REPORT NO. DOT-TSC-OST-77-23, II

HYBRID VEHICLE TECHNOLOGY CONSTRAINTS AND APPLICATION ASSESSMENT STUDY

Volume II: Sections 1 through 4

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Environment and Energy Conservation Division
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NOVEMBER 1977

FINAL REPORT

DOCUMENT IS AVAILABLE TO THE U.S. PUBLIC THROUGH THE NATIONAL TECHNICAL INFORMATION SERVICE, SPRINGFIELD, VIRGINIA 22161

Prepared for

U.S. DEPARTMENT OF TRANSPORTATION
OFFICE OF THE SECRETARY
Office of the Assistant Secretary for Systems Development and Technology
Office of Systems Engineering
Washington DC 20590

Technical Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
DOT-TSC-OST-77-23, II		

4. Title and Subtitle	5. Report Date
HYBRID VEHICLE TECHNOLOGY CONSTRAINTS AND APPLICATION ASSESSMENT STUDY Volume II: Sections 1 - 4	November 1977
	6. Performing Organization Code
7. Author(s) D. E. Lapedes, M. G. Hinton, L. Forrest, J. Kohlenberger, T. Ryan, H. Sampson, W. Smalley, C. Speisman, H. White.	8. Performing Organization Report No. DOT-TSC-OST-77-23, II
9. Performing Organization Name and Address The Aerospace Corporation* Environment and Energy Conservation Division El Segundo, CA 90245	10. Work Unit No. (TRAIS) OS714/R7508
	11. Contract or Grant No. F04701-76-C-0077
12. Sponsoring Agency Name and Address U. S. Department of Transportation, Office of the Secretary, Office of the Asst. Sec. for Sys. Dev. & Tec., Office of Systems Engineering Washington, DC 20590	13. Type of Report and Period Covered Final Report April 1975 - June 1976
	14. Sponsoring Agency Code

15. Supplementary Notes

*Under contract to: U. S. Department of Transportation
Transportation Systems Center
Kendall Square
Cambridge, MA 02142

16. Abstract

This four-volume report presents analyses and assessments of both heat engine/battery-and heat engine/flywheel-powered hybrid vehicles to determine if they could contribute to near-term (1980-1990) reductions in transportation energy consumption under several sets of operational conditions: urban driving, highway driving, and stop-start, low-speed delivery service conditions. In addition, the impact of such hybrid vehicle use on vehicle-related exhaust emissions is determined, and the ability to accommodate a different energy resource base in the longer term is evaluated; i.e., by permitting a portion of the recharge energy for the on-board energy storage device (battery or flywheel) to be provided by wall-plug electric power from the utility industry instead of from the on-board heat engine. Alternative paths for power transmission from the heat engine to the vehicle drive wheels are considered along with the potential of regenerative braking to reduce vehicle energy consumption.

This second of four volumes contains the first four sections of the full report. It introduces the methods used in the study and the data base employed in simulation modeling of each vehicle powertrain. It also includes a technology review of powertrain components and various hybrid systems developed in recent years.

17. Key Words	18. Distribution Statement
Hybrid Automotive Vehicles, Transportation Energy Consumption, Automotive Vehicle Exhaust Emissions, Batteries, Flywheels, Motors-Generators, Heat Engines, Transmissions, Regenerative Braking.	DOCUMENT IS AVAILABLE TO THE U.S. PUBLIC THROUGH THE NATIONAL TECHNICAL INFORMATION SERVICE, SPRINGFIELD, VIRGINIA 22161

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
Unclassified	Unclassified	226	

Form DOT F 1700.7 (8-72) Reproduction of completed page authorized

PREFACE

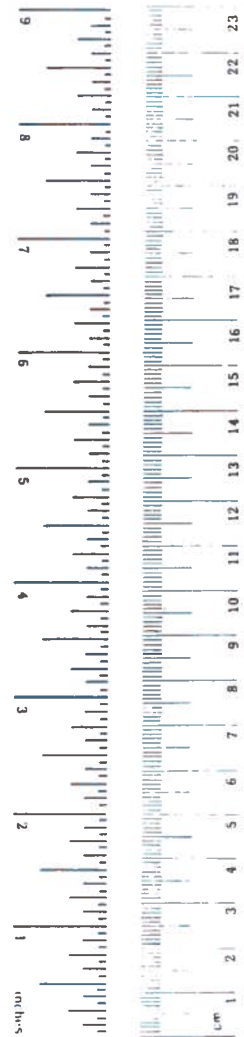
This four-volume report presents the results of an analysis conducted by The Aerospace Corporation for the U.S. Department of Transportation, Transportation Systems Center, as part of the Automotive Energy Efficiency Project, sponsored by the Energy and Environment Division of the Office of the Secretary, U.S. Department of Transportation.

Appreciation is hereby extended to the Technical Monitor at the Transportation Systems Center, Mr. Joseph Abbas, whose guidance and suggestions were most helpful to the conduct of the study.

METRIC CONVERSION FACTORS

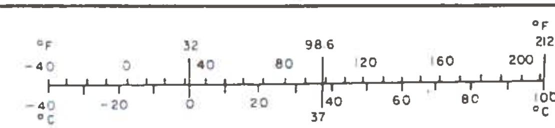
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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1. INTRODUCTION

1.1 STUDY OBJECTIVES

A shortage in domestic fuels* for supplying the nation's energy needs has prompted numerous investigations into potential means for energy conservation and for use of alternative sources of energy. The transportation sector, because of its large and growing energy needs, has been the subject of intense study in recent years. In particular, the fuels and vehicle propulsion systems used for personal transportation have been scrutinized carefully to ascertain possible fuel savings.

In this regard, one possible scheme that is proposed is the use of a hybrid powertrain in automobiles. This system is envisioned as combining an energy storage device (that can meet rapidly fluctuating propulsion power demands) with a source of recharge energy such as a heat engine that runs almost independent of power demands and hence functions near its most efficient operating point. Additionally, the use of an on-board heat engine relieves the current restrictions in range found with vehicles relying solely on contemporary energy storage systems, e.g., batteries in electric cars. Until energy storage capability of electric vehicles is improved, the hybrid vehicle could be an adequate intermediate system and would likely have less impact on operations of the petroleum industry or the electric utility industry.

Systems of this nature have been investigated previously, with major emphasis on determining the potential for reduction in automotive exhaust emissions. The vehicle performance capabilities were required to match fully those of conventionally powered automobiles, and recharge of the energy storage device was to be accomplished totally by an on-board heat engine.

*Primarily petroleum-based fuels

By contrast, the current study permits relaxation of vehicle performance requirements. It also allows recharge of the energy storage device to be shared between an on-board heat engine and an electric wall outlet. With these revised guidelines, the effect of reductions in vehicle design cruise speed and acceleration on fuel consumption can be determined. Furthermore, because power at an electric wall outlet is supplied from stationary electric generating plants burning a variety of fuels, there exists the potential for consuming less petroleum-based fuels. Of course, the vehicle-related exhaust emissions continue to be a major factor in evaluation of hybrid vehicle systems.

Therefore, the overall objective of this study is to provide the U.S. Department of Transportation (DOT) with a clear indication as to what forms of hybrid vehicles could contribute to reductions in transportation energy consumption under a given set of operational conditions; i.e., determine if there is an application for hybrid vehicles and if they have clear-cut advantages over other types of vehicles.

More specifically, the objectives of this study are (1) to identify and characterize meaningful hybrid vehicle configurations having potential applicability to passenger car and delivery van use modes, (2) to determine the feasibility of implementation of selected hybrid vehicle configurations as a function of performance constraints, (3) to determine the impact of such hybrid vehicle usage on petroleum-based fuel consumption reduction and on vehicle-related exhaust emissions, and (4) to determine the impact of hybrid vehicle use on the ability to accommodate a different energy resource base in the longer term.

1.2 METHODOLOGY FOR STUDY IMPLEMENTATION

Fundamental to the basic approach in this study is the use of empirical data for analytically predicting the operating performance of each major element in the vehicle powertrain. This required the acquisition, review, updating, and summarization of data from previous hybrid vehicle studies and test programs, as well as from manufacturers' component data sheets.

Next, a versatile computer program has been developed to use these data in a simulation model of powertrain operation during vehicle motion over specific driving cycles. Wherever possible, each component was modeled with a full performance map based on contemporary data. This program is also capable of sizing components and of altering component performance characteristics and specified vehicle performance requirements in sequential steps from baseline values so that parametric studies can be performed.

Lastly, the calculated hybrid vehicle-related energy requirements and exhaust emissions were compared with those for other types of vehicles to determine relative advantages and disadvantages. Results from the parametric studies show how changes in vehicle or powertrain component performance (due to degradation or enhancement of assumed values) affect the aforementioned comparison. The subsequent systems analysis indicates those technological constraints that could inhibit vehicle implementation and determines applications in which hybrid vehicles have a meaningful advantage over other types of vehicles.

2. HYBRID VEHICLE CONCEPTS

2.1 POWERTRAIN CONCEPTS AND CONFIGURATIONS

As noted in Section 1, the specific form of hybrid vehicle addressed in this study is one that combines an on-board heat engine with an energy storage system that is recharged by both the heat engine and by power from an electric wall outlet. The hybrid powertrain in this case has been proposed as a propulsion scheme for conserving fuel by virtue of the concept that, if the energy storage device can meet the rapidly fluctuating propulsion power demands, the heat engine can be effectively divorced from these requirements and run at the most efficient operating point. Secondarily, petroleum-based fuels can be conserved if the electric generating plant burns coal, for example. The validity of the concept was examined in a limited study performed earlier for the U.S. Environmental Protection Agency (EPA) (Ref. 2-1).

The energy sources for vehicle propulsion are the fuels consumed by the heat engine and by the stationary electric generating plant that delivers power to the wall outlet. Interposed between these sources of energy and the vehicle drive wheels, a variety of energy storage devices, power flow paths, and powertrain component combinations can be envisioned.

Indeed, as discussed in Section 3, Technology Review of Hybrid Vehicle Systems, the designs that have been analyzed or tested in the form of experimental hardware are quite unique and different. Assessment of the relative advantages and disadvantages of each type of powertrain from a quantitative standpoint would require a study unto itself. Suffice to say that, in general, the fuel consumed for vehicle propulsion would depend more on the operating efficiencies of specific components than on the means of providing power to the drive wheels or, in some cases, on the operating modes selected for various components.

In the present study, the selection of energy storage devices for use in analyses of hybrid powertrains was limited to two types: Batteries and flywheels. The rationale for this selection is that only these systems were sufficiently characterized for automotive application by a complete, empirically-derived operating map. Additional elements in the powertrain needed for power conversion were selected for compatibility with each of these energy storage devices and for good efficiency combined with operating flexibility, as discussed in Section 2.4.

In regard to the means of transmitting power to the vehicle drive wheels by different flow paths, hybrid powertrain concepts can be grouped into two broad classes, as shown in the simplified schematic in Figure 2-1. The first class, series configuration, is characterized by the principle that energy flowing from the heat engine to the rear wheels must first pass through multiple intermediate energy conversion devices. This means of decoupling the engine from the rear wheels provides a large degree of flexibility in engine operating modes.

In the case of a series-configured heat engine/battery hybrid, the heat engine drives an electrical generator that transmits energy to the electric drive motor and thence to the wheels. A portion of the generator energy is directed to recharging the batteries as needed. For a series-configured heat engine/flywheel hybrid, the heat engine drives the flywheel through a transmission and the flywheel drives the rear wheels through another transmission.

The second class, parallel configuration,* is characterized by the principle that a portion of energy flowing from the heat engine to the rear wheels passes, at most, through one energy conversion device. This type of coupling to the rear wheels is somewhat more limited in terms of engine flexibility than for the series configuration, but the power transmission losses are generally less and the overall system efficiency could be higher if engine efficiency were maintained at levels equivalent to that of the series configuration. Furthermore, some of the nonenergy-storing

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*This term was originally used for hybrid heat engine/electric systems to denote the parallel flow of mechanical and electrical energy to the rear wheels. The broader definition as used herein is necessary because of application of the term to heat engine/flywheel systems.

components that are not driven by the engine in the parallel configuration are required to supply acceleration power only, whereas, in the series configuration they are required to supply cruise plus acceleration power. Hence, the size and weight of components in the parallel configuration are reduced from those for the series configuration. This is particularly true in the heat engine/battery system where the electric drive motor (which can operate at three to four times rated power for brief acceleration periods) can be markedly reduced in size.

For a parallel-configured heat engine/battery hybrid, heat engine power in one of the energy flow paths drives a generator to recharge the batteries that are used to provide acceleration power to an electric motor that is differentially geared to the heat engine drive shaft. For a parallel-configured heat engine/flywheel hybrid, heat engine power in one of the energy flow paths drives a flywheel through a transmission; the flywheel then delivers power to the vehicle drive shaft through a transmission and differential gear system. In the parallel energy flow path for both systems, the engine also delivers power directly to the rear wheels through a transmission linked to the differential.

2.2 ENGINE OPERATING MODES

Different operating modes have been considered for the hybrid vehicle. A number of designs are based on the unimodal operating concept, whereby a portion of the heat engine energy is used continually to replace all or a portion of energy drained from the on-board energy storage device (battery or flywheel). Other designs have resulted in a form of trimodal operating scheme whereby the vehicle can be driven alternatively in the (1) "hybrid" (engine on-board recharging) mode, (2) all-battery (or all-flywheel) mode, or (3) all-engine mode. A somewhat simpler version of this design is the bimodal operating scheme whereby the vehicle is driven only in an all-battery (or all-flywheel) mode or all-engine mode. For this case, the vehicle would normally be driven in the battery (or flywheel) mode with recharging provided by a source external to the vehicle; the engine is then used merely to extend vehicle operating range whenever required.

Several forms of heat engine operating modes can be conceived for the single "hybrid" mode of vehicle operation (Figure 2-2). These modes are discussed first for the series configuration and then for the parallel configuration. In either case, a design can be evolved to ensure either partial or full recharging of the on-board energy storage device.

2.2.1 Heat Engine Operated Continuously at Constant Speed (rpm) and Power Output

In this mode of operation, a severe problem arises in relation to sizing the heat engine. If the heat engine is sized only to produce the total energy required in the time duration of a given driving cycle (including inefficiencies of the powertrain system), then the heat engine may not provide the proper continuous high-speed wheel power required for highway operation. This results in discharge of the energy storage device at high speeds (if the heat engine size is too small). Conversely, if the heat engine is sized for the maximum continuous wheel power demanded for highway operation, excessive energy loss to a dissipation heat-sink occurs at lower wheel power demands.

This mode of heat engine operation is of course attractive from the standpoint of heat engine exhaust emissions and/or fuel economy in that it should be possible to select a single operating point (i.e., rpm, air-fuel ratio, etc.) most amenable to reduced emissions and/or improved fuel economy. However, its apparent inflexibility with regard to heat engine sizing and operation eliminates it from consideration as a viable mode of operation.

2.2.2 On-Off Operation of Heat Engine

As an alternative to continuous operation, it is possible to operate a constant power output heat engine in an on-off mode. Here, the heat engine would be sized to meet the continuous high-speed power demand for highway operation, and would operate intermittently during urban driving conditions. The heat engine could be turned on or off in response to vehicle power demands or state-of-charge of the energy storage device.

In general, this form of operation would require clutching the engine in and out of the powertrain at periodic intervals determined by the recharging needs of the energy storage system. Effects on engine and clutch lifetime are unknown, and the engine starting energy must be accounted for in system losses. Also, a more complicated control system would likely be required. In addition, for a hybrid heat engine/battery powertrain, this mode of operation can result in very high energy losses during those periods when the electric drive motor power demand is low and a good portion of heat engine power output must be dissipated because the battery cannot accept power at the rate being supplied.

2.2.3 Heat Engine Operated Continuously with Variable Speed and Power Output

Many of the deficiencies of the constant-power output mode of operation can be avoided by allowing the power output of the heat engine to vary. In this case, the heat engine can be sized for the maximum continuous power requirement and be allowed to operate at lower power levels for those periods of vehicle driving cycles that require less power. The heat engine power output might be scheduled in various ways as a function of the vehicle velocity (heat engine produces more power as the load increases). If engine speed is also allowed to vary to produce changes in power output (as in conventional internal combustion engines), it is envisioned that the control system can effectively modulate throttle setting response times so that engine speed and power changes take place at a controlled rate. This can be accomplished in such a manner that no true vehicle acceleration demands are imposed on the heat engine in the conventional sense.

The matching of all possible vehicle duty cycle energy requirements may not be possible. To overcome this difficulty, a throttle "bias" feature in the heat engine fuel control system could increase the baseline heat engine power output schedule in accordance with an input signal related to the state-of-charge of the energy storage device.

2.2.4 "Step Mode" Operation

Another technique for varying heat engine power output is to schedule power output in discrete steps. Figure 2-2 includes an illustration of one approach, wherein three power output levels are used. A low level could be scheduled for a low-velocity range (e.g., 0 to 30 mph), an intermediate level for velocities between the low-velocity range and vehicle top speed, and a peak level for operating at maximum continuous cruise speed.

2.3 SELECTION OF ENGINE OPERATION MODE

The method of engine operation selected for use in analysis of the hybrid powertrain is based on simplicity of operation combined with maintaining acceptable powertrain efficiency. Generally speaking, fixing power and speed would offer two advantages. First, as already noted, the engine could be run near its most efficient operating point. Second, were a new engine to be designed, considerable simplicity would be expected to result from this approach. However, for the heat engine/battery hybrid, fixed conditions imply an engine setting at large power levels to avoid extensive energy extraction from the less efficient battery (e.g., during vehicle high-speed cruise). An engine operating in this manner would waste considerable fuel during periods of vehicle deceleration and stopping. Alternatively, the unknown factors influencing the overall powertrain system lifetime and the control system complexity preclude selection of on-off operation at this time. Hence, as a compromise, the following approach, elected for the heat engine/battery hybrid in an earlier study (Ref. 2-2) is carried on in this study. For consistency, the heat engine/flywheel system is operated in the same manner. Other forms of operation may have equal or superior merit, but these alternatives were not evaluated quantitatively in this study. Suffice to say that each contending design must be examined in detail separately.

In essence, heat engine power at the generator output (for battery system) or engine transmission output (for flywheel system) is maintained at a fixed value up to a given vehicle speed. Figure 2-3 illustrates this power variation. As can be seen, the shift from fixed power takes place only when the vehicle road load, as determined at the generator or engine transmission output, exceeds the fixed power level. Then the generator or transmission power output is increased and simply follows the increasing road load requirement as vehicle speed increases. At the fixed power setting of the generator or transmission the heat engine power and speed are essentially fixed; slight variations would occur if the generator efficiency varied (due to load changes) or if the transmission efficiency varied (due to flywheel speed changes).

The various vehicle power demand situations are noted on Figure 2-3. At point A, the vehicle is at a low speed cruise condition. Excess power is then used to recharge the energy storage system (assuming it can be accepted). Energy that can't be accepted must be diverted to a separate load and is considered wasted. At point B, the vehicle undergoes a mild acceleration (or powered deceleration), and again the excess power is directed to the energy storage system. A higher acceleration is shown for point C, and no excess power exists. An even higher acceleration is found at point D, and with the fixed power setting exceeded, the additional power must come from the energy storage system. A high speed cruise condition is shown at point E, where the heat engine output was ncessarily increased. And finally, at point F, a vehicle acceleration is called for during high speed cruise (e.g., a highway passing maneuver) and the energy storage system must provide supplemental power.

As is evident, the power and speed changes from the heat engine are slowly varying and could permit efficient operation of the engine. In addition, the choking required for conventional carbureted gasoline engines during cold starts could be reduced or possibly eliminated. Therefore, improvements in exhaust emissions and fuel consumption might be expected.

When the vehicle is stopped or decelerating, the generator (transmission) power output is also kept fixed (unless it is experiencing a high-speed powered deceleration). If the energy storage system is fully charged, some energy savings would be possible if the engine were throttled back to an idle condition. This degree of sophistication, however, was not introduced into the computer program model of the hybrid powertrain. Indeed, it would prove somewhat more difficult to achieve for the flywheel system because the direct mechanical linkage provided by the transmission would require a declutching mechanism.

In regard to recharge energy from the electric wall outlet, it was assumed that this was accomplished only at the end of a given period of driving. This is advantageous for the battery system because higher recharge currents can be accepted by a depleted battery than by a battery near a full state-of-charge. (Hence, less waste of recharge energy from the recharging generator is possible.) However, achievement of maximum vehicle acceleration may be marginal with a heavily depleted battery for the case where the electric wall outlet is relied on for providing the major portion of recharge energy. Of course, manual override of the programmed engine operation could provide high power on-board recharge at driver option if a series of maximum accelerations were contemplated in advance.

2.4 COMPONENT DESIGN CONSIDERATIONS AND SELECTION

2.4.1 General

Assembly of all required powertrain components into a lightweight, compact, efficient, and low-cost unit will undoubtedly prove to be a difficult task for design teams planning a prototype hybrid automotive system. Considering the numerous complex interactions between various components, great care must be taken during design studies to assure that there is proper selection and integration of all elements in the powertrain. This will avoid an inadvertent conclusion that a given system has poor performance and is therefore unacceptable. Within the scope of this study, it was decided to rely on a simple set of guidelines and, wherever possible, to conduct a parametric analysis that would give some indication as to the potential advantages or disadvantages of alternative approaches. A discussion of the design alternatives follows.

The basic vehicle to be analyzed is a full-size, 6-passenger automobile with a test weight of 4000 pounds.* This is the inertia weight (the loaded weight) used in calculations of vehicle propulsion power requirements while operating over a given driving cycle, and is for a car with passenger and cargo weight added to the basic curb weight. It is also the weight used in selecting the fuel economy of the reference gasoline-powered, conventional car from data compiled by the U.S. Environmental Protection Agency.

Allowable weight and volume for the hybrid powertrain were obtained from the data given in Ref. 2-2. As formerly, the same philosophy

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*A 2500-lb car and a 6000-lb delivery van were also analyzed.

prevailed; that is, weight and volume for all components other than the energy storage system are subtracted from the allowable value for the powertrain. The remainder (if any) is then assigned to the energy storage device and thereby becomes a design goal. The design goals are expressed in terms of specific power (watts/pound) and specific energy (watt-hours/pound). Additional discussion relating vehicle performance requirements to component size and energy storage system design goals is given in Section 9.

With regard to choice of components for each type of hybrid vehicle, there is a large list from which to make selections, as can be seen from Table 2-1. In addition, various means of using these components can be envisioned. For example, power to be delivered from a flywheel energy storage system to the vehicle drive wheels can be accomplished by use of a transmission or by use of a motor/generator. In the case of a battery storage system, one can consider ac versus dc electric drive motors. The relative advantages or disadvantages can be argued at length, but an experimental test program will be required for resolving alternative choices for a given design. The discussion that follows presents the components selected for each type of powertrain, and offers a brief rationale for each selection. Of uppermost consideration, however, was the use of components based on existing technology for near-term (within a decade) application and for which a reliable, extensive data base was available. Figure 2-1 and Table 2-1 can be referred to for reference purposes throughout the succeeding discussion.

The heat engine selected for evaluation with both battery and flywheel hybrid powertrains is the spark ignition, reciprocating piston (SIRP) engine, fueled with gasoline. It is well characterized by specific engine performance maps for both fuel consumption and exhaust emissions -- a pairing necessary for realistic evaluation of any engine candidate. This information, coupled with near-term availability, makes this engine a first choice (although the rotary piston engine could also suffice). Except for diesel engines, performance maps for other engines were considered inadequate.

The arguments advanced for choice of the SIRP engine could well apply to the compression-ignition (diesel) engine. However, a previous study (Ref. 2-2) showed that this engine as presently designed is too large and heavy for use in a hybrid powertrain. Furthermore, progress in making major reductions in exhaust emissions seems to be found more often with the spark ignition engine linked with exhaust treatment (catalyst) devices. Lastly, because the relative advantages and disadvantages of hybrid vehicles are to be contrasted with conventional vehicles in current operation, and because the majority of these conventional vehicles are powered by the SIRP engine, it would seem appropriate to use a compatible engine in the hybrid vehicle.

While discussing the heat engine, it is appropriate to note the means selected for powering engine auxiliaries and vehicle accessories. For purposes of this study, all such needs are extracted from the heat engine power output shaft. The power absorption of such units is then dependent on engine speed; therefore, for a given level of propulsion power requirement, operation of the engine at low speeds is preferable, providing that the engine efficiency or lifetime is not markedly degraded under these conditions.

2.4.2 Heat Engine/Battery Powertrain

For the battery hybrid, an ac electric generator running at 12,000 rpm was selected to transfer power from the engine to the drive motor and to the battery. An ac unit was preferred over a dc unit because of its smaller size and weight. The added complexity of controlling and rectifying output ac to dc was considered a minor problem. The speed selection results in reduced weight and yet is not so high as to require expensive bearings.

The electric drive motor selection was especially difficult because of competing design philosophies found in the literature. A series-wound, dc motor was chosen because its torque-speed characteristics matched well with those needed for automotive application, and it represented a reasonable compromise between weight, volume, cost, lifetime, flexibility, proven performance, and availability.* Although an ac motor would be lighter and possibly more efficient, the added weight, volume, and cost of the associated power inverter system precluded its selection at this time. An experimental test bed program would be desirable to fully evaluate the relative merits of the ac system with modern power inverter systems.

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*The externally excited dc motor is preferred by some designers, particularly for avoiding high current switching in regenerative braking systems.

As for regenerative braking by the drive motor, previous studies of its effectiveness for energy savings are discussed in Section 3. These studies of hybrid and all-electric vehicles have made differing claims for energy savings to be expected from this form of operation. Long, slow decelerations are penalized because most of the energy to be regenerated is never utilized, being lost in road and aerodynamic losses. Rapid decelerations result in high power pulses, leading to larger than normal losses in the motor (acting as a generator), and in limitations in power acceptance for energy storage devices such as batteries. Also, extended periods of vehicle cruising would not benefit from regenerative braking.

One other consideration is worthy of mention. The efficiency of the motor decreases considerably when it is required to deliver high torque at low speed. To offset this effect, a manual gear shift could be interposed between the motor output shaft and the rear end differential gear box.* Hence, one gear ratio might suffice for speeds of, say, 0-20 mph and another gear ratio for speeds in excess of 20 mph. Of course, an automatic transmission would result in even superior overall efficiency, but added weight, volume, and cost could be considered detrimental to its use. Neither manual nor automatic gear shifting, however, were included in the computer program. This added sophistication might be considered for future studies in terms of minimizing the combined motor-transmission size and cost.

Although other types of batteries possess superior specific power/specific energy characteristics, for reasons cited earlier of near-term applicability and availability of complete performance maps, the selection of rechargable batteries for consideration in this study was rapidly narrowed to three types: lead-acid, nickel-zinc, and nickel-cadmium. The nickel-cadmium battery was finally deleted from consideration because of high cost. For the remaining two types of batteries, performance maps related the voltage to a given discharge current or charge current. To prolong battery lifetime, a limitation was placed on the maximum discharge current and charge current at any given battery state-of-charge. A further approach

*Additional direct benefits are also apparent: (1) current drain on the battery is reduced, (2) the limits on peak motor speed can be relieved, and (3) motor size and cost can be reduced.

to extending lifetime was to limit the energy extracted to three-fourths of the total battery capacity. A solid-state chopper control was envisioned as the means of regulating battery power delivered to the electric drive motor.

2.4.3 Heat Engine/Flywheel Powertrain

Two major studies (Refs. 2-3, 2-4) have previously reviewed flywheel designs for hybrid vehicles, but with recharging accomplished only by the on-board heat engine. These studies were relied on for selecting and characterizing flywheel systems (primarily Ref. 2-3). The flywheel system consists of more than just the rotor wheel. In addition, there is the power delivery shaft and associated bearings, a housing to permit the operation of the rotor in a reduced pressure atmosphere so that windage losses are minimized, a vacuum pump to initiate and maintain the reduced pressure, shaft seals to inhibit leakage of outside air into the housing, and lastly, an impact barrier ring surrounding the periphery of the rotor.

All of these auxiliaries result in a flywheel system with a substantial increase in volume and weight over that for the rotor alone. Also, by mounting the rotor shaft in line with the heat engine shaft to simplify gearing, the rotor diameter is restricted to about 1 foot for a 4000-lb car (limiting the energy stored). One additional restriction on energy is that imposed to prolong rotor cyclic fatigue lifetime by restricting the operational speed range of the wheel. To be compatible with batteries, wherein three-fourths of the stored energy is considered usable, the flywheel speed range is limited from maximum speed down to one-half maximum speed (because of the relationship that energy stored is proportional to speed squared).

For a given flywheel inertial mass, the greater the rotational speed, the greater the energy stored. Hence, the highest possible speed is most desirable if the specific energy of this type of storage system is to be maximized. However, following the design considerations of Ref. 2-3, speeds were generally limited to the neighborhood of 25,000 rpm because of the nature of the contemplated automotive environment loads (precession, shock, etc.), as well as the expense and lifetime problems associated with bearings capable of operation at higher speeds.* The problem of bearing loads induced by gyroscopic torques was not addressed in the review of various flywheel designs. Counter-rotating wheels are recognized, however, as a means of relieving the torques that ultimately act on the vehicle chassis.

The flywheel inertial mass is dictated by the configuration of the rotor, the size of the rotor, and the type of material chosen. Generally, a rotor would be designed in the shape of a constant stress disc which is tapered in decreasing axial thickness radially from the hub out to the rim. This type of design, when rotated at speeds resulting in maximum allowable material stress, offers a maximum value for specific energy. Unfortunately, the rotor size necessary to achieve an optimum design, when combined with a guard ring and housing, is too large to fit within the confines of an automobile. Hence, with these space restrictions, a simple, constant-thickness disc profile was selected with a thickness-to-radius ratio of 0.4. (A greater thickness ratio would of course increase the energy stored, but could also result in problems in achieving uniform strength properties).

The choice of material for flywheels has been generally found to fall into two categories: (1) various grades of high-strength nickel steel and (2) reinforced plastics with various reinforcing filamentary materials ranging from glass to graphite. Again, because of the size restrictions in the automobile, the reinforced plastics would require on the order of double the rotor speed of steel for the case of inducing a maximum allowable working stress in the material and achieving equivalent energy density. Therefore, with the speed restrictions mentioned earlier, steel is the logical candidate for a primary system in this study.

An impact barrier ring is a necessity with steel flywheels. Proponents of reinforced plastic flywheels claim that this ring is not necessary because most of the plastic binder and reinforcement parts of the rotor vaporize during a wheel failure (as opposed to release of high-energy pieces with metallic wheels); however, a greater degree of experimental evidence

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*Advanced reinforced plastic flywheel systems operating in excess of 40,000 rpm were briefly examined in the overall analysis.

will be required to allow a system without a barrier ring to be installed in a car. Even with a barrier ring, the safety aspects of the flywheel system design will require the utmost scrutiny and verification.

There are other safety aspects in addition to the apparent ones posed by vehicle impact. These involve the need for precision balancing of the rotating mass so as to minimize critical speed loads on the bearings and avoid premature failure. Material uniformity in the rotor is also essential to avoid imbalances produced by nonuniform stretching of the rotor due to centrifugal forces. Furthermore, failure of the vacuum system must be anticipated and a rapid braking system used to preclude runaway heating of the rotor by air friction. Such friction could lead to rotor failure by thermal shock or by a substantial reduction in the working strength of the material.

Another eventuality to be accounted for in flywheel system design is the sudden release of stored energy to the vehicle drive wheels, causing unexpected vehicle acceleration. This event might be more possible with a mechanical or hydromechanical linkage to the wheels than with an electromechanical linkage (in which failure might occur before excessive power is transferred, due to overheating of the motor windings).

In the present study, the decision was made to transfer power into and out of the flywheel by hydromechanical means. An equally acceptable design would be one relying on motor-generator systems. In fact, such a unit could conceivably be placed within the flywheel housing. With power transferred electrically, there would be no flywheel shaft seals required, and air leakage into the housing could be minimized. The possible increase in weight and volume of such a massive housing would have to be investigated to further evaluate such a concept.

As described previously, with the design approach selected for this study, the flywheel system requires more than just a simple rotor. With the auxiliaries and with the restrictions on installation volume and rotor speed, the overall system results in specific energy less favorable than for batteries. However, the specific power for flywheel systems is much higher and is only restricted by the power transfer devices used between the flywheel and the vehicle drive wheels.

Because of the expected wide range in speed variation between the heat engine and the flywheel and between the flywheel and the vehicle drive wheels, a power transfer system with a wide range in input-output speed capabilities, accompanied by a consistently high level of efficiency, was deemed desirable. Such a system is best represented by the class of continuously variable transmissions (CVTs). A review of various transmissions for flywheel hybrid application was conducted previously (Refs. 2-5, 2-6), and based on near-term availability, low cost, and overall performance, the mechanical CVT and the power-splitting hydromechanical CVT were superior. With the immediate availability of a performance map, the latter system was chosen for this study. The mechanical CVT is an equivalent contender as a power transfer device and has not been eliminated from further consideration. With further development, it may even prove to be a superior unit, particularly for use during regenerative braking of the vehicle because of its inherent ability to readily multiply torque at either input or output shafts.

With the particular input-output speeds found in the transmission performance map for the selected CVT, a mechanical gear system was still needed for speed matching of the engine-flywheel-drive wheel combination. Hence, gears were introduced at the input and output shafts of the CVT that linked the engine to the flywheel and at the input and output shafts of the CVT that linked the flywheel to the vehicle drive wheels (with an interposing standard rear end differential gear train). The proper selection of gear ratio in each case was found to be essential to attainment of acceptable levels of powertrain efficiencies throughout the speed and torque range demanded of each component during vehicle negotiation of a given driving cycle.

2.5 REFERENCES

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2-4. G. L. Dugger, et al., Heat-Engine/Mechanical-Energy-Storage Hybrid Propulsion Systems for Vehicles, Final Report, Johns Hopkins University, Applied Physics Laboratory, Report No. CP011 (March 1972).

2-5. M. A. Cordner and D. H. Grimm, Hybrid Propulsion System Transmission Evaluation, Phase I, Final Report, Sundstrand Aviation Division of Sundstrand Corporation, Report No. AER 640 (25 February 1972).

2-6. Feasibility Analysis of the Transmission for a Flywheel/Heat Engine Hybrid Propulsion System, Mechanical Technology, Inc., Report No. MTI-71 TR 75 (APTD-1181) (November 1971).

TABLE 2-1. POSSIBLE COMPONENTS FOR HEAT ENGINE HYBRID POWERTRAIN

Motors
Alternating current induction Direct current externally excited Direct current series wound Direct current compound wound Torque motors Direct current brushless
Generators
Direct current Alternating current (alternators)
Power Conditioning and Control
Pulse-width modulation Frequency modulation Variable-frequency inverters Cycloconverters Integrated circuits Relays/switches Current limiters Cicruit breakers and fuses Filters (inductor-capacitor) Storage battery system
Heat Engines
Spark ignition Diesel Gas turbine Stirling
Energy Storage Systems
Battery Flywheel Pneumatic Hydraulic Thermal

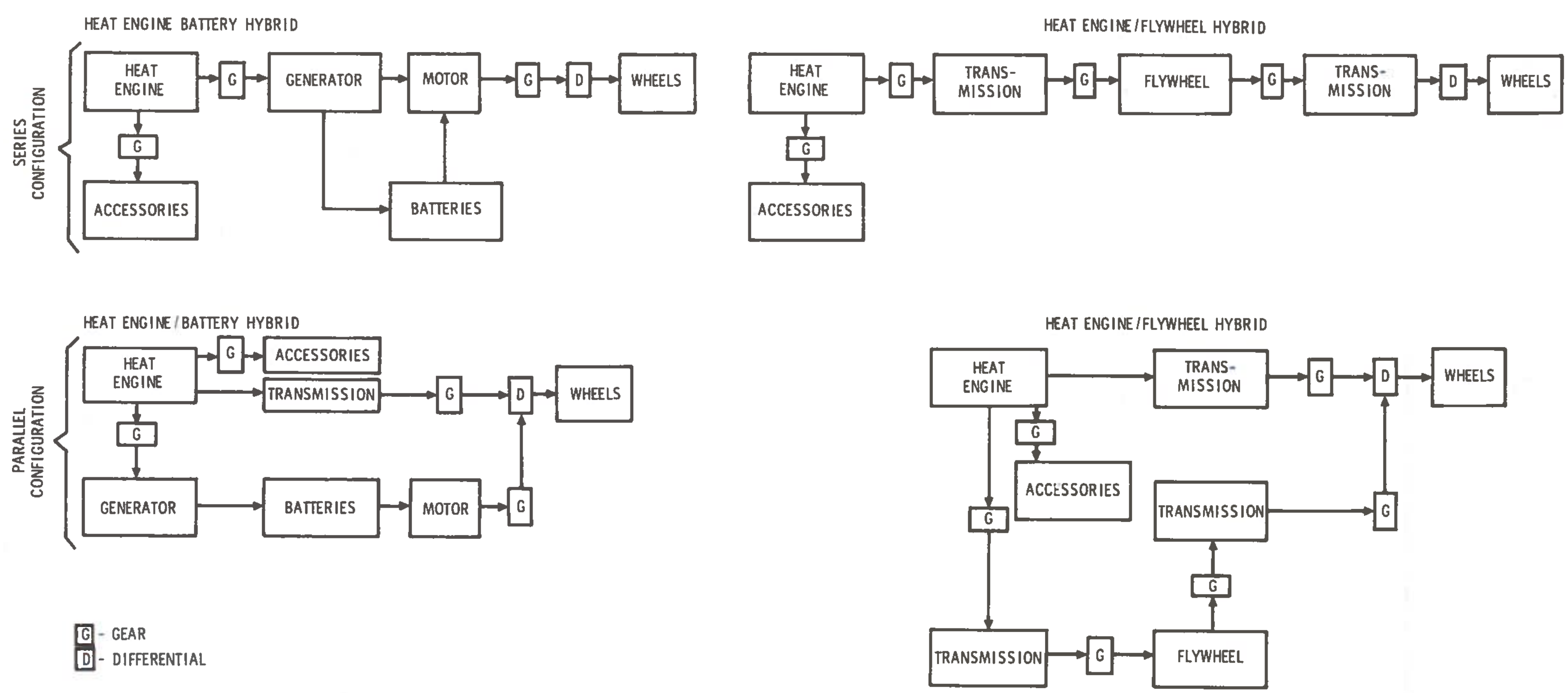

FIGURE 2-1. SIMPLIFIED SCHEMATICS OF HEAT ENGINE HYBRID VEHICLE POWERTRAIN CONCEPTS

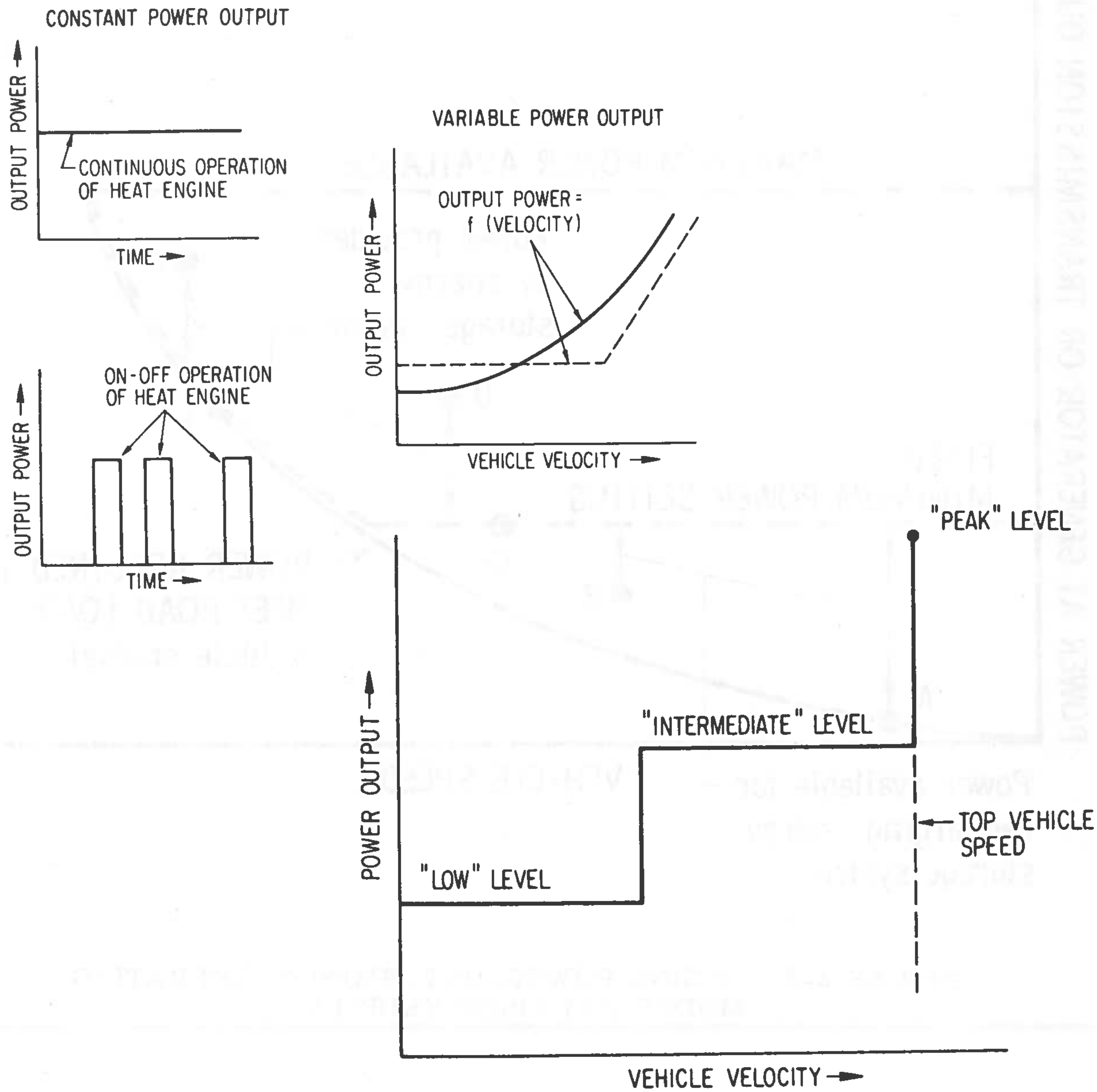

FIGURE 2-2. VARIOUS HEAT ENGINE OPERATIONAL MODES

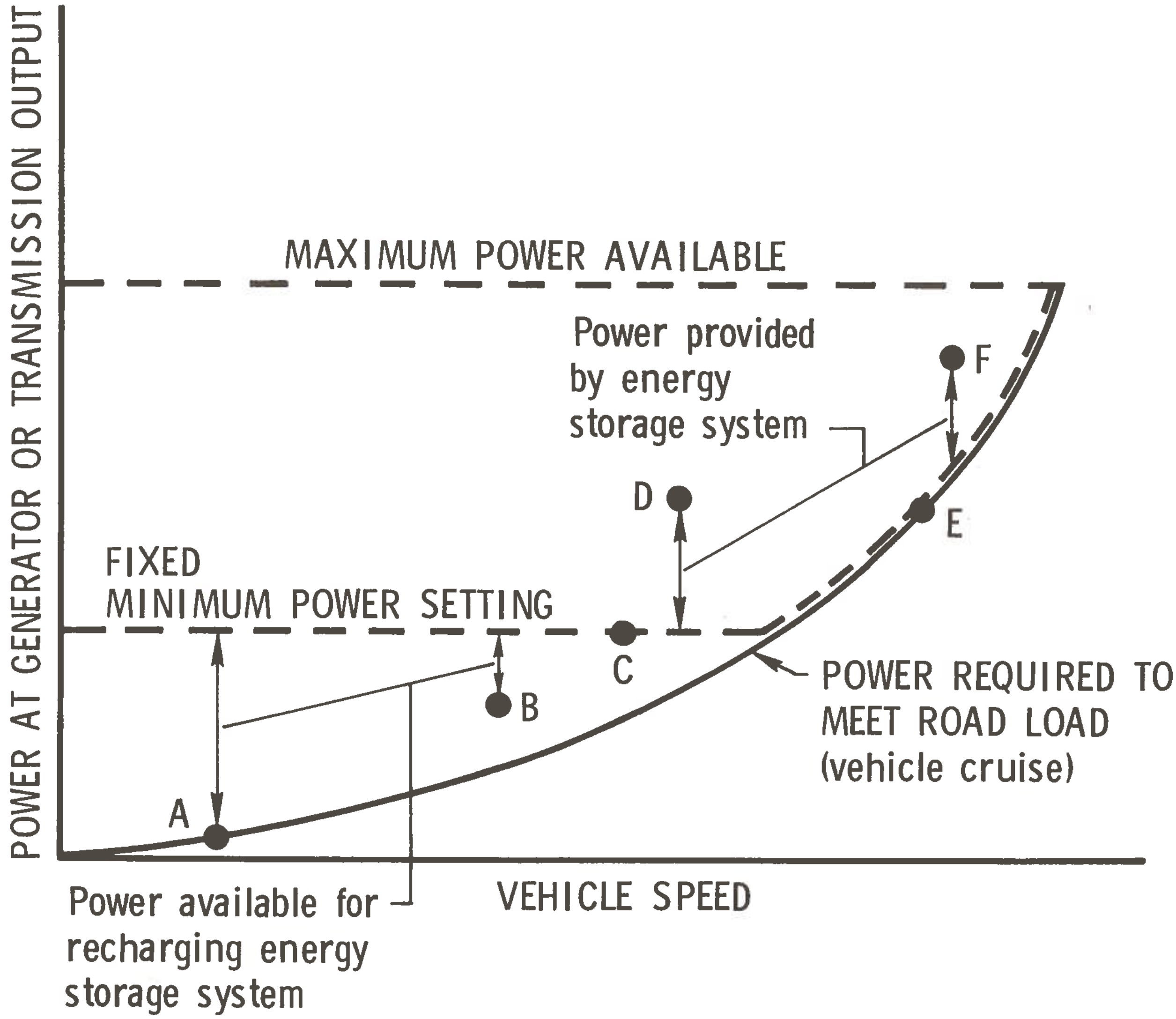

FIGURE 2-3. ENGINE POWER-DETERMINING OPERATING MODES ON HYBRID VEHICLE

3. TECHNOLOGY REVIEW OF HYBRID VEHICLE SYSTEMS

Hybrid vehicle and component performance data from the most significant government or private sponsored investigations, as well as from manufacturers' specification listings, were reviewed, summarized, and assessed. Information published in the open literature constituted the main source of data.

