the following battery current requirements:

FHDC:

a.	Number of current peaks of 450 ±10A	14
b.	Approximate duration of peak pulses	≈1 sec
c.	Range of intervals between peaks	2-197 sec
d.	Approximate A-hr (from 0 to 450A, with pulse durations of item b above)	1.8 A-hr
e.	Number of current peaks of 400 $^{+40}_{-10}$ A	23
f.	Approximate duration of peak pulses (range)	≈1-4 sec
g.	Range of intervals between peaks	2-186 sec
h.	Approximate A-hr (from 0 to 400A, plus the A-hr from 400 to 450 A for the 14 peaks to 450A)	4.0 A-hr
i.	Number of current peaks of 300 $^{+140}_{-10}$ A	49
j.	Approximate duration of peak pulses (range)	≈1-9 sec
k.	Range of intervals between peaks	3-72 sec
l.	Approximate A-hr (from 0 to 300A, plus the A-hr from 300 to 450A for the 14 peaks to 450A and the 23 peaks to 400A)	12.7 A-hr

One peak pulse of approximately 10 seconds duration was noted at the 300A level; at this level, most of the pulses were three seconds or shorter.

Some battery requirements are dictated by vehicle packaging and cycle life considerations. A summary of the battery requirements is presented below.

a.	Total EMF (volts)	72
b.	Number of batteries in series	12
c.	Energy content at the 3hr rate, kW-hr	12.6
d.	Weight, kg [lb]	336 (740)
e.	Recharge time, hr	5-7
f.	Cycle life at 80 percent depth of discharge	800
g.	Energy density, W-hr/kg [W-hr/lb]	37.5 (17)
h.	Peak current, A (available for 2 sec*)	450
i.	Secondary load pulse, A (available for 5 sec*)	400
j.	Tertiary load pulse, A (available for 10 sec*)	300

*According to CARSIM computer simulation

k. Size** per battery, cm [in] < 41 (height) x 20 x 30 [< 16 (height) x 8 x 12]

6.3 BATTERY CHARGER

The on-board charger (120V 60Hz input; 15 Adc and 30 Adc maximum outputs), through proper design, offers the possibility of combining the charger and the field supply controller (especially since the charger and controller have about the same power rating). Such a combination can provide a weight savings of about 10 kg. In this preliminary design, the charger already incorporates weight savings features by having the solid state circuitry provide elevated frequency pulses (as compared to 60 cycles for a direct line conversion model), thereby reducing the iron mass requirements. The fast moving improvements and cost reduction of solid state components, such as may be needed in this charger, may provide an interesting and durable unit.

The on-board charger should be able to satisfy most battery charging requirements. If needed, the off-board charger (208V or 240V, 60 Hz and 60 Adc maximum) is less constrained by weight. So, simpler designs (SCR rectifications) may be considered. Both on-board and off-board chargers are required to emit a minimum of EMI radiation.

If the cost penalties are not severe, the off-board charger could conceivably be designed to (eventually) supply even heavier charge currents at the start of a charge cycle, when the state of charge is low and charge acceptability is high. This reduces the time required for a full charge, and provides a quick partial charge for modest (emergency or other) travel in the afternoon or evening. The size of the current input to the battery would be related to battery temperature, state of charge, and cost of the charger. Some battery designers feel that such high inputs to batteries in a nearly discharged state are not harmful if temperatures are kept under control, if gassing is prevented or kept at a minimum, and if individual cell voltages (more correctly, individual electrode potentials) are not beyond the design limits. Perhaps

**Represents compartment dimensions in the vehicle. A real battery must be smaller to allow room in the compartment for insulation, and for forced air passages for heating/cooling and for battery gas scavenging.

future battery designs could incorporate sensors of temperature, electrode potential, presence of gassing, specific gravity, electrolyte conductivity, etc., giving signals to feed back to the charger and thereby control its output. It is to be expected that electric vehicle chargers will be more sophisticated than typical industrial chargers, because human monitoring cannot be relied upon.

Since excessive overcharge (gassing), frequent undercharge (plate sulfation) and elevated battery temperature (especially beyond 43°C [110°F]) reduce the cycle life, the charger should accommodate the battery status and adjust to avoid outputs detrimental to the battery. However, it must provide for a modest overcharge (to guarantee that the battery has a full charge, and to provide a small amount of gassing to assure electrolyte mixing), despite the fact that such overcharge causes a small amount of shedding of active material and/or grid corrosion.

6.4 BATTERY THERMAL REQUIREMENTS

The range of temperature which the vehicle and battery will experience is -30°C to 50°C (-22°F to 122°F). Since even a fully charged battery is unable to deliver good performance at subfreezing temperatures, it must be warmed to some acceptable temperature to perform adequately, or even marginally. As discharge continues, internal self-heating increases the electromechanical activity, lowers the acid viscosity and resistivity, and improves performance.

Electrolyte freezing is a risk at low temperatures when the battery is partially or totally discharged (such conditions reduce the specific gravity of the electrolyte). Freezing is a factor when the specific gravity falls to 1.200 (as measured at 25°C) or 1.235 (as measured at -30°C). As an example, consider a battery with a full charge specific gravity of 1.285 at 25°C and a fully discharged specific gravity of 1.160 at 25°C — a drop of 125 "points." When the specific gravity drops from 1.285 to 1.200 (85 points), the battery is at $\frac{85}{125} \times 100 = 68$ percent depth of discharge. So, when a user in a cold climate operates his vehicle to this or greater DOD and parks for an extended time where temperatures drop enough to cool the battery to -30°C, he risks battery freezing. One protective scheme is to put the battery on charge promptly at the parking location to bring up the specific gravity.

Elevated temperatures present a different problem. Temperatures of 40°C and higher, even for extended periods (in a heat wave), are not uncommon in southern and southwest portions of the U.S. Even temperatures approaching 48°C for several hours are not uncommon. Although battery electrical performance is not especially harmed at these temperatures, battery life is compromised. Since a working battery can have internal temperatures of at least 5 to 10°C above the ambient, the effect on life is greater. So, when ambient temperatures are high, prospects of cooling below ambient vanish. One then accepts the degraded cycle life and depends on the control strategy to revert to diesel power.

For low temperature environments, opportunities exist for heating the battery with a heater or with waste heat from the diesel engine. A battery heated to 5°C will manifest about 85 percent of its full capacity, more or less, depending on design. If the time required to bring the battery to this temperature is tolerable, an acceptable performance may be achieved, and subsequent temperature rise from current draw will improve it. A warm up temperature of 15°C adds approximately 5 percent to the battery capacity.

If range may be sacrificed at low temperatures, shorter battery heating times (which conserve fuel) would allow warm-up to temperatures less than 5°C. A warm-up to only -18°C still produces an available battery capacity of about 65 percent. Subsequently, self warming would elevate the capacity to a higher level.

6.5 BATTERY CONTAINER DESIGN

The design criteria for the battery container were developed from reviews of the designs of existing electric vehicles and from the proposed Electric Vehicle Safety Standards. The general criteria are as follows:

1. Light weight, integrated with structure
2. Simple battery access from above
3. Positive lock-in of the battery cases
4. Improvement (or no degradation) of crashworthiness
5. Good flow paths for a hot air circulation system
6. Non-corrosive plastic surfaces
7. Drain hole provision for flushing with water.

The battery container preliminary design was begun by first identifying the volume and locations of the batteries and the case size. The general design concept is applicable to various case sizes and numbers of batteries.

6.5.1 Basic Construction

The battery containers will be constructed of molded figerglass (Figure 6-1). The front container will be mounted on the front bumper extension tubes and the rear surface of the front bumper channel. Additional support will be provided at the center of the container.

The rear battery container and its batteries will be supported with lateral steel tubing cross members. These will be welded to the rear side frame members of the X body. Longitudinal straps running under the battery compartment may be used to provide additional support.

Both containers will carry molded-in positioning lugs, so that each case in the compartment is correctly located. The heated air flow paths will also be molded integrally. The upper edge of each battery container will carry a molded-in flange, which will mate with a similar flange on the battery covers. The lower surface of the battery containers will have adequate drain holes, so that they can be completely flushed with water to remove any residual materials from the battery compartment.

6.5.2 Battery Hold-downs

The design of the battery hold-downs is quite simple. It consists of a fiberglass channel which is molded to fit across the top of the batteries and to clear the water-venting system. The hold-down is mounted on the battery container with studs which are welded to the compartment support frames. A captive wing nut is used to secure the channel to the stud. The hold-downs fit completely inside the covers.

6.5.3 Battery Container Cover

In order to protect the batteries and collect any hydrogen escaping from the water level control and venting system, a tightly fitting vent cover (Figure 6-2) is located over each battery compartment. The battery container cover is molded (vacuum-formed) of rigid polypropylene plastic. A vent opening is located on the ramp-shaped upper surface of the cover, and is fitted with a flexible polypropylene exhaust tube leading to the hydrogen collector. The battery covers are completely removable for service.

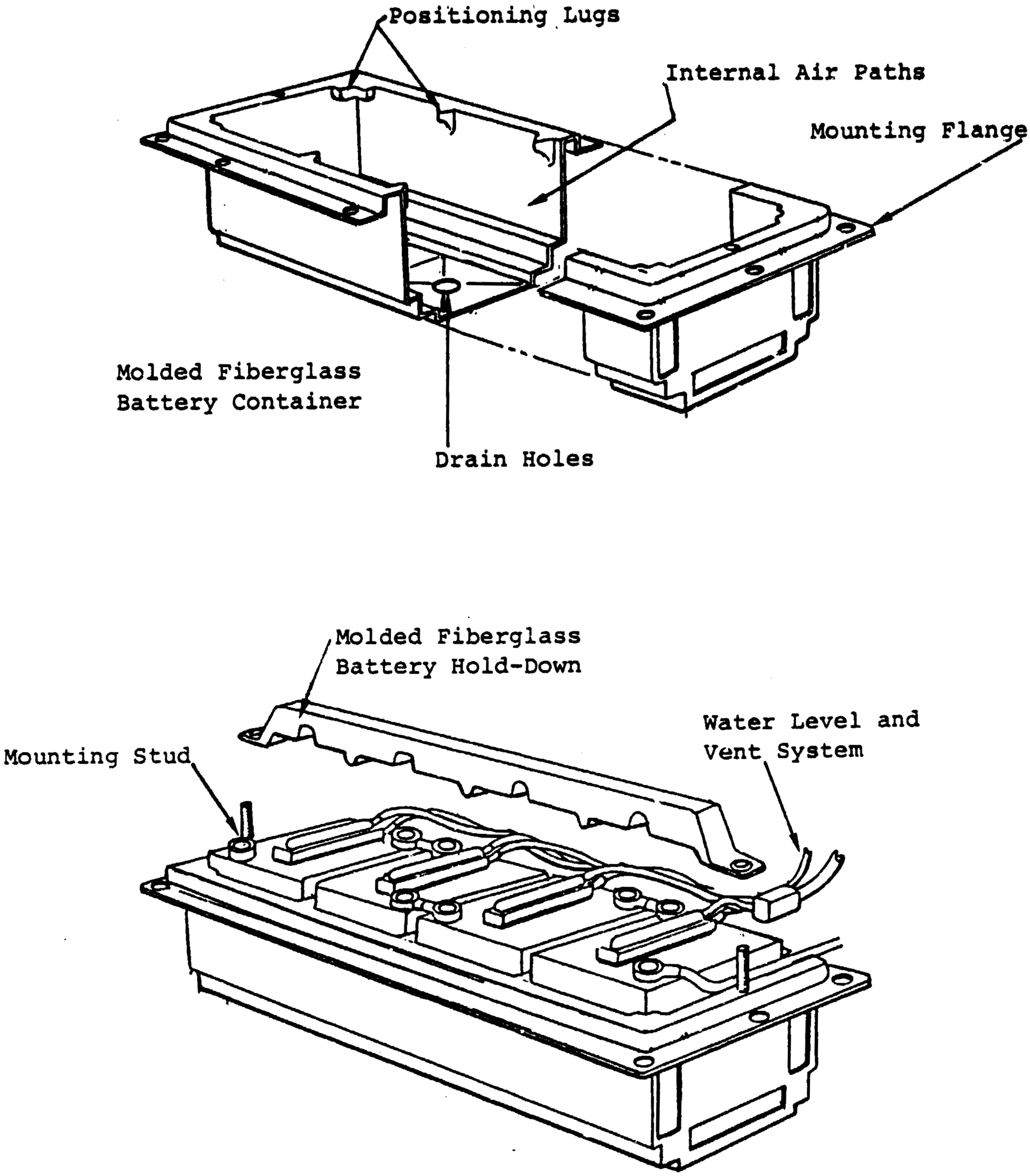


Figure 6-1. Battery Container Construction

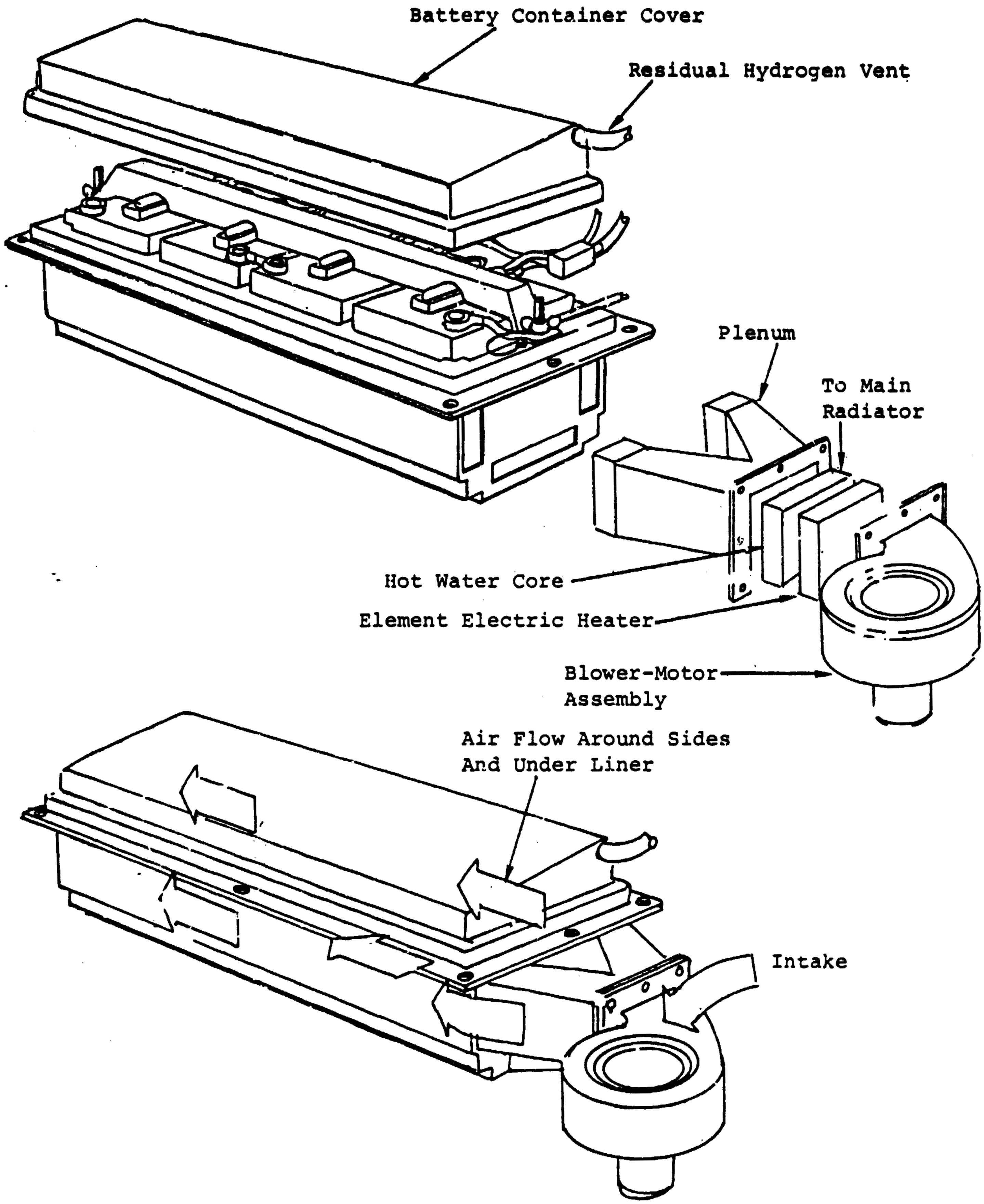


Figure 6-2. Battery Container Assembly

SECTION 7

CONTROL SYSTEM

7.1 NTHV CONTROL SYSTEM PRELIMINARY DESIGN

The NTHV powertrain control system is one of the new technology elements of the NTHV program. The preliminary design of the power train control system, and the rationale for its development, are presented in this section.

The methodology utilized to arrive at the preliminary design is shown in Figure 7-1. The discussion below briefly identifies the motivation for each element of the methodology and gives the rationale for the conclusions reached.

7.2 IDENTIFICATION OF CONTROL SYSTEM REQUIREMENTS

The first step in the methodology was to identify the control requirements of the various vehicle subsytems. A review of the Mission Analysis and Performance Specification Studies Report[24] and the Design Tradeoff Studies Report[1] indicated that in order to meet the project constraints and maximize petroleum savings, the powertrain operational strategy would involve a fairly complex set of control requirements. The configuration of the powertrain did not preclude direct mechanical driver control, but two other considerations made an electronic control system for this subsystem mandatory. The first was the required maximization of petroleum savings, given a set of performance constraints. The second was driver convenience and acceptance, i.e., marketability.

7.2.1 Powertrain Control Alternatives

Three general alternatives were available for selection. The first was an overall manual operation, in which all functions involving the clutches, motor and engine were controlled by the driver. There were many problems with this approach. First, in order to meet the performance requirements, the driver is forced to assume a number of tasks when the performance requirements call for dual power plant operation. From the driver's viewpoint, the operation is unacceptably complex. Further, the driver might not understand how to maximize petroleum savings (and might not want to); hence, the goal of maximizing the petroleum savings would probably not be attained.

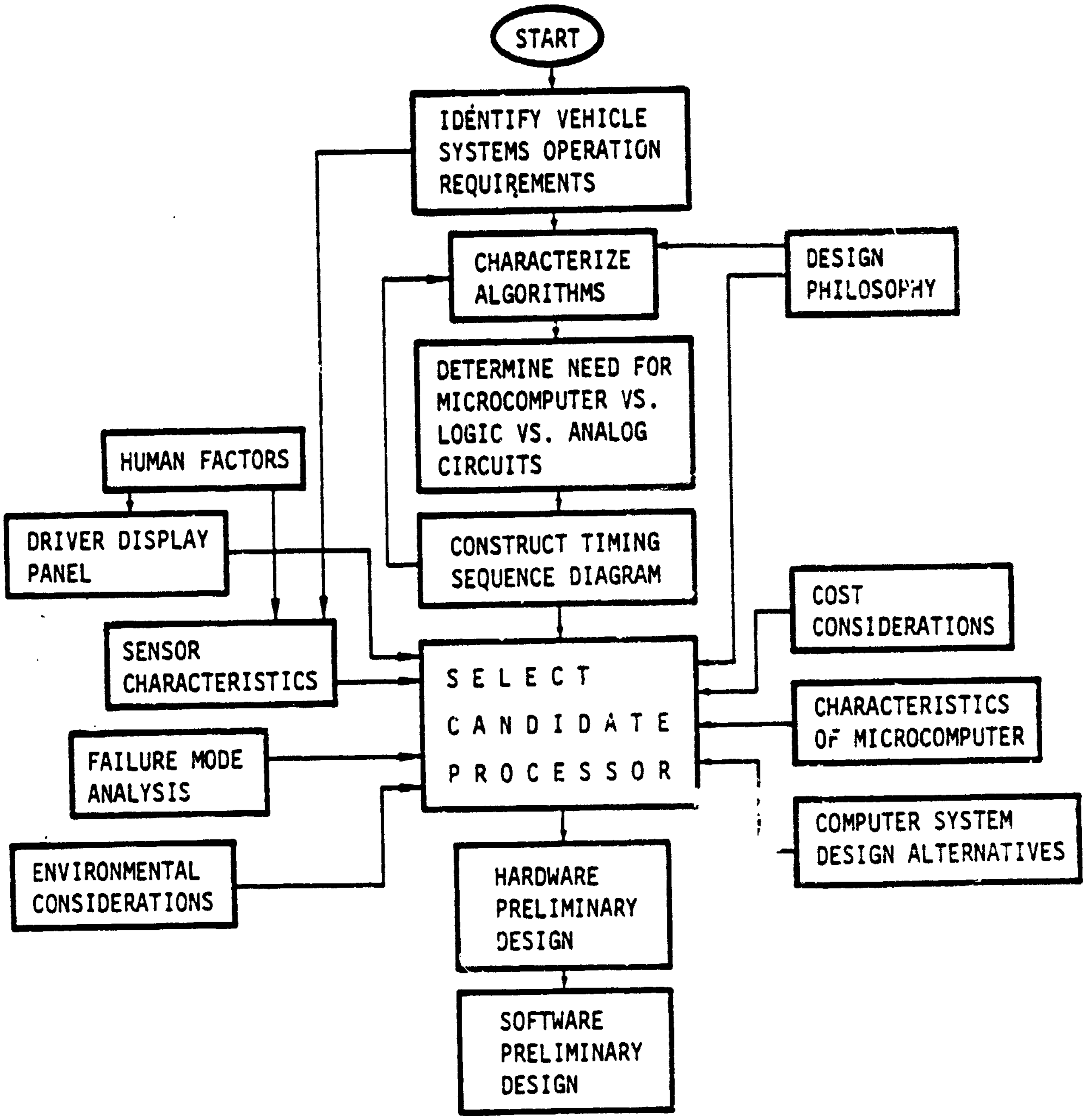


Figure 7-1. Methodology Utilized in the Development of the Control System Preliminary Design.

One conclusion about this first approach is that driver control is not practical for the situation in which performance requirements mandate using both power sources. Another conclusion is that any system which allows the driver to make judgments about what the operating mode (diesel, electric or both) should be (and when both systems are operating, what the power split should be), will not, in general, maximize the petroleum savings. These conclusions rule out the possibility of direct driver control (in normal driving operation) and suggest the involvement of a logic system. However, the logic system needed to control a hybrid powertrain to maximize petroleum savings and provide driver acceptability must also be flexible and adaptable. Therefore, these requirements and the complexity mandate the use of a computer-based system.

The second configuration was computer control. This system accepts driver inputs and takes into consideration the power train condition and the operational strategy. The latter is designed to maximize the petroleum savings. The computer system adjusts the powertrain control according to the driver demands, under the requirements of the operational strategy. This approach could adequately conform to the requirement of maximization of petroleum savings, consistent with the performance constraints. However, this "drive-by-wire" approach does not provide the driver with the capability to drive the vehicle in the event that the computer fails, since there are no direct links between the driver and the powertrain. Since a computer system failure would come without warning, the driver could be greatly inconvenienced, or even possibly placed in a precarious situation. Thus the approach would not satisfy the requirements of driver convenience and acceptance.

Two new conclusions were drawn from these considerations. First, the vehicle must be driveable (at least in a degraded mode) if the computer control system is inoperative. Second, failsafe operation of the vehicle under computer control is required.

This leads to the third configuration which has two operational modes, normal and minimal (which involves computer system override).

This control system appears to the driver as a drive-by-wire system under normal operating conditions, in which all aspects of the computer control system and powertrain are working properly. But in other circumstances the system would exhibit its failsafe and/or operational override capabilities. (The particular control system responses, and the mechanisms by which the responses will occur, are discussed below.)

The driver has the capability to directly control the essentials of the powertrain, using the backup electronics, either under a failsafe mode of the computer system or under an override directive.

This configuration maximizes the petroleum savings whenever the system is within its normal range of operating conditions. And it meets the requirements of customer convenience, safety, and acceptance by allowing manual operation when the computer fails.

The control system concept developed in the preliminary design of the NTHV was derived from the third concept and was a computer-based control system with (1) fail-degraded/fail-safe operational capabilities and (2) a capability for manual override and direct operator control of the heat engine and transmission. However, in this latter mode the electric motor is locked out of the powertrain loop.

7.2.2 Control and Monitoring Requirements

In step one of the control system design methodology we identified the specific subsystems which would require monitoring and control, or simply monitoring. These subsystems are listed in Table 7-1. We also identified the actions required of the computer system for each of its functional responsibilities. Table 7-2 gives a summary of the control system functions and the sensors and actuators required. In general, the sensors and actuators available for use seem to meet the needs of this project, although currently they may not be inexpensive.

Table 7-1. Control System Functional Responsibilities

System Monitoring and Control	System Monitoring
Diesel Engine	Accessory & engine battery
Electric motor & motor controller	Brake pedal & lines
Transmission	Accelerator pedal
Motor clutch	Accessories
Engine clutch	Fuel tank
Lock up torque convertor	Environmental conditions
Driver display	Power batteries state of charge
Power battery on board & off board charging	
Power battery compartment temperature	

Table 7-2. Control System Functions, Sensors and Actuators

Function	Purpose	SENSORS Function	SENSORS Description	ACTUATORS Function	ACTUATORS Description
I.C. Engine	Control speed and power of I.C. engine plus monitoring of engine operating system parameters and fuel level	RPM sensor*	Magnetic pickup and interface	Throttle position	Two solenoid valves for hydraulic ram - small
		Throttle position	Potentiometer and interface	Starter	Starter relay
		Oil pressure	Pressure Transducer and interface	Fuel pump	Fuel pump relay
		Fuel level	Potentiometer and interface		
		Water temperature*	Thermistor and interface		
		Oil pressure	Thermistor and interface		
I.C. Torque Converter	Control torque converter luck-up	Lockup	Microswitch showing lock-up	Lock-up control	Solenoid valves for lock-up clutch actuation
		Engine speed*	Magnetic pick-up and interface		
		Converter out RPM*	Magnetic pick-up and interface		
I.C. Engine Clutch	Control I.C. engine clutch	Clutch	Pressure switch showing clutch open or closed	Clutch control	Two solenoid valves controlling hydraulic ram
		Clutch input RPM*	Magnetic pick-up and interface		
		Clutch output RPM	Magnetic pick up and interface		
Electric Motor and Controller	Control speed and power of motor	Field current armature	Current sensor and interface	Field current	Serial I/O line to field controller
		Armature current	Current sensor and interface	Armature current	Relay to turn on/off
		Field voltage	Voltage sensor and interface	Armature current	Relay for starting motor
		Armature voltage	Voltage sensor and interface		
		Motor speed*	Magnetic pick-up and interface		

Table 7-2. Control System Functions, Sensors and Actuators (cont'd)

Function	Purpose	SENSORS Function	SENSORS Description	ACTUATORS Function	ACTUATORS Description
Motor Clutch Control		*Clutch input RPM**	Magnetic pick-up and interface	Clutch closure and opening	Two solenoid valves controlling *hydraulic ram*
Transmission Control	Control transmission shifting and gear selection	Transmission input RPM	Magnetic pick-up and interface	Shifting	Six solenoid actuated hydraulic valves
		Transmission output RPM	Magnetic pick-up and interface		
		Accelerator pedal*	Pot		
		Gear select*	Switches		
Monitor Driver Inputs	Control of information to and from the driver	Brake position	Potentiometers and interface		
		Accelerator position	Potentiometers and interface		
			Full out and idle		
			Switches		
			Accelerator Pedal		
			Circuit monitor		
		Transmission shift position			
		Manual input	Key-in system		
		Key position	Circuit monitor		
Power batteries, condition monitoring and battery environmental control	Control and monitor battery conditions	State of charge	Battery state of charge	Battery charger	Battery charger control from wallplug
		Battery compartment temperature*	3 thermistors and interface	Temperature	Solenoid controlling air flow
		Battery module voltage	13 voltage sensor circuits		
		On-board charger	Switch		

Table 7-2. Control System Functions, Sensors and Actuators (cont'd)

		SENSORS		ACTUATORS	
Function	Purpose	Function	Description	Function	Description
Monitor engine and accessory battery	Monitor charging and voltage of engine battery	Alternator output voltage	Alternator output voltage interface	Diagnostic Display	Indicate Vehicle Status and Problems
Environmental conditions	Control air temperature for passengers	Inside air temperature	3 thermistors and interface	A/C control	Air conditioner clutch control solenoid
		Outside air temperature	3 thermistors and interface	Combustion heater control	Combustion heater control relay plus additional actuators required for motor and engine control exist elsewhere
		Engine temperature*	1 thermistor and interface		
		Battery compartment temperature*	1 thermistor and interface		
		Environmental control switches	Circuit status monitoring		
Accessories and diagnostics	Control of accessories and diagnostic checks	Lights	Interfaces to detect bulb or circuit failures	Diagnostic display	Indicate vehicle status and problems
		Brakes	Circuit to monitor brake system failure sensor	Diagnostic interface	Interface to plug-in diagnostic box
			Brake system - pressure transducers		
			Brake pressure transducers		

*Sensor shared with another function.

In addition to its subsystem control responsibilities, the computer system is required to execute the operational strategy algorithm, which provides the operating guidelines to the control strategy algorithm. The operating guidelines include a statement as to whether the operating mode should be diesel topped by electric (DE) or electric topped by diesel (ED) and what the maximum acceptable electric motor power is. Other major tasks are to provide diagnostic capabilities and to control the driver display.

7.2.3 Time Interval Requirements for Subsystem Control

In order to provide acceptable vehicle powertrain response characteristics, the microcomputer system must, at various times, update its information on the driver demands, the current system status, and the overall operating strategy. It must then adjust the various subsystem component controls to be consistent with these inputs. The frequency with which it does each of these operations should be consistent with the vehicle performance constraints and the project goals of maximizing petroleum savings. Optimally, the project goals are accomplished through adherence to the operational strategy, in which the performance constraints have been taken into account.

In step one of the methodology the time intervals required for updating the control commands for each of the subsystems were estimated. Those for the system control commands and vehicle status data were judged using worst case conditions, in conjunction with estimated acceptable performance tolerances under these conditions. For example, in order to estimate the interval for updating the transmission shift control, we used the worst case condition of shifting from first to second gear under maximum acceleration. We considered this problem for the situation in which either the motor or the engine was the source of the acceleration. A two percent tolerance in the shift-point speed was used as the maximum acceptable error in initiating a shift. Maximum loaded slew rates of 2000 and 1000 rpm/sec were used for the electric motor and engine, respectively, under these conditions. The effects of the engine and motor speed adjustments during shifting were neglected in order to approximate the worst case. Using shift-points of 3600 rpm for the motor and 3000 rpm for the engine, the sampling times must be 36 msec for the motor and 60 msec for the engine. Of course, the motor and engine may be working together, but we feel that using the maximum slew rates and the appropriate shift-point a 25 msec transmission command update time interval will not intolerably affect the 2 percent tolerance band. Thus, in our specification for the transmission command update intervals, we have used 25 msec as the update time interval.

The estimated update time intervals for the individual subsystem components are shown in Table 7-3. It should be noted, however, that the actual time intervals used will have to be evaluated once the control system is matched with the actual powertrain hardware to provide optimal performance for maximizing petroleum savings and satisfying the driver's responsiveness requirements.

Further, the use of the "Executive VOS (Vehicle Operational State)" software philosophy will allow the time intervals to be adjusted easily, as is described below.

7.3 SOFTWARE ALGORITHM CHARACTERIZATIONS

As was shown in Figure 7-1, the second step of the methodology was to characterize the algorithms which conform to the operational requirements. In order to consider the nature of the algorithms, we had to consider the context in which they were being used. We therefore developed a software design philosophy which is discussed in the following section. The key elements of this philosophy are modularity, redundancy and diagnostic checking. These techniques have been found to be appropriate in the automotive environment.*

7.3.1 Software Design Philosophy

The operational part of the microprocessor software will be broken down in to several scenarios which we call "vehicle operational states" (VOS). Each VOS software section will run the vehicle for a particular VOS. An executive module will oversee all of the VOS and will call each section as it is needed. The sections, in turn, will call other subprograms to handle data collection and subsystem component control. This highly modularized approach to the software will provide insurance against software bugs, as well as an easy means of developing and modifying the software system. All code that is specific to one device will be local to one routine. This will isolate sections and allow sensors and actuators to be changed easily. It must be recognized that the software will go through a series of evolutionary (and perhaps even radical) changes as the control system is modified and fine tuned to provide smooth operation.

Each time a VOS software section is called, the executive module will initialize an external hardware timer. Upon return from the VOS section, the exec module will reset the timer. This timer will serve as a watchdog timer and will be set to allow ample time for each VOS section to complete its function. Should something go wrong (with either the hardware or the software) which prevents the VOS section from completing its function, the watchdog timer will time out, generating an interrupt. Control will then return to the exec Reset pulses will repeatedly be issued to reset the microcomputer for automatic restart and at the

*A few of the key algorithms are briefly described in Appendix A.

same time hardware logic will force the use of the hardware backup electronics for fail degraded operation. The control system will stay in the backup mode until the microcomputer restarts successfully and extinguishes the watchdog timer. Such an approach will prevent the processor from falling into an unrecoverable state without proper action being taken.

A partial list of scenarios is as follows:

1. Initiation — starting the vehicle
2. Starting from stop — vehicle idling or just starting out
3. Cruising — vehicle maintaining speed
4. Stopping — vehicle coming to a stop
5. Termination — vehicle turned off
6. Power battery charging — vehicle off

The number of scenarios needed will depend on the system performance characteristics. Each of these operational states may be a different module in the software, since each has different requirements and timing. The executive will call each one as it is needed and switch to another as the driver input changes. Since the vehicle can run with the electric motor, diesel engine, or both, these modes may be further subdivided into three sections each. There will also be modes of starting or stopping each powerplant.

For example, the acceleration mode requires extensive speed and rpm sampling for transmission control. In the idle or cruise modes, this may not be as critical, and can, therefore, be slowed. If acceleration is suddenly needed (e.g., the accelerator pedal position suddenly changes, or more power is needed for hill climbing) the executive module must quickly respond and call the proper VOS software section.

An alternative to the "Executive-VOS" organization of the software is to have one large control loop which is continually executing. In this approach one would execute the most often occurring or "bottom level" algorithm, then branch to a higher level where another algorithm, or part of an algorithm, would be executed. Successive branches to the higher levels would be sequenced, to guarantee the execution of all necessary control algorithms. Furthermore, the higher level algorithms would be broken into segments small enough so that a return to the bottom level soon enough to meet its timing requirements would be guaranteed. This "loop" organization of the software requires that one know fairly precisely the requirements demanded of the microprocessor control system. A change in the requirements at the lowest level, for example, could adversely affect the entire software organization. Thus, the "loop" approach is somewhat inflexible; it is difficult to modify parts of the program without affecting other parts. In the Phase II NTHV program, the detailed knowledge of the demands of the vehicle subsystems on the microprocessor system will evolve only as the vehicle is built and tested, both in the laboratory and on the road. Therefore we have chosen the "Executive-VOS" approach to the organization of the software.

7.3.2 Algorithm Characteristics

Once the executive module has decided which scenario the vehicle is in, it will set up a series of calls to various routines to perform that scenario. The algorithms will be executed in a looping fashion, with the algorithms that require the quickest control updating being executed most frequently. Any algorithms that change the VOS will do so by calling an executive routine to perform the change. This keeps the executive constantly aware of any changes, and allows it to quickly change scenarios if necessary once the routine currently executing returns to the executive. An example of the timing layout for one scenario of the system is shown in Figure 7-2. It should be noted that (consistent with the software design philosophy) the frequency that a particular control algorithm is executed is dependent on the VOS, and hence the evaluation times (Table 7-3) will vary from the generalized evaluation times which were subsequently developed for evaluation purposes. Further, shifts in the evaluation times within a particular VOS software section will generally mean only a change in the performance, rather than an inability to carry out the function.

The major software operations required to accomplish the control system functions have been identified (Table 7-4). For each of the major algorithms, the number of machine instruction operations required to accomplish the necessary task was estimated, based on Minicar's past experience with microprocessor software development projects, and assuming a hardware multiply capability.

The numbers presented in Table 7-4 under the title "Machine Instruction Operations" are not the number of instructions in the particular algorithm, but the total number of instructions executed in a call to that algorithm. The required operations are here estimated on the basis that most required operations will be done in software. It must be recognized that the number of operations required can be significantly affected by the inclusion of hardware functions, which effectively preprocess the data and thus

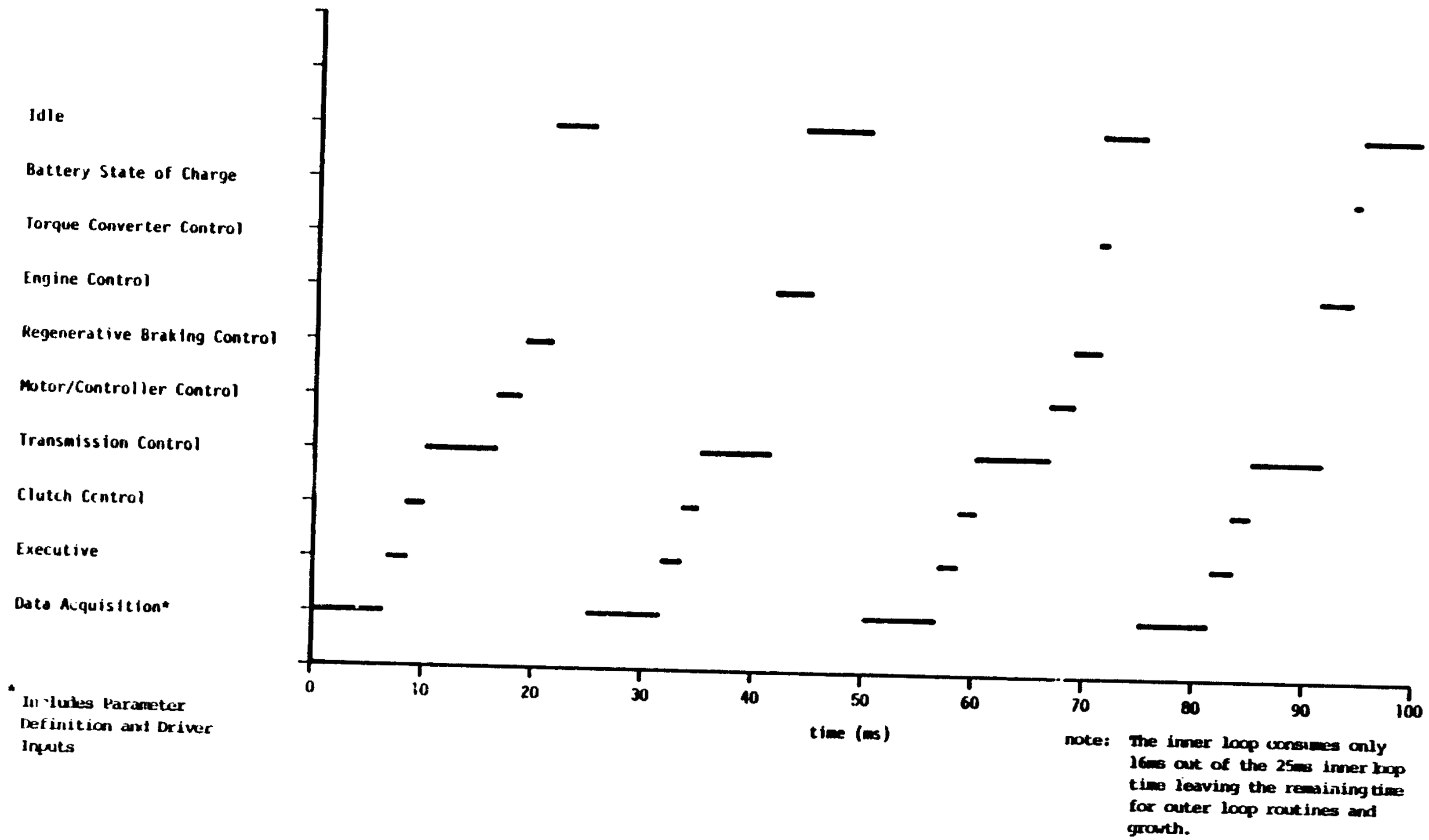


Figure 7-2. Example of Controller Software Timing Layout in a Selected Executive-VOS Operating Mode for Use on Powertrain Controller Microprocessor (#1)

Table 7-3. Summary of Estimated Command Update Time Interval Requirements

Algorithm	Output Response Time (sec)
Driver Inputs	.025
Parameter Definition	.025
Executive	.025
Clutch Control	.025
Transmission Control	.025
Motor/Controller Control	.050
Engine Control	.050
Regenerative Braking Control	.050
Torque Converter Control	.075
Battery State of Charge	.100
Display Control	.500
Accessory Control	.500
Diagnostics & Self Test	1.000
Operational Strategy	1.000

Table 7-4. **Summary of Preliminary Design Software Algorithm Characteristics**

Algorithm	Machine Instruction Operations	Processor Time (Sec)[a]	Memory Size Requirements[b] (Bytes)
Clutch Control[c]	400	.0016	100
Transmission Control	1600	.0064	4000
Driver Inputs[e]	600	.0024	1000
Parameter Definition[d]	1000	.0040	250
Battery State of Charge	100	.0004	200
Engine Control	600	.0024	2000
Motor/Controller Control	600	.0024	2000
Executive	500	.0020	1000
Torque Convertor Control	100	.0004	50
Regenerative Braking Control	600	.0024	1000
Display Control	3000	.0120	3000
Accessory Control	1500	.0060	2000
Diagnostics & Self Test	8000	.0320	4000
Operational Strategy	8000	.0320	4000

a. Assumes 4×10^{-6} sec. per instruction.
b. May not include all data storage requirements.
c. Only needed in some executive-VOS operational modes.
d. Much of this function is done in hardware.
e. Three samples are taken and stored at 0.008 ms intervals. This algorithm then operates on the samples. The time required to read and store the samples is negligible.

significantly reduce the work a processor must do. The time required of a CPU to accommodate the algorithms was estimated using a conservative average instruction time of 4×10^{-6} sec. These algorithms were laid out in a timing sequence under the condition of a worst-case VOS in which all were required. It was found that some distribution of the processing functions might be desirable, in order to maintain margins of flexibility, and that as much preprocessing of the data as possible (for the purposes of filtering and averaging) should be done prior to the use of the data in the algorithms themselves. The preprocessing could be done either in hardware or in peripheral processors. In addition to the operational characteristics of the algorithms, we also estimated the number of bytes required for each algorithm. While these numbers are shown as memory requirements, it should be noted that substantial sharing of code will likely occur, since each algorithm will use many routines common to other algorithms.

7.4 SELECTION OF SYSTEM DESIGNS AND CANDIDATE PROCESSORS

The next major step in the methodology was to select a preliminary design candidate microprocessor for use in the system. In order to do this we considered a number of factors; the major ones were

- Microprocessor performance and capabilities
- Hardware design philosophy
- Software design philosophy (discussed in Subsection 7.3.1)
- Environmental conditions
- Cost considerations
- Computer system design alternatives

7.4.1 Microprocessor Performance and Capabilities

A review was made of a number of available microprocessors, including the Harris 6100, 1800, 8080, 8085, 8086, 8088, Z-80 Z-8000, 6800, 6801, 6809, TMS-9900, and Fairchild 9440. The processors were compared on the basis of data size, addressing space, instruction set, addressing modes, maximum clock frequency, execution times of typical instructions, and execution times of simple operations (such as an interpolation). Table 7.5 gives this information for each of the 13 processors listed. We considered the availability of appropriate development tools - for example, a development system, an In-Circuit Emulator card, a cross-assembler and cross-compiler for some high level language. One final consideration is that the chips must be available in

Table 7-5. Characteristics of 13 Microprocessors

μP	Type	Data Size (bits)	Address Space (bytes)	Maximum Clock Frequency (MHz)	Typical Instruction Time Range (μsec)[c]
Harris 6100	CMOS	12	4K	8	5-11 @ 4 MHz
CDP1802	CMOS	8	65K	5[a]	3.2-4.8 @ 5 MHz
8080	NMOS	8	65K	2	2-9 @ 2 MHz
8085	NMOS	8	65K	5	1.3-5.75 @ 3.1 MHz
8086	NMOS	16	1M	8	0.4-6.0 @ 5 MHz[d]
8088	NMOS	8	65K		
Z-80	NMOS	8	65K	4	1-5.75 @ 4 MHz
Z-8000	NMOS	16	8M	4	
6800	NMOS	8	65K	1	2-12 @ 1 MHz
6801	NMOS	8	65K	1	2-4 @ 1 MHz[k] typical
6809	NMOS	8	65K	1	
TMS 9900	NMOS	16	65K	3	3-10 @ 3 MHz
Fairchild 9440	I^3L	16	65K	12[b]	

μP	Multiplication: Method (Hardware/Software)	Multiplication: Number of Bits	Multiplication: Number of Clock Cycles[e]	Multiplication: μsec[h]	One-Dimensional Interpolation: #Clock Cycles[j]	One-Dimensional Interpolation: μsec[h,j]
Harris 6100	S	12	1700	425	1000	250
CDP1802	S	8	1246	249	690	138
8080	S	8	390	195	215	108
8085	S	8	390	125	215	69
8086	H	8 or 16	124+EA[f,g]	~25	167	33
8088	H	8 or 16				
Z-80	S	8	390	97.5	215	54
Z-8000	H	8, 16 or 32	72[f]	18		
6800	S	8				
6801	H	8	10	10	59	59
6809	H	8				
TMS 9900	H	16	60	20	186	62
Fairchild 9440	H	16				

[a]RCA literature gives 6 MHz, but Reference 25 reports that users have experienced design problems when using this chip above 5 MHz.