The intent of this review was threefold. First, the hybrid vehicle systems that had been analyzed, designed, and/or tested, were summarized so that different approaches to powertrain design could be understood, screened, and used as background material when formulating the powertrain design for the present study. Second, both systems and components were reviewed in order to ascertain what potential attributes and constraints contributed to specific use in a given application. Third, wherever data were available, system and component performance were examined and summarized for use as data inputs to the hybrid vehicle simulation computer program developed for this study. This section of the report reviews data on vehicle systems and includes a brief review of regenerative braking systems. Section 4 presents a similar technical review of powertrain components.

A synopsis containing a brief description of major components of the powertrain is given for each hybrid automotive vehicle system design that has been examined in recent years. Only battery or flywheel energy storage devices were considered for use in these designs. The majority of designs reviewed have evolved from the work sponsored by the U.S. Environmental Protection Agency (EPA) in the period 1970-1972, and were oriented primarily toward achieving major reductions in exhaust emissions. (Fuel consumption was also a consideration.) Tables 3-1 and 3-2 are tabular summaries of pertinent information for each systems investigation.

3.1 HYBRID HEAT ENGINE/BATTERY SYSTEMS

3.1.1 Petro-Electric Motors Prototype Hybrid Car

Petro-Electric Motors, Ltd., developed a prototype standard-size hybrid car for entry into the Federal Clean Car Incentive Program

(FCCIP), formerly sponsored by EPA (Ref. 3-1). Table 3-3 summarizes the Phase I and Phase II vehicle requirements for the FCCIP multiphase program. In the pre-Phase I portion, a one-dollar contract is negotiated between EPA and a contractor who provides an acceptable program proposal. In Phase I, the prototype vehicle was to be leased to the government for 90 days and tested by EPA for compliance with the requirements of Table 3-3. Under Phase II of the FCCIP, the federal government was to purchase ten demonstration vehicles for extended testing. In Phase III, the final portion of the program, the government would buy up to 500 vehicles for field testing. The Petro-Electric Motors pre-Phase I contract began on 17 May 1971. Their prototype vehicle was submitted for Phase I testing in January of 1974. The Phase I testing by the EPA was conducted from 25 January 1974 to 16 May 1974 and from 8 October 1974 to 25 November 1974.

The prototype vehicle built by Petro-Electric Motors consists of a converted 1972 Buick Skylark (Refs. 3-1 and 3-2). Figure 3-1 shows the location of powertrain components in the car. Figure 3-2 is a block diagram of the parallel configuration powertrain combining a rotary spark ignition engine and a dc electric motor. The motor and engine are tied together and run at identical speeds. The motor/engine is connected to the wheels through a standard transmission and a differential speed gear. Electrical power is supplied to the electric motor from a set of batteries. The flow of power is determined by electronic controls in response to driver accelerator position. Exhaust gas recirculation and a thermal reactor are used to control exhaust emissions. Data on the individual components of the hybrid powertrain are contained in Table 3-1.

The electronic control system is designed to reduce power transients on the heat engine and to operate the engine over a restricted range by sensing the manifold vacuum which is a strong function of engine power. Engine power can be maintained near-constant by supplying additional propulsion power demands of the vehicle from the electric motor. When the propulsion power demand decreases below the level supplied by the engine, the electric motor is operated as a generator (driven by excess engine power) and charges the batteries. The control electronics do not sense battery state-of-charge, which means that engine power in excess of propulsion power demand is wasted whenever the batteries are fully charged. If the driver demand on the powertrain cannot be satisfied by power from the motor and the engine operating at constant manifold vacuum, then an override control is actuated which opens the engine throttle for increased engine power.

EPA tests on the hybrid car are summarized in Table 3-1 under the performance heading. The results show a remarkable reduction in hydrocarbon (HC) and carbon monoxide (CO) exhaust emissions, but with oxides of nitrogen (NO_x) levels still in excess of the original Federal Emission Standards for 1976. Fuel economy was poor compared to conventional cars of equivalent inertia weight.

Conclusions reached by the EPA investigators were as follows:

a. The Petro-Electric prototype demonstrated a capability to achieve the levels of the Phase I requirements for emissions control.

b. The Phase I requirement of 4000 miles of mileage accumulation with a vehicle configuration meeting the emissions requirement was not demonstrated.

c. On the basis of the Phase I test results, many deficiencies in the developmental prototype were identified and addressed by the Contractor in a vehicle improvement package proposal. While EPA considered the solutions to the deficiencies as sound engineering approaches, the technological gap between Phase I performance and Phase II requirement was considered too great to permit EPA extrapolation and prediction of Phase II performance with any degree of confidence.

d. It was recommended that the candidate vehicle not be admitted to Phase II of the FCCIP program.

3.1.2 Mercedes-Benz Hybrid Bus

The Mercedes-Benz OE-302 16 ton, 66-passenger research vehicle is a prototype hybrid diesel electric bus (Ref. 3-3). The purpose of this vehicle is to provide quiet, exhaust-free operation in the inner city. The powertrain selected by Mercedes-Benz is of the series configuration, and is illustrated schematically in Figure 3-3. A driver input from the accelerator pedal is sensed by the box labeled "electronics." This commands the box labeled "controls" to drive the electric traction motor with a series of electric pulses whose width is determined by the driver input. The rpm indicator senses the motor speed and adjusts the frequency of the electrical pulse train. Energy to run the motor is obtained from the 380-volt battery in the energy storage unit. The energy storage unit also contains the power conditioning that is required to run all of the vehicle accessories. The 380-volt battery is recharged both from the regenerative braking path, which is indicated by the dashed arrows, and also by the diesel-driven generator. The diesel engine is operated at constant speed and minimum fuel consumption. The battery, diesel engine, generator, and electric motor are sized so that it is possible to operate the bus for one or two shifts before the batteries have to be completely recharged. Table 3-1 summarizes the performance of these components. Figure 3-4 identifies the location of the powertrain elements in the hybrid diesel-electric bus.

The electric traction motor is connected to the rear axle through a reduction gearbox and a propeller shaft. This combination eliminates the interruption in driving force which is inherent in a manual transmission. Redundant braking is provided by a backup set of dual air brakes. The air brakes only come into operation when the operator sharply depresses the brake pedal. The operator may also turn off the diesel engine for operation in congested metropolitan areas. On intercity runs, the diesel engine/generator unit is turned on to produce power for the traction motor.

3.1.3 TRW Hybrid Vehicle Study and Test

The purpose of TRWs study funded by EPA (Ref. 3-4) was fivefold: first, to select and to analyze the performance of series and parallel heat engine/battery hybrids; second, to define relative efficiencies, weight, and costs of the hybrids; third, to generate data of selected three-component catalysts for control of exhaust emissions; fourth, to develop a hydrocarbon emission trap (accumulator); and fifth, to demonstrate that a hybrid car could meet the then-promulgated 1975-1976 emission goals.

Three series powertrain configurations and one parallel powertrain configuration with electromechanical transmission (EMT) were considered. Using a LA-4 driving cycle, it was determined that one of the series hybrid designs, which included an alternator power control unit, was very inefficient. The TRW group also believed that the second design, a two-generator series hybrid design, would involve a potentially more difficult control scheme due to control stability arguments. The third series scheme, nominally "a full voltage hybrid," was deemed the best of the three series hybrids.

The "full voltage hybrid" series powertrain configuration is shown in Figure 3-5. In this scheme an alternator is directly coupled through fixed gearing to the engine. The rectified voltage and power of the heat engine is delivered to the battery bus where the power can be used either to charge the battery or power the traction motor. The alternator runs at one-third rated speed in urban traffic, and its speed is proportional to road speed in highway operation. A feedback system using the alternator field and voltage is used to keep the engine running at predetermined urban or highway reference values. For highway travel the heat engine puts out a percentage of the power needed. The batteries supply the remainder of the required power.

The parallel powertrain configuration in the urban traffic mode is illustrated in Figure 3-6. The parallel system with EMT consists of a planetary gear, alternator, alternator power control unit (PCU), traction motor, and motor PCU. The engine input power drives the sun gear of the planetary set; the ring gear drives the powertrain propeller shaft, and the planet carrier drives the alternator. The planetary gear algebraically splits the input power of the engine between the ring gear and planet carrier in direct proportion to the speeds of the respective elements. Thus, when the propeller shaft is stopped (car at rest), the planet carrier and alternator turn at maximum speed and all engine power goes into the alternator, which in turn recharges the batteries. As the car accelerates and the propeller shaft speeds up, the planet carrier slows down and less engine-produced energy is absorbed by the generator.

In urban operation the engine runs at constant speed and power. The throttle is preset and the speed is controlled by increasing or decreasing the alternator's load to reflect a balancing torque back on the engine shaft.

During highway operation, the planet carrier and the alternator are locked out and the engine speed increases linearly with road speed. Engine throttle levels are slowly changed by a form of battery feedback loop. The electric motor takes up the rest of the power requirements, much as the series hybrid.

In sizing the components in both systems, the wheel power demands of a 4000-lb car were determined for different driving conditions and speeds. These data were then used to define power requirements of either the series or parallel hybrid system. The components finally chosen for the series and parallel system are listed in Tables 3-4 through 3-7.

The calculated efficiencies for the series configuration varies from 43 to 55 percent and for the parallel configuration from 54 to 76 percent. TRW studies showed that the series configuration was more costly, heavier, and less efficient than the parallel configuration.

The TRW group also experimented with a hydrocarbon accumulator—a device used to control the cold start hydrocarbon emissions from an engine. The accumulator's function is to absorb a large percentage of the hydrocarbons while the catalytic converter is cold. The hydrocarbons are temporarily stored in an absorbing material. When the converter is at its operating temperature, the hydrocarbons are returned to the engine or catalyst for oxidation. Six absorbent materials were tested using the exhaust of one cylinder from a Volkswagen 1600-cc engine as the input. The tests were used to determine which material was best suited and what temperature range was to be employed in the accumulator use. The testing was not suficiently detailed to establish whether or not the absorbent degrades with use. Of the different absorbents tested, an activated carbon (Pittsburgh BPL type) seemed to be the best suited absorbent.

Three catalytic converters were also tested. The first was a TRW design composed of copper oxide on aluminum pellets, the other two converters were Universal Oil Products using platinum coated pellets. The Universal Oil Products' converters differed in size only. The three converters were tested on a Vega engine. The results were about the same except that the TRW design seemed to excel in oxides of nitrogen (NO_x) conversion. It is believed that the main reason for this is due to sizing, as all the converters were of different sizes, which caused differences in the time the exhaust remained in the converter. The high performance of the TRW converter is attributable to its being the largest in size.

A prototype parallel configuration powertrain was built which included the hydrocarbon accumulator and the TRW catalytic converter.

The heat engine was a Chevrolet Vega engine with Volkswagen fuel injection. Only the engine and emission control units were tested. The powertrain was tested on a dynamometer over the LA-4 driving cycle. The emission results are given in Table 3-8. There was no accumulator used in the 4 January 1972 test, resulting in much higher cold-start emissions for hydrocarbons (HC) and carbon monoxide (CO). Note that all cold-start CO emissions are high. The TRW group believes that this is due to poor carburetion and can be remedied by a better designed carburetor system. Also, note that all results except the 4 January 1972 test are below the then-promulgated 1975 emission standards.

None of these hybrids has a scheme to measure the battery state of charge. There is also no scheme for a man in the control loop. This would be warranted for short stop and go trips; the car could then be run as an electric car and recharged by external means at the end of the journey. A summary of the TRW hybrid system is given in Table 3-1.

3.1.4 Minicars, Inc. Hybrid Car

Under EPA funding, Minicars, Inc., investigated the relative reduction in exhaust emissions obtainable from a heat engine/electric hybrid vehicle when compared to the same vehicle powered by a heat engine alone (Ref. 3-5). The vehicle was built and tested in its initial configuration (a heat engine-powered vehicle) as well as in three hybrid configurations. The differences in hybrid designs were in heated engine intake manifold air, heated engine intake manifold legs, a throttle delay mechanism, and 24 or 48 volt electrical systems. The basic hybrid configuration is shown in Figure 3-7. In this system the heat engine is continually charging the batteries and/or powering the car. There are two potential disadvantages of the system design. First, there is no way of running the car in an all-electric mode; in localities of high pollution or high emissions this feature might almost be a necessity. Also, when the batteries are fully charged there is no need for the heat engine to operate. Second, the electric motor/generator draws power from the heat engine when the car is stationary. This is due to back electromotive forces in the motor/generator. To alleviate this problem, it was proposed that the electric motor/generator could be turned off electrically. This modiciation was not made.

A major design contribution is in a throttle delay mechanism. The philosophy behind this design is that heat engines have best emission performance when run at a constant speed. In this hybrid configuration the direct coupling of the heat engine and the drive wheel necessitates heat engine speed variation, but the main objective of hybrid operation is to keep heat engine transients to a minimum. The mechanical system employed is basically a damper on the foot throttle input. The damping alleviates the high frequency variations and allows the low frequency (or slow changes) of the foot throttle to adjust the set point on the carburetor throttle (Figure 3-8).

It is also desirable for the electric motor to produce some of the extra power called for by the foot throttle input. The electric motor control system is therefore attached to the carburetor input as illustrated in Figure 3-9. In summary, the advantage of the throttle control is as follows: The engine provides all the power plus charging power to the storage batteries independent of steady-state driving conditions for steady-state operation on level, uphill, or downhill grades. When the throttle position is changed quickly, the engine power changes very slowly, thereby operating nearly as a steady-state engine. This keeps the air/fuel mixture ratio nearly constant, which in turn should keep the exhaust emission concentration more nearly constant. The increase in power needed to satisfy the throttle setting comes from the electrical system. A possible disadvantage of this throttle delay is when rapid acceleration is needed, such as in an emergency situation; there is then no bypass of the delay circuit or indication of the response from a large step input. If an analysis were to be carried out and it was found that the response (rise time) was too slow, a simple mechanism could be designed to directly couple the foot throttle to the carburetor for large, rapid excursions in power demand.

The battery system in the hybrid designs were varied from 24 to 48 volts. It was found that the electric motor experienced sharp drops in both speed and power output at 24 volts since sufficient power could not be delivered from the electrical system. The best battery system included twelve 12-volt batteries that were connected through a parallel/series relay to provide voltage switching from 24 to 48 volts, and therefore could supply the increase in power desired by the electric motor.

The hybrid vehicle weight was 3200 pounds. The heat engine and electric motor/generator were sized to meet weight, speed, and power requirements, as well as losses from the transmission, rear end drive, and tires. The final design is summarized in Table 3-1.

Emission results are shown in Table 3-9 for each hybrid configuration. All of the results are from dynamometer tests with the heat engine using Indolene-30 gasoline. In general, when an engine operates under retarded ignition timing, the peak combustion temperatures are reduced, thereby reducing the NO_x emissions. In these hybrid configurations, these results were not confirmed.

Minicars concluded that the hybrid powertrain did lower exhaust emissions, especially when the throttle delay is used, but did not reduce emissions to acceptable levels. Spark timing variations did not show any conclusive results; also, HC and CO emissions decreased, but NO_x emissions increased when the air-fuel ratio was increased. It was also found that engine misfiring during steady-state operation increased HC emission up to tenfold, while CO remained about the same and NO_x increased by approximately 50 percent.

Minicars also concluded that 48-volt, heavy duty, lead-acid batteries were found acceptable, the shunt dc motor/generator with armature and field control performed satisfactorily and the electric machine used was capable of supplying peak requirements.

Minicars recommended more experimentation with heat engines capable of higher air-fuel ratios and a more complete examination of the throttle delay mechanism, both in mechanical design and delay times. It was also stated that (1) the motor/generator should be resized, based on a more general driving cycle and (2) a stabilized shunt motor should be tried with an eye to increasing the torque-producing capabilities in the low-speed range without the need of over excitation.

3.1.5 General Motors Corporation Stir-Lec I Hybrid Car

In 1968, General Motors Corporation designed and built a Stirling engine/battery hybrid (Ref. 3-6). The Stirling engine was investigated for possible use as a future automotive power unit with low exhaust emissions.

Stirling engines are capable of burning a wide variety of liquid fuels (unleaded gas, diesel fuel, and kerosene) at large air-fuel ratios. The continuous nature of the external combustion process should make for more complete combustion than an internal combustion heat engine. Thus, it was expected that the Stirling engine would emit fewer pollutants. Also, due to the design of this engine, it is inherently balanced and therefore quieter than an internal combustion engine.

The prototype vehicle consisted of a converted 1968 Opel Kadett (Figure 3-10). The powertrain employs a series configuration (Figure 3-11). The Stirling engine drives a three-phase alternator, the output of which is rectified to charge the batteries. The controller included a modulated inverter to provide variable frequency and voltage to a three-phase induction motor coupled to a differential pinion shaft. The engine was operated without any emission control devices.

The battery pack consists of 14 automotive grade lead-acid batteries connected in series. The depth of discharge was limited to 75 percent to increase battery life.

With two passengers, the car weighs 3200 pounds. The heat engine alone can propel the vehicle at 30 mph on level ground with fuel economy of 30 to 40 mpg. The vehicle is capable of accelerating from zero to 30 mph in 10 seconds with a top speed of 55 mph.

At speeds over 30 mph the batteries supply the excess energy; below 30 mph, they are recharged. At 55 mph the vehicle range is 30-40 miles. In the all-electric mode the range is from 15-30 miles depending on driving style.

The vehicle curb weight is 2930 pounds as opposed to the Opel's 1990 pounds. The powertrain weight of 1189 pounds is distributed between the inverter, the control electronics, the motor and gearbox, the battery charger and alternator, the battery pack and the Stirling engine (see Table 3-10). A vehicle specification summary is provided in Table 3-1.

The major disadvantage of the system is the large powertrain to total weight ratio coupled with the low road performance (0-30 mph acceleration time).

In Figure 3-12 the vehicle emissions data are shown both with and without preheating of burner inlet air to 1200°F. The HC and CO emissions are low, but the NO_x emissions are much higher than expected for this engine. General Motors stated that engine modifications could be expected to provide reductions in the NO_x levels.

3.1.6 The Aerospace Corporation Hybrid Vehicle Study

The study was funded by EPA and was oriented toward determining the feasibility of using a hybrid heat engine/electric propulsion system as a means of reducing exhaust emissions from street-operated vehicles (Ref. 3-7). These vehicles are categorized as commuter cars, family cars, high- and low-speed vans, as well as high- and low-speed buses.

Study guidelines for vehicle operating strategy and road performance were:

a. The hybrid vehicles would match the road performance of conventional vehicles.

b. The vehicles would not require external recharging of the batteries.

c. The batteries would discharge only when the vehicle was undergoing acceleration.

d. The heat engine would supply road load power and not be required to undergo rapid acceleration.

e. Only design concepts compatible with near-term (1972-1975) prototype vehicle development were to be considered.

f. All other operating strategies were left variable.

A computer program was written to compare different strategies and components in terms of vehicle emissions for different driving cycles. The vehicle design was based on meeting specified acceleration and sustained speed-on-grade requirements; its fuel economy and emissions were evaluated over a federal emissions test driving cycle. From the prescribed vehicle road performance, power requirements were calculated as a function of velocity. Assuming a tire size and a powertrain efficiency, motor torque (as a function of motor speed) and powertrain gear ratios were determined. The performance of a baseline configuration for both classes of vehicles was established for each type of heat engine and design strategy. Since a specific detailed design was not a goal (rather a design feasibility), the effect on vehicle emissions was determined by varying parameters, such as component efficiencies, regenerative braking, vehicle weight, powertrain weight, types of driving cycle and types of battery. Because of the number of vehicle classes and engines considered, the vehicle powertrain weight was held fixed for a given vehicle curb weight.

A summary of the conclusions drawn from the computer simulation is listed below:

a. For the performance specified, only spark ignition internal combustion engines and gas turbines can be packaged into the vehicle.

b. The hybrids show marked calculated emission reductions over the then-current conventional vehicles.

c. Based on available technology, no version of the 4000-lb family car could meet the promulgated 1975-76 emission standards (no catalytic converters or thermal reactors were employed).

d. Based on advanced technology, the commuter car could meet the then-promulgated 1975-76 emission standards, but the spark ignition family car could not meet NO_x standards (the diesel family car does have some potential with engine improvements).

e. For a 3600-lb family car with reduced performance, the parallel powertrain configuration with advanced technology could meet the then promulgated 1975-76 standards.

f. Parallel powertrain configurations are 10 to 15 percent lighter than series configurations, but have a more complex design.

g. An improved lead-acid battery with increased power density capabilities and longer lifetime is needed. Nickel-zinc batteries look promising in the near future.

h. Regenerative braking has little effect on emissions due to poor battery charge acceptance. For regenerative braking to be effective, the batteries should be able to accept around 400 amps at 95 percent state-of-charge.

i. Vehicle weight increase of several hundred pounds has little effect on emissions, but causes reduced road performance.

j. Battery recharge efficiency (realistically varied) has little effect on emissions.

k. Fuel consumption over the 1972 DHEW driving cycle* are:

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*U.S. Department of Health, Education, and Welfare Cycle, forerunner of EPA Federal Emissions Test Cycle.

Vehicle	Series Configuration (mpg)	Parallel Configuration (mpg)
Commuter Car	26	30.5
Family Car	11	12.5

The air-fuel ratio (A/F) used is 14 to 16, and it is assumed that horsepower does not degrade at the higher A/F ratios used in the emission calculations. If degradation occurs, then as much as 20 percent reduction in fuel economy can be expected.

l. An approximate cost analysis follows, but should be viewed with caution due to the necessity in the analysis of assigning production costs to components still in the design phase:

Vehicle	Relative Cost
Current Conventional Car	1
Hybrid Car:	
Spark ignition	1.4 - 1.6
Diesel	1.5 - 1.7
Gas turbine	1.6
Rankine	2+
Stirling	2.25 +

m. A comparison of DHEW and New York driving cycles effect on emissions showed the New York cycle resulted in a major increase in vehicle emissions. The parallel system seemed a little more sensitive to the change than the series (49 to 55 percent increase versus 45 to 54 percent increase) and it had less emissions than the series in both driving cycles. The increased emissions of the New York cycle apparently are the result of more accelerations, more decelerations, and more time stopped, which poses a greater energy demand per mile from the powerplant.

3.1.7 University of Wisconsin Hybrid Vehicle Study

The motivation for this work was the Urban Vehicle Design Competition held at General Motors Proving Grounds in 1972. The principal design emphasis of the Urban Vehicle Design Competition was improved fuel economy in urban driving situations.

The sizing was initiated by assuming a 3000-lb car capable of a 3.0 g initial acceleration (Refs. 3-8, 3-9, 3-10). A parallel powertrain with a 53 hp Wankel engine as the prime mover, an 18 hp motor/generator, and a 450-lb lead-acid battery pack of 36 volts was chosen. A special transmission design allows operation of the powertrain in all three modes:

a. All-electric mode
b. Hybrid mode (drive and charge)
c. Parallel or power mode

An all-electric control system regulates the excitation and armature voltage of the electric motor/generator. The operation of the internal combustion engine and the motor/generator are controlled in accordance with speed and road characteristics, as well as driver input.

The powertrain is the most unique part of the design. Figure 3-13 illustrates the mechanical arrangement of the powertrain. The power is combined in the Volkswagen transaxle. When clutch 2 is engaged there is no differential action, that is, the transmission acts as a simple shaft. Only when clutch 2 is disengaged does the differential operate. This occurs only in the hybrid mode, thus increasing the unit's life expectancy.

3.1.7.1 <u>All-Electric Mode</u>

Clutch 1 is disengaged to isolate the heat engine from the rest of the powertrain. Clutch 2 is engaged, the electric motor drives the transmission (as a simple shaft), which in turn drives the rear axle. In regenerative braking the reverse occurs, in which case the motor acts as a generator.

3.1.7.2 <u>Hybrid Mode</u>

Clutch 1 is engaged and clutch 2 is disengaged. In this mode the differential operates, bringing into effect its torque and power balance relationships. The power needed to propel the car goes through a direct mechanical path from the heat engine to the wheels. The remainder of the power drives the generator, which in turn recharges the battery. The generator rotates at twice the engine speed when the car is motionless. This speed ratio decreases, due to differential action, as the car increases its speed. The heat engine thus operates at its peak efficiency in this mode.

The car speed is controlled by field excitation of the generator. The greater the field, the more torque required to turn the generator, and in turn, more torque is made available by differential action to propel the car. In this way the generator/motor is run in the same torque range in either of its operational modes resulting in higher efficiency. Battery recharging, which occurs only in mode two, varies invesely with car speed.

3.1.7.3 Parallel Mode

Both clutches are engaged and the transmission acts as a simple shaft. This mode is capable of high speed and power. The heat engine's total power is available for propelling the car. The electric motor adds additional power as needed.

3.1.7.4 Performance

No test results were available. It was believed that the car would travel 5 to 10 miles in mode 1. The batteries could then be recharged externally or by mode 2 operation. It was felt that it would take 20 percent more time in mode 2 to recharge the batteries than the mode 1 operation time. The cost projection is a price increase of 10 to 20 percent over the conventional model year car. Factors included in the cost analysis were: (1) the transmission needs only two gears because the motor can be reversed, and (2) there is no need for a heat engine starter motor because the electric motor can be used. Emission levels would be equal to a Wankel engine running at its most efficient condition if the car were in mode 2. In mode 1, there is virtually no emissions, while in mode 3 the emissions are a little worse than mode 2. A summary of emissions for the LA-4 driving cycle* is shown in Table 3-11 with no emission controls or devices. These data are simulated results based on assumed system efficiencies and regenerative braking. A summary of the hybrid system is given in Table 3-1.

*Forerunner of EPA Federal Emissions Test Cycle

3.2 HYBRID HEAT ENGINE/FLYWHEEL SYSTEMS

3.2.1 Lockheed Hybrid Vehicle Study

The Lockheed Missiles and Space Company (LMSC) undertook a two-part investigation of inertial energy storage (flywheel) systems for EPA. One part of the investigation (Ref. 3-11) had as its objective the analytical determination of the feasibility of the flywheel hybrid as a low emission propulsion system for urban vehicles (family car, commuter car, delivery van, and intracity bus) and the demonstration and performance evaluation of full-scale flywheels for hybrid applications. The other part of the investigation (Ref. 3-12) was directed toward advancing the development of flywheel systems technology including the development of designs of flywheel auxiliary equipment (housing, bearings, seals, etc.). It included the experimental demonstration of positive energy containment in burst tests of flywheels, safety analyses, the use of heat engine emission data provided by the U.S. Bureau of Mines (Ref. 3-13) for analysis of hybrid vehicle emissions, and systems coordination for the transmission studies conducted by Mechanical Technology, In. (Ref. 3-14), and Sundstrand (Ref. 3-15). This section covers the vehicle system; the flywheel subsystem is discussed in Section 4.4. Table 3-2 summarizes the family car system.

The drivetrain selected by LMSC (Figure 3-14) consists of a heat engine coupled to a flywheel and the vehicle wheels through a continuously variable planetary/hydrostatic power splitting transmission.

In the LMSC system, the heat engine provides cruise power, drivetrain losses, accessory power, and flywheel recharging power. An internal combustion, spark ignition engine was selected for operation under variable speed (2:1) and load conditions.

The steel flywheel provides the power required for vehicle acceleration, and it was sized on the basis of matching the vehicle kinetic energy at maximum speed. Flywheel charging is accomplished by the heat engine and by regenerative braking during deceleration. The preliminary configuration selected by LMSC (Figure 3-15) was 20.4 inches in diameter. This flywheel configuration was fabricated and tested during the feasibility study. Subsequent analysis showed, however, that incorporation of the burst containment structure dictated a reduction in flywheel diameter to about 13 inches in order to meet space limitations in the vehicle. The final baseline flywheel configuration, shown in Figure 3-16, had a design energy storage capacity of 0.5 kW-hr at a design speed of 24,000 rpm. The weight of the rotor was 86 pounds, which constitutes approximately 46 percent of the complete flywheel assembly weight (185 pounds).

Two flywheel drive transmission configurations (Figure 3-17) were considered by Lockheed. The double transmission (series configuration) was preferred to the single transmission (parallel configuration) because it offered greater control flexibility by allowing the heat engine speed to be controlled independently of either flywheel speed or vehicle speed.

A comparison of several types of family car transmissions studied by Lockheed in both the single and double configuration is presented in Table 3-12. The efficiency is calculated for operation over the DHEW Urban Dynamometer Driving Schedule, with the assumption that all braking is regenerative. The best transmission on the basis of highest efficiency and lowest cost was the power splitting transmission. Powertrain comparisons of the power splitting transmissions, with each of four different heat engines as summarized by Lockheed is given in Table 3-13.

As part of the Flywheel Drive Systems Study (Ref. 3-12) Lockheed calculated fuel consumption on the basis of an EPA-supplied brake specific fuel consumption (BSFC) map for a medium-sized V-8 engine and EPA-furnished accessory power loads. Results of computer runs over the DHEW Urban Driving Schedule showed that average fuel economy ranged from 7.3 to 13.7 mpg for the hybrid engine/flywheel vehicle, depending on assumed values for transmission efficiency and the operating regime on the BSFC map. Comparable figures for the conventional passenger car with automatic transmission ranged from 11 to 12 mpg, depending on the assumed transmission efficiency.

Emission calculations made by Lockheed were based on emissions data provided by the U.S. Bureau of Mines (Ref. 3-13). These data were taken on two 350-CID engines at various engine speeds, percent load, air-fuel ratios, spark advance, exhaust gas recirculation (EGR) rates and with and without an oxidizing catalyst.

Results of the computer simulations over the DHEW driving cycle are shown in Table 3-14 for both a conventional three-speed automatic transmission and a flywheel/hybrid vehicle. Results are predicated on both vehicles being equipped with an oxidizing catalyst and with EGR. (Cold start effects are not included because of insufficient data). The hybrid vehicle only showed significant reductions in emission for oxides of nitrogen when compared to the conventional car, but failed to meet the then-promulgated Federal Standards for 1976.

3.2.2 Johns Hopkins University Hybrid Vehicle Study

An experimental and analytical study of high specific energy flywheel systems was conducted by Johns Hopkins University, Applied Physics Laboratory (JHU), for EPA (Ref. 3-16). This study had two objectives: (1) proof of principal demonstration of the use of filamentary or composite materials of high uniaxial tensile strength that would have high specific energy (i.e., 30 W-hr/lb) and (2) theoretical evaluation of the performance of such flywheels alone and in hybrid combination with heat engines, in four classes of vehicles: family car, commuter car, delivery van, and intracity bus.

Vehicle evaluation studies by Johns Hopkins indicated that the heat engine/flywheel hybrid system would satisfy performance requirements for all four classes. Using a flywheel-only propulsion system, the city bus was the only one of the four classes of vehicles that could meet performance requirements.

The general layout of a hybrid propulsion system selected by Johns Hopkins for a commuter car is shown in Figure 3-18. A series configuration powertrain was envisaged by JHU with the heat engine mounted in the front of the car and the flywheel-transmission system mounted in the rear. This arrangement provides space for a disc flywheel with an energy storage capability of 2 kW-hr or a bar flywheel with about 1.5 kW-hr capability. The use of a system of this capacity will depend on the ability to contain the rotor, acceptable vehicle handling characteristics, and the development of low friction bearings and seals. The heat engine would be operated in an on-off mode in supplying energy to the flywheel. All drive-wheel power would be supplied by the flywheel in the proposed series configuration. A continuously variable transmission of the power-splitting type was felt by Johns Hopkins to best satisfy the requirements for a hybrid vehicle.

The drive train schematic is shown in Figure 3-19. The power required for accessories is taken from the central gearbox and an input shaft is provided for externally supplying power to the flywheel in the event of rundown or while the vehicle is parked. Regenerative charging can also be accomplished.

The analytical results for the family car and commuter car are summarized in Tables 3-15 and 3-16. The heat engine was sized to permit continuous cruise at the design speed with all accessories operating. Note that the flywheel subsystem weight (which includes a protective housing in addition to the flywheel) is significantly greater than the weight of the flywheel alone. The family car system is summarized in Table 3-2.

3.2.3 University of Wisconsin Hybrid Automobile — Design and Simulation

An automobile, based on a 3000-lb chassis and equipped with a powerplant incorporating a high-speed energy-storage flywheel, has been analyzed and designed by the University of Wisconsin under contract to the U.S. Department of Transportation (Ref. 3-17). Construction of the automobile is underway with demonstration tests scheduled for the middle of 1976 to measure fuel consumption and exhaust emissions (Ref. 3-18). Design and fabrication of the vehicle has been augmented by computer simulation studies of fuel economy and emissions.

A pictorial representation of the powertrain configuration is shown in Figure 3-20, and the proposed installation in the car is shown in Figure 3-21. A reciprocating piston, spark ignition, gasoline engine, calibrated for minimum emissions, is connected through a clutch to the flywheel. When the flywheel speed drops below a predetermined value, the engine is turned on and run at full throttle for maximum efficiency. The engine is shut off when the flywheel reaches a maximum design speed. The vehicle system is summarized in Table 3-2.

A four-speed manual shift transmission is used in combination with a hydrostatic power-split, continuously-variable transmission (CVT) to allow for proper matching of the flywheel. Power is transferred partly through a hydrostatic transmission (pump and motor) and partly through a mechanical gear train. The system is designed to absorb regenerative braking energy during vehicle deceleration.

Basic specifications for the flywheel are:

a. Usable energy storage of 2/3 hp-hr (about 0.5 kW-hr)
b. Maximum windage loss of 1 hp
c. Overspeed protection
d. Locked bearing protection
e. Alloy steel construction
f. 250 ft-lb torque capability

Principal features of the CVT are:

a. A ratio range of 3.5:1
b. Torque control
c. 400 ft-lb torque capability
d. Designed for 80 mph maximum vehicle speed

Vehicle acceleration and regenerative braking are controlled through the CVT by varying the hydrostatic pressure. Although a production vehicle would have automatic controls, the demonstration vehicle will be manually controlled.

A computer simulation was developed by the University of Wisconsin to predict the fuel economy and other performance characteristics of the flywheel vehicle. Based on 1976 emission standards, Table 3-17 shows comparative predictions for three different types of 3000-lb vehicles and includes a breakdown of energy disbursement in each case. A potential improvement of 58 percent over the conventional car is shown by the near-term flywheel car. This would appear to be due primarily to the claimed ability of operating the engine at a brake specific fuel consumption (BSFC) of 0.50 lb/hp-hr. Also assisting in reduction of energy disbursement is (1) the ability to shut the engine off when the flywheel is not being recharged and (2) the use of regenerative braking.

An examination of the calculated energy losses shows the CVT to have the greatest loss. The arrows designate those components that the University of Wisconsin believes can be significantly improved in efficiency.

3.2.4 Technical School at Aachen, West Germany Hybrid Van — Design and Test

Development of this hybrid drive system with flywheel energy storage has been sponsored by the West German Federal Ministry of Research and Technology since 1973. The ability to recover energy during vehicle braking and to operate the heat engine at improved efficiency account for the reduction in fuel consumption when compared to a conventional drive train (Ref. 3-19).

A schematic of the parallel configuration drive train is shown in Figure 3-22. The major components are an electric-motor/generator, a heat engine, a differential gear train, flywheel primary energy storage system, and a battery secondary energy storage system. Modulation of the speed and torque of the motor/generator controls the torque and speed of the drive shaft leading to the wheels during vehicle motion, as well as the energy recharging of the battery and flywheel when the vehicle is stationary. Power from the battery is used for producing the necessary motor torque and speed. Power from the flywheel is used to augment heat engine power for vehicle acceleration. The system is summarized in Table 3-2.

The aforementioned components were tested in the laboratory and on the road. A photograph of the assembled main drive unit is shown in Figure 3-23. The electric motor is flange-connected to the left-hand side and a Wankel rotary engine to the right-hand side with the differential gear in the middle. The flywheel housing is shown above the differential with the flywheel shaft oriented in a vertical direction.

A 2100 kilogram Volkswagen van was used for road tests of the installed hybrid drive train (Figure 3-24). The vehicle top speed is 70 km/hr; the system start-up time is about 20 seconds and the heat engine normally runs at about 3500 rpm (acceleration of the vehicle has been improved above that shown in Table 3-2 according to Ref. 3-20).

Road test results are shown in Figure 3-25 where fuel consumption is plotted versus the dynamic factor (a term developed to correlate the energy requirements of various types of driving cycles) for a conventional van and the hybrid van. Noted on the plot are dynamic factors corresponding to various driving cycles. The hybrid system shows fuel consumption reductions of about 40 percent (with as much as 45 percent at the large dynamic factors). As compared with conventional vehicle engine efficiencies ranging from 10.5 to 14 percent, the hybrid vehicle engine efficiencies were about 19 to 20 percent. Between 10 and 30 percent of the energy available for recovery during dynamic braking was actually recovered. For the case of 30 percent energy recovery, a 45 percent reduction in fuel consumption was achieved. Of this reduction, about 31 percent was attributed to recuperation of braking energy and 69 percent to improved operating efficiency of the heat engine.

Additional development work on system optimization is expected to lead to reductions of 50 percent in fuel consumption for a wide range in driving cycles.

3.3 REGENERATIVE BRAKING SYSTEMS

The ability to recover kinetic energy during vehicle deceleration, store it, and then reuse it for vehicle propulsion could possibly provide significant reductions in vehicle energy consumption. Of major importance for a given weight vehicle is the manner in which vehicle deceleration occurs and the efficiency for energy conversion in the vehicle powertrain. Vehicle deceleration of course varies with driver habits, type of roadway, and traffic conditions.

The maximum possible energy that could be extracted for recovery would be the vehicle kinetic energy, but this would require instantaneous deceleration. In actuality, vehicle aerodynamic drag and road load absorb a significant portion of vehicle kinetic energy during deceleration (the longer the period of deceleration, the more energy absorbed and the less available for recovery). Therefore, the maximum possible energy that could be extracted for recovery is the energy dissipated in the vehicle brakes. In the succeeding discussion both analytical and experimental determinations of regenerative braking energy are briefly reviewed.

In a generalized, parametric study (Ref. 3-21), the percentage of energy supplied to the vehicle drive axle that is available for recovery was calculated for a series of vehicle weights and for three types of driving. The figures for a 4200-lb car, for example, were 32.7 percent for city driving, 10.5 percent for urban driving, and 2.2 percent for highway driving. This is the maximum available for a 100 percent efficient powertrain. For a 75 percent efficient powertrain, these figures dropped to 18.4, 5.9, and 1.2 percent, respectively (75 percent efficiency from drive wheels to energy storage unit and 75 percent efficiency from energy storage unit back to drive wheels). Calculations for other vehicle weights produced similar results. The differences are merely due to the variations in aerodynamic and tire drag loads for each class of vehicle. This effect produces small variations in the ratio of road load to inertial energy needs on a given driving cycle.

Reference 3-22 showed by calculation that, for a 4000-lb vehicle operating on the EPA Urban Driving Cycle, 27 percent of the drive axle energy was the maximum possible available for recovery. For the EPA Highway Driving Cycle, the figure was only about 2.5 percent.

An all-electric vehicle operating on a metropolitan city-type driving cycle was examined in Ref. 3-23. Considering all the powertrain losses and the charge acceptance of the battery, a figure of 18 percent was calculated as being the best improvement possible in energy consumption at the vehicle drive wheels. By assuming that the battery will generally be recharged before experiencing large depth-of-discharge (thereby further limiting charge acceptance), a value of only 10 percent was calculated for energy consumption improvement.

Regenerative braking effects for a hybrid heat engine/battery system evaluation was reported in Ref. 3-4 for both series and parallel configurations. For a 4000-lb family car operated on the LA-4 Driving Cycle, it was calculated that without regenerative braking, the system efficiency from the engine output shaft to the drive wheels is reduced by 10 percent for the series configuration and by 13.5 percent for the parallel configuration. A 33.1 percent maximum possible reduction was calculated for a 100 percent efficient powertrain.

In Ref. 3-24, a regenerative braking system was evaluated during actual road tests on a converted Austin sedan, weighing 1950 pounds, and having in the powertrain a dc motor, a 4-speed transmission, and a fuel cell/battery energy storage system. The results showed that the gain did not warrant the additional weight (60 pounds) and circuit complications of the regenerative system. It was recommended instead that the weight of the storage batteries be increased by 60 pounds. (The implication here is that the "gain" referred to was operational range and not energy expenditure.)

With a combined propulsion mode and regenerative mode efficiency of 38.9 percent, Ref. 3-25 showed a possible energy gain of 12.5 percent for the 512 Electric Vehicle on a city driving schedule. These results assumed that the recharge system could be designed to use 100 percent of the energy available at all car speeds and that no mechanical braking would be required. With a practical system, these assumptions were felt not to be valid and, therefore, less than 12.5 percent could be expected to be gained. The conclusion was that the system complexity required to obtain regenerative braking did not warrant its use in the vehicle.

The Mercedes-Benz hybrid-electric bus discussed in Ref. 3-3, used regenerative braking to augment the battery energy storage system. A major portion of the battery energy provided in the driving cycle was for vehicle acceleration. Therefore, the potential existed for a significant portion of vehicle kinetic energy to be returned to the battery. Indeed, a figure derived from the data presented showed that about 25 percent of the total energy expended by the battery was regenerated and returned to the wheels.

Reference 3-23 noted that flywheel systems can accept regenerative energy at all speeds up to the limit of their storage capacity, and therefore should derive improved energy savings compared with battery systems. As configured in the present study, however, the transmission probably is the weak link in the system. Operating the CVT in reverse would imply more reliance on the inefficient hydraulic elements for power delivery to the flywheel. Of course, an additional transmission could be considered and clutched in to the flywheel shaft during the regeneration phase.

For a 4000-lb family car operating on the DHEW city driving cycle, Ref. 3-16 initially determined that 55 percent of hybrid vehicle kinetic energy could be recovered at the flywheel during decelerations, or 15.8 percent of the total energy expended on the cycle. These figures were reduced to 50 percent and 14.2 percent, respectively, for final system performance calculations. Using the assumed value of 72 percent for flywheel-to-vehicle drive wheels power transmission efficiency, the energy savings at the wheels would be about 10 percent. This value is compatible with the savings percentage determined by other investigators for battery systems.

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TABLE 3-1. PROTOTYPE HYBRID VEHICLE SYSTEMS: HEAT ENGINE/BATTERY SYSTEM

Organization	Petro-Electric, Ltd.	Mercedes-Benz
Objectives and Goals	Develop standard-size prototype car which meets FCCIP requirements	Develop a hybrid diesel electric bus
Powertrain Elements		
Configuration	parallel	series
Heat Engine	continuous operation, variable power, Mazda rotary, 70 cu in, 9.4:1 compression ratio, 4-barrel carburetor, 130 hp max @ 7000 rpm, 115 ft-lb max torque @ 4000 rpm, weight 273 lb	OM 314 diesel engine, 232 cu in, 4-cylinder, 65 hp @ 2200 rpm, operated for range extension of vehicle or during highway driving
Emission Control System	thermal reactor and EGR	INP[a]
Electric Traction Motor	dc shunt, separately excited motor, 120 volts, 115 amps continuous or 600 amps surge, 20 hp continuous rating, 60 hp max @ 5500 rpm, 190 ft-lb max torque, weight 240 lb	dc shunt, separately excited motor, 120 hp continuous rating, 201 hp max, 4800 rpm max motor speed, motor weight with driving gear, 1330 lb
Electric Power Conditioning and Control	INP[a]	combined pulse width and pulse frequency modulation max pulse frequency, 250 Hz, min pulse duration, 1 ms, max current, 600 amps
Electric Power Generator	INP[a]	3-phase generator with rectifier
Batteries	8 Gould, 12-volt, lead-acid batteries; 48 or 96 volts; 90 amp hours at 10-hr rate; 600 amps max current; weight, 300 lb	supplied by VARTA; rated voltage, 380 volts; 5-hour capacity discharge, 275 A-hr; storage capacity 104 kW-hr, weight, 3.86 tons
Transmission	manual 1973 Vega - 1st gear ratio, 3:1 2nd gear ratio, 1.85:1 3rd gear ratio, 1:1	none
Differential	gear ratio, 5:1	single reduction gear between electric traction motor & wheels

[a] Information not provided in references

TABLE 3-1. CONTINUED

Organization	Petro-Electric, Ltd. (continued)	Mercedes-Benz (continued)
Performance		
Emissions		
HC	0.38 gm/mi, 40 mi, EPA test	INP[a]
CO	2.42 gm/mi, 40 mi, EPA test	
NO_x	0.76 gm/mi, 40 mi, EPA test	
Acceleratior	0-60 mph, 17.5 sec @ 4950 lb	≤ 2.25 mph/sec from stop due to standees
Fuel Economy	8.75 mi/gal, EPA Urban Cycle	INP[a]
Noise	79 dB (A) max	INP[a]
Max Speed	INP[a]	43.5 mph
Gradeability	INP[a]	11% @ 13.7 mph
Range	INP[a]	34.2 mi with stops every 0.25 mi on batteries only. All-day operation in hybrid mode.
Payload	INP[a]	66 - 110 passengers

[a] Information not provided in references

TABLE 3-1. CONTINUED

Organization	TRW Systems, Inc.	
Objectives and Goals	(1) Select and analyze performance of hybrid vehicle designs. (2) Define relative weights and costs. (3) Acquire data on catalysts. (4) Develop an accumulator. (5) Meet 75/76 emission goals.	
Powertrain Elements		
Configuration	parallel	series
Heat Engine	continuous operation, variable power, 71 Chevrolet Vega, 2300 cc (140 CID), 90 hp (max), aluminum block	INP[a]
Emission Control System	hydrocarbon accumulator of activated carbon catalytic converter with copper oxide on aluminum pellets	INP[a]
Electric Traction Motor	series dc, 30 hp at 7200 rpm, 180 terminal volts, 145 amps at 22.5 ft-lb continuous duty, 9-inch diameter frame GE BT 2448, 15-1/2 inches long, 150 lb	series dc, 65 hp at 4650 rpm, 235 terminal volts, 235 amps at 74 ft-lb continuous duty, 12-1/2-inch diameter frame GE CD 280/2508, 21 inches long, 325 lb
Electric Power Conditioning and Control	200 Vdc rectifier	240 Vdc rectifier
Electric Power Generator	salient pole alternator with sliprings, 10 kW @ 12,000 rpm, 95% efficient, rpm range 1200-12,000, 3-phase continuous duty, 40 lb, 8-in. diameter, 4 in. long, 400 Hz base speed	salient pole alternator with sliprings, 58 kW @ 4000 rpm, 95% efficient, rpm range 4000-12,000, 3-phase, continuous duty, 160 lb, 10-in. diameter, 19 in. long, 400 Hz, top speed
Batteries	INP[a]	INP[a]
Transmission	electromechanical transmission; 2:1, 0 to 42.5 mph; 1.5:1, 42.5 to 55 mph; 1:1, 55 to 85 mph; planetary gear train	INP[a]
Differential	INP[a]	INP[a]

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[a]Information not provided in references

TABLE 3-1. CONTINUED

Organization	TRW Systems, Inc. (continued)	
Performance		
Emissions	EPA Urban Cycle, gm/mi	INP[a]
	Cold / Hot	
HC	2.84 / 0.29	
CO	46.8 / 3.26	
NO_x	3.84 / 0.32	
Acceleration	designed for 440 ft in 10 sec	INP[a]
Fuel Economy	INP[a]	INP[a]
Noise	INP[a]	INP[a]
Max Speed	85 mph calculated at dry weight	85 mph calculated at dry weight
Gradeability	INP[a]	INP[a]
Range	INP[a]	INP[a]
Payload	INP[a]	INP[a]

[a]Information not provided in references

TABLE 3-1. CONTINUED

Organization	Minicars, Inc.	General Motors Corp.
Objectives and Goals	Determine relative reduction in exhaust emissions obtainable from a heat engine/battery hybrid car compared to same vehicle powered by heat engine alone.	Basic in-house research to explore use of low emission Stirling engine in a hybrid car.
Powertrain Elements		
Configuration	series hybrid, elements in line	series, 1189 pounds
Heat Engine	continuous operation, variable power Corvair engine, 6-cylinder opposed, clockwise rotation, 164 cu in displacement, 8:1 compression ratio, single venturi carburetor with idle, main, and power jets, and an accelerator pump, heat air intake and manifold legs	continuous operation, fixed power, Stirling engine (GPU-3), converted Army design with hydraulic controls, single-cylinder, 8 hp at 3000 rpm, hydrogen working fluid at 1000 psi using combustion air blower
Emission Control System	INP[a]	INP[a]
Electric Traction Motor	Lear-Siegler G22-3, 24V, 300 amp, 9.7 hp rated, shunt motor-generator, 94 lb, 2000-6500 rpm range	AC induction motor, 3-phase, 24 Vdc, 20 hp over 3:1 speed ratio
Electric Power Conditioning and Control	modulated with both shunt field control and variable armature voltage, throttle delay mechanism	variable frequency and voltage, all solid state, modulating inverter frequency and amplitude control
Electric Power Generator	(see Electric Traction Motor)	3-phase alternator, 19 kV nominal, 5500 rpm
Batteries	24 Vdc or 48 Vdc controlled by parallel-series relay (based on throttle depression) 12-12 volt batteries in different parallel-series configurations	14 series-connected lead-acid SLI batteries, 44 amp-hr at 20-hr discharge rate, 6.6 kW-hr
Transmission	automatic transmission, 2:1 stall ratio at 1400 rpm, 1.82:1 low and reverse, 1:1 high gear	planetary gear set
Differential	3.57 axle gear ratio,	3.45:1.0

[a]Information not provided in references

TABLE 3-1. CONTINUED

Organization	Minicars, Inc. (continued)	General Motors Corp. (continued)
Performance		
Emissions	DHEW Cycle, constant volume sampling (engine air-fuel ratio set at 16.5 to 1.0)	with 25 to 1 air-fuel ratio, and 1200°F heated combustion air
HC	3.15 gm/mi	HC (c6) 0.03 gm/hp-hr
CO	29.6 gm/mi	CO 0.5 gm/hp-hr
NO_x	1.0 gm/mi	NO_x 3.3 gm/hp-hr
Acceleration	0-60 mph, 23.2 sec on 3000-lb car (5.64 mph/sec peak); 32.1 sec on 4000-lb car (4.20 mph/sec peak)	0 - 30 mph in 10 sec on 3200-lb car (weight includes 2 passengers)
Fuel Economy	internal combustion engine: 14.5 mpg at 15 mph, 10.3 mpg at 30 mph, 12.6 mpg at 50 mph; hybrid: 11.8 mpg at 15 mph, 8.8 mpg at 30 mph, 12.4 mpg at 50 mph; no all-electric mode possible	30-40 mpg at 30 mph, engine-only operation
Noise	INP[a]	INP[a]
Max. Speed	75 mph	55 mph with heat engine and batteries, 30 mph with heat engine alone
Gradeability	INP[a]	INP[a]
Range	≈ 200 miles on heat engine, believed 2-5 miles on batteries only	heat engine: 30-40 miles at 55 mph; electric power only: 15-30 miles at 30 mph
Payload	INP[a]	INP[a]

[a]Information not provided in references

TABLE 3-1. CONTINUED

Organization	The Aerospace Corporation	
Objectives and Goals	(1) Establish through computerized analysis and hardware data the design feasibility and potential for major reductions in exhaust emissions through use of hybrid heat engine/battery vehicles. (2) Establish general design goals for components & subsystems in the vehicle powertrain.	
Powertrain Elements		
Configuration	series	parallel
Heat Engine	continuous operation, variable power, 93 hp spark ignition engine operating at best brake specific fuel consumption of 0.5 and weighing 335 lb, for powering 4000-lb full-size hybrid family car	continuous operation, variable power, 84 hp spark ignition engine operating at best brake specific fuel consumption of 0.5 and weighing 319 lb for powering 4000-lb full-size hybrid family car
Emission Control System	lean carburetion (A/F=22), oxidizing catalyst, exhaust gas recirculation	lean carburetion (A/F=22), oxidizing catalyst, exhaust gas recirculation
Electric Traction Motor	forced-air cooled, 8000 rpm, 64 hp, dc, shunt-wound with step voltage and field control weighing 337 pounds. 90% peak efficiency, 80% average efficiency	forced-air cooled, 8000 rpm, 35 HP, DC, shunt-wound with step voltage and field control weighing 250 lb. 90% peak efficiency, 80% average efficiency and capable of 3:1 short-term overload
Electric Power Conditioning and Control	step voltage augmented with field control and armature current sensing, 12.5 lb motor controller, 18 lb ac rectifier, 3 lb generator controller, 99.5% control system efficiency	step voltage augmented with field control and armature current sensing, 12.5 lb motor controller, 9 lb ac rectifier, 2 lb generator controller, 99.5% control system efficiency
Electric Power Generation	12,000 rpm alternator, rated at 51 kW, weighing 80 lb with rated efficiency of 90% and average efficiency of 80%	12,000 rpm alternator rated at 7 kw weighing 18 lb with rated efficiency of 90% and average efficiency of 80%
Batteries	38 amp-hr lead-acid or nickel-zinc with capacity of 8.36 kW-hr, weighing 398 lb and required to deliver 92.5 kW, 10 shallow charge-discharge cycles (3.5% of capacity max) per vehicle mi.	38 amp-hr lead-acid or nickel-zinc, with capacity of 8.36 kW-hr weighing 460 lb and required to deliver 92.5 kW, 10 shallow charge-discharge cycles (3.5% of capacity max) per vehicle mi.

TABLE 3-1. CONTINUED

Organization	The Aerospace Corporation (continued)	
Powertrain Elements (cont.)		
Transmission	transmission not necessary	modified conventional 3-speed automatic, rated at 64 hp; 90% efficiency, weighing 59 lb
Differential	95% efficiency, total rear axle drive weighing 80 lb	95% efficiency; total rear axle drive weighing 80 lb
Performance		
Emissions	Calculated for DHEW[b] Urban Driving Cycle	Calculated for DHEW[b] Urban Driving Cycle
HC	0.361 gm/mi	0.323 gm/mi
CO	0.494 gm/mi	0.442 gm/mi
NO_x	0.504 gm/mi	0.451 gm/mi
Acceleration	0-60 mph in 13 sec, peak acceleration of 5 mph/sec	0-60 mph in 13 sec, peak acceleration of 5 mph/sec
Fuel Economy	calculated 11 mpg over DHEW Urban Driving Cycle	calculated 12.5 mpg over DHEW Urban Driving Cycle
Noise	INP[a]	INP[a]
Max Speed	80 mph	80 mph
Gradeability	40 mph on 12% grade for 8 miles	40 mph on 12% grade for 8 miles
Range	200 miles	200 miles
Payload	min. of 300 lb, passengers and luggage	min. of 300 lb, passengers and luggage

[a] Information not provided in references
[b] U.S. Department of Health, Education & Welfare, forerunner of EPA cycle.

TABLE 3-1. CONTINUED

Organization	University of Wisconsin	
Objectives and Goals	Urban Vehicle Design Competition - 1972 - High fuel economy, low emissions	
Powertrain Elements		
Configuration	parallel	
Heat Engine	continuous operation, variable power, Wankel, 53 hp (powertrain also functions in other modes with engine off)	
Emission Control System	INP[a]	
Electric Traction Motor	dc, continuous rating of 18 hp	
Electric Power Conditioning and Control	manual control	
Electric Power Generator	INP[a]	
Batteries	lead-acid, run between 50% and 90% full charge, 450 lb, 36 volts	
Transmission	1:1 gear ratio	
Differential	gear ratio 1:1 to 1:2 depending on drive mode	
Performance		
Emissions	simulated LA4-1370 sec driving cycle, 60% efficient powertrain, no emission controls or devices	
HC	0.559 gm/mi	
CO	27.7 gm/mi	
NO_x	1.26 gm/mi	
Acceleration	0.3 g (11 ft/s^2)	
Fuel Economy	simulated LA4-1370 sec, 60% efficient powertrain, 21.6 mpg	

[a]Information not provided in references

TABLE 3-1. CONTINUED

Organization	University of Wisconsin (continued)	
Performance (cont.)		
Noise	INP[a]	
Max Speed	INP[a]	
Gradeability	INP[a]	
Range	Mode 1, 5-10 miles (simulated); modes 2, 3 INP[a] (probably limited by gas tank size only)	
Payload	INP[a]	

[a]Information not provided in references

TABLE 3-2. PROTOTYPE HYBRID VEHICLE SYSTEMS: HEAT ENGINE/FLYWHEEL SYSTEM

Organization	Lockheed Missiles and Space Company	Johns Hopkins University, Applied Physics Laboratory
Objectives and Goals	(1)Determine through analysis, feasibility of flywheel system as a low-emission propulsion system. (2)Demonstrate and evaluate performance of full-size flywheels.	Conduct proof-of-principal tests of the "superflywheel" concept and evaluate through analysis the use of such flywheels to reduce automotive emissions.
<u>Powertrain Elements</u>		
Configuration	series	series
Heat Engine	continuous operation, variable power, medium-size V-8, 350 CID engine characteristics scaled from 176 hp for full-size 4300-lb car	on-off operation, 94 hp, 357 lb spark ignition engine for 4300-lb car, characteristics derived from literature sources
Emission Control System	exhaust recirculation and Engelhard oxidizing catalyst	INP[a]
Transmission	hydrostatic power-splitting, 238-lb Sundstrand Version 8C	hydrostatic power-splitting
Control System	INP[a]	INP[a]
Flywheel	tapered steel disk with rim flange, 6.52-inch radius, 24,000 rpm, 86 lb weight for rotor only; usable energy storage of 0.5 hp-hr (max of 1.0 hp-hr), shaft mounted in-line with engine, total flywheel system weight of 187 lb.	reinforced plastic composite, 5.3-inch-thick bar with spin diameter of 24 inches, 32,000 rpm, 163-lb rotor weight, 7.1 hp-hr energy storage, 3.5 min recharge time, system weight of 255 lb.
Differential	INP[a]	INP[a]
Batteries	not applicable	not applicable
<u>Performance</u>		
Emissions	calculated for 4300-lb car over EPA Urban Cycle, hot start	calculated for 4300-lb car over EPA Urban Cycle, hot start
HC	0.378 gm/mi	0.127 gm/mi
CO	1.12 gm/mi	1.97 gm/mi
NO_x	1.21 gm/mi	0.692 gm/mi
Acceleration	b	b

[a]Information not provided in references

[b]Specified in <u>Vehicle Design Goals-Six Passenger Automobile</u>, EPA Advanced Automotive Power Systems Program, Revision C, 28 May 1971

TABLE 3-2. CONTINUED

Organization	Lockheed Missiles and Space Company (cont.)	Johns-Hopkins University, APL (cont.)
Performance (cont.)		
Fuel Economy	calculated 10 mpg over EPA Urban Driving Cycle	calculated 14.4 mpg over EPA Urban Driving Cycle
Max Speed	b	b
Gradeability	b	b
Range	[b]200 mi	[b]200 mi
Payload	[b]1700 lb max	[b]1700 lb max

[a]Information not provided in references
[b]Specified in Vehicle Design Goals - Six Passenger Automobile, EPA Advanced Automotive Power Systems Program, Revision C, 28 May 1971)

TABLE 3-2. CONTINUED

Organization	University of Wisconsin	Technical School at Aachen, West Germany
Objectives and Goals	(1)Study & evaluate methods for improving fuel utilization efficiency in autos emphasizing advanced powerplant concepts, (2)Complete a demonstration flywheel vehicle and test its fuel consumption and emission characteristics.	Development and demonstration of vehicle powertrain designed to reduce fuel consumption in urban traffic.
Powertrain Elements		
Configuration	parallel	parallel
Heat Engine	2.3 liter spark ignition engine operating on-off at wide-open throttle and calibrated for low emissions and fuel consumption. Clutch connects engine to flywheel for recharging.	continuous operation, Wankel rotary, max power of 15 kW, max speed of 523 rad/sec, weight of 28 kg, operates normally at 366 rad/sec
Emission Control System	INP[a]	INP[a]
Transmission	four-speed manual - in conjunction with University of Wisconsin designed CVT hydrostatic power splitting unit with 400 ft-lb capability and 3.5 to 1.0 speed ratio range.	INP (Direct mechanical link to vehicle drive wheels via differential controlled by electric motor torque.)
Control System	hydrostatic system linked to CVT transmission and sensitive to position of accelerator pedal and use of brake pedal.	INP (Electric motor torque and speed are mechanical input to control of drive shaft output torque and speed.)
Flywheel	AiResearch steel design with 2/3 hp-hr energy storage and 250 ft-lb torque capability	moment of inertia of 0.621 kg-m^2, max speed of 1832 rad/sec, weight of 50 kg
Differential	INP[a]	INP[a]
Batteries	not applicable	Energy content of 4.1 kW-hr weight of 150 kg
Electric Motor		power output of 11 kw, max speed of 701 rad/sec, weight of 65 kg

[a]Information not provided in references

TABLE 3-2. CONTINUED

Organization	University of Wisconsin (continued)	Technical School at Aachen, West Germany (continued)
Performance		
Emissions	designed to meet model year 1976 emission standards of HC = 1.5 gm/mi CO = 15.0 gm/mi NO_x = 3.1 gm/mi	INP[a]
Acceleration	INP[a]	max acceleration of 1.2 m/sec^2, max deceleration of 1.4 m/sec^2, reaches 50 km/hr in 10.8 sec[b] (with flywheel operating, up to 100 kW can be delivered to drive shaft)
Fuel Economy	calculated 38 mpg for 3000-lb car	0.11 ℓ/km (reduced fuel consumption up to 45% compared to conventional vehicle)
Max Speed	80 mph	70 km/hr[b]
Gradeability	INP[a]	INP[a]
Range	INP[a]	INP[a]
Payload	INP[a]	INP[a]

[a]Information not provided in references

[b]By road test for 2100 kg vehicle; test cycle not specified

TABLE 3-3. SUMMARY OF FCCIP PHASE I AND PHASE II VEHICLE REQUIREMENTS (Ref. 3-1)

Requirement	Phase I	Phase II
Emissions		
Hydrocarbons	0.41 gm/mi for 4000 mi	0.41 gm/mi for 16,000 mi
Carbon Monoxide	3.4 gm/mi for 4000 mi	3.4 gm/mi for 16,000 mi
Oxides of Nitrogen	1.0 gm/mi for 4000 mi	0.4 gm/mi for 16,000 mi
Evaporative Control	2.0 gm/test	2.0 gm/test
Safety		
Federal Motor Vehicle Safety Standards	Compliance required	Compliance required
Inherent Safety	Analysis	Analysis
Performance		
Startup Self-sustaining idle	Within 30 sec at 60°F	Within 25 sec at 40°F
Start Without Aids	a	At 0°F after 72-hr soak
Acceleration, 0-60 mph	16 sec	In 14.5 sec
Acceleration, 25-70 mph	a	17 sec
Top Speed - Level Grade	75 mph for 1 mile	80 mph for 1 mile
Speed up 5% Grade	a	60 mph
Range-Level Grade	200 mi at 50 mph or 150 mi at 65 mph	200 mi at 60 mph or 175 mi at 70 mph
DOT High-Speed Pass	a	17 sec
Fuel Economy, EPA Urban Cycles FTP	a	10 mpg
Reliability	a	Not degraded from contemporary conventional vehicles
Serviceability		
Power Plant	Enclosed in vehicle	Enclosed in vehicle - volume <35 ft^3
Passengers/Comfort	4/heater	5/heater + air conditioning
Controls	Functionally similar to conventional vehicles, single driver operation	Functionally similar to conventional vehicles, single driver operation
Shifting	Simple, interlocked	Simple, interlocked
Roadability	Equivalent to contemporary vehicles	Equivalent to contemporary vehicles
Engine Braking	Required	Required
Curb Weight	a	4700 lb
Fuel Availability		
Availability/Year	2.5 M vehicle mi/yr	250 M vehicle mi/yr
Filler Opening	Accessible, suitably marked	Accessible, suitably marked
Noise		
Maximum	80 dB(A) at 50 ft	77 dB(A) at 50 ft
Low Speed	a	65 dB(A) at 30 mph
Idle	a	63 dB(A) at 10 ft
Interior	a	78 dB(A) at 50 mph
Maintenance Cost	a	Similar to contemporary vehicles

[a] No requirement

TABLE 3-4. TRW SERIES HYBRID TRACTION MOTOR (Ref. 3-4)

Motor Type	Series dc
Horsepower (hp)	65 at rated speed
Rated Speed (rpm)	4650
Terminal Voltage (volts)	235
Input Current (amps)	235 at rated torque
Rated Torque (lb-ft)	74
Duty	Continuous
Frame	Similar to GE CD 280/250B
Diameter (inches)	12-1/2
Length, Less Shaft (inches)	21
Approximate Weight (lb)	325
Ambient Temperature, Operating (°C)	-30 to +50
Overspeed (rpm)	150% of rated speed
Brush Life	50,000 car miles between changes
Cooling Requirement, Forced Ambient Air	200 cfm at 1.6 in. H_2O, 50°C maximum

TABLE 3-5. TRW PARALLEL HYBRID TRACTION MOTOR (Ref. 3-4)

Motor Type	Series dc
Horsepower (hp)	30 at rated speed
Rated Speed (rpm)	7200
Terminal Voltage (volts)	180
Input Current (amps)	145 at rated torque
Rated Torque (lb-ft)	22.5
Duty	Continuous
Frame	Similar to GE BT 2348
Diameter (inches)	9
Length, Less Shaft (inches)	15-1/2
Approximate Weight (lb)	150
Ambient Temperature, Operating (°C)	-30 to +50
Overspeed (rpm)	150% of rated speed
Brush Life	50,000 car miles between changes
Cooling Requirement, Forced Ambient Air	120 cfm at 1.0 in. H_2O, 50°C maximum

TABLE 3-6. TRW FULL VOLTAGE SERIES SYSTEM ALTERNATOR-RECTIFIER CHARACTERISTICS (Ref. 3-4)

Alternator Type	Salient pole with slip rings
Rating (kW) at Top Speed	58 (X3)[a]
Base Speed (rpm)	4000
Power Factor	0.95
Rectifier Voltage (nominal)-Vdc	240
Speed Range at Nominal Voltage (rpm)	4000 to 12,000
Overspeed (mechanical)	150% of rated speed
Number of Phases	3
Frequency at Top Speed (Hz)	400
Duty	Continuous
Ambient Temperature, Operating (°C)	-30 to +50
Weight (lb)	160
Diameter, Approximately (inches)	10.0
Length, Approximately (inches)	19.0
Brush Life, Between Changes	50,000 car miles
Cooling Requirement with Specified Load Schedule	Integral fan

[a] 58 kW rating is available at base speed, top speed rating is available but never used.

TABLE 3-7. TRW PARALLEL SYSTEM ALTERNATOR-RECTIFIER CHARACTERISTICS (Ref. 3-4)

Alternator Type	Salient pole with slip rings
Rating at Base Speed (kW)	10
Base Speed (rpm)	12,000
Top Speed	Equal to base speed
Power Factor	0.95
Maximum Rectified Voltage (volts)	200 at base speed and rated load
Speed Range (rpm)	1200 to 12,000
Overspeed (mechanical)	150% of base speed
Number of Phases	3
Frequency at Base Speed (Hz)	400
Duty	Continuous
Ambient Temperature, Operating (°C)	-30 to +50
Weight (lb)	40
Diameter, Approximately (inches)	8
Length, Approximately (inches)	4
Brush Life, Between Changes	50,000 car miles
Cooling Requirement, Forced Ambient Air	100 cfm at 1.0 in. H_2O
Suggested Field Voltage (vdc)	200

TABLE 3-8. TRW FULL SCALE HYBRID SYSTEM EMISSION TEST RESULTS (Ref. 3-4)

Test Date	Cold Start Bag (gm)			Hot Start Bag (gm)			Stabilized Bag (gm)			Emissions (gm/mi)			Remarks
	HC	CO	NO_x [a]	HC	CO	NO_x [a]	HC	CO	NO_x [a]	HC	CO	NO_x [a]	
12/22/71	1.63	55.3	0.52	0.24	1.08	0.16	0.77	14.7	0.12	0.21	5.22	0.06	Poor hot start excessive enrichment
12/27/71	1.25	55.3	0.87	0.25	2.20	0.54	0.38	4.3	0.21	0.14	3.92	0.12	
12/28/71	1.72	59.5	0.28	1.27	22.1	0.54	0.41	4.41	0.33	0.25	5.65	0.10	Too slow choke relief
12/30/71	1.68	41.9	3.22	0.40	6.60	0.21	0.38	4.3	0.32	0.18	3.47	0.24	
1/3/72	2.56	49.6	3.88	0.40	2.66	0.44	0.90	9.54	0.43	0.30	4.33	0.31	Very poor hot start
1/4/72	17.5	45.2	0.66	0.21	2.43	2.66	0.95	6.95	0.50	1.15	3.69	0.31	No hydrocarbon accumulator
1/6/72	2.84	46.8	3.48	0.84	1.63	0.56	0.50	2.64	0.58	0.29	3.20	0.32	No detectable NH_3

[a] NO_x as NO_2

TABLE 3-9. MINICARS HYBRID VEHICLE EXHAUST EMISSIONS TESTS: 15-30-50 MPH CRUISES (Ref. 3-5)

	Velocity (mph)	ICE Power (hp/whls)	Fuel (mpg)	Engine (rpm)	Manifold (Hg VAC)	A/F (Ratio)	HC (ppm)	CO (%)	NO (ppm)
Phase I Tests									
ICE[a]	15	1	13.7	1550	11	15/1	101	0.9	459
	30	3	17.8	1625	7	15/1	73	1.0	972
	50	8	16.0	2650	9.5	15/1	112	1.0	3383
Hybrid B	15	1[b]	9.6	1525	9	15/1	54	0.4	766
	30	6.6	13.4	1650	6.5	15/1	91	1.8	1051
	50	14	9.1	2650	3.5	12.5/1	177	7.5	445
Phase II Tests									
ICE	15	1	16.5	1500	12	15/1	167	1.0	534
	30	3	21.6	1650	7.5	15/1	110	2.0	1117
	50	8	17.1	2700	9	15/1	132	1.8	3143
Hybrid C	15	0[b]	8.6	1500	12	15/1	133	0.2	376
	30	6.6	11.2	1600	3	12.5/1	240	7.7	310
	50	14	9.8	2650	2.5	12.5/1	216	7.8	462
Phase III Tests									
ICE	15	1	14.5	1530	13.5	15.5/1	188	1.5	468
	30	4	10.3	2900	13.5	12/1	297	9.2	336
	50	16	12.6	2640	8	12.5/1	230	8.6	427
Hybrid C-1	15	— —	11.8	1570	7.5	16/1	145	0.2	127
	30	— —	8.8	2960	7.5	15.5/1	177	0.2	1056
	50	— —	12.4	2650	6	15/1	512	3.5	1492

[a]ICE = internal combustion engine

[b]Generator not charging, but in motoring mode, therefore requiring very little power from ICE.

TABLE 3-10. GENERAL MOTORS STIR-LEC I POWERTRAIN WEIGHTS (Ref. 3-6)

Inverter (air cooled)	82 lb
Control Electronics	24 lb
Motor and Gearbox	85 lb
Battery Charger and Alternator	58 lb
Battery Pack	490 lb
Stirling Engine	450 lb
Total	1189 lb

TABLE 3-11. MILEAGE, CO, HC, AND NO_x VALUES FOR UNIVERSITY OF WISCONSIN HYBRID VEHICLE (Ref. 3-9)

Vehicle	Generator/Battery Motor Efficiency (η_e)	Weight (lb)	Driving Cycle	Distance Travelled (mi)	Mileage (mpg)	CO (gm/mi)	HC (gm/mi)	NO_x (gm/mi)
UW Hybrid (generator disabled)	—	2900	LA4 1370 s	6.71	22.5	33.2	1.07	1.07
UW Hybrid	100%	2900	LA4 1370 s	6.94	25.5	23.3	0.473	1.06
UW Hybrid	60%	2900	LA4 1370 s	6.94	21.6	27.7	0.559	1.26
UW Hybrid	25%	2900	LA4 1370 s	6.94	18.22	32.8	0.663	1.49
UW Hybrid	0%	2900	LA4 1370 s	6.94	15.8	37.8	0.765	1.72
Wankel-Powered Galaxie (1964)	—	~3500	California (9 × 7) Cold Start	—	—	53.4	10.7	2.53

TABLE 3-12. LOCKHEED FAMILY CAR TRANSMISSION COMPARISON (Ref. 3-11)

Item	Electric		Hydrostatic		Power Splitting	
	Single	Double	Single	Double	Single	Double
DHEW Schedule Efficiency (Normalized)	0.588	0.369	0.745	0.626	1.000	0.835
Volume (ft^3)	3.4	4.0	3.3	3.7	3.7	5.1
Weight (lb)	488	536	280	289	311	391
Cost ($)	641	689	403	469	261	341

TABLE 3-13. LOCKHEED FAMILY CAR POWERTRAIN COMPARISON (Ref. 3-11)

Item	1970 Family Car	Engine Plus Power Splitting Transmission			
		Spark Ignition	Diesel	Turbine	Rankine
Volume (ft^3)	16	21.0	24.2	8.5	37.2
Weight (lb)	817	1,034	1,390	672	1,492
Cost ($)	958	1,050	1,771	5,749	1,476

TABLE 3-14. LOCKHEED VEHICLE EXHAUST EMISSION COMPARISON (Ref. 3-12)

Drive System[a]	Calculated Hot Start Emissions (gm/mi)		
	HC	CO	NO_x
Conventional Three-Speed Automatic Transmission	0.39	0.95	3.98
Hybrid Heat Engine/ Flywheel Vehicle	0.38	1.12	1.21

[a]4300-lb family car spark ignition engine with oxidation catalyst and EGR. Engine specific emissions from U.S. Bureau of Mines data.

TABLE 3-15. JOHNS HOPKINS UNIVERSITY RESULTS FOR FAMILY CAR WITHOUT AIR CONDITIONER OPERATING (Ref. 3-16)

Performance Parameters	Flywheel Only	Hybrids		
		Gas Turbine	Otto	Steam
Gross Engine Horsepower	NA	91.0(200.0)[a]	94.0	91.0
$W_{fw}/W_{(fw+e)}$	1.0	0.62(0.52)	0.31	0.27
Flywheel Rotor Weight, W_R, lb	584	353(389)	163	138
$W_{fw} - W_R$, lb	239	159(143)	92	85
Fuel Economy at V_{max}, mpg	NA	9.2(11.8)	11.8	10.0
Flywheel Only Range at 40 mi/hr, mi	42.3	25.6(21.0)	12.2	10.0
Flywheel Only Range on Grade, mi	5.5	3.3(2.7)	1.5	1.3
No. of Accelerations per Flywheel Charge	49	30(24)	13	11
Flywheel Charge Time, minutes	NA	7.7(2.9)	3.5	3.0
Cycle Performance				
Fuel Economy, mpg	NA	11.3(14.4)	14.4	12.2
HC Emission Ratio[b]	NA	0.02(0.01)	0.31	0.46
NO_x Emission Ratio[b]	NA	1.97(1.97)	1.73	0.88
CO Emission Ratio[b]	NA	0.20(0.15)	0.58	0.57
Flywheel Cycles per 100 Miles	4.5	7.4(9.0)	16.0	18.9
Flywheel Only Range, mi	22.4	13.5(11.1)	6.2	5.3
Percentage of Time Engine On	NA	21.1(9.6)	21.1	21.1

[a]Numbers in parenthesis are for a gas turbine engine sized for same fuel economy as Otto engine.

[b]Emissions ratioed to the original 1976 Federal emission standards.

NA = Not Applicable

W_{fw} = weight of flywheel subsystem, lb

$W_{(fw+e)}$ = weight of flywheel subsystem plus engine, lb

W_R = weight of rotor, lb

TABLE 3-16. JOHNS HOPKINS UNIVERSITY RESULTS FOR COMMUTER CAR WITHOUT AIR CONDITIONER OPERATING (Ref. 3-16)

Performance Parameters	Flywheel Only	Hybrids		
		Gas Turbine	Otto	Steam
Gross Engine Horsepower	NA	32.2(109)[a]	33.2	32.2
$W_{fw}/W_{(fw+e)}$	1.0	0.53(0.33)	0.41	0.25
Flywheel Rotor Weight, W_R, lb	217	103(64)	74	35
$W_{fw} - W_R$, lb	112	72(45)	61	49
Fuel Economy at V_{max}, mpg	NA	10.9(25.5)	25.5	17.8
Flywheel Only Range at 40 mi/hr, mi	38.3	18.5(11.5)	13.1	6.2
Flywheel Only Range on Grade, mi	4.9	2.3(1.4)	1.6	0.8
No. of Accelerations per Flywheel Charge	42	20(12)	14	6
Flywheel Charge Time, minutes	NA	6.4(1.2)	4.6	2.2
Cycle Performance				
Fuel Economy, mpg	NA	13.0(30.6)	30.6	21.2
HC Emission Ratio[b]	NA	0.01(0.01)	0.11	0.20
NO_x Emission Ratio	NA	0.76(0.76)	0.67	0.34
CO Emission Ratio	NA	0.10(0.08)	0.22	0.24
Flywheel Cycles per 100 Miles	4.7	9.9(15.8)	13.8	28.9
Flywheel Only Range, mi	21.5	10.2(6.3)	7.3	3.5
Percentage of Time Engine On	NA	23.1(6.8)	23.1	23.1

[a]Numbers in parentheses are for a gas turbine engine sized for same fuel economy as Otto engine.
[b]Emissions ratioed to the original 1976 Federal emission standards.

NA = Not Applicable

W_{fw} = weight of flywheel subsystem, lb

$W_{(fw+e)}$ = weight of flywheel subsystem plus engine, lb

W_R = weight of rotor, lb

TABLE 3-17. EPA-CVS CYCLE ENERGY COMPARISON BY UNIVERSITY OF WISCONSIN (Ref. 3-17)

Items	Standard 1976 Vehicle with 4-Speed Transmission and 2.3 Liter Engine (hp-sec)	1976 Flywheel Hybrid Vehicle with 2.3 Liter Engine (hp-sec)		Future Flywheel Hybrid Potential from Continued R&D (hp-sec)
Road Load	3700		3702	3702
Rear Axle	470		536	536
Transmission	648		648	200[b]
Deceleration and Brakes	2555	Flywheel	855	400
		CVT	1809	900[b]
Total (+) Work	7373	FW Gears	172	172
		Charge Pump	634	100
		Excess Brakes	50	50
		Engine Clutch	99	99
		Engine Inertia	93	93
Idle and Coast Fuel 0.25 lb	≈1111[a]	Engine Start	66	66
Total Work	8484		8664	6318
Fuel for (+) Work	1.655 lb	1.202 lb		0.876 lb
Fuel Total	1.905 lb	1.202 lb		0.876 lb
(+) BSFC	0.808 lb/hp-hr	0.50 lb/hp-hr		0.50 lb/hp-hr
Mileage	24.0 mpg	38.0 mpg		52.0 mpg
Improvement		58%		117%

[a] Equivalent work computed at 0.808 lb/hp-hr

[b] A single CVT package will replace both units

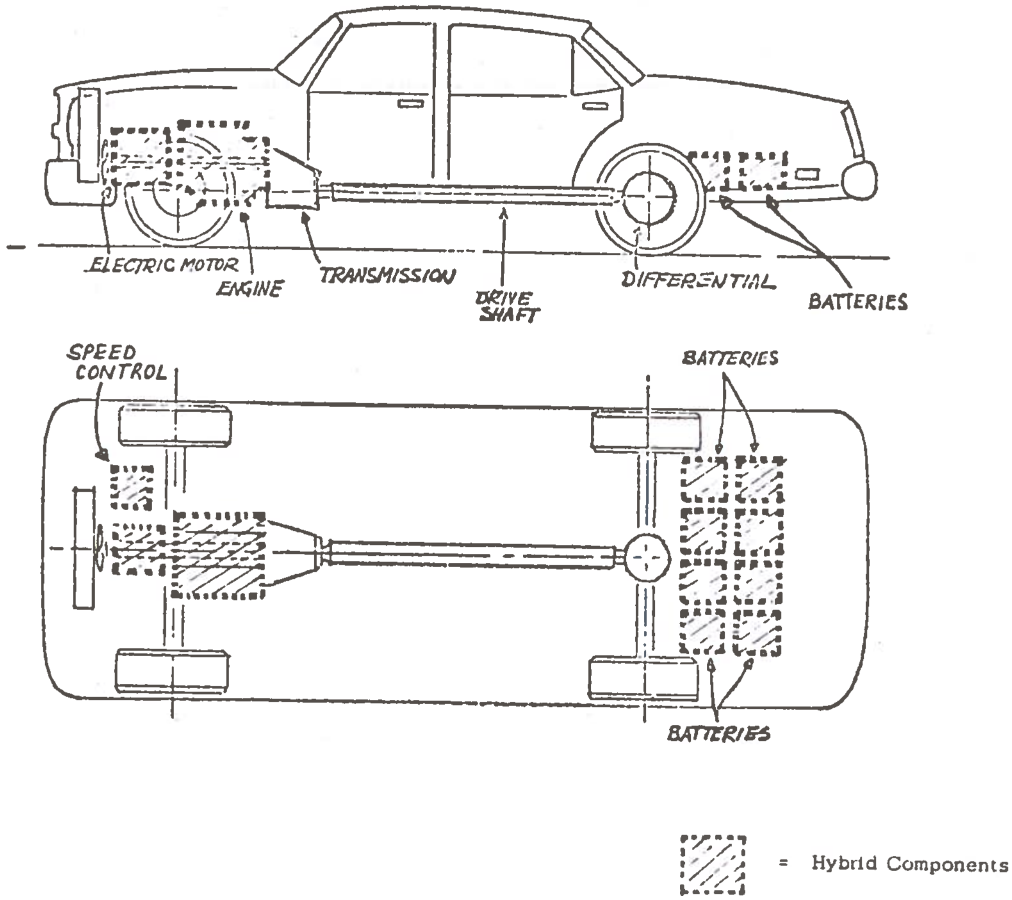


FIGURE 3-1. COMPONENT LAYOUT FOR PETRO-ELECTRIC MOTORS HEAT ENGINE/BATTERY ELECTRIC HYBRID VEHICLE

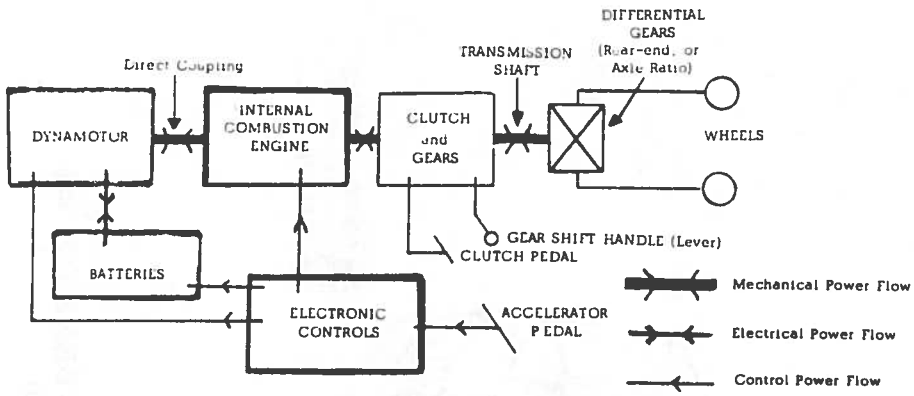


FIGURE 3-2. BLOCK DIAGRAM OF POWERTRAIN OF PETRO-ELECTRIC MOTORS PROTOTYPE VEHICLE

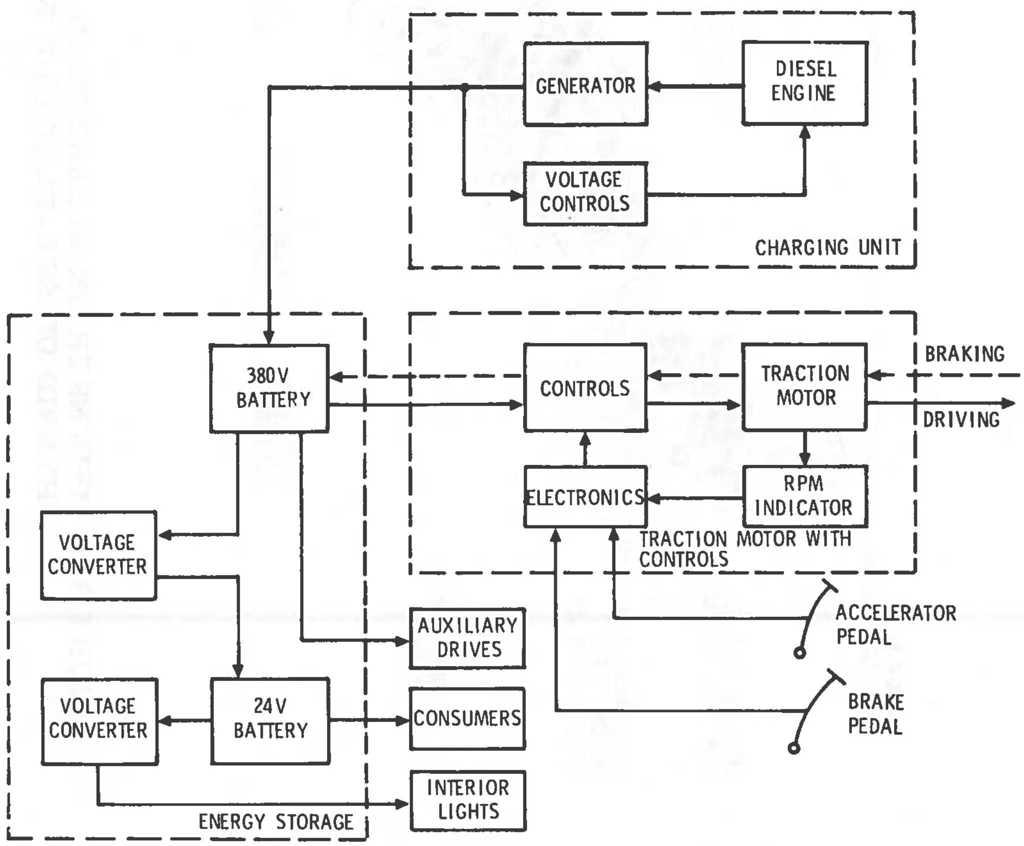


FIGURE 3-3. ELECTRIC DRIVE SYSTEM SCHEMATIC FOR MERCEDES-BENZ OE 302 HYBRID DIESEL-ELECTRIC BUS (Ref. 3-3)

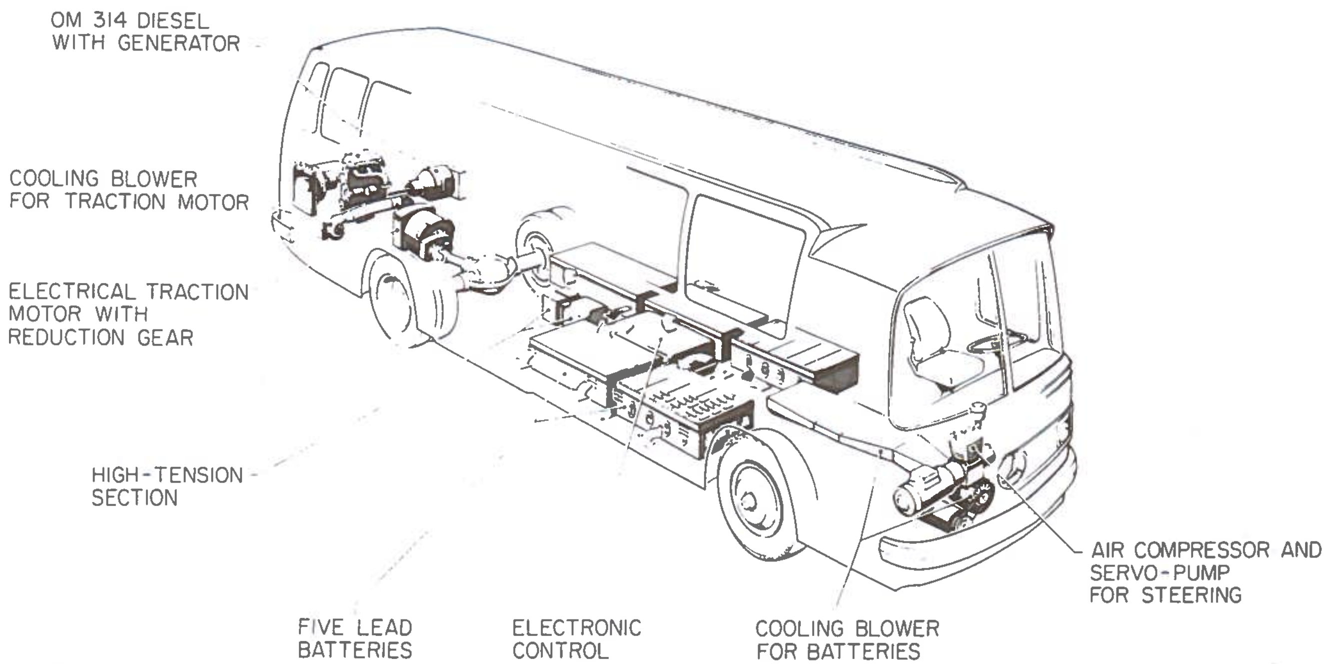


FIGURE 3-4. POWERTRAIN ELEMENTS IN MERCEDES-BENZ OE 302 HYBRID DIESEL-ELECTRIC BUS (Ref. 3-3)