[b]May want to avoid the excessively high speeds and operate at 4 or 5 MHz.

[c]Reference 25.

[d]Typically 2.90 exclusive of multiply and divide.

[e]For software multiplication these times should be regarded as a rough estimate.

[f]Assume 16 x 16 bit multiply.

[g]EA is the time required to calculate the effective address. These times will range from 0 to 18 clock cycles (see Reference 26).

[h]Same clock frequency as instruction time range column.

[j]These times should be regarded as a rough estimate.

[k]Hardware multiply time 10 μsec.

single unit quantities. No special consideration was given to CMOS since, as noted in Subsection 7.4.1, the noise problem will have to be solved on a much broader scale, and that solution would allow the use of an NMOS processor. Our estimates of the processor time required to handle the control functions suggest that an 8-bit processor will be sufficient, provided that it have an 8 x 8 bit hardware multiply instruction. Thus, there is no need to go with the added complexity and cost of a 16-bit processor. Also, the 16-bit processors do not usually have memory efficient instruction sets, and would tend to use more memory.

It appears that the Motorola 6801 is well suited to automotive control system application. This is supported by General Motor's use of the 6801 in automotive control applications. GM presented a paper to the SAE National Congress in March 1979 on their custom microprocessor.

7.4.2 Hardware Design Philosophy

The next step in the process was to outline a preliminary hardware design philosophy. Several key issues were identified as relevant to the preliminary design and to work which can be accomplished in Phase II.

A major aspect of the hardware design philosophy is that the Phase II effort will not necessarily produce the most cost-effective production system. Rather, the principal output from the Phase II effort will be 1) proof of principle; and 2) development of control system software. The production engineering of such a system consistent with current automotive practice would take a level of effort far outside the scope of this contract. This would require custom IC's such as mask-programmable ROM, that would be high-volume items.

Another specific hardware design issue recognized is that in a production engineered system a distributed system in a master-slave configuration may be the most cost effective. In this configuration a main processor might do the bulk of the decision making, while the peripheral processors provide filtered and smoothed data, and perhaps operate units like the display or execute functions related to subsystems not directly involved in the powertrain control process. The reason this approach may be the most cost effective would be the assumption by the peripheral processors of many functions (e.g. signal filtering) which will require hardware in a development/demonstration system. In a production system it is desirable to minimize the number of components peripheral to the processors. Further, in order to implement peripheral processor systems, the exact hardware and software functions to be performed by the peripheral processors must be very well defined.

Additional major considerations for safety and reliability are:

1) Feedback
There must be methods (directly using sensors or indirectly using other information) to indicate that the appropriate action has taken place. Example: send command to close clutch - read microswitch to see if clutch is closed or check speeds on each side of clutch to see if they track.

2) Dual Control
Two lines are needed to control each actuator. Thus, noise on one line will not cause inadvertant actuation.

3) Redundancy
The hardware must have the capability to run without microprocessor. If the microprocessor fails, the vehicle will still be driveable to enable movement to a repair facility.

4) Noise Protection
Input, output, and bus lines must be protected from the external EMI and that generated in the car. This requires twisted pair shielded wiring in places, and the incorporation of filters for the lines.

5) Use of hardware filtering

6) Packaging consistent with requirements.

7.4.3 Environmental Considerations

The major environmental considerations in this application are electromagnetic interference (EMI), temperature conditions, and, to some extent, humidity and vibration effects. While the exact levels of EMI in this particular hybrid vehicle are not known in Phase I of the program, it is expected that they will be quite severe. One positive effect of using a diesel rather than a gasoline engine will be the elimination of the distributor and spark plug system. But the presence of the electric motor and motor controller will likely make the EMI problem much worse than the typical conducted 10-?0 volt noise levels in today's automobiles.

As a result, it is felt that the NTHV EMI levels will be sufficiently high that the choice between CMOS and NMOS microprocessor chips will not significantly affect the degree of buffering and isolation required for the microcomputer system; hence the selection of the microcomputer chip set can be made on the basis of the features that the specific microprocessor provides, such as its architecture, speed and instruction set. However, most of the support chips, etc., will likely be CMOS.

The temperature and humidity aspects will be a problem addressed through use of MIL SPEC chips, as required, in combination with appropriate package design and construction. The latter might include, for example, humiseal coatings, component position considerations, heat sinks, and cooling and air flow designs.

However, as a critical element of the Phase II effort the electrical conditions of the hybrid will be simulated with an actual electric motor and controller, similar to that to be used in the NTHV. Also, conducted and radiated noise will be measured. A simple development configuration of the selected microprocessor will be tested in the simulated conditions, to assure that the selected microprocessor will work in the hybrid vehicle EMI environment.

7.4.5 Cost Considerations

As mentioned in the discussion of hardware design philosophy, we recognize that time and cost limitations will require that the system will not be production engineered, as it would be in current automotive practice. We could take cost into consideration when selecting a microprocessor for use in the development/demonstration system. But, because of the well known strong decay of electronics cost with the recovery of development costs and mass production, the consideration of current microprocessor costs is not appropriate; their prices generally reflect how long they have been available and the number sold. Thus we have not considered electronics costs in the selection of components, except to the extent that units with highly divergent capabilities are recognized as likely to result in cost differences. We therefore gave some consideration to the reasonableness of the processor in its particular application and whether its capabilities far exceed what is needed.

7.4.6 Computer System Design Alternatives and Selection

A number of computer system configurations were considered for use in the NTHV Phase II effort, including both single and multiprocessor designs. The designs considered were

1. Single processor system
2. Distributed processing system - equal processors
3. Distributed processing system - master-slave
4. Distributed processing system - independent processing
5. Distributed processing system - independent master-slave.

Each of these is discussed below.

An example of the single processor system is given in Figure 7-3, using a Motorola 6801 Microprocessor. The major drawback of this system is the requirement that the processor be fast enough to read all the data, make all the decisions, and control all the peripherals in the allotted times required to provide vehicle performance. The advantage of this system is low cost and simplicity, for there are no communication problems between processors.

All distributed processing systems use two or more processors. The system using equal processors has two or more which share periphials and memory. An example of this type of system, again using Motorola 6801 Microprocessors, is shown in Figure 7-4. In this system, each processor can read data, make decisions, and control the peripherals, thus sharing the load and reducing the speed requirements for each. The disadvantages of this system are that the processors must now do extensive communication with each other, and that they must contend for the same memory and peripherals.

In a master-slave configuration there is one master processor to direct all the slaves. This is shown in Figure 7-5 using a 6801 Microprocessor and 6801 slave processors. The slave processors control all the functions and monitor their performance, while the master takes the data, makes the decisions, and sends commands to the slaves to set new performance. The communication is all back and forth to the master, so the communication problem is not extreme. However, each function to be put in a slave processor must be well defined, so as to preclude the need for communication with other functions that are in other slaves or in the master. Although this may be desirable in a production environment, it is extremely hard to achieve in a development program.

Using two independent processors, the load can also be split up. One processor does one part, and the other does another independent part. There is no sharing of memory, and communication goes through input/output ports. It is not as fast as a shared bus system, but there are no shared memory contention problems. Also,

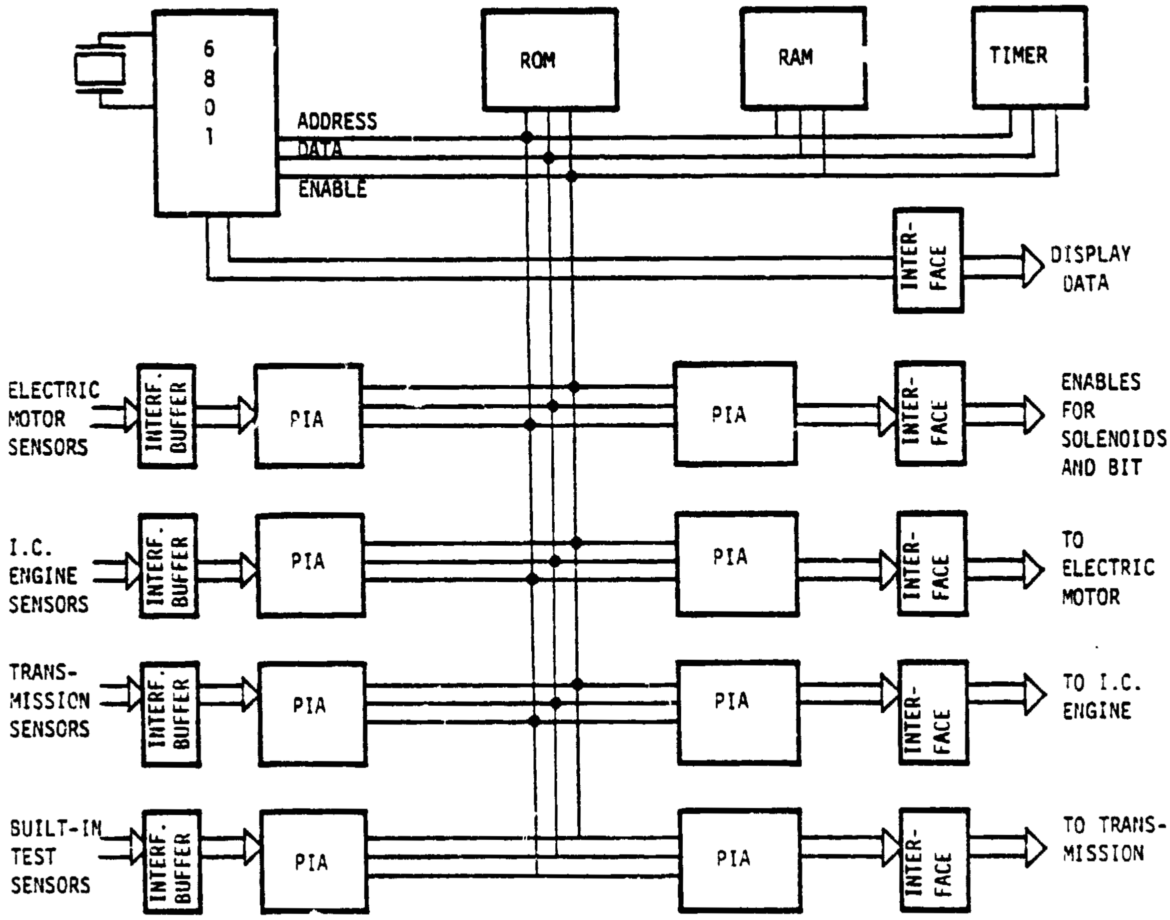


Figure 7-3. Single Microprocessor System

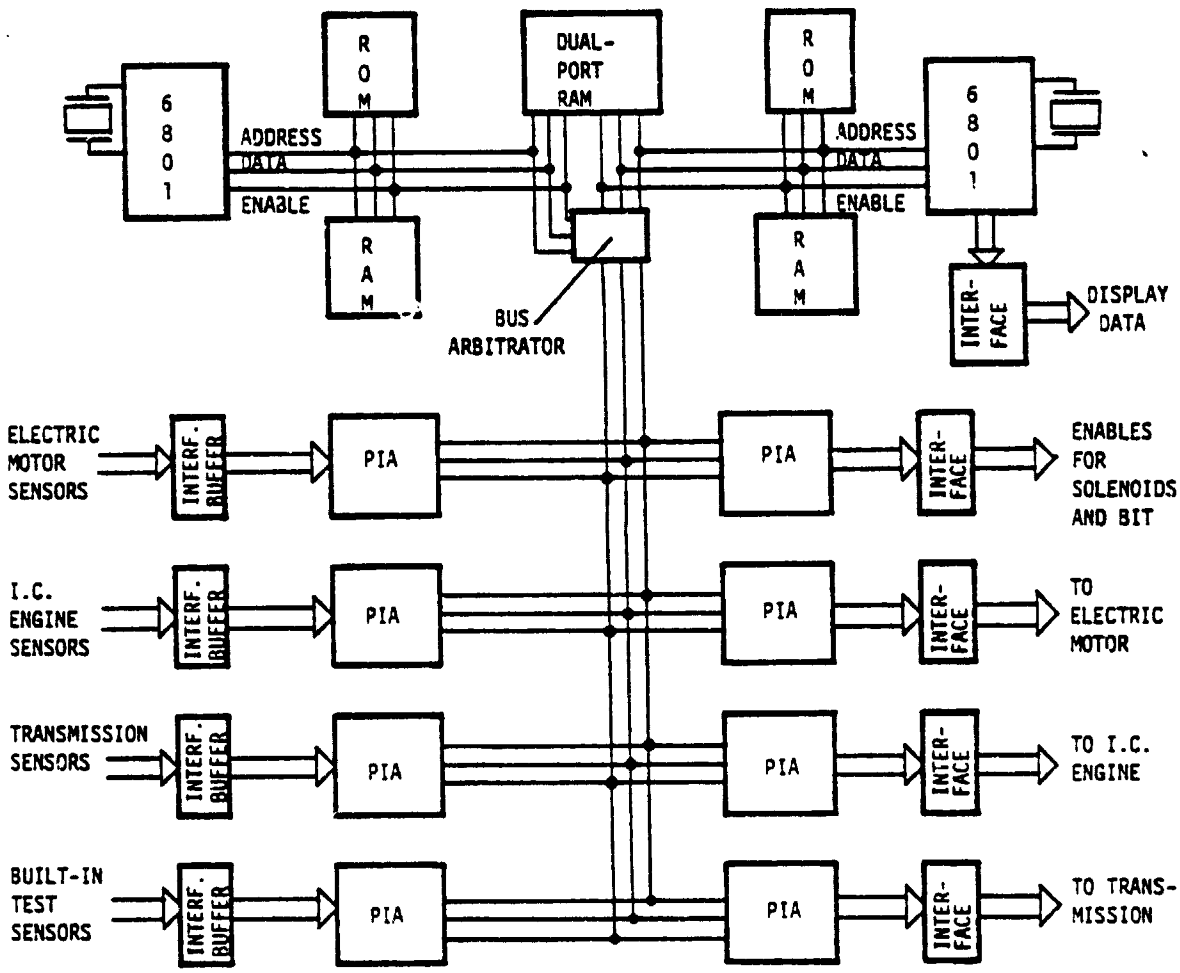


Figure 7-4. Dual Microprocessor System

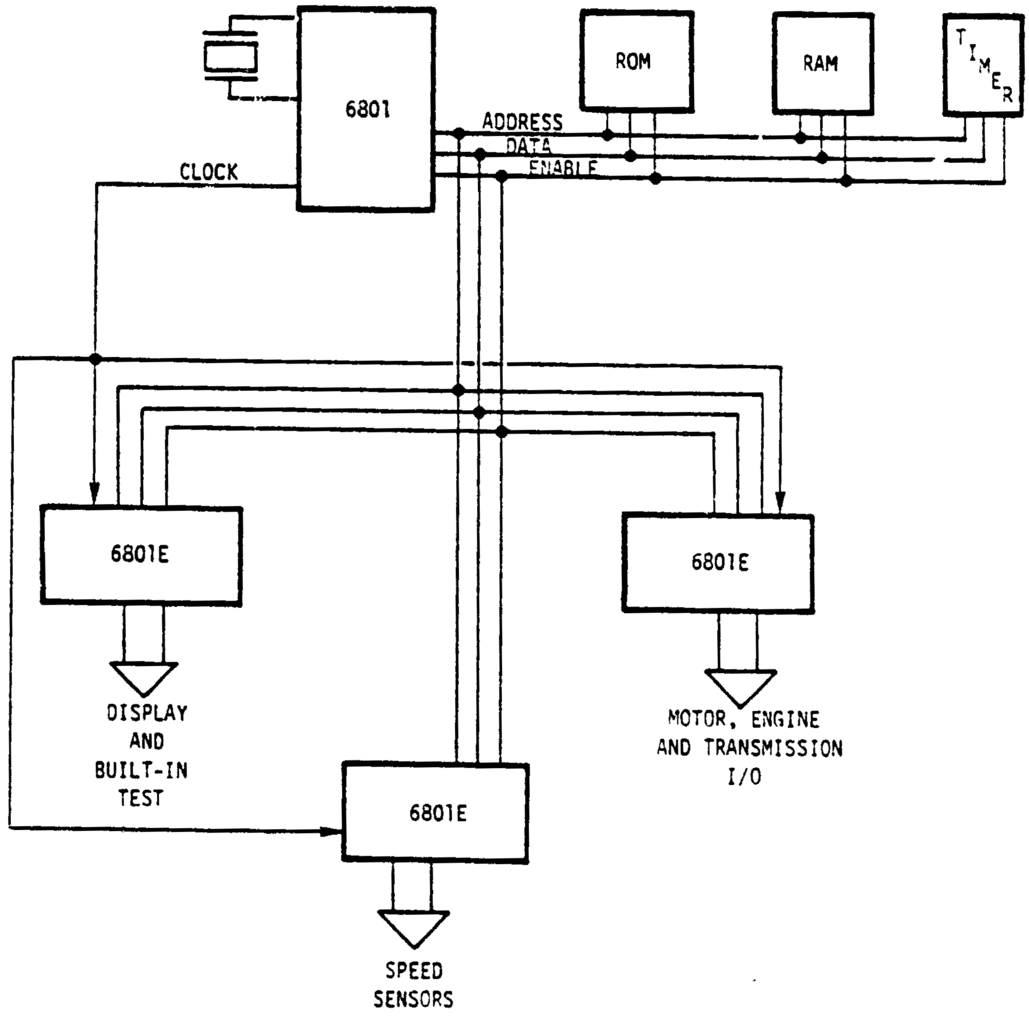


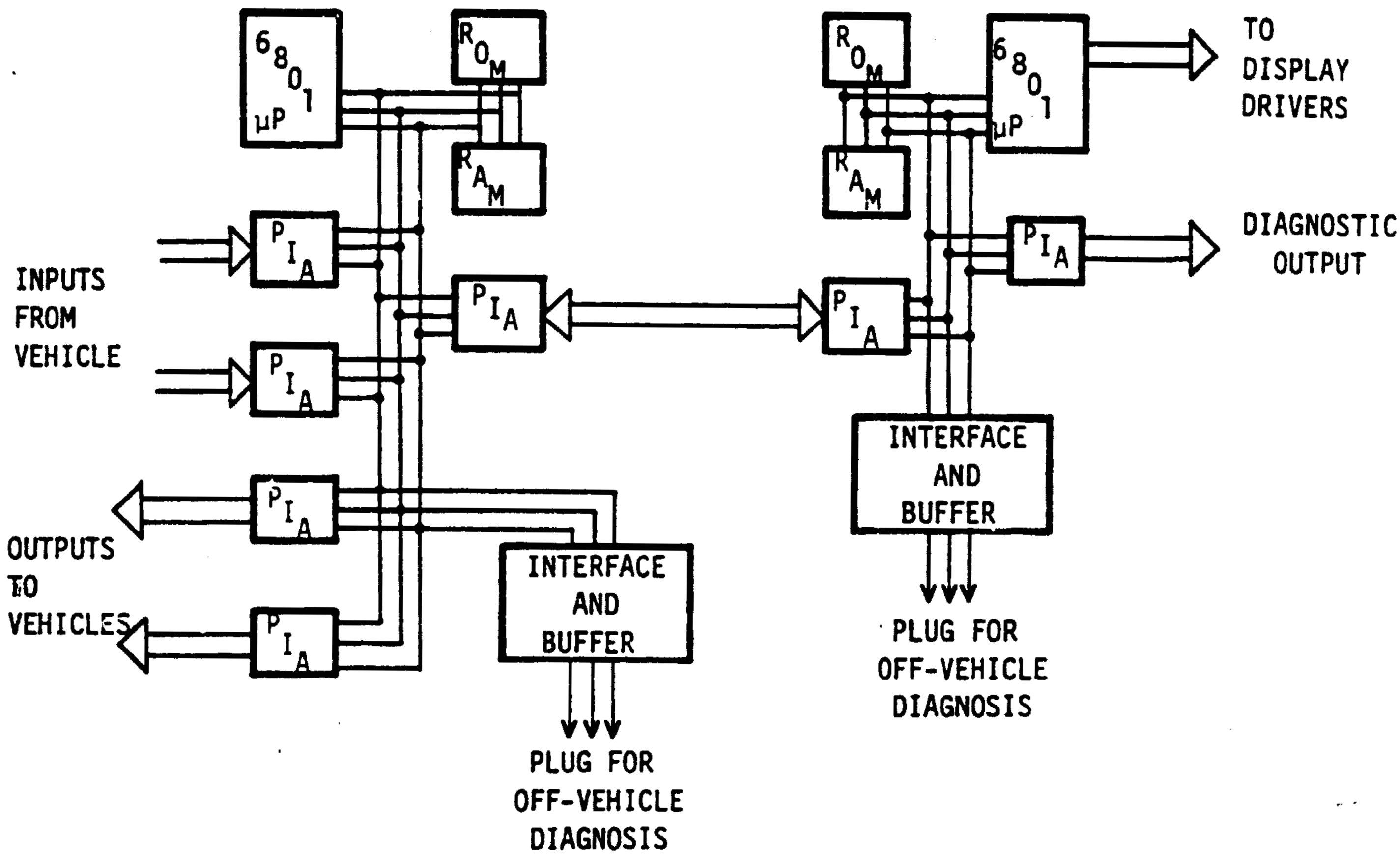
Figure 7-5. Microprocessor and Slave System

one system can constantly check on another. This system is the one we have chosen for the development of the hybrid vehicle, and is shown in Figure 7-6 using two Motorola 6801 microprocessors. Although identical microprocessors are not required, we use them to eliminate the development and familiarity problems with two different microprocessors. This approach gives us a low risk because each processor is independent. There is no constraint on the supervisor software to meet response times required by the controller. Each system can be developed almost separately, and can run separately without requiring one to stop so the other can run. Thus, changes made to one system produce little if any changes on the other.

Finally, we considered the possibility of combining a master-slave system with an independent processor system. This is shown in Figure 7-7, using the 6801 and 6801E microprocessors.

This type of system may turn out to be a good candidate for a production system, once all the functions are well defined. The independent processor gives the additional advantage (over the master-slave system) of further isolating specific sections of the program and checking on the operations of both processors.

A few comments about our choice of a distributed independent dual processor system are in order here. We recognize that both the control strategy and the operating strategy will very likely continue to develop, as experience with the actual vehicle is gained. Thus, in order to guarantee adequate processing power and ease of software system development and modification, we have specified a dual processor system. It may well be, that a single processor will be adequate to handle all of the processing once the development work is completed. But this question will not be answered definitively until the vehicle is working and finalized. Only at that time will it be possible to specify a system for production. Further, we believe now that a distributed master-slave (DMS) configuration would be the most cost effective in production. However, the development of a DMS system should not be attempted until after all of the control strategy and operational strategy software is fully written and checked out and the vehicle is developed and tested (at least to the prototype level).



(For use in the Phase II development effort)

Figure 7-6. Microprocessor System Concept for the Phase II Development/Demonstration Effort

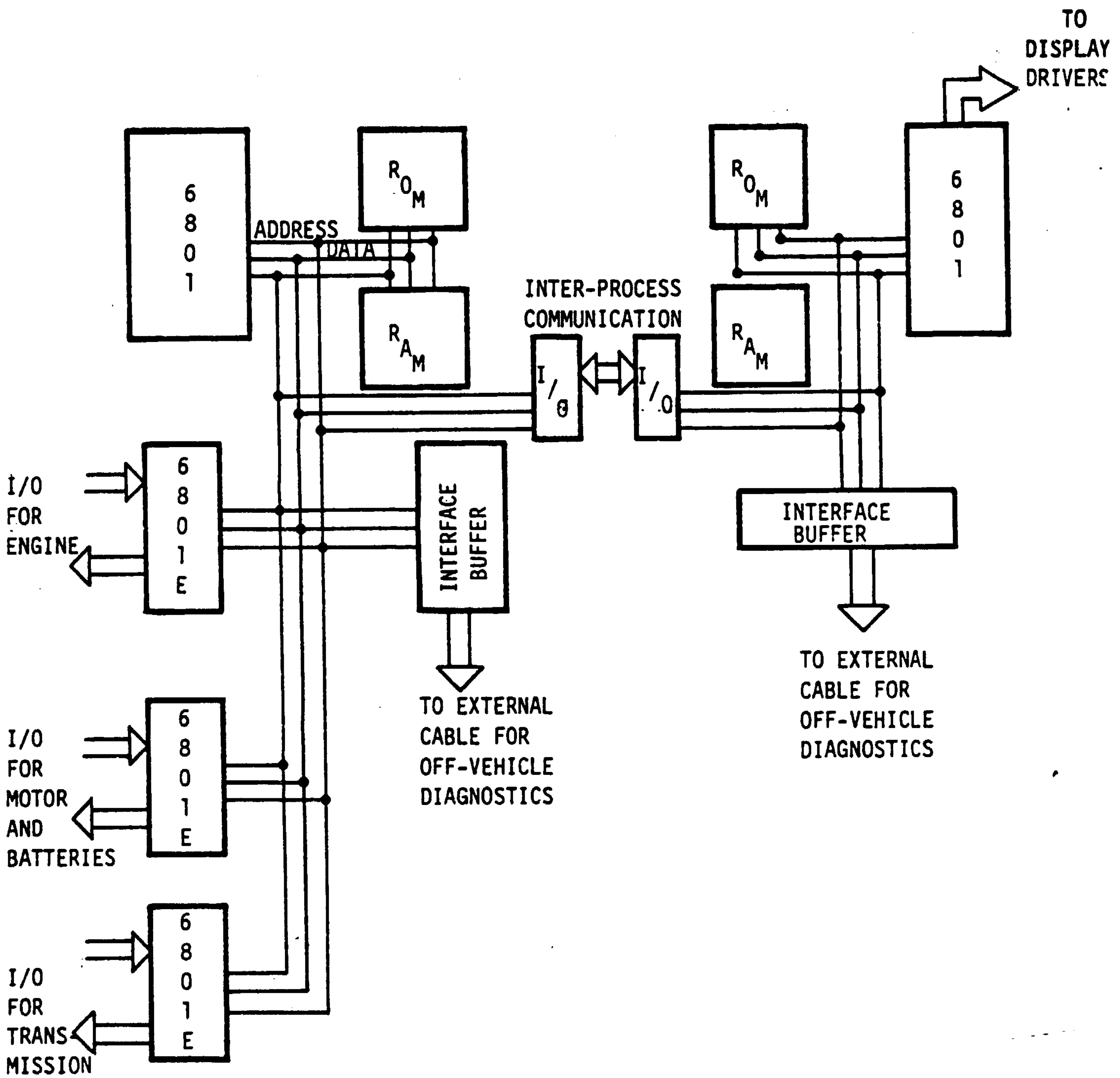


Figure 7-7. NTHV Microprocessor System Concept for a Production System

7.4.6 Processor Selection

The microprocessor unit for the distributed independent processor development/demonstration control system was selected based on the data considered with regard to the computer system configuration, costs, environment, the hardware design philosophy, the software requirements, and the software design philosophy.

The family selected was the Motorola 6800 NMOS family, and, specifically, the 6801 unit.

The primary reason for using this microprocessor is that it is a fast (2 to 4 msec typical instruction execution time), 8 bit microprocessor with built-in hardware for an 8 bit multiply. (See previous Table 7-5.) In our preliminary analysis, the algorithms will use many multiplications and interpolations. This microprocessor will perform an 8 x 8 bit multiply resulting in a 16 bit product in 10 microseconds, while the same multiply done in software would take about 250 microseconds. The only other microprocessors that are able to do a hardware multiply (which is important in order to accommodate the software requirements) are the 16 bit microprocessors, such as the INTEL 8086, and the 8088 (a 16 bit processor converted to two 8 bit processors) which may be too slow for use in the development/demonstration system. The 16 bit based devices are much more expensive than the 8 bit microprocessors, and this added expense is unjustified, since the additional capability that these devices allow is not needed for this program. Also, they are not usually as memory efficient as the 8 bit processors.

The 6801 also offers advantages for the NTHV program itself. Motorola has a development system (the "EXORciser") which will enable us to more easily design the NTHV hardware and software. It will allow us to perform in-circuit emulation and checkout of the vehicle system using the development system debug and system analysis boards. We could also compile and simulate the software as it is being written, checking each routine against the other routines and the system hardware. The use of the MPL high-level language offered by Motorola to aid in our software effort will be very helpful in accomplishing the project on schedule. It will allow faster development, checkout, and modification of the software, while making it more readable. The cross-compiler, cross-assembler and simulator that are written in ASCII Fortran will be installed on our in-house VAX 11/780 computer. This will allow several programmers to simultaneously write, compile, and check out their software on one large time-sharing system, without having to wait for time on the development system. For final checkout, the software will be transferred to the development system via EPROM's. The combination of the EXORciser, the in-circuit emulator, the MPL language, and the capability of using the VAX all contribute to a low-risk effort.

Features of the 6801 Microprocessor

The Motorola 6801 microprocessor has many features which make it optimum for the NTHV microprocessor system:

- It has a wide variety of addressing modes which apply to most of the instructions. This capability allows the programmer to write smaller programs and to use larger addressing space, making such things as table lookup easier and faster.
- It has the 16 bit operations of load, store, add, and subtract. This can be done in one instruction, instead of the several required on 8 bit machines.
- It has an 8 x 8 bit hardware multiply that is much faster than an equivalent software multiply.
- Instructions have been optimized to reduce the amount of memory used and the execution speed.
- If it becomes necessary to use two or more processors in a multi-processing mode, this microprocessor has a hardware busy signal built in. This will make it easier to interface two or more processors.
- It has auto-incrementing and auto-decrementing capability, which allows quick searching through tables and block operations.

Some disadvantages of this microprocessor are

- It has no single bit operations other than AND, OR and EXCLUSIVE OR. Testing and setting individual bits must be done by a small amount of software, but this is not often used.
- It has no undefined operation code traps. If the program fails and an illegal instruction is executed, it will not interrupt the processor. However, there seems to be no 8 bit microprocessors with this feature. Thus, a special timer will be used instead to reset the system in the event of a problem causing the microcomputer to "hang".

- It has only two 8 bit accumulators (which can be used as one double register), and two 16 bit index registers. This means more memory references must be used.

Of the processors examined, the 6801 seems best suited for the job; so we have selected this processor. Of course, the area of microprocessors is progressing rapidly; we will pick the 6801 as our preliminary design choice, but will constantly review new technology in the event that newly available products and support equipment are more appropriate for this application. Table 7-6 shows all its features, addressing modes, and instruction speeds. Appendix E shows some benchmark results between the 6801, 6800, and 6809. Figure 7-8 shows the 6801 microcomputer and its peripheral controller configuration, the 6801E.

Table 7-6. Summary of MC6801 Features

SUMMARY OF FEATURES

Hardware

- M6800 Bus Compatible
- Single 5V Power Supply
- 8-Bit Word Size
- 16-Bit Address Field
- TTL-Compatible Inputs and Outputs
- On-Chip Oscillator/Driver
- On-Chip 16-Bit Dual Function (Input and Output) Programmable Timer
- On-Chip Serial Input/Output
- On-Chip 128 Byte RAM
- On-Chip 2K Byte ROM
- Vectored Priority Interrupts for Timer and Serial I/O
- Four Programmable Input/Output Ports
- Eight Hardware Programmable Modes of Operation
- Mask Option for External Clock Input
- Peripheral Controller Mask Option
- EROM Version for all Mask Option
- Mask Relocatable ROM Address
- Mask Relocatable RAM Address
- Programmable External Address Space to 64K
- Multiplexed Address Data Bus
- Valid Address Strobe
- On-Chip Standby RAM for 64 Bytes
- Vectored Restart
- Maskable Vectored Interrupt
- Separate Non-Maskable Interrupt
- Full Duplex Programmable Serial I/O for either NRZ or Bi-phase
- Four Baud Rate Programmable Selection for Serial I/O

Software

- MC6800 Upward Compatible Architecture
  - Two 8-Bit Accumulators
  - One 16-Bit Index Register
  - One 16-Bit Stack Pointer
- MC6800 Upward Compatible Instruction Set
  - All M6800 Instructions Included (Many contain fewer cycles)
  - M6800 Object Code Compatible

Table 7-6. Summary of MC6801 Features (continued)

8x8-Bit Unsigned Hardware Multiply
16-Bit Arithmetic Capabilities (Load, Store, Add, Subtract, Shift)
Additional Index Register Instructions (PUSH, PULL, ADD B to X)

ADDRESSING MODES

Accumulator (ACCX) Addressing - In accumulator only addressing, either accumulator A or Accumulator B is specified. These are one-byte instructions.

Immediate Addressing - In immediate addressing, the operand is contained in the second byte of the instruction except LDS and LDX which have the operand in the second and third bytes of the instruction. The MPU addresses this location when it fetches the immediate instruction for execution. These are two or three-byte instructions.

Direct Addressing - In direct addressing, the address of the operand is contained in the second byte of the instruction. Direct addressing allows the user to directly address the lowest 256 bytes in the machine i.e., locations zero through 255. Enhanced execution times are achieved by storing data in these locations. In most configurations, it should be a random access memory. These are two-byte instructions.

Extended Address - In extended addressing, the address contained in the second byte of the instruction is used as the higher eight-bits of the address of the operand. The third byte of the instruction is used as the lower eight-bits of the address for the operand. This is an absolute address in memory. These are three-byte instructions.

Indexed Addressing - In indexed addressing, the address contained in the second byte of the instruction is added to the index register's lowest eight bits in the MPU. The carry is then added to the higher order eight bits of the index register. This result is then used to address memory. The modified address is held in a temporary address register so there is no change to the index register. These are two-byte instructions.

Table 7-6. Summary of MC6801 Features (continued)

Implied Addressing - In the implied addressing mode the instruction gives the address (i.e., stack pointer, index register, etc.). These are one-byte instructions.

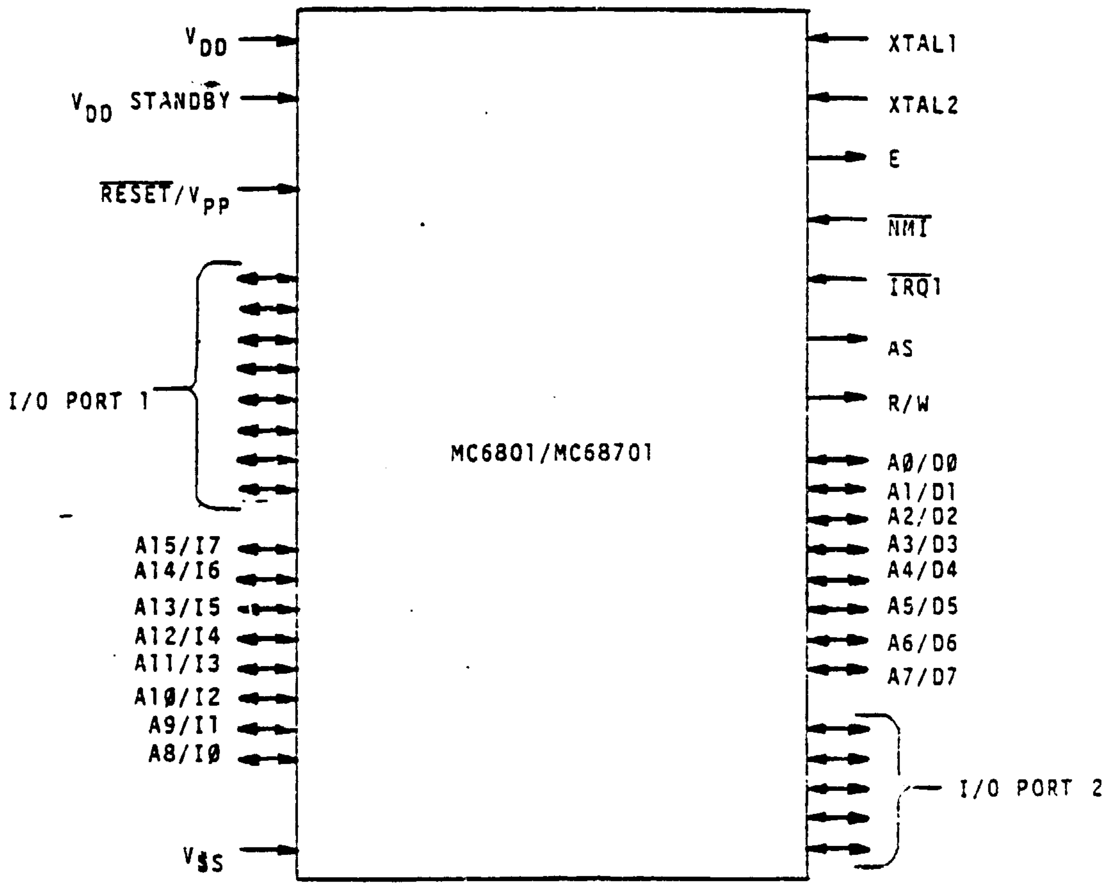
Relative Addressing - In relative addressing, the address contained in the second byte of the instruction is added to the program counter's lowest eight bits plus two. The carry or borrow is then added to the high eight bits. This allows the user to address data within a range of -126 to +129 bytes of the present instruction. These are two-byte instructions.

Table 7-6. Summary of MC6801 Features (continued)

Instruction Execution Times in Machine Cycles

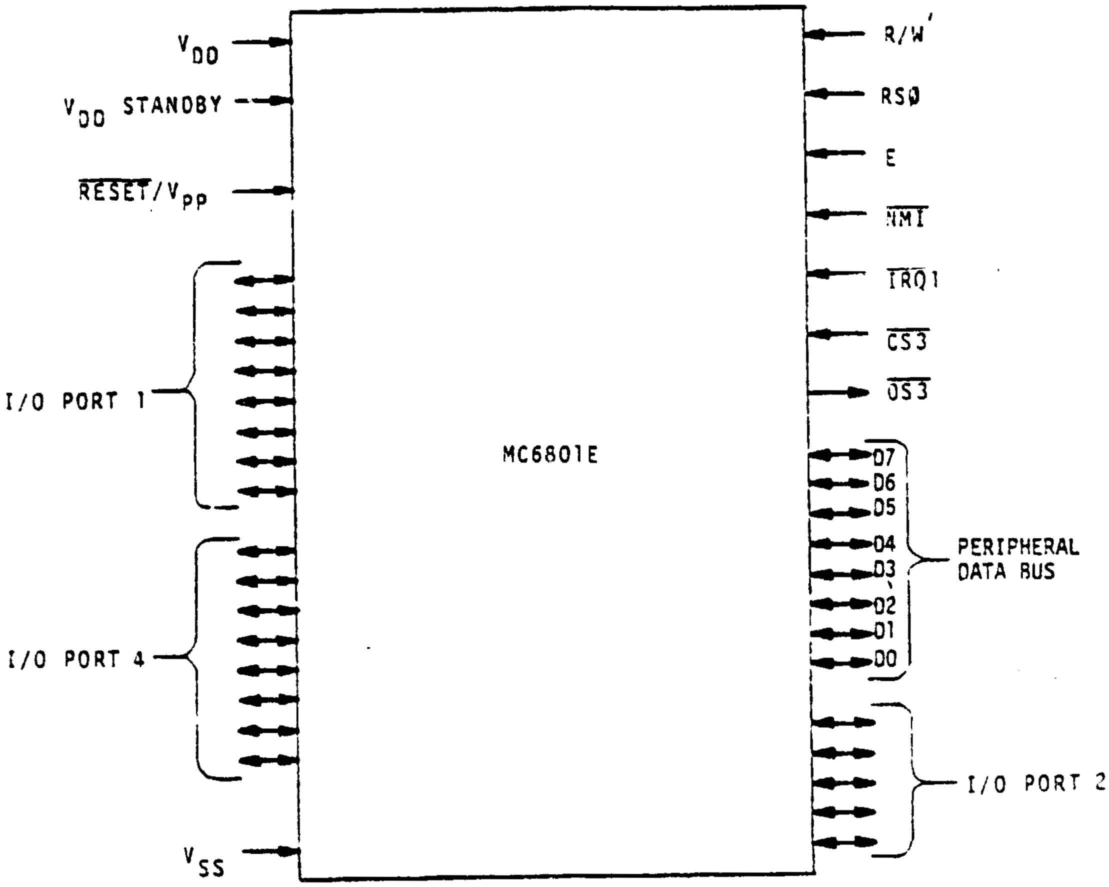
	ACCX	Immediate	Direct	Extended	Indexed	Inherent	Relative
ABA	•	•	•	•	•	2	•
ABX	•	•	•	•	•	3	•
ADC	•	2	3	4	4	•	•
ADD	•	2	3	4	4	•	•
ADDD	•	4	5	6	6	•	•
AND	•	2	3	4	4	•	•
ASL	2	•	•	6	6	•	•
ASLD	•	•	•	•	•	3	•
ASR	2	•	•	6	6	•	•
BCC	•	•	•	•	•	•	3
BCS	•	•	•	•	•	•	3
BEQ	•	•	•	•	•	•	3
BGE	•	•	•	•	•	•	3
BGT	•	•	•	•	•	•	3
BHI	•	•	•	•	•	•	3
BIT	•	2	3	4	4	•	•
BLE	•	•	•	•	•	•	3
BLS	•	•	•	•	•	•	3
BLT	•	•	•	•	•	•	3
BMI	•	•	•	•	•	•	3
BNE	•	•	•	•	•	•	3
BPL	•	•	•	•	•	•	3
BRA	•	•	•	•	•	•	3
BRN	•	•	•	•	•	•	3
BSR	•	•	•	•	•	•	6
BVC	•	•	•	•	•	•	3
BVS	•	•	•	•	•	•	3
CBA	•	•	•	•	•	2	•
CLC	•	•	•	•	•	2	•
CLI	•	•	•	•	•	2	•
CLR	2	•	•	6	6	•	•
CLV	•	•	•	•	•	2	•
CMP	•	2	3	4	4	•	•
COM	2	•	•	6	6	•	•
CPX	•	4	5	6	6	•	•
~~DAA~~	•	•	•	•	•	2	•
DEC	2	•	•	6	6	•	•
DES	•	•	•	•	•	3	•
DEX	•	•	•	•	•	3	•
EOR	•	2	3	4	4	•	•
INC	2	•	•	6	6	•	•
INS	•	•	•	•	•	3	•

	ACCX	Immediate	Direct	Extended	Indexed	Inherent	Relative
INX	•	•	•	•	•	3	•
JMP	•	•		3	3	•	•
JSR	•	•	5	6	6	•	•
LDA	•	2	3	4	4	•	•
LDD	•	3	4	5	5	•	•
LDS	•	3	4	5	5	•	•
LDX	•	3	4	5	5	•	•
LSR	2	•	•	6	6	•	•
LSRD	•	•	•	•	•	3	•
MUL	•	•	•	•	•	10	•
NEG	2	•	•	6	6	•	•
NOP	•	•	•	•	•	2	•
ORA	•	2	3	4	4	•	•
PSH	3	•	•	•	•	•	•
PSHX	•	•	•	•	•	4	•
PUL	4	•	•	•	•	•	•
PULX	•	•	•	•	•	5	•
ROL	2	•	•	6	6	•	•
ROR	2	•	•	6	6	•	•
RTI	•	•	•	•	•	10	•
RTS	•	•	•	•	•	5	•
SBA	•	•	•	•	•	2	•
SBC	•	2	3	4	4	•	•
SEC	•	•	•	•	•	2	•
SEI	•	•	•	•	•	2	•
SEV	•	•	•	•	•	2	•
STA	•	•	3	4	4	•	•
STD	•	•	4	5	5	•	•
STS	•	•	4	5	5	•	•
STX	•	•	4	5	5	•	•
SUB	•	2	3	4	4	•	•
SUBD	•	4	5	6	6	•	•
SWI	•	•	•	•	•	12	•
TAB	•	•	•	•	•	2	•
TAP	•	•	•	•	•	2	•
TBA	•	•	•	•	•	2	•
TPA	•	•	•	•	•	2	•
TST	2	•	•	6	6	•	•
TSX	•	•	•	•	•	3	•
TXS	•	•	•	•	•	3	•
WAI	•	•	•	•	•	9	•



MC6801 in the Expanded Mode

Figure 7-8. Versions of the MC6801



PERIPHERAL CONTROLLER
VERSION OF THE 6801

Figure 7-8. Versions of the MC6801 (continued)

7.5 SENSOR AND ACTUATOR CHARACTERISTICS

The sensor and actuator types required for the NTHV application were characterized in the first step of the preliminary design methodology. A tabular summary of the sensor and actuator types required is contained in Table 7-2 (previous).

In order to measure rotational frequency efficiently, we will be using a modified method which substantially reduces the required number of teeth per revolution that would normally be required for proper sensor functioning and reliability. In the approach we use, each sample will represent the time required for n-teeth to pass the magnetic pickup. This approach eliminates great problems with asymmetrical spacing between teeth and helps smooth the data. An example of such a circuit is shown in Figure 7-9.

During the early part of Phase II the actual sensors and actuators to be used will be selected after the finalization of system space and response requirements. In order to handle the noise problem, high level outputs will be desired, with the lines filtered to eliminate high frequency transients. 16-bit counters will likely be required, due to the frequencies needed to accurately clock the counting of the n-teeth sets. A list of sensors that are needed for the different functions is shown in Appendix D.

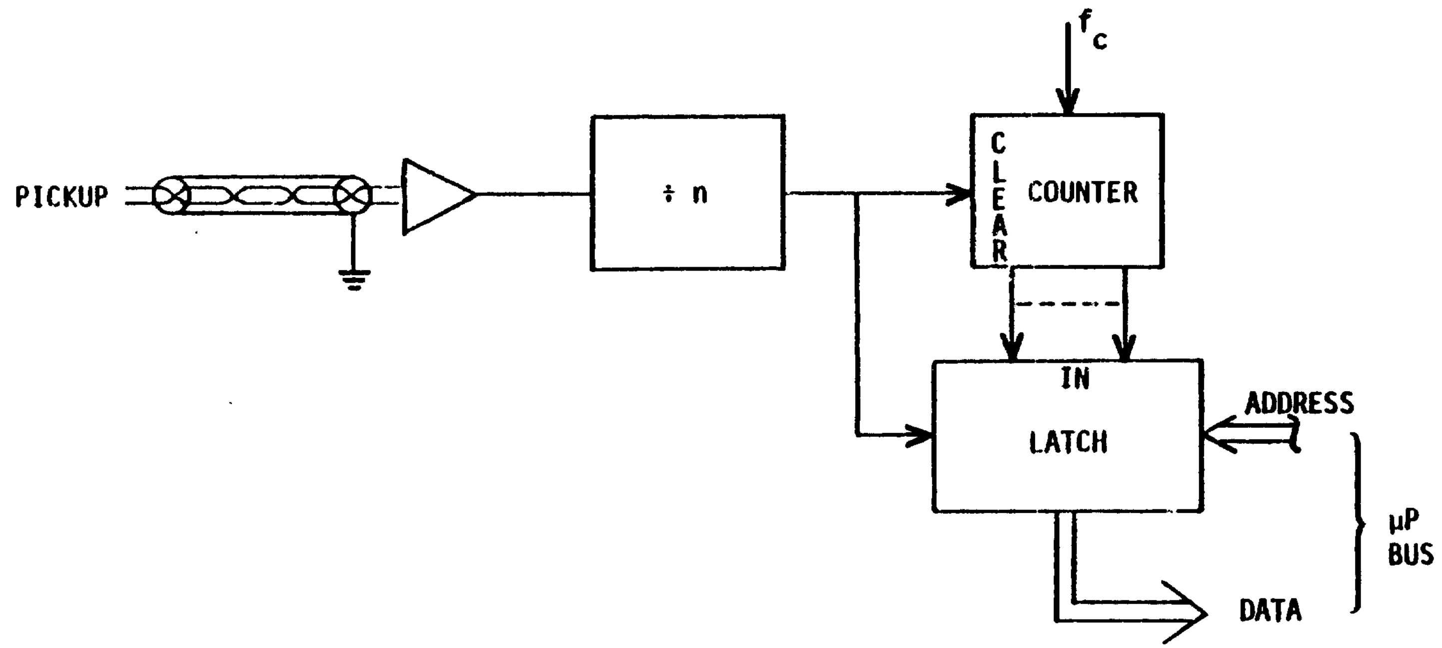
7.6 DISPLAY CHARACTERISTICS AND HUMAN FACTORS CONSIDERATIONS

7.6.1 Available Displays

Our examination of the alphanumeric display market found no product was equal to or better than the Burroughs units. Although it was not feasible to contact every possible manufacturer in this area, those contacted did not offer displays even approaching the capacity of the Burroughs.

A new display from Burroughs is now available to supplement the 32-character display used in the RSV. The Self-Scan II 20-character display has the advantage of larger character height 0.7 inch vs. 0.4 inch on 32-character display). However, the dot matrix in this display may have too much spread for the viewing distance in the NTHV. We are awaiting additional data which should provide the information necessary to choose between the units.

Of course, neither of these units is ideal. A large quantity of stimulus information is being packed into a limited space. For that reason, General Motors is privately developing a specialized



,ure 7-9. Sensor Circuit

display, and, in the near term, there may be available new displays of the vacuum flourescent type which provide very large character heights. These would be very desirable, since they require lower voltages than the high voltage gas discharge displays (for instance, the Burroughs). Also to be considered will be the Beckman Displays and the vacuum flourescent displays from Industrial Electronic Engineers.

7.6.2 Ambient Light Filters

Optical Coating Labs produces a low cost filter that is compatible with the Burroughs unit and can minimize display wash-out due to glare and the sun's rays. The increased contrast may also enhance the display readability in all light conditions. The range of viewing angles is uncertain at this time; we will soon receive additional information. The use of an optical filter may reduce the need for an extensive hood over the display.

7.6.3 Suggested NTHV Displays

The following is a suggested list of NTHV displays, along with appropriate comments. The panel locations for these displays can be established according to the zoning criteria in NHTSA Contract DOT-HS-800-742, with the following two exceptions: (1) some displays may appear on the Burroughs unit that would normally appear elsewhere, e.g. temperature, and (2) the positioning of the driver air bag in the steering wheel hub may prohibit a strict adherence to the suggested zoning.

Diesel Fuel and Battery Pack State of Charge

These displays are complementary and should appear together. If the state of charge is low, the diesel fuel level must be sufficiently high, and vice versa. The Burroughs dual-reset bar graphs would display this information nicely. The two bar graphs would be oriented horizontally and parallel, to the left of the steering wheel, and would be microprocessor controlled.

Mode Change Indicator

Although an advance warning of a subsequent mode change seems appropriate for the NTHV driver, it does not appear feasible for the following reasons:

1. The mode change may actually be unobtrusive.

2. An advance warning requires a programmed delay which cannot always be afforded, e.g., when a power demand is urgent.

3. Even if the microprocessor is programmed to send as much of an advance warning as it determines is possible, this would require a variable time interval between warning and onset, which would likely be very upsetting to the driver.

4. An advanced warning may cause the driver to abruptly change the pedal position and thus change the demand on the vehicle.

While an advance warning does not seem feasible, a visual message concurrent with the mode change would be helpful. This would appear as a short message on the Burroughs unit. This display may actually be a novelty for the NTHV driver, telling him what the hybrid is doing, when it is doing it, and giving him a greater sense of involvement with the vehicle.

Microprocessor Failure Warning

As part of an overall microprocessor failsafe procedure, the driver should be informed immediately of a microprocessor failure and be instructed what to do. Depending on the actual microprocessor design, the driver may be required to manually shift the transmission into an appropriate gear following the failure, in order to continue to a repair station. The immediate demands on the driver in this event should be minimal.

Speedometer

Unlike the RSV arrangement, the use of a third Burroughs bar graph is suggested here, located above an alphanumeric display. These bar graphs can be purchased in 100 or 200 element increments and would consequently provide a near-analog speed readout. And, like conventional gauge displays, they also give the driver a sense of relative speed change or rate of acceleration.

Oil Pressure and Odometer Temperature

These displays should be digital, microprocessor controlled, and appear in the main Burroughs unit.

Engine Speed

Because of the automatic transmission, this information cannot be acted upon by the driver and is not necessary.

7.6.4 Amount of Display

One feature of the above design suggestions is the reduction of the information load on the alphanumeric Burroughs display. Compared with the RSV. the NTHV should not display the following: time, economy, amperes, fuel engine speed, or travel speed. Therefore, the density of the information will be reduced, and there will be no need to switch between an upper and lower display mode (all information will be continually displayed).

7.7 NTHV MICROCOMPUTER SYSTEM DESIGN

The computer control system we will use for the NTHV consists of two microprocessor based subsystems which communicate with each other through a pair of parallel I/O parts.

The system is shown in Figure 7-10. The left portion will control the engine, electric motor, transmission and clutches, while the right portion will run the displays, determine the operational strategy and provide diagnostic output.

All interfaces to the vehicle will be designed for self-testing and for automatic hardware backup in the event of failure (Appendix C). For example, each solenoid will be tested to insure that it did energize, whenever it was so commanded. Each reading from a sensor will be checked for reasonable values, and a count of unreasonable values will be kept. If many bad values come in over a given time, the microprocessor will consider the sensor input bad and take appropriate action. There will be watchdog timer circuits to indicate microprocessor failure or software hang up, which will force a hardware backup. If any serious error is indicated, the microprocessor systems will be overriden and some minimum backup circuitry will come into operation. The backup will give the control of the I.C. enginer and transmission to the driver. This will insure that, for any failure in the microprocessor system, the driver will not be stranded.

The Phase II hardware will serve to functionally define the production system and will be directly translatable to it. The production configuration, shown in Figure 7-7, will be carried along on paper in the Phase II effort.

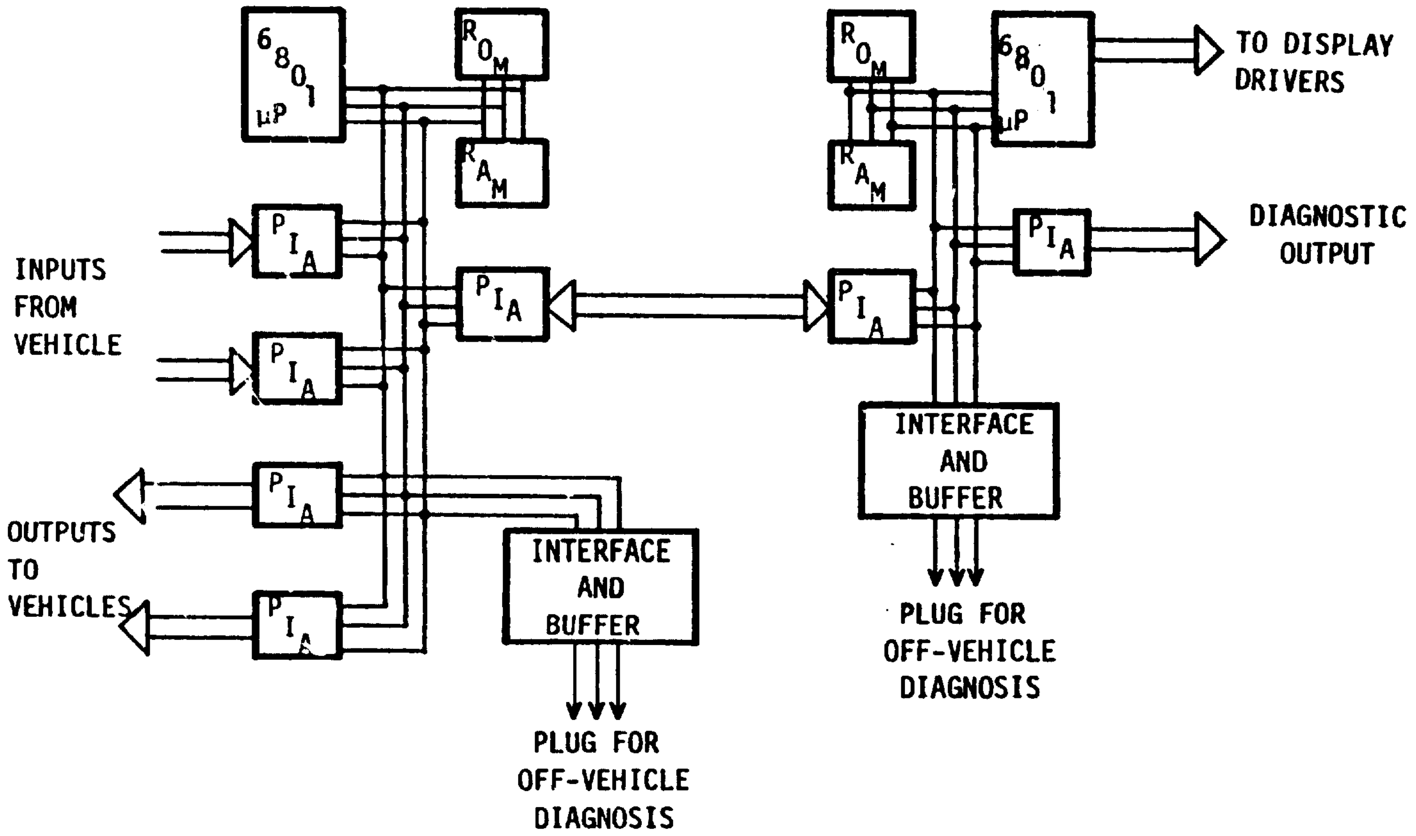


Figure 7-10. Computer Control System

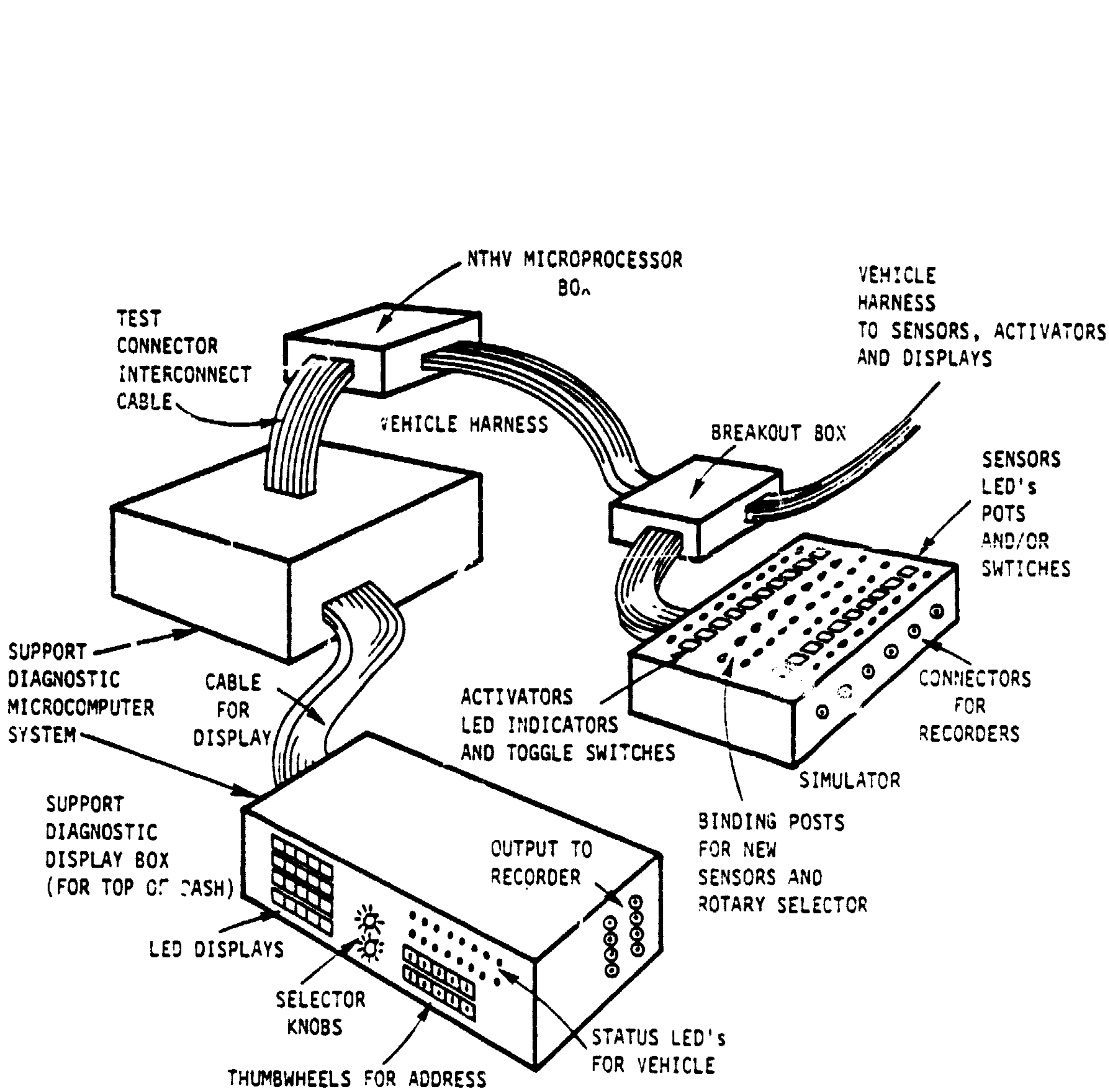
7.8 NTHV DIAGNOSTICS

There will be two different levels of diagnostics designed into the NTHV. They consist of an on-board system and a support system. There is also a diagnostic display unit carried along in the vehicle for limited diagnostic information.

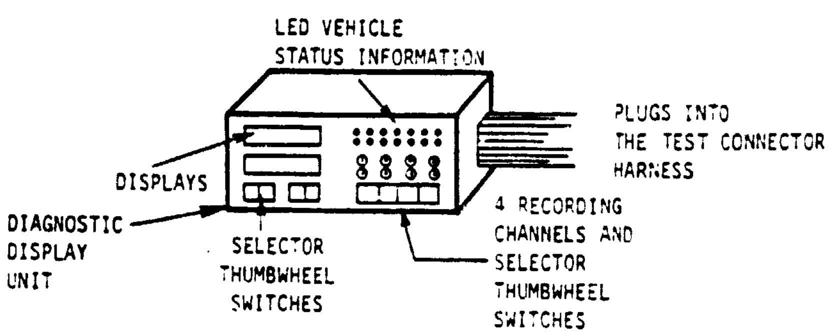
The on-board diagnostic procedure will watch vehicle sensors and actuators during vehicle operation and record any times when improper operation occurs. This will give long-term failure indication as well as information on any intermittant failure. It will also exercise the CPU and RAM to check for any failure there. A display will show any serious errors, and the cumulative errors for a given period of time will be available for observation or for access by an off-board diagnostic system. Through these three diagnostic systems, most failures will be easily detectable and repairable.

In addition to the built-in test capability, there will be a diagnostic computer for development and testing. (Figure 7-11). This unit will connect to the operational microprocessor and read the address and data bus. It will be able to pick up the address and data of any significant item (such as engine rpm, speed, etc.) and display it for the operator. It will also be able to show the vehicle state, as sent out by the computer LED's which indicate clutch on or off, gear, etc. The display for this unit will be on a flexible cable, to allow it to be placed on the top of the dashboard. It will also have a switch allowing it to interface to and display the status of the display processor. In this manner, both processors can be observed to see if they are operating correctly.

For more extensive testing, there will be a break-out box available. This will be a "T" connector which will plug in between the microprocessor system and the vehicle and connect to a simulation box. From this box, any output will be available for display or recording. Also, any sensor will be available and could be switched to a control on the box, or patched to another sensor. In this manner, all inputs and outputs to the computer are available and can be read or set to specific values for simulating the vehicle. Also, new sensors could be patched in for testing without rewiring the vehicle.



OFFBOARD SUPPORT DIAGNOSTIC TEST EQUIPMENT



DIAGNOSTIC DISPLAY UNIT

Figure 7-11. Supplementary Diagnostic Units

Diagnostic Information Display

Figure 7-11 also shows a small box which will be provided with each vehicle to allow a limited diagnostic capability (LDC). This box will plug into the vehicle harness and provide a digital display for a number of controller addresses, such as error counts, rpm, speed, motor rpm, etc. It will also display the contents of the diagnostic error counters to indicate what errors have occurred in the immediate past. It will have digital displays and thumbwheel switches to select the information to be displayed, and will have several BNC connectors for digital and analog data, and a thumbwheel selector for each. The LDC instrument will allow critical data to be sent out to a recorder for later analysis by a computer. There will be a digital-to-analog converter for converting data to be observed on an oscilloscope or recorder.

7.9 POWER SUPPLY

Two important features of the NTHV powertrain control system are its single point ground, and its battery backup, both of which will provide increased system reliability. The preliminary design of the power supply system is shown in Figure 7-12.

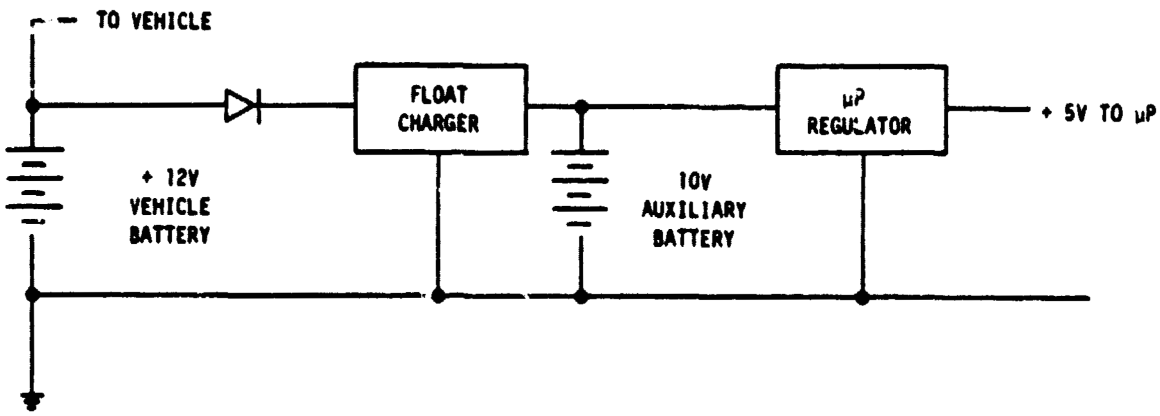


Figure 7-12. Power Supply

7.10 POWER AND GROUND DESIGN CONSIDERATIONS

The selected NTHV microcomputer control system power and grounding approach is shown in Figure 7-13. The vehicle battery (a non-propulsion source) has been chosen as the prime power source. Suitable gauge wire will be routed directly to its terminals. A separate, lower capacity microcomputer battery/charging circuit was chosen to protect against severe voltage drops brought about, for example, by starter motor actuation.

The direct battery terminal connection eliminates the effect of voltage drops in other vehicle cables associated with engine and accessory power circuits. Also, this represents the minimum voltage drop point during ICE cranking.

The engine/transmission, vehicle frame, and vehicle body are connected to the battery via a single point ground in the engine/transmission structure. This single point ground represents a compromise due to the adverse cost penalties associated with a single point connection directly at the battery terminal. The battery cable will be sized to the ICE starter motor requirements. Due to the potential inclusion of electrically non-conducting vibration and shock isolator NTHV body mounts, both the frame and the body will be tied to the above single point ground with braid straps. These straps will have a 4 to 1 length to width ratio.

The prime power wiring to the NTHV microcomputer control system will be twisted at a minimum of 10 turns per foot and will be

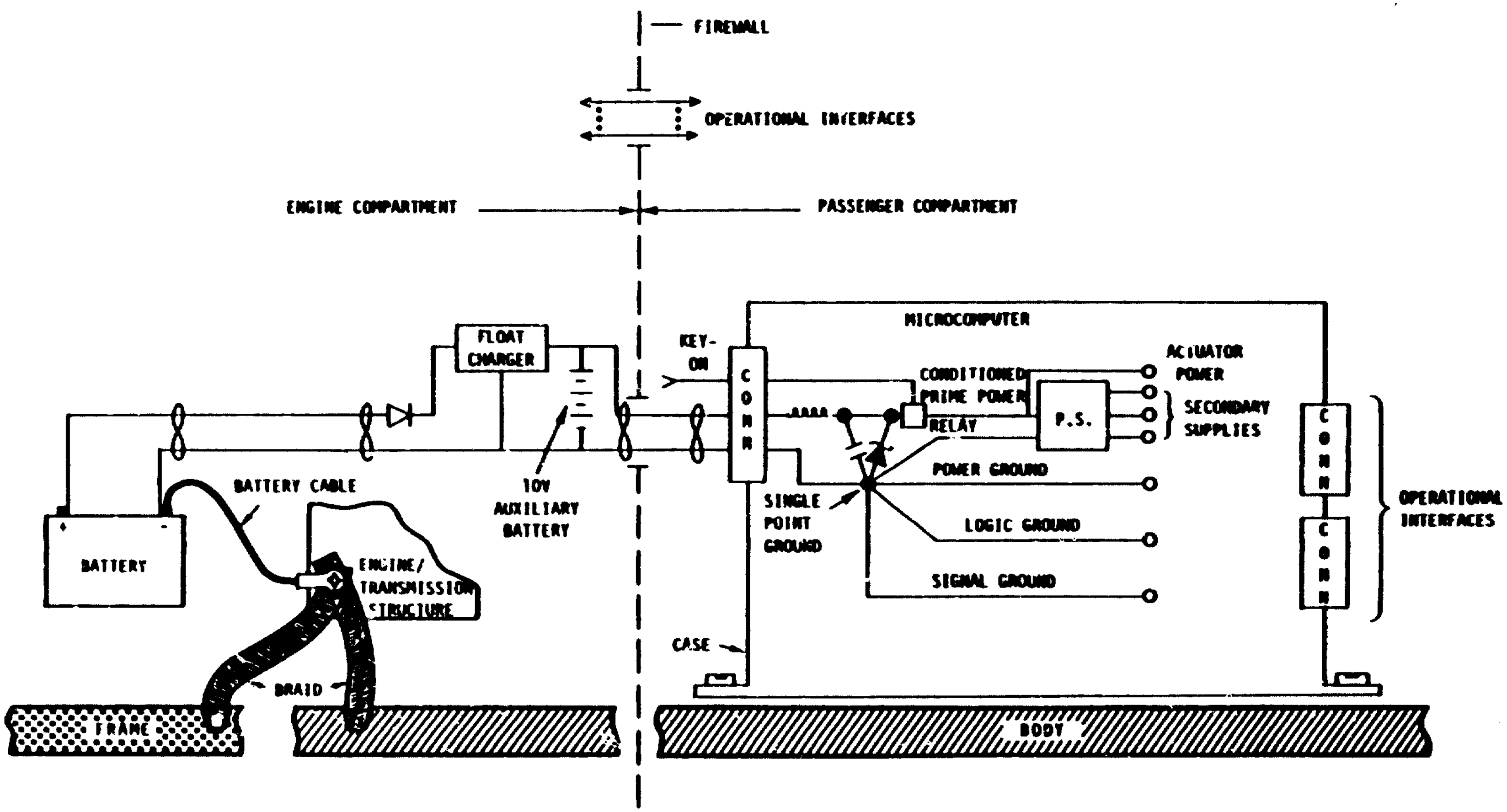


Figure 7-13. NTHV Microcomputer Control System Power and Ground Structure.

routed as close as practical to metal structure, both in the engine and passenger compartments. A separate power connector, physically separated from operational connectors will be provided at the microcomputer enclosure. The prime power wiring will also be physically separated from the rest of the microcomputer cabling. These techniques will significantly reduce electromagnetic interference effects within the constraints of a cost-effective design. The internal microcomputer electronics will rely on separate grounds, as illustrated in Figure 7-13. The grounds will be tied to a single point, immediately at the power connector, and the single point will not be tied to the case/enclosure. This connection has been deliberately avoided because the vehicle frame/ body does not represent a very good ground plane at high frequencies and large circulating ground currents with attendent appreciable voltage drops can be present. The separate microcomputer ground design was selected because it provides the best noise-immunity for the processor from power circuit interference and, similarly, noise-immunity for signal conditioning circuits from the processor and power circuits. The microcomputer system electronics will be partitioned among the internal ground structure, as follows:

A. Power Ground

- Secondary power supply subsystem
- Actuator and high current output drivers

B. Logic Ground

- Processor subsystem
- Memory subsystem
- Low-level I/O

C. Signal Ground

- Analog signal conditioning
- A/D converter subsystem

A filter and transient suppressor will be located in close proximity to the power connector and will be referenced to the microcomputer single point ground. A choke input filter mechanization will be used on the prime power input high. This will serve two purposes. It will reduce the potential interference both entering and leaving the microcomputer enclosure. It will also provide a series impedence for transient suppressor operation while presenting a minimum voltage drop under dc power drain conditions.

The microcomputer case/enclosure will be mechanically secured to the NTHV body or body member. This will provide an electromagnetic shield for the microcomputer electronics and will also help dissipate the microcomputer self-generated heat. High volume, cost-effective design techniques prevent the use of special metal-to-metal surface treatments. Therefore, star type lock washers will be employed to insure an acceptable electrical connection. This will also be the case for the engine/transmission and body connection points.

7.11 ELECTROMAGNETIC COMPATIBILITY CONSIDERATIONS

One of the most severe requirements for automotive electronics is reliable operation in the typical vehicle electromagnetic interference and power line transient environment. This is particularly true of present day microcomputers whose MOS-type logic interfaces are more sensitive than contemporary bipolar-type devices, i.e., TTL. The MOS device is a voltage sensitive element. This, together with the typical small geometry low input capacitance devices associated with LSI chip interfaces, results in a significant vulnerability to relatively low levels of electromagnetic interference. The prevalent electromagnetic interference energy coupling mode is through magnetic field coupling into the system cabling harness which causes significant voltages to appear at high impedance interface circuits which can, in turn, cause invalid system response. Direct coupling through the system enclosure is a second, and in most cases a third, order effect.

The choice of MOS LSI for the NTHV microcomputer control system was driven by a need for high performance, high-functional-density devices at low unit cost. These three criteria could not simultaneously be met by any bipolar device logic family. The only notable exception is IIL, which, being low voltage, current-summation logic, requires special interface devices and power supply voltages. This is further complicated by a present lack of peripheral support devices and industry multiple sources. The selection of NMOS over the wider temperature range and relatively higher noise immunity CMOS was again driven by functional density considerations. There are very few CMOS microcomputer designs at present of the performance and functional density required by the NTHV system. Additionally, the typical automotive electronic interface can experience electromagnetic induced noise of from volts to tens of volts, which is well above the response threshold of even CMOS. The magnitude of the interference problem is such that "electromagnetic hardening" of automotive electronics cannot be achieved simply by device technology selection. The selection of microcomputer chip set technology was based on

- High performance (processor throughput)
- High functional density (functions per chip)
- Low cost
- Availability of development support equipment.

Electromagnetic hardening will be accomplished at the system level as described in the following sections.

There are two additional automotive electronics interference problem areas. Power line transients are very severe in the typical modern day production vehicle. These are primarily due to solenoid and motor-type loads which are generally unsuppressed, (e.g., air conditioner clutch) or which require large currents, (e.g., starter motor). This necessitates special design considerations for vehicle power interface circuits. Also, the typical microcomputer operates over a clock frequency range of 1 to 10 MHz. The logic switching transients at these clock rates are particularly effective in coupling into other automotive accessory electronics such as radio and even tachometer circuits. This is a particularly severe problem for automotive radios which can have front end sensitivities down to 0.5 microvolts. The result is highly audible noise on AM and station blanking on FM. Again, hardening can only be effectively accomplished at the system level.

7.11.1 Design Considerations

The requirements of MIL-STD-461A, Notice 3 will be used as a design goal. These requirements will be modified to reflect the anticipated NTHV environment as indicated in Table 7-7. Notice 3 pertains to Air Force procurements and reflects more experience with imbedded digital computers in weapons systems. Specific design guidelines will be generated at the system and circuit levels for both development and prototype hardware configurations.

7.11.1.1 Radiated Susceptibility

The requirements of RS02 will be used as specified. RS03 will be modified to include the type of field strengths recently reported for automotive applications. ("Exposure of Automobiles Used by the General Public to High-Strength RF Fields", D.R. Kerns, et al, Southwest Research Institute Report 14-4379, May 76). These are due mainly to vehicle proximity to transmitter antennas, e.g. voice of America.

Table 7-7. Electromagnetic Interference Summary for Automotive-Type Environments

Requirements: MIL-STD-461A, Notice 3	Modifications	Additions
Conducted Emissions		
CE01 - 30 Hz to 20 kHz, power leads		
CE03 - 20 kHz to 50 MHz, power leads		
CE02 - 30 Hz to 20 kHz, control/signal leads		
CE04 - 20 kHz to 50 MHz, control/signal leads		
Conducted Susceptibility		
CS01 - 30 Hz to 50 kHz, power leads		
CS02 - 50 kHz to 400 MHz, power leads		
CS06 - Spike power leads	SAE J113 - Load dump, power leads - in place of CS06	
Radiated Emissions		
RE02 - 14 kHz to 10 GHz, electric field		
RE05 - 150 kHz to 1 GHz, vehicles and engine driven equipment		AM, FM, CB and radio phone band interference measurement
Radiated Susceptibility		
RS02 - Magnetic induction field		
RS03 - 14 kHz to 10 GHz	RS03 modified for 14 kHz to 500 kHz at 50 V/M and 500 kHz to 1 GHz at 200 V/M	

7.11.1.2 Conducted Susceptibility

The requirements of CS01 and CS02 will be used as specified. A special SAE requirement will be employed in place of CS06. This reflects the effect of "Load Dump", one of the most severe power transients associated with the automotive power bus.

7.11.1.3 Radiated Emission

The limits of RE02 and RE05 will be used as listed. A special requirement will be added for automotive-type radio equipment.

7.11.1.4 Conducted Emissions

The limits of CE01, CE02, CE03 and CE04 will be used as shown.

7.11.2 Cabling

The proper design of the NTHV microcomputer control system cabling will be one of the most significant contributors to electromagnetic hardness. A balanced design approach will be employed. This is considered necessary because a single brute force technique such as continuous sheath shielding with metal back-shell connectors, while providing an excellent design, is not very practical or cost effective for an automotive production-type design.

7.11.2.1 Wiring

All interface wiring will be twisted pairs. No single wire interfaces with vehicle frame/body for the return path will be allowed. All interface wiring will be twisted at a minimum of 10 and a maximum of 20 turns per foot whether shielded or not. This will significantly reduce electric field and, in particular, magnetic field coupling effects. The twisting requirement will pertain to power as well as signal interface wiring. The turns per foot criteria was chosen to be representative of present day high volume automotive production capabilities.

7.11.2.2 Shielding

In addition to twisting, shielding will be required for a limited number of critical interfaces. At a minimum, all speed pickup interface twisted wire pairs and microcomputer to microcomputer serial communications twisted interfaces will be shielded. The shielding material will be continuous and will be terminated to vehicle ground (engine, frame or body) at both ends. The shielded termination may be run through a connector pin, but the combined length of the connector pin and shield braid pigtail will not exceed 3 inches. This is required to keep the series inductance of the pigtail low to insure proper high frequency current flow through the shield to ground. The shield material will be insulated over the run of the cable and will be grounded only at the ends. Termination of the shield at both ends is required to provide good magnetic field shielding.

7.11.2.3 Routing

Power, actuator, and sensor wiring interfaces will be physically separated. The cable assemblies thus formed will be routed away from EMI generators such as the electric motor, electric propulsion batteries, and electric motor controller. The same approach will be used for susceptible devices such as radio antennas. Those cable assemblies which must interface with the above equipment will do so at right angles and with short, direct runs. All cabling will be routed and dressed as close to metal structure as possible. This will greatly reduce electric field coupling and significantly reduce magnetic field coupling.

7.11.3 Case/Connectors

No particular restrictions will be placed on the microcomputer case and connector design. The only major consideration will be a physical separation of power, actuator and sensor connectors. It is anticipated that the case will not be fully continuous due to thermal dissipation requirements. No attempt will be made to achieve a design with controlled seams and apertures. The case will follow typical automotive radio design standards with a requirement that a means be provided to achieve a good electrical bond with vehicle ground to the microcomputer case. All internal circuitry will be insulated from the case.

7.11.4 Interface Circuits

Specific design techniques will be employed to harden interface circuits. This will be accomplished in conjunction with the single point power, logic and sensor grounding system discussed earlier.

7.11.4.1 Power

A choke input filter together with a power transient suppressor will be provided at the prime power input. The inductive series element of the filter will serve to reduce the entry of high frequency transients and also will suppress any microcomputer power supply emissions. The addition of the power transient suppressor will provide a "conditioned" prime power source for actuator power as well.

7.11.4.2 Sensor

All sensor interfaces will have an analog filter (RC) incorporated into the receiver design. The RC time constant of the filter will be maximized consistent with the bandwidth of the sensor. All high speed sensor interface circuits will employ a differential receiver along with the RC filter. High frequency pulse-type sensors will rely on RC filters, differential receivers, followed by a digital filter. Ferrite bead (inductive) filters will not be used due to the lack of a practical and cost effective circuit board installation technique.

7.11.4.3 Actuator

All actuator drivers will be referenced to power ground and bypassed with a tantalum (energy storage) and a ceramic capacitor (high frequency response). Inductive loads will have an anti-parallel suppression diode at the driver, not at the load. This will snub the inductive turn-off transient associated with the actuator and with the interface wiring simultaneously.

7.11.5 Microcomputer Design Features

Specific system design techniques at the system design and system partitioning levels will be used to insure minimum susceptibility and emissions. These techniques involve hardware and software design. The emphasis will be on operating safely through transients rather than preventing them.

7.11.5.1 Address/Data Bus Design

The NTHV microcomputer address and data bus will be restricted to one circuit card assembly. This will be the CPU circuit card. The CPU circuit card will contain no operational connector interfaces. These interfaces will be made through the I/O card(s). The address/data bus and microprocessor and memory control signals will be made available at a test connector and will be suitably buffered. The CPU assembly will contain the processor, memory, clodk, and low level interface devices. The restriction of the bus structure will reduce both the susceptibility and emissions of these MOS-type low level high speed signals. In particular, it will not allow an external direct EMI path to exist to the memory devices.

7.11.5.2

The microcomputer hardware and software will have a provision for a software driven hardware timer. The timer will be wet in the software minor loop and reset alternately in the outer loop(s). Should the processor or software "hang up" for any reason, the timer will reset the system and restart the execution of the control program. The time constant of the hardware timer will be set by the longest possible disturbance tolerated by the NTHV control system without causing an irreversible action. If this restart is not immediately achieved, the system will go into automatic hardware backup.

7.11.5.3 Software/Hardware Initialization

All microcomputer outputs will be conditioned with the system reset signal. The presence of reset will initialize the output circuits to a non-control or a safe state. Software will continuously set all software programmable devices (I/O ports) and every major loop. This will insure that in case of a noise induced runaway, the microcomputer system will time out, safe itself, restart, and resume the operational program.

7.12 CONTROL SYSTEM INTEGRATION

As shown in Figure 7-14, the microprocessor based distributed processing system controls a large number of vehicle subsystem components. The 7 by 8 by 6 inch computer box will be located in the passenger compartment on the right firewall.

Figure 7-15 provides an expanded view of the powertrain control sensor and actuator locations. Table 7-8 presents a summary list of various aspects of the vehicle electronic control system.

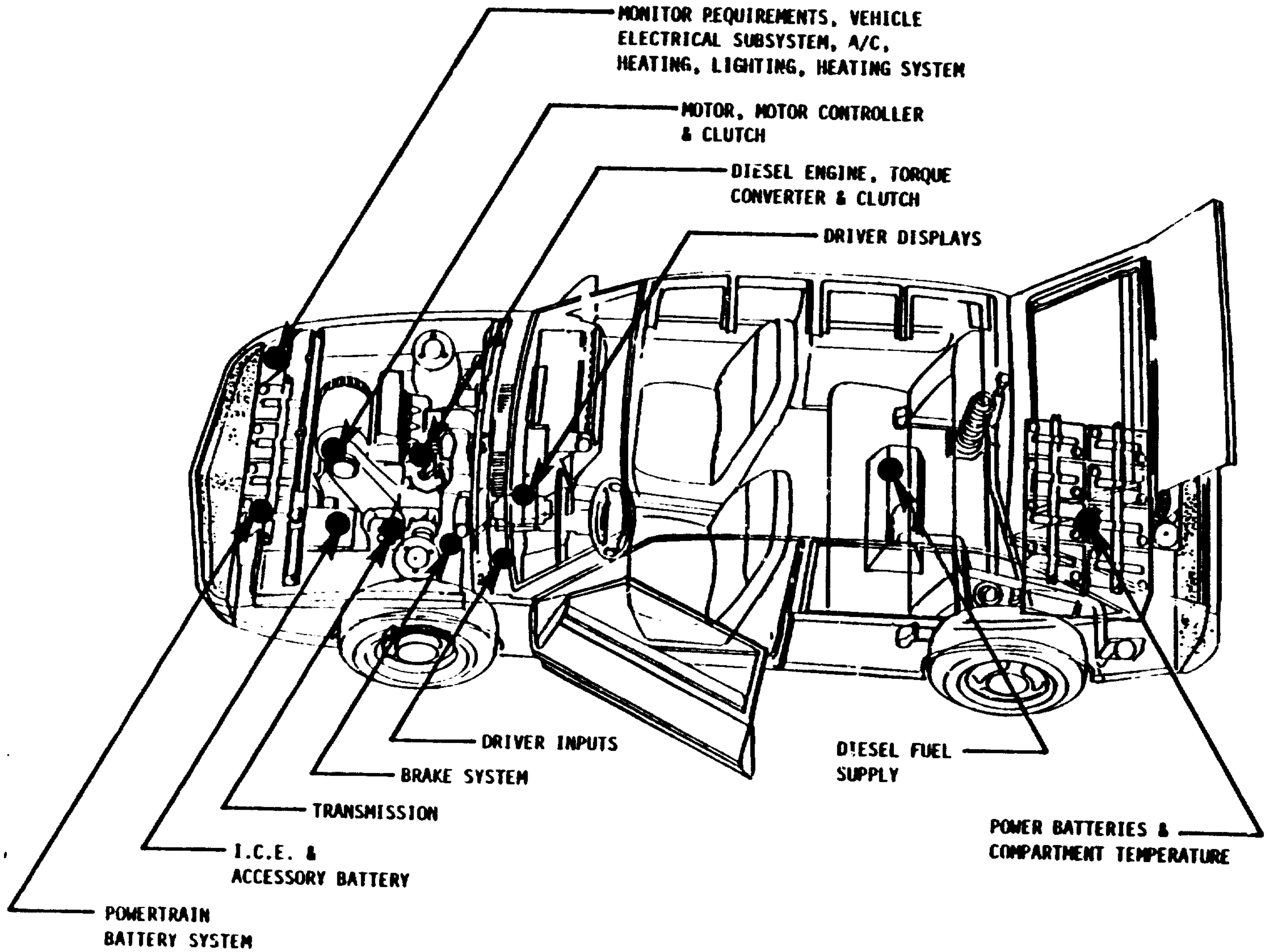


Figure 7-14. Components Controlled or Monitored by the Computer System

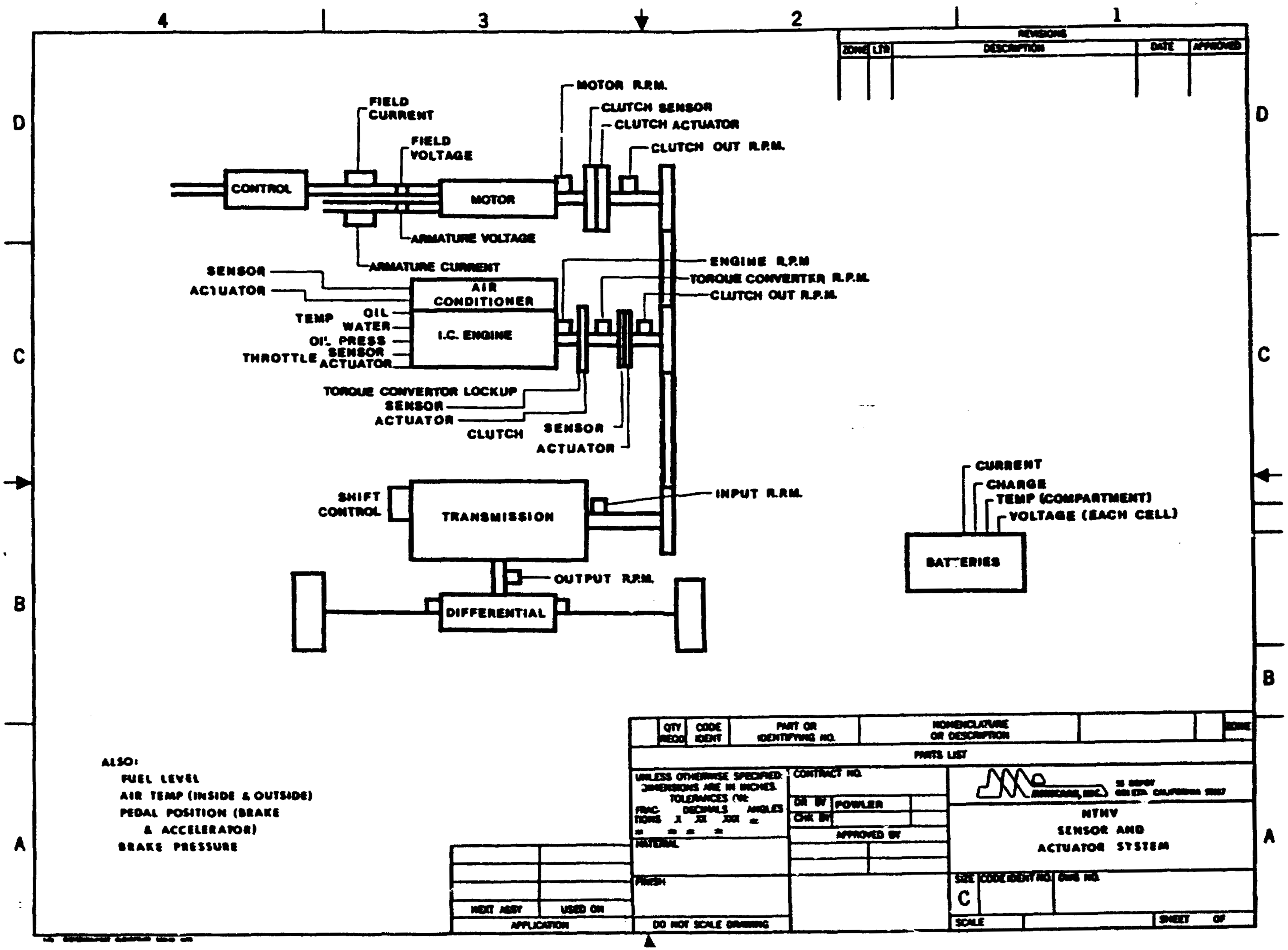


Figure 7-15. Sensor and Actuator System

Table 7-8. Summary of the NTHV Electronics System

I. Microprocessor System

A. Microprocessor
1. Motorola 6801
2. Two Separated Systems
3. Communication Through I/O Port
4. Development System
5. Vehicle Simulator

B. Memory
1. 64 k EPROM Distributed Between the Two Subsystems
2. 4 K RAM Each Subsystem
3. Separate Memory for Each Subsystem

C. Support Circuitry
1. Clock
2. Auxiliary Battery Power Supply
3. Input/Output Interface and Filtering
4. Displays
   a. Vacuum Fluorescent or Gas Discharge
   b. Easy to See Light Color and Level
   c. Digital Output (or Digital Analog)

II. Vehicle System

A. Functions, Sensor Types and Interfaces
1. State of Battery Charge
2. Current
3. Voltage
4. Frequency and RPM
5. Pressure
6. Temperature
7. Position

B. Functions, Activator Types and Interfaces
1. Hydraulic Valves
2. Current Control
3. Voltage Control
4. Display
5. Solenoids
6. Relays

Table 7-8 (Cont'd)

C. Wiring
   1. Wiring Appropriate to Current and Voltage
   2. Connection and Handling
   3. Wiring Kept Near Ground Plane

III. Electromagnetic Interference (EMI)

A. Sensors and Interfaces
   1. Noise Protection Shielding
   2. Shielded Twisted Wiring Near Ground Plane
   3. Interface Isolation from Microprocessor

B. Actuators and Interfaces
   1. Noise Protection Shielding
   2. Shielded Twisted Wiring Where Needed
   3. Interface Isolation from Microprocessor

C. Microprocessor and Support
   1. Power Supply Isolation and Protection from Vehicle
   2. Filtering and Bypass on All Lines
   3. Signal Line Conditioning

IV. Reliability Analyses

A. Sensors

B. Actuators

C. Microprocessor

D. Memory

E. Wiring

F. Power Supply

V. Diagnosis and Repair

A. Built-in Test
   1. Functional Testing

Table 7-8 (Cont'd)

2. Status Display
3. Limp-Home Capability

B. Diagnosis
   1. Interface to Off-Board System
   2. On-board Limited Diagnosis
   3. Failure Modes and Frequency Stored for Off-Board System Use

C. Modular Parts

VI. Environmental

A. Devices for Full Temperature Range

B. Clamped and Supported Boards to Protect from Vibration

C. Sheet Metal Boxes and Covered Connectors to Protect from Contamination (modeled after General Motor's C-System)

SECTION 8

ENVIRONMENTAL SYSTEM

Both heating and cooling systems are required for the NTHV, the heating system for the passenger and battery compartments, the cooling system for the passenger compartment alone. Since 80% of new, U.S. built cars were fitted with air conditioning in 1978, air conditioning is an obvious requirement for the NTHV, which is intended to be a potential replacement for all similar-sized vehicles on the road.