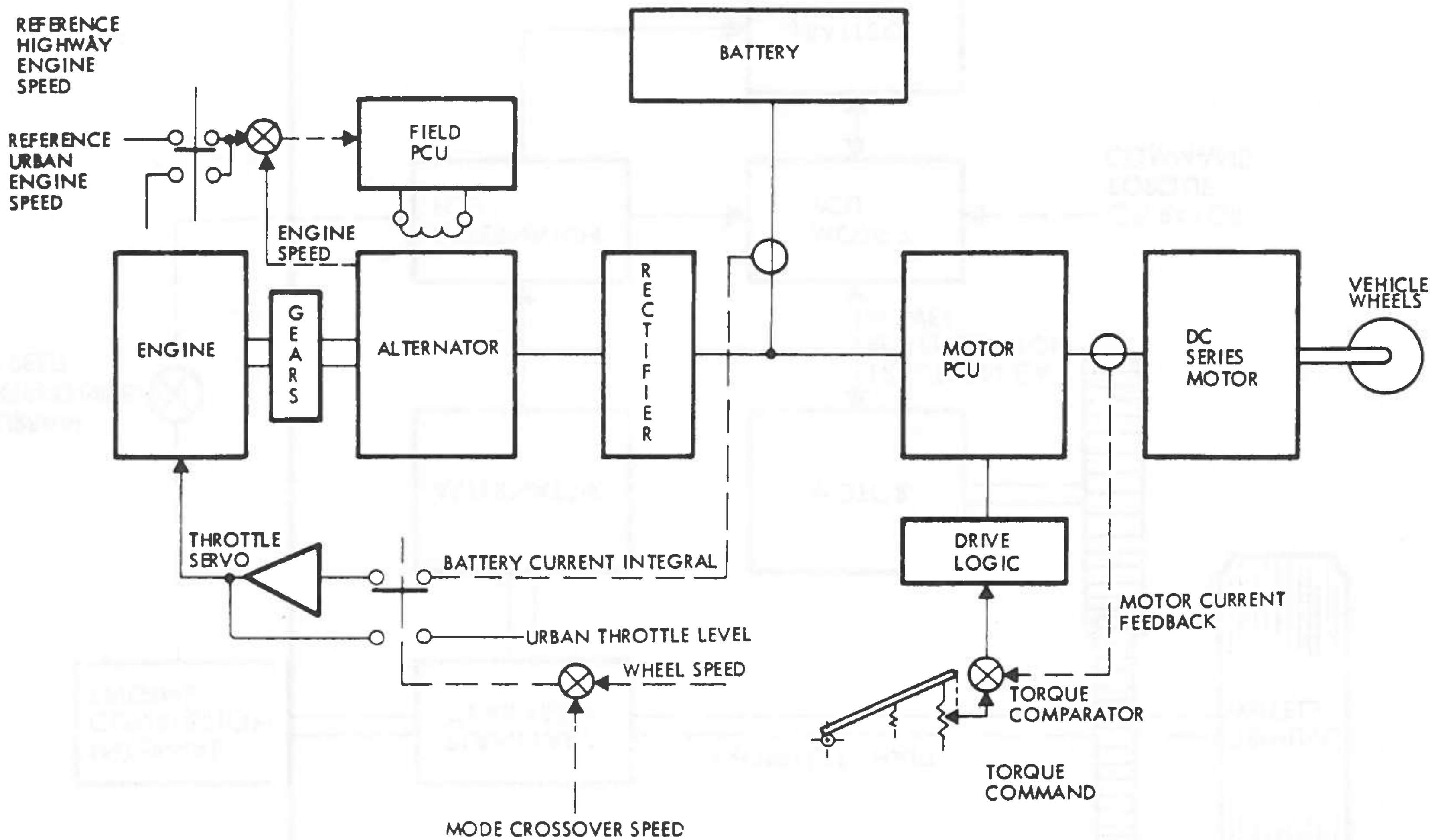


FIGURE 3-5. TRW FULL VOLTAGE SERIES-CONNECTED HYBRID SYSTEM (Ref. 3-4)

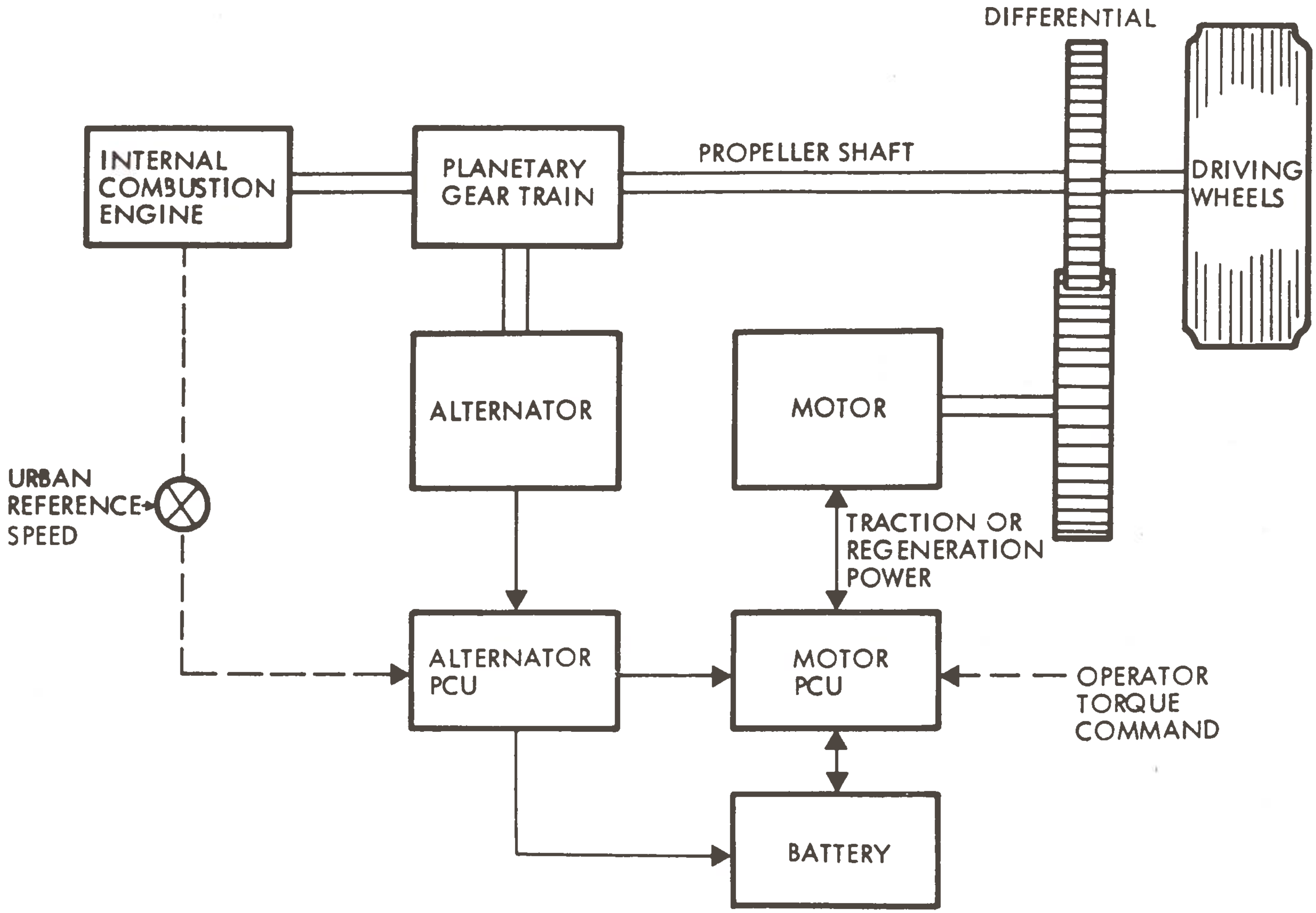


FIGURE 3-6. TRW EMT PARALLEL HYBRID SYSTEM IN URBAN TRAFFIC MODE (Ref. 3-4)

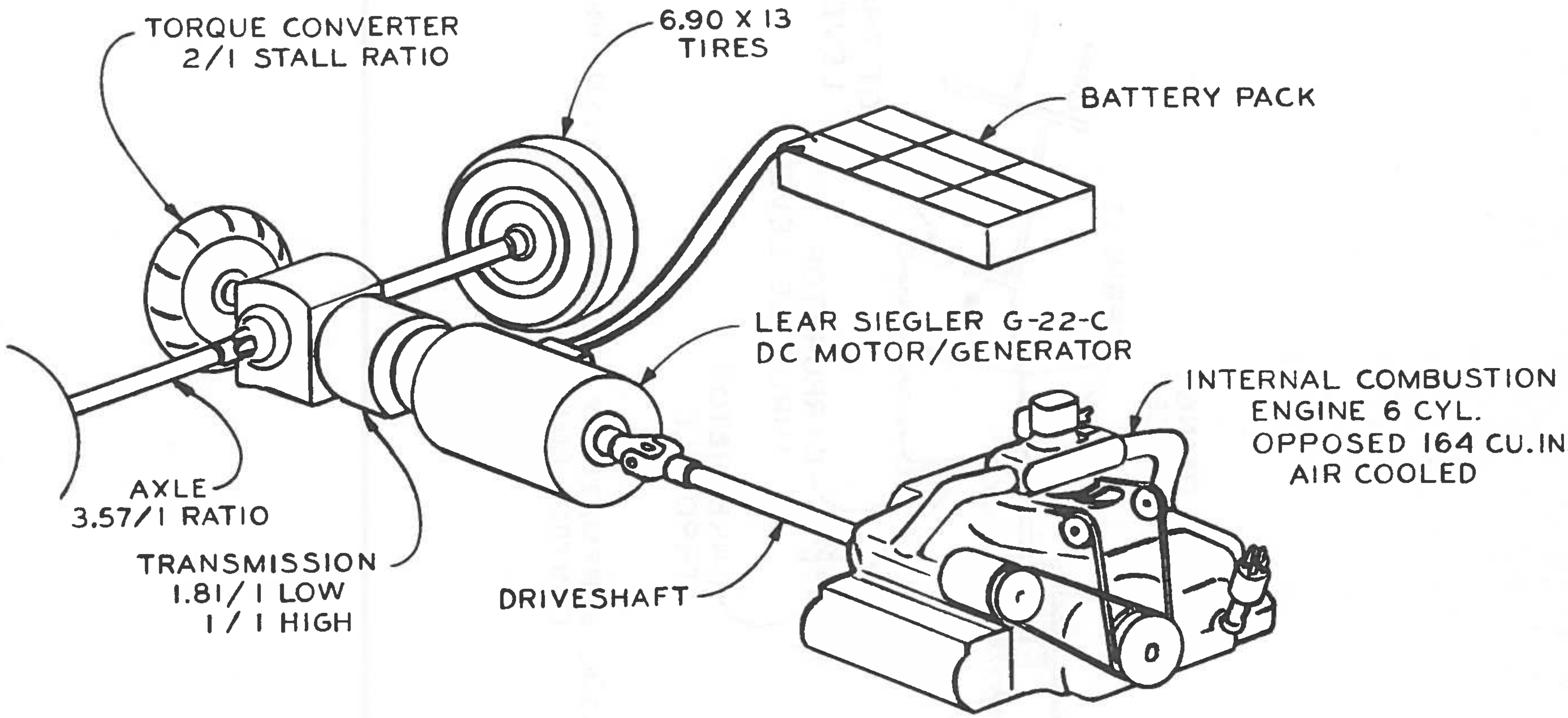


FIGURE 3-7. MINICARS HEAT ENGINE/ELECTRIC HYBRID POWERTRAIN (Ref. 3-5)

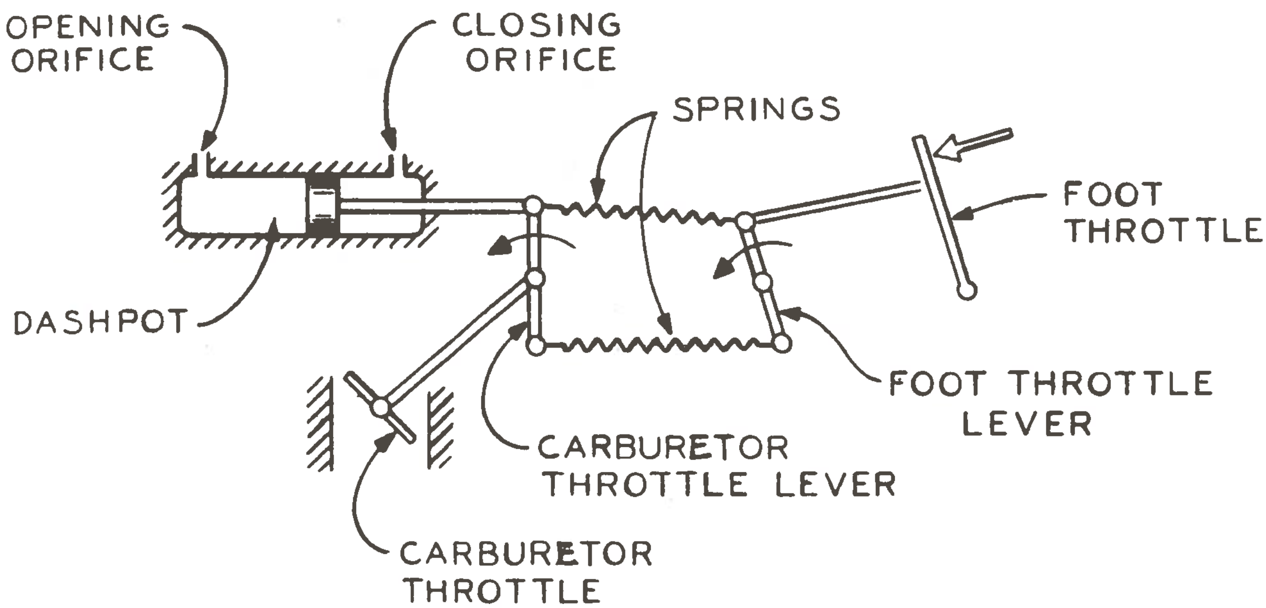
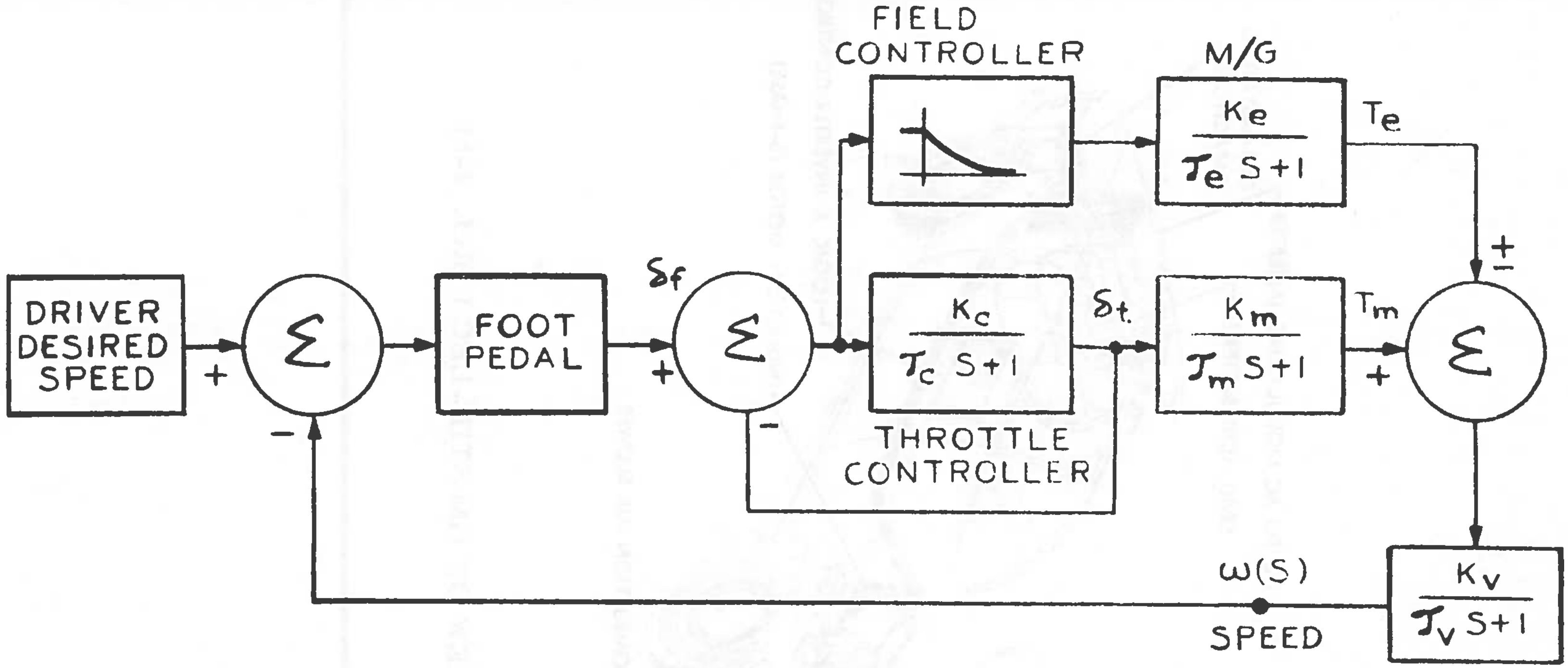


FIGURE 3-8. SCHEMATIC OF MINICARS DELAYED THROTTLE CONTROL (Ref. 3-5)



δ_f – FOOT PEDAL ANGLE		e – ELECTRICAL	
δ_t – THROTTLE ANGLE		m – MECHANICAL	
T – TORQUE		c – CONTROLLER	
K – GAIN		v – VEHICLE	
S – LAPLACE TRANSFORM			
τ – TIME CONSTANT			

FIGURE 3-9. SCHEMATIC DIAGRAM OF THE MINICARS THROTTLE DELAY AND ELECTRICAL CONTROL SYSTEMS FOR HYBRID CAR (Ref. 3-5)

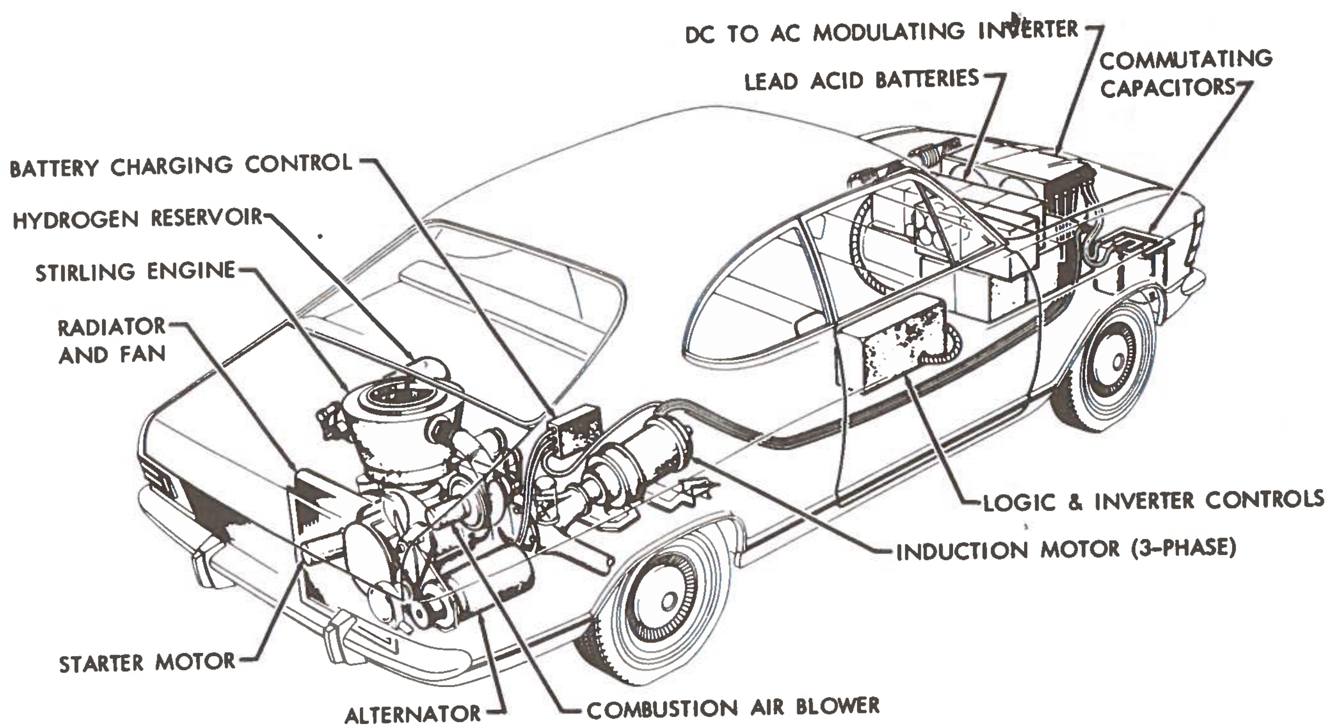


FIGURE 3-10. PHANTOM VIEW OF GM STIR-LEC I (Ref. 3-6)

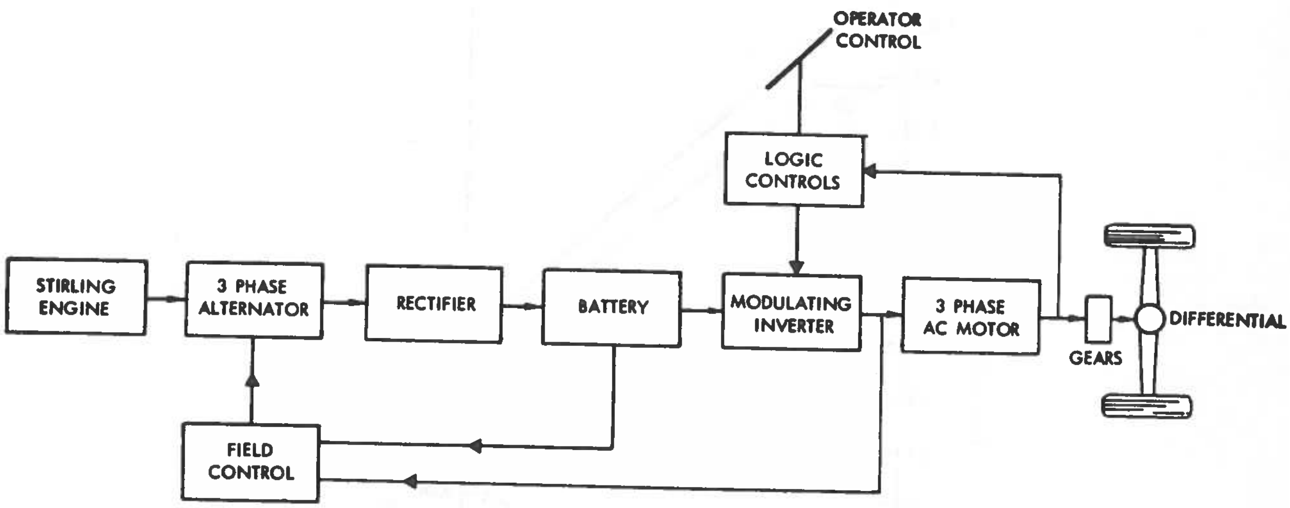


FIGURE 3-11. BLOCK DIAGRAM OF GM STIRLING-ELECTRIC HYBRID SYSTEM (Ref. 3-6)

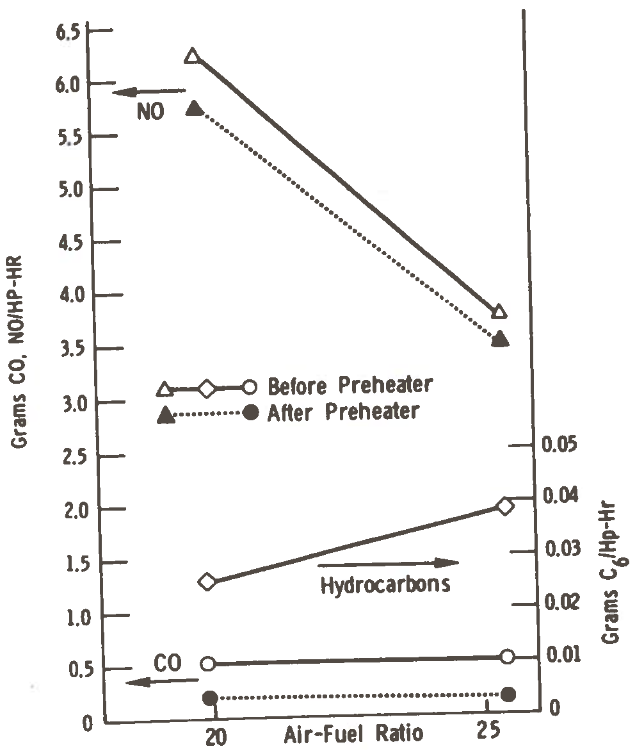
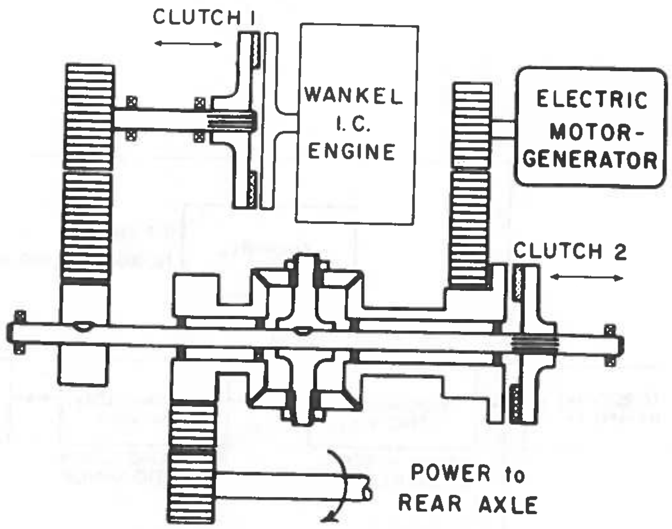


FIGURE 3-12. GM STIR-LEC I ENGINE MASS EMISSIONS (Ref. 3-6)



MODES of OPERATION

① Battery drive, or regenerative braking
Clutch I disengaged, Clutch 2 engaged

② Drive and generate
Clutch I engaged, Clutch 2 disengaged

③ Power from Wankel and electric motor
Both Clutches engaged

FIGURE 3-13. UNIVERSITY OF WISCONSIN HYBRID ELECTRIC POWER-PLANT SCHEMATIC (Ref. 3-10)

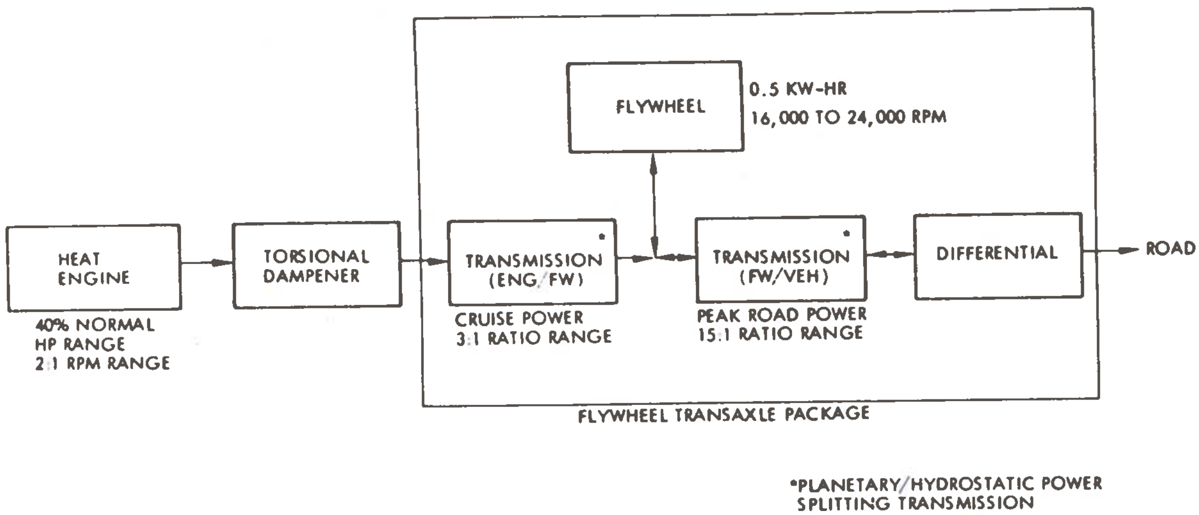


FIGURE 3-14. LOCKHEED TRANSAXLE FLYWHEEL/HYBRID TRANSMISSION CONFIGURATION (Ref. 3-11)

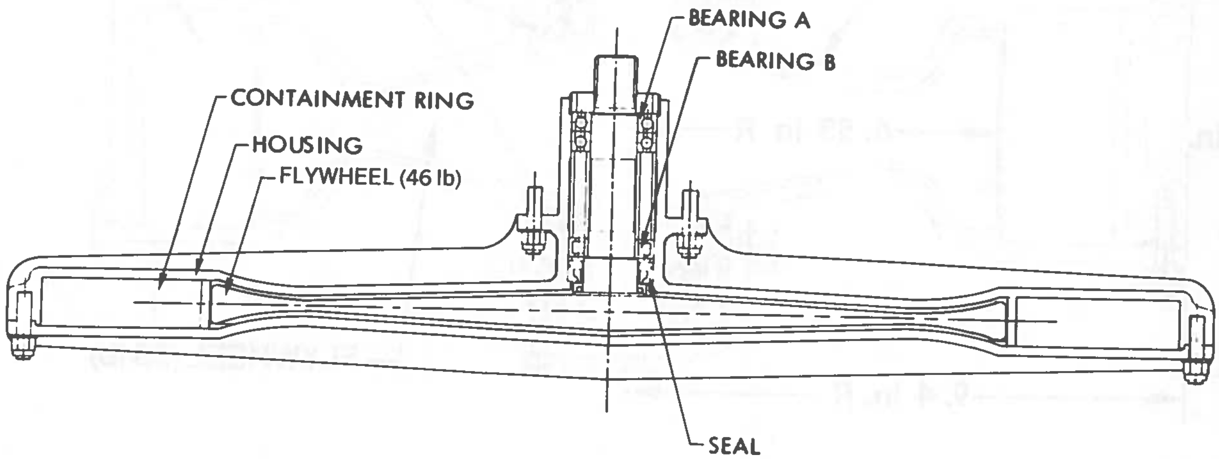


FIGURE 3-15. PRELIMINARY LOCKHEED FLYWHEEL DESIGN AND TEST CONFIGURATION FAMILY CAR (Ref. 3-12)

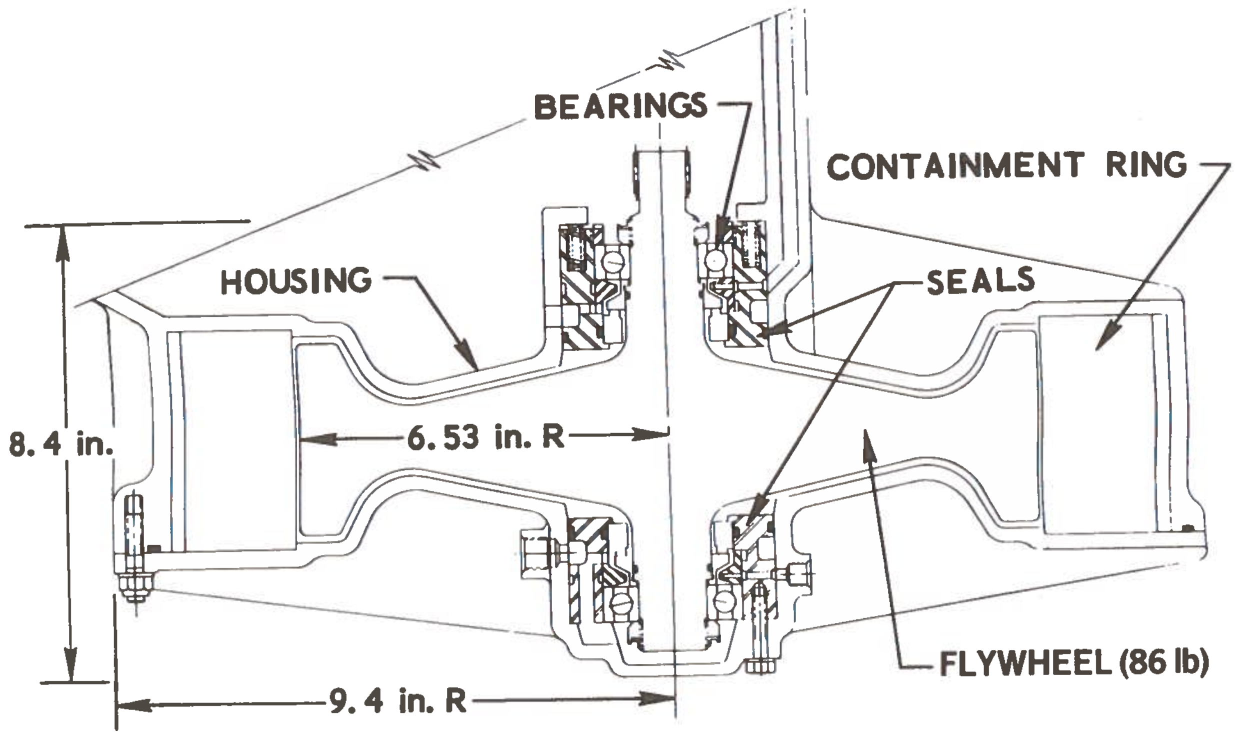


FIGURE 3-16. LOCKHEED BASELINE FLYWHEEL (Ref. 3-12)

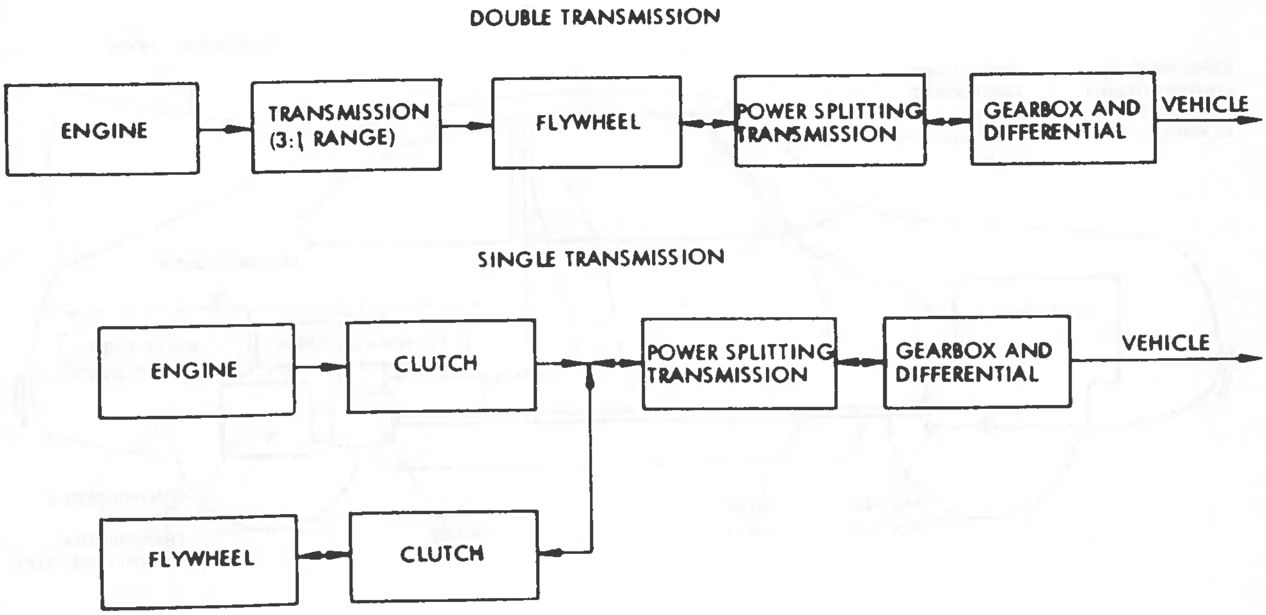


FIGURE 3-17. LOCKHEED POWER-SPLITTING TRANSMISSION CONFIGURATIONS (Ref. 3-11)

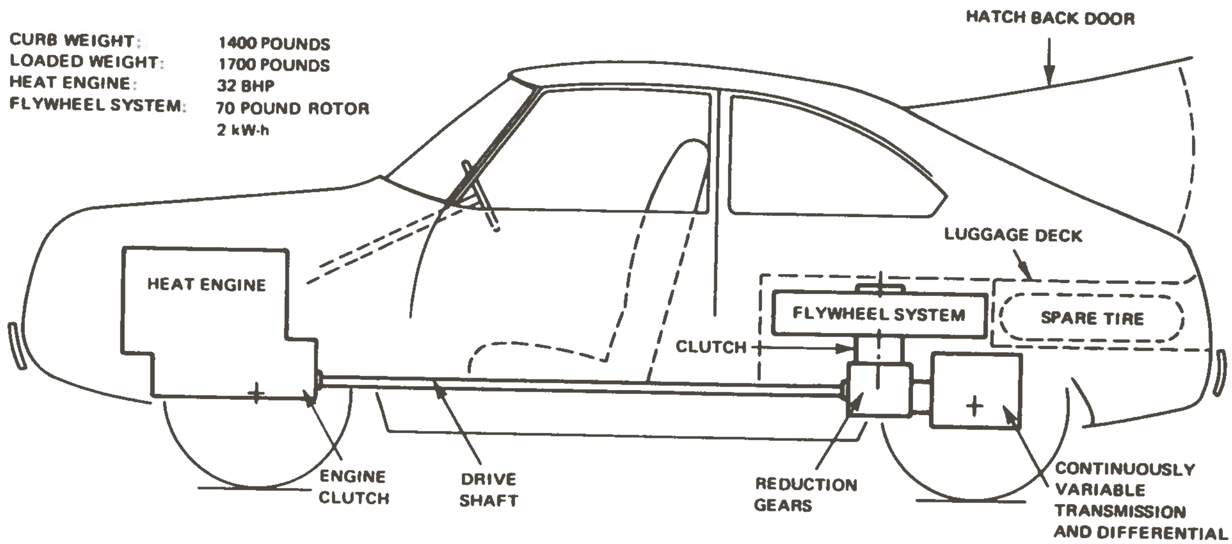


FIGURE 3-18. JOHNS HOPKINS HEAT ENGINE/FLYWHEEL HYBRID COMMUTER CAR (Ref. 3-16)

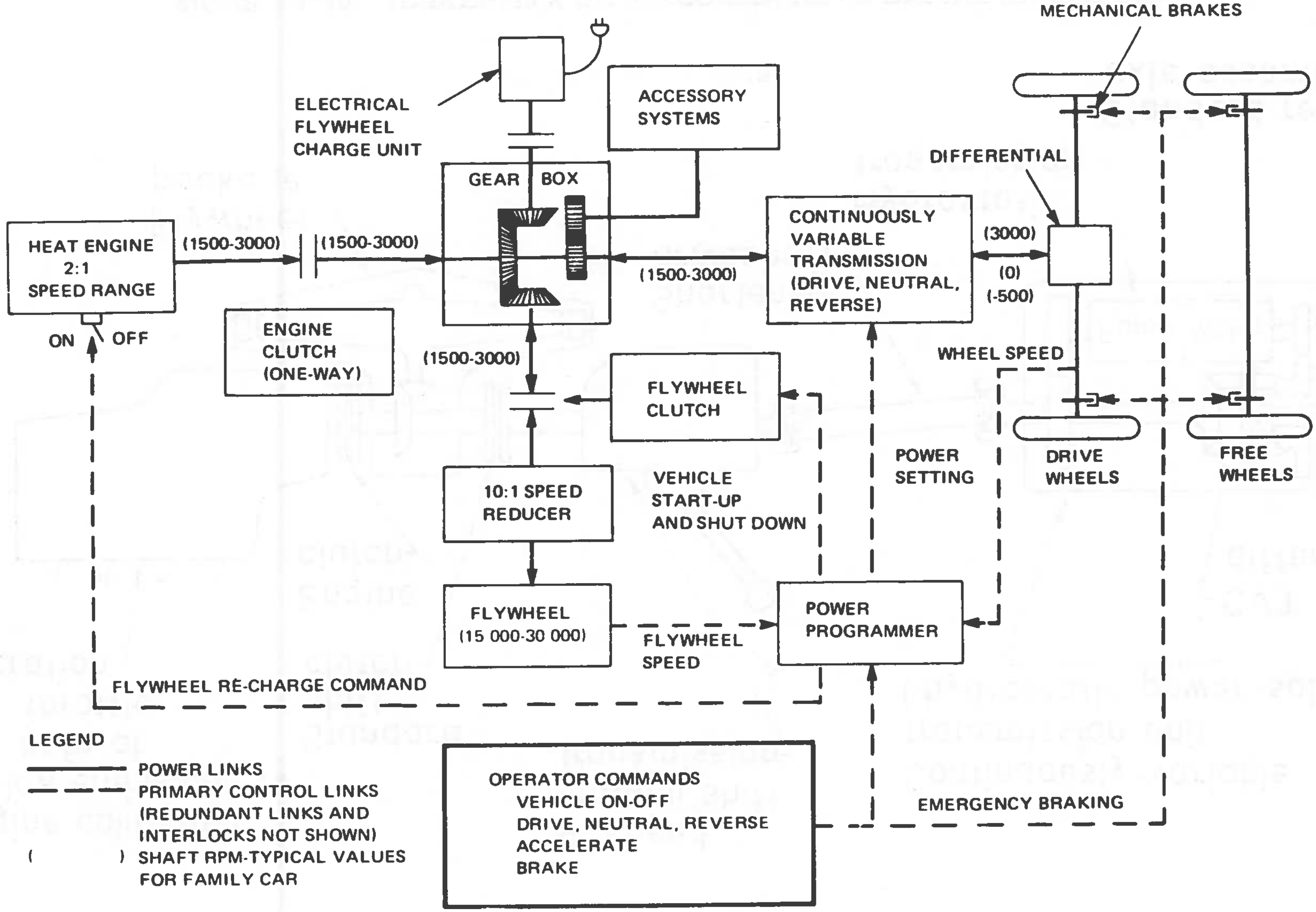


FIGURE 3-19. JOHNS HOPKINS FLYWHEEL HYBRID POWER CONTROL SYSTEM (Ref. 3-16)

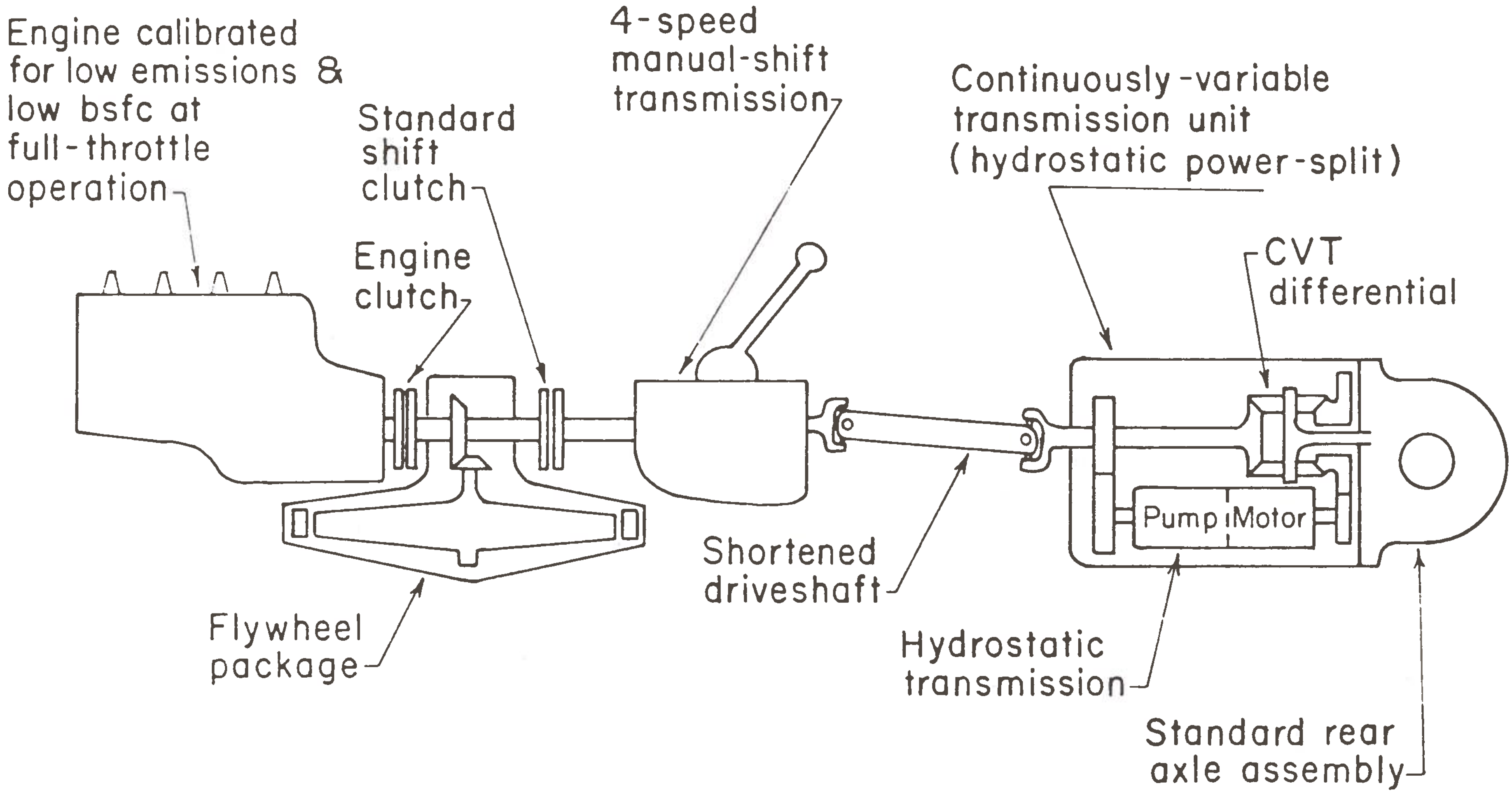


FIGURE 3-20. UNIVERSITY OF WISCONSIN HEAT ENGINE/FLYWHEEL SYSTEM CONFIGURATION (Ref. 3-17)