8.1 HEATING

The heating system must perform two different functions, the obvious function of heating the passenger compartment and defrosting the windshield, and the less obvious function of heating the battery compartment. Figure 8-1 is a curve showing the loss in battery capacity with lowered temperature. The battery system will lose 50 to 60 percent of its capacity, if the battery temperature drops to -20°C. Since this much loss would require an increase in the

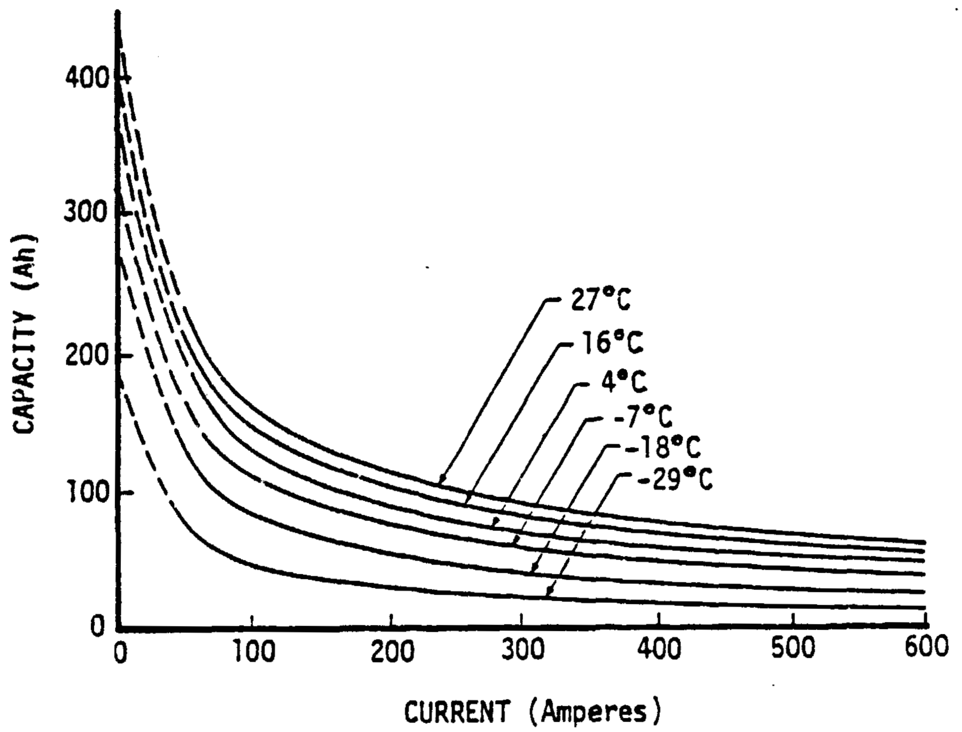


Figure 8-1. Effect of Temperature on Battery Capacity

use of diesel fuel, there should be provision for keeping the batteries warm.

There are only two practical sources of heat while the vehicle is on the road: the diesel engine or a separate combustion heater The waste heat from the engine is the normal source for passenger heating, and performs this function very well. However, the waste heat is available only when the engine is running under load. If this were the source of vehicle heat, the engine would not only have to be running, but sharing a significant portion of the driving load all of the time the heat is being used to warm the vehicle. The engine, particularly in the colder areas, would be used to a much greater extent than necessary for vehicle driving (especially on days in which only short distances are driven).

The other potential source of heat for the vehicle is a combustion heater that burns diesel fuel directly to create heat and uses either air or water as a medium to heat the car and the batteries. The combustion heater makes more efficient use of the fuel burned, since it uses the energy for nothing but heat. However, the combustion heater is a redundent use of fuel when the engine is running.

The best heating system is the combination of a combustion heater and a system to use the waste heat from the diesel engine. A schematic of such a system is shown in Figure 8-2. In this case the heat will be provided by the engine when it is running, and by a combustion heater heating the same coolant when the engine is not running—or by both together if a rapid warm up is required. The heat in both cases will be transferred by the coolant to the standard heater/defroster of the X-body. The same coolant will be used to heat the battery compartments. Since the engine water pump is driven from the accessory drive system, and so can be run by either the engine or the motor, the coolant will be circulated through the system, even when the vehicle is being run by the motor. When the ambient temperature is low, the combustion heater can help to keep the diesel engine warm. The diesel must be kept above 40°C in order to be able to start quickly when required.

The total heating load of the vehicle will be in the range of 6 to 9 kW (20,000 to 30,000 BTU/hr), at -20°C. The combustion heater should have an output in this range, so that it will not be necessary for the engine to be used when it is not required for driving.

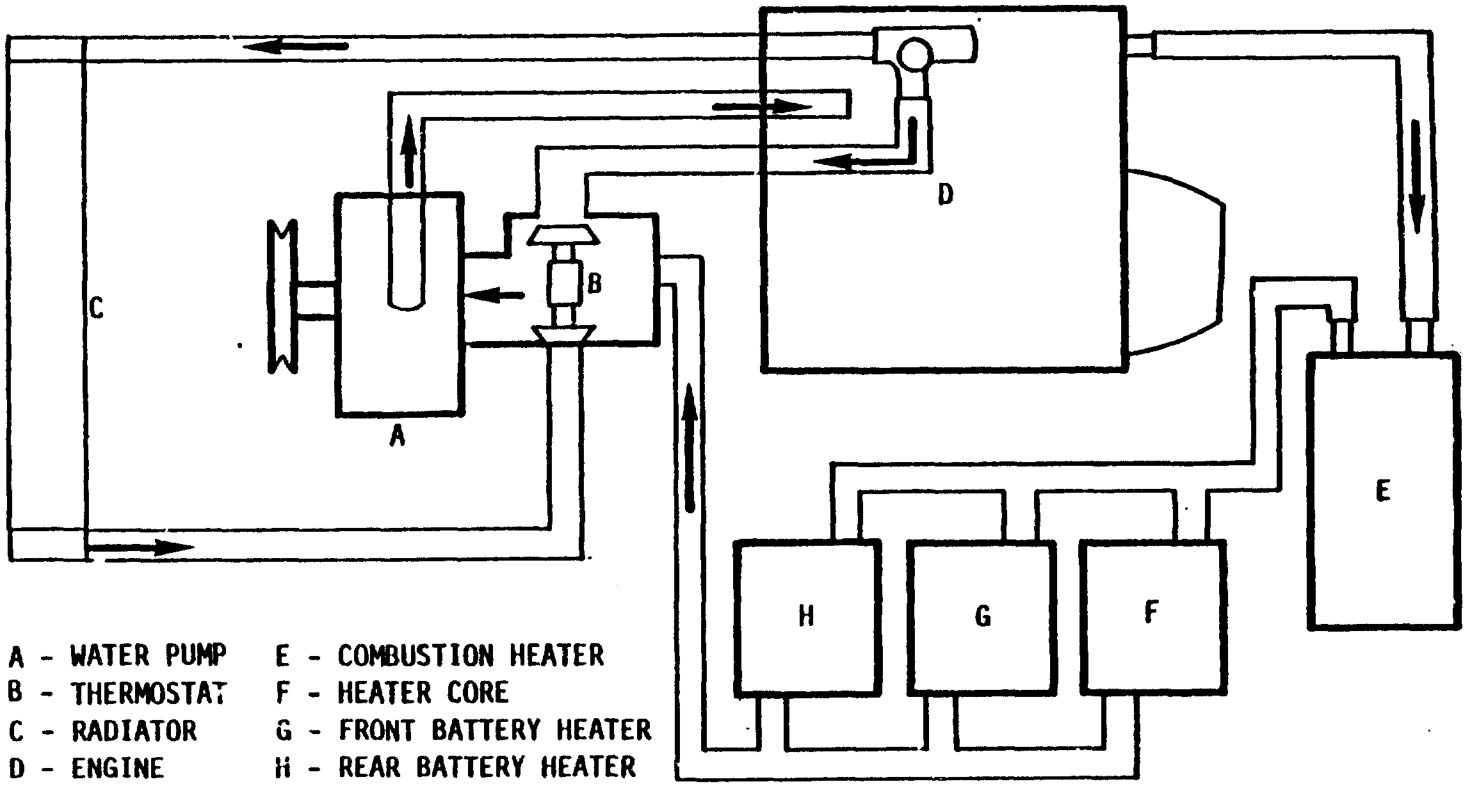


Figure 8-2. Schematic of the Heating System

The battery heating is required to keep the batteries at a satisfactory operating temperature. The amount of heat energy required to bring the batteries up to temperature from a low ambient temperature is probably too much to make this a practical operation, but electric heating elements, connected through the battery charger power cord, could provide electric heating to keep the batteries at 10°C when the battery charger is plugged in. This would let the batteries be at an adequate temperature when the battery charger is unplugged in the morning, so that at least the morning portion of the daily travel could be made with battery power. If the vehicle was parked during the day, without the charger plugged in, then the batteries would cool down toward ambient and the remaining portion of the day's travel would be primarily by diesel power (if the batteries were too cool to provide sufficient power).

8.2 AIR CONDITIONING

The only practical cooling system for the near term is a conventional compressor-powered system using R-12 refrigerant. There are no other systems available in the near term that will satisfactorily cool the NTHV. The cooling system will work with the standard X-body air conditioning.

8.3 COMPRESSOR DRIVE

The air conditioning compressor can be driven from the engine, from the motor, or from the accessory drive shaft which is driven by both. As discussed in Subsection 4.4, the high power requirement of the compressor entails that it be driven by the engine. Therefore, the engine must be running, and contributing the power, if necessary, whenever the air-conditioning compressor is needed.

8.4 EFFECTS ON OPERATIONAL STRATEGY

There are several different operational strategies that could be used in cold weather conditions. The primary problem is getting the diesel engine up to operating temperature as soon as possible, so that it will be available to provide power, if needed. The diesel engine will not start and produce power quickly if its temperature is under 40°C. The engine can be brought up to this temperature by three methods, all of which could be used under certain circumstances. These methods are: starting the engine and using it to drive the vehicle, heating the engine with the combustion heater, or preheating the engine with an electric block heater powered by wall plug electricity. If the wall plug electricity is available (when the battery charger is plugged in), the third method should be the best, since it uses no petroleum to heat the engine. The other two methods would be usable when the car had been parked in cold weather where it could not be connected to wall plug electricity. The combustion heater should heat the engine with less petroleum usage than running the engine, and would thus be the preferred method of heating the engine. The exception to this rule would occur when the engine is needed to power the vehicle.

The electric block heater could be used to keep the engine up to 40°C all of the time the power cord is plugged in, or the engine could be allowed to cool down, with the heater turned on at a preset time in the morning so that the engine is up to temperature when the driver is ready to leave. In both cases the engine would have to be insulated to minimize the heat loss. About 5 cm of fiberglass insulation would cut the heat loss rate down to a theoretical level of 50 Watts. In actual practice, the heat loss rate would be greater, but could still be kept low. The fuel injection pump and fuel lines would be outside the insulation, to eliminate vapor lock problems. Tests would have to be run on an insulated engine to discover the best method of achieving the desired engine temperature for morning running: maintaining the temperature, or letting the engine cool off and reheating it under clock control.

To run the air conditioner, the least petroleum will be used if the engine drives the compressor at the minimum acceptable speed and does not provide power to drive the vehicle. Under these conditions the specific fuel consumption of the engine is not particularly good, but the actual fuel flow is lower than when the engine is providing driving power. Therefore the diesel engine will be turned on whenever air conditioning is needed.

SECTION 9
PRELIMINARY DESIGN NTHV SPECIFICATIONS

This section presents the physical characteristics, component characteristics, performance specifications, and energy consumption measures of the preliminary design NTHV.

The dimensions of the NTHV are given in Table 9-1.

The component and subsystem weight breakdown of the preliminary design NTHV is presented in Table 9-2.

A summary of the major component specifications is presented in Table 9-3.

The preliminary design NTHV life cycle costs are obtained by using component and subsystem unit cost data information from References 27, 28, 29, and 30. The consumer costs and other performance specifications of the vehicle are given in Table 9-4.

The energy consumption measures of the preliminary design NTHV are determined by using the data on Tables 9-5, 9-6, and 9-7. The data for the manufacturing energy and the scrapped energy are obtained from Reference 31. The energy consumption measures of the preliminary design NTHV are presented in Table 9-8.

Table 9-1. Preliminary Design NTHV Dimensions

Wheelbase	265 cm
Length	493 cm
Height	139 cm
Width	173 cm
Front Tread	149 cm
Rear Tread	145 cm
Front Seat Headroom	97 cm
Front Seat Leg Room	107 cm
Front Seat Shoulder Room	143 cm
Front Seat Hip Room	140 cm

Table 9-2. Preliminary Design NTHV Weight Breakdown

Body in white with modifications	454 kg
Trim, glass, body electrical, etc.	190
Suspension System	78
Brake System	104
Steering System	34
Tires and Wheels (4)	80
Restraints	14
Heater/Air Conditioner	27
Transaxle, Modified for Motor	103
Drive Shafts	18
Diesel Engine	131
Electric Motor	91
Motor Controller/Charger	11
Traction Batteries	336
Power Harness	22
Battery Compartments	34
Microprocessor, Sensors	17
Curb Weight	1746 kg

Table 9-3. Summary of Preliminary Design Component Specifications

Weight	
Curb Weight	1746 kg
Inertia Weight	1890 kg
Dimensions	
Length	493 cm
Width	173 cm
Height	139 cm
Wheelbase	266 cm
Battery	
Type	ISOA Lead/Acid
Capacity (3 hr rate)	12.6 kW-hr
Voltage	72V
Weight	336 kg
Size	242 DM^3
Heat Engine	
Type	4-cylinder Turbocharged VW Diesel
Displacement	1475 cc
Power	48.5 kW @ 5000 rpm
Torque	119 Nm @ 3000 rpm
Maximum Speed	5000 rpm
Electric Motor	
Type	Compound wound dc
Power Rating	24 kW intermittent; 15 kW continuous
Field Control	Transistor
Maximum Speed	10,000 rpm
Transaxle	
Type	3-speed, computer controlled automatic with lock-up torque converter
Number of Gears	3
Gear Ratios - 1	2.84:1
2	1.60:1
3	1.00:1
Final Drive Ratio	2.53:1

Table 9-3 (Cont'd)

<u>Brakes</u>	
Type	Disc/drum with regenerative braking diagonal split hydraulic system
<u>Suspension</u>	
Type	Front independent; rear beam axle
<u>Steering</u>	
Type	Powered rack and pinion
<u>Tires</u>	
Type	Radial ply P205/75 R14
<u>Microprocessor</u>	
Type	Distributed processing system utilizing the Motorola 6800 processor family

Table 9-4. Preliminary Design NTHV Performance Specifications*

P1	Minimum Non-Refueled Range	
	P1.1 FHDC	718 km
	P1.2 FUDC	505 km
	P1.3 J227a(B)	413 km
P2	Cruise Speed	88 km/h
P3	Maximum Speed	
	P3.1 Maximum speed	180 km/h
	P3.2 Length of time maximum speed can be maintained on level road	5 min
P4	Accelerations	
	P4.1 0-50 km/h (0-30 mph)	5 sec
	P4.2 0-90 km/h (0-56 mph)	13 sec
	P4.3 40-90 km/h (25-56 mph)	10 sec
P5	Gradeability (Heat Engine only)**	
P6	Payload Capacity	520 kg
P7	Cargo Capacity	0.5 m^3
P8	Consumer Costs	
	P8.1 Consumer purchase price (1978 $)	9,212
	P8.2 Consumer life cycle cost (1978 $)	26,111

P5 Gradeability (Heat Engine only)**

		Grade	Speed
P5.1		3%	118 km/hr
P5.2		5%	86 km/hr
P5.3		8%	80 km/hr
P5.4		15%	25 km/hr
P5.5	Maximum Grade	25%	

*72V Near Term Hybrid Vehicle with the accessories on.
**Distance is not included, because in diesel drive the distance is limited only by the fuel tank capacity.

Table 9-4 (Cont'd)

P9	Emissions	
	P9.1 Hydrocarbons (HC)	0.13 gm/km
	P9.2 Carbon monoxide (CO)	0.31 gm/km
	P9.3 Nitrogen oxides (NO_x)	0.56 gm/km
P10	Ambient Temperature Capability	
	Temperature range over which minimum performance requirements can be met	-20°C to +40°C
P11	Rechargeability	
	Maximum time to recharge from 80% depth-of-discharge	6-8 hrs
P12	Required Maintenance	
	Routine maintenance required per month	1 hr
P13	Unserviced Storeability	
	Unserviced storage over ambient temperature range of -30°C to +50°C (-22°F to +122°F)	
	P13.1 Duration	120 day
	P13.2 Warm-up required	1-2 min
P14	Reliability	
	P14.1 Mean usage between failures - powertrain	40,000 km
	P14.2 Mean usage between failures - brakes	40,000 km
	P14.3 Mean usage between failures - vehicle	40,000 km
P15	Maintainability	
	P15.1 Time to repair - mean	5.0 hrs
	P15.2 Time to repair - variance	2.0 hrs
P16	Availability	
	Minimum expected utilization rate (i.e., 100 x time in service ÷ (time in service + time under repair)	97%
P17	Additional Accessories and Amenities	
	Fuel-burning heater air conditioner, power steering, and power brakes	

Data Used to Obtain the NTHV Energy Consumption Measures:

Table 9-5. Petroleum and Electricity Consumptions for the NTHV with Accessories on and with Electric Motor as the Primary Drive Component.

FUDC (ℓ/cycle)	FUDC (Δs/cycle)	FHDC (ℓ/cycle)	FHDC (Δs/cycle)	SAE J227a(B) (ℓ/cycle)	SAE J227a(B) (Δs/cycle)
0.041	0.4046	0.036	0.4969	0.0	0.0105

Table 9-6. Petroleum and Electricity Consumptions for the NTHV with Accessories on and with Heat Engine as the Primary Drive Component.

(ℓ/cycle)	(Δs/cycle)	(ℓ/cycle)	(Δs/cycle)	SAE J227a(B) (ℓ/cycle)	SAE J227a(B) (Δs/cycle)
0.994	0.0	0.955	0.0	0.034	0.0

Table 9-7. Life Cycle Energy Consumption Data (Reference 31)

Manufacturing Energy = 85.92 MJ/kg
Scrapped Energy = 11.06 MJ/kg

Table 9-8. Preliminary Design NTHV Energy Consumption Measures

E1. Annual petroleum based fuel energy consumption per vehicle compared to reference ICE vehicle over Mission A.

NTHV = 753 liters/yr, Ref. ICE = 1730 liters/yr
NTHV = 28672 MJ/yr, Ref. ICE = 65873 MJ/yr

E2. Annual total energy consumption per vehicle compared to reference ICE vehicle over Mission A.

NTHV = 42435 MJ/yr, Ref. ICE = 65873 MJ/yr

E3. Potential annual fleet petroleum based fuel energy savings compared to reference ICE vehicle over Mission A.

11.16×10^{10} MJ/yr

E4. Potential annual fleet total energy consumption compared to reference ICE vehicle over Mission A.

15.29×10^{10} MJ/yr

E5. Average energy consumption over maximum non-refueled range.

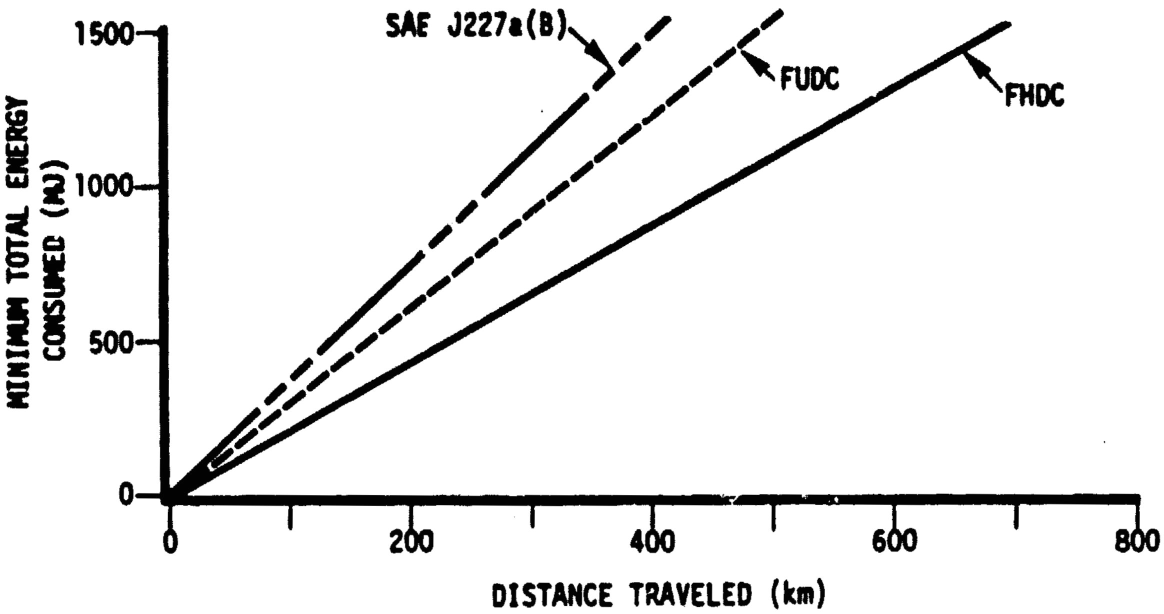
FHDC = 2.19 MJ/km
FUDC = 3.12 MJ/km
SAE J227a(B) = 3.79 MJ/km

E6. Average petroleum based fuel energy consumption over maximum non-refueled range.

FHDC = 2.12 MJ/km
FUDC = 3.03 MJ/km
SAE J227a(B) = 3.68 MJ/km

Table 9-8. (Cont'd)

E7. Minimum total energy consumed vs. distance traveled starting with full charge and full tank over FHDC, FUDC, and SAE J227a(B).



E8. Minimum Petroleum based fuel energy consumed vs. distance traveled starting with full charge and full tank over FHDC, FUDC, and SAE J227a(B).

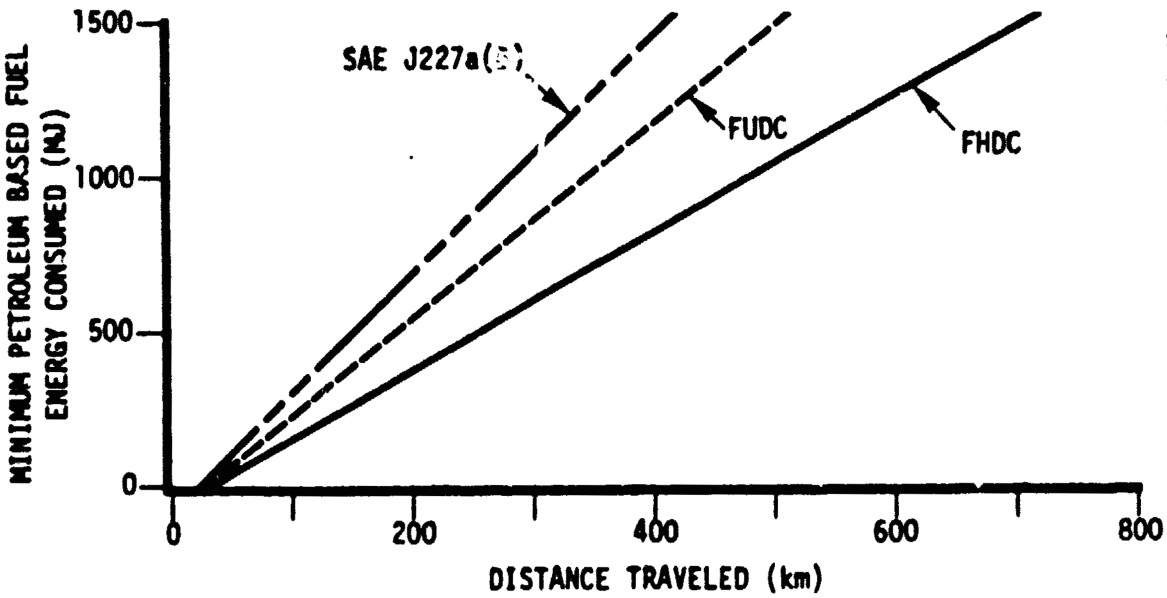


Table 9-8. (Cont'd)

E9. Life cycle energy consumption per NTHV compared to reference ICE vehicle.

	NTHV (MJ)	Ref. ICE (MJ)
Vehicle Fabrication	141,174	109,366
Vehicle Operation and Maintenance (including fabrication of replacement parts)	506,795	676,866
Vehicle Disposal	18,179	14,083
Total Life Cycle Energy Consumption	629,790	772,149

SECTION 10

TECHNOLOGY REQUIREMENTS

In order to achieve the preliminary design for the NTHV, the technology developments listed below must be developed. We believe that all of these items will be achieved during the term of Phase IIA of this program.

- Development of electronic interfaces with the hydraulic control system of the transmission
- Development of the slipping clutch for the motor, and its control system
- Development of a variable fill fluid coupling for the motor and the hydraulic and electronic control systems for the coupling
- Development of an electrically driven transmission oil pump with its control system and an accumulator
- Development of the mechanical interfaces between the motor input to the transmission through a slipping clutch (or a variable fill fluid coupling) and the engine input through a lock-up torque converter and a connecting clutch
- Development and implementation of shifting algorithms for the electric, diesel and hybrid modes of operation
- Development of the Improved State of The Art Lead-acid batteries (or equivalent) to a satisfactory performance and durability level
- Development of a satisfactory battery heating system for use when the vehicle is in operation and when the batteries are being recharged
- Development of a variable speed belt drive for use between the engine and the accessory drive shaft
- Development of the vehicle heating system using both the diesel engine and the combustion heater as heat sources

- Development of a braking control system that will provide both good driver control of vehicle braking and maximum regenerative energy returned to the batteries

- Development of an air cushion restraint system for the NTHV that will provide 30 mph passive protection

- Modification of, and addition to, the base vehicle front end structure, to provide good crush characteristics in a frontal impact and support for the extra weight on the front of the vehicle

- Integration of the various computer and control system components into an integrated, reliable control system

- Refinement and implementation of the necessary control strategies

- Refinement and implementation of the necessary operational strategies

- Development of a reliable "drive by wire" control system for the NTHV with a failsafe backup, "get you home," system.

SECTION 11

DATA SOURCES AND ASSUMPTIONS

11.1 LIST OF DATA SOURCES USED

Most of the data sources are listed in the References. Other data sources were as follows:

1. Automotive News, 1979 Market Data Issue.

2. Chevrolet Catalog for the Citation, February 1979.

3. Federal Motor Vehicle Safety Standards.

4. Final Report, Vol. II, Comprehensive Technical Results, Research Safety Vehicle, Phase II, Minicars, Inc., Contract DOT-HS-5-01215.

5. "Energy Management," Bailey Division, USM Corporation, brochure on rubric bumper parts.

6. Material published by the NHTSA on the Volkswagen passive belt system.

7. Honda Accord and Volvo service manuals.

8. International Rectifier HEXFET product literature.

9. Motorola Microcomputer data library.

10. Numerous system design and data sheet publications by Motorola, Intel, Texas Instruments, Fairchild, National Semiconductor, etc.

11.2 LIST OF SIGNIFICANT ASSUMPTIONS

1. The primary emphasis in the program is the integration of the hybrid propulsion system and its controls. The scope of the program does not allow for a complete, from the ground up, design of the vehicle.

2. It is neither feasible nor necessary that the vehicle be able to satisfy the JPL Minimum Requirements (or other performance requirements) all of the time (e.g., independently of the power

source being used, the battery state of charge, the number of passengers, etc.).

3. Diesel and spark ignition engines are the only ones that qualify as near-term heat engines.

4. To be a totally socially responsible vehicle, the NTHV must possess a degree of occupant protection equal to or greater than those of the vehicles it is replacing.

5. It is necessary that the NTHV meet the safety standards in existence at the time when it would be on the road, and, therefore, that it can be designed to meet the standards in force during the time frame of the contract. Specifically, this assumption calls for passive protection of front seat occupants.

6. The passive belts in use will experience usage rates of 40 to 50 percent and consumers are likely to prefer airbag systems.

7. Operation of the NTHV should not require any special training, skills, or workload requirements beyond those required of drivers of conventional vehicles.

8. The minimum acceptable battery charging efficiency is 70 percent.

9. Many of the deficiencies of lead-acid batteries are likely to be alleviated in the near term.

10. A loss of 50 to 60 percent of battery capacity at $-20^{\circ}C$ would require an increase in the use of diesel fuel, and, hence, there should be a provision for keeping batteries warm to prevent this.

11. Air conditioning is a requirement for the NTHV, since 80 percent of new U.S. cars were fitted with this option.

12. A production-engineered computerized control system is beyond the scope of the Phase II effort; rather, the effort should focus on 1) proof of concept, and 2) control system software.

13. Failsafe operation of the vehicle is required while under computer control; the vehicle must have "limp home" capability if the computer control system is inoperative.

14. Supplementary diagnostic aids are desirable for the NTHV program.

15. Assurance of a successful control system demonstration is a prerequisite in the selection of a control system configuration; hence, simplicity, reliability, and ease of development are of primary importance.

16. EMI conditions are at a sufficiently high level in the NTHV as to produce very little benefit from the use of a CMOS microprocessor relative to an NMOS unit. Most of the EMI protection will result from the use of CMOS peripheral chips and proper design techniques.

17. Power consumption is not be considered an issue in the selection of a development/demonstration control system.

18. Only gross differences resulting from significantly differing capabilities are to impact the microprocessor selection for the development/demonstration system.

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2. Environmental Protection Agency, "1979 Gas Mileage Guide - California," Second Edition, January 1979.

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4. General Motors Corp., "1979 General Motors Car and Body Dimensions," 30 April 1979.

5. C. Strother, B. Broadhead and T. Zinke, "Small Car Driver Inflatable Restraint System Evaluation," Final Report, Contract DOT-HS-6-01412, July 1978.

6. M.O. Dustin and R.J. Denington, "Test and Evaluation of 23 Electric Vehicles for State-of-the-Art Assessment," SAE Paper 780290, February 1978.

7. "Determination of the Effectiveness and Feasibility of Regenerative Braking Systems on Electric and Other Automobiles," Final Report, Contract W-7405-ENG-48, February 1978.

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APPENDIX A

CONTROL SYSTEM ALGORITHMS

CLUTCH CONTROL ALGORITHM

The clutch control algorithm checks a flag to see what the desired state of the clutch should be. It then assesses the current motor motion RPM and the clutch output rpm to identify the relative speeds of the two surfaces. The clutch control valves are then activated as appropriate to bring the clutch surfaces together or to separate them as indicated by the flag.

Inputs: Control flag (in, out)
Motor rpm
Clutch output rpm

Outputs: Pulsing of two hydraulic valves on clutch control

DRIVER INPUT ALGORITHM

The driver input algorithm assesses the brake pedal and acceleration pedal positions to determine what the driver wants to have happen. It also has switches to indicate no pedal ("true idle") and fuel pedal ("fuel power") modes. It also assesses the state of the automatic shift lever and the key start. Based on these inputs and the current vehicle speed, it sets flags for the transmission state (F,R,P,N) and calculates a desired driveline torque. The desired driveline torque considers the current speed and the pedal angle (or depression). The braking needs consider the current velocity and the pedal depression.

Inputs: Acceleration pedal position
Brake pedal position

Output: Overall torque requirement
Brake status
Transmission control status

MOTOR/CONTROLLER ALGORITHM

The motor/controller algorithm accepts as input the overall torque requirements and the operational strategy data associated with the total torque to be generated by the motor. In addition, the motor rpm, vehicle speed, clutch output speed, and system status flag will be considered as input data already stored in memory.

The algorithm will then calculate the needed field current and transmit the information to the field controller. If the motor is not up to the baseline motor speed, the clutch status will be checked. If the clutch is disengaged, the armature resistor switch will be activated. If the armature switch resistor is on and the motor is at base speed, the switch will be reset to off, and the field current instruction will be sent. The regenerator braking flag will be reset if the flag is 1 and the motor torque requirement is positive.

Input: Data from memory

Output: Field controller current signal

ENGINE CONTROL ALGORITHM

The engine control algorithm accepts as input the overall torque requirements and the operational strategy data associated with the fraction of total torque to be provided by the engine. In addition, the engine rpm vehicle speed, torque converter output speed and transmission input speed, the engine temperature, throttle position, fuel level and the system status flag will be considered as input from memory. The algorithm will then be compared until the engine rpm exceeds the idle rpm, at which time the starter flag will be reset and the remainder of the algorithm with regard to throttle position can be executed.

Input: Engine rpm
Torque converter output rpm
Transmission input rpm
Engine temperature
Fuel level

Output: Throttle position signal

BATTERY STATE OF CHARGE ALGORITHM

The battery state of charge algorithm updates the state of charge memory location by accepting input from the Curtis Instruments charge device, computing this information, or looking it up in stored table. It will also correct the data as necessary to compensate for environmental conditions as necessary.

Input: State of charge measure

Output: Update of memory location containing state of charge data

TRANSMISSION CONTROL ALGORITHM

The transmission control algorithm evaluates the system operating strategy input, the driver torque requirements, the current transmission status flags, the driver line rpm, the transmission input rpm, the torque lockup status, the clutch status, the engine rpm, the motor rpm, the shift lever status, the braking status, and the time since last shift. Based on these inputs a decision is made as to the proper gear. The result is then checked against the current transmission status. If the gear is not currently implemented, or in the process of being implemented, a shift initiation is begun and the shift sequence flags are set. The algorithm then begins the shift sequence. For safety, it resets those sequence flags which are accomplished. If the algorithm is called and the shift flags are positive, the algorithm continues in the shifting sequence starting from the point where it left off. For safety, all actions are checked by feedback signals to indicate completion of each step in the sequence. At the completion of the sequence, a flag is set and the evaluation sequence is begun again.

Inputs: All powertrain dynamic inputs
Feedback signals from valve or solenoid actuation to status signals

Outputs: Solenoid or valve actuation signals

TORQUE CONVERTER CONTROL ALGORITHM

The torque converter control algorithm controls the lockup mechanism, although it can be reactivated in other algorithms. The program takes as input the engine rpm, the torque converter output rpm, and the transmission input rpm, as well as the brake status, the gear shift status, electric motor output rpm, clutch status, and torque requirements. The algorithm evaluates the condition and either engages or disengages the torque converter lockup. The lockup would be initiated if the transmission input, TC output and engine output rpm are within some value of each other. The lockup would be disconnected if it appeared that some rpm mismatches were occurring or were about to occur (such as through clutch engagement).

Inputs: Torque requirements
Engine speed
Torque converter output speed
Clutch status
Spare status

Output: Lockup signal

DISPLAY CONTROL ALGORITHM

The display control will evaluate the status of various systems and update the display information, as appropriate. The display will be latched; hence it will not require continuous updating, except as the parameters change. In addition, the algorithm will check the status flags from the diagnostic algorithms and display any errors by appropriate diagnostic messages. The message to be displayed will be determined by a priority system. Potentially, this message could alternate with display information.

Input: System status data
Diagnostic messages flags

Output: Update of display latches as needed
Possible override of display with diagnostic messages

ACCESSORY CONTROL ALGORITHM

The accessory control algorithm will evaluate the status of all vehicle accessory controls. The status of these will be translated to power requirements and stored in a dedicated memory location.

Input: System accessory sensors

Output: Power requirements data for use in operational strategy assessments

OPERATING MODE ALGORITHM

Considers the output from the operational strategy algorithm and decides in conjunction with the current conditions what the correct requirements are from each power plant.

Input: Input on current conditions
Operational strategy output

Output: Determines required output from each power unit

OPERATIONAL STRATEGY ALGORITHM

Considers the current conditions input in conjunction with mission input, analysis data, and potential driver mission input to derive the correct operational strategy for maximizing petroleum savings based on the system analysis studies conducted in Phase I.

Input: Mission information
System status information

Output: Decision on operating mode (i.e., electric, diesel, diesel topped by electric, or electric topped by diesel) and maximum allowable motor power
Range remaining computations and display

APPENDIX B

CONTROL SYSTEM DESIGN CONSIDERATIONS

— OVERRIDE

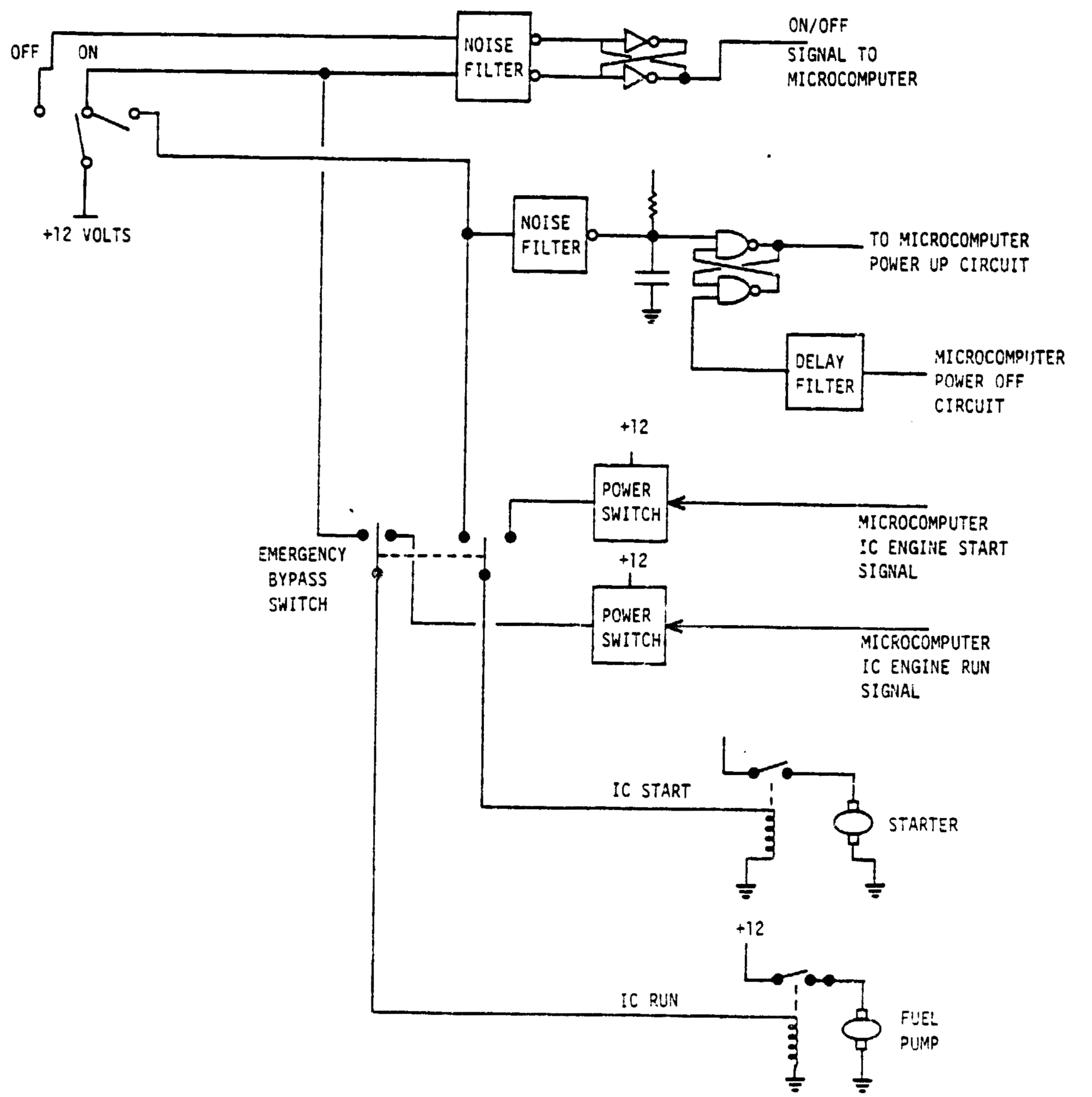
CONTROL SYSTEM DESIGN CONSIDERATIONS - OVERRIDE

1) Override switch
   (a) Starter relay switched to key switch
   (b) Fuel turned on
   (c) Microprocessor disabled
   (d) Electric motor disabled (or given to driver, if appropriate).

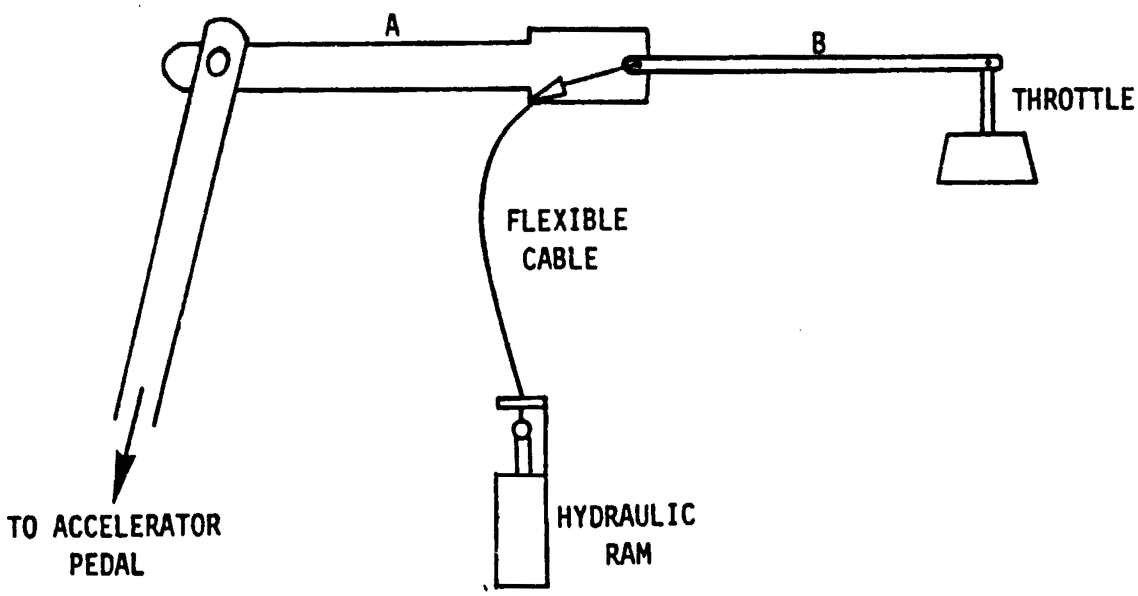
2) Transmission works normally in park, neutral, reverse, second, and low. Microprocessor given control only in drive.

3) Timer switch on fuel pump — if diesel does not run in specified time, fuel pump is shut off.

KEY SWITCH EXAMPLE

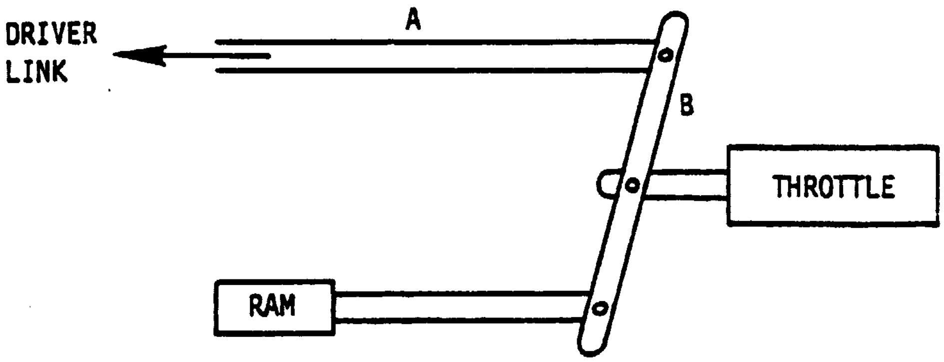


THROTTLE (OR BRAKE) OVERRIDE



As the driver moves the accelerator pedal, A and B work together to move the throttle. The hydraulic ram can move the inner cable of the flexible cable to move B in relation to A. Thus, if the driver pushes A to the right, the computer can make the ram move B to the left, in relation to A. Thus there may be no net motion of the throttle. If the ram fails for any reason, the driver has complete control.

2) Links



The driver has a direct control through link A. The ram can counteract this by moving in the opposite direction. However, if the ram fails, the driver still has complete control.

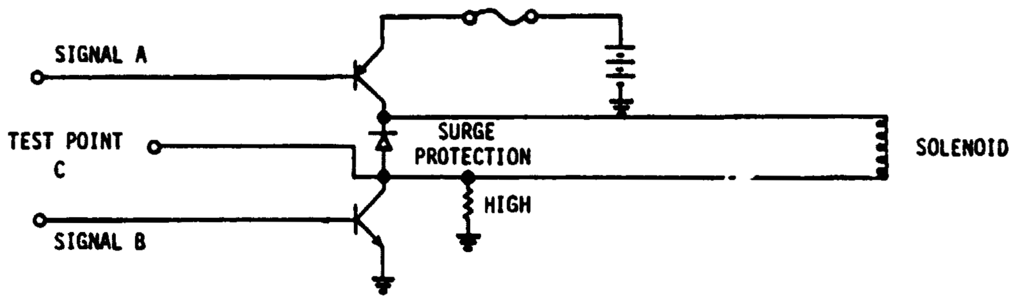
APPENDIX C

BUILT-IN TESTING DESIGN

BUILT-IN TESTING DESIGN

The following circuits are examples of designs for built in testing of sensors and actuators by the microprocessor system.

1) Solenoid



Initially solenoid is off - A = High, B = Low

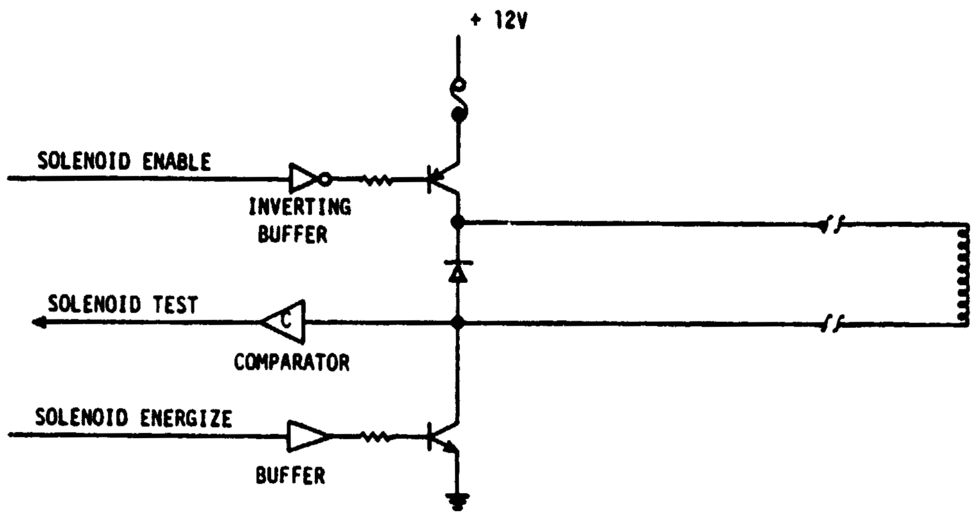
Turning Solenoid On

1) Lower point A
2) Read point C
3) If it goes high, the connection from battery through fuse, transistor, and coil is good
4) If it does not go high, something is bad
5) Raise point B
6) Read point C
7) If it goes low, the coil is ok (electrically)
8) If it does not, coil, transistor or connection is bad

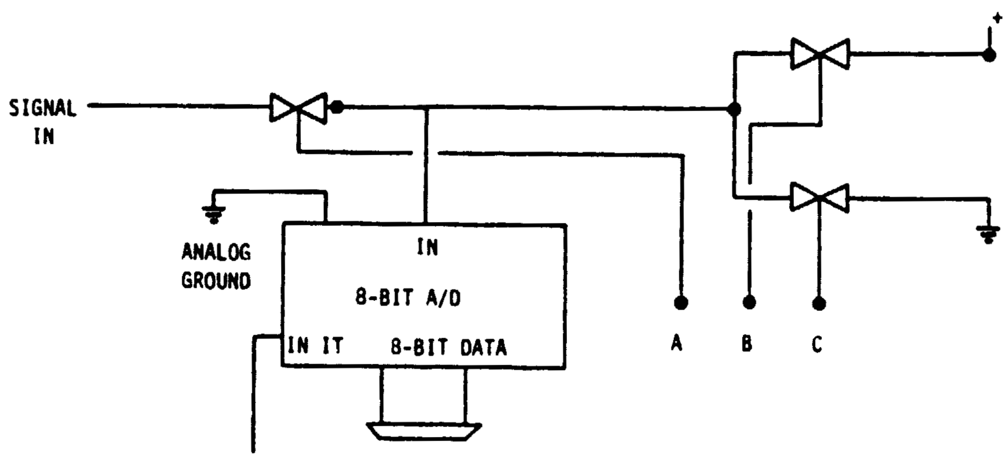
Turning Solenoid Off

1) Lower point B
2) Read point C
3) If it went high, ok. else bad
4) Raise point C
5) Read point C
6) If it went low, ok. else bad.

Solenoid Interfaces



2) A/D Converters (thermistors, pots, pressure gauges)



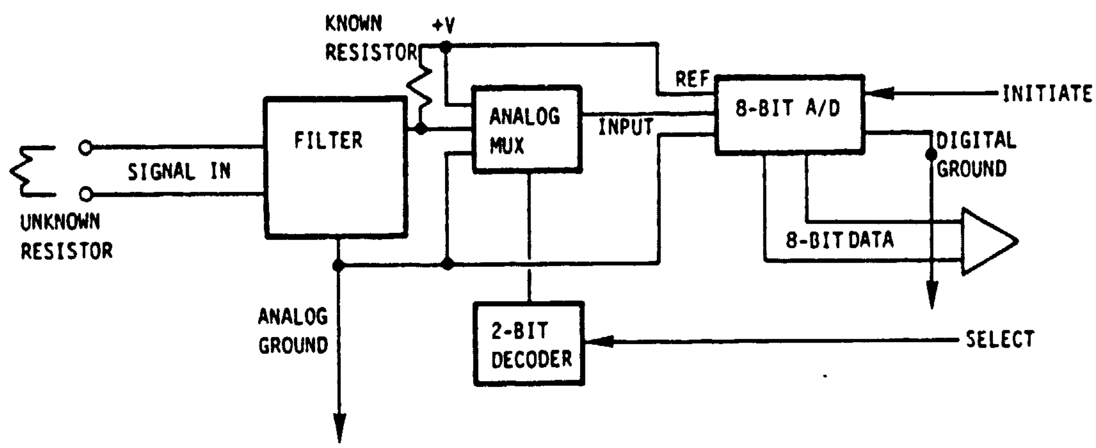
To take reading

1) Set B
2) Take Reading
3) Check to see if it is within error range
4) Reset B - Set C
5) Take reading
6) Check to see if it is within error range
7) Reset C Set A
8) Take Reading
9) Reset A

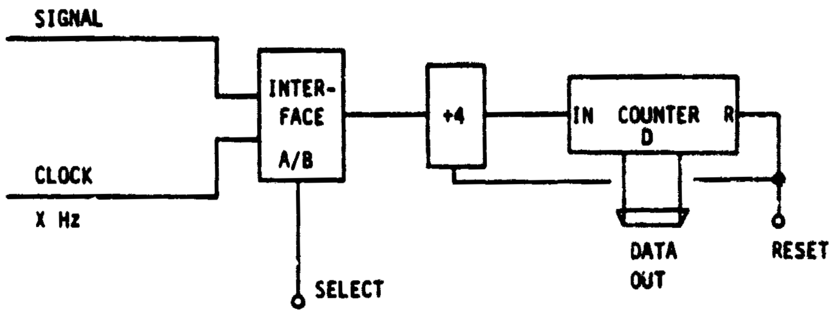
Use ratiometric measurements
Take ratio between value and full-scale voltage
Do not need absolute voltage

A/D Converter

A/D Converter



3) Frequency Sensor



To take reading

1) Select B (clock)
2) Reset counter
3) Wait x msec
4) Read counter - check within limits
5) Select A (data)
6) Reset counter
7) Wait x msec
8) Read data from counter

APPENDIX D

PRELIMINARY SENSOR AND ACTUATOR REQUIREMENTS

Preliminary Sensor and Actuator Requirements

	Type	Sealed	Temperature (°C)	Vibration	Size	Manufacturer	Description
I.C. ENGINE							
Sensors							
Engine speed	Magnetic pickup	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Throttle position	Potentiometer	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			Rotary pot, 0 to 65° rotation Log taper Resistance, 0 to 50K Ω
Oil pressure	Pressure Potentiometer	Yes	-40 to 150	0 to 5 Gs 0 to 2 kHz			Pressure pot Log taper Resistance 0 to 50K Ω
Fuel level	Fuel float pot or Capacitive	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Water temperature	Thermistor	Yes	-40 to 150	0 to 5 Gs 0 to 2 kHz			
Oil temperature	Thermistor	Yes	-40 to 150	0 to 5 Gs 0 to 2 kHz			
Actuators							
Throttle postion	Solenoid valves	Yes	-40 to 150	0 to 5 Gs 0 to 2 kHz			
Starter relay	Relay	Yes	-40 to 150	0 to 5 Gs 0 to 2 kHz			
Fuel pump relay	Relay	Yes	-40 to 150	0 to 5 Gs 0 to 2 kHz			
I.C. TORQUE CONVERTER							
Sensors							
Lockup	Microswitch	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Engine RPM	Magnetic Pickup	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Converter out RPM	Magnetic Pickup	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			

Preliminary Sensor and Actuator Requirements (cont'd)

	Type	Sealed	Temperature (C°)	Vibration	Size	Manufacturer	Description
I.C. TORQUE CONVERTER (Cont'd)							
<u>Actuators</u>							
Lockup control	Solenoid valves	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
I.C. ENGINE CLUTCH							
<u>Sensors</u>							
Clutch position	Pressure switch	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Converter out RPM	Magnetic pickup	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Clutch out RPM	Magnetic pickup	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
<u>Actuators</u>							
Clutch control	Solenoid valves	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
ELECTRIC MOTOR							
<u>Sensors</u>							
Field current	Current	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Armature current	Current	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Field voltage	Voltage	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Armature voltage	Voltage	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Motor speed	Magnetic pickup	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Motor temperature	Thermistor	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			

Preliminary Sensor and Actuator Requirements (cont'd)

	Type	Sealed	Temperature (C°)	Vibration	Size	Manufacturer	Description
ELECTRIC MOTOR (Cont'd)							
Actuators							
Field current	Current controller	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Armature relay	Relay	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
ELECTRIC MOTOR CLUTCH							
Sensors							
Motor RPM	Magnetic pickup	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Clutch out RPM	Magnetic pickup	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Actuators							
Clutch control	Solenoid valves	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
TRANSMISSION CONTROL							
Sensors							
Transmission input RPM	Magnetic pickup	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Transmission output RPM	Magnetic pickup	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Oil temperature	Thermistor	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Gear positions	Microswitch	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Actuators							
Shifting control	Solenoid valves	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			

Preliminary Sensor and Actuator Requirements (cont'd)

	Type	Sealed	Temperature (C°)	Vibration	Size	Manufacturer	Description
DRIVER INPUT & OUTPUT							
Sensors							
Accelerator position	Pot	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Brake pedal position	Pot	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Transmission select	Switches	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Key position	Switch position	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Actuators							
Driver display	Display tubes	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
BATTERY STATUS							
Sensors							
State of charge	Charge	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Battery temperature	Thermistor	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Battery module voltage	Voltage	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
On-board charger	Current	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Actuators							
Battery charger	Current control	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Battery compartment temperature	Solenoid	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			

Preliminary Sensor and Actuator Requirements (cont'd)

	Type	Sealed	Temperature (C°)	Vibration	Size	Manufacturer	Description
ENGINE AND ACCESSORY BATTERY							
<u>Sensors</u>							
Alternator output	Voltage	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Battery voltage	Voltage	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
<u>Actuators</u>							
ENVIRONMENTAL CONDITIONS							
<u>Sensors</u>							
Inside air temperature	Thermistor	Yes	-40 to 150	0 to 5 Gs 0 to 2 kHz			
Outside air temperature	Thermistor	Yes	-40 to 150	0 to 5 Gs 0 to 2 kHz			
Engine temperature	Thermistor	Yes	-40 to 150	0 to 5 Gs 0 to 2 kHz			
Battery compartment temperature	Thermistor	Yes	-40 to 150	0 to 5 Gs 0 to 2 kHz			
Environmental control	Switches	Yes	-40 to 150	0 to 5 Gs 0 to 2 kHz			
<u>Actuators</u>							
A/C control	Clutch solenoid	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Combustion heater	Heater control relay	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			

Preliminary Sensor and Actuator Requirements (cont'd)

	Type	Sealed	Temperature (C°)	Vibration	Size	Manufacturer	Description
ACCESSORIES							
<u>Sensors</u>							
Lights	Voltage	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
<u>Actuators</u>							
Diagnostic display	Indicators	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			
Diagnostic interface	Plug to diagnostic unit	Yes	-40 to 125	0 to 5 Gs 0 to 2 kHz			

APPENDIX E

BENCHMARK COMPARISON

FOR 6801 MICROCOMPUTER

Subroutine	6800	6801	6809 Reentrant
8 x 8 Multiply			
ROM Bytes	19	1	1
RAM Bytes	1	-	-
Stack Bytes	0	0	0
Instructions		1	1
Cycles	145-177	10	11
8 x 16 Multiply			
ROM Bytes	37	25	17
RAM Bytes	3	2	-
Stack Bytes	0	0	0
Instructions		12	10
Cycles	209-277	59	57
16 x 16 Multiply			
ROM Bytes	48	53	44
RAM Bytes	4	6	-
Stack Bytes	0	0	6
Instructions		23	24
Cycles	504-632	116-128	140-154
16/16 Divide			
ROM Bytes	51	44	50
RAM Bytes	4	4	0
Stack Bytes	0	0	6
Instructions		22	24
Cycles	761-931	643-778	626-770
2-D Table Lookup			
ROM Bytes	51	28	26
RAM Bytes	3	1	0
Stack Bytes	0	1	2
Instructions		21	17
Cycles	761-931	59	57-61
3-D Table Lookup			
ROM Bytes	70	37	36
RAM Bytes	8	3	0
Stack Bytes	5	7	7
Instructions		23	20
Cycles	828-1146	251	233-242
Set/Clear Bit			
ROM Bytes	8	8	8
RAM Bytes	0	0	0
Stack Bytes	0	0	0
Instructions		3	3
Cycles	11	10	12

Subroutine	6800	6801	6809 Reentrant
Branch on Bit			
ROM Bytes	7	7	7
RAM Bytes	0	0	0
Stack Bytes	0	0	0
Instructions		3	3
Cycles	10	9	10

```
                               NAM    2DLOOK
                         *
                         ***************************************************
                         *
                         * M6809 REENTRANT
                         *
                         * ENTRY:
                         *  X POINTS TO TABLE START (ACTUAL DATA)
                         *  THE MINIMUM VALUE OF THE INDEPENDENT VARIABLE I
                         *     IN THE LOCATION PRECEEDING THE ACTUAL DATA (-
                         *  A CONTAINS INPUT VALUE
                         *
                         * EXIT:
                         *  A CONTAINS RESULT
                         *
                         ***************************************************
                         *
0000 A0    1F          5 LKUP2D SUBA   -1,X       OFFSET = X-XMIN
0002 C6    10          2 LKUPQ  LDAB   #$10       SEPARATE OFFSET INTO TA
0004 3D               11        MUL               AND INTERPOLATION PARTS
0005 30    86          5        LEAX   A,X        GET INDEX INTO TABLE (P
0007 A6    84          4 LKUPR  LDA    ,X         Y1
0009 A0    01          5        SUBA   1,X        Y1-Y2
000B 25    08          3        BLO    LUP1
000D 3D               11 LKEN1  MUL               Y2 < Y1
000E 40                2        NEGA              Y1 + (-((Y1-Y2)*(X-X1)/16
000F AB    84          4        ADDA   ,X
0011 58                2        ASLB              ROUNDOWN IF NECESSARY
0012 82    00          2        SBCA   #0
0014 39                5        RTS
0015 40                2 LUP1   NEGA              Y2<Y1
0016 3D               11        MUL               Y1+(Y2-Y1)*(X-X1)/16
0017 A9    84          4        ADCA   ,X
0019 39                5        RTS
                         *
                         * 57 - 61 CYCLES
                         * 17 INSTRUCTIONS
                         * 26 BYTES
                         *
```

```
                                 *
                                 ***********************************************
                                 *
                                 * 3 DIMENSIONAL TABLE LOOKUP
                                 * M6809 REENTRENT VERSION
                                 *
                                 * ENTRY:
                                 *  Y POINTS TO 4TH VALUE IN TABLE HEADER
                                 *    -1,Y RNUM
                                 *    -2,Y Q MIN
                                 *    -3,Y R MIN
                                 *  U & S ARE VALID STACK POINTERS
                                 *  D CONTAINS R:Q
                                 *
                                 * EXIT
                                 *  A CONTAINS RESULT
                                 *
                                 * EXECUTION STACKS
                                 *      S         U
                                 * +---------+----------+
                                 * |INTER R  |    Q     |
                                 * +---------+----------+
                                 * |   Q1    |
                                 * +---------+
                                 *
                                 ***********************************************
                                 *
001A A3   3D      7 LKUP3D SUBD  -3,Y      R-RMIN AND Q- QMIN
001C 36   04      5        PSHU  B         SAVE Q
001E C6   10      2        LDAB  #$10      GET R INDEX & INTERPRET
0020 3D          11        MUL
0021 34   04      5        PSHS  B         INTERPOLATION VALUE FOR
0023 E6   3F      5        LDAB  -1,Y      CALCULATE ADDR OF R1 TA
0025 3D          11        MUL
0026 30   A5      5        LEAX  B,Y
0028 A6   C4      4        LDA   ,U        CALCULATE Q1
002A 8D   D6      7        BSR   LKUPQ
002C 34   02      5        PSHS  A         Q1
002E E6   3F      5        LDB   -1,Y      GET ADDR OF R2 TABLE
0030 3A           3        ABX
0031 37   02      5        PULU  A         CALCULTE Q2
0033 8D   CD      7        BSR   LKUPQ     Q2
0035 40           2        NEGA            Q1-Q2
0036 AB   E0      6        ADDA  S+
0038 35   04      5        PULS  B         INTERP VALUE OF R
003A 24   D1      3        BHS   LKEN1
003C 20   D7      3        BRA   LUP1
```

```
                                  NAM    8X16->16
                             *****************************************
                             *
                             *    M6809 REEENTRANT VERSION
                             *
                             *    ENTRY:
                             *     X POINTS TO 16 BIT MULTIPLICAND
                             *     Y POINTS TO 8 BIT MULTIPLIER
                             *    EXIT:
                             *     U CONTAINS ANSWER ON EXIT
                             *     X,Y ARE UNCHANGED; D IS DESTROYED
                             *
                             *      TU
                             *       V
                             *      --
                             *      UV
                             *     TV
                             *     ---
                             *
                             *****************************************
                             *
0000 A6    A4             4  M8X16   LDA    ,Y        V
0002 E6    84             4          LDB    ,X        T
0004 3D                  11          MUL              TV
0005 1F    03             6          TFR    D,U
0007 A6    A4             4          LDA    ,Y        V
0009 E6    01             5          LDB    1,X       U
000B 3D                  11          MUL              UV
000C 89    00             2          ADCA   #0        ROUNDUP UVH
000E 33    C6             5          LEAU   A,U       UVH +TU -> RESULT
0010 39                   5          RTS
                             *
                             * 17 BYTES
                             * 10 INSTRUCTIONS
                             * 57 CYCLES
           0000                      END

TOTAL ERRORS   00000
TOTAL WARNINGS 00000
```

```
****************************************************************
*                                                              *
*           M6809 CODE                                         *
*                                                              *
* 16X16 MULTIPLY SUBROUTINE - REENTRANT                        *
*  16 BIT X 16 BIT = 32 BIT                                    *
*                                                              *
*  (T:U) X (V:W) =             UWH:UWL                         *
*                    +     TWH:TWL                             *
*                    +     UVH:UVL                             *
*                    + TVH:TVL                                 *
*                    ----------------                          *
*                                                              *
*  ENTRY FORMAT:                                               *
*   X REGISTER CONTAINS ADDRESS OF 16 BIT MULTIPLICAND         *
*   Y REGISTER CONTAINS ADDRESS OF 16 BIT MULTIPLIER           *
*   S REGISTER MUST CONTAIN STACK POINTER DECREMENTED BY 4     *
*   (VIA THE INSTRUCTION; LEAS -4,S) TO RESERVE 4 BYTES ON     *
*   STACK BEFORE ENTERING SUBROUTINE                           *
*                                                              *
*  RETURN FORMAT:                                              *
*   32 BIT PRODUCT IS ON STACK, MS AT SP                       *
*                                                              *
****************************************************************


*                                                       STACK
*                                                     +++++++++
*                                                     +  UWL  +
*                                                     +++++++++
*                                                     +  UWH  +
*                                                     +++++++++
*                                                     +  TVL  +
*                                                     +++++++++
*                                                     +  TVH  +
*                                                     +++++++++
*                                                     +  PCL  +
*                                                     +++++++++
*                                                     +  PCH  +
*                                                     +++++++++
*                                                     +   B   +
*                                                     +++++++++
*                                              SP ->  +   A   +
*                                                     +++++++++


MULTX    PSHS     D          SAVE A,B REGISTERS

         LDA A    1,X        U -> A
         LDA B    1,Y        W -> B
         MUL                 U X W
         STD      6,S        STORE UWH,UWL ON STACK

         LDA A    0,X        T -> A
         LDA B    0,Y        V -> B
```

```
         MUL                    T X V
         STD        4,S         STORE TVH,TVL ON STACK

         LDA A      0,X         T -> A
         LDA B      1,Y         W -> B
         MUL                    T X W
         ADDD       5,S         ADD TW PARTIAL PRODUCT
         STD        5,S         ..TO
         BCC        MULTX1      ..PARTIAL PRODUCT
         INC        4,S         ..SUM

MULTX1   LDA A      1,X         U -> A
         LDA B      0,Y         V -> B
         MUL                    U X V
         ADDD       5,S         ADD UV PARTIAL PRODUCT
         STD        5,S         ..TO
         BCC        MULTX2      ..PARTIAL PRODUCT
         INC        4,S         ..SUM

MULTX2   PULS       D,PC        RESTORE A,B REGISTERS
```

```
                                  NAM    6801
                             ***********************************************
                             *
                             * 6809 REENTRANT
                             *
                             * ENTRY:
                             *  X CONTAINS DENOMINATOR
                             *  D CONTAINS NUMERATOR
                             *  U & S ARE VALID STACK POINTERS
                             *
                             * EXIT:
                             *  D CONTAINS RESULT
                             *  IF NUMERATOR > DENOMINATOR THEN $FFFF IS RETURNE
                             *
                             * DURING EXECUTION:
                             *  S POINTS TO DENOMINATOR
                             *  U POINTS TO QUOTIENT
                             *
                             ***********************************************
                             *
0000 108E 0000          4 DIV     LDY    #0        INIT QUOTIENT
0004 36   20            6         PSHU   Y
0006 34   10            6         PSHS   X         PUT DENOMINATOR ON STAC
0008 10A3 E4            7         CMPD   ,S        CHECK FOR OVERFLOW ONCE
000B 24   1C            3         BHS    DIV5
000D 20   0A            3         BRA    DIV2      FORCE A 1 AS FIRST BIT
                          *                            FOR TERMINATION.
000F A3   E4            6 DIV4    SUBD   ,S        WILL NEXT BIT BE A 1 OR
0011 24   06            3         BCC    DIV2
0013 E3   E4            6         ADDD   ,S        ZERO, ADD BACK VALUE
0015 1C   FE            3         CLC              ZERO TO SHIFT IN
0017 20   02            3         BRA    DIV1
0019 1A   01            3 DIV2    SEC              ONE TO SHIFT IN
001B 69   41            7 DIV1    ROL    1,U       SHIFT IN QUOTIENT BIT
001D 69   C4            6         ROL    ,U
001F 25   0D            3         BCS    DONE      A 1 MEANS WE HAVE DONE
0021 58                 2         ASLB             SHIFT DENOMINATOR
0022 49                 2         ROLA
0023 24   EA            3         BCC    DIV4      GO TO NEXT BIT
0025 A3   E4            6         SUBD   ,S
0027 20   F0            3         BRA    DIV2
                          * HERE ON OVERFLOW
0029 CC   FFFF          3 DIV5    LDD    #$FFFF
002C ED   C4            5         STD    ,U
                          * HERE TO EXIT
002E 37   06            6 DONE    PULU   D         GET QUOTIENT IN D
0030 35   90            8         PULS   X,PC      CLEAN STACK & EXIT
```

+16/16 BIT DIVIDE +

```
                      *
                      * 50 BYTES
                      * 24 INSTRUCTIONS
                      * TIMING:
                      *                       SINGLE LOOP           TOTAL
                      *                     BEST     WORSE      BEST      WORSE
                      *  INIT                31       31         31        31
                      *  FIRST LOOP          38       47         38        47
                      *  (TO DIV1)
                      *  MIDDLE LOOPS        35       44        525       660
                      *  (DIV1 TO DIV1)
                      *  LAST LOOP           32       32         32        32
                      *                                         -----     ------
                      *                                         626       770
                      *
            0000              END

TOTAL ERRORS   00000
TOTAL WARNINGS 00000
```

```
*******************************************************************
*                                                                 *
*                   CODE                                          *
*                                                                 *
*  8X16 MULTIPLY SUBROUTINE - NON REENTRANT                       *
*   8 BIT X 16 BIT = 16 BIT                                       *
*   16 BIT PRODUCT IS MS 2 BYTES ROUNDED WITH HI BIT OF LOWEST    *
*   (DROPPED) BYTE                                                *
*                                                                 *
*   (T:U) X V =          UVH:UVL                                  *
*                      + TVH:TVL                                  *
*                      -------------                              *
*                                                                 *
*   ENTRY FORMAT:                                                 *
*    X REGISTER CONTAINS ADDRESS OF 16 BIT MULTIPLICAND           *
*    A REGISTER CONTAINS 8 BIT MULTIPLIER                         *
*                                                                 *
*   RETURN FORMAT:                                                *
*    A,B REGISTERS CONTAIN 16 BIT PRODUCT, MS IN A                *
*                                                                 *
*******************************************************************


MUL8X16  STA A   TEMP          V -> TEMP
         LDA B   1,X           U -> B
         MUL                   U X V

         ADC A   #0
         STA A   TEMP+1        UVH ROUNDED -> TEMP+1

         LDA A   0,X           T -> A
         LDA B   TEMP          V -> B
         MUL                   T X V
         STA A   TEMP          TVH -> TEMP
         CLR A
         ADDD    TEMP          0:TVL + TVH:(UVH ROUNDED) -> A:B

         RTS
```

```
***********************************************************************
*                                                                     *
*                    6801   CODE                                      *
*                                                                     *
*  TWO DIMENSIONAL TABLE LOOKUP - NON REENTRANT                       *
*                                                                     *
*   ENTRY FORMAT:                                                     *
*    X REGISTER CONTAINS TABLE START ADDRESS                          *
*    A REGISTER CONTAINS INPUT VALUE (X)                              *
*    B REGISTER OPTIONALLY CONTAINS THE MINIMUM VALUE OF THE          *
*               INDEPENDENT VARIABLE                                  *
*                                                                     *
*   RETURN FORMAT:                                                    *
*    A REGISTER CONTAINS THE RESULT Y = F(X)                          *
*                                                                     *
*   NOTES:                                                            *
*    1. MAXIMUM DIFFERENCE BETWEEN TABLE VALUES MAY BE 255 ($FF)      *
*    2. THIS ROUTINE PERFORMS LINEAR INTERPOLATION FOR TABLES         *
*       WITH 8 BIT UNSIGNED ENTRIES AND FIXED SPACING , H, OF THE     *
*       INDEPENDENT VARIABLE.  THE INTERPOLATION IS OF THE FORM:      *
*          Y = Y1 + (Y2 - Y1) * (X - X1) / H                          *
*       WHERE: Y, THE RESULT, IS INTERPOLATED FROM THE TABLE          *
*              VALUES Y2, Y1 FROM X2, X1.  X2 AND X1 ARE ADJACENT     *
*              POINTS WHICH BRACKET THE INPUT X.  H IS FIXED AT       *
*              16.                                                    *
*                                                                     *
***********************************************************************


LKUP2DA  SBA                       OFFSET = X - XMIN

LKUPQ    LDA B    #$10             SEPERATE OFFSET INTO TABLE ENTRY AND
         MUL                       ..INTERPOLATION PORTIONS
         PSH B
         TAB
         ABX
         LDA D    0,X              PICK UP Y1 AND Y2

LKUPR    STA A    TEMP
         SBA                       Y1 -Y2
         PUL A
         BCS      LUP1

         MUL                      Y2 < Y1
         NEG A                    ..COMPUTE Y1 + (Y2 - Y1) * (X - X1) / 16
         ADD A    TEMP
         ASL B
         SBC A    #0              ..ROUND DOWN IF NECESSARY
         RTS                      ..AND RETURN

LUP1     NEG A                    Y2 > Y1
         MUL                      ..COMPUTE Y1 + (Y2 - Y1) * (X - X1) / 16
         ADC A    TEMP

                                  ..ROUND UP IF NECESSARY
         RTS                      ..AND RETURN
```

ADDENDUM TO PRELIMINARY DESIGN DATA PACKAGE

DESIGN OPTIONS

DESIGN OPTIONS

With the extra time that has been allowed for the submission of the NTHV Phase II proposal, we have had an opportunity to investigate several additional items of the preliminary design. The items investigated are powertrain configurations and a turbo-generator system to regain some of the energy lost in the exhaust of the diesel engine.

POWERTRAIN OPTIONS

Four additional powertrain configurations were shown to be viable alternatives to the system recommended in the Preliminary Design Data Package. These systems offer various advantages of simplicity and reduced risk in development, or improvements in overall energy efficiency. However, these gains are balanced by increased petroleum or electric energy usage, greater risk in development, or loss of simplicity of operation for the driver. The four potential systems are listed below.

In the first alternative, the motor as well as the engine feeds its power into the torque converter of the automatic transmission. This would reduce the mechanical modifications to the base transmission; there would be no modifications required of the transmission housing or the major internal parts of the transmission. However, for this proposed system to work, it would be necessary to accelerate the vehicle to 15 km/hr with the engine or modify the motor control system to permit the motor to start the vehicle from rest at low acceleration levels.

The second alternative powertrain has the engine driving through the torque converter and transmission and the motor connected to the driveline at the transmission output. This would result in some loss in vehicle acceleration above 70 or 80 km/hr, but would produce a very simple transmission system. If the Chrysler A-404 transmission were used, the motor input could be installed with no modification to the basic transmission, and no major change to the existing transmission shifting control system would be needed. This powertrain would also require the use of the variable-fill fluid coupling, the slipping clutch, or the other starting methods referred to above. The overall energy efficiency of this system will be almost identical to the system in the preliminary design, up to approximately 75 km/hr.

The third alternative is the automatically controlled manual transmission, such as the one Minicars is currently developing for the Research Safety Vehicle program. This powertrain will give fuel and energy efficiency almost as good as a full manual transmission (being reduced by the oil pump needed to operate the gear shift and clutch control systems), without the need for any greater driver skill than an automatic transmission. This transmission does have the disadvantage that it will shift, with consequent interruption of power flow, at relatively unexpected times. While such a system would have questionable driver acceptability, the potential gains in vehicle efficiency make it a viable alternative. This transmission system does require the most development work of the various combinations proposed, and is, therefore, the least likely of the systems to result in a fully consumer acceptable package during the period of the NTHV program.

The fourth alternative system is a fully manual transmission, with the shifting and clutch operation under the control of the driver. To aid the driver in the complex selection of gear ratios in the hybrid vehicle, the vehicle control computer would still determine the optimum transmission ratio, but, instead of selecting that gear ratio, the computer would indicate the gear selection to the driver by means of a display, and the driver would do the actual shifting. This powertrain system would result in the best energy efficiency of all of those considered, and the lowest risk of failure. However, the high efficiency is only available if the driver follows the shifting instructions, and it also requires the driver to have enough skill to drive a vehicle with a manual transmission.

In this addendum the first and fourth powertrains will be discussed in detail, since they show the range of powertrain complexity and variations that Minicars is prepared to consider for the NTHV program.

MODIFIED AUTOMATIC TRANSMISSION POWERTRAIN (ALTERNATIVE 1)

The powertrain recommended in the main text of this Data Package consists of a turbo-charged diesel engine connected to an automatic transmission through a lock-up torque converter, and an electric motor connected to the transmission (at the converter output) through either a slipping clutch or a variable-fill fluid coupling. The motor is not connected through the torque converter, due to the inherently poor match between the characteristics of a hydrodynamic torque converter and an electric motor which uses field control alone. The resultant powertrain is one that we feel will give the best overall package for the NTHV, but its development will require extensive modification and rebuilding of the Turbo Hydramatic 125 transmission, as well as a major development program for the slipping clutch and/or variable-fill fluid coupling.

As an alternative to this powertrain layout we have considered a powertrain in which both power sources feed into the torque converter, and no major changes are required to the transmission itself, except for the electric shifting control (and there is also an alternative for this). The result is a clutch/transfer case that is manufactured as a one piece assembly and is mounted between the diesel engine and the automatic transmission. This assembly also includes the mounting for the electric motor and the chain drive between the motor and the transmission.

Figure 1 is a schematic of this proposed unit, connected to the Turbo Hydramatic 125 transmission. This combination includes the flywheel for the diesel engine, a clutch to disconnect the diesel engine, a clutch to disconnect the electric motor, a chain drive to transfer the power of the electric motor, and an input to the stock torque converter of the TH 125 transmission. The torque converter of the transmission remains the standard unit that is normally installed in this transmission, and the lock-up clutch option is either added to this converter or deleted in the interest of simplicity.

If no substitutions for the slipping clutch were made, this proposed driveline would have a major shortcoming - the poor compatibility between the field control electric motor and the torque converter. Figure 2 shows the low speed acceleration potential of the proposed compound electric motor and the standard TH 125 torque converter. This curve shows the maximum acceleration, which is of a satisfactory level, as well as the minimum acceleration level. As explained in the main text of the Data Package, this minimum acceleration level is the lowest acceleration that can be achieved for this motor/torque converter package, and, since it produces a maximum 0-20 km/hr time of only 4 seconds, it would be very difficult to drive this vehicle in slow traffic with electric power only.

Instead of either the slipping clutch or the variable-fill fluid coupling proposed earlier, there are several other methods that

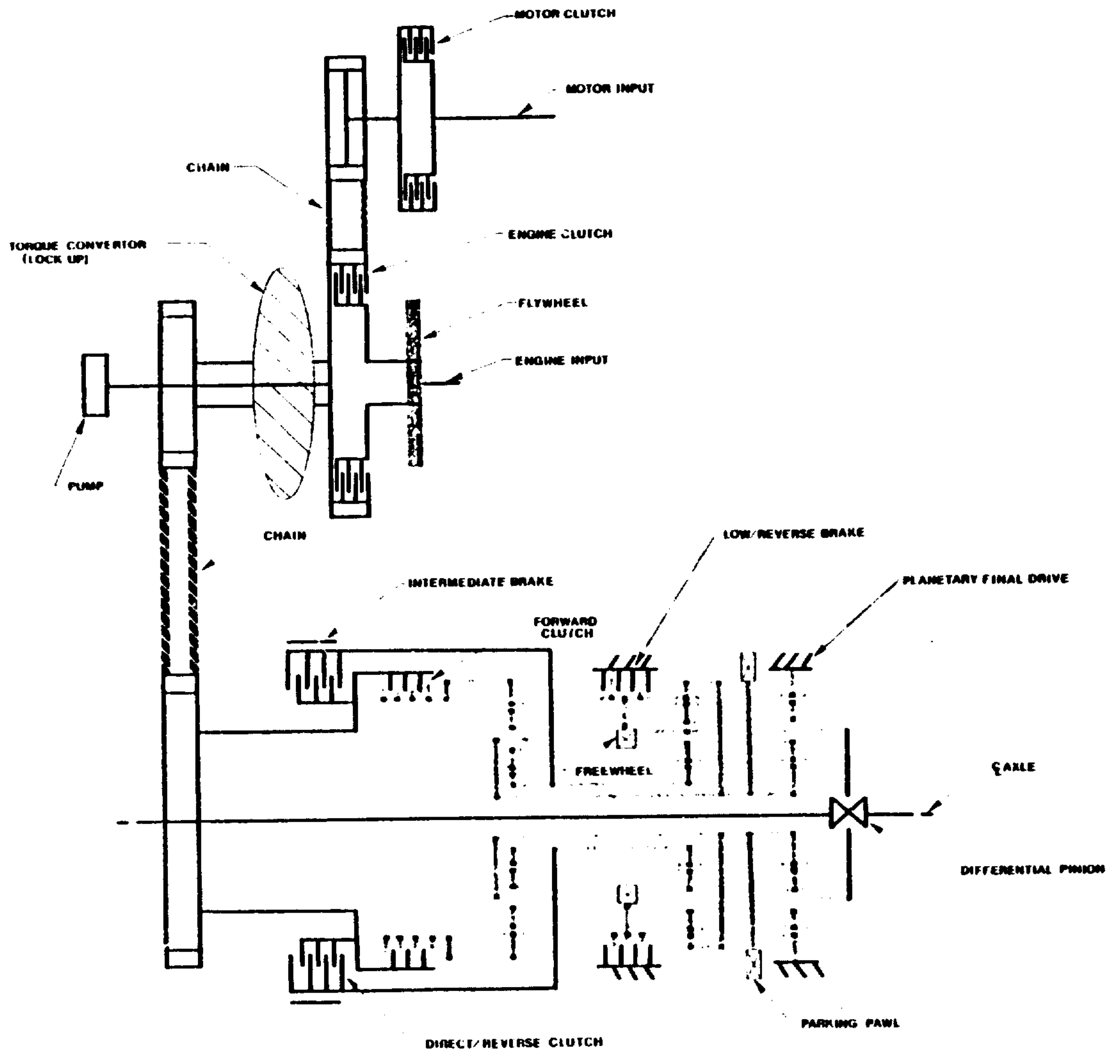


Figure 1. General Motors TH-125 Automatic Transmission Modified for NTHV

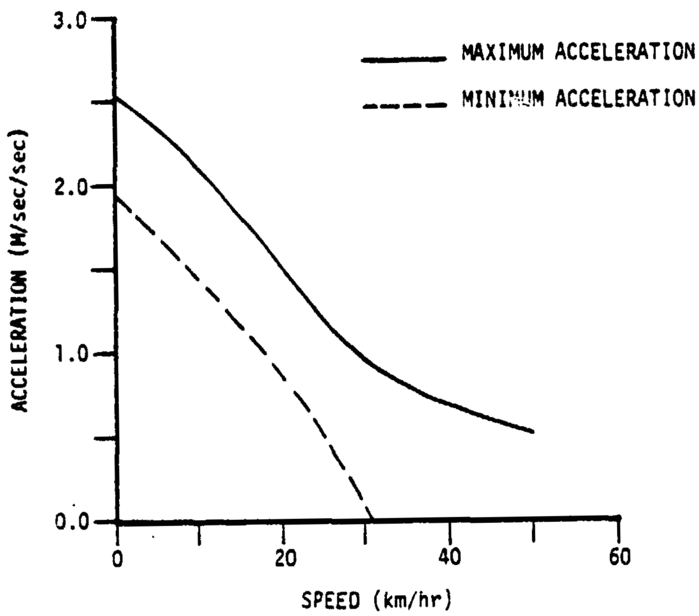


Figure 2. Low Speed Maximum and Minimum Acceleration, Proposed NTHV Motor with TH 125 Torque Converter

could be used to start the vehicle from rest while still using only field control for the motor. All of these methods introduce some loss of efficiency, compared to the originally proposed systems, but they will only have this inefficiency for short periods of time at low vehicle speeds. Thus there would be a very small effect on overall operating efficiency. (Armature chopper control was not seriously considered due to its weight, cost and noise potential, and because it would be used only for short durations at low vehicle speeds, due to the transmission.)

The possible methods of starting the vehicle to run in electric drive include using the engine from 0 to 15 km/hr or such systems as field forcing of the motor to lower its base speed, using a one or two stage starting resistor under load, or series/parallel battery switching. All of these systems introduce some complexity or complication, but can provide an alternative for starting the vehicle from rest. Figure 3 shows the maximum and minimum accelerations that could be expected with one or more of these starting methods. The minimum acceleration has been lowered significantly compared to Figure 2; the driver could now take as much as 14.5 seconds to accelerate from 0 to 20 km/hr. This would provide much more acceptable operation in heavy traffic conditions.

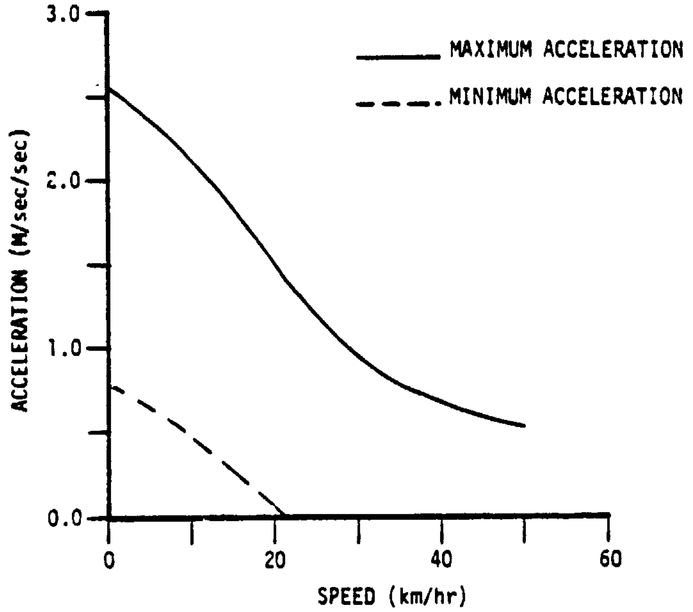


Figure 3. Low Speed Maximum and Minimum Acceleration, Proposed NTHV Motor with Modified Motor Starting, TH 125 Torque Converter

MANUAL TRANSMISSION (ALTERNATIVE 4)

At the other end of the complexity spectrum is the fully manual transmission. This transmission would be the simplest, lightest, cheapest, most efficient, and easiest to develop of any of the combinations considered during the Phase I NTHV program. However, it requires a certain level of competence to drive (as with any manual transmission), and a greater level of driver understanding to obtain the potential high efficiency from the powertrain. To improve the system operation, we would have the control computer signal (display) the optimum gear to use for the current driving condition. If the driver followed these recommendations, the overall vehicle efficiency would be outstanding; if he did not follow the recommendations, then powertrain efficiency would suffer. The vehicle control computer would still be responsible for the selection of the power source or sources to drive the vehicle.