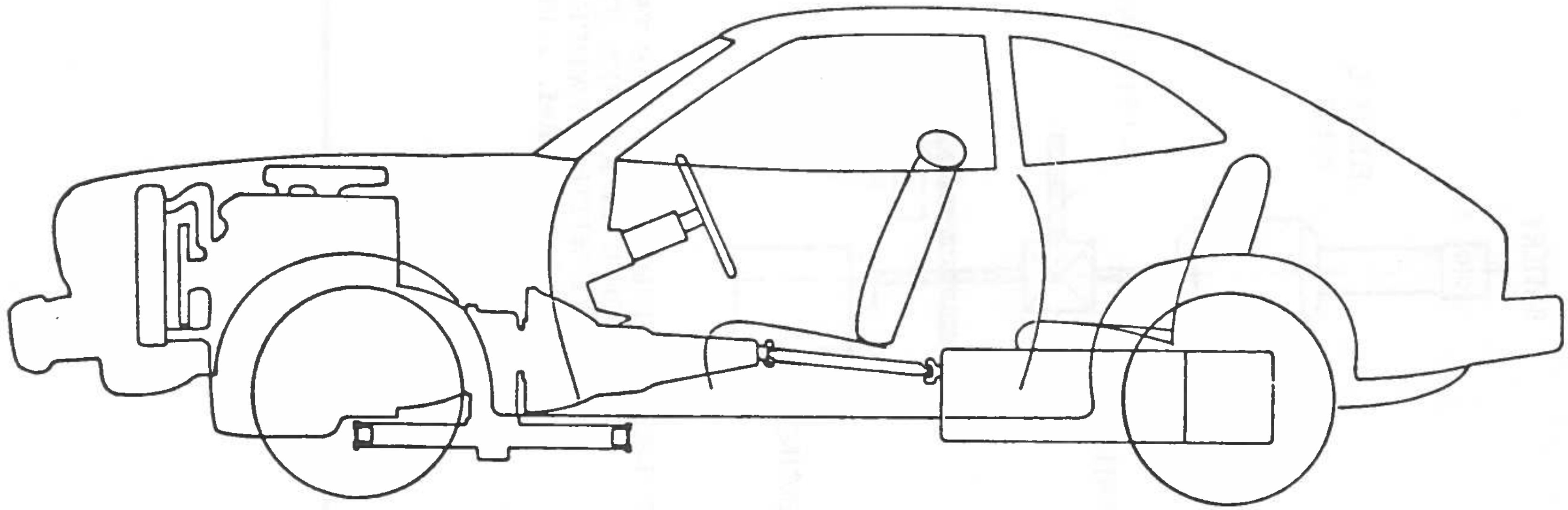


FIGURE 3-21. INSTALLATION SKETCH -- UNIVERSITY OF WISCONSIN HEAT ENGINE/FLYWHEEL SYSTEM (Ref. 3-17)

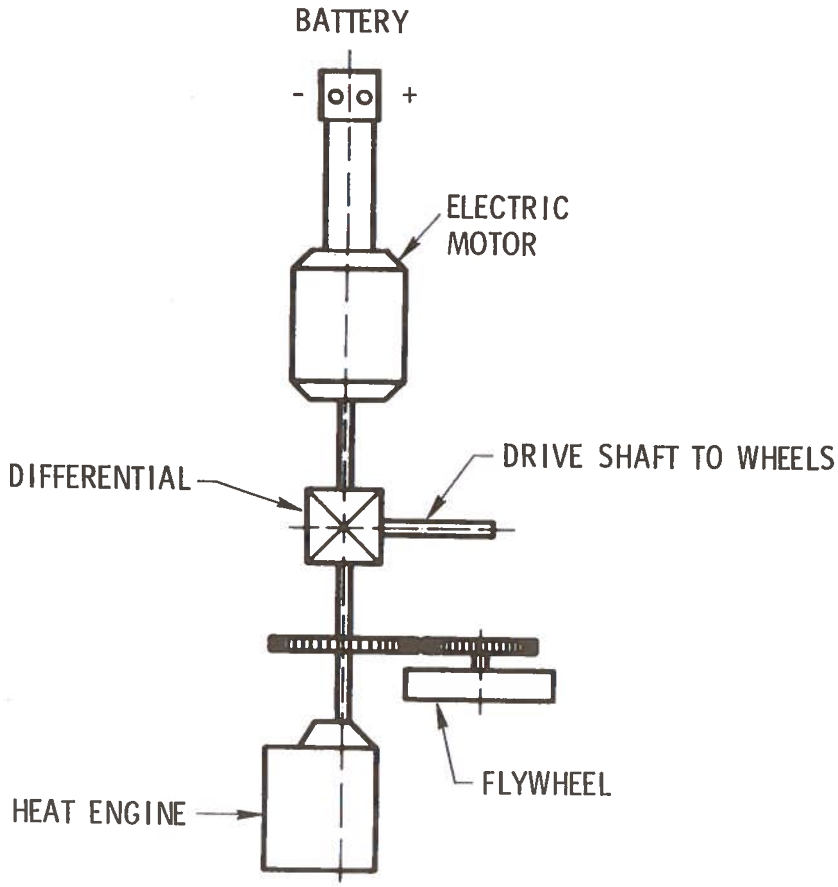


FIGURE 3-22. SCHEMATIC OF THE TECHNICAL SCHOOL AT AACHEN HYBRID-DRIVE WITH FLYWHEEL COMPONENT (Ref. 3-19)

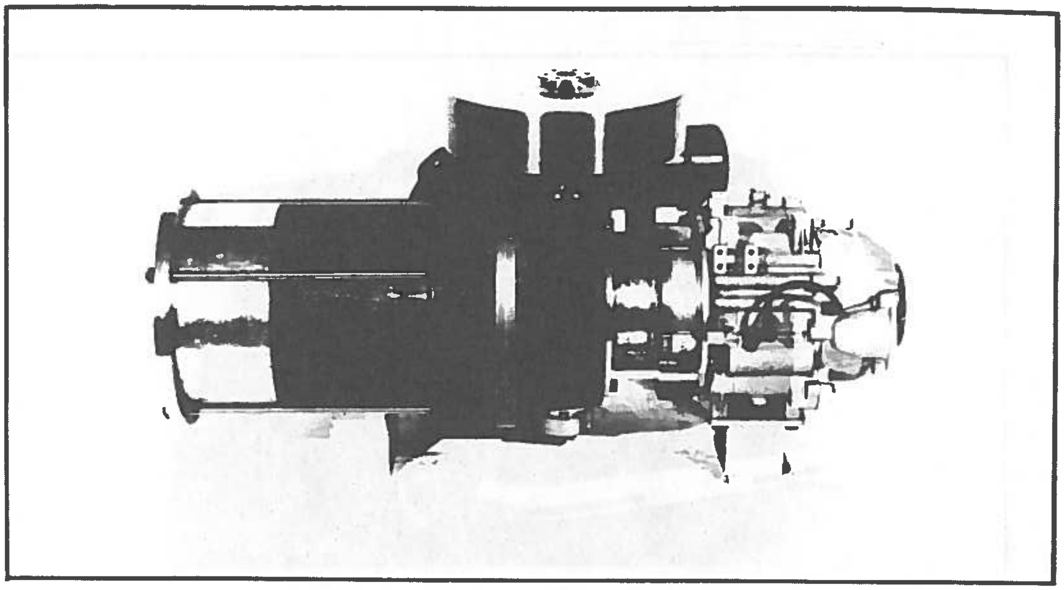


FIGURE 3-23. TECHNICAL SCHOOL AT AACHEN HYBRID-DRIVE UNIT WITH FLYWHEEL ENERGY STORAGE (Ref. 3-19)

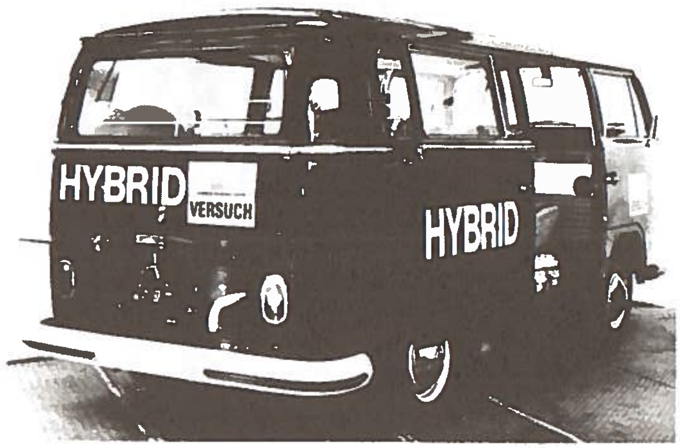


FIGURE 3-24. TECHNICAL SCHOOL AT AACHEN TEST VEHICLE WITH HYBRID DRIVE (Ref. 3-19)

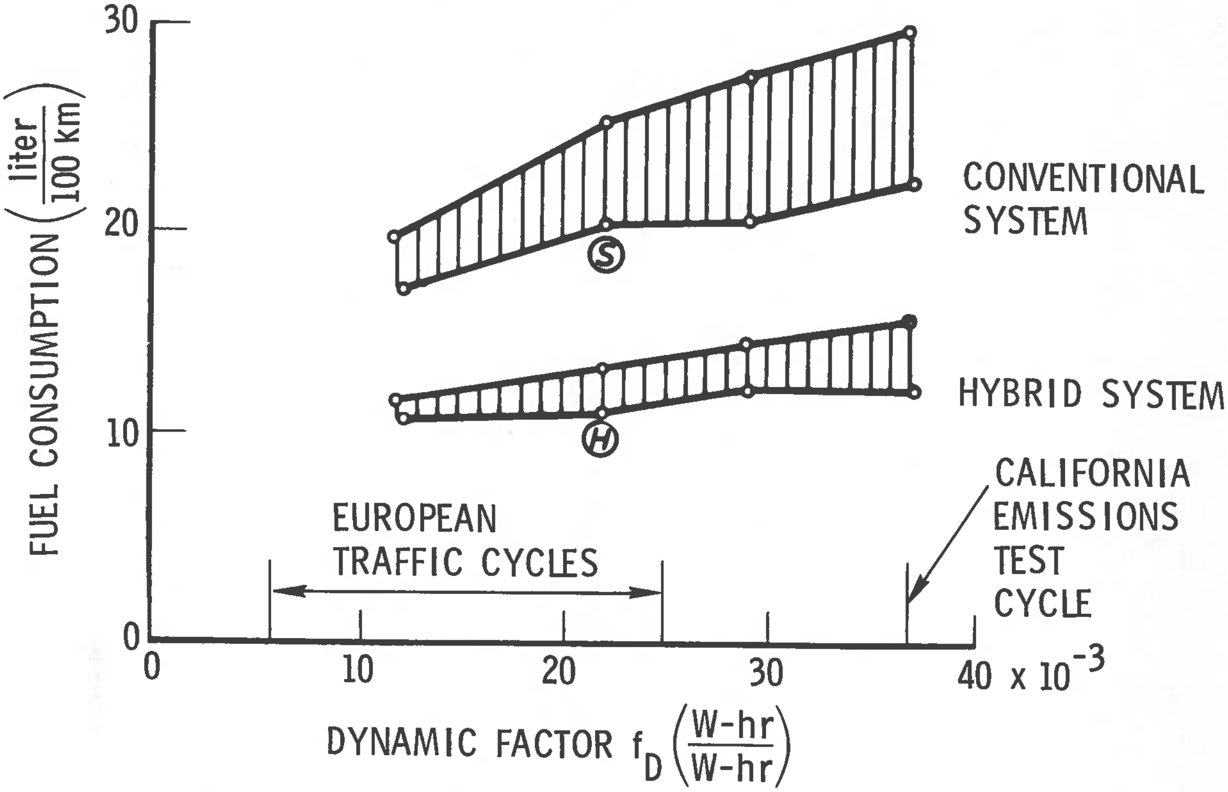


FIGURE 3-25. FUEL CONSUMPTION FOR 2100-KG VEHICLE AS A FUNCTION OF THE DYNAMIC FACTOR (Ref. 3-19)

4. TECHNOLOGY REVIEW OF POWERTRAIN COMPONENTS

A brief technological review is presented for each major element in the powertrain for both heat engine/battery and heat engine/flywheel systems. This review concentrates on an evaluation of expected component performance as applied to hybrid vehicles and avoids, wherever possible, a repetition of material readily available in textbooks or handbooks. The information presented herein formed the basis for selection of component performance that was used in the vehicle simulation computer program.

4.1 HEAT ENGINES

The heat engines suitable for hybrid application can be divided into two classes: (1) internal combustion engines and (2) external combustion engines.

The conventional spark ignition engine and the diesel engine belong to the first class. Supercharged engines, rotary piston engines, and stratified charge engines are modifications of internal combustion engines. The second class is represented by the Stirling, Rankine, and gas turbine engines. From a fuel economy point of view, the diesel and the Stirling engines dominate the field at present levels of technology. Steam engines are large and heavy and high in fuel consumption; thus they are not candidates for hybrid automotive applications. Gas turbines are compact and light weight, but suffer from a lack of flexibility; their fuel economy at part load is poor. However, they could be considered for hybrid applications provided the majority of power variations are handled by the engine storage system. The rotary piston engine has poorer fuel economy than the baseline spark ignition engine; its only advantage lies in its smaller specific volume and weight.

Figure 4-1 shows some fuel economy results obtained over the Federal Emissions Test Driving Cycle for various heat engines as reported in Ref. 4-1. Although these data points were obtained for conventional vehicles, the fuel economy ordering that can be expected from the use of various heat engines in hybrid vehicles of a given inertia weight is not expected to change too significantly from that of Figure 4-1. Another form of presentation is to show the potential efficiency of each of the candidate heat engines as in Figure 4-2 for operation of a conventional vehicle over the Federal Emissions Test Driving Cycle. It is not expected that the use of these heat engines in hybrid vehicles will change their relative position on the bar chart shown, although their absolute values may change. Translated to a fuel economy point of view, again, the two best heat engine candidates with existing technology are the diesel and Stirling engines.

For hybrid automobile applications, weight and volume are important parameters to be considered because of the need to house not only the heat engine but also the energy storage system. Figure 4-3 shows comparisons of engine weight and volume. Figures 4-4 through 4-6 show comparative results for bare engine exhaust emissions. The best and worst emissions when examined over the horsepower range show no consistent superiority of any engine for all three pollutants combined.

Each of the engines discussed above on a comparative basis will be reviewed individually in the following paragraphs.

4.1.1 Spark Ignition Engines

The spark ignition engine is the most extensively used automobile powerplant, and consequently can be taken as a baseline engine. Much of the success of this engine is due both to its ability to change power output rapidly and to its good specific weight and specific volume characteristics. However, the need to save gasoline and problems with control of exhaust emissions have prompted research into new designs and methods of operation.

In recent years, the use of air injection in the exhaust, exhaust gas recirculation, and modifications to compression ratio, timing, carburetion, and choking has led to marked reductions in engine exhaust emissions, although there have been some accompanying reductions in fuel economy as emission regulations become more stringent. The promulgated emission standards for light-duty vehicles in model year 1975-1976 have required the further need for catalytic treatment of exhaust gases in some cases. With modern emission control systems, the engine fuel consumption increase has been ameliorated and, considering the engine specific weight and volume and its widespread use, it is a suitable choice for hybrid vehicles.

A set of performance curves representative of a 1973 model year, V-8 engine is given in Figures 4-7 through 4-10. Contour maps of brake specific fuel consumption (BSFC) and brake specific emissions for hydrocarbons, carbon monoxide, and oxides of nitrogen (BSHC, BSCO, and $BSNO_x$) are shown for the parameters brake mean effective pressure (BMEP) and engine revolutions per minute (rpm).

4.1.2 Diesel Engines

The diesel engine is a well developed engine which is being mass produced. While it has been used mostly for industrial or commercial applications, a few foreign manufacturers, such as Mercedes Benz, Opel (Germany), and Peugeot (France) have used the diesel engine in passenger cars; Perkins (England) and Daihatsu (Japan) have also manufactured the engine.

The advantages of the diesel engine are its excellent fuel economy, particularly at part load. This is seen in Figure 4-11 where the specific fuel consumption (SFC) of the diesel varies between 0.42 and 0.64 lb/hp-hr while the corresponding spread of SFC for the spark ignition engine is between 0.45 and 1.34 lb/hp-hr.

The diesel has excellent hydrocarbon (HC) and carbon monoxide (CO) emissions characteristics (Ref. 4-1) (HC ≤ 0.4 gm/mi, CO ≤ 2 gm/mi on the Federal Emissions Test Cycle); however, according to the data shown in Figures 4-4 through 4-6, careful design is required to obtain these low exhaust emission rates.

While the production of oxides of nitrogen (NO_x) can be reduced to very low levels, it is still too high to satisfy the ultimate statutory federal standards of 0.4 gm/mi. Typically, NO_x of current diesel engine-powered automobiles is below 1.5 gm/mi and some reduction to the 1 gm/mi level might be achieved by means of retarded injection timing, exhaust gas recirculation, or water injection.

In general, diesels which use a prechamber give a lower NO_x and slightly higher SFC than direct injection diesels. Increased compression ratio increases NO_x but reduces soot formation. The use of a turbocharger would reduce SFC somewhat, but there is a danger of increasing soot formation and NO_x. Water injection reduces NO_x and soot formation.

Smoke and odor are negative characteristics of the diesel engines, although they have been reduced to tolerable levels. The mechanism of odor formation is not well understood and thus, its appearance cannot always be predicted.

Other negative features of the diesel engines are its specific weight and specific volume which are higher than for similar spark ignition engines. This is seen in Figures 4-12 and 4-13.

The relatively small use of diesel engines in automotive applications has largely resulted from the higher specific weight and specific volume, the cost of this engine, and its sluggish response to acceleration demands. However, sluggish response should not be detrimental to hybrid applications where most of the acceleration demands are handled by the energy storage system. The major factor remains that until such time as weight and volume are significantly reduced, this engine is generally considered unsuitable for application to hybrid vehicles.

4.1.3 Stirling Engines

Displacement-type Stirling engines have operated in prototype buses and boats in Europe; they have never been mass produced for automotive applications. Among the advantages of the Stirling engine is its inherent high efficiency (Figures 4-2 and 4-14) which result in excellent fuel economy. Improvements in fuel consumption have brought the Stirling engine to the level of the diesel engine. Specific fuel consumption and torque maps are shown in Figure 4-15. However, these data may be obsolete. More up-to-date information has not been found for specific fuel consumption, but preliminary torque data from Ford are given in Figure 4-16. Ford has been under contract to EPA/AAPS* to develop a pre-production 100-hp Stirling engine suitable for a 3000-lb car. Target date for the first production car is 1981 (Ref. 4-7).

At first, Stirling engines were very heavy and bulky (0.06 ft^3/lb), but the use of a swash plate instead of the rhombic drive has reduced

*U.S. Environmental Protection Agency/Advanced Automotive Power Systems — program has been transferred to ERDA (Energy Research and Development Agency).

this problem to manageable levels. The 7 lb/hp obtained by General Motors in 1970 (Ref. 4-8) has been reduced recently to 4 lb/hp by Ford.

New burners developed by N. V. Philips (Netherlands) have resulted in improved fuel consumption and exhaust emission characteristics for this engine. Current best results yield 74 percent fuel consumption improvement over spark ignition engines while achieving HC = 0.10 gm/mi, CO = 0.31 gm/mi, NO_x = 0.17 gm/mi using EGR. These results make the Stirling engine one of the main contenders for future hybrid vehicle applications and forecast compliance with the 1978 federal emission standards (Ref. 4-7).

The Stirling engine is a very quiet (70 dB(A)) and smooth-running engine and is practically smokeless and odorless. Also, Stirling engines can burn a wide variety of fuels.

Among the main disadvantages of the Stirling engine is its complexity, which makes it more expensive to fabricate than the spark ignition engine. It requires a cooling system two or three times the size of a conventional spark ignition engine. Also, for best efficiency, hydrogen has been used as the working fluid, but there have been hydrogen diffusion problems in the past. Now, however, a solution has been reported by Philips (Netherlands). It is claimed that a proprietary coating, when applied to the heater heads, reduces hydrogen diffusion by a factor of 100. In addition, piston seal life is a problem which needs a solution.

Although, as with many external combustion engines, the Stirling engine has lower acceleration response than the conventional spark ignition engine, this is not considered detrimental in hybrid vehicle applications where acceleration transients can be handled by the energy storage system. A long startup time (20 sec minimum), however, may be considered objectionable.

At the present time, the lack of verifiable specific weight, specific volume, and performance maps for fuel consumption and exhaust emissions prevents further consideration of this engine for hybrid vehicle applications; but the potential for future consideration appears to be good.

4.1.4 Gas Turbines

The development of gas turbines as applicable to automotive needs has been slowed by poor fuel economy and estimates of high production costs. It is a compact, high-speed, high-temperature machine, but its fuel economy has been limited by high-temperature material availability, durability, and cost. The effect of temperature is depicted in Figures 4-17 and 4-18, where it is shown that low specific fuel consumption can be achieved only with high turbine inlet temperature. Development of ceramic components for the turbine section is necessary for operation at high temperature. Work of this nature is under way (Ref. 4-9).

No automotive turbine engine is in production today. Several experimental gas turbines have been built and tested in the last few years. These are listed in Table 4-1. Figures 4-19 and 4-20 are based on Ref. 4-1 data. The specific weight to be expected in the 100-hp to 200-hp gas turbine range is about 3 lb/hp, with a corresponding specific volume of about 0.10 ft^3/hp. The data spread indicates the state of infancy of the gas turbine for automotive applications. Much more development work is needed before a commitment to production can be made.

An analytical study of various types of basic performance parameters for gas turbines is shown in Table 4-2. The values were obtained with the inputs shown in Table 4-3.

Chrysler Corporation has been developing a gas turbine for EPA/AAPS* (Ref. 4-10). The minimum fuel consumption achievable with the 150-hp baseline engine is shown in Figure 4-21. It is not competitive with spark ignition or diesel engines, particularly at part load, but an upgraded and slightly smaller engine (123 hp) is being fabricated by Chrysler with improved specific fuel consumption. (This upgraded gas turbine has a specific weight of 3.25 lb/hp.) However, until a full engine map for emission and fuel consumption is available, these parameters are not known for various combinations of engine speed and power output. Furthermore, to achieve the stated minimum curve would require the use of a continuously variable transmission (which is a possibility).

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*This program has been transferred to ERDA.

Low hydrocarbon and carbon monoxide emissions have been reported in the literature (see Table 4-4). It is seen that the oxides of nitrogen emissions are generally too high to satisfy future federal standards. General Motors, though, has succeeded in achieving NO_x emissions below 0.4 gm/mi, and, based on advanced combustor data, Solar has predicted gas turbine vehicle NO_x emission below 0.4 gm/mi over the Federal Emissions Test Driving Cycle.

The poor fuel consumption exhibited by gas turbines subjected to rapid changes in power output need not be present in the applicable hybrid vehicles because transient accelerations can be handled by the energy storage system. This should also relieve the problem of gas turbine response time in meeting driver power demand (Figure 4-22).

Generally, for all external combustion engines there is a delay in starting time when compared to the conventional spark ignition, reciprocating piston engine. This is shown for the gas turbine in Figure 4-23. Again, for the hybrid vehicle, this is not a necessary detriment if the energy storage system can temporarily supply the necessary power.

Because of its very low specific weight and volume, this engine is considered suitable for application to hybrid vehicles. But further consideration must be delayed until fuel consumption is improved and full engine maps are available.

4.1.5 Rankine (Steam) Engines

Like all external combustion engines, the steam engine has multifuel capability and good emission characteristics. However, all the systems tested to date have exhibited very poor fuel economy. The best specific fuel consumption obtained by Scientific Energy Systems Corporation on a 150 hp steam engine with inlet steam at 1000 psi and 1000°F was only 0.707 lb/hp-hr (Ref. 4-11).

The vapor generator for the engine can be used with different types of expanders like the piston, rotary, or turbine types. It is a more expensive and complex engine than the spark ignition engine and it has excessive weight and bulk (Figures 4-3 and 4-24). It requires a long start time, and some designs respond sluggishly to acceleration demands.

At present levels of fuel consumption, there appears to be little incentive to use the Rankine engine. More development work is needed before it could be considered for possible use. Table 4-5 is a list of recent development efforts.

4.1.6 The Roesel Heat Engine

The Roesel heat engine is being developed by Roesel Laboratories (Ref. 4-12). It is a heat engine of the external combustion type very similar to the Stirling engine except that the working fluid never leaves the power cylinder. As such it is a pneumatic engine.

The Roesel heat engine can act directly as a hydraulic pump and thus is readily integrable into a hydraulic system using hydraulic motors, accumulators, and regenerative braking.

Reference 4-12 claims that the specific volume for this engine could be comparable to the diesel specific volume. However, only laboratory models have been built so far and this claim cannot be substantiated as yet.

All the advantages and disadvantages of an external combustion engine might be found in this engine. Thus, low emission, multifuel capability, poor response to load demands, high specific weight and high specific volume could be experienced with this engine. The system efficiency of the Roesel heat engine is not yet known, although it is claimed in Ref. 4-12 that it should be excellent. Fuel economy data are not available.

While this engine appears to have merit, it has not been sufficiently developed to be ranked in an evaluation of other engines competing for applications in hybrid vehicles.

4.2 ELECTRIC MOTORS/GENERATORS/CONTROL SYSTEMS

4.2.1 General

Many of the components suitable for use on hybrid vehicles have in the past been reviewed and screened in regard to type and operating mode for application to electric vehicle powertrains. Indeed, a wide selection of automotive-type motor and controller combinations has been used in electric cars. The controllers vary in complexity from the use of carbon-pile resistance stacks employed on early streetcars to three-phase, silicon-controlled rectifier (SCR) time-ratio controls used in some of the advanced experimental ac motor drives. No specific approach has yet been universally adopted, but the pulsewidth modulation SCR system has been used extensively to control brush-type dc motors for electric vehicles; the step voltage, parallel-series switching system has also been used in this application.

Of the remaining motor types available, only a small number can be considered for vehicle drive applications. Many, such as the stepper motor and hysteresis motor, are designed for special-purpose applications very different from those for an electric car. Others, such as the ac induction motor and the ac reluctance motor, require very complex controllers that must handle large currents for the amount of torque generated. The latter two types of motors must be supplied with both variable voltage and variable frequency. A chopper (or modulator) is used to convert fixed battery voltage to a smoothly variable dc voltage. Then an inverter converts this variable dc to ac voltage for an induction motor. Thus, by controlling the chopper and inverter simultaneously, the voltage and frequency of the motor can be varied smoothly (Ref. 4-13).

4.2.2 Drive Motors

Electric motors, particularly dc motors, have not been developed to provide combined optimization of efficiency, weight, size, and cost for vehicle propulsion. It is possible that they might be designed lighter than those on the market today with equal reliability and lifetimes because weight in many designs has not been a prime consideration. Some gains may be effected by installing improved (low hysteresis loss) core materials, replacing all the frame with the lightest weight materials at minimum structural rigidities that will maintain gaps and bearing integrity, and using high energy density fields (new rare earth cobalt permanent magnet materials). The usual motor frame structure is very thick, heavy, and strong. The weight saved could be used to increase the motor core cross-sections, reducing core hysteresis losses and magnetizing currents. Also, increasing the copper cross-sections will decrease the resistance I^2R copper losses. Motor efficiency at design load could thus possibly be increased from 75 to 90 percent at the same weight per unit horsepower. With proper design, the working portion of the motor can be larger for a given size and weight, as compared to floor-mounted models where weight is unimportant.

Currently, efficiency can be traded off against motor weight. For a given design power level, the weight per unit horsepower can be decreased if the efficiency is allowed to decrease as shown in Figure 4-25. Extrapolation from present technology indicates that weight-to-power ratios of 5.5 to 8 lb/hp should be achieved at a reasonable efficiency and cost by merely optimizing the design for the particular application and using lightweight materials whenever possible (Figure 4-26). Based on these figures, a 4000-lb vehicle with a series powertrain configuration capable of cruising at 80 mph would require a motor weighing about 375 pounds (rated at 57 hp).

4.2.2.1 Types of Motors

The following list describes motors that are candidates for electric vehicle propulsion and their characteristics:

a. The series wound dc motor has its field winding connected in series with the motor armature so the field strength is a function of load current. This provides a very steep speed-torque characteristic at light loads, which becomes fairly flat at overloads. The series motor has high starting torque capability because the field is strengthened as the load increases, thereby compensating for the demagnetizing effect of armature reaction. The series motor speed can be controlled by varying the applied voltage, but since changes in load can cause relatively large changes in speed, precise control is difficult. For example, if the maximum speed on level terrain is reached through application of full battery voltage to the series motor, as soon as the vehicle reaches an incline or a headwind, it will always slow down (series field strength increases) rather than draw additional power from the batteries to maintain speed. This motor is also difficult to control for power transfer during regenerative braking.

b. The shunt wound dc motor has its field connected in parallel across the motor armature. This provides a field strength independent of load current and directly proportional to applied voltage. This results in a fairly flat speed-torque characteristic. It is important that this type of motor have compensating windings to nullify the degradation that would result from armature reaction.

Without compensation, the demagnetization effect on the field poles of armature reaction will cause field weakening. Then, the motor draws more current, producing an avalanche effect, which further weakens the field.

To obtain good speed control with this type of motor, it is necessary to provide an independent power control for the separately excited field to obtain controlled field strength. Then, voltage to the armature can be varied in order to vary motor speed.

With constant field strength despite armature reaction, this motor will draw additional current at a given voltage on occasions of increased load (with the vehicle on an incline or encountering a headwind) to help maintain the same velocity. In addition, field weakening is sometimes used to increase speed up to three times base speed. During field weakening, the motor can operate at constant power, providing higher speeds. The power to operate the separately excited field is from 1 to 10 percent of the rated motor power. Accurately controlled regenerative braking is available when this motor is equipped with a separately excited field.

c. The compound wound dc motor provides both a shunt field and a series field so that performance will fall somewhere between a shunt and a series speed-torque characteristic, depending on the ratio of field strengths selected. If additional speed and regenerative braking control are desired, the parallel winding can be separately controlled as in a separately excited motor, rather than the usual shunt connection across the single-line input.

d. The permanent magnet dc motor has permanent magnet pole pieces in place of field windings. Field strength is constant and is not affected by armature reaction as in a wound field motor. This is due to the low permeance coefficient of the permanent magnet material. This provides a straight line speed-torque characteristic with relatively low no-load speed and high starting torque. By varying the applied armature voltage, precise speed control is obtained. Since this motor does not have a wound field, there are no field losses. This means that for an equivalent rating, it is more efficient at all equivalent power levels and speeds. This motor has strong potential for electric vehicles, but is not yet fully developed, because the permanent magnet material has not been available in sizes and field strengths needed for electric cars.

e. The brushless dc motor has often been used in fractional horsepower space applications, but has not yet been fully developed for electric vehicle use. The integral horsepower brushless dc motor will use rare earth cobalt permanent magnets and Hall effect detectors to trigger the SCRs for commutation. The brushless motor is self commutating. Load demand is met by variation of average current, and velocity demand is met by a change of effective voltage through the pulse-width modulation system.

f. The synchronous ac motor energized from a battery requires a controller incorporating both voltage control and frequency conversion. The voltage and frequency control can be achieved either in a separate chopper followed by an inverter or combined in a modulating inverter. The ac synchronous motor will come to a stop if its maximum torque is exceeded. A rotor position sensor can be incorporated into the motor which will force the motor to operate like the brushless dc motor.

g. The ac induction motor operating at a fixed frequency is not practical for variable-speed applications. However, when driven by a variable-frequency inverter or cycloconverter, torque-speed characteristics similar to dc motors can be obtained. The induction motor has the advantage in specific power when compared to dc motors at the required electric vehicle horsepower levels. The requirement for a complex device to control frequency and voltage has restricted its use in electric vehicles.

In the case of electric vehicle builders, the overwhelming majority use the brush-type dc motor. Brushes are easy to replace and are capable of lasting 50,000 vehicle miles. Power input to the motor is usually through a single circuit requiring only variable voltage; this greatly simplifies controller complexity and weight. Some builders found it worthwhile to use a two-circuit dc motor providing control of both armature and field circuits. This provides an increase in load, velocity, and regeneration control available from the low-power second circuit of a separately-excited field.

Despite the high power density of the ac induction motor, its requirement for advanced controller technology is a severe limitation. This was demonstrated by General Motors when it used this motor for a converted Corvair, the "Electrovair." Its requirement for the three separate circuits providing both variable voltage and frequency caused the controller weight to be greater than the motor weight. Two versions of the Electrovair were built. The first required a total controller weight of about 480 pounds, compared to 160 pounds for the motor. The Electrovair II controller combined some functions by building voltage and frequency controls in the same component. This unit weighed about 240 pounds, including the cooling system.

4.2.2.2 Design Considerations

4.2.2.2.1 General Operation

A data summary required to size a dc electric motor for the hybrid vehicle contains three main criteria; these are (1) the continuous duty rating, (2) the maximum torque requirement, and (3) the rated speed. The continuous duty rating of an electric motor is influenced primarily by the steady-state heat dissipation of the motor. The continuous power rating is set for a specified torque output and a specified speed. There is also a value for efficiency which characterizes the motor at this torque and speed.

The maximum torque requirement is related to the motor capability by the motor torque constant and the motor current. For a compensated, separately excited dc motor, the torque constant is not dependent on the motor current so that the maximum torque for the motor is limited only by the current overload capability.

The last item which influences motor sizing is the rated speed of the motor. Direct current motors are considered to be low speed motors if their rated speed is between 0 and 3000 rpm. The mid-speed range lies between 3000 and 8000 rpm. The high speed range starts at 8000 rpm and progresses to about 24,000 rpm. The higher speed motors tend to have higher peak efficiencies and to be lighter and more compact than low speed motors.

Motor size is represented by weight and volume, which in turn depend on both the power and the torque requirements referred to the motor shaft. Data on motors can be used to form motor weight sizing curves by forming the ratio of the motor weight to its continuously-rated power and plotting this number versus the rated power. The motor rated speed is usually presented as a parameter on this sizing curve. Volume sizing curves can also be formed in a similar manner. Similarly, the motor efficiency can be parameterized by the rated speed and displayed against the rated horsepower. This form of data presentation on weight sizing, volume sizing, and efficiency is illustrated in Figures 4-27, 4-28, and 4-29, respectively.

Motor sizing curves can be also based on the motor torque. The ratio of the motor weight and volume to the motor starting torque at five times rated current is formed. These two ratios can then be plotted against the motor starting torques as illustrated in Figures 4-30 and 4-31.

The numbers for continuous rated motor power, starting torque, and each one of the sizing parameters are expected to bound requirements for any vehicle use mode to be investigated. The vehicle torque and power requirements are determined by aerodynamic drag, the mass of the vehicle, and the vehicle design performance specifications. Design performance factors to be considered are gradeability, cruise speed, maximum speed, and acceleration time from 0 to 60 mph. The motor weight and volume should be sized for each separate torque and/or power requirement which result from the design performance specifications. The final determination of motor size would be made on the basis of which design performance criterion is the most restrictive.

Hybrid vehicle motor performance requirements are considerably greater for the series configuration than for the parallel configuration.

All the heat engine mechanical energy must be converted to electrical energy and then back to mechanical torque at the drive wheels in the series configuration. In various parallel arrangements, mechanical energy is transferred directly to the drive wheels. Only the differences between the mechanical level transmitted and the vehicle demand level would be supplied by the motor. This allows a lower continuous duty rating for the parallel system motor, with corresponding lower weight and lower electrical losses than for the series system.

4.2.2.2.2 Regeneration Mode

In the hybrid vehicle, braking force can be supplied when kinetic energy is converted back to electrical energy through the motor (acting as a generator) and directed to recharging the battery system. The braking drag is limited, therefore, by both the short-term over-current rating of the motor and the charge acceptance rate capability of the batteries. In both hybrid configurations, the heat engine/generator system may be charging the batteries at the same time braking drag is required. The simultaneous supply of electric energy from both the heat engine/generator and motor/generator during regenerative braking can exceed the charge acceptance rate of the battery system. In many hybrid designs, the battery is kept near a full charge. Therefore, both the state of charge of the battery and its charge acceptance rate are two severe constraints to utilization of regenerative braking energy in the hybrid vehicle.

4.2.2.2.3 Cooling Requirements

Air density and the motor cooling system temperature control impacts the design, particularly under load. Allowable temperature rise is the long-term constraint that must be met by sufficient sizing or by an adequate cooling system. In the initial running period, the power output capability is restricted by the degree of thermal lag that determines transient temperatures in the motor. After continuous operation, the power is limited by heat emission rates of the motor and its cooling system heat removal capability. Cooling must be provided to ensure that the temperature rise does not exceed the temperature rating (which is dependent on the type of material used for insulation). Running the motor above this rating will shorten its operating life due to insulation deterioration and subsequent shorts. The temperature rise of the motor that would result from climbing a mountain could be a serious constraint.

4.2.2.2.4 Motor-Transmissions

A means of changing the gear ratio between the motor and the drive wheel is advisable for hill climbing and start-and-stop driving. Though dc motors can provide their highest torque at zero velocity, the disadvantage of high current required to accelerate a vehicle with a fixed gear ratio must be considered against the losses and added weight of a transmission.

4.2.2.2.5 Motor Rating and Overloads

Figure 4-32 shows the overload capability of compensated brush-type dc motors suitable in size for hybrid vehicle use. These overload limits are established both by commutation constraints (which become more severe at higher speeds) and by thermal conditions (Ref. 4-14). Notice that the "Frequently Repeated" curve in Figure 4-32 allows one minute overload of two and one-half to three and one-half times design-rated continuous duty load. This could provide adequate acceleration of a hybrid electric car to highway speeds. Bursts of power for up to 5 seconds allow three and one-third to five and one-half times the overload which is "Occasionally Repeated."

The overload capability is very important in a parallel hybrid configuration. At cruise velocity, all the power is mechanically transferred from the heat engine to the wheels. Since the motor is not supplying continuous power for cruise, it can be sized for supplying transient power, resulting in a small, lightweight unit. It was thus determined in Ref. 4-2 that for a 4000-lb passenger car, a 38 hp motor rated for continuous duty would, at a factor of three overload, provide 114 hp for acceleration to maximum vehicle speed. It can also provide a starting torque of three and one-half times the rated current torque for one minute. In the curves of Figure 4-32, continuous duty at 100 percent of rated load before overloads occur is assumed. The temperature rise of the motor in the parallel hybrid configuration would be lower because the motor is not continuously loaded.

On the other hand, the series hybrid configuration requires that all continuous and overload power be provided by the electric motor. This motor must be sized for the more rigorous requirement of continuous cruise power and is much heavier than the parallel system motor. Reference 4-2 thus calls for the use of a continuous duty rating of 61 hp in a 4000-lb passenger car for this configuration. This is the minimum input power required to provide vehicle cruise at 80 mph.

4.2.2.2.6 Cost

Motors that are suitable for automotive vehicles and the controllers to regulate them have not been mass produced to date. The resulting high cost of these powertrain elements is the greatest single obstacle to widespread use in automotive vehicles. A single compound wound, compensated motor of the type that could drive a family car has been listed in recent times in the General Electric catalog as costing $2300 to $3200, depending on the features. The controller is an additional cost. A large production base will thus be required to make these powertrain elements economically viable.

4.2.3 Control Systems

4.2.3.1 Types of Control Systems

In a comparison tradeoff of drivetrain designs, it is very important to include control system cost and weight. In terms of performance and versatility, the selection of an adequate low-cost control system is as significant as motor selection. Because all of the contending electric motors need variable voltage applied for varying speed, control and regulation must include the modification of a fixed-voltage source (the batteries). This can be achieved by means of a chopper circuit, a variable resistance in the armature circuit, or a step-voltage change combined with field control.

The chopper circuit (generally using SCRs) provides an efficient means for transforming a fixed battery voltage to a smoothly varying effective voltage matching the requirement of the motor at all speeds of operation and providing a smoothly varying speed. The chopper system provides pulse frequency variation, pulse-width modulation, or a combination of the two. While chopper control is effected by switching high currents on and off, the use of proper filtering elements can avoid the introduction of high current pulses in the battery. Compared to pure dc control, the chopper introduces higher losses during high-frequency operation, but generally has lower losses during low-frequency operation. The high-frequency losses can be partially reduced by special motor design and adequate filtering. Since the forward voltage drop of the high-current SCR is about 1 volt, 0.5 kW would be lost in the SCR at 500 amps. This power loss presents a heat dissipation problem. The higher the maximum system voltage, the lower the loss of the SCR controller system at a given motor power.

The main disadvantage of the chopper is the high cost of the power switching components and the associated control circuitry. However, if industry has the incentive for high production levels, it is projected that at some period beyond 1975, the price of high-current, high-voltage SCRs will be reduced sufficiently to make them economically viable. Recent price reductions have already made them practical for experimental vehicles. However, the SCR protection circuit and the current-smoothing filters will continue to remain significant cost factors.

A variable resistance in the armature circuit is a simple type of controller that was used on streetcars and some early electric cars. Although simple, this type of control introduces high losses because of the voltage drop across the resistance, and is an inefficient method of voltage control for a vehicle required to operate over a wide speed range.

Step voltage systems have been used in which multiple-pole relays switch batteries from parallel to series in steps as vehicle speed increases. This may be undesirable, because the discrete velocity increments may prevent one vehicle from following another vehicle at the same velocity, and the relays are constantly working under load, thereby shortening the operating life. To provide adequate voltage matching over a wide speed range, several stages of voltage switching are required to obtain reasonable motor efficiency and avoid excessive loading of the battery. The number of switching steps may be reduced by combining field control with voltage switching (i.e., by using armature current sensing to provide feedback information for controlling the field). This would control current surges and corresponding jerks.

A comparison of operational characteristics using the three types of controllers is given in Table 4-6.

Of greater complexity are the controllers for the brushless dc and the ac induction motors. Controllers for ac motors must provide not only variable voltage, but also variable frequency to the motor. At present, three-phase, variable-frequency, and variable-voltage inverters at power levels associated with electric vehicles are very expensive and heavy, because they are complex (12 SCRs or more are needed, with at least six having high current ratings). The voltage control may be incorporated into the inverter or a separate chopper may be used.

4.2.3.2 Motor Control

The motor control system for the series configuration hybrid electric vehicle is identical to that for an all-electric vehicle. However, a simple separate control of the generator field is required in the series configuration hybrid to modulate the power output from the heat engine for a fixed engine speed and convert it to electrical energy to be delivered to the motor. A more complex control logic will have to be inserted for a heat engine/generator system with variable engine speed and designed for transmitting higher levels of power to the motor, enabling the vehicle to operate at high speed on level roads or at sustained speeds on grades.

The motor controller for the parallel configuration hybrid system, by contrast, is more complex than the series hybrid because it requires a special logic to control the electric motor that is augmenting power from the heat engine. That is, the road power demand less the mechanical power provided by the heat engine must be supplied by the electric motor. A sensor and logic system must, therefore, be capable of accepting foot throttle position information and heat engine power output information to provide the difference. Therefore, it is a heavier, costlier system by virtue of the increased logic, but it is lighter and cheaper on the basis of reduced power handling requirements.

4.2.3.3 Generator Control

The type and form of control of the generator output are dependent largely on the particular hybrid configuration selected (series, parallel, etc.) and its unique operating mode. Thus, a proper assessment of the control system requirements demands a detailed review and analysis of each powertrain design, and is beyond the scope of this study. It is sufficient to state that feedback to the heat engine/generator would, at any one time, come from one of two sources: (1) the allowable battery charge current rate, and (2) the motor demand voltage acting in response to driver demand for a given vehicle speed. In either case, the generator output voltage would have to be controlled. In addition to any solid state regulation devices in the power flow circuit, overall control would be achieved by adjusting heat engine torque and speed and by adjusting generator field strength. Response rates and system stability are also pertinent concerns.

4.3 SECONDARY BATTERIES

4.3.1 General

A survey was conducted to determine the current and projected status of secondary battery technology and to assess the applicability of this technology to hybrid vehicle propulsion requirements. A large number of different secondary battery systems appear to be potentially feasible candidates for hybrid vehicle propulsion. The scope of the current effort, however, has been limited to those types of systems which are either currently commercially available or which are under development and appear to have a chance of attaining a "technology readiness" status within the next 5 years. (The term "technology readiness," as used here, is defined as a stage of advanced development where a potential user can assume that all of the fundamental design and operating problems have been solved, performance and life of prototype units have been demonstrated, and the battery is ready for confident selection for a hybrid vehicle.)

Organization and comparison of the data were made with respect to several key battery performance parameters relevant to hybrid vehicle propulsion requirements, namely:

a. Specific energy (Wh/lb)
b. Energy density (Wh/in^3)
c. Specific power (W/lb)
d. Cycle life (cycles)

For hybrid vehicles using a battery to provide power for acceleration, specific power is the most important battery performance parameter. Since the range of the hybrid vehicle is not necessarily directly related to installed capacity, specific energy is generally a less important performance parameter. However, when battery recharging is partially accomplished by use of electric power from stationary sources (rather than relying totally on the on-board heat engine), the vehicle range will be affected by the specific energy. Therefore, as specific energy is improved, more reliance can be placed on recharging the battery from electric wall outlets, and hybrid vehicles could be replaced eventually by all-electric vehicles for meeting a portion of urban transportation needs.

Manufacturing and operating costs are also important criteria to consider when selecting a battery system because the hybrid vehicle must be economically feasible and competitive with other modes of transportation.

Cycle life of a battery in determining the number of miles that can be driven before battery replacement is necessary, is the main factor affecting operating costs. The term cycle life, as used here, is defined as the number of charge/discharge cycles available from the battery before the capacity degrades to a point where it is no longer usable. Calendar life is directly related to cycle life and ideally should be at least equal to the time between major overhauls of the heat engine.

The battery systems selected for consideration here are lead-acid, nickel-cadmium, nickel-zinc, lithium-sulfur, sodium-sulfur, and lithium-chlorine.

4.3.2 Types of Batteries

4.3.2.1 Lead-Acid

Of the many types of lead-acid batteries now commercially available, the SLI (starting, lighting, and ignition) and the motive type are of prime interest for hybrid vehicle propulsion. SLI and motive power batteries can be made available in single and multiple cell units in a capacity range as high as 1800 to 2175 amp-hr (Ref. 4-15). The open circuit voltage range is 2.10 to 2.20 volts and the specific energy ranges from 9.0 to 15 W-hr/lb with a corresponding energy density range from 1.08 to 1.80 W-hr/in.^{3}

Both Tyco Laboratories (Ref. 4-16) and TRW (Ref. 4-17) have tested commercially available lead-acid batteries and they consider their performance marginal relative to hybrid vehicle requirements. Both studies identified key design factors affecting the performance of commercial batteries under the high charge and discharge mode of operation found with high performance hybrid vehicles. These are summarized as follows:

a. The high-rate charge and discharge capability required for hybrid vehicle propulsion are limited by diffusion of the electrolyte (H_2SO_4) across the electrode thickness and IR losses along the grid of the plate.

b. Cycling lifetime is reduced by high localized depth-of-discharge accompanied by high localized shedding of paste material which is caused by IR losses and heating along and down a plate, away from the terminal.

c. Positive plate density is not optimized for hybrid vehicle applications. High plate densities favor improved cycle life whereas low plate densities favor high-rate discharge capability.

d. Separator material would deteriorate rapidly under the range of temperatures and cycle conditions expected for hybrid vehicle applications.

It appears that the development of advanced lead-acid batteries will occur in the following areas:

a. Incorporation of a large number of thin plates in order to increase power density

b. Reducing resistance losses in grids to improve high charge rate capability

c. Optimization of plate density and devising manufacturing techniques to improve the uniformity of active material to increase usable capacity

d. Development of improved separator materials with low deterioration rates in order to increase cycle life.

e. Optimization of specific gravity of the electrolyte (H_2SO_4) for high rate performance.

Other developments in lead-acid batteries relevant to hybrid vehicle applications include:

a. The continued improvement of sealed, maintenance-free systems

b. The development of lightweight electrodes

c. The increased use of high-temperature, lightweight plastics as container materials

d. The development of alternate design concepts

The development of permanently sealed batteries, both by U.S. and foreign companies, involves the use of proprietary alloys such as silver, calcium, platinum, palladium, arsenic, and antimony as catalysts which are incorporated within the electrodes either as catalytic stoppers or as "third" electrodes which recombine hydrogen and oxygen to form water. This reaction eliminates the buildup of internal gas pressure created during charging or deep discharge and maintains constant battery electrolyte level, conductivity, and specific gravity which increases the service life of the battery by 50 percent to 100 percent (Ref. 4-18).

Techniques to reduce the weight of lead-acid batteries have involved the development of lightweight electrodes (such as shown in Figures 4-33 and 4-34) and the use of low density thermoplastics such as polypropylene polyethylene, polyvinylchloride, and polycarbonates for battery containers. These transparent plastics enhance the ease of monitoring electrolyte levels. They also reduce size and weight which increases specific energy and specific power of the battery. In addition, the plastics permit a greater internal area which allows additional electrodes or greater electrode surface area which increases battery energy density.

Both Gould (formerly under subcontract to TRW) (Ref. 4-19) and Tyco Laboratories considered the bipolar lead-acid battery as an alternate concept. The conventional bipolar lead-acid battery with Plante' formed plates is manufactured electrolytically by anodic oxidation of pure lead substrate to active lead oxide. Plante' plate batteries have a potential for very long life because of the corrosion resistance of the pure lead substrate and exceptional adherence of the active material to the substrate. However, because the bipolar plate has to have sufficient thickness to support the active mass and to prevent plate buckling, the conventional bipolar battery is too heavy for vehicle propulsion applications. Work at Gould and Tyco was oriented toward the development of a lightweight substrate material onto which thin layers of an active material could be formed. The Tyco approach was to develop a quasi-bipolar plate design in which the active material support is plastic and conduction through the plate is achieved by parallel conducting strips laid vertically over the top of each plate and along it. Limited testing of these experimental plates showed a 15 percent improvement in specific energy compared to a commercial SLI lead-acid battery.

The Gould approach to a modified bipolar plate design involved the development of conductive plastic substrates using vitreous carbon and various types of epoxy resins. The objective was to achieve 300 W/lb at 1 to 3 W-hr/lb. At present there is no bipolar plate development in progress either at Gould or at Tyco.

4.3.2.2 Nickel-Cadmium

Sealed nickel-cadmium batteries are widely used for space applications and are also available commercially in a range of capacities from 3 to over 60 amp-hr. They are superior to lead-acid batteries in service life, low-temperature discharge, chargeability, construction and long duration charge maintenance. Nickel-cadmium batteries have several features which are attractive relative to hybrid vehicle propulsion:

a. High-rate charging capability (≤1 C rate continuous)

b. High-discharge capability (≤60 C rate pulse, 20 C rate continuous)

c. Good cycle life

d. Moderate self-discharge

e. Maintenance-free operation

Their main disadvantage is high cost. In the U.S., commercially available nickel-cadmium battery manufacturing costs are approximately three to five times greater than those for a similar size lead-acid battery. Fabrication techniques and materials involved in the production of the electrodes and case account for the relatively high cost.

At present there are no ongoing programs in the U.S. directed towards increasing specific power and/or the specific energy of nickel-cadmium batteries. Current nickel-cadmium development is centered on increasing battery life and improved charge control techniques (Refs. 4-20, 4-21). The two primary wear-out modes in nickel-cadmium batteries are loss of electrolyte from the separator via diffusion into the plates and deterioration of the separator material. Improved separator materials (organic and inorganic) and methods to increase electrolyte in the cell are under investigation by SAFT (France) (Ref. 4-20).

New and improved manufacturing techniques are also under development. SAFT (France) has developed the "voltabloc" process for fabricating NiCd electrodes for both cylindrical and prismatic cells. The "voltabloc" electrodes are thin, sintered plate electrodes, which are coiled for cylindrical cells or in plate form for rectangular cells.

4.3.2.3 Nickel-Zinc

Relative to nickel-cadmium and lead-acid, nickel-zinc batteries are at an earlier stage of development. The potential advantages which make nickel-zinc batteries attractive for hybrid vehicle propulsion are:

a. High specific energy
b. Deep discharge capability (70-80 percent)
c. Flat discharge voltage characteristic
d. High charge rate capability
e. High discharge rate capability
f. Low projected cost (compared to nickel-cadmium and molten salt types)

At present a nickel-zinc development program is underway at Eagle-Picher Co. in Joplin, Missouri. The two and one-half year effort is being funded by the Air Force Aero Propulsion Laboratory and has the objective of achieving 33 W-hr/lb with 500 amp discharge capability, 36-month wet life, and a life of 300 cycles (Ref. 4-22).

Several nickel-zinc batteries were evaluated for hybrid vehicle operation in a study performed by the U.S. Army Electronics Technology and Devices Laboratory (Ref. 4-23). The results of the work showed that these batteries have a high discharge rate capability suitable for vehicle applications but have inadequate cycle life.

The principal failure mechanisms observed in this study were shorting and loss of discharge voltage due to high carbonate concentration in the electrolyte (potassium hydroxide) resulting from degradation of the cellulose separator material.

More recently, the Deutche Automobilgesellschaft Forschungslaboratorium (DAUG) in Germany has reported the development of a 21 W-hr/lb nickel-zinc battery (Ref. 4-24). The design feature of this battery is that calcium hydroxide is added to the zinc electrode in order to reduce the solubility of the zinc, thereby keeping the concentration of zinc ions formed during charge at a low level. The conventional nickel-zinc design incorporates a pure zinc electrode which completely dissolves during discharge. This is the source of dendrite formation and shape-change problems that limit the cycle life of conventional nickel-zinc batteries. DAUG reported the completion of over 3000 charge/discharge cycles in a half-cell configuration with a loss of capacity of less than 17 percent.

The Furukawa Battery Co., Ltd. has reported the development of a nickel-zinc prototype battery incorporating nylon separators with a service life of over 200 cycles. Furukawa projects that by 1978 a cycle life of over 1000 cycles with a specific energy of 100 W-hr/lb and cost of $2.00/lb will be achieved.

The Soviets at the Yeravan Polytechnical Institute have reported that one small-scale prototype passenger van powered by nickel-zinc batteries has attained a cycle life of over 200 cycles at deep discharge and 800 cycles at nominal discharge of from 25 to 50 percent.

Energy Research Corporation, Bethyl, Conn., has reported the development of a sealed nickel-zinc cell incorporating an inorganic separator to increase cycle life, and an auxiliary electrode (oxygen recombination electrode) to attain increased cycle life by maintaining a zinc-oxide reserve (Ref. 4-25). In sealed systems approximately 200 cycles have been obtained at 100 percent depth of discharge.

ESB, Incorporated has reported the development of a milling technique for fabricating the positive nickel electrode. This technique is expected to significantly reduce the cost of the conventional sintered nickel electrode (Ref. 4-26).

Although some of the basic problems appear to have been solved, a cycle life sufficiently long for hybrid vehicle application remains to be demonstrated.

4.3.2.4 Molten Salt

Theoretically, one method of electrochemically obtaining a specific energy in excess of 500 W-hr/lb is to combine the lightest weight oxidizing materials such as oxygen, air, the halogens, or sulfur with lightweight reducing materials such as lithium and sodium.

Several laboratories in the U.S., Japan, United Kingdom, France, West Germany, Czechoslovakia and USSR are now working on batteries using various combinations of the above mentioned and other materials. In general these batteries operate at temperatures greater than 550°F and contain a molten salt type electrolyte which is compatible with the highly reactive materials.

4.3.2.4.1 Lithium-Sulfur

The theoretical specific energy of the lithium-sulfur system operating at approximately 750°F is 700 W-hr/lb and it has an open circuit voltage of 2.25 volts (Ref. 4-18). Since, typically, only an average of 15 to 20 percent of the theoretical capacity is obtained in operational systems, it appears that one can expect between 100 and 140 W-hr/lb from this system.

Work at Argonne Laboratory has involved the development of cells with electrodes of solid lithium-aluminum alloy and solid metal sulfides (LiAl/FeS) with a molten (at 665°F) electrolyte of LiCl-KCl eutectic (Ref. 4-27). Boron-nitride is the separator material. A prototype of this cell concept for automobile propulsion is shown in Figure 4-35.

Various material problems associated with stability of FeS and FeS_2 type electrodes remain to be solved. Cells having specific energies in the range between 55 and 70 W-hr/lb have been built and tested. One cell has demonstrated 250 charge/discharges.

4.3.2.4.2 Lithium-Chlorine

Advanced battery research and development at General Motors has succeeded in reducing the operating temperature of the lithium-chlorine cell from 1200°F to 800°F by using an unconventional electrolyte consisting of a molten salt mixture of lithium-fluoride, lithium-chloride, and potassium-chloride. The electrolyte used in the conventional lithium-chlorine cell is molten lithium-chlorine requiring an operating temperature of 1200°F which causes severe corrosion and thermal expansion problems. The GM cell under test delivered a specific energy of 150 W-hr/lb and a specific power of 200 W/lb. However, this cell failed after 600 hours of operation.

Lithium-chlorine research efforts at Standard Oil Co. have been directed toward the development of a cell that uses fused lithium-chloride/potassium-chloride as the electrolyte. This cell operates in a temperature range between 662°F and 932°F. However, temperature instability is a problem; the cell tends to operate near the high end of the temperature range. The projected specific energy for this cell is 70 W-hr/lb at a specific power level of 100 W/lb.

4.3.2.4.3 Sodium-Sulfur

The original development work on the sodium-sulfur battery occurred at Ford Motor Company in 1966 and at a time when laboratory cells demonstrated a specific energy of between 90 and 100 W-hr/lb, a cycle life of 250 to 500 cycles and an operational temperature range of from 575°F to 660°F. The Ford battery uses molten electrodes (sodium anode and sulfur cathode) and a solid ceramic electrolyte (beta-alumina) which allows the passage of sodium ions from the molten sodium electrode to the molten sulfur electrode as shown in Figure 4-36. The cell configuration is shown in Figure 4-37.

An inherent advantage of the sodium-sulfur battery is that no gases are evolved during charge or discharge which allows the battery to be permanently sealed in a stainless steel container. The low cost and availability of the battery reactants is another potential advantage. However, a major problem with sodium-sulfur batteries is the chemical and thermal stability of the solid beta aluminum (Na_2O $11Al_2O_3$) separator and seals.

Toshiba, Toyota Motor Co., and Yuasa are jointly sponsoring the development of an advanced sodium-sulfur battery concept shown in Figure 4-38. A cycle life of 500 cycles for a prototype 10 kW power pack has been demonstrated. Actual road-testing of a sodium-sulfur battery-powered van has been achieved in the United Kingdom. A four-year development program by Chloride Silent Power (United Kingdom) is under way with the goal of producing commercially available sodium-sulfur batteries for electric buses and vans (Ref. 4-28).

TRW has a sodium-sulfur battery program oriented toward establishing basic technology for both utility load leveling needs and vehicular propulsion applications. Performance parameters are 50 W-hr/lb and 50 W/lb with a capability of 100 charge-discharge cycles (Ref. 4-29).

4.3.3 Performance Comparison of Candidate Systems

The performance and design characteristics of candidate battery systems are summarized in Table 4-7. Although the raw material cost is relatively low for the molten salt systems the technology of these systems has not been developed to a level that allows a realistic estimate of fabrication cost.

Specific energy and specific power of the candidate batteries are plotted in Figure 4-39. Also shown are requirements for hybrid and electric vehicles. The conventional hybrid vehicle requirements are based on the results of a previous study (Refs. 4-2 and 4-35) and reflect vehicle acceleration capabilities which are comparable to conventional gasoline engine-powered vehicles. The second set of points plotted for the hybrid reflect a reduced vehicle acceleration capability (holding vehicle weight constant). The vehicle acceleration capabilities assumed for this analysis are given in Table 4-8. Requirements for an all-electric vehicle (Ref. 4-36) are included to illustrate the point that battery requirements for the two types of vehicles are not necessarily congruent. In addition, the installed battery capacity requirement for an electric vehicle will be significantly greater than that for a similar weight hybrid vehicle that relies totally on the on-board heat engine for recharging the battery.

Several inferences can be drawn from the results plotted in Figure 4-39. First, it is clear that lead-acid batteries cannot meet either the specific energy or specific power requirements derived from Refs. 4-2 and 4-35. Second, the requirements of the van and buses having conventional acceleration capabilities can be met by either nickel-zinc or nickel-cadmium batteries. Third, it appears that by reducing the acceleration capabilities of the family and commuter car, the required specific power and energy requirements for the vehicle can be reduced to a level which is consistent with the objectives of current nickel-zinc development programs. Similarly, the energy and power requirement for a modified intracity bus can be almost met by advanced lead-acid batteries. Finally, while important for the development of electric vehicles with extended range capability, the development of molten salt batteries having relatively high specific energy does not appear to be crucial for the development of a hybrid vehicle with performance requirements given in Refs. 4-2 and 4-35.

4.3.4 Summary of Battery Review

All of the candidate battery systems except for lead-acid appear to be feasible for hybrid vehicle propulsion. Further, the specific energy and specific power requirements for hybrid vehicles having reduced acceleration capability can be met using either nickel-zinc or commercially available nickel-cadmium batteries. Because nickel-cadmium batteries are relatively expensive, their use may be limited to hybrid vans and buses. The problem of most concern with nickel-zinc batteries is increasing the cycle life to a practical level.

4.4 FLYWHEELS

Two major investigations of flywheel systems for hybrid vehicles have been conducted by Lockheed Missiles and Space Company and Johns Hopkins University, under federal government sponsorship. These efforts are briefly reviewed, along with a parametric analysis that was conducted under private sponsorship by General Electric Company.

4.4.1 Lockheed Missiles and Space Company

Concurrent with the hybrid vehicle applicability and configuration studies conducted by Lockheed for EPA (see Section 3.2.1), flywheel design studies were also conducted prior to fabricating and testing candidate flywheels (Ref. 4-3). Six basic flywheel geometries were considered by Lockheed in the preliminary design studies for automobiles and buses. These included the pierced uniform disc, an unpierced uniform disc, a constant stress disc, a truncated conical disc, a rim-type flywheel, and the bar-type configuration.

Only those materials that could be obtained in mill-run quantities were considered for the flywheel design studies. Eleven materials were chosen on the basis of high strength or low cost, as shown in Table 4-9. The normalized cost represents the cost for each material to provide equivalent energy storage capability for a given flywheel configuration. From

Table 4-9, the most cost-effective materials are seen to be E-glass, S-glass, and 4340 grade steel. The filamentary composites seem to be readily applicable to only the bar-type flywheel geometry. These filamentary composites might be used in a rim-type flywheel, but in Lockheed's opinion the web attachment and balancing requirements appeared to present significant problems. The 4340 grade steel was felt to be an excellent candidate material for application to low cost flywheels in a disc configuration.

A total of 24 designs were found to be suitable based on the design capacity of 0.5 kW-hr for a hybrid family car and 1.0 kW-hr for a hybrid city bus. These are summarized in Table 4-10. The optimum design for the family car from the standpoint of minimum assembly cost, size, and weight is seen to be the constant stress disc of 4340 grade steel. The cost shown represents the projected assembly cost in automotive quantities after amortization of necessary tooling. On the basis of this preliminary screening, two 46-pound, 20.4-inch diameter, 24,000 rpm test flywheels were fabricated from 4340 grade steel.

Similarly, the optimum flywheel for the city bus appeared to be the 4340 grade steel flywheel in the constant-stress disc configuration. However, it was decided by Lockheed that little additional information would result from fabrication and test of the 1.0 kW-hr flywheel in essentially the same configuration and of the same material as chosen for the 0.5 kW-hr flywheel. Therefore, it was decided to fabricate a bar flywheel of S-glass filamentary composite bonded with epoxy.

Tests on the first steel flywheel were conducted to verify windage losses and to determine the specific energy at disintegration speed. In the test-to-failure, the flywheel was accelerated until disintegration occurred at 35,590 rpm. This represented a peripheral velocity of 3170 fps, a specific energy of 26.1 W-hr/lb for the flywheel rotor, and a total stored energy of 1.1 kW-hr, which exceeded the design specification by a factor of about two. The failure may have been initiated by an occlusion on the surface of the flywheel. (Tests for verification of momentum transfer and containment of fractured wheels are described in Ref. 4-37.)

On the second steel flywheel, specific power tests verified that at least 5000 W/lb of usable power could be extracted at various speeds.

Spindown tests were conducted in the test pit evacuated to 0.2 mm Hg. After 20 minutes the flywheel speed was down to 8700 rpm. The major sources of drag between 24,000 and 20,000 rpm were attributable to the bearings and brake disc. Bearing losses were high because of oil flooding of the bearings.

Two S-glass bar-type flywheels with a design speed of 20,000 rpm and with an energy storage capacity of 1.0 kW-hr were tested. In the test of the first glass flywheel, disintegration of flywheel and hub occurred prior to reaching the design speed. The maximum energy storage capacity at failure (15,070 rpm) was 0.568 kW-hr. The energy storage test of the second glass flywheel resulted in disintegration at 14,690 rpm. Again, the flywheel and hub were destroyed.

4.4.2 Johns Hopkins University - Applied Physics Laboratory

Experimental work conducted by Johns Hopkins University (JHU) was directed toward a demonstration of the "superflywheel" concept in which specific energy of 30 W-hr/lb for flywheel rods or bars could be achieved (Ref. 4-38). Spin tests of small-diameter composite rods (up to 0.25 inches) and filamentary single strands were conducted in which the candidate materials exceeded the goal of 30 W-hr/lb. Follow-on tests of 30-inch-long rods or bars achieved 82 to 94 percent of the desired goal.

Results of the 30-inch-long, small diameter rod tests conducted by JHU are summarized in Table 4-11. The boron filaments were tested in a spin fixture powered by a motor having a rated speed of 39,000 rpm. This was insufficient to fail the boron rods, all of which were intact after spindown. All other materials were tested to failure.

Based on the test results of the small rods, 1-lb bars were fabricated of S-glass epoxy and graphite/epoxy composites. The S-glass was selected on the basis of low cost (~$1.00 per pound). The filamentary graphite was selected over boron because it has equivalent strength and a potential for future price reduction. Results of the test are summarized in Table 4-12. The first test, as indicated, was a facility checkout run using an available E-glass bar. Because of leaks in the vacuum system (attributable to the turbine drive spindle seal) it was not possible to achieve the desired vacuum levels. The highest pressure reached during the tests was 25×10^{-2} Torr. This was felt to be satisfactory for a short-duration test because the bending stresses induced in the rod by aerodynamic drag were less than 50 psi.

The failure of the bars to achieve the desired energy storage capacity of 30 W-hr/lb were attributable to several factors. Among these were inability to (1) maintain uniform fiber alignment and distribution in the plastic materials, and (2) achieve more uniform properties in the fiber and the plastic.

4.4.3 General Electric Company

In a recent study conducted at General Electric Company (Ref. 4-39), comparative analyses were made between the isotropic flywheel materials examined by Lockheed, the undirectional fiber composites evaluated by Johns Hopkins, and other types of energy storage devices (i.e., batteries). The specific energy analyses included allowance for other subsystem weights and the cost analyses included the vacuum pump, the housing, bearings, and seals. Results shown in Figure 4-40 are presented both with and without allowance for a containment ring. The specific energies are quite low even for the advanced reinforced plastic composite materials. Of these, Kevlar shows superior performance capabilities, but it is more costly than E-glass or S-glass.

For comparison, Ref. 4-39 noted that commercial lead-acid batteries provide about 20 W-hr/$, 12 W-hr/lb, and 2 kW-hr/ft^3 when rated for charge and discharge times of about 10 hours. Ni-Cd batteries provide about 1 to 1.5 W-hr/$ and 15 to 22 W-hr/lb. The development goal for Na-S batteries is about 60 W-hr/$ and 60 W-hr/lb.

4.4.4 Summary of Flywheel Investigations

The capability of the flywheel as a competitive energy storage system is dependent upon several factors: the specific energy of the flywheel system, the bearing power losses and other losses, and safety conditions.

The energy storage of the steel flywheel, as typified by the Lockheed studies had a design capacity of 0.5 kW-hr and achieved 1.1 kW-hr when tested to destruction. On the basis of flywheel rotor weight alone, this would amount to a design capacity of approximately 6 W-hr/lb for the 86 pound, 13.06-inch diameter rotor selected by Lockheed for the baseline configuration. However, when coupled with the bearings, seals, housing and containment ring, the flywheel assembly weight of 186 pounds reduces the effective specific energy to less than 3 W-hr/lb. This would indicate that the fiber composite flywheel could have a considerably greater potential since the containment housing required for the steel flywheel will in general, result in a reduction of 50 to 60 percent in the effective specific energy when the entire system is taken into consideration.

However, the experience of Johns Hopkins indicates that while considerably higher specific energy capacities are possible with the composite flywheel, fabrication capability needs to be developed to ensure uniformity in (1) properties of the material and (2) fiber distribution in the plastic matrix and fiber alignment.

Spindown losses are of particular significance in their impact on the energy requirements of the hybrid vehicle. This requires the careful design of seals and bearings, and forces the designer to consider smaller-diameter, lower speed rotors along with lower-pressure atmospheres in the housing. Power absorbing accessories in the form of vacuum pumps or oil pumps for bearing lubrication would also significantly reduce the effectiveness of the system.

4.5 TRANSMISSIONS

4.5.1 General

The transmission converts energy output of the engine to useful levels of torque at the vehicle wheels. Ideally, the transmission should have the following characteristics:

a. High efficiency over the normal operating range
b. Control simplicity for optimum performance
c. Low volume for compactness
d. Low noise
e. Low specific weight
f. Reverse-power and braking capability
g. Capability of absorbing road shocks
h. Low power consumption during engine start and at idle.

The operational and economic feasibility of a hybrid system depends in large part on the above features and on a reasonable low cost for the transmission.

Several types of continuously variable transmissions were investigated for possible application in the heat engine/flywheel vehicle powertrain. Included in these are the mechanical (straight gear ratio), the hydrostatic, the multiple V-belt (variable pitch sheaves), the traction drive and the hydromechanical transmission. Because of numerous limitations (size, weight cost, durability, and/or efficiency), the most promising candidates appear to be either the traction drive or the hydromechanical transmissions.

The selection or design of a transmission for a hybrid system will depend primarily on the type of hybrid powertrain arrangement, the energy storage method used, and the relative speed between various powertrain elements. In a heat engine/flywheel hybrid, the transmission links three main components: engine, flywheel, and drive wheels. Figure 4-41 shows the power connection for both parallel and series hybrid configurations. In the parallel system shown, power can be delivered to the drive wheels directly from the engine or through a flywheel. Because the flywheel spins at very high speeds, unique forms of wide speed range transmissions are required.

In a heat engine/battery hybrid, the electric drive motor acts as the transmission for the series configuration, and the transmission for the parallel configuration can be simply a gear train without need of a torque converter. Hence, the discussion of this section addresses only the subject of unique, more complex transmissions for heat engine/flywheel hybrid vehicles.

4.5.2 Parallel Configuration Operation

Under EPA sponsorship two studies were conducted on transmission designs for a parallel configuration flywheel hybrid system. These studies were conducted by Sunstrand Aviation (Ref. 4-40) and Mechanical Technology, Inc. (MTI) (Ref. 4-41). Both studies examined (1) the development of total energy transfer systems from the heat engine to the drive wheels and (2) the management of the energy storage system.

As shown in Figure 4-41, for parallel operation, two types of transmissions are required. From flywheel to the load an infinitely variable transmission is needed (Ref. 4-40). Generally, a standard three-speed transmission is adequate between the engine and the full load.

4.5.2.1 Sundstrand Aviation Study

The Sundstrand study assessed the practicality of a transmission for use in a heat engine/flywheel hybrid system for a full-size family car. In this study, a number of possible types of links between the engine, flywheel, and drive wheels were analyzed: mechanical, hydrostatic, and hydromechanical.

An infinitely variable hydromechanical transmission was selected between the flywheel and the drive wheels. For the engine-flywheel transmission, the engine speed was fixed at each power level to ensure operation near minimum SFC or minimum emission levels. A fixed speed ratio between the engine and the flywheel was not sufficient and, hence, a transmission was required. Sundstrand selected a combination mechanical, hydromechanical, and hydrostatic transmission system for links 1, 2, and 3, respectively (Figure 4-41). This transmission, called Baseline 8A, is made up of a five-element differential, several hydraulic units (variable and fixed displacement), clutches, controls, and associated gearing. By controlling the displacement of the variable hydraulic unit, it is possible to control the reaction torques in the five-element differential. By controlling these torques, it is possible to control the direction of power flow and hence extract energy from the flywheel and supply this energy to the "output" or to take energy from the "output" and supply it to the flywheel, as required.

In considering the Federal Emissions Test Driving Cycle, it was discovered that with the Baseline 8A transmission the engine was not running continuously at minimum fuel consumption conditions. But the required engine speed versus vehicle speed characteristics for minimum SFC could be very closely approximated by putting a clutch on the hydraulic unit input, such that at light accelerator pedal loads below 50 mph, the engine power input comes into the variable hydraulic unit (V-unit), and at heavy accelerator pedal loads below 50 mph, or at any load above 50 mph, the engine power input comes directly into the differential gear set. This arrangement, the alternate 8C transmission, shown in Figure 4-42, allows the engine to run at slower and more economical speeds at the slower road speed and lighter load conditions, but allows higher engine speed operation (when the engine would otherwise be power limited) at the higher load and/or higher road speed conditions.

On the basis of fuel consumption calculations utilizing ideal energy storage versus ideal nonenergy storage transmission systems, Sundstrand concluded that the amount of energy available for storage and reuse, as regenerated on deceleration by a light vehicle operated over the Federal Emissions Test Driving Cycle, was relatively small. This fact was reflected in Federal Emissions Test Driving Cycle fuel consumption calculations made for the Baseline 8A and Alternate 8C transmission and for a typical three-speed automatic. These results, shown in Table 4-13, indicate that the Baseline 8A system has a poorer fuel economy than the Alternate 8C system, and that both of these have poorer fuel economy than the standard transmission when transmission losses are estimated and included in the energy computation ("real" case).

The results of constant speed fuel consumption calculations for the two hybrid storage/transmission systems compared with results for a typical three-speed transmission are shown in Table 4-14. As expected, transmission 8C has fuel economy that is superior to transmission 8A up to approximately 50 mph. It can also be seen that the three-speed automatic transmission has better fuel economy below 50 to 60 mph. Above this value, the two hydromechanical flywheel transmissions exhibit superior fuel economy. These results largely reflect the fact that the flywheel transmissions are configured to permit the engine to operate at or near minimum SFC at higher speeds.

In conclusion, Sundstrand stated that:

a. A combination of mechanical, hydromechanical, and hydrostatic transmissions is a practical means of providing power for the flywheel, heat engine, and drive wheel links.

b. The selected transmission provides an infinitely variable ratio between the flywheel and the vehicle wheels, and a nonlinear ratio (fixed by vehicle speed) between the heat engine and flywheel. Although the engine speed is not independent of the flywheel speed, it does operate near its minimum SFC line.

c. The specified spark ignition heat engine with the selected transmission has a greater computed fuel consumption over the Federal Emissions Test Driving Cycle than that of a typical three-speed automatic transmission. Cruise fuel consumption is greater than for the three-speed automatic below 50 mph and less above this speed.

d. The theoretical fuel economy benefits that can be gained from the flywheel energy storage concept over a "light duty" cycle such as the Federal Emissions Test Driving Cycle is minimal because of the small amount of energy available for storage and reuse. In fact, when the "cost" of storage in terms of power loss is included, there is no benefit. The more "severe" the acceleration/braking duty cycle relative to maximum vehicle capability and the heavier the vehicle, the greater are the benefits derived from the flywheel energy storage concept.

4.5.2.2 Mechanical Technology, Inc. Study

The study performed by MTI basically arrived at the same conclusions as Sundstrand. Mechanical Technology proposed a power splitting transmission (Ref. 4-41). This transmission is an infinitely variable, step-less unit that obtains torque multiplication and control by hydraulic principles. It is intended for use in a medium-size automobile.

The unit differs from the torque converter or fluid coupling hydrodynamic-type transmissions in that the power in the hydraulic circuit is transferred by fluid static pressure at low flow rates, whereas the hydro-dynamic unit uses high flow rates and the inertial motion of the fluid to transfer power. Basically, the transmission system consists of a flywheel planetary gear train, hydraulic variable displacement elements, connecting drive gears, an output planetary gear train, and a control system.

As shown in schematic form in Figure 4-43, three planetary gear trains are used in the assembly to (1) provide a power path for the flywheel, (2) direct the output power when the vehicle is in the high-ratio range, and (3) provide a low-ratio range power path.

The overall transmission consists of two separately controlled split-path hydrostatic links: the primary path that establishes a given ratio between the engine and vehicle for optimum torque-speed loading of the engine in a steady-state mode, and the secondary path that controls the direction and magnitude of power flow to and from the flywheel during vehicle velocity transient. In each of the primary and secondary sections, the hydrostatic transmission consists of two identical positive displacement units - Elements I and II for the primary, and Elements III and IV for the secondary circuits.

The output torque is a function of the hydraulic pressure and the displacement of the hydrostatic units. If the torque increases, the displacement or pressure of the units must increase.

The secondary or flywheel drive section operates over a relatively small speed ratio and only operates for short bursts of power. The major power to and from the flywheel is transmitted by the planetary. The hydrostatic drive functions on both sides of the mechanical drive serve only as a positive speed control.

Figure 4-44 shows a comparison between the efficiencies of powertrains for flywheel hybrid and the standard automobile for cruise operation. As shown, the efficiency of the flywheel/hybrid powertrain is substantially lower even though the transmission efficiency for the hybrid transmission is higher (Figure 4-45). The fuel economy of the hybrid automobile compared to the standard automobile is poor up to a cruise speed of 50 mph, but at higher speeds it has superior fuel economy.

4.5.3 Series Configuration Operation

In the series configuration for the hybrid heat engine/flywheel as discussed in Ref. 4-38, the engine drives the flywheel and the flywheel drives the car. In such a scheme, an infinitely variable transmission is required between the engine and the flywheel as well as between the flywheel and the load. Mechanical Technology and Sundstrand selected a hydromechanical infinitely variable transmission for this configuration.

4.5.4 Other Transmission Designs

There are several other types of transmissions that have been evaluated. In an EPA-sponsored program, "Automobile Gas Turbine Optimization Study," several contractors studied transmission systems for use with gas turbine automotive powerplants. The single-shaft gas turbine requires an infinitely variable speed transmission. Traction and belt transmissions are two types of infinitely variable speed transmissions discussed in Ref. 4-42 and summarized in Ref. 4-43. These transmissions, which are candidates for use in the hybrid vehicle, are briefly discussed in the following paragraphs.

4.5.4.1 Traction Transmission

Traction transmissions are not currently commercially available for large power output devices. A recent company development effort in this area by Tracor, Inc., has resulted in the design of a special metal traction device for transmitting torque at the high power levels associated with automotive drives (Ref. 4-42).

The Tracor design uses toroidal discs and rollers, special hydrostatic thrust bearings, and a specially prepared lubrication oil (Monsanto's Santotrac 30). Roller position is controlled so that the transmission can operate in a speed step-up, or in a speed step-down, or in a direct-drive mode. According to Tracor, the favorable features of the traction transmission include low noise, high-speed operation (up to 10,000 rpm input speed), compact size, a wide-speed range capability, and comparatively low cost.

The estimated efficiency of the Tracor traction transmission serving a 250-hp engine is shown in Figure 4-46. Over a wide range of vehicle speed, the transmission efficiency is between 80 and 90 percent, somewhat lower than the performance of the hydromechanical system shown in Figure 4-45.

4.5.4.2 Belt Transmission

The belt drive is used for high-torque transmissions. It benefits from a recent development in high-strength rubberized composites and is based on a unique bent-axis concept that affords nearly optimum design (Ref. 4-44). Estimated efficiency versus speed performance of this type of transmission is presented in Figure 4-46. As can be observed, the belt drive transmission has very high efficiency (substantially higher than the toroidal traction drive). In general, the belt drive transmission is presently limited to lower absolute speeds than the toroidal traction system. Further development of this system would be required to match its characteristics for several of the candidate heat engines considered for use in the hybrid system.

4.6 OTHER ENERGY STORAGE SYSTEMS

4.6.1 Hydraulic Accumulator

The hydraulic accumulator used as an energy storage device for hybrid vehicle application has not yet been tested on a vehicle; however, several laboratory models have been tested with various degrees of success.

Hydraulic drive systems discussed in the literature use an all-hydraulic system (Figure 4-47) and do not mix hydraulic and electrical components. It appears more efficient to take full advantage of a hydraulic design that can run the vehicle wheels directly, eliminating the need for a vehicle transmission and mechanical brakes. In addition, regenerative braking can be obtained without any additional equipment. This is especially advantageous to vehicles which travel in heavy stop-go traffic patterns such as delivery vans and city buses.

A typical system taken from Ref. 4-45 is shown in Figure 4-48. There, the heat engine, A, is a Sachs Wankel which drives two hydraulic pumps, B and C. The system contains two accumulators, F and G, and two motors, D and E. Table 4-15 gives the results of a computer simulation. It should be noted that the predicted high fuel economy is mostly the result of energy savings through use of regenerative braking.

A major disadvantage of hydraulic accumulators is that they are bulky. In addition, when energy is rapidly provided to the system, as during regenerative braking, the gas tends to heat up under compression (Refs. 4-46 and 4-47). Subsequent cooling reduces the system pressure, thereby extracting a portion of the stored energy. This is depicted in Figure 4-49. It is seen that adiabatic compression makes hydraulic accumulator efficiency a time-dependent quantity. Attempts to minimize this problem by introducing heat regenerator devices to keep the compressed gas hot are reported in Refs. 4-46 and 4-47. Plastic insulating foam in the gas well tends to maintain the gas temperature, thereby minimizing the losses as depicted in Figure 4-49. A similar solution is proposed in Ref. 4-47, where metal strands act as heat regenerators. In Ref. 4-46, it is claimed that the work loss is reduced from 15 to 25 percent with a conventional system down to 1.4 percent with the foam added.

Another means to reduce adiabatic compression loss is by use of gases with low specific heat ratio such as Freon 13 or 13B1. The use of condensible mixtures like a CO_2-N_2 mixture is discussed in Ref. 4-47. All these schemes attempt to replace adiabatic compression by isothermal compression to offset the time-dependency associated with heat loss through the walls of hydraulic accumulators.

References 4-48 and 4-49 discuss hydraulic system feasibility. References 4-50 and 4-51 give an evaluation of a hydraulic hybrid drive system using a steam engine as the powerplant.

4.6.2 Electrical Capacitor Storage Systems

These systems, at least with current state-of-the-art capacitor technology, would have to be very large in volume to store the necessary energy. Practical energy densities in capacitors are about 0.0001 of that of a good battery. Because of high internal energy leakage rates, energy storage in very high voltage capacitors could only be for brief periods. Even if they could be built, the power transfer efficiencies would probably be very low.

4.6.3 Pneumatic Energy Storage Systems

These systems (not involving a liquid system, as in a hydraulic accumulator) are inherently inefficient because of the large work required to pump gases to high pressures.

4.6.4 Thermal Energy Storage Systems

These systems are subject to the same large thermodynamic efficiency losses suffered by a heat engine during initial energy generation. No other temporary energy storage schemes have been proposed.

4.6.5 Fuel Cell Battery Systems

These systems have been proposed, but are subject to the limitations imposed by the large volume, high cost, and limited lifetime of current fuel cell systems.

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TABLE 4-1. GAS TURBINE ENGINE/VEHICLE BASIC CHARACTERISTICS (Ref. 4-1)

Manufacturer	Type of Engine	hp	Peak Operating Speed, rpm	Dimensions, in. Length	Width	Height	Specific Weight lb/hp	Design SFC lb Fuel/ hp Hr
Ford	Three Shaft with Intercooling, Re- heat, Recuperator		46,500/91,500/ 37,500[2]					
		320	45,300/38,200[3]				5.31	0.41
	Regenerative Cycle with Ceramic Regenerator	450	37,500/31,650	40.8	34.6	41.8	3.78	0.41
		525	37,500/31,650				3.23	0.43
GM	Free Turbine	225	44,000/ ?	?	?	?	?	0.55[4]
	Regenerative Cycle with Metal Regen- erator	225	33,000/27,000	37	26.4	24	2.68	0.54
		280	-	36	30	36	3.4	0.45
		325	37,100/ 30,830	47	28	42	5.23	0.475
AiResearch	Single Shaft Regen- erative Cycle with Metal Regenerator	155[1]	70,230	-	-	-	2.2	0.41
	Single Shaft Recu- perative Cycle with Ceramic Recuperator	125[1]	83,600	-	-	-	3.2	0.43
	Free Turbine Re- generative Cycle with Ceramic Regenerator	175[1]	67,500/55,200	-	-	-	2.5	0.42
United Aircraft	Simple Cycle Single Shaft	150[1]	130,000	17	-	19	1.07	0.51
	Simple Cycle Free Turbine	150[1]	130,000/40,000	21	-	17	1.33	0.51
	Regenerative Cycle with Regenerative Single Shaft	163[1]	112,000	26	-	21	1.78	0.45
	Regenerative Cycle with Regenerative Free Turbine	166[1]	66,500/71,300	27.5	-	28	2.95	0.53
Williams Research	Regenerative Cycle with Ceramic Regenerator	80	61,000/51,000	24	24	16	3.13	0.59
	Regenerative Cycle with Regenerative Free Turbine	130[1]	-	-	-	-	1.4	0.40
Chrysler	Regenerative Cycle with Regenerative Free Turbine	150	44,600/45,700	33	27.5	29.5	4.42	0.60
Rover	Recuperative Cycle with Metal Recuperator	150	65,000/36,000	38	30	27	3.17	0.55
Fiat	Simple Cycle Free Turbine	200	29,000/22,000	-	-	-	2.90	0.95
Volvo	Single Shaft Regenerative Cycle	250	43,000	51.2	26.4	29.5	3.26	0.40
Boeing	Simple Cycle Free Turbine	400	-	39	27	24	1.0	0.75
GE	Regenerative Cycle with Ceràmic Regenerator	150[1]	40,000	34.5	26.5	24.1	4.79	0.65
	Recuperative Cycle with Ceramic Recuperator	150[1]	80,000/50,000	31.5	27.0	24.5	4.57	0.50
Solar	Recuperative Cycle with Metal Recu- perator Free Turbine	300	36,400/31,000	50	36	41	5.3	0.60

[1] Design Engine, not built or under test

[2] Three speeds shown for low pressure spool, high-pressure spool, and free turbine, respectively

[3] Two speeds shown for dual-shaft (free turbine) engines; first figure for first stage turbine, second figure for power turbine

[4] Experimental test bed engine.

TABLE 4-2. GAS TURBINE CYCLE ANALYSIS RESULTS (Ref. 4-2)

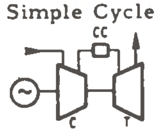
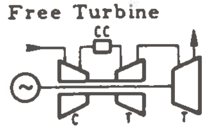
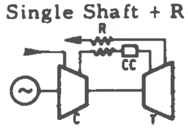
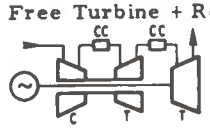
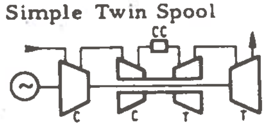
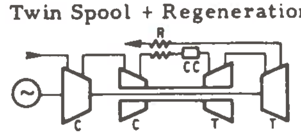
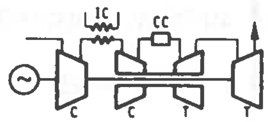
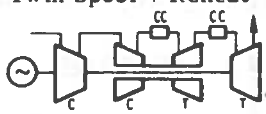
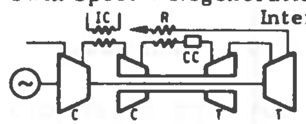
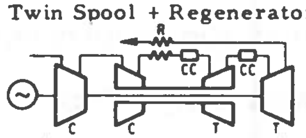
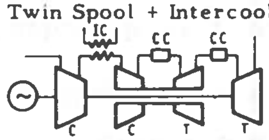
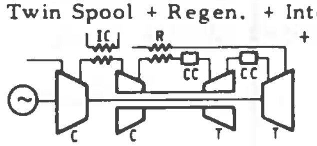
Cycle	Thermal Efficiency	BSFC, lb/bhp-hr	Specific Output, hp/lb (air) min	Exhaust Volume Flow, ft^3/hp (rated)	Low Speed Idle	
					Turb. Inlet Temp., °F	Fuel Consumption, Percent of Maximum
1. Simple Cycle 	0.11	1.24	0.94	44.2	703	18.2
2. Free Turbine 	0.11	1.24	0.94	44.2	703	18.2
3. Single Shaft + Regeneration 	0.33	0.42	0.84	24.5	1035	11.9
4. Free Turbine + Reheat 	0.10	1.43	0.97	48.6	703	15.8
5. Simple Twin Spool 	0.15	0.93	0.97	34.2	703	22.0
6. Twin Spool + Regeneration 	0.16	0.84	0.89	30.6	1035	6.0

TABLE 4-2. CONCLUDED

Cycle	Thermal Efficiency	BSFC, lb/bhp-hr	Specific Output, hp/lb (air) : .m	Exhaust Volume Flow, ft^3/hp (rated)	Low Speed Idle Turb. Inlet Temp., °F	Low Speed Idle Fuel Consumption, Percent of Maximum
7. Twin Spool + Intercooling 	0.18	0.74	1.59	21.0	703	30.4
8. Twin Spool + Reheat 	0.15	0.88	1.37	28.9	703	25.6
9. Twin Spool + Regeneration + Intercooling 	0.31	0.44	1.52	18.8	1035	11.4
10. Twin Spool + Regenerator + Reheat 	0.21	0.64	1.23	23.0	1035	7.8
11. Twin Spool + Intercooling + Reheat 	0.17	0.80	1.74	21.7	703	28.1
12. Twin Spool + Regen. + Intercooling + Reheat 	0.29	0.47	1.65	12.6	1035	10.7

Key: (~) Generator; C Compressor; CC Combustion Chamber; T Turbine; R Regenerator; IC Intercooler

TABLE 4-3. GAS TURBINE CHARACTERISTICS ASSUMED IN CYCLE ANALYSIS (Ref. 4-2)

Compressor Inlet Pressure	14.7 psia
Inlet Temperature	$80^{\circ}F$
Pressure Ratio/Stage	2.8
Stage Efficiency	0.80
Turbine Inlet Temperature	$1600^{\circ}F$
Stage Efficiency	0.83
Mechanical Efficiency	0.95
Combustor Efficiency	0.98
Pressure Drop	5 Percent of Inlet Absolute Pressure
Regenerator Effectiveness	0.95
Pressure Drop	2.5 Percent Inlet Absolute Pressure/Leg
Intercooler Effectiveness	$10^{\circ}F$ Above Ambient
Pressure Drop	2.5 Percent Inlet Absolute Pressure
Turbine Weight Flow = Compressor Weight Flow	
Specific Heat	1.395
Fuel Lower Heating Value	18,700 Btu/lb

TABLE 4-4. GAS TURBINE ENGINE EXHAUST EMISSION DATA OVER THE FEDERAL DRIVING CYCLE (Ref. 4-1)

Engine/Manufacturer	Emissions (gm/mi)			Remarks
	CO	HC	NO_x	
Conventional Combustors				
United Aircraft Research Laboratories RGSS-6	0.53	0.15	2.72	Calculated values over simulated Federal Driving Cycle, based on emission data from GM engine GT-309.
United Aircraft Research Laboratories SSS 10	1.86	0.31	1.03	Calculated values over simulated Federal Driving Cycle, based on emission data from GM engine T-56.
Chrysler sixth-generation engine	3.99	-0.26	2.21	1975 Federal Test Procedure emissions corrected (background level subtracted from measured values).
Williams Research/WR26/ AMC Hornet	7.43 6.92	0.62 0.72	2.8 2.5	Cold start Hot start } test data, 1975 Federal Test Procedure
Williams Research/131Q/ Volkswagen	4.5	0.34	1.81	Test data, 1972 Federal Test Procedure.
Advanced Combustors				
Solar	3.34	0.11	0.34	Calculated values for simulated Federal Driving Cycle, based on advanced combustor data.
General Motors 225 hp regenerative engine	2.4	0.015	0.315	Experimental test bed engine, chassis dynamometer test with 5000-lb car, 1975 Federal Test Procedure
AiResearch	1.6	0.67	0.44	Calculated values for simulated Federal Driving Cycle, for recuperated engine with 10% bypass.
United Aircraft of Canada	3.64	0.49	0.52	Calculated values over simulated Federal Driving Cycle based on advanced combustor data.
Clean Air Act Requirements	3.4	0.41	0.4	Future Federal standards

TABLE 4-5. SUMMARY OF RANKINE CYCLE ENGINE/VEHICLE DEVELOPMENT PROGRAMS (Ref. 4-1)

Developer/Contractor	Vehicle	Rated (or Estimated) Horsepower	Type of Expander and Rated Speed in rpm	Working Fluid	Status of Program
EPA AAPS Program					Preprototype engine development program completed in December 1973. One contractor (SES) selected to proceed with prototype engine development, followed by installation in vehicle.
Aerojet Liquid Rocket Co.	6-Passenger car[a]	150, Expander gross	Turbine; 32,000	AEF-78 (Proprietary)	
Lear Motors Corp.	5-Passenger car[a]	123, Expander gross	Turbine; 65,000	Water	
Scientific Energy Systems	6-Passenger car[a]	158, Expander gross 138, Transmission in	Piston; 2,500	Water	
Thermo Electron Corp.	6-Passenger car[a]	146, Expander gross	Piston; 1,800	Fluorinol - 85	
DOT/California Steam Bus Program					Program completed September 1972. Consisted of demonstration use in public service and vehicle emissions testing.
Brobeck & Associates	51-Passenger bus	240, Gross bhp 200, Net bhp	Piston; 2,100	Water	
Lear Motors Corp.	51-Passenger bus	240, Gross bhp 180, Net bhp	Turbine; 65,000	Water	
Steam Power Systems Co.	51-Passenger bus	275, Gross bhp 224, Net bhp	Piston; 1,850	Water	
DOT/Dallas Program					Program completed. Consisted of service in transit system and vehicle emissions testing.
Sundstrand Corp.	25-Passenger bus	120, Expander gross 90, Gear box out	Turbine; 35,000	CP-25; a single chemical compound comprised of methyl benzene, toluene, and toluall.	
California Clean Car Project					Testing and vehicle installation in progress. Target date for vehicle delivery is May 15, 1974.
Aerojet Liquid Rocket Co.	4-Passenger car[a]	60, Expander gross	Turbine; 60,000	Water	
Steam Power Systems Co.	4-Passenger car[a]	65, Expander gross	Piston; 2,400	Water	
Other Domestic Programs					
Carter Enterprises, Inc.	4-Passenger car (Volkswagen)	70	Piston; 5,000	Water	Vehicle first driven March 1972. Performance measurements being made.
Kinetics, Inc.	4-Passenger car	—	Gerotor type	Refrigerant 113	No published emissions or fuel economy data.
Williams Engine Co.	4-Passenger car	—	Piston	Water	
Foreign Programs					Component development. Recently initiated.
Pritchard Steam Power Proprietary, Ltd.	6-Passenger car	—	Piston	Water	
SAAB-SCANIA	Passenger car[a]	160	Piston	Water	
Politecnico Milano	Bus[a]	100	Turbine	Organic	

[a]Simulated or installation design target. bhp = brake horsepower DOT = U.S. Department of Transportation

TABLE 4-6. COMPARISON OF MOTOR CONTROLLERS (Ref. 4-2)

Item	dc Chopper	Variable Resistance	Step Voltage with Field Control
Types of Motor Controlled	All dc motors	All dc motors	Only separately excited, stabilized or compound wound
Velocity Range	Zero to maximum speed	Start only	Wide with three steps or more
Smoothness of Velocity Change	Very smooth	Jumpy	Initial jump 0.5 mph then smooth
Controller Protection	Solid state only-circuit breakers and fuses too slow	Circuit breakers and fuses sufficient	Circuit breakers and fuses
Controller Cost (1975)	High	Low	Medium
Controller Efficiency	Medium	Very Low	High with controller logic
Special Sensors and Control Logic	Complex	Simple	Complex
External Smoothing Filter	Heavy filter req'd	Not required	Not required
Starting Torque	High but inefficient	Medium and very inefficient	High with inefficient over-excitation
Velocity Stability	Stable with shunt motor, decreasing with load on series motor	Somewhat unstable varying with load	Stable up to torque limit
Torque at High Speed	High	Low	Medium-limited by field weakening ratio
Power Conditioning Characteristics	Modulation of full power used by motor	High switching currents with much dissipation	With small signal field control, high contactor currents at switch closing but zero contactor currents on switch opening.[a]

[a]Before a change of armature voltage takes place, the field is momentarily increased to the point where armature current reaches zero. The feedback from the current sensor then allows the armature relay to open. The usual problem of interrupting direct current is thus avoided.