Figure 4 shows a possible layout for the hybrid powertrain using the standard General Motors X-body 4-speed manual transmission. The engine drives the transmission through a normal clutch system. The electric motor drives the opposite end of the transmission

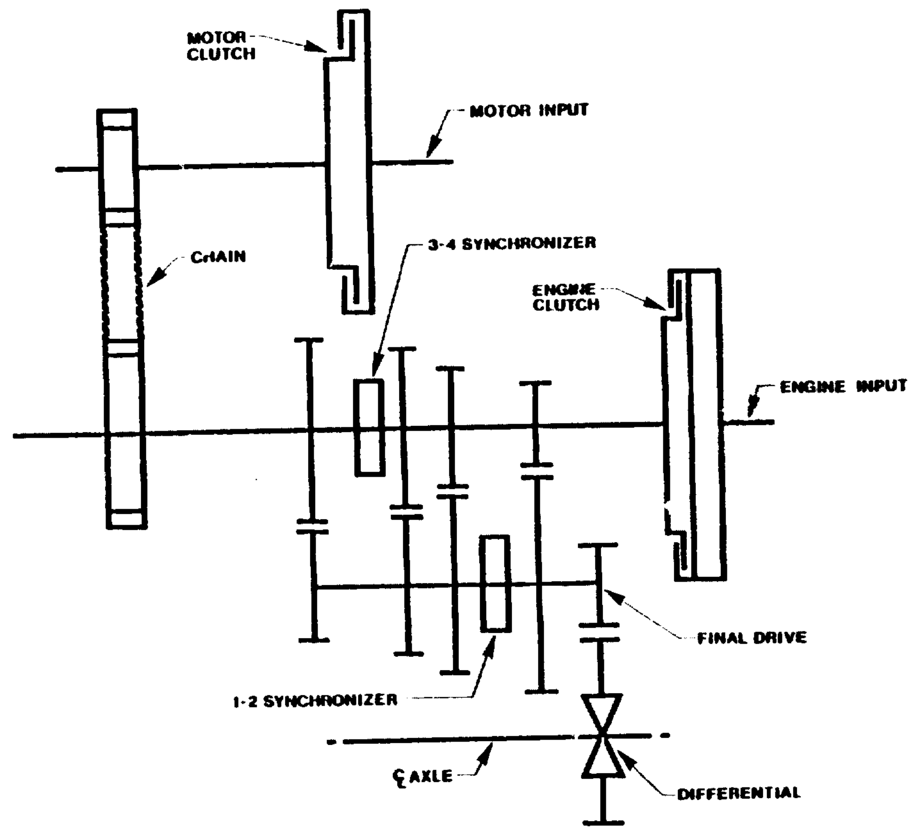


Figure 4. General Motors X-Body 4-Speed Manual Transmission Modified for NTHV

input shaft through a chain drive from a second clutch on the motor. The clutch pedal would operate either or both clutches, depending on the operational strategy of the vehicle. Figure 5 shows the proposed system of controlling the two clutches. Both clutches would be operated by a normal cable system, but the clutches would also have a hydraulic control cylinder in parallel with the cable operation. This cylinder would be operated by an on/off electric control valve that would connect the cylinder to the oil pressure of the power steering system. The control computer would open the valve for the power source that was not to be used, which would hold its clutch disengaged regardless of the clutch pedal position. The clutch pedal would then connect only the power source selected by the computer. Under command of the computer, the second power source could be added at any time. The control valve would be closed, permitting the oil in the cylinder to escape through an orifice, which would engage the clutch of the second power source, bringing it on line.

This manual transmission system would be by far the easiest powertrain system to develop for the NTHV, and the one with the lowest risk of failure. With the currently increasing acceptance of manual transmissions by the public as a whole, in keeping with the greater desire by the public for improved fuel economy, a manual transmission on the NTHV becomes a more practical alternative.

POWERTRAIN SUMMARY

The powertrain originally proposed in the Preliminary Design Data Package is the most versatile and the most likely to provide a fully acceptable vehicle to the ultimate customer. However, it is a complex system that will be more expensive to develop than either of the other powertrains. As shown in the addendum to the Design Trade-Off Study Report the manual transmission based powertrain (alternative 4) would be slightly more efficient than the original design, and the automatic transmission powertrain (alternative 1) would be almost as fuel economical (with the engine going on and off), less complex and easier to develop than the original design. Therefore, JPL will be asked to clarify and re-interpret the consumer acceptance and petroleum consumption requirements before a choice is made.

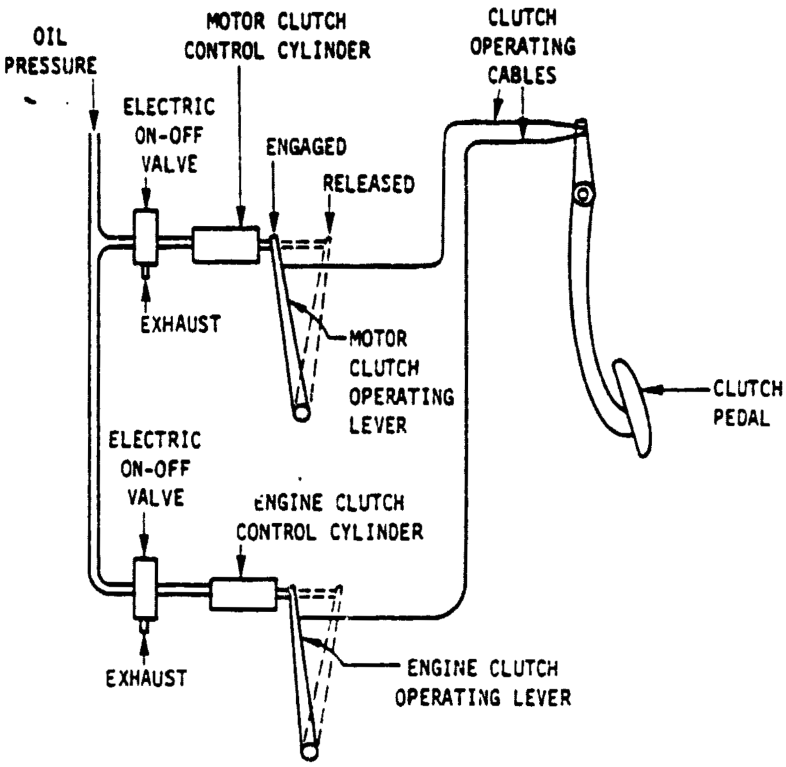


Figure 5. Clutch Operating System for NTHV
with Manual Transmission

ACCESSORIES AND ACCESSORY DRIVE

In accordance with the recommendations implicit in the addendum to the Design Trade-off Studies Report, the preliminary design has been modified to utilize the variable ratio pulley system described on page 102 of this volume. Furthermore, the addendum on accessories in the Design Trade-off Studies Report is here with incorporated in the preliminary design and substituted for the material in Subsection 4.4.1, pages 100 to 102.

TURBO GENERATOR FOR IMPROVED THERMAL EFFICIENCY

As discussed in the addendum to the Trade-Off Studies Report, there is significant potential for reduction in overall petroleum usage by regaining some of the waste energy from the exhaust of the diesel engine and using it to recharge the traction batteries of the NTHV. This energy would be additional to the energy regained to drive the turbocharger of the diesel engine.

As a rule of thumb for internal combustion engines, approximately 30 percent of the total chemical energy in the petroleum is converted into useful work, 30 percent is lost in the exhaust, 30 percent is lost to the cooling system, and the remaining 10 percent is lost in engine friction. A diesel engine is more efficient than a gasoline engine, but its losses are still substantial. Of the various energy losses, those to the exhaust system are the best candidates for recovery (since the temperature of exhaust gasses is much higher than the temperature of cooling water).

A turbocharger is the most widely used means of regaining some of the waste energy from the engine exhaust. Turbochargers are designed to provide most of their energy return at high engine speeds and at high loads, just where it is needed for maximum increases in the power output of the engine. However, a turbocharger offers very little gain at the levels of engine power and speed where most driving is done. At 2500 engine rpm and 10-12 kW power output, which is a typical operating condition for the NTHV when it is in the diesel mode, the turbocharger gives a potential power increase of less than 10 percent. There is, nevertheless, a considerable amount of energy left in the exhaust (after the turbocharger) that could be put to use. At these operating conditions the total energy in the exhaust, if brought down to ambient temperature, is over 9 kW. It is not practical to regain all of this power, but it is reasonable to expect to regain approximately one-third, or about 3 kW. The trade-off studies have shown that regaining this much power would result in an overall reduction of 12 percent in yearly petroleum usage.

There are several potential methods of regaining this energy from the exhaust system, but only two seem to merit study - a Rankine bottoming cycle and an exhaust turbine. Both of these systems are now being developed under Department of Energy contracts for large highway trucks, and both have given indications of significant gains in overall fuel efficiency.

The Rankine bottoming cycle takes the heat from the exhaust of a turbocharged diesel engine and transfers it to an organic fluid (water and trifluorethanol). This heated fluid is used to drive a turbine through a Rankine (steam) cycle. The turbine is then geared to the engine crankshaft. This system is currently working quite well, but its size and complexity make it difficult to fit into a hybrid passenger vehicle.

The other system under development for DOE is a turbo-compound diesel. In this engine the exhaust output from the turbocharger is fed into a second exhaust turbine wheel, which is connected through a reduction gear to the crankshaft. The return of this extra power to the crankshaft results in significant improvements in overall fuel consumption of the diesel engine. This turbo-compounding system appears to work very well on a large truck engine, which in over the road service is essentially a constant speed/constant power device. For passenger car usage, however, the variations in engine speed and power are much greater, which makes it very difficult to obtain major gains in efficiency from a turbo-compound system geared to the crankshaft.

There is a much more promising way to return the lost energy of the exhaust system to the NTHV - by means of a turbine driven generator that will return the regained energy to the traction batteries. This energy can then be used for electric topping in the diesel operating mode, and, when enough energy is stored, the vehicle can revert to the electric driving mode with diesel topping.

Figure 6 shows two possible layouts for a turbo-generator system to regain some of the lost energy from the exhaust system. The two systems differ only in the relative layout of the two turbine devices (turbocharger and turbo-generator): in series or parallel. In either case the two turbines would be sized for different functions. The turbine of the turbocharger would stay the same as it currently is and would be sized to drive the compressor for maximum boost at high engine speeds and high engine output levels. This is what is needed, since the only time that a turbocharger is of benefit to a diesel engine is when the required output is greater than what could be provided by the same engine in normally aspirated form. In short, the turbocharger is only used for maximizing the wide open throttle performance of the engine. The turbo-generator turbine, on the other hand, is sized for maximum efficiency at the normal speeds and loads of the engine. This would be in the 2000-3000 rpm range and at powers up to 15-20 kW. A turbine of this size would be less efficient at higher loads, and would probably have to be bypassed at high exhaust flows to keep from causing excessive exhaust back pressures. The potential for regaining otherwise lost energy under normal driving conditions would be significant, however.

The simplest system would be to connect the turbine directly to a high speed alternator. Alternatively, the turbine speed could be reduced to allow the use of a more conventional, lower speed alternator. In either case the alternator should be capable of generating up to 3-4 kW of electric power to feed back to the battery system.

Such an alternator was developed in 1953 for the Sidewinder missile and was used in the production of that weapon system until recent years. The alternator consists of a laminated unwound rotor switching the magnetic field produced by two permanent magnets through field coils in the stator. The system is capable of operating at extremely high temperatures (well above the 700°F of exhaust gases) and at shaft speeds from 60,000 to 100,000 rpm. It is driven by a turbine wheel spun by a solid propellant hot gas generator. Such a system can have a small size at any output power level because of the high speeds involved.

A turbo-generator system could have a significant effect on the overall operating efficiency of the NTHV, but it would be essentially an add-on feature. It would improve the vehicle, but the vehicle would still produce major petroleum savings without its use.

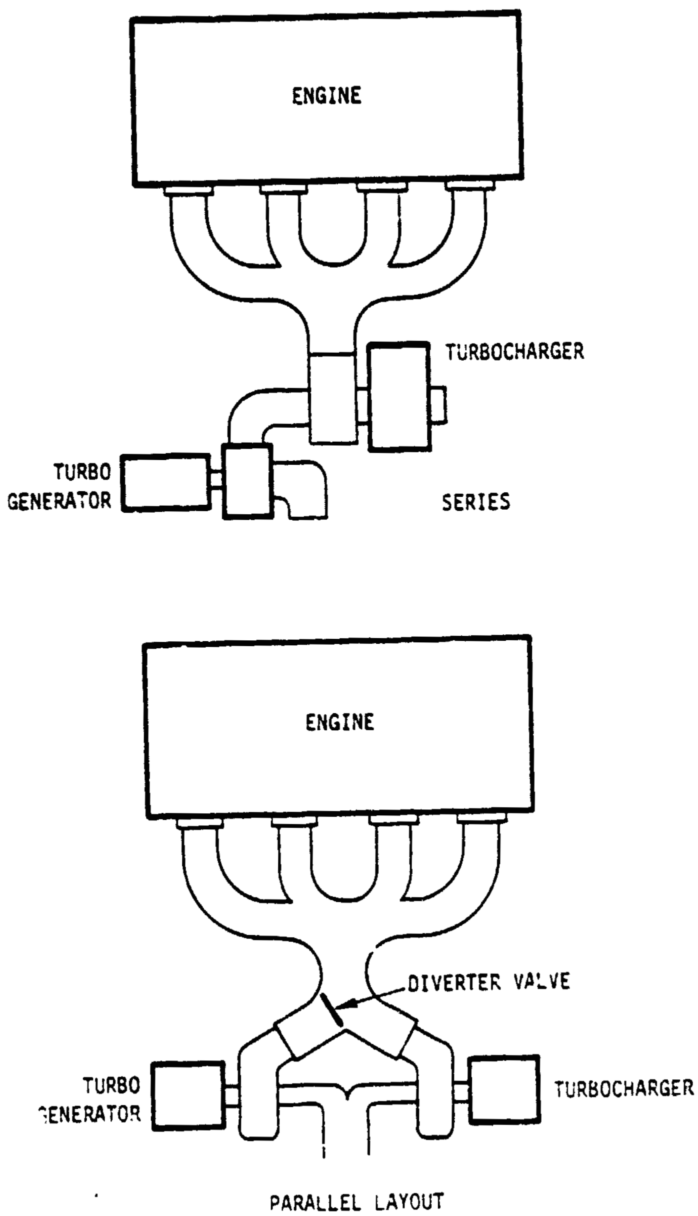


Figure 6. Turbo-Generator for NTHV

APPENDIX D

SENSITIVITY ANALYSIS REPORT

SR-4500-05-79(2)

NEAR TERM HYBRID PASSENGER VEHICLE DEVELOPMENT PROGRAM

PHASE I

CONTRACT 955188

SENSITIVITY ANALYSIS REPORT

25 May 1979

Submitted to:

JET PROPULSION LABORATORY
CALIFORNIA INSTITUTE OF TECHNOLOGY
4800 Oak Grove Drive
Pasadena, California 91103

Submitted by:

MINICARS, INCORPORATED
55 Depot Road
Goleta, California 93017

This report contains information prepared by Minicars, Inc. under JPL sub-contract. Its content is not necessarily endorsed by the Jet Propulsion Laboratory, California Institute of Technology, or the National Aeronautics and Space Administration.

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SUMMARY

This report describes the Minicars, Inc., Near Term Hybrid Vehicle (NTHV) sensitivity studies, which were carried out in close conjunction with the NTHV mission analysis, References 1 and 2, and design trade-off studies, Reference 8. The contents of this report are summarized in accordance with the format of the Jet Propulsion Laboratories' Data Requirement Description given in Reference 10.

1. DATA SOURCES

The principal data sources for tripmaking behavior were References 1, 2, 3 and 4. A critique and synthesis of these sources is presented in Section 1.

The principal data source for the NTHV design candidate system characteristics was Reference 8 which, in turn, contains additional references.

Further key data sources are listed at the end of this report and as part of Reference 1.

2. METHODOLOGY

The principal analytical tool used in this study is a Monte Carlo tripmaking computer program, TRAVEL, similar to that described in Reference 6. The program, however, contains a number of new features, including the method of assigning increased travel to trip length and trip frequency in any desired proportion. Trip distributions by mission are another feature not found in earlier programs.

The principal output of the program is the amount of fuel used by any candidate system when driven over any particular mission. A detailed computer program description is given in Section 2.

Results from program TRAVEL were closely integrated with those of two other large scale Minicars computer programs - CARSIM, which computes fuel consumption over any particular driving cycle, and the Present Value Life Cycle Cost Analysis computer program, which calculates candidate system costs. These programs are documented in Appendices A and C of Reference 8.

3. INTERIM DATA

Interim data were developed as a matter of course as part of this study and are found throughout the report. Nothing of particular note needs to be mentioned here.

4. RESULTS

a. Mission analysis results are quite sensitive to the large uncertainties in the "tails" of the trip length distributions, i.e., in the length and frequency of very long trips.

b. Mission analysis results are insensitive to the manner in which increased travel is apportioned between longer trips and more frequent trips.

c. Variations in annual travel of the magnitude given in Reference 11 do not strongly affect the choice of the preferred candidate system; but they do affect the total fuel consumed. The change in fuel used can be approximated by assuming that the incremental distances are driven on the internal combustion engine only.

d. Variations in fuel and electricity prices in the range specified in Reference 11 do not significantly affect the design trade-off study results.

e. Larger variations in fuel prices do have important effects. The break-even price for motor fuel at which the NTHV ten-year life cycle cost becomes less than that of the reference vehicle are:

   Mission AA, All Travel - 55¢/ℓ or $2.08/gal

   Mission A, Restricted General Purpose Travel - 65¢/ℓ or $2.45/gal

   Mission C, Family and Civic Business - 30¢/ℓ or $1.13/gal.

f. The number of passenger cars affects the national petroleum consumption and thus, indirectly, the total petroleum imports and the balance of payments. Within the range specified by JPL, these effects are moderate. Details will be found in Table 12, Section 6.2.

5. CONCLUSIONS

Neither the mission analysis nor the design trade-off study results are strongly sensitive to variations in annual travel, the number of passenger cars, and fuel and electricity prices, so long as these variations do not exceed the limits set by JPL in Reference 11.

All aspects of the evaluation of the NTHV are very sensitive to the selection of the reference vehicle.

PART I

ASSUMPTIONS AND METHODOLOGY

SECTION 1

DISTRIBUTION OF TRIP LENGTH, TRIP FREQUENCY, DAILY AND ANNUAL TRAVEL

1.1 DATA SOURCES AND THEIR INTERPRETATION

In Reference 1 the data pertaining to travel patterns for a series of missions were developed principally on the basis of References 2 and 3. The purpose of such data is, of course, to provide the requirements for, and to test the adequacy of, candidate hybrid vehicle systems.

In this report trip and travel patterns are further examined, particularly in the light of the additional data given in Reference 4. An attempt was made to unify the various data bases so as to provide the best possible starting point for the sensitivity studies. This work is described below.

1.1.1 Nationwide Personal Transportation Study

The most comprehensive set of trip making data is given in the Nationwide Personal Transportation Study (NPTS)[3], which was designed to obtain up-to-date information on national patterns of travel. Data were collected in 1969 and early 1970 by the Bureau of the Census. In essence, people in a representative set of geographical locations were asked to reconstruct where and for what purpose they traveled on a specified day. From this, a variety of determinants of travel behavior were aggregated. The ones of greatest interest to us are trip purpose, average number of trips per day, average trip length, annual travel and trip length destribution. Reference 3 does not make it entirely clear how results were obtained or how they are to be interpreted. There are many references to "unpublished tables." For example, was the average trip length of 14.3 km per household calculated from the observed trip length distribution or, as seems more likely, was it obtained by summing the length of all trips and dividing by their number? (Our own comparison of these two approaches follows shortly.)

Again, Report 2 in Reference 3 gives the annual distance traveled per household as 18,676 km per year, while Report 7 favors 19,964 km per year. A possible reason for this discrepancy is that one result may have been derived by multiplying the product of average number of trips per day and average trip length (i.e., average

C-4

daily travel—assuming independence) by 365, while the other was obtained directly from a survey question which asked respondents to estimate how far they drove during the previous year.

These are not meant to be niggling objections. We merely want to emphasize that all of the available tripmaking data is of but limited accuracy, and that we must keep this well in mind when using it to evaluate closely competing vehicle designs.

Some particular difficulties with the NPTS are that results are presented on a per household rather than on a per car basis, and that no data concerning the distribution of daily travel is given. It is also not clear whether the "per household" refers to all households surveyed or only to "active households," i.e., those that took at least one trip on the survey day. From the internal evidence it appears more likely that all surveyed households were included in the figures and this is what we have assumed.

There has been a consistent trend in the U.S. in recent years for the number of cars to closely approximate 1.25 per household[5]. Using this figure, the 3.81 trips per household per day equate to 3.05 trips per car per day. Trip length distributions are, of course, unaffected by whether data is collected by household or by car.

Since daily travel distributions are not given in the NPTS, they must be generated from the distribution of trip length, which is available, and from an assumed distribution of trip frequencies. This was done in a manner similar to that of Reference 6, as will be described below.

1.1.2 Los Angeles and Washington D.C. Origin-Destination Studies

General Research Corporation (GRC) of Goleta, California, under subcontract to Minicars, Inc., obtained and analyzed trip making patterns from computer tapes of Los Angeles (1971) and Washington, D.C. (1968) origin-destination travel surveys (Reference 2). The Washington data appears suspect and, therefore, we concentrate on the Los Angeles results.

This study contains distributions of trip frequency and daily travel but not of trip length. Further difficulties are that the data is not diaggregated by trip purpose (mission), and while the results are on a per car basis, cars which took no trips on the survey day were not included.

The GRC results were modified to allow for some of the above shortcomings by means of the following analysis:

Let

n_i = the number of cars which took i trips on the survey day

M = the greatest value of i

λ = the average number of trips per car on the survey day

Then,

$$\lambda = \frac{\sum_{i=0}^{M} i n_i}{\sum_{i=0}^{M} n_i} = \frac{\sum_{i=1}^{M} i n_i}{n_o + n_A} = \frac{T}{n_o + n_A} \qquad (1)$$

Where,

T = total trips travelled

n_A = number of active cars

n_o = number of inactive cars

$$\lambda_A = \frac{T}{n_A} = \text{average trip length per active car} \qquad (2)$$

Substituting (1) in (2),

$$\lambda_A = \frac{\lambda}{n_A/(n_o + n_A)} \qquad (3)$$

In Reference 4 (see below), the ratio of active car days compared to total car days, averaged over six cities, was found to be

$$\frac{n_A}{n_o + n_A} = \frac{4475}{4797} = 0.933 \qquad (4)$$

Thus,

$$\lambda = 0.933\ \lambda_A = 0.933 \times 3.82 = 3.56$$

The GRC study also gives an average daily travel distance of 47 km. Using the above derived trip frequency, this implies an average trip length of

$$L = 47/3.56 = 13.2 \text{ km}$$

1.1.3 Systems Development Corporation Driving Patterns Study

Systems Development Corporation (SDC) of Santa Monica, California, performed a study[4] to survey the characteristic use patterns of privately operated automobiles in six major metropolitan areas: Los Angeles, Houston, Cincinatti, Chicago, Minneapolis, St. Paul, and New York City. A major finding of the study was that driving patterns as expressed by such variables as trip lengths and trip frequencies in all these cities were, on the whole, quite similar.

In the SDC study, 946 private vehicles were selected, instrumented, and observed over a total of 4,797 car days. The test sample was made up of "employed volunteers" – employed by SDC one assumes. Thus the sample consists of people who are almost certain to drive to work and who are, in general, keener drivers than most. We have attempted to allow for this bias in the manner shown below.

Let

p = the probability that a car is driven to work

N = the total number of cars.

λ = the average number of trips for all purposes per car per day.

Np = the expected number of cars driven to work.

Then,

$2Np$ = expected number of work trips

$N\lambda$ = expected number of all trips

Let μ = the fraction of all trips which are work trips.

Then,

$$\mu = \frac{2Np}{N_\lambda} = \frac{2p}{\lambda}$$

and,

$$p = \frac{\mu\lambda}{2}$$

Using $\mu = 0.319$ and $\lambda = 3.05$ from Reference 3 and the discussion above, we obtain,

$$p = \frac{0.319}{2} \times 3.05 = 0.48$$

This happens to be the fraction of all workers who drive to work, say q, as given in Report 8 of Reference 3.

While p and q are not identical quantities, they are obviously closely related and their agreement enforces the credibility of our analysis.

The fact that only about half of all workers drive their own cars to work may come as a surprise to the reader. It did to us. However, Report 8, Page 27, Reference 3, gives the nationwide results shown in Table 1.

The number of work trips per car per day is $2Np/N = 2p = 0.96$, or $1.25 \times 0.96 = 1.2$ trips per household. This compares reasonably well with the 1.4 work trips per household per day given in Reference 3.

SDC gives the six-city average of trips per car per day as 4.57. Because of the special nature of the sample, we may assume that each car took two work trips per work day instead of the 0.96 national average derived above. Accordingly we estimate $\lambda = 4.57 - (2 - 0.96) = 3.53$. The average daily distance calculated by SDC is 34.19 miles which we adjust on the basis of the foregoing to $34.19 \times 3.53/4.57 = 26.4$ miles or 42.5 km. The average trip length, L, now becomes $42.5/3.53 = 12.0$ km.

For Los Angeles, the SDC study estimates an average of 4.42 trips and 32.05 miles/per car per day. In a manner identical to that used above, this adjusts to $\lambda = 3.38$ and $L = 11.7$ km.

Table 1. Percent of Employed Persons Classified by Major Mode of Home-to-Work Transportation

Mode of Transportation	Percent Distribution
Automobile	67.4
Driver	48.4
Passenger	19.0
Public transportation	7.2
Truck	5.7
Walking	5.0
Automobile, public transportation, and other	2.9
All other	11.8
Total	100.0

Our adjusted trip making results (obtained from References 2, 3, and 4) are summarized in Table 2. The table on the whole shows good agreement among the various studies and lends credence to the assumptions that the average number of trips per car per day is about 3.5, that the mean trip length is of the order of 12 km, and that on the average, cars travel approximately 40 km per day.

1.2 TRIP LENGTH DISTRIBUTIONS

Table 3 shows the cumulative distribution of trip length for various trip purposes as given in Report 10, P. 16, of the NPTS[3]. Data on trip length, trip frequency, and fraction of trips by trip purpose, taken from the same reference, are also presented. The data of Table 3 are plotted in Figure 1 in which the Family and Civic trip purposes were combined into a single mission, Mission C, as discussed in Reference 1. Mission B corresponds to the "Commute to Work" trip purpose, while Mission A is "Limited all Purpose Travel." In Reference 1, Mission A was called "All Purpose City Driving," but we now feel that this is too restrictive since small town or country driving is equally acceptable provided no very long trips are taken. Thus, the new name appears to better describe the intended mission.

Table 2. Summary of Trip and Travel Parameters*

Location	Source	Reference	Approximate Date	Average Trip Length (km)	Average Number Trips/Car/Day	Average Daily Distance Per Car (km)	Average Annual Distance/Car (km): Estimated as 365 × Daily Distance	and Estimated From Gas Tax Data
Los Angeles	GRC	2	1967	12.9	3.64	47.0	17,155	
Los Angeles**	SDC	4	1970	11.7	3.40	38.6	14,089	
6 Cities	SDC	4	1970	12.0	3.45	41.5	15,148	
Nationwide	NPTS	3	1969-70	14.3	3.05	43.8	15,987	
Nationwide	Schwartz	7	1972	14.3	3.145	44.9	16,386	
Nationwide	Highway Statistics	8	1969					15,749
		8	1972					16,396
		8	1974					15,285

*Adjusted in accordance with the discussion of this section.

**Trips longer than 50 miles were not considered.

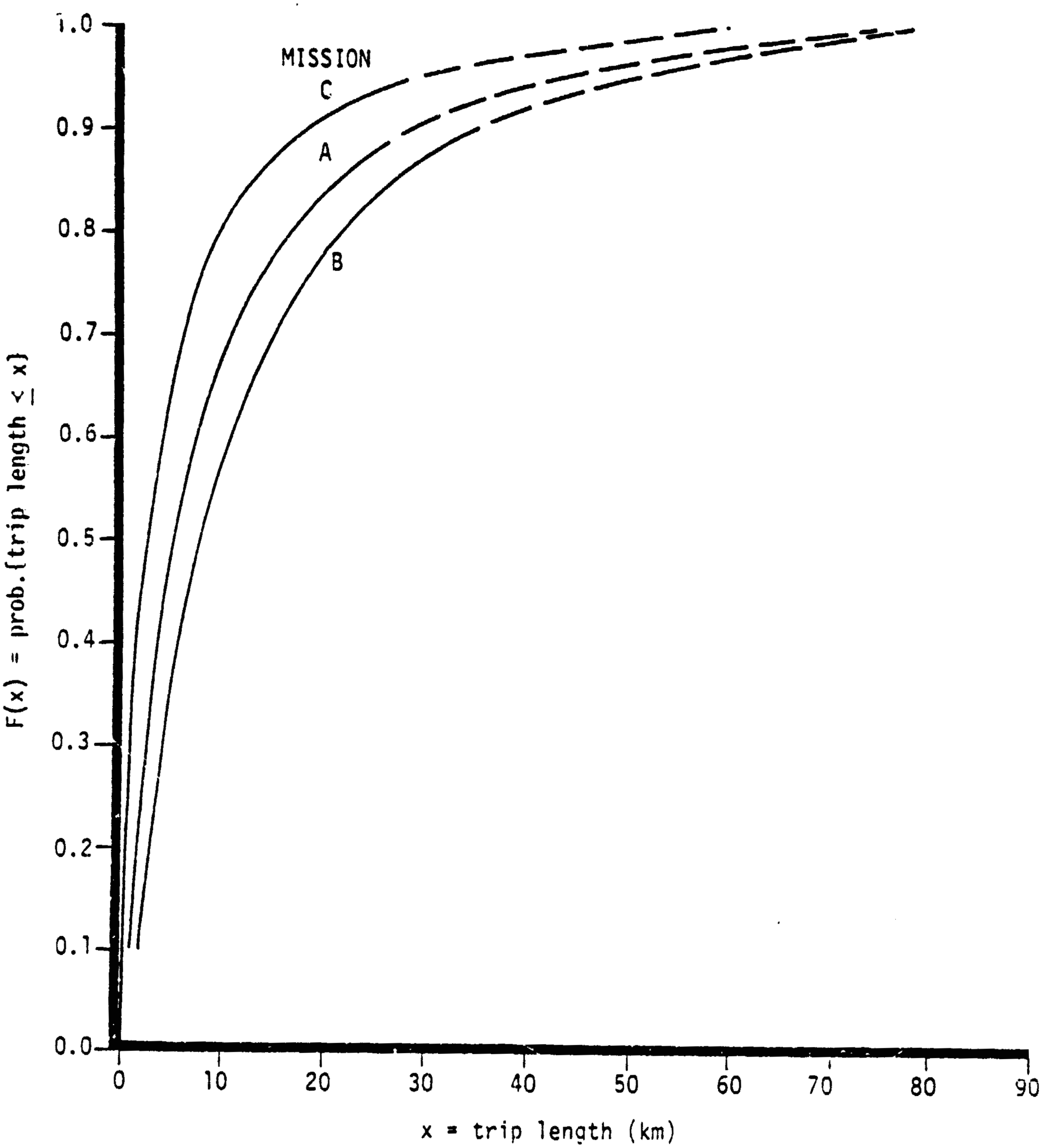

Figure 1. Trip Length Distribution
Functions by Mission

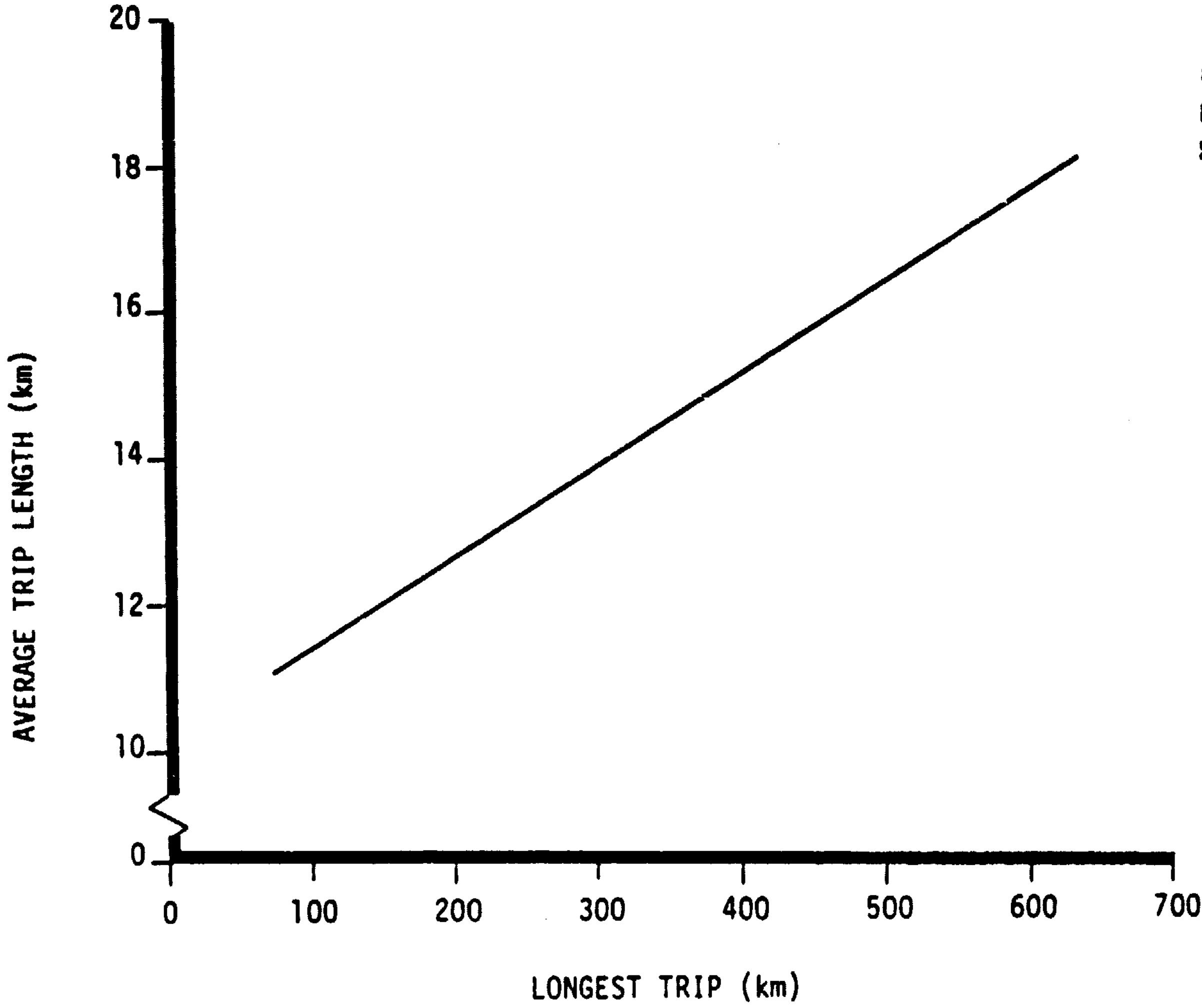

Figure 2. Average Trip Length as a Function of the "Longest" Trip - Mission AA, All Travel

Table 3. Distribution of Trip Length by Trip Purpose

Trip Length Interval (x) (km)	Distribution Function Probability {trip length < x}				
	All Travel	Commute	Family	Civic	Soc. & Rec.
8.05	62.4	51.8	74.2	78.8	56.6
16.1	79.2	71.9	88.5	90.8	73.7
32.2	91.7	89.2	96.3	97.0	87.3
>32.2	8.3	10.8	3.7	3.0	12.7
Average trip length (km)	14.3	16.4	9.01	7.56	21.1
Fraction of all trips	1	.362	.310	.093	.224
Average number of trips per household per day	3.8	1.4	1.2	0.4	0.8

The average trip length and consequently the annual distance driven (one of the important sensitivity parameters to be studied) is quite dependent on how the tails of the distributions of Figure 1 are drawn. Since the last real data point is at 32 km, this leaves a lot to the imagination.

In Figure 2 we show how the average trip length varies with the "longest trip," which is taken to be at the 99.9th percentile. The observed average trip length of 14.3 m corresponds to a longest trip of 330 km. While this result is, of course, entirely reasonable, it could hardly have been deduced from the shape of the curves shown in Figure 1.

In level terrain a hybrid vehicle could undertake such long trips. Studies[8] indicate that regenerative breaking recharges the batteries sufficiently so that electric power will always be available to supplement the internal combustion engine for level terrain power demands. On long grades, however, the hybrid would exhaust its battery and ICE power alone will provide inadequate performance.

For this reason we have restricted Mission A to a maximum single trip length of 80 km, hence the name "Limited all Purpose Travel." However, for possible application, handy reference, and comparison, we also carried out calculations for Mission AA, "All Travel."

A summary of the missions used in the Sensitivity Analysis and some of their attributes is shown in Table 4.

Table 4. Missions Selected for Sensitivity Analysis

Mission	All Travel	Limited All Purpose Travel	Augmented Commuting	Family & Civic Business
Designation	AA	A	BB	C
Average trip length (km)	14.3	11.2	16.4	8.66
Fraction of trips	1	0.98	0.362	0.403

SECTION 2

COMPUTER PROGRAMS

The principal tool used in the sensitivity studies was a Monte Carlo trip making program similar to the one described in Reference 6. The program, originally written by the General Research Corporation of Goleta, California, was expanded and adapted to the new Minicars Vax 11/780 computer. Chief program modifications were the simulation of trips by individual missions, and the ability to assign variations in annual travel to trip length and trip frequency in arbitrary proportions. The mofified program is called TRAVEL.

Further program improvements, including a feature which permits the insertion of arbitrary, rather than Poisson, trip frequency distributions, are currently in progress and will be described in our revised Mission Analysis report.

A typical program output is shown in Figure 3. An interpretative key is provided below.

PRINTOUT	MEANING
Inputs:	
RANGE	Range the hybrid vehicle can travel using electric motor power only wherever possible (This variable equates directly to battery capacity)
NEW KM/YR	Assumed annual travel, this case km
OLD KM/YR	Assumed annual travel, base case (km)
Outputs:	
ANNUAL ELEC	Annual distance traveled (km) using electric motor power only - in the 5th land the 10th years
AVERAGE ELEC	Annual distance traveled (km) using electric motor power only - averaged over 5th land 10 years
ANNUAL GAS & AVE GAS	As above for km traveled under heat engine power

ANNUAL AVE	As above for total km traveled
TRPNO	Average number of trips per car per day
TRPLEN	Average trip length
DADIST	Average distance traveled per car per day
FRACFUEL	Fraction of all travel on electricity
TOTFUEL	Total fuel used (liters/yr)
LITERS SAVED	Liters saved per car per year
DOLLARS SAVED	Dollars saved per car per year

Missions:

Mission 1	Mission A - Limited all purpose travel
Mission 2	Mission BB - Augmented commuting
Mission 3	Mission C - Family and civic business
Mission 4	Mission AA - All travel

Other inputs are the trip length distributions for each mission which are entered in data statements in the program, and a set of input parameters inserted in a data file; the parameters are:

1. Number of years the simulation is to run.
2. Constant which apportions travel to trip length and number of trips.
3. Trip frequency, Mission AA.
4. Fuel consumption, FUDC cycle, diesel topped by electric, km per liter.
5. Fuel consumption, FUDC cycle, electric toped by diesel, km per liter.
6. Fuel consumption, FUDC cycle, reference vehicle.
7. Fraction of trips on each mission.

Figure 3 shows a program output for the base case. A nominal annual distance of 15,919 km, corresponding to 3.05 trips per day and an average trip length of 14.3 km (see the discussion in the previous section), was assumed. As the program output shows, the simulation produced average trip lengths of 14.18 and 14.45 km for Mission AA, all travel and also accumulated annual totals quite close to the nominal value. The program correctly calculates this trip frequency for Mission C, family and civic business as 1.23 trips per car per day, and matches the average trip length of 8.66 km quite well. We thus have confidence in the accuracy of simulation results.

A separate set of assumptions is used in analyzing Mission BB, augmented commuting. Here we base the annual total on 230 working days per year and use two trips per day per commuting car as opposed to 1.4 trips per day averaged over all cars. Proper allowance for these assumptions is of course made when national totals are considered.

The logical approach used in the computer program TRAVEL is essentially the following: first the ratio of the assumed annual distance which is to be studied to the baseline annual distance of 15,919 km is found. The trip length and trip frequency are now stretched by appropriate factors as will be explained further on in this report. Each day, the NTHV to be analyzed starts out with a freshly charged battery. The computer program first chooses the number of trips to be taken today. It then selects the first trip from the trip length distribution and tests whether it can be driven in the electric mode; it next selects the second trip and repeats the procedure until all of the trips for that day have been accomplished. Once the vehicle electric range has been reached, the remainder of the current trip and all subsequent trips are driven in the diesel mode. Results for each day are tallied and yearly averages and totals are found.

While this procedure generates a different driving pattern each day, the days are "satistically identical;" that is, trip length and frequencies are drawn from the same distribution each day. This is only a first approximation to reality. A more authentic simulation should at least differentiate between workdays and weekends and possibly also include seasonal and geographic variations. We are at present conducting an investigation of these matters as part of the computer program modifications already mentioned. Further study, however, is needed to determine

RANGE(KM)	NEW KM/YR	OLDKM/YR
36.0	15919.0	15919.0

MISSION NUMBER 1

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	8529.8	8337.0	3499.8	4117.6	12029.5	12454.7	2.99	11.42	34.12
10	7715.0	8281.4	3569.8	4110.0	11284.8	12391.4	2.99	11.36	33.95

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.3317	321.4	702.2	83.4

MISSION NUMBER 2

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	5841.3	5936.9	1492.3	1624.4	7333.5	7561.3	2.00	13.44	26.88
10	6048.3	6008.1	2085.5	1853.8	8133.8	7861.9	2.00	14.09	28.18

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.2358	158.1	491.3	58.4

MISSION NUMBER 3

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	3277.8	3488.4	570.5	454.9	3848.3	3943.4	1.23	8.79	10.80
10	3398.8	3432.1	466.0	401.4	3864.8	3833.5	1.23	8.54	10.50

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.1047	46.6	270.0	32.1

MISSION NUMBER 4

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	8490.5	8419.5	6677.3	7256.6	15167.8	15676.0	3.05	14.08	42.95
10	8896.0	8459.8	8972.0	7646.0	17868.0	16105.7	3.05	14.47	44.13

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.4747	557.5	772.8	91.8

Figure 3. Typical Program Output

whether these additions in model sophistications can really be justified in view of the quality of the available basic data.

In addition to the Monte Carlo simulation, TRAVEL, described above, the other large scale computer programs were used to provide the Sensitivity Analysis results. The CARSIM program available from previous studies but mofified to handle hybrid vehicles was used to obtain candidate system fuel consumptions, and the Life Cycle Cost Program (LCC) developed as part of the NTHV design-trade off studies was used to calculate candidate system costs. A third program MISSIM which links CARSIM and LCC was also written for the design trade-off studies. These programs are described in detail in the Minicars Design Trade-Off Studies Report[8].

SECTION 3

CANDIDATE SYSTEMS

3.1 SYSTEM DEFINITION

3.1.1 NTHV Systems

In the sensitivity studies a candidate system is defined by means of three basic parameters, battery capacity, electric motor peak power and heat engine peak power. The latter two are related by the fact that each vehicle design must meet the power requirements dictated by the FUDC and FHDC driving cycles. Thus a "design" can be defined by only two independent variables, battery capacity and motor peak power. Consideration of the more detailed features of car design such as type of power plant, transmission and battery or environmental controls are all treated in the design trade-off studies, Reference 8. A very brief summary of the general features of our design will however be presented here for the sake of convenient reference.

While the ultimate design selection and proposal for Phase II has not as yet been finalized, we may assume the basic vehicle to be a 1980 GM X body, two-door fastback. The curb weight of the NTHV will be in the 3500-3900 lb range. The X body structure will be used as built, with additional structure added to a) support batteries and other new items and b) reinforce the vehicle for its weight increase. It should not be necessary to rebuild the front end structure except to accomodate weight increases since the necessary engine/motor/transmission system will be able to fit in the X body engine compartment.

The basic X body suspension and steering systems, strengthened, where necessary, will be used. Wheel bearings, hubs, tire and wheel sizes will be selected to match the vehicle weight.

Three locations for storing the 8 to 16 lead-acid 6-volt batteries are receiveing close study; These are: split between the very front and very rear of the body; in the rear quarter panels above the rear wheels; and in the bottom of and behind the doors.

The heat engine will be a turbocharged VW Rabbit diesel; the electric motor, a compound or shunt DC motor powered by 9 to 16 lead-acid 6-volt batteries. A computer-controlled automatic transmission with a lock-up torque converter will be used. The automatic transmission will be the Turbo-Hydromatic 125 (GM X body) with a modified valve body for electronic control, a Chrysler Omni torque converter modified to take a lock-up clutch, and two clutches on the torque converter input to connect the transmission to the engine and/or the motor. The computer-controlled manual transmission, being developed for the High Technology RSV, will be considered as an alternative transmission.