TABLE 4-7. DESIGN AND PERFORMANCE CHARACTERISTICS EXPECTED FROM DEVELOPMENT OF CANDIDATE BATTERIES FOR HYBRID VEHICLES (Refs. 4-30 to 4-34)

Type	Temp. (°F)	W-hr/lb	W/lb	W-hr/in.^{3}	Cycle Life[a]	$/lb	Availability	Cell Reactions Charge $\longleftarrow$ Discharge $\longrightarrow$
Lead-Acid	-40 to 115	8-16	75-300	1.1-1.6	300-400	0.3-1.0	Now	$Pb + PbO_2 + 2H_2O + 2H_2SO_4 \rightleftarrows 2PbSO_4 + 2H_2O$
Nickel-Cadmium	30 to 95	18-25	100-150	0.7-1.0	< 3000	3-12[b]	Now	$Cd + 2N_iO(OH) + 2H_2O \rightleftarrows Cd(OH)_2 + 2N_i(OH)_2$
Nickel-Zinc		25-35	100-300	2-3	150-300	2-3	1976	$2N_iOOH + Zn + H_2O \rightleftarrows 2N_i(OH)_2 + ZnO$
Lithium-Sulfur	700	100-150	100-150	ID[c]	300	ID	1980	$2Li + S \rightleftarrows Li_2S$
Sodium-Sulfur	660	90-100	90-100	ID	250-500	ID	1980	$2Na + 3S \rightleftarrows Na_2S_3$
Lithium-Chlorine	1200	100	100	ID	ID	ID	1980	$2Li + Cl_2 \rightleftarrows 2LiCl$

[a] High discharge rate/high depth-of-discharge service
[b] Commercial batteries
[c] ID Insufficient data

TABLE 4-8. HYBRID VEHICLE SPECIFICATIONS (Ref. 4-2)

Vehicle Type	Weight (lb)	Acceleration Capability (sec)	
		Conventional	Reduced
Family Car	4000	0-60 mph, 13 sec	0-60 mph, 20 sec
Commuter Car	1700	0-60 mph, 15 sec	0-60 mph, 20 sec
Delivery Van	7000	0-60 mph, 25 sec	0-60 mph, 30 sec
Intercity Bus	30,000	0-60 mph, 80 sec	No Change
Intracity Bus	30,000	0-30 mph, 10 sec	0-30 mph, 22 sec

TABLE 4-9. FLYWHEEL MATERIALS STUDIED BY LOCKHEED (Ref. 4-3)

Material	Density (ρ) (lb/in.^{3})	Poisson's Ratio (ν)	Ultimate Tensile (F_{tu}) ksi	Yield Tensile (F_{ty}) ksi	Working Stress (σ) ksi	$\frac{\sigma}{\rho}$ ($\times 10^6$)	Material Cost ($/lb)	Normalized Cost ($/lb)
18NI-400 (Maraging Steel)	0.289	0.26	409	400	260	0.900	2.25	5.30
18NI-300 (Maraging Steel)	0.289	0.30	307	300	200	0.692	2.25	6.89
4340 Steel	0.283	0.32	260	217	130	0.459	0.60	2.76
1040 Steel	0.283	0.30	87	58	36	0.127	0.30	5.00
1020 Steel	0.283	0.30	68	43	25	0.088	0.30	7.23
Cast Iron	0.280	0.30	55	37	20	0.071	0.30	8.94
2021-T81 (Aluminum)	0.103	0.33	62	52	26	0.252	0.53	4.45
2024-T851 (Aluminum)	0.100	0.33	66	58	35	0.350	0.50	3.03
6Al-4V (Titanium)	0.160	0.32	150	140	82	0.512	4.00	16.55
E-Glass	0.075	0.29	200	–	67	0.890	0.42	1.00
S-Glass	0.072	0.293	260	–	87	1.210	0.75	1.31

TABLE 4-10. FLYWHEEL CONFIGURATIONS STUDIED BY LOCKHEED (Ref. 4-3)

Capacity (kw-hr)	Material	Geometry	Flywheel Speed (rpm)	Flywheel Weight (lb)	Assembly Weight (lb)	Assembly Volume	Assembly Cost ($)
0.5	4340	Pierced Disc	21,749	111.7	124.8	0.75	240.44
0.5	4340	Solid Disc	22,821	57.6	74.9	0.39	155.65
0.5	1040	Solid Disc	13,453	206.6	222.0	1.39	232.40
0.5	2021-T81	Solid Disc	13,394	105.0	127.9	1.95	235.65
0.5	2024-T851	Solid Disc	21.940	75.8	91.5	1.45	160.70
0.5	4340	Conical	23,410	52.0	69.1	0.20	145.18
0.5	2021-T81	Conical	18,698	92.1	112.3	1.0	201.08
0.5	2024-T851	Conical	22,970	66.2	82.5	0.75	148.17
0.5	4340	Constant-Stress	24,000	42.4	59.5	0.12	127.51
0.5	2021-T81	Constant-Stress	24,000	75.2	90.9	0.56	166.50
0.5	2024-T851	Constant-Stress	24,000	56.62	73.52	0.43	135.63
1.0	4340	Pierced Disc	17,308	225.9	242.0	1.52	454.85
1.0	4340	Solid Disc	21,746	115.1	133.7	0.78	262.86
1.0	1040	Solid Disc	10,678	413.1	433.1	2.78	431.52
1.0	2021-T81	Solid Disc	13,394	210.1	233.0	3.89	402.60
1.0	2024-T851	Solid Disc	17,714	151.5	171.7	2.89	287.88
1.0	4340	Conical	22,873	99.5	119.3	0.4	238.26
1.0	2021-T81	Conical	18,097	180.2	199.5	2.0	344.30
1.0	2024-T851	Conical	20,857	130.8	149.8	1.49	293.22
1.0	4340	Constant-Stress	24,000	83.8	104.8	0.22	213.66
1.0	2021-T81	Constant-Stress	24,000	150.3	166.0	1.10	286.00
1.0	2024-T851	Constant-Stress	24,000	110.0	127.9	0.83	218.56
1.0	E-Glass	Bar	15,132	87.6	136.5	9.90	260.59
1.0	S-Glass	Bar	19,199	63.3	108.9	7.63	274.54

TABLE 4-11. SUMMARY OF COMPOSITE MATERIALS, ROD TESTS (JOHNS HOPKINS) (Ref. 4-38)

Test	Drive System[1]	Mounting System[2]	Max. Speed (rpm)	Max. Stress[3] (ksi)	E/W[3] (W-h/lb)	Remarks
AVCO Boron Filaments			W_r = 0.00035 and 0.00014 pound			
4-mil	Diehl	Tube	38 000	424	48	Rod did not fail; speed limited by motor
8-mil	Diehl	Tube	34 000	340	38	
General Technology Corp. Boron/Magnesium 20-Mil Preform			W_r = 0.00075 pound			
1	Diehl	Epoxy	31 500	254	33	No failure; speed limited
Hercules Graphite/Epoxy 1/8-Inch Square			W_r = 0.027 pound			
1	S/I	RTV	24 500	105	20	
2	G/S	RTV	19 000	66	12	
3	G/S	Epoxy	31 300	180	33[4]	
Fothergill & Harvey Graphite/Epoxy 1/16 Inch Square			W_r = 0.0066 pound			
1	S/I	RTV	22 400	91	17	
2	S/I	RTV	22 800	94	18	
3	S/I	RTV	22 400	91	17	
4	S/I	RTV	28 900	151	28	
5	S/I	RTV	26 200	124	23	
6	S/I	RTV	22 100	88	16	
7	G/S	Epoxy	33 000	195	36[4]	
PPG/BBI Type 525 E-Glass/Polyester (48% Fiber Volume) 0.098 Inch in Diameter			W_r = 0.015 pound			
1	G/S	Epoxy	23 600	116	19	
2	G/S	Epoxy	23 400	118	18	
3	G/S	Epoxy	24 000	123	19	
4	G/S	RTV	~19 800	~84	~13	Counter stopped just before rod failure
5	G/S	RTV	24 000	123	19	
6	G/S	RTV	23 200	115	18	
PPG/BBI Type 1055 E-Glass/Polyester (58% Fiber Volume) 0.098 Inch in Diameter			W_r = 0.016 pound			
1	G/S	Epoxy	27 000	165	24	
2	G/S	RTV	26 700	160	24	
PPG/BBI R-Composition Glass/Polyester (55% Fiber Volume) 0.098 Inch in Diameter			W_r = 0.015 pound			
1	G/S	Epoxy	28 900	177	28[4]	
2	G/S	Epoxy	30 700	194	31[4]	
3	G/S	RTV	<18 000	<65	<11	Failure before strobe synchronization
4	G/S	RTV	26 400	149	23	
Columbia Products (Shakespeare) E-Glass/Epoxy, 0.250 Inch in Diameter			W_r = 0.091 pound			
1	S/I	RTV	14 400	41	7	Some indications rod pulled out of holder
2	S/I	RTV	14 600	42	7	

TABLE 4-11. CONCLUDED (Ref. 4-38)

Test	Drive System[1]	Mounting System[2]	Max. Speed (rpm)	Max. Stress[3] (ksi)	E/W[3] (W-h/lb)	Remarks
Corning Glass Rods, 0.106 Inch in Diameter, 30.9 Inches Long			W_r = 0.026 pound			
1	G/S	Acrylic	9 240	29	3	
2	G/S	Acrylic	9 480	28	3	
3	G/S	Acrylic	9 780	26	3	Failed away from point of max. stress
4	G/S	Acrylic	9 900	32	4	d. o.
5	G/S	Acrylic	9 540	30	3	
6	G/S	Acrylic	10 260	36	4	
PPG Glass Rods in Steel Holder			W_{glass} = 0.0015 pound; W_{steel} = 0.32 pound			
1	G/S	Acrylic	25 100	165	14	E/W based on equivalent full circular brush (see text)

1 Drive System:

- S/I: Speed increaser
- G/S: Globe motor with spindle
- Diehl: Diehl motor

2 Mounting System:

- RTV: Dow Corning Silastic 734, room temperature vulcanizing
- Epoxy: Armstrong epoxy
- Acrylic: Acrylic cement
- Tube: Stainless steel tube support with epoxy cement

3 Stress and specific energy calculations assume constant rod cross-section, uniform mass distribution, and 30-inch length (spin diameter) except where noted; 1 ksi = 1000 lb/in^2.

4 For the graphite/epoxy and R-glass/polyester materials, the better results were obtained with the Globe motor and epoxy mounting system.

TABLE 4-12. TEST RESULTS FOR 1-LB BAR: SPEED, STRESS, AND SPECIFIC ENERGY AT FAILURE (JOHNS HOPKINS) Ref. 4-38)

Parameters	Test Number					
	JH-1[a]	JH-2	--	JH-3	JH-4	JH-5
Material	E-Glass/ Polyester	S-Glass/ Epoxy	S-Glass/ Epoxy	Graphite/ Epoxy	Graphite/ Epoxy	S-Glass/ Epoxy
Cross Section	13/16-in.-dia	0.57-in.-square	0.56-in.-square	0.79-in.-square	0.78-in.-square	0.57-in.-square
Weight, lb	1.10	0.73	0.72	1.00	0.99	0.72
Speed, rpm	19,900	29,100	b	28,000	28,200	27,200[c]
Stress						
ksi	89	204	b	136	137	178
percent[d]	--	79	b	68	69	69
E/W, W-h/lb	13.2	28.2	b	26.1	26.5	24.7
Vacuum, Torr	1.8×10^{-2}	6.3×10^{-2}	b	17×10^{-2}	25×10^{-2}	19×10^{-2}

[a] Facility and instrumentation checkout

[b] No test--facility failure

[c] Test aborted at this speed. Rod subsequently failed at 24,700 rpm

[d] Percent of expected value quoted by Hercules

TABLE 4-13. SUNDSTRAND TRANSMISSION EVALUATION -- FEDERAL EMISSIONS TEST DRIVING CYCLE (Ref. 4-40)

Transmission	Results (mpg)		
	Flywheel Energy Storing Transmission		Nonenergy Storing Three-Speed Automatic Transmission
	Baseline	Alternate	
"Real" (with estimated transmission losses)	7.96	9.26	11.14
"Ideal" (zero transmission losses, fly-wheel losses are included)	9.78	12.66	11.99
Note: Vehicle weight 4,300 lb			

TABLE 4-14. SUNDSTRAND ESTIMATE OF CONSTANT SPEED FUEL CONSUMPTION (Ref. 4-40)

Constant Speed, (mph)	Results (mpg)		
	Baseline 8A	Alternate 8C	Three-Speed Automatic
20	9.82	12.62	15.58
30	11.41	12.47	17.86
40	14.42	15.59	17.92
50	16.04	16.80	16.92
60	16.59	16.59	14.30
70	16.25	16.25	11.91
80	13.32	13.32	10.34
Note: Vehicle weight 4,300 lb			

TABLE 4-15. HYDRAULIC HYBRID-COMPUTER SIMULATION RESULTS (Ref. 4-45)

VEHICLE TYPE / DRIVING CYCLE	LOW SPEED DELIVERY				URBAN "MINI-CAR"		
	LA-4	TR	SR	RR	LA-4	LA-4	LA-4
LOADED MASS (lb.)	7000.0	7000.0	7000.0	7000.0	1700.0	1700.0	1700.0
LOADED INTERTIAL MASS (lb)	7200.0	7200.0	7200.0	7200.0	1750.0	1750.0	1750.0
AERODYNAMIC DRAG COEF.	0.85	0.85	0.85	0.85	0.35	0.35	0.35
FRONTAL AREA (ft²)	42.0	42.0	42.0	42.0	18.0	18.0	18.0
REAR AXLE RATIO	4:1	4:1	4:1	4:1	3 08 1	3 08 1	3 08 1
ROLLING FRICTION COEF.	0.012	0.012	0.012	0.012	0.011	0.011	0.011
TIRE ROLLING RADIUS (in.)	15.0	15.0	15.0	15.0	12.5	12.5	12.5
MAXIMUM VELOCITY (mph)	40.0	40.0	40.0	40.0	60.0	50.0	40.0
ENGINE HORSEPOWER	43.2	43.2	43.2	43.2	21.6	14.0	8.5
ACCUMULATOR SIZES (gal.)	6.6/0.6	6.6/0.6	6.6/0.6	6.6/0.6	3.6/0.4	2.5/0.2	1.6/0.2
FIXED PUMP DISP (in.³/rev)	0.21	0.21	0.21	0.21	0.11	0.07	0.04
VARIABLE PUMP MAX. DISP (in.³/rev)	1.27	1.27	1.27	1.27	0.65	0.42	0.25
VARIABLE MOTORS DISP (in. 3/rev)	3.85/5.78	3.85/5.78	3.85/5.78	3.85/5.78	1.42/2.20	1.09/2.41	0.82/2.61
SPEED ON GRADE mph 1%	15.6/8.0	15.6/8.0	15.6/8.0	15.6/8.0	22.5/12.0	15.0/12.0	9.3/12.0
MILES PER GALLON (SI ENGINE)	13.3	13.6	12.3	11.8	51.1	56.7	64.2
ADD'L. TRIP TIME (sec)	41.0[a]				0.0	21.0[a]	60.9[a]

[a]THESE VEHICLES DO NOT HAVE ENOUGH TOP SPEED TO FOLLOW THE LA-4 DRIVING CYCLE, AND ADDITIONAL TIME IS REQUIRED TO COMPLETE THE 7.5 MILE TRIP.

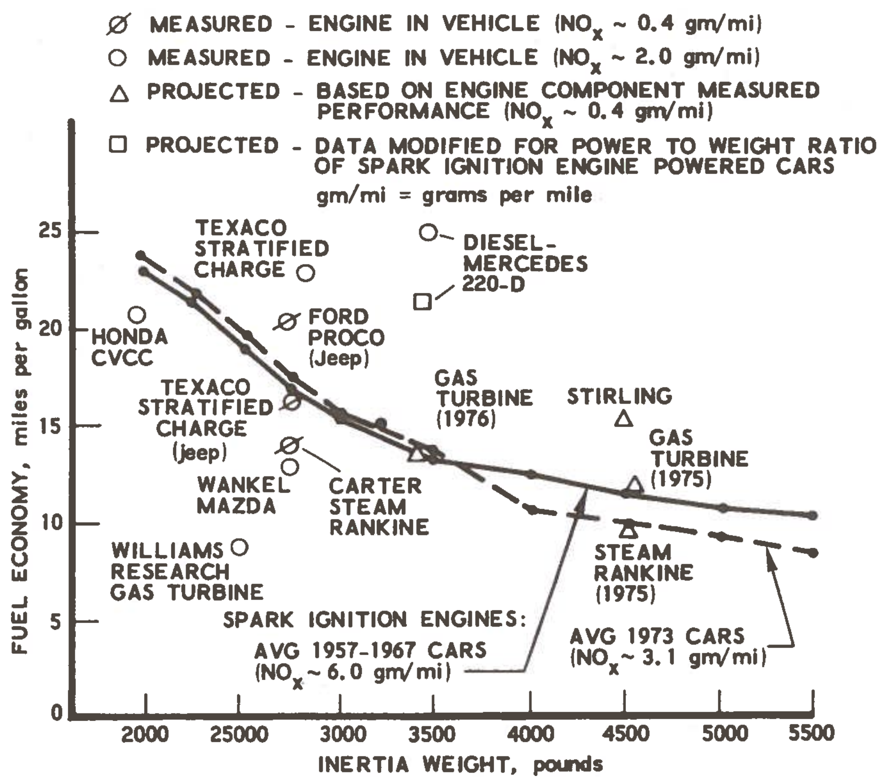


FIGURE 4-1. FUEL ECONOMY OVER THE FEDERAL EMISSIONS TEST DRIVING CYCLE (Ref. 4-1)

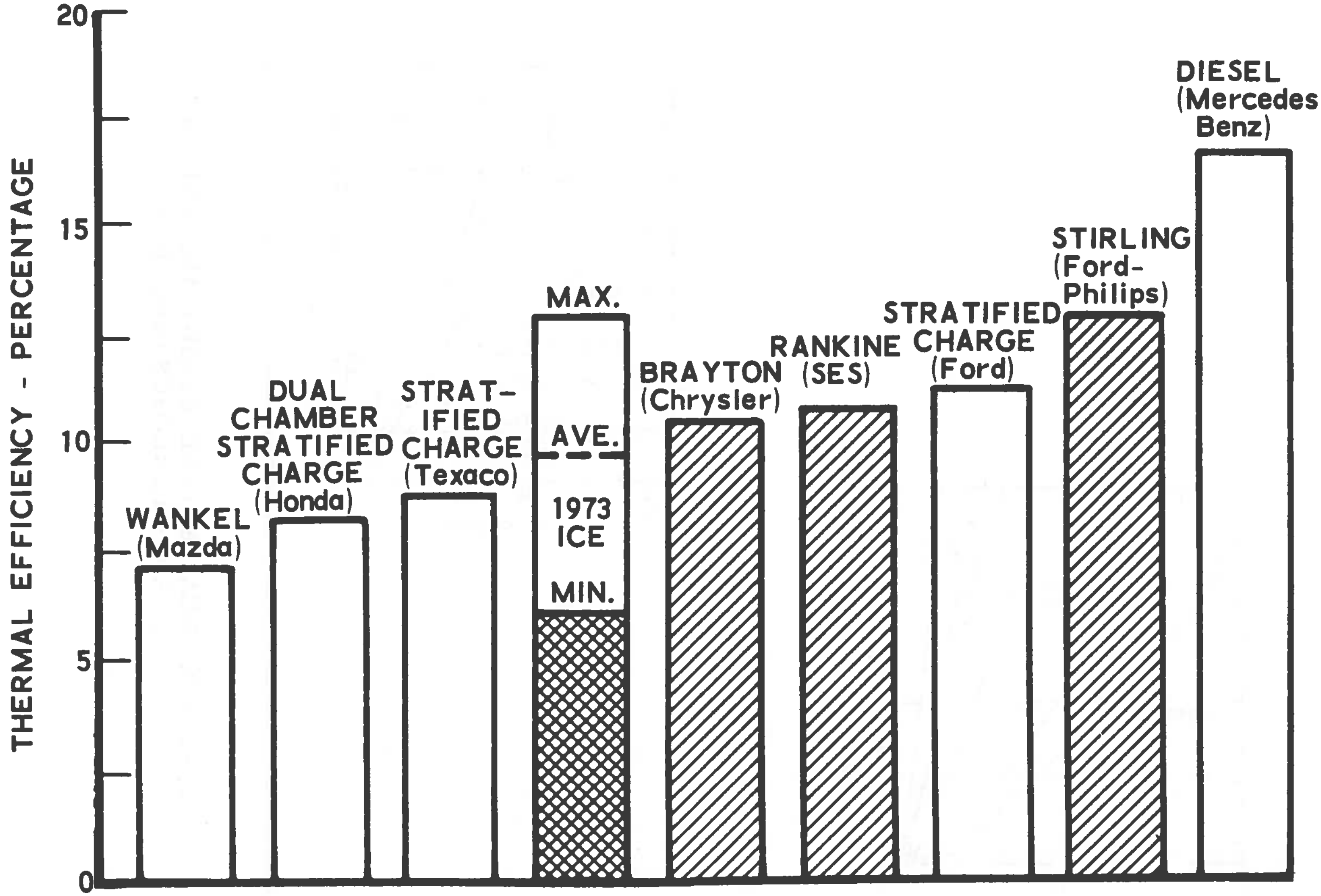
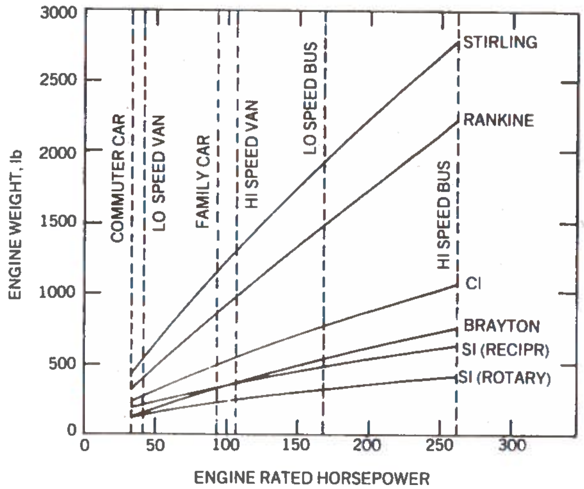


FIGURE 4-2. THERMAL EFFICIENCY OVER THE FEDERAL DRIVING CYCLE (Ref. 4-1)



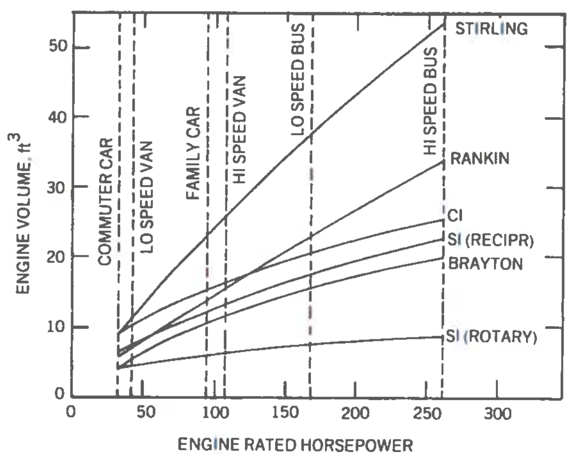


FIGURE 4-3. HEAT ENGINE WEIGHT AND VOLUME CHARACTERISTICS (Ref. 4-2)

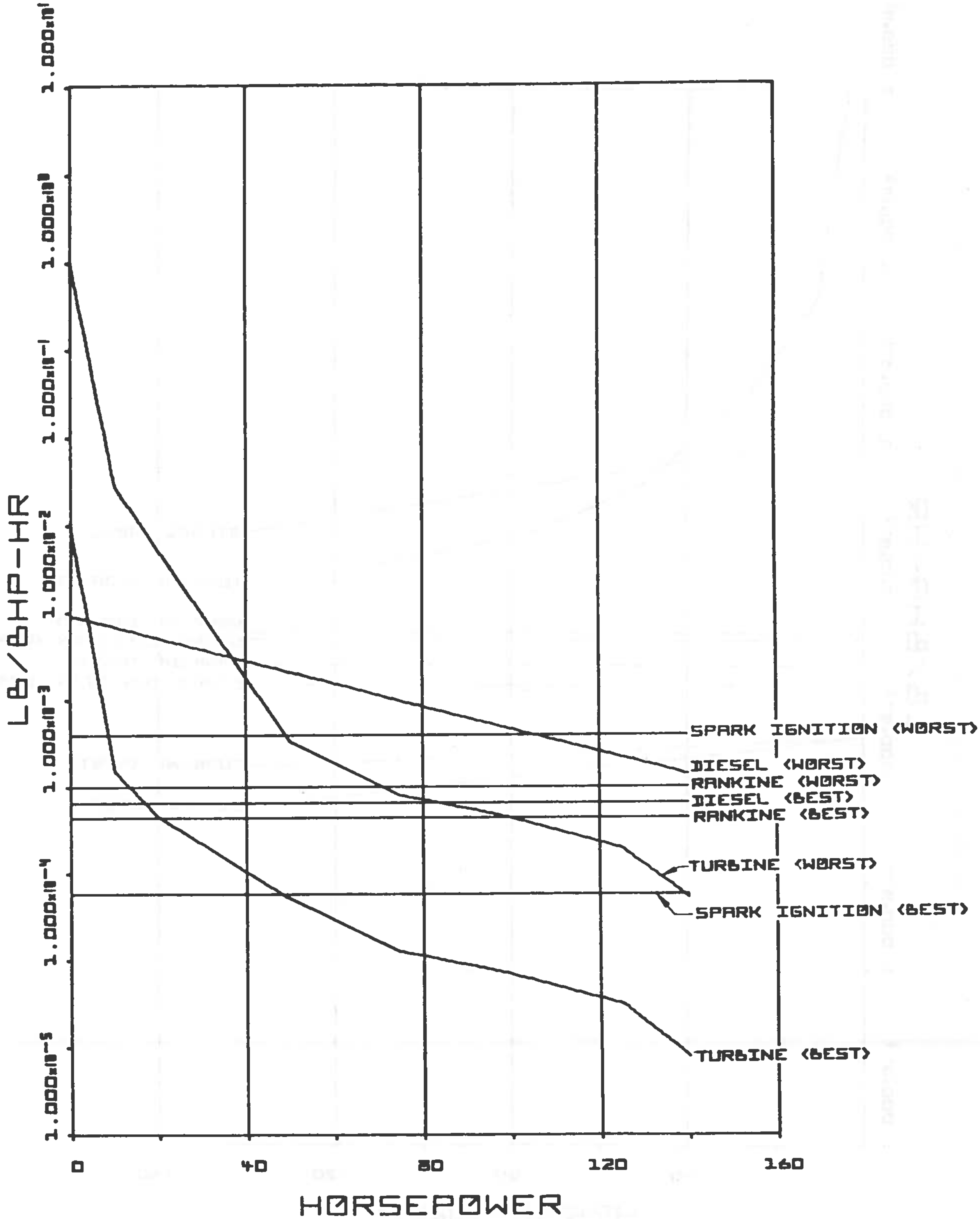


FIGURE 4-4. HC EMISSIONS VERSUS CONTINUOUS HORSEPOWER (Ref. 4-3)

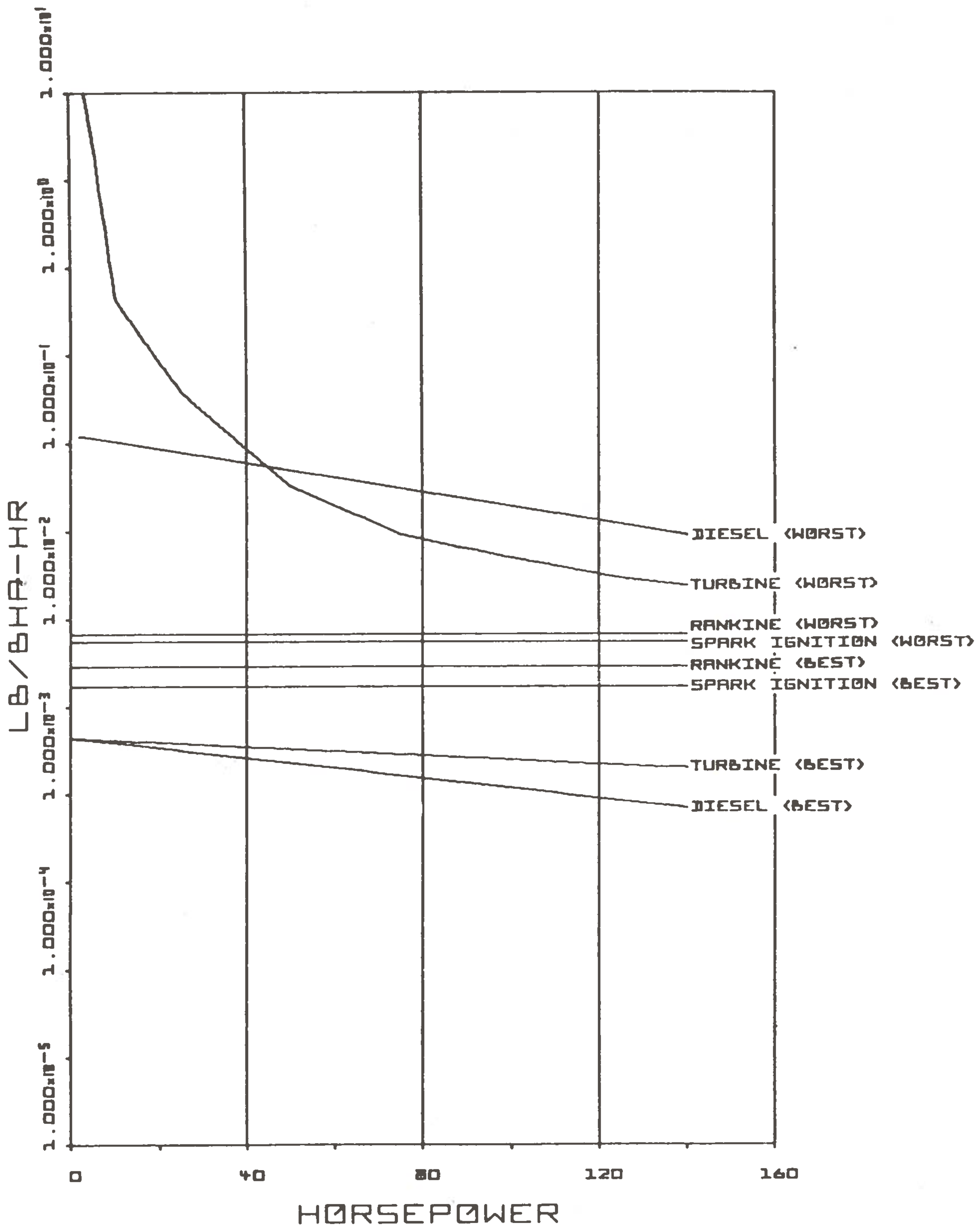


FIGURE 4-5. CO EMISSIONS VERSUS CONTINUOUS HORSEPOWER (Ref. 4-3)

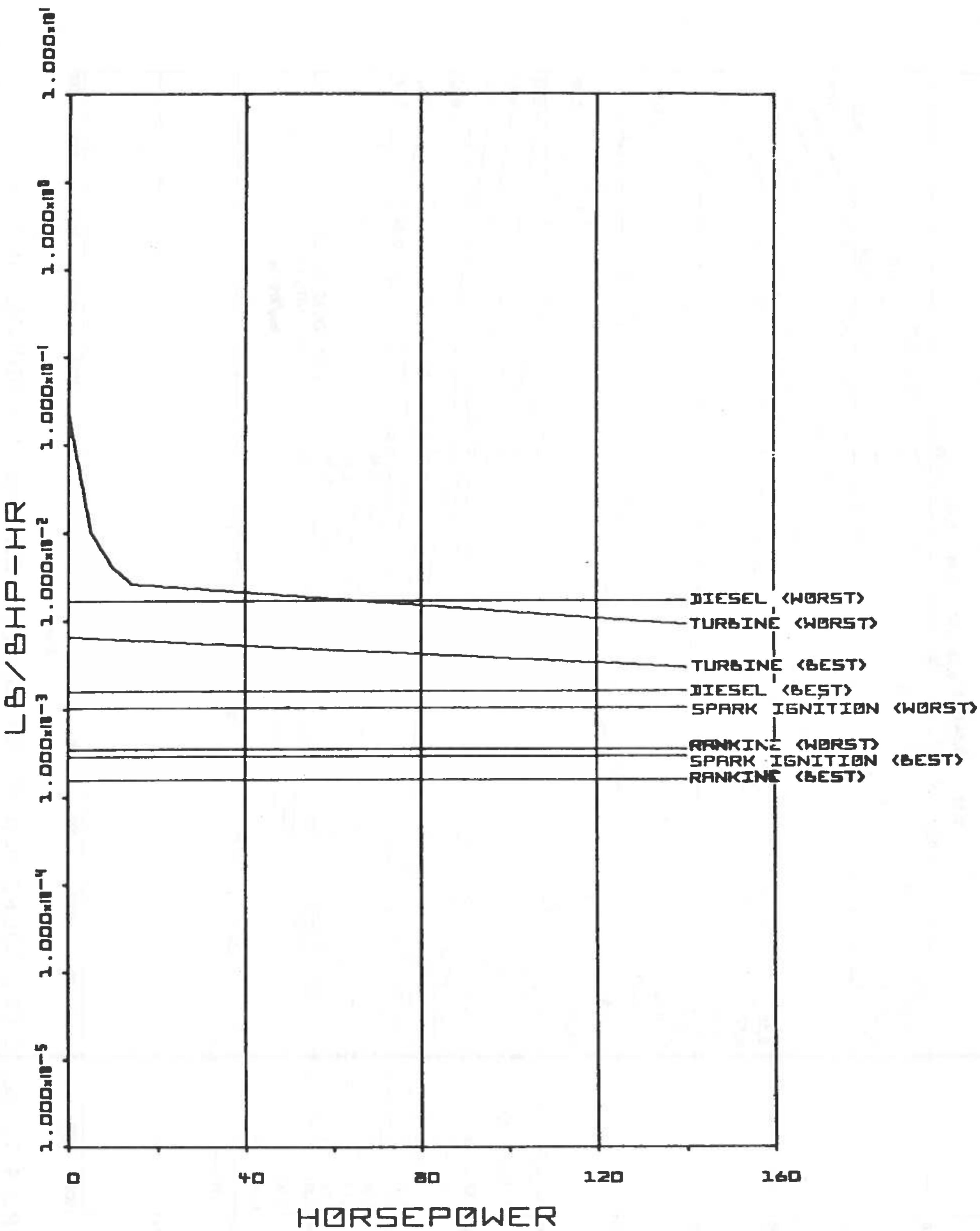


FIGURE 4-6. NO_x EMISSIONS VERSUS CONTINUOUS HORSEPOWER (Ref. 4-3)

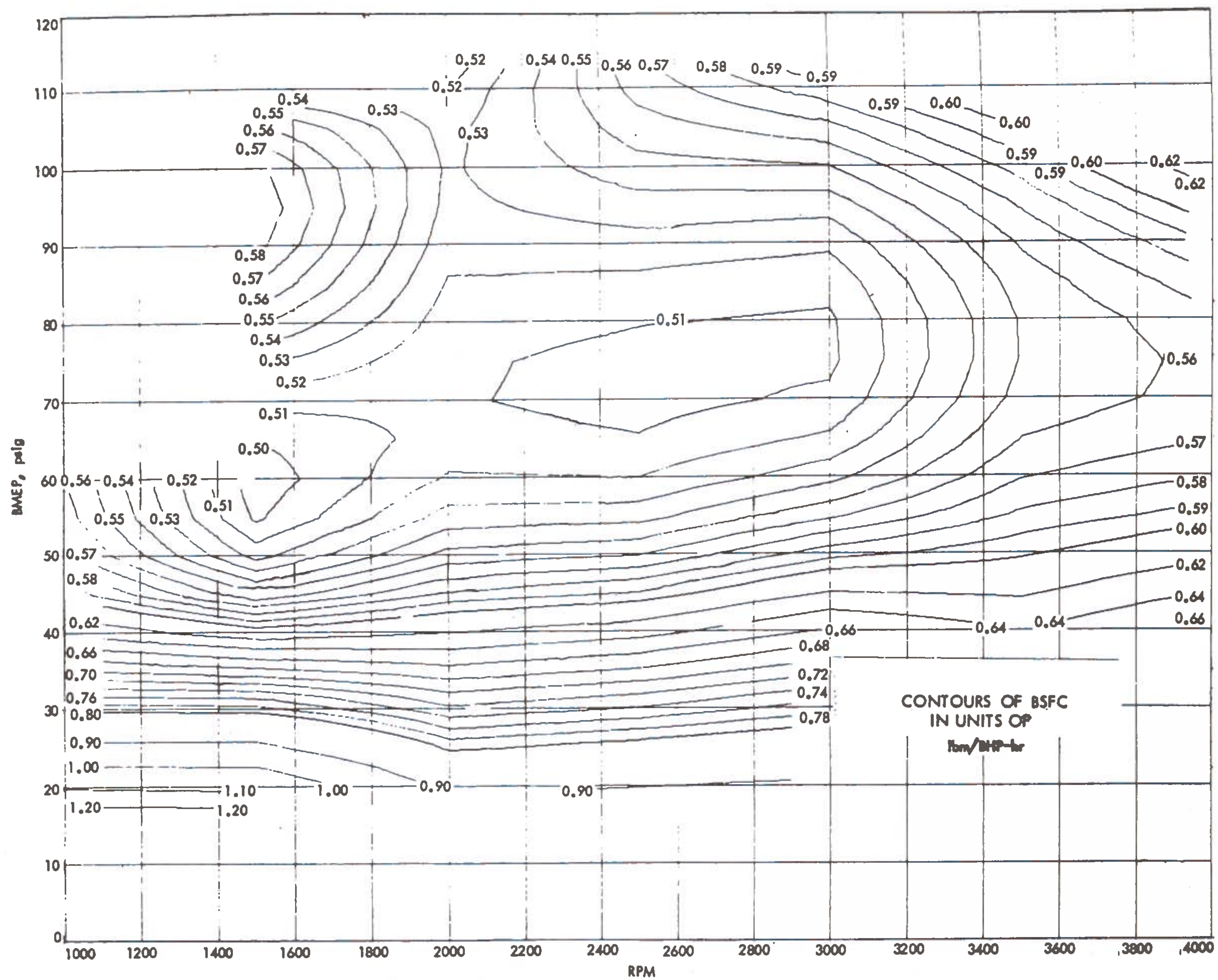


FIGURE 4-7. BSFC CONTOURS FOR SPARK IGNITION ENGINE, LB/BHP-HR (Ref. 4-4)