The NTHV will use regenerative braking, but the initial design will not have a system that will proportion the hydraulic braking to compensate for the level of regenerative braking. The initial part of the brake pedal travel will apply regenerative braking and the remainder, the hydraulic brakes. During Phase II a more complex braking system which will allow the hydraulic brakes to compensate for the differences in regenerative braking will be considered.

The spectrum of system designs considered in the Sensitivity Analysis are summarized in Table 5.

Table 5. Candidate Systems for Sensitivity Analysis

Number of 6-volt lead-acid batteries	8	10	12	14	16
Voltage (Volts)	48	60	72	84	96
Capacity (kW-hrs)	8.4	10.5	12.6	14.7	16.8
Vehicle weight (kg)	1630	1713	1797	1880	1964
Electric Motor Power (kW)	IC Engine Power (kW)				
14	51.2	54.5	57.9	61.2	64.6
19	46.2	49.5	52.9	56.2	59.6
24	41.2	44.5	48.5	51.2	54.5
29	36.2	39.5	42.9	46.2	49.6
34	31.2	34.5	37.9	41.2	44.6

As already noted, there are only two independent variables in Table 5, battery capacity and motor peak power. Consider, for example, the 10 battery design; a 14 kW motor is chosen, the IC engine power is 54.5 kW, while if a 34 kW motor is selected the IC engine has 34.5 kW. In either case, the total power is 68.5 kW because that is the peak power required by the driving cycles for a car which weighs 1713 kg. Evidently, the power requirements are 0.04 kW per kg. Thus, the 16 battery candidate system with a 49.6 kg IC engine requires a total of 0.04 × 1964 or 78.5 kW. It follows that a 29 kW motor is required, which, in fact, is what Table 5 shows.

3.1.2 Reference Conventional ICE Vehicle

The reference ICE vehicle was selected to represent the vehicles that the near term hybrid vehicle could replace. It is a gasoline engine automobile with an inertia weight of 1360 kg and an EPA composite fuel economy of 12.1 km/ℓ. This choice is explained below.

We assume that the NTHV designed in this program will be able to perform well on all but very long automotive missions - short trips on electric power alone, and long trips on a combination of both electric and ICE power. On this basis, hybrid vehicles could replace all sizes and types of conventional vehicles.

However, the size requirement for the proposed NTHV puts a practical limitation on the vehicles which can be replaced. The proposed hybrid will be a five-passenger car, which, even with downsizing and weight reductions, would be too large and too heavy to have the efficiency required to replace small or subcompact cars. On the other hand, the NTHV would be too small to replace the largest vehicles. Therefore, the required size limits its potential to that of a replacement for compact and full sized vehicles. Since each of these sizes of vehicles is estimated tin the JPL Guidelines to make up 30 percent of the vehicle market in 1985, a replacement would have the potential of capturing up to 60 percent of the total vehicle market.

The inertia weight and composite fuel economy numbers are averages of the estimated weights and economies of 1985 compact and full-size cars. They give only a coarse description of the baseline vehicle, and do not contain enough information to allow a direct comparison of the acceleration and economy of this vehicle with the NTHV over specific missions. We therefore used the data developed by BURKE (Reference 9) to further define the baseline vehicle. This yields an automobile with the specifications and performance shown in Table 6.

Table 6. Reference ICE Vehicle Performance Specifications

Vehicle type	Mid-size, five-passenger
Inertia weight (kg)	1360
Length (cm)	470
Width (cm)	185
Height (cm)	137
Engine	Gasoline, 63-67 kW
Transmission	4-speed manual or 4-speed OD automatic with lockup torque converter
Acceleration	0 → 96.6 km/hr in 14 sec
Fuel economy (km/ℓ)	
Composite	12.1
City	10.8
Highway	14.3
SAE J227a(B)	7.1

3.2 FUEL AND ELECTRICITY CONSUMPTION

Candidate system ICE fuel consumptions and electric ranges over the FUDC cycle were calculated by means of the CARSIM program documented in detail in Reference 8. The results are shown in Figures 4 and 5. The analysis was based on an operational strategy in which the hybrid vehicle is run on batteries until the battery reserve limit, say 20 percent of full capacity, is reached. Thereafter, vehicle propulsion is provided mainly by the IC engine.

During the first phase, power requirements will at times exceed electric motor peak power. When this occurs, the IC engine will be used to supplement the electric motor. We call this mode of

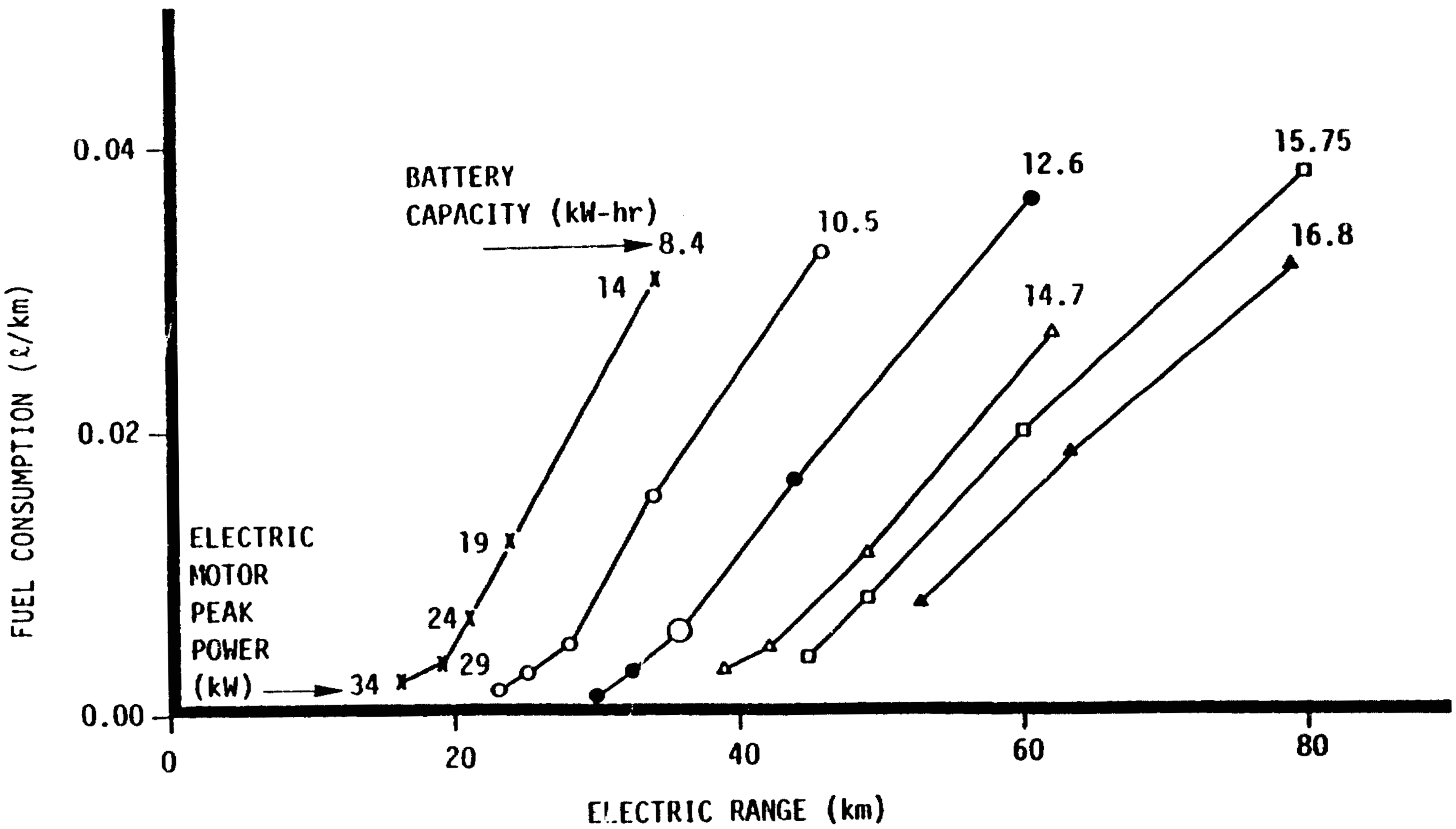

Figure 4. Fuel Consumption, Electric Motor Primary Drive, FUDC Cycle

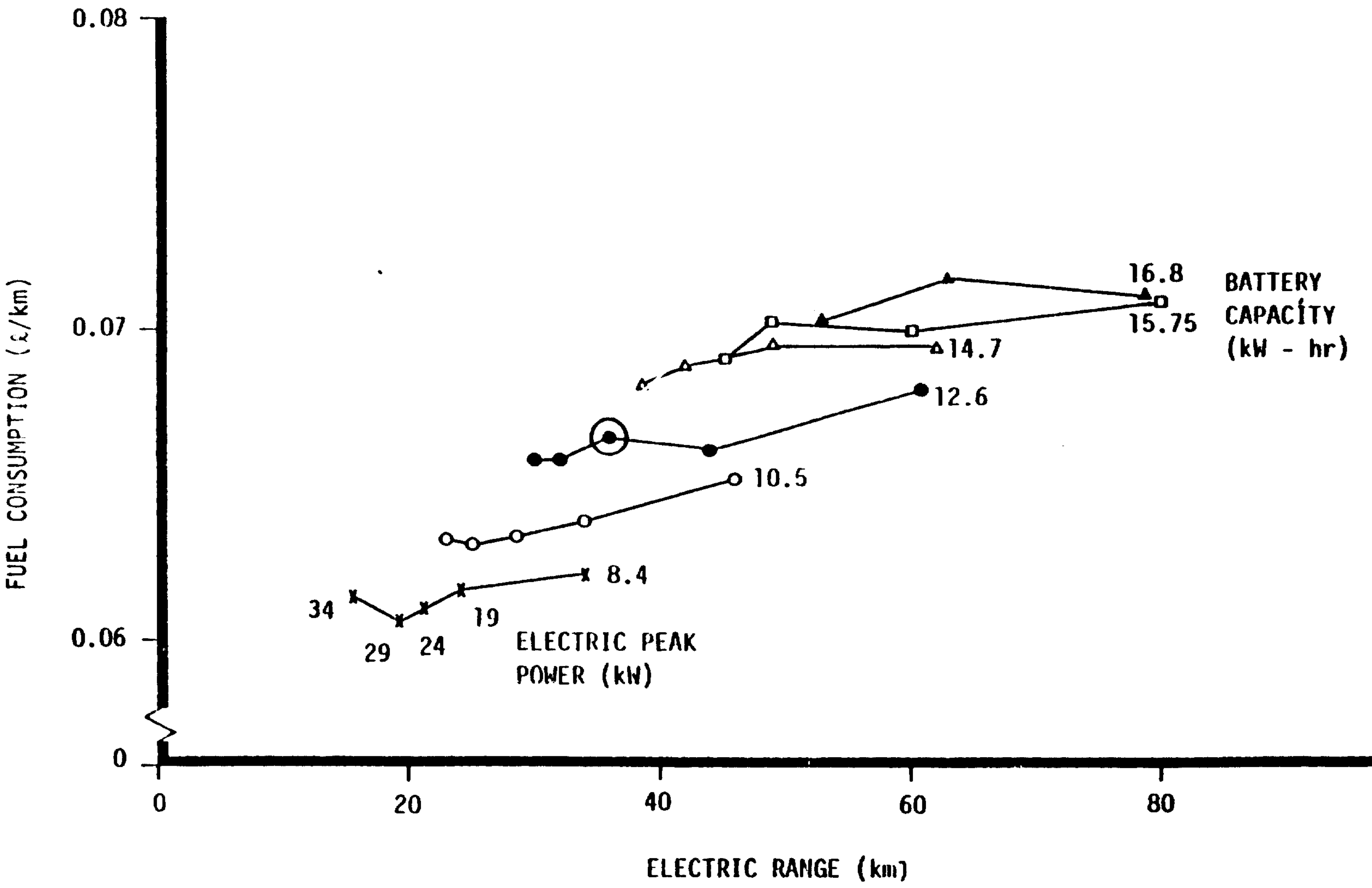

Figure 5. Fuel Consumption, Diesel Primary Drive, FUDC Cycle

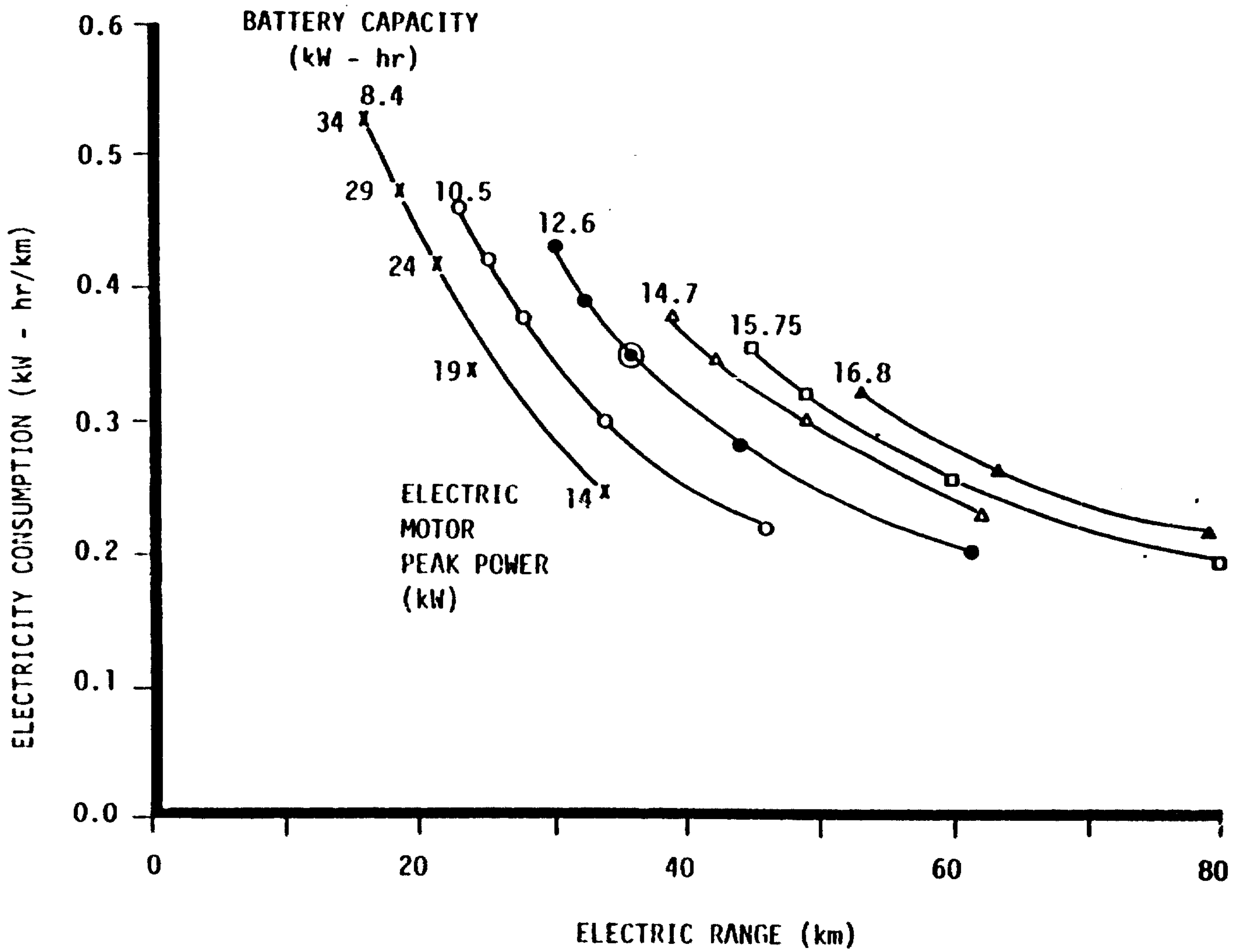

Figure 6. Electricity Consumption, Electric Motor Primary Drive, FUDC Cycle

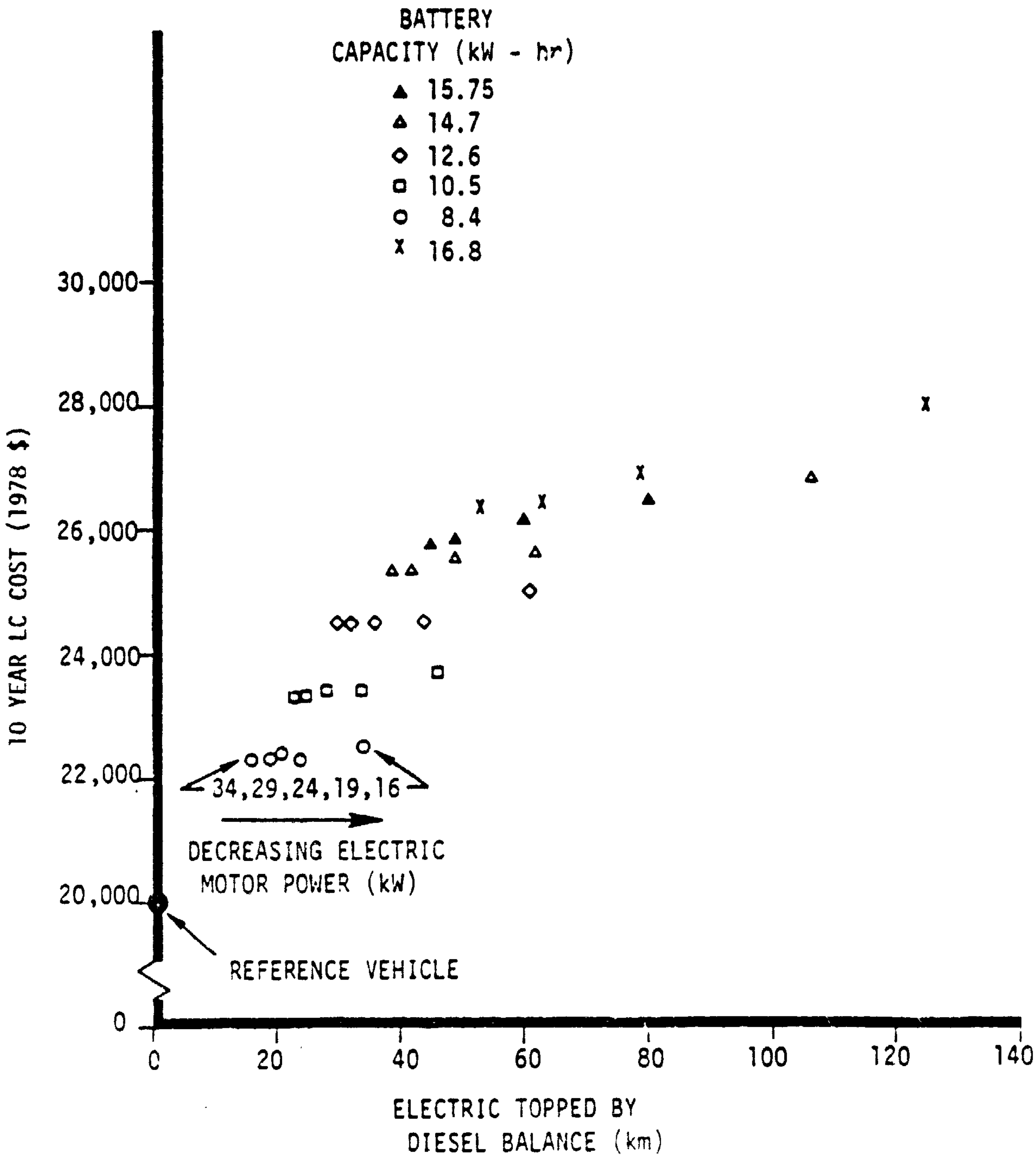

Figure 7. Candidate System Life Cycle Costs, Mission A

operation "electric-topped-by-diesel," and the distance a car can travel in this mode the "electric range." As one might expect, Figure 4 shows that the IC fuel consumption in the electric-topped-by-diesel-mode is quite low, one or two liters are used to travel 100 km. In familiar terms, the fuel economy is well over 100 miles per gallon.

Once the battery reserve limit is reached, the NTHV proceeds under ICE power. However, again on occasion, power requirements will exceed maximum ICE capability. In this case the electric motor is engaged to make up the deficiency. This mode of operation is called "diesel-topped-by-electric." Typical fuel consumptions in this mode are shown in Figure 5; for example, a candidate system of 12.6 kW-hr battery capacity using a 24 KW motor has a fuel consumption in the diesel-topped-by-electric mode of 0.0665 ℓ/km. As Table 5 shows, this system features a 48.5 kW diesel engine, 12 batteries for a total of 72 volts, and the overall vehicle weight is 1797 kg.

Figure 6 provides complementary information on the electricity consumption in the electric-topped-by-diesel mode. Values are of the order of one-third of a kilowatt hour per kilometer; for example, the design just discussed uses 0.35 kW-hrs/km. Since the battery capacity is 12.6 kW-hrs, it follows that if this candidate system were driven over the FUDC cycle only, as opposed to an appropriate mix of all three driving cycles, the resultant electric range would be $(12.6/0.35) \times 0.8 = 28.8$ km.

Figure 7 presents ten-year life cycle costs of candidate system as developed in the design trade-off studies.[8] The figure shows that, in the region of interest (electric ranges between 20 and 60 km) cost is essentially a linear function of electric range, and nearly independent of electric motor peak power. Thus, cost can be approximated as a linear function of a single variable, battery capacity.

As can be seen from the figure, none of the candidate systems can compete with the reference vehicle on a cost basis given the costs of fuel, assumed in Figure 7, namely those listed on Page 6 in the JPL Guidelines, Reference 10. The effect of increased fuel prices on these relationships will be studied in the next part of this report when we consider sensitivity to fuel prices.

PART II

RESULTS AND INTERPRETATIONS

SECTION 1

COMPARISON OF ANALYTICAL METHODS

Convenient measures for comparing the various approaches to tripmaking and fuel consumption calculations are 1) the fraction of all travel carried out in the diesel-topped-by-electric mode, hich we will abbreviate to "diesel mode," and 2) the fuel used per car per year. Ideally both of these measures would be zero.

In Figure 8, we compare the fuel used per car per year on Mission A as obtained by means of the TRAVEL AND MISSIM programs. In view of the differences in the assumptions used and in the analytical methods, the agreement of results is quite satisfactory.

In Figure 9 the fraction of all driving in the diesel mode - which is essentially equivalent to the fraction of fuel used - was calculated from the daily travel distribution data given in Reference 2, by means of the relationship derived on page 18, Reference 1. Results computed by the Monte Carlo simulation TRAVEL are also shown for comparison. The agreement here is remarkable.

In summary, the various analytical methods developed by Minicars for tripmaking and fuel consumption studies give entirely consistent and credible results.

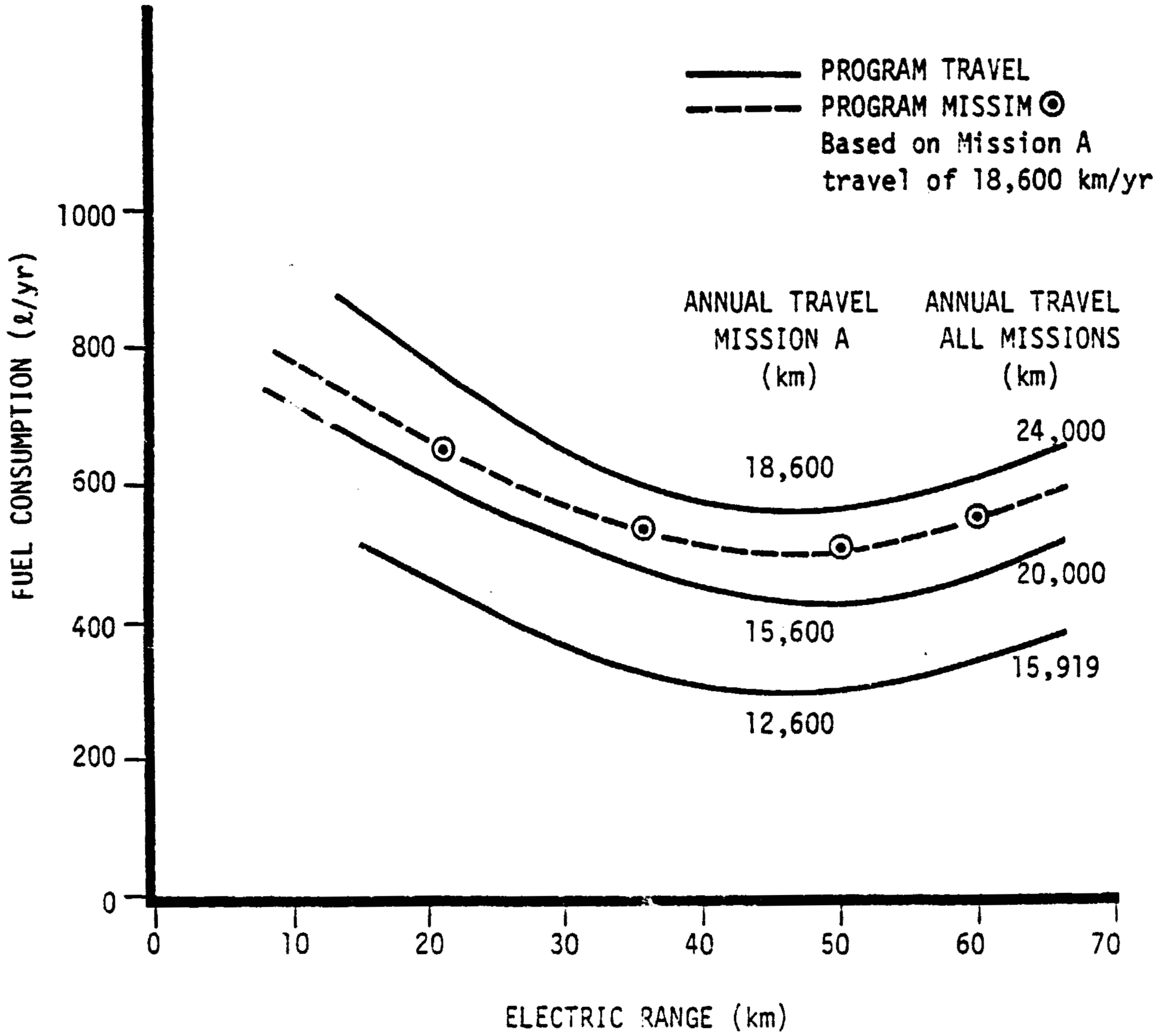

Figure 8. Fuel Used per Car per Year - Mission A, All Purpose Travel

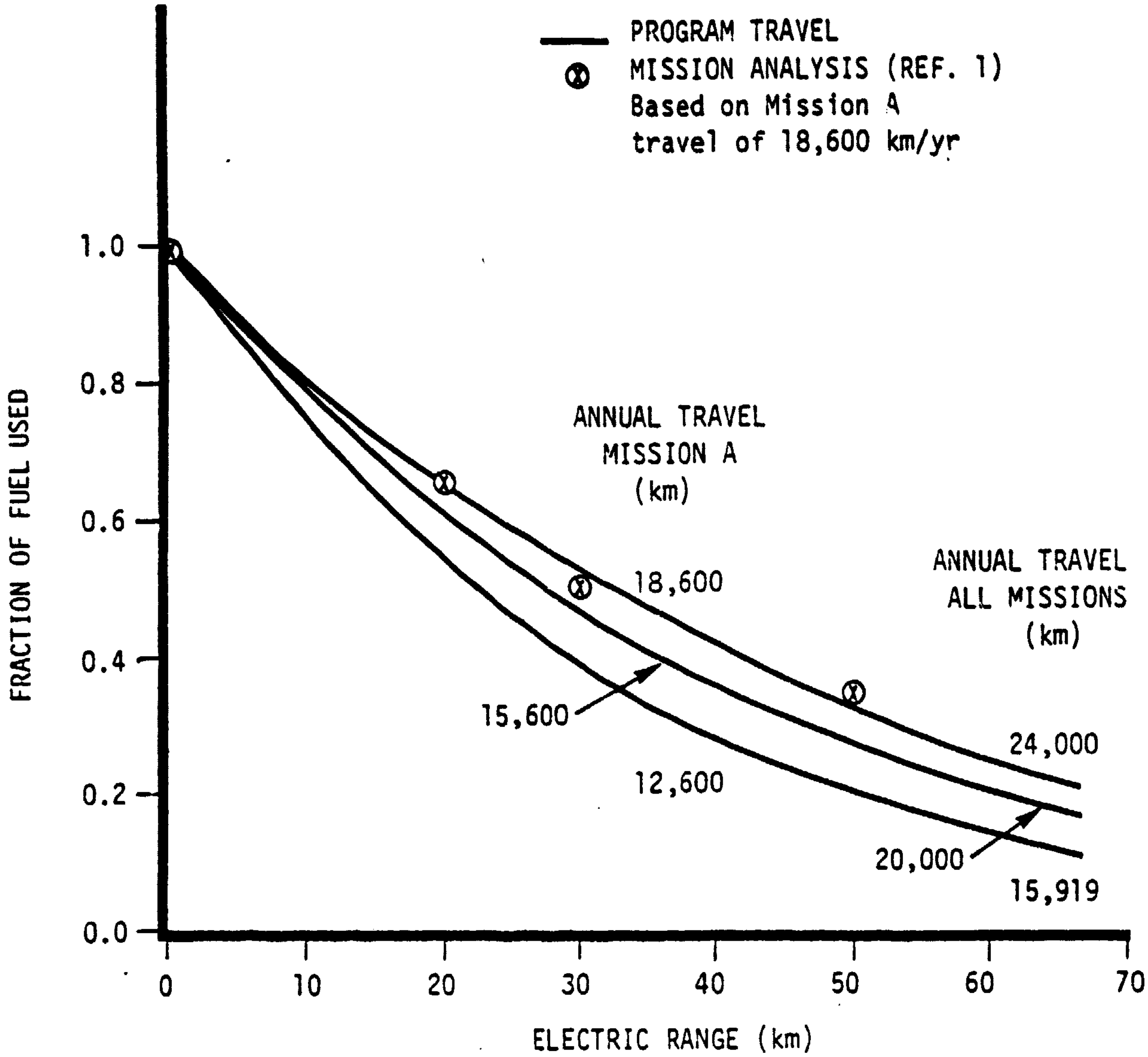

Figure 9. Fraction of Fuel Used per Car per Year - Mission A, Limited All Purpose Travel

SECTION 2

SENSITIVITY TO METHODS OF APPORTIONING ANNUAL TRAVEL

As discussed earlier, the baseline (1969) case consists of an average trip length of 14.3 km, an average daily trip frequency of 3.05, and an annual distance per car for all travel of 15,919 km.

Now if the annual distance is to be increased to some other value, say 20,000 km, how is this increase to be apportioned? In other words, is the greater annual travel to be ascribed to the fact that more trips are taken, or that longer trips are taken, or to both? Since no data appears to exist to resolve this question, General Research Corporation in the original version of program TRAVEL assigned increased annual travel to trip frequency and length in equal amounts. That is,

let

$$C = S_1/S_o$$

where,

S_o = annual travel baseline case

S_1 = annual travel new case

Then,

$$\lambda_1 = \sqrt{C}\,\lambda_o$$

$$L_1 = \sqrt{C}\,L_o$$

where,

λ = trip frequency, trips per day

L = trip length (km)

And

$$S_1 = 365\ \lambda_1 L_1 = 365\ C\ \lambda_o L_o = CS_o$$

In order to investigate the sensivity of results to the method of apportioning changes in annual travel we modified this procedure by the following elementary scheme.

Let

$$\frac{1}{C} \leq N \leq C$$

$$C_1 = \sqrt{\frac{C}{N}}$$

$$C_2 = NC_1$$

Then

$$C_1\ C_2 = C$$

And let,

$$L_1 = C_1 L_o$$

$$\lambda_1 = C_2 \lambda_o$$

Then

$$S_1 = 365\ L_1 \lambda_1 = 365\ C_1 C_2 L_o \lambda_o = CS_o, \text{ as before.}$$

If $N = 1$, $C_1 = C_2 = \sqrt{C}$; this corresponds to the previous case.

If $N = C$, $C_1 = 1$, $C_2 = C$, that is all of the increase is due to more trips.

If $N = \frac{1}{C}$, $C_1 = C$, $C_2 = 1$, now all of the increase is due to longer trips.

Intermediate cases arise when other values of N are chosen.

The extent to which the value of the apportionment constant, N, influences simulation results is investigated in Figures 10, 11 and 12. In each of these computer runs a hybrid vehicle of 36 km electric range was driven (on the computer) over an annual distance of 24,000 km. Thus $C = S_1/S_o = 24{,}000/15919 \simeq 1.5$. The values of N = 1, N = 1.5, and N = 0.67 were selected in Figures 10, 11, and 12, respectively; all else remained the same. We observe by comparing the three computer outputs that the choice of N affects the results in a relatively minor way. Accordingly, we chose N = 1 in all subsequent runs. Of course, the computer program retains the capability of apportioning increases in annual travel in any arbitrary ratio to trip length and frequency should data become available which suggests how this split should be selected.

RANGE(KM)	NEW KM/YR	OLDKM/YR
36.0	24000.0	15919.0

MISSION NUMBER 1

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	9827.0	9662.3	8225.5	9063.8	18052.5	18726.1	3.00	17.08	51.30
10	9152.8	9615.0	7895.1	9027.2	17047.9	18642.1	3.00	17.00	51.07

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.4842	656.1	883.8	105.0

MISSION NUMBER 2

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	6694.4	6809.4	3948.1	3949.3	10642.5	10758.8	2.00	20.39	40.78
10	6779.4	6835.7	4579.2	4298.1	11358.6	11133.7	2.00	21.20	42.41

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.3860	325.5	594.2	70.6

MISSION NUMBER 3

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	4280.0	4580.6	1352.0	1362.4	5632.0	5943.0	1.24	13.18	16.28
10	4392.5	4536.6	1371.1	1249.5	5763.6	5786.2	1.24	12.83	15.55

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.2160	109.4	368.5	43.8

MISSION NUMBER 4

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	9864.5	9754.3	12385.6	13698.9	22250.1	23453.1	3.07	20.96	64.26
10	10069.2	9789.4	16402.0	14355.1	26471.3	24144.5	3.07	21.58	66.15

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.5946	1011.4	982.9	116.8

Figure 10. Effect of Increased Travel Distance (N=1.0)

RANGE(KM)	NEW KM/YR	OLDKM/YR
36.0	24000.0	15919.0

MISSION NUMBER 1

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	10243.7	10362.6	8296.8	8123.5	18540.5	18486.1	4.49	11.27	50.65
10	10373.6	10336.9	8737.1	8003.1	19110.7	18340.0	4.49	11.18	50.25

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.4364	592.2	922.7	109.6

MISSION NUMBER 2

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	6968.4	6891.0	4241.0	4533.1	11209.4	11424.1	2.00	21.83	43.67
10	6844.3	6878.3	4168.3	4389.0	11012.6	11267.3	2.00	21.49	42.99

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.3895	331.8	598.9	71.2

MISSION NUMBER 3

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	4102.9	4241.3	791.8	687.3	4894.7	4928.5	1.85	7.31	13.50
10	4451.5	4276.2	831.8	692.8	5283.4	4969.1	1.85	7.37	13.61

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.1394	70.9	339.6	40.3

MISSION NUMBER 4

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	10528.3	10556.8	13781.3	13704.9	24309.6	24261.7	4.59	14.49	66.47
10	10288.0	10562.4	13447.2	13778.5	23735.2	24340.9	4.59	14.54	66.69

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.5661	977.5	1033.0	122.7

Figure 11. Effect of Increased Travel Distance (N=1.5)

RANGE(KM)	NEW KM/YR	OLDKM/YR
36.0	24000.0	15919.0

MISSION NUMBER 1

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	9934.2	9936.9	7417.2	8319.3	17351.5	18256.1	3.67	13.63	50.02
10	9772.1	9902.6	7397.9	8274.6	17170.1	18177.2	3.67	13.57	49.80

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.4552	607.7	893.7	106.2

MISSION NUMBER 2

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	6780.6	6824.1	3886.0	4546.2	10666.6	11370.3	2.00	21.72	43.44
10	6944.2	6854.8	5137.0	4609.2	12081.2	11464.0	2.00	21.92	43.84

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.4021	346.3	600.7	71.4

MISSION NUMBER 3

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	4564.7	4369.9	930.6	942.8	5495.3	5312.7	1.51	9.64	14.56
10	4219.2	4373.9	730.9	941.4	4950.1	5315.3	1.51	9.65	14.56

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.1771	88.0	351.1	41.7

MISSION NUMBER 4

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	10326.3	10171.6	12917.3	13307.8	23243.6	23479.4	3.74	17.18	64.33
10	10152.6	10094.3	12722.1	14136.6	22874.7	24230.8	3.74	17.73	66.39

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.5834	998.6	1002.8	119.1

Figure 12. Effect of Increased Travel Distance (N=0.67)

SECTION 3

SENSITIVITY TO ANNUAL TRAVEL

In Reference 10, JPL specified the 1985 value for the distance traveled per car per year to be 19,073 km. In Reference 11, we were asked to study the effect of varying annual travel between the limits of 17,739 and 20,408 kilometers (±0.7 percent).

In order to cover this and a larger range of variations, the TRAVEL program was used; computer runs were made using the candidate systems shown in Table 7. Each system was simulated over four missions for annual travel distances of 15,919, 20,000 and 24,000 kilometers. The results are presented in the computer print-outs, Figures A-1 thru A-9 in Appendix A.

Table 7. Characteristics of Candidate Systems

	Candidate Systems		
	Number 1	Number 2	Number 3
Electric Range (km)	21	36	49
Assumed Motor Peak Power (kW)	24	24	24
Battery Capacity (kW-hrs)	8.4	12.6	14.7
Number of Batteries	8	12	14

In Figures 13, 14 and 15, the fuel used per NTHV per year on Missions AA, BB and C as given in Figures A-1 through A-9 is crossplotted with range for the purpose of further discussion. Mission BB results have not as yet been fully evaluated. The design and costing of a two passenger car which does not meet the JPL constraints will be required to complete this task. This will be accomplished as part of the revision of the Mission Analysis and Performance Specification Studies Report.

The basic conclusion to be drawn from Figures 13, 14, and 15 is that the choice of the optimal car is influenced very little by total annual travel. It appears that an electric range of 45-50 km, corresponding to 14 six-volt batteries, is optimal for the NTHV in the sense that it minimizes fuel consumption per car, and that this result is true over very wide variations in the yearly distance driven. It is further clear that there is no point in considering designs with an electric range greater than

50 km since beyond this point both cost and fuel consumption increase. It is, however, necessary to continue to evaluate cars of lesser electric range. Their lower cost may increase their acceptability to the point where the greater number of cars sold more than makes up for the smaller fuel savings per car.

As one might expect, the fuel used per car is essentially proportional to the annual range. To a first order, this relation for a car of 50 km electric range may be expressed by

$$f = 740 + .06\ (d-20{,}000)$$

where f is the fuel consumption in liters per year and d is the annual distance traveled. For example, an annual distance of 17,739 km corresponds to 604 liters per car per year. Similar sensitivity relations can easily be constructed from Figures 13 through 15 for other electric ranges and missions. Reference 8 shows fuel consumption for an NTHV of 50 km electric range is .07 km in the diesel mode. Thus, the above equation indicates that 6/7 or about 85 percent,of increased annual driving is done in the diesel-topped-by-electric mode. Another way of saying the same thing is that, to a good approximation, the variations in fuel used per hybrid vehicle over the range of the annual driving distance of interest to JPL can be calculated by assuming that any change in the distance driven is driven in the diesel-topped-by-electric mode only; that is, essentially with the internal combustion engine.

In summary:

1. Variations of the order of ± 10 percent in annual distance driven have very little effect on the choice of the optimal car.

2. The changes in fuel consumption per car which results from ± 10 per cent variation in annual driving distance can be well approximated by assuming that the distances are driven in the diesel mode only.

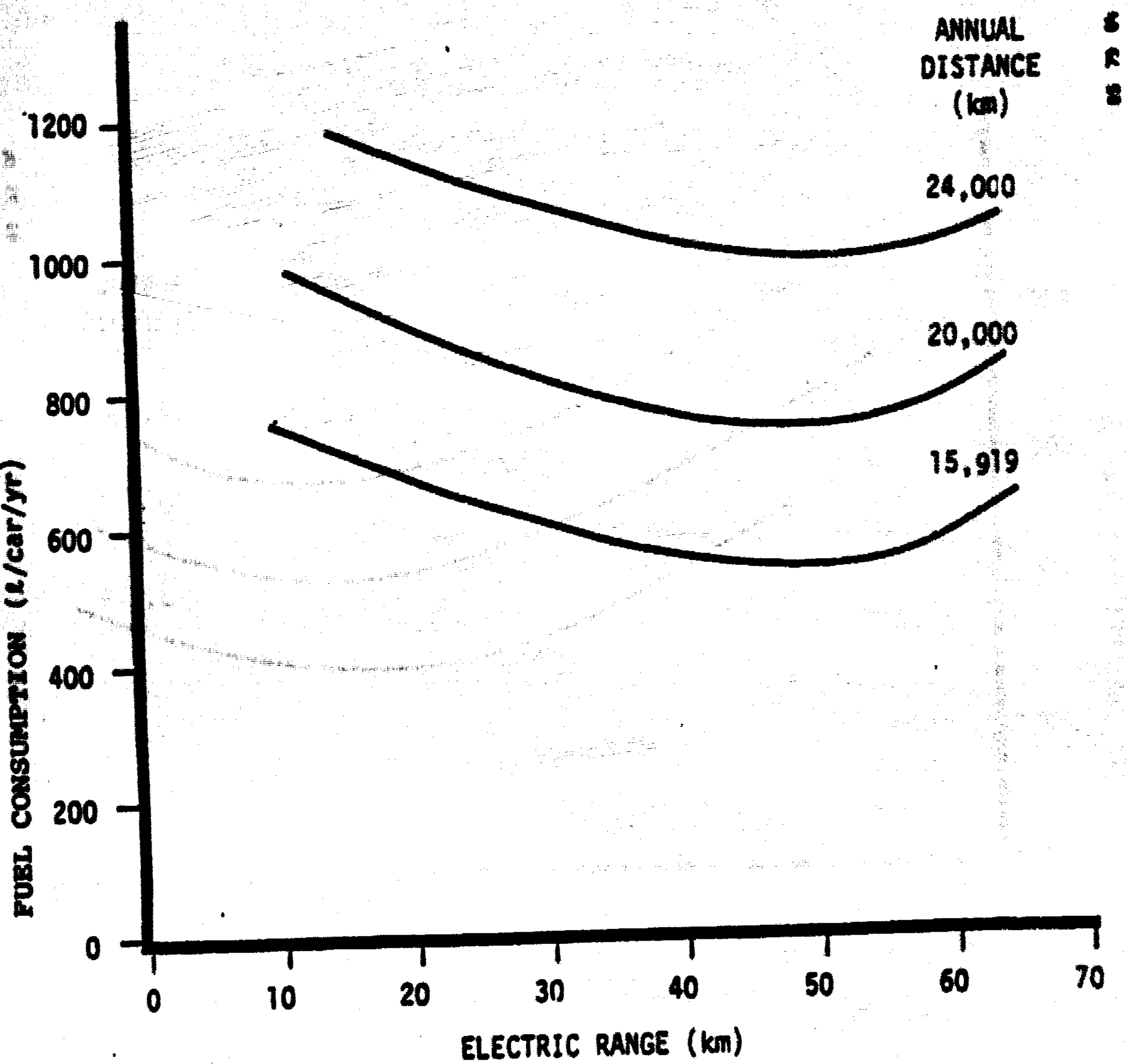

Figure 13. NTHV Petroleum Consumption as a Function of Annual Distance and Electric Range - Mission AA, All Travel (24 kW Motor)

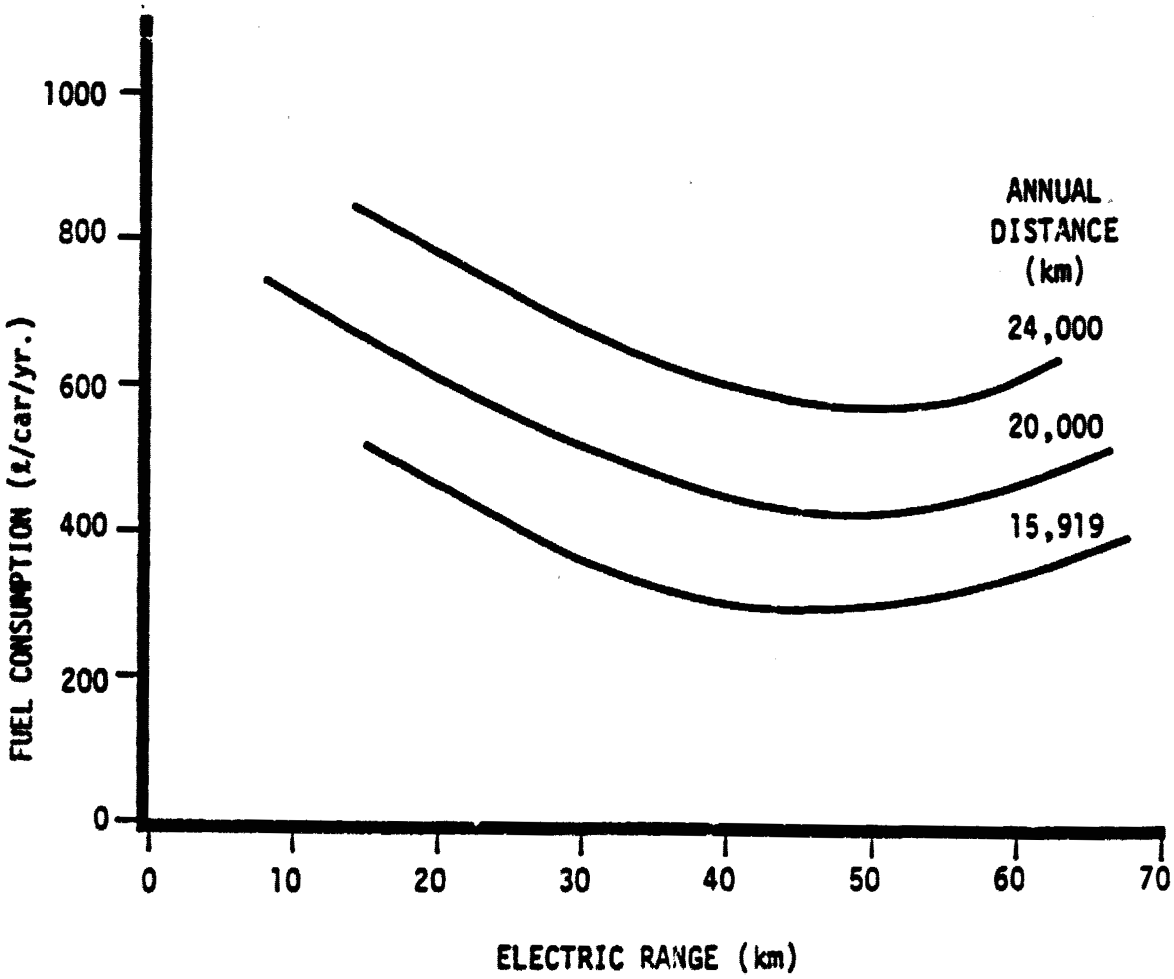

Figure 14. NTHV Petroleum Consumption as a Function of Annual Distance and Electric Range - Mission A, All Travel (24 kW Motor)

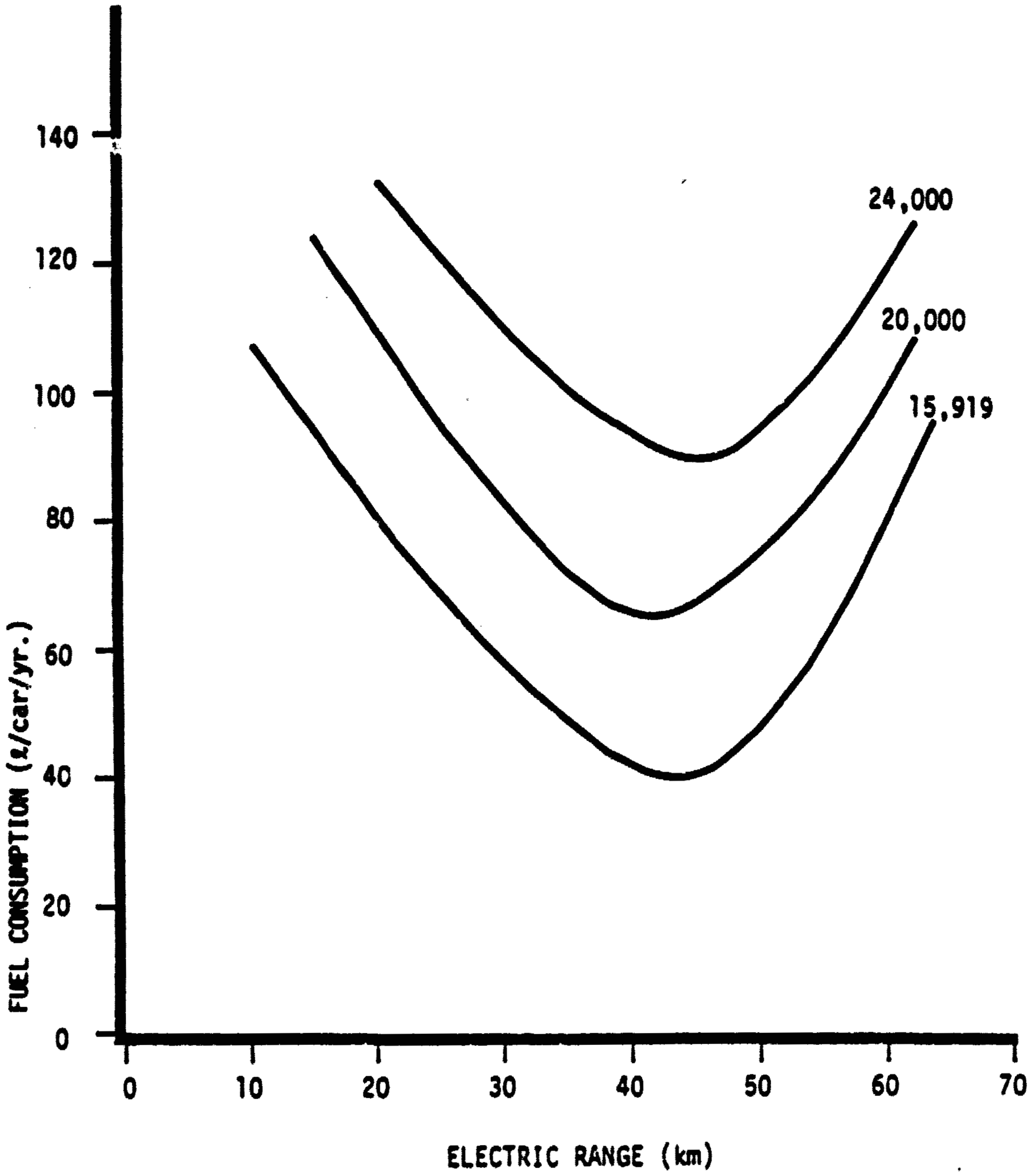

Figure 15. NTHV Petroleum Consumption as a Function of Annual Distance and Electric Range - Mission C, All Travel (24 kW Motor)

SECTION 4

SENSITIVITY TO FUEL PRICES

The 10 year life cycle cost for each of the candidate systems and for the reference vehicle were obtained by means of the Minicars (LCC) Life Cycle Cost computer program described at length in Reference 8. Fuel economics for each candidate were derived by exercising the CARSIM program (see Reference 8) over the FUDC driving cycle. The results are presented in Table 8.

Table 8. Candidate System Costs and FUDC Fuel Economy

	Systems			
	Ref. Vehicle	1	2	3
Electric Range (km)	0	21	36	49
10-year LC Cost Without Fuel (1978 $)				
Missions A and AA	15,549	20,883	23,217	24,313
Mission C	10,047	10,910	12,110	12,736
Fuel Consumption (ℓ/km)	.093			
Diesel-Topped-by-Electric		.061	.0665	.070
Electric-Topped-by-Diesel		.0065	.0060	.020

The data of Table 8 were combined with the annual fuel consumptions calculated by the TRAVEL program in order to arrive at the total 10 year life cycle cost as a function of fuel prices. These results are shown in Figures 16 through 18.

Figure 16 shows that a hybrid vehicle with a 21 km electric range becomes price competitive on Mission A with the reference vehicle when petroleum reaches a price of approximately 65 cents per liter or $2.46 per gallon (1978 dollars). Vehicles of greater electric range, and cost, cannot compete pricewise until fuel reaches 80 or 90 cents per liter. Note that the 21 km range hybrid remains the least expensive over the full range of fuel prices considered. That is, the curves for hybrids of different battery capacity do not cross each other in the range of fuel prices of interest to this study.

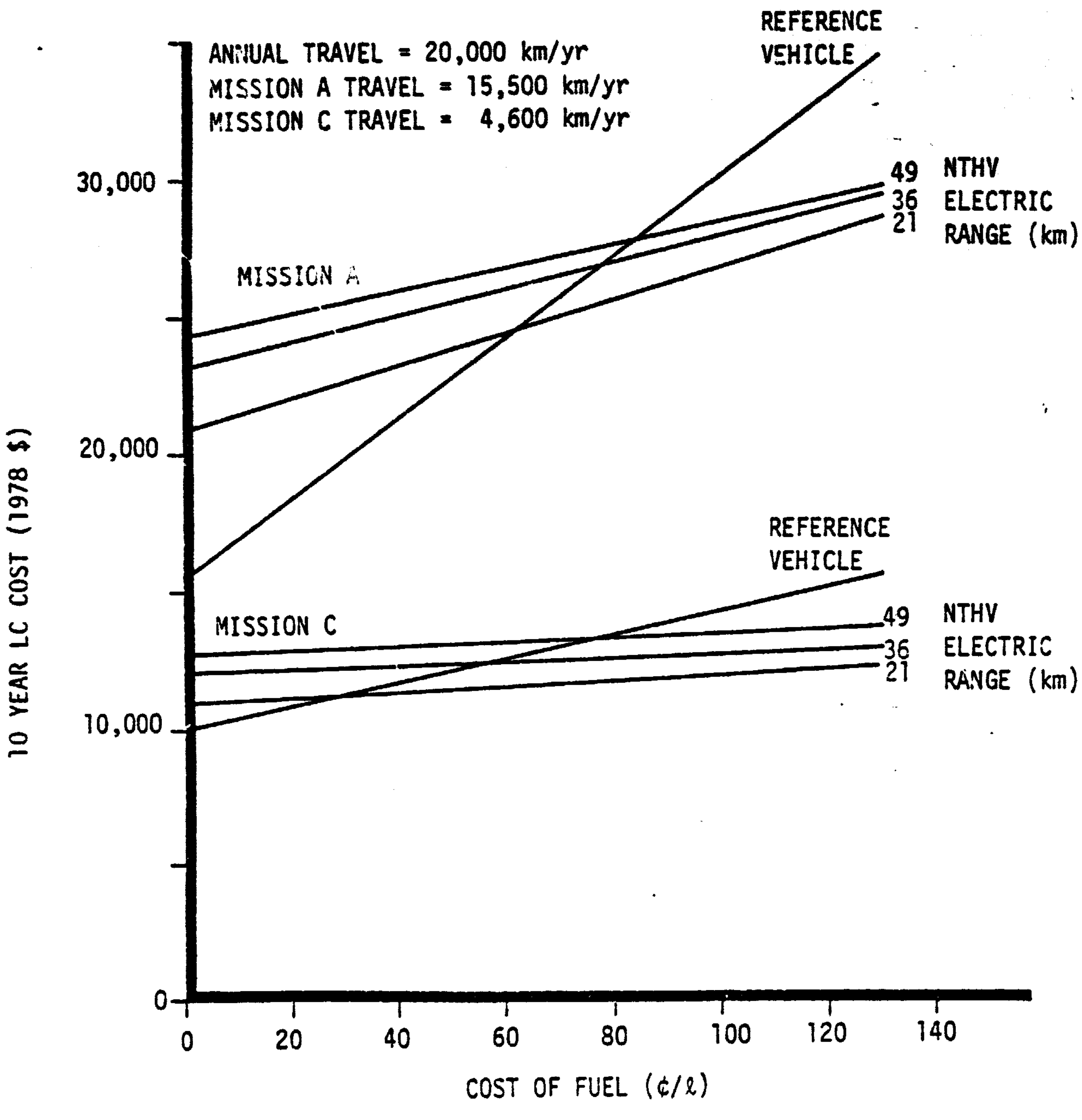

Figure 16. Effect of Fuel Prices on Candidate System Life Cycle Cost

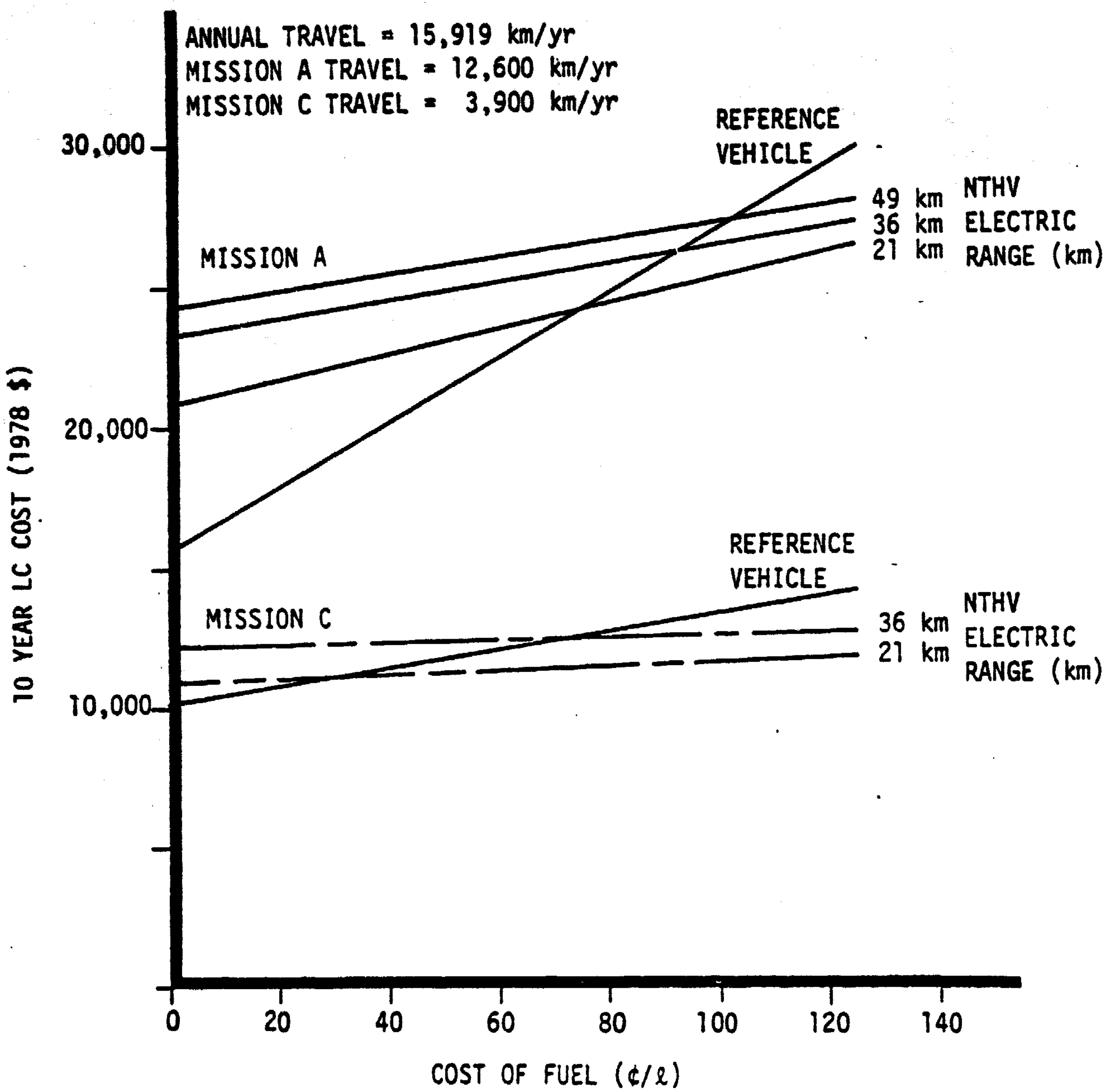

Figure 17. Effect of Fuel Prices on Candidate System Life Cycle Cost

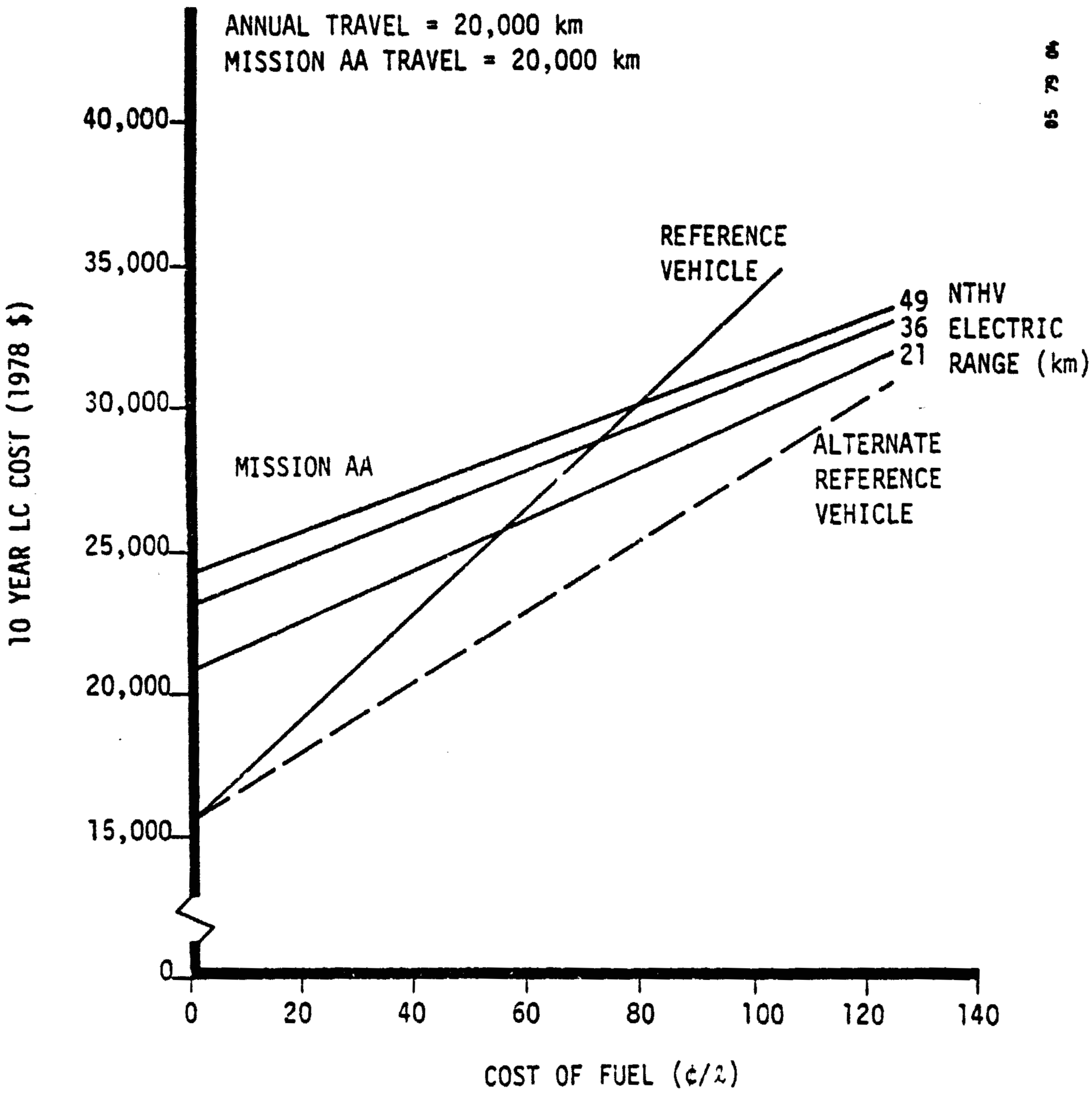

Figure 18. Effect of Fuel Prices on Candidate System Life Cycle Cost

JPL, in Reference 11, asked us to consider fuel prices ranging from 62.2 to 124.2 cents per gallon (16.5 to 32 cents per liter). As Figure 16 shows, general purpose hybrid vehicles as typified by Mission A are not price competitive in this region. However, Figure 16 also indicates that for more limited travel, (Mission C, Family and Civic Business), a hybrid vehicle with a 21 km electric range (8 batteries) can compete pricewise once fuel reaches 30 cents per liter or $1.13 per gallon. This price is nearly a reality in the u.S. today and is far below the two to three dollars per gallon price which is not uncommon elsewhere in the world.

A design based on Mission C may present an opportunity for the early introduction of hybrid vehicles. At a later date, the battery capacity could be increased, say from 8 to 14 batteries, and an all-purpose car could thus be developed. A further possibility is the export of Mission C designed cars to other countries (ideally to places in which driving distances are limited and oil is expensive). The U.S. balance of payments problem could thus be attacked not only be restricting the import of oil but also by fostering the export of cars.

The break even price of fuel for an all-purpose car is somewhat dependent on the annual distance driven. This effect may be studied by comparing Figure 16 with Figures 17 and 18 as is done in Table 9 below.

Table 9. Cross Sensitivities

Annual distance driven (km)	20,000	15,500	12,600
Break Even Price of Fuel (¢/ℓ)	55	65	75

The evaluation of hybrid vehicles is very sensitive to what one wishes to assume about the reference vehicle. We have selected the reference vehicle (see Section 3.12) to be typical of the cars the hybrid vehicle might replace assuming the car population in 1985 and thereafter to be as characterized by JPL in Reference 10. In Figure 18, we show an alternative and perfectly plausible reference vehicle; one which has the same fuel consumption as a typical hybrid vehicle in the diesel-topped-by-electric mode. The best (21 km range) hybrid now becomes price competitive at $1.63 per liter or $6.16 per gallon — in other words never, as long as anything at all like the present "free market" for motor fuel exists.

SECTION 5

SENSITIVITY TO ELECTRICITY PRICES

The effect of variations in electricity prices ranging from 10 percent below to 30 percent above the nominal value of 4.2 cents per kW-hr were considered as required by Reference 11.

A brief calculation will show that changes in electricity prices in this range are of little consequence.

The baseline NTHV design of 36 km electric range employs a 12.6 kW-hr battery. If this battery is drained to 80 percent of capacity, the energy available per day is 10.08 kW-hrs, say 10 kW-hrs per day or 3650 kW-hrs per year. At 4 cents per kW-hr, the yearly cost of electricity is $146. This number is of the same order as the cost of diesel fuel. As Table 8 shows, the ten year Life Cycle (LC) cost of the vehicle without diesel fuel is $23,217. Since the car uses 470 liters per year, the yearly cost at 25 cents per liter (95.5 cents per gallon specified as the nominal value in Reference 10) is $117. The total 10 year LC cost is thus $24,387. Wall plug electricity therefore represents 6 percent and diesel fuel represents 4.8 percent of the 10 year LC cost.

A 30 percent change in electricity (or diesel) prices thus equates to a two percent variation in total cost. Uncertainties in such characteristics as future trip making behavior or customer preferences far overshadow effects of this magnitude in determining either the choice of the optimal NTHV or the fuel savings that might be obtained by it.

The above "back of the envelope" calculation is borne out by the more detailed studies of Reference 8. Figure 19 presents the LCC program computer printout which breaks down the costs for the candidate system under discussion. As the figure shows, the 10 year diesel fuel cost is $1306 or $136/year, while wall plug electricity is $1372 or $132/year. The respective percentages are 5.31 and 5.38 and thus in excellent agreement with the calculation results presented above.

JPL NEAR TERM HYBRID PASSENGER VEHICLE DEVELOPMENT PROGRAM-PHASE I MARCH 1979

PRESENT VALUE OF TOTAL LIFE CYCLE COSTS — RUN NO. 2

(1978 $) — DATE MARCH 28 1979

DISCOUNT RATE= 2.00 % — SYSTEM PACKAGE NO. 1

	$/KM	1985	1986	1987	1988	1989	1990	1991	1992	1993	1994	1995	TOTAL $	% OF LCC
ACQUISITION COSTS														
PURCHASE PRICE	0.04329	2087.9	2047.0	2006.8	1967.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	8109.2	32.98
SALES TAXES	0.00223	417.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	417.6	1.70
INTEREST	0.01370	661.7	648.8	636.0	623.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0	2570.1	10.45
SALVAGE VALUE OF VEHICLE	0.00183	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	-342.6	-342.6	1.39
TOTAL ACQUISITION COST	0.05734	3167.2	2695.7	2642.9	2591.1	0.0	0.0	0.0	0.0	0.0	0.0	-342.6	10754.4	43.74
OPERATING COSTS														
ENERGY COSTS														
DIESEL FUEL	0.00696	198.4	167.9	150.3	138.0	128.7	120.3	111.4	103.2	96.7	91.1	0.0	1306.1	5.31
ELECTRICITY	0.00732	217.1	180.9	159.7	144.5	133.0	122.8	113.9	106.2	99.7	94.5	0.0	1372.2	5.58
TOTAL ENERGY COST	0.01428	415.5	348.8	310.0	282.5	261.7	243.1	225.3	209.4	196.4	185.6	0.0	2678.3	10.89
MAINTENANCE + REPAIR COSTS														
PROPULSION SYSTEM MAINT	0.00448	57.6	148.4	43.5	124.7	100.4	57.4	92.8	169.5	24.6	20.7	0.0	839.6	3.41
REPAIRS	0.00701	218.9	179.8	156.5	139.7	126.9	115.6	105.8	97.3	90.2	84.4	0.0	1315.1	5.35
TOTAL MAINT + REPAIR COST	0.01149	276.6	328.2	200.0	264.4	227.2	172.9	198.6	266.8	114.8	105.1	0.0	2154.7	8.76
BATTERY REPLACEMENT COSTS														
BATTERY PURCHASE PRICE	0.01692	0.0	0.0	136.8	536.5	482.1	300.8	505.5	289.1	445.4	476.4	0.0	3172.5	12.90
SALES TAXES	0.00089	0.0	0.0	59.3	0.0	0.0	55.9	0.0	0.0	52.6	0.0	0.0	167.8	0.68
INTEREST	0.00328	0.0	0.0	26.6	104.2	93.6	58.4	98.2	56.1	86.5	92.5	0.0	616.0	2.51
SALVAGE VALUE	0.00179	0.0	0.0	-118.6	0.0	0.0	-111.7	0.0	0.0	-105.3	0.0	0.0	-335.6	1.36
TOTAL BATTERY REPL COST	0.01931	0.0	0.0	104.1	640.6	575.7	303.3	603.7	345.2	479.2	568.9	0.0	3620.8	14.73
OTHER OPERATING COSTS														
TIRE REPL EVERY 50000 KM	0.00209	65.4	53.7	46.7	41.7	37.9	34.5	31.6	29.1	26.9	25.2	0.0	392.8	1.60
INSURANCE	0.00902	281.5	231.2	201.2	179.6	163.1	148.6	136.0	125.3	116.0	109.5	0.0	1690.8	6.88
ANNUAL REG. AND LICENSE	0.00164	51.2	42.0	36.6	32.7	29.7	27.0	24.7	22.8	21.1	19.7	0.0	307.4	1.25
ACCESSORIES	0.00173	54.0	44.4	38.6	34.5	31.3	28.5	26.1	24.0	22.3	20.8	0.0	324.5	1.32
GARAGING,PARKING,TOLLS	0.01384	432.2	355.0	309.0	275.8	250.4	228.1	208.8	192.1	178.1	166.6	0.0	2596.0	10.56
TOTAL OTHER OPERATING COST	0.02832	884.2	726.4	632.1	564.3	512.4	466.7	427.2	393.1	364.3	340.9	0.0	5311.6	21.60
TOTAL OPERATING COST	0.07390	1576.3	1403.4	1246.2	1751.8	1577.0	1186.1	1454.8	1214.6	1154.8	1200.4	0.0	13765.4	55.99
R+D COST AMORT (500000 VEH)	0.00036	67.7	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	67.7	0.28
PRESENT VALUE OF TOTAL LCC	0.13110	4811.2	4099.2	3889.1	4342.9	1577.0	1186.1	1454.8	1214.6	1154.8	1200.4	-342.6	24587.4	100.00

BENEFIT PER NTHV = 4370.51 - 1306.10 = 3064.41

COST OF ACCRUING THIS BENEFIT PER NTHV = (24587.43 - 1306.10) - (19927.82 - 4370.51) = 7724.02

NET BENEFIT PER NTHV = -4659.60

Figure 19. Candidate System 10 year LC Cost

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SECTION 6

INTERPRETATION OF RESULTS ON A NATIONAL SCALE--SENSITIVITY TO NUMBER OF PASSENGER CARS

6.1 NATIONAL FUEL CONSUMPTION AND RELATED DATA

The previous sections of this report presented results on a per car basis. In order to discuss the effect of variations in the number of cars, we obviously must be concerned with total, nationwide results. Table 10 summarizes U.S. motor fuel consumption as derived from the data given in Reference 10.

Table 10. Projected U.S. Annual Passenger Vehicle Motor Fuel Consumption

Year:	1975	1980	1985	1990	2000
Households ($\times 10^6$):	71.1	79.4	86.5	94.3	101
Passenger Cars ($\times 10^6$)	95.2	107.3	113.2	118	127
Annual Travel per Car (km $\times 10^3$)	17.5	18.4	19.0	19.6	20.2
Fuel economy (km/ℓ)	5.78	6.51	8.12	9.36	9.87
Liters of Fuel per Car per Year	3920	2830	2340	2160	2050
Cost of Fuel per Liter ($/ℓ)	0.150	0.202	0.252	0.291	
Cost of Fuel per Car per Year ($)	453	572	587	629	
Total Fuel per Year (ℓ $\times 10^9$)	287	304	264	250	
Total Cost per Year ($ $\times 10^9$)	43	61	67	73	

In estimating the number of passenger cars, one must distinguish between cars registered and cars available. Generally, only 85 percent of the registered cars are available, because some are registered in different states in the same year, some are junked after being registered, and so on. General Research Corporation (Reference 5) found that the number of available cars can be closely estimated by assuming that there are 1.25 cars per household. The use of this figure corresponds well with Reference 10 and we assume, therefore, that the numbers given there represent available cars.

Nationwide motor fuel consumption data collected by the Federal Highway Administration of the U.S. Department of Transportation are given in Reference 7. It appears, for example, that passenger cars used 76.5 billion gallons of motor fuel in 1975, equivalent to 290 billion liters, and thus is in excellent agreement with the 287 billion liters for 1975 given in Table 10. This is not surprising, since JPL used national fuel consumption data to arrive at its estimates.

We note, in passing, that the total amount of motor fuel used in the United States is expected to decline after 1980 while the total cost will continue to rise modestly.

Table 10 shows that a total of $67 billion is expected to be spent on motor fuel in 1985. One incentive for reducing oil imports is to lessen the balance of payments deficit which in 1978 was approximately $30 billion.

At present, oil is imported at about $16 per barrel or 38¢/gal, while gasoline sells for about 95¢/gal (JPL's 1985 price. Thus, 1985 appears to be here.

Assume then that the total motor fuel used represents 38/95 x 67 or $28 billion of imported oil. Thus, roughly speaking, the fraction of passenger car motor fuel which could be saved by hybrid vehicles equals the fraction by which the balance of payments could be reduced. For example, if hybrid vehicles found sufficient acceptance to reduce passenger car motor fuel consumption by half, the balance of payments deficit would also be reduced by approximately half.

In addition to the desire to protect the balance of payments and the dollar, there is the even more fundamental idea that there simply may not be enough oil to go around, and that consumption will have to be reduced, like it or not. Pursuing this line of argument, we note that in 1975 about half (56 percent) of all the petroleum consumed in the U.S. was used in transportation, as shown in Table 11.

Table 11. 1975 U.S. Allocation of Petroleum Consumption

User	Amount (billion barrels*)
Households	1.006
Industrial	1.049
Transportation	3.334
Electricity	0.520
Miscellaneous	0.049
Total	5.958

*1 barrel = 42 gallons or 159 liters

Consumption today is higher on the order of 21 million barrels per day* or 7.665 billion barrels per year; however, the proportion allocated to each use, shown in Table 11, has probably remained fairly constant. At today's price of $16 per barrel for imported oil the total spent in the U.S. on oil is $122 billion.** Of this amount, 56 percent is spent on transportation, and 70 percent of the petroleum used in transportation (i.e., 39 percent of all petroleum) is used in passenger cars, the remainder going mostly to trucks with lesser amounts to public transport. Thus, the oil bill for passenger cars is $48 billion** when calculated in this manner. Since about half of the oil used in the U.S. is imported and 39 percent is used in passenger cars, a reduction in imported oil, of say 10 percent, could be totally absorbed by a reduction in passenger car fuel of approximately the same amount (actually about 50/39 × 10 = 12.8 percent.)

In summary, the fraction of fuel saved by hybrid vehicles is approximately equivalent to the fractional decrease in the balance of payments deficit, or the fractional decrease in oil imports. If all passenger cars could somehow be eliminated, there would be, broadly speaking, no oil imports and no balance of payments deficit.

*Secretary Schlesinger quoted in the Santa Barbara News Press, 19 March 1979.

**At present domestic oil sells at $8 per barrel so that, today, this figure is too high but chances are good that domestic oil will be decontrolled and that its price will rise to waste market values.

6.2 SENSITIVITY TO NUMBER OF PASSENGER CARS

In Reference 10 we were asked to consider the sensitivity of mission analysis and design trade-off study results to the number of passenger cars. The principal result of varying the number of cars is the obvious one of changing the amount of fuel used and the potential fuel savings. These effects can be studied by means of Table 12 which shows the liters of fuel used and dollars expended by reference vehicles and the baseline NTHV as a function of the number of passenger cars. Dollar amounts were computed on the basis of an imported oil price of $16 per barrel or, as it happens, almost exactly 10 cents per liter. The table indicates that on Mission A about nine billion dollars could be cut from the bill for passenger cars. As the table further shows, the results are not overly sensitive to variations in the total number of passenger cars.

Table 12. Effect of the Number of Passenger Cars on U.S. Motor Fuel Consumption (Annual Distance = 19073 km)

(Liters of fuel $\times 10^9$ or Billion $ $\times 10^{-1}$)

	Reference Vehicle			36 km Range NTHV			Savings		
No. of Pass. Cars	105	113	121	105	113	121	105	113	121
Mission AA	208	224	240	76	82	88	132	142	152
Mission A	145	156	167	63	67	72	82	88	95
Mission C	46	49	53	6.7	6.7	7.7	39	42	45

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APPENDIX A

CANDIDATE SYSTEM

MONTE CARLO COMPUTER SIMULATION RESULTS

(See Table 7, Section 3)

RANGE(KM)	NEW KM/YR	OLDKM/YR
21.0	15919.0	15919.0

MISSION NUMBER 1

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	5918.5	5824.5	6111.0	6630.1	12029.5	12456.7	2.99	11.42	34.12
10	5535.3	5803.5	5749.5	6587.9	11284.8	12391.4	2.99	11.36	33.95

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.5317	438.4	585.1	69.5

MISSION NUMBER 2

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	4317.8	4337.3	3015.8	3224.0	7333.5	7561.3	2.00	13.44	26.88
10	4381.8	4342.7	3752.0	3519.1	8133.8	7861.9	2.00	14.09	28.18

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.4476	242.0	407.4	48.4

MISSION NUMBER 3

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	2741.0	2895.3	1107.3	1048.1	3848.3	3943.4	1.23	8.79	10.80
10	2795.5	2870.8	1069.3	962.6	3864.8	3833.5	1.23	8.54	10.50

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.2511	76.8	239.8	28.5

MISSION NUMBER 4

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	5897.3	5863.6	9270.5	9812.4	15167.8	15676.0	3.05	14.08	42.95
10	6016.5	5880.9	11851.5	10224.9	17868.0	16165.7	3.05	14.47	44.13

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.6349	460.8	669.6	79.5

Figure A-1

RANGE(KM)	NEW KM/YR	OLDKM/YR
21.8	20000.0	15919.0

MISSION NUMBER 1

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	6388.2	6247.8	9100.3	9218.6	15488.5	15466.4	3.35	12.65	42.37
10	6223.2	6226.0	8145.0	9034.7	14368.2	15260.8	3.35	12.48	41.81

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.5920	590.3	670.2	79.6

MISSION NUMBER 2

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	4485.0	4478.3	5254.1	5072.9	9739.2	9551.1	2.00	17.76	35.53
10	4502.3	4491.0	5813.0	5172.3	10315.2	9663.3	2.00	18.01	36.01

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.5353	343.8	454.4	54.0

MISSION NUMBER 3

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	3011.1	3221.4	1169.8	1417.0	4180.9	4638.5	1.38	9.22	12.71
10	3123.1	3212.9	1537.2	1404.2	4660.3	4617.2	1.38	9.18	12.65

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.3041	105.9	275.5	32.7

MISSION NUMBER 4

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	6349.3	6328.0	13484.6	13566.4	19833.9	19894.4	3.42	15.94	54.51
10	6292.5	6338.2	13541.1	13940.4	19833.6	20278.6	3.42	16.25	55.56

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.6874	890.3	784.7	93.2

Figure A-2

RANGE(KM)	NEW KM/YR	OLDKM/YR
21.0	24000.0	15919.0

MISSION NUMBER 1

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	6546.4	6540.0	10805.0	11716.2	17351.5	18256.1	3.67	13.53	50.02
10	6528.3	6541.3	10641.7	11635.9	17170.1	18177.2	3.67	13.57	49.80

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.6401	751.0	750.4	89.2

MISSION NUMBER 2

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	4591.2	4582.3	6075.5	6787.9	10666.6	11370.3	2.00	21.72	43.44
10	4617.6	4589.4	7463.6	6874.5	12081.2	11464.0	2.00	21.92	43.84

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.5997	448.3	498.7	59.2

MISSION NUMBER 3

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	3629.1	3523.1	1866.2	1789.6	5495.3	5312.7	1.51	9.64	14.56
10	3450.2	3527.2	1499.9	1788.1	4950.1	5315.3	1.51	9.65	14.56

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.3364	131.3	307.7	36.6

MISSION NUMBER 4

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	6696.7	6630.4	16546.9	16849.0	23243.6	23479.4	3.74	17.18	64.33
10	6687.7	6611.3	16186.9	17619.6	22874.7	24230.8	3.74	17.73	66.39

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.7272	1116.4	885.0	105.1

Figure A-3

RANGE(KM)	NEW KM/YR	OLDKM/YR
36.0	15919.0	15919.0

MISSION NUMBER 1

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	8529.8	8337.0	3499.8	4117.6	12029.5	12454.7	2.99	11.42	34.12
10	7715.0	8281.4	3569.8	4110.0	11284.8	12391.4	2.99	11.36	33.95

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.3317	321.4	702.2	83.4

MISSION NUMBER 2

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	5841.3	5936.9	1492.3	1624.4	7333.5	7561.3	2.00	13.44	26.88
10	6048.3	6008.1	2085.5	1853.8	8133.8	7861.9	2.00	14.09	28.18

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.2358	158.1	491.3	58.4

MISSION NUMBER 3

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	3277.8	3488.4	570.5	454.9	3848.3	3943.4	1.23	8.79	10.80
10	3398.8	3432.1	466.0	401.4	3864.8	3833.5	1.23	8.54	10.50

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.1047	46.6	270.0	32.1

MISSION NUMBER 4

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	8490.5	8419.5	6677.3	7256.6	15167.8	15676.0	3.05	14.08	42.95
10	8896.0	8459.8	8972.0	7646.0	17868.0	16105.7	3.05	14.47	44.13

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.4747	557.5	772.8	91.8

Figure A-4

RANGE(KM)	NEW KM/YR	OLDKM/YR
36.0	20000.0	15919.0

MISSION NUMBER 1

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	9610.5	9268.8	5878.1	6197.6	15488.5	15466.4	3.35	12.65	42.37
10	9019.6	9179.0	5348.6	6081.8	14368.2	15260.8	3.35	12.48	41.81

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.3985	457.7	802.9	95.4

MISSION NUMBER 2

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	6465.7	6441.9	3273.4	3109.2	9739.2	9551.1	2.00	17.76	35.53
10	6582.1	6474.1	3733.1	3189.3	10315.2	9663.3	2.00	18.01	36.01

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.3300	249.6	548.6	65.2

MISSION NUMBER 3

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	3607.0	3924.5	573.9	713.9	4180.9	4638.5	1.38	9.22	12.71
10	3804.1	3925.8	856.2	691.4	4660.3	4617.2	1.38	9.18	12.65

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.1497	68.7	312.6	37.1

MISSION NUMBER 4

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	9550.1	9434.3	10283.8	10460.0	19833.9	19894.4	3.42	15.94	54.51
10	9434.5	9447.5	10399.1	10831.1	19833.6	20278.6	3.42	16.25	55.56

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.5341	775.1	899.9	106.9

Figure A-5

RANGE(KM)	NEW KM/YR	OLDKM/YR
36.0	24000.0	15919.0

MISSION NUMBER 1

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	9934.2	9936.9	7417.2	8319.3	17351.5	18256.1	3.67	13.63	50.02
10	9772.1	9902.6	7397.9	8274.6	17170.1	18177.2	3.67	13.57	49.80

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.4552	607.7	893.7	106.2

MISSION NUMBER 2

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	6780.6	6824.1	3886.0	4546.2	10666.6	11370.3	2.00	21.72	43.44
10	6944.2	6854.8	5137.0	4609.2	12081.2	11464.0	2.00	21.92	43.84

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.4021	346.3	600.7	71.4

MISSION NUMBER 3

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	4564.7	4369.9	930.6	942.8	5495.3	5312.7	1.51	9.64	14.56
10	4219.2	4373.9	730.9	941.4	4950.1	5315.3	1.51	9.65	14.56

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.1771	88.0	351.1	41.7

MISSION NUMBER 4

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	10326.3	10171.6	12917.3	13307.8	23243.6	23479.4	3.74	17.18	64.33
10	10152.6	10094.3	12722.1	14136.6	22874.7	24230.8	3.74	17.73	66.39

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.5834	998.6	1002.8	119.1

Figure A-6

RANGE(KM)	NEW KM/YR	OLDKM/YR
49.0	15919.0	15919.0

MISSION NUMBER 1

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	9951.3	9774.7	2078.3	2680.0	12029.5	12454.7	2.99	11.42	34.12
10	8918.5	9698.8	2366.3	2692.7	11284.8	12391.4	2.99	11.36	33.95

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.2173	299.4	724.2	86.0

MISSION NUMBER 2

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	6496.3	6662.6	837.3	898.6	7333.5	7561.3	2.00	13.44	26.88
10	6912.3	6806.6	1221.5	1055.3	8133.8	7861.9	2.00	14.09	28.18

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.1342	152.2	497.2	59.1

MISSION NUMBER 3

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	3581.0	3747.8	267.3	195.6	3848.3	3943.4	1.23	8.79	10.80
10	3681.3	3672.5	183.5	161.0	3864.8	3833.5	1.23	8.54	10.50

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.0420	53.8	262.9	31.2

MISSION NUMBER 4

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	9865.5	9854.5	5302.3	5821.6	15167.8	15676.0	3.05	14.08	42.95
10	10571.5	9923.5	7296.5	6182.3	17868.0	16105.7	3.05	14.47	44.13

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.3839	544.2	786.2	93.4

Figure A-7

RANGE(KM)	NEW KM/YR	OLDKM/YR
49.0	20000.0	15919.0

MISSION NUMBER 1

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	11543.7	11128.5	3944.8	4337.9	15488.5	15466.4	3.35	12.65	42.37
10	10674.9	10990.0	3693.3	4270.7	14368.2	15260.8	3.35	12.48	41.81

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.2799	423.9	836.7	99.4

MISSION NUMBER 2

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	7665.1	7552.8	2074.1	1998.4	9739.2	9551.1	2.00	17.76	35.53
10	7820.8	7604.8	2494.4	2058.5	10315.2	9663.3	2.00	18.01	36.01

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.2130	231.1	567.1	67.4

MISSION NUMBER 3

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	3881.9	4279.8	299.0	358.7	4180.9	4638.5	1.38	9.22	12.71
10	4160.5	4275.8	499.8	341.4	4660.3	4617.2	1.38	9.18	12.65

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.0739	73.3	308.1	36.6

MISSION NUMBER 4

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	11437.6	11318.6	8396.3	8575.8	19833.9	19894.4	3.42	15.94	54.51
10	11358.3	11353.0	8475.3	8925.6	19833.6	20278.6	3.42	16.25	55.56

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.4402	751.1	923.9	109.8

Figure A-8

RANGE(KM)	NEW KM/YR	OLDKM/YR
49.0	24000.0	15919.0

MISSION NUMBER 1

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	12169.1	12156.5	5182.4	6099.7	17351.5	18256.1	3.67	13.63	50.02
10	11817.7	12098.8	5352.3	6078.4	17170.1	18177.2	3.67	13.57	49.80

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.3344	562.2	939.3	111.6

MISSION NUMBER 2

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	7975.9	8187.7	2690.8	3182.5	10666.6	11370.3	2.00	21.72	43.44
10	8345.8	8228.3	3735.3	3235.6	12081.2	11464.0	2.00	21.92	43.84

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.2822	320.0	626.9	74.5

MISSION NUMBER 3

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	5020.4	4813.9	474.9	498.8	5495.3	5312.7	1.51	9.64	14.56
10	4550.2	4807.8	399.9	507.4	4950.1	5315.3	1.51	9.65	14.56

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.0955	91.0	348.1	41.3

MISSION NUMBER 4

YEAR	ANNUAL ELEC	AVE ELEC	ANNUAL GAS	AVE GAS	ANNUAL	AVERAGE	TRPNO	TRPLEN	DADIST
5	12680.5	12519.4	10563.1	10960.0	23243.6	23479.4	3.74	17.18	64.33
10	12436.9	12425.5	10437.8	11805.4	22874.7	24230.8	3.74	17.73	66.39

FRACFUEL	TOTFUEL	LITERS SAVED	DOLLARS SAVED
0.4872	963.4	1038.0	123.3

Figure A-9