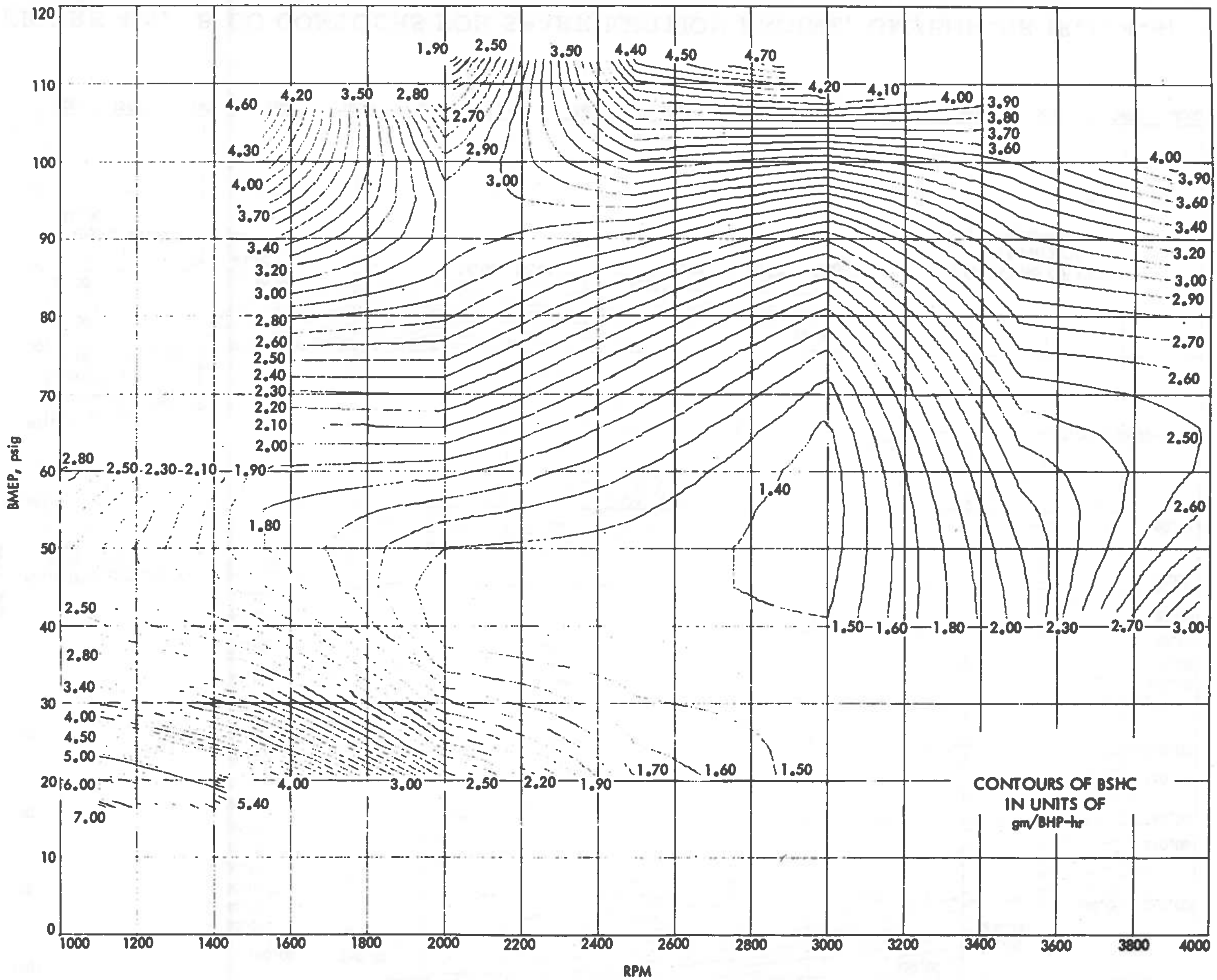


FIGURE 4-8. BSHC CONTOURS FOR SPARK IGNITION ENGINE, GM/BHP-HR (Ref. 4-4)

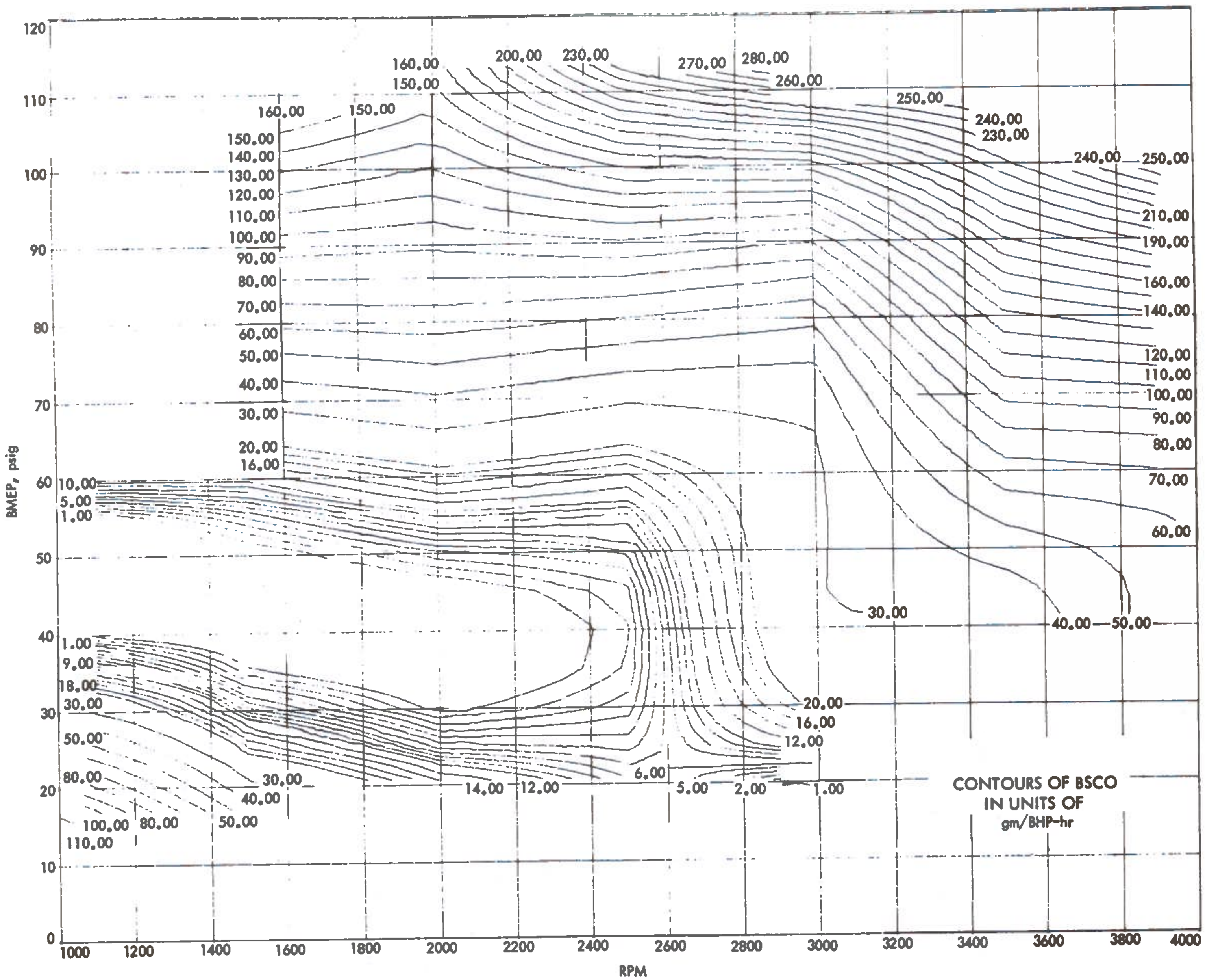


FIGURE 4-9. BSCO CONTOURS FOR SPARK IGNITION ENGINE, GM/BHP-HR (Ref. 4-4)

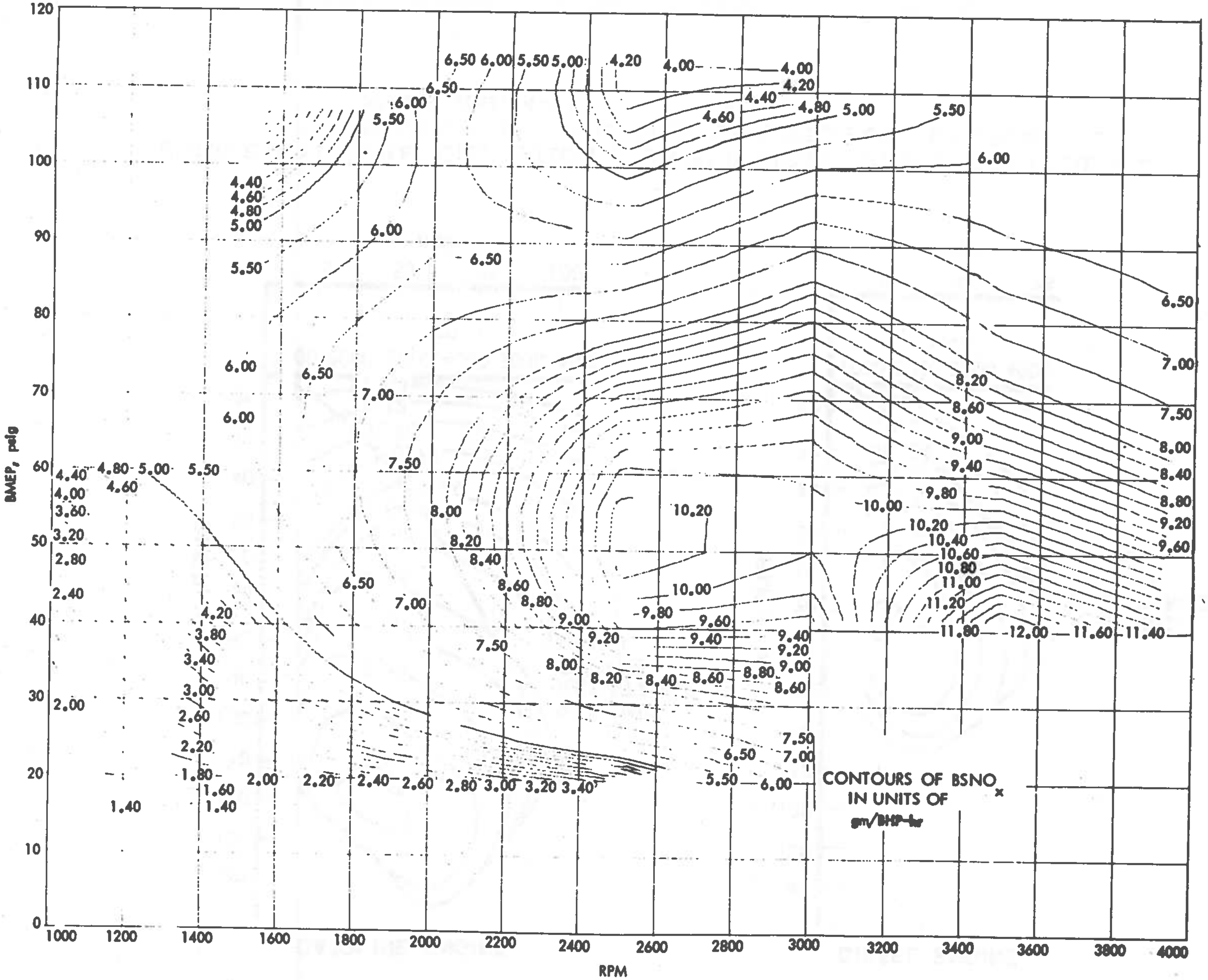


FIGURE 4-10. BSNO$_x$ CONTOURS FOR SPARK IGNITION ENGINE, GM/BHP-HR (Ref. 4-4)

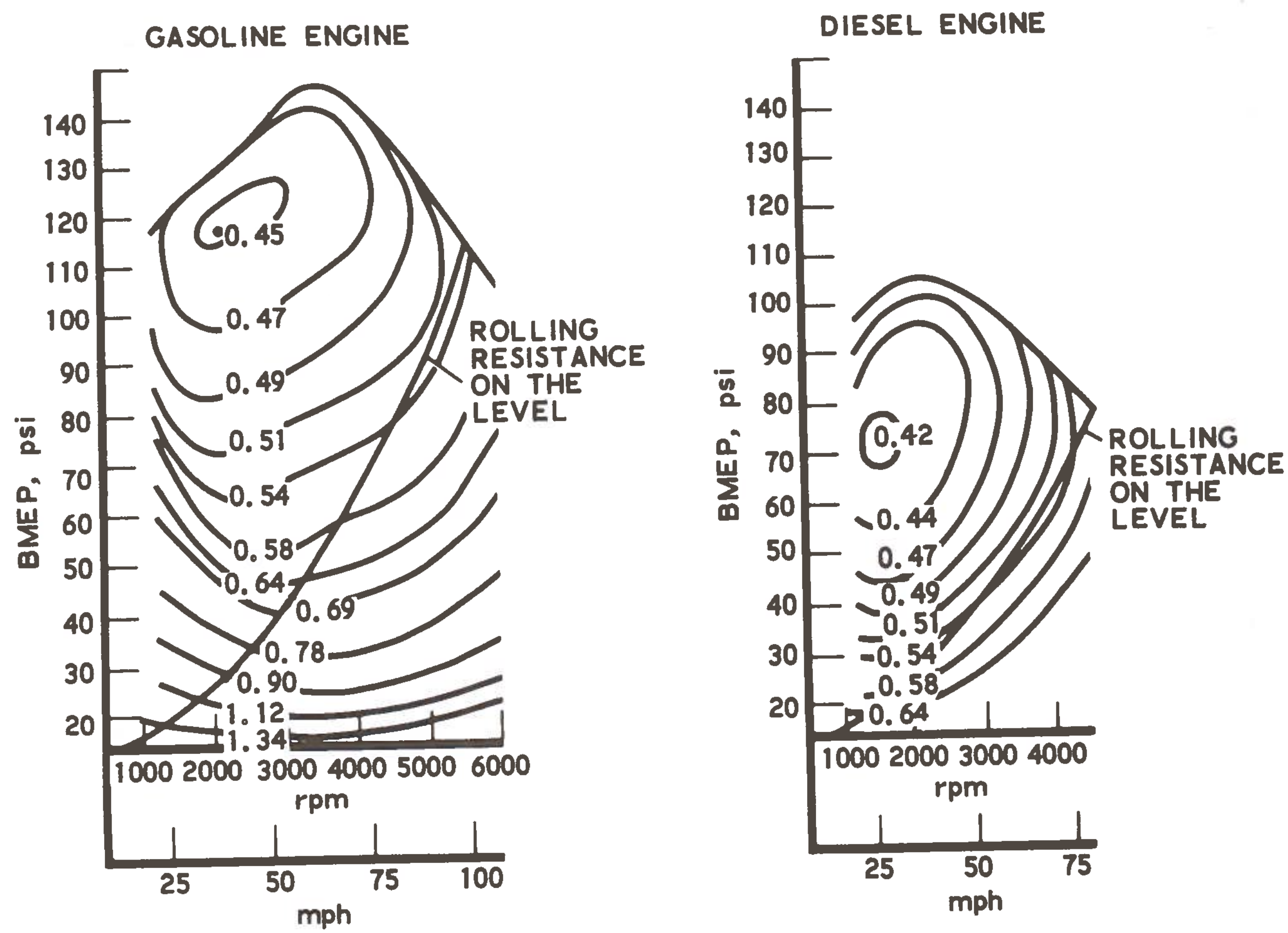


FIGURE 4-11. PRECOMBUSTION CHAMBER DIESEL AND UNCONTROLLED SPARK IGNITION ENGINE SPECIFIC FUEL CONSUMPTION MAPS (Ref. 4-5)

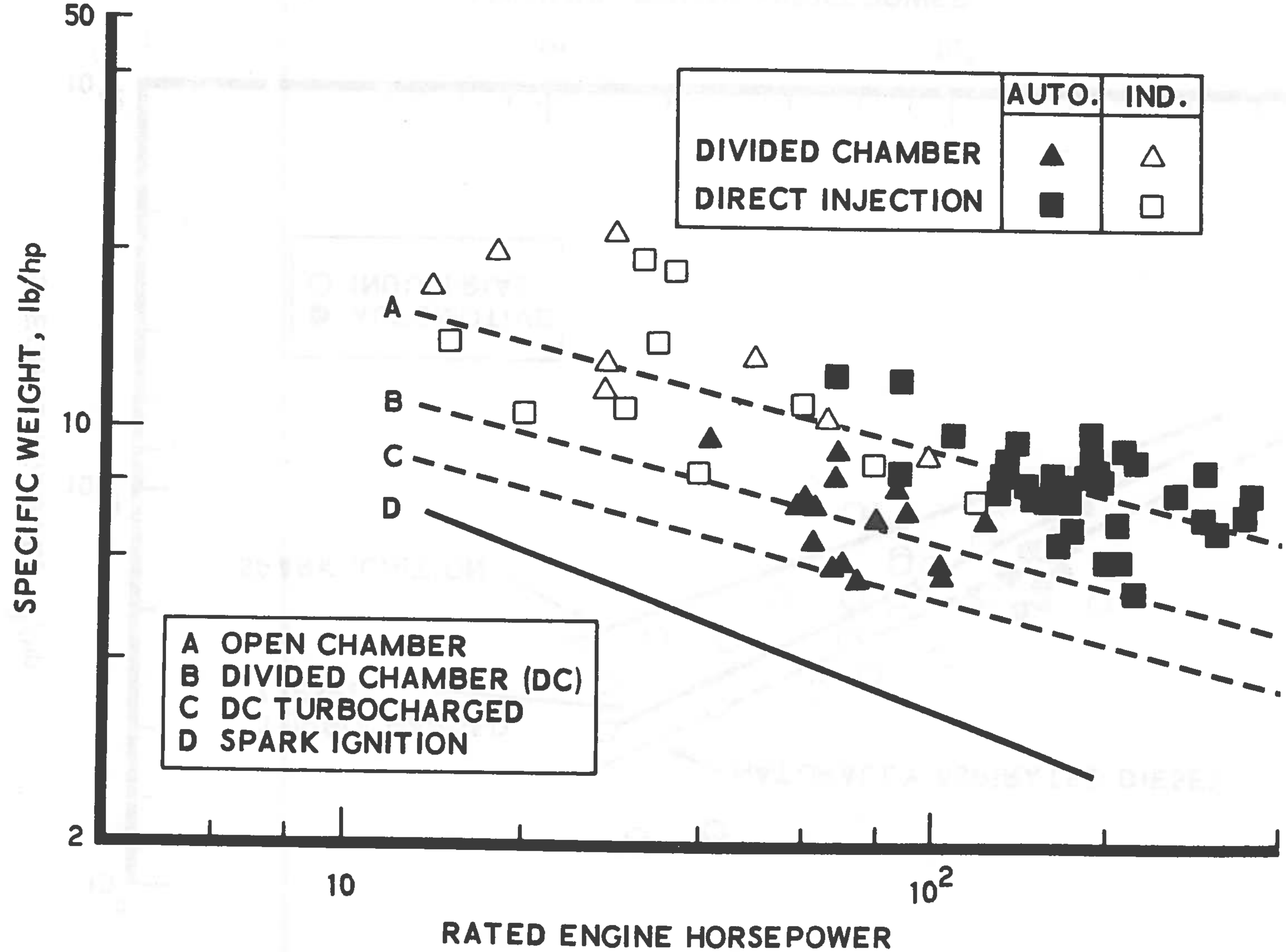


FIGURE 4-12. SPECIFIC WEIGHT OF DIESEL AND SPARK IGNITION ENGINES (Ref. 4-1)

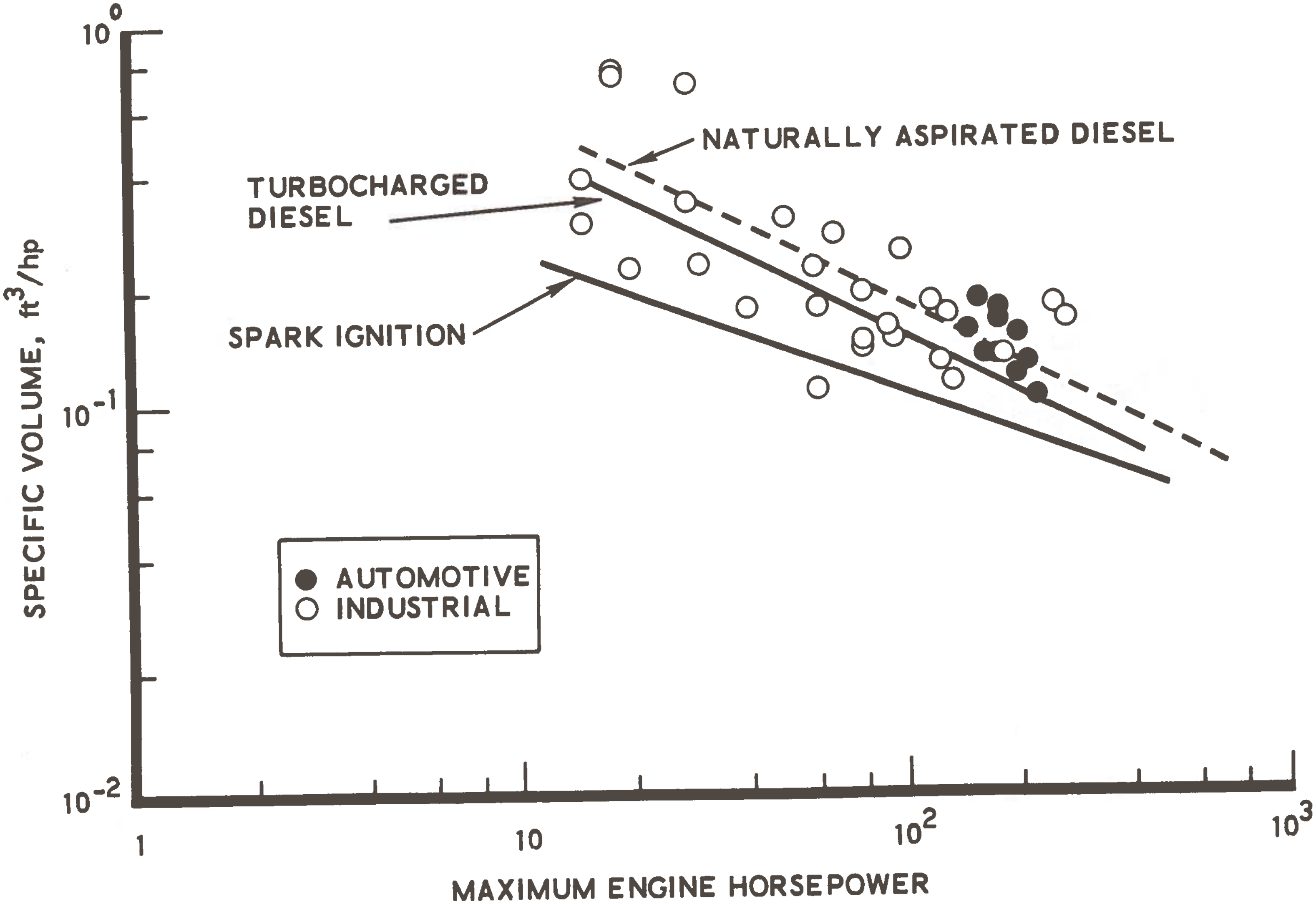


FIGURE 4-13. SPECIFIC VOLUME OF DIESEL AND SPARK IGNITION ENGINES (Ref. 4-1)

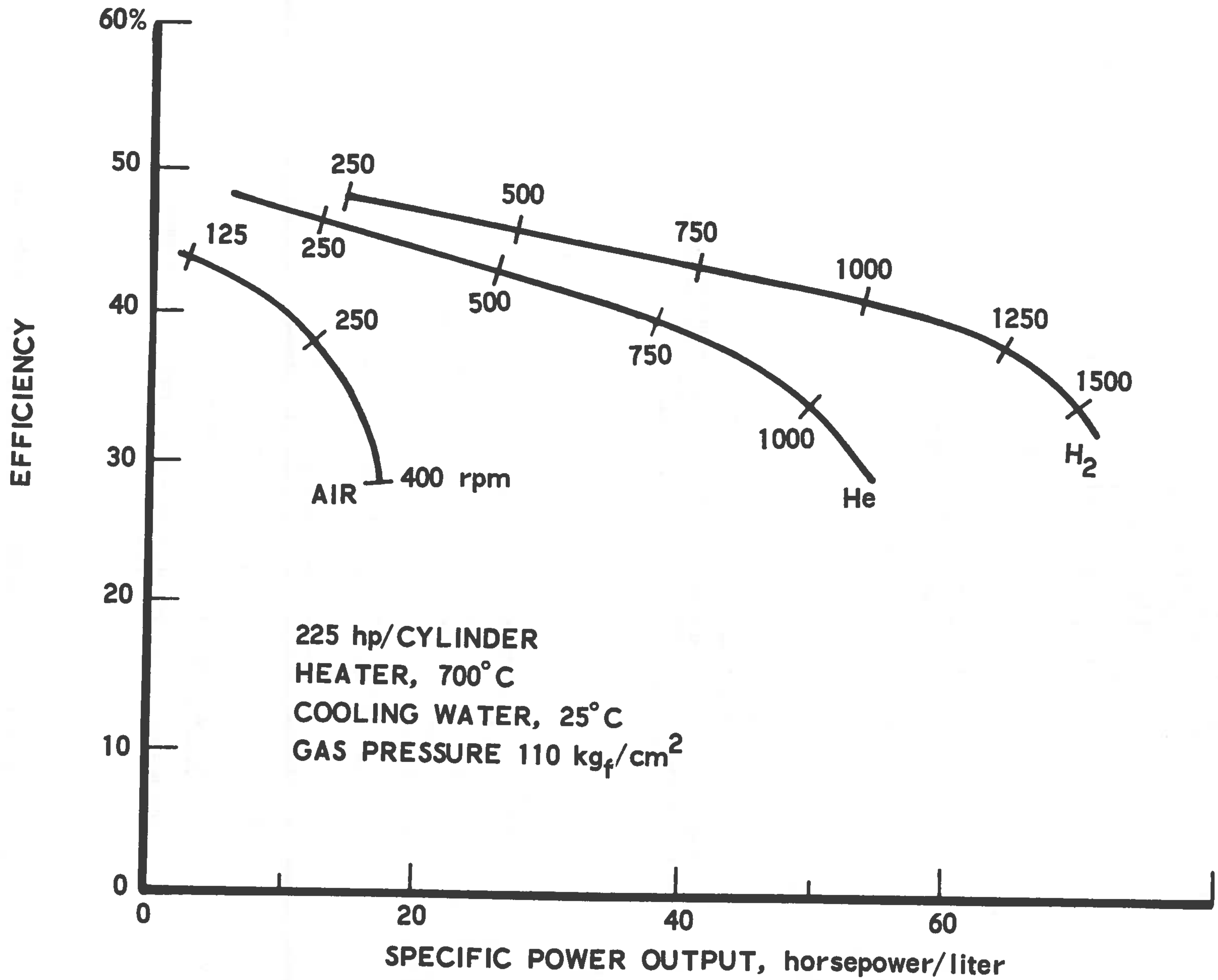
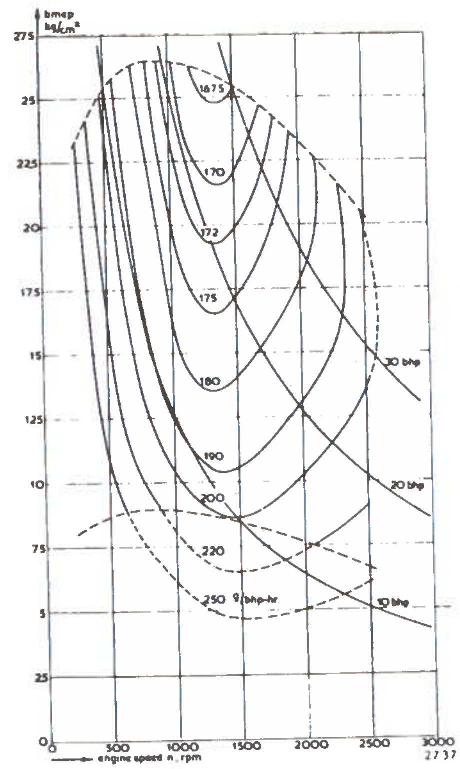
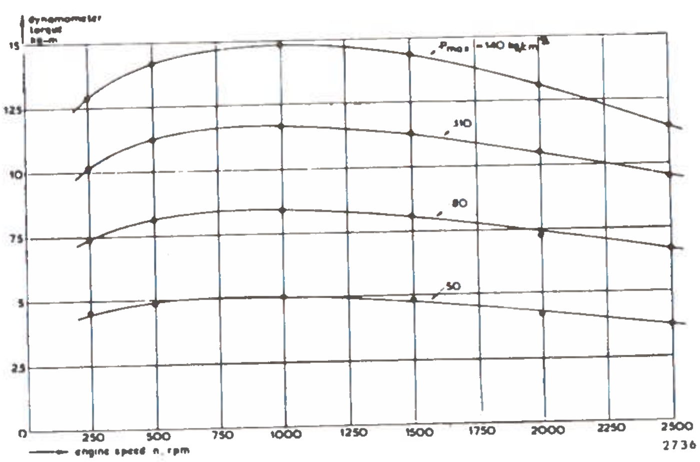


FIGURE 4-14. STIRLING ENGINE EFFICIENCY VERSUS SPECIFIC POWER OUTPUT (Ref. 4-1)



Lines of constant specific fuel consumption (g/b.h.p.-hr) and constant brake horse-power as a function of the mean effective pressure and the engine speed n (fuel consumption based on a lower heat value of 10,000 kcal/kg).



Dynamometer torque in kg-m plotted as a function of the speed n for various values of p_{max}.

FIGURE 4-15. RHOMBIC DRIVE STIRLING ENGINE PERFORMANCE MAP (Ref. 4-6)

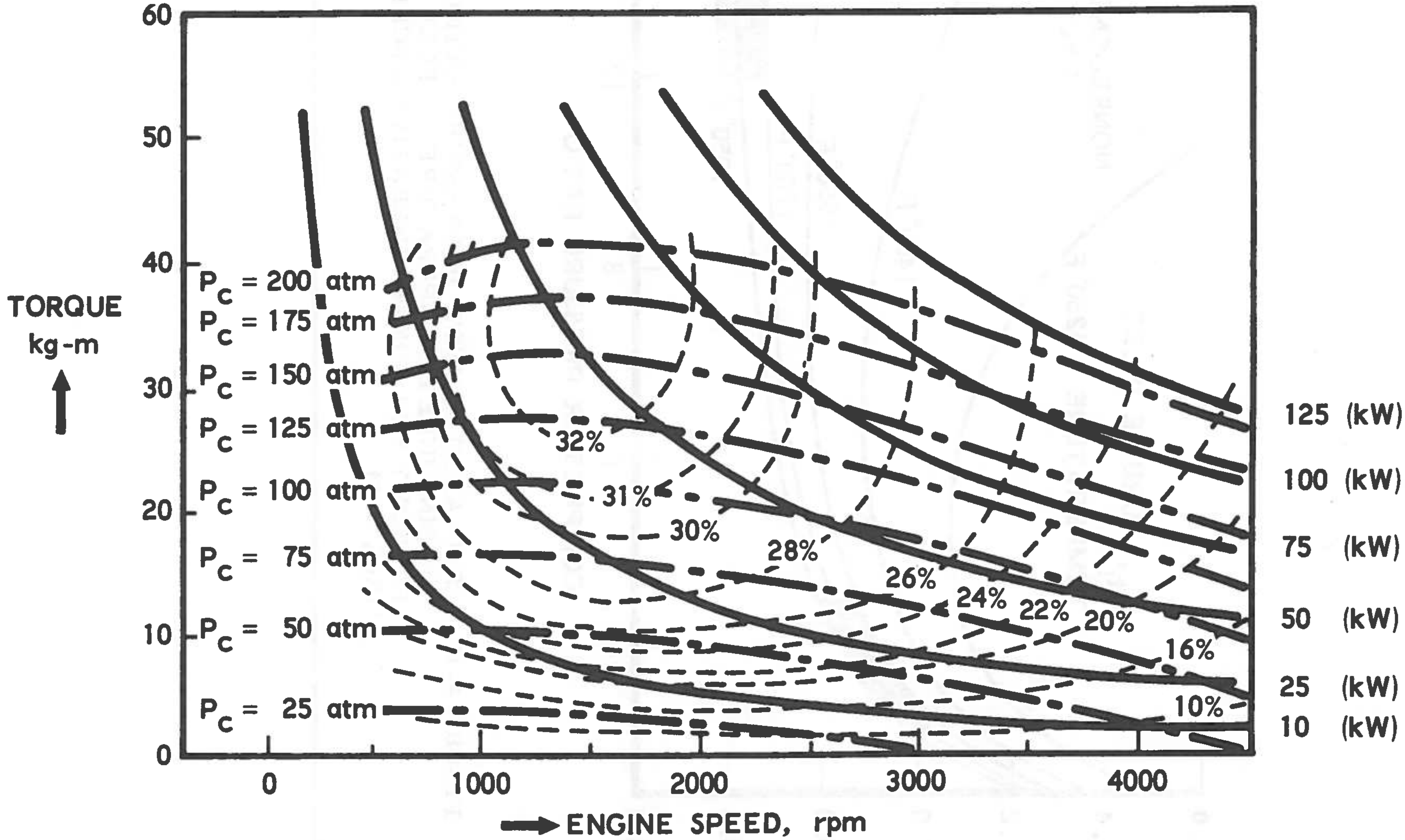


FIGURE 4-16. PERFORMANCE MAP OF FORD TORINO STIRLING ENGINE (Ref. 4-1)

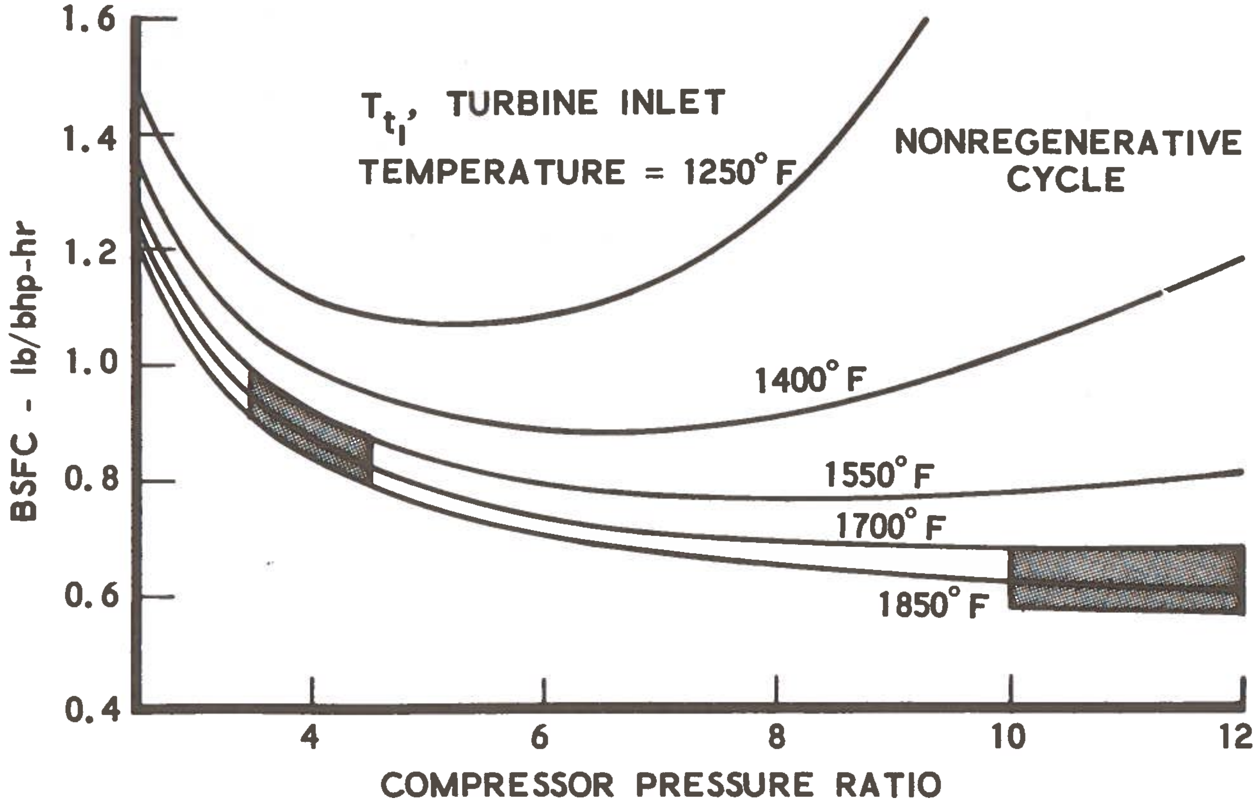


FIGURE 4-17. ESTIMATED EFFECT OF PRESSURE RATIO AND TURBINE INLET TEMPERATURE ON FUEL ECONOMY FOR NONREGENERATIVE CYCLE (Ref. 4-1)

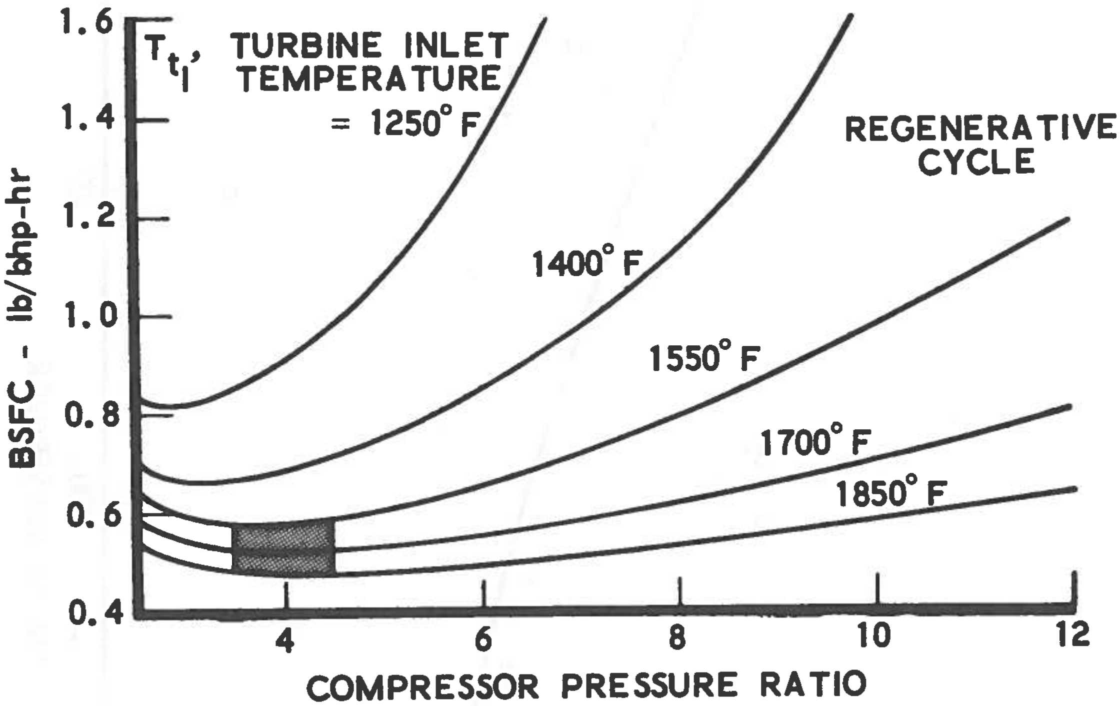


FIGURE 4-18. ESTIMATED EFFECT OF PRESSURE RATIO AND TURBINE INLET TEMPERATURE ON FUEL ECONOMY FOR REGENERATIVE CYCLE (Ref. 4-1)

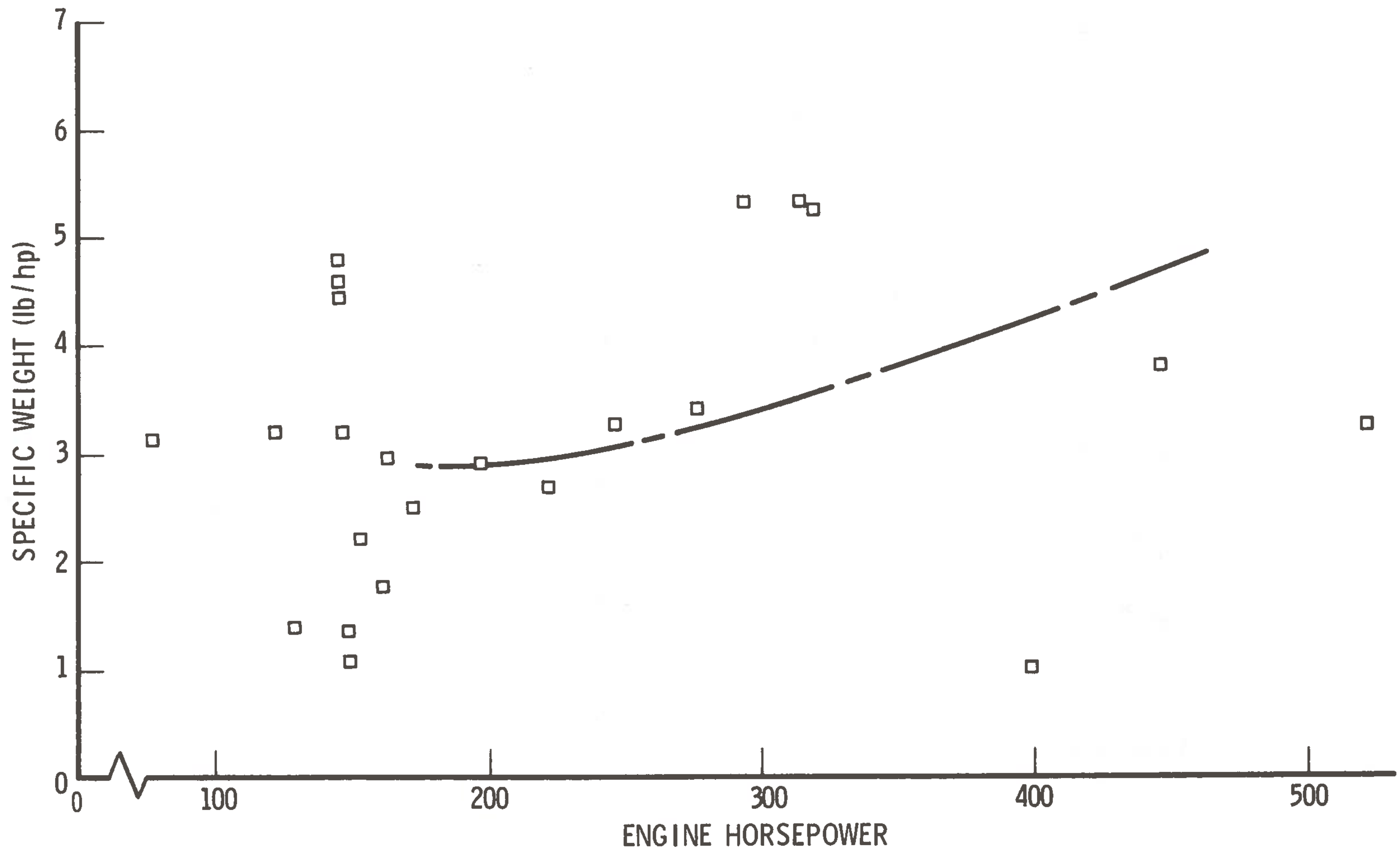


FIGURE 4-19. SPECIFIC WEIGHT OF GAS TURBINES FOR AUTOMOTIVE APPLICATIONS (Derived from Table 4-1)

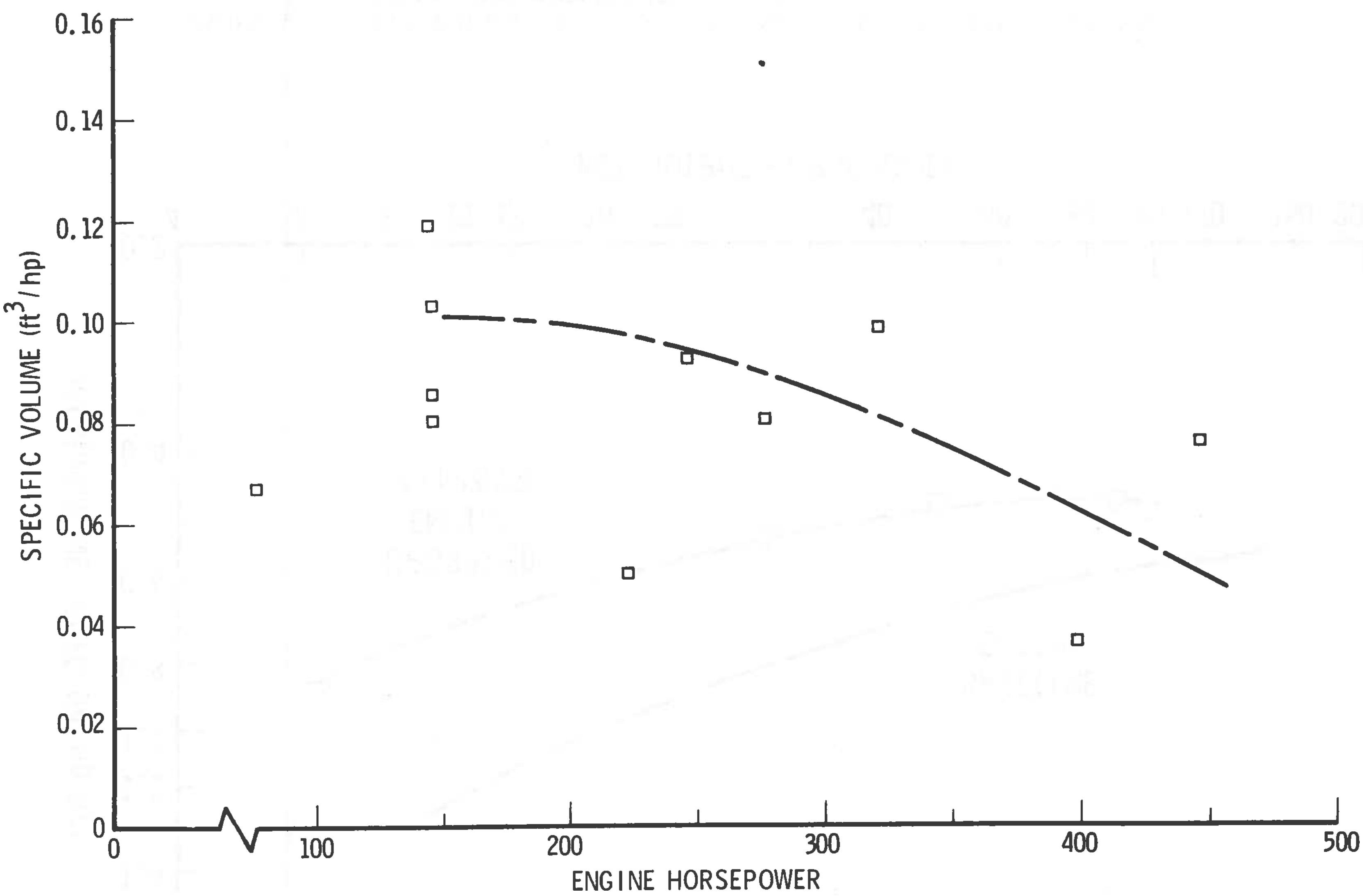


FIGURE 4-20. SPECIFIC VOLUME OF GAS TURBINES FOR AUTOMOTIVE APPLICATIONS (Derived from Table 4-1)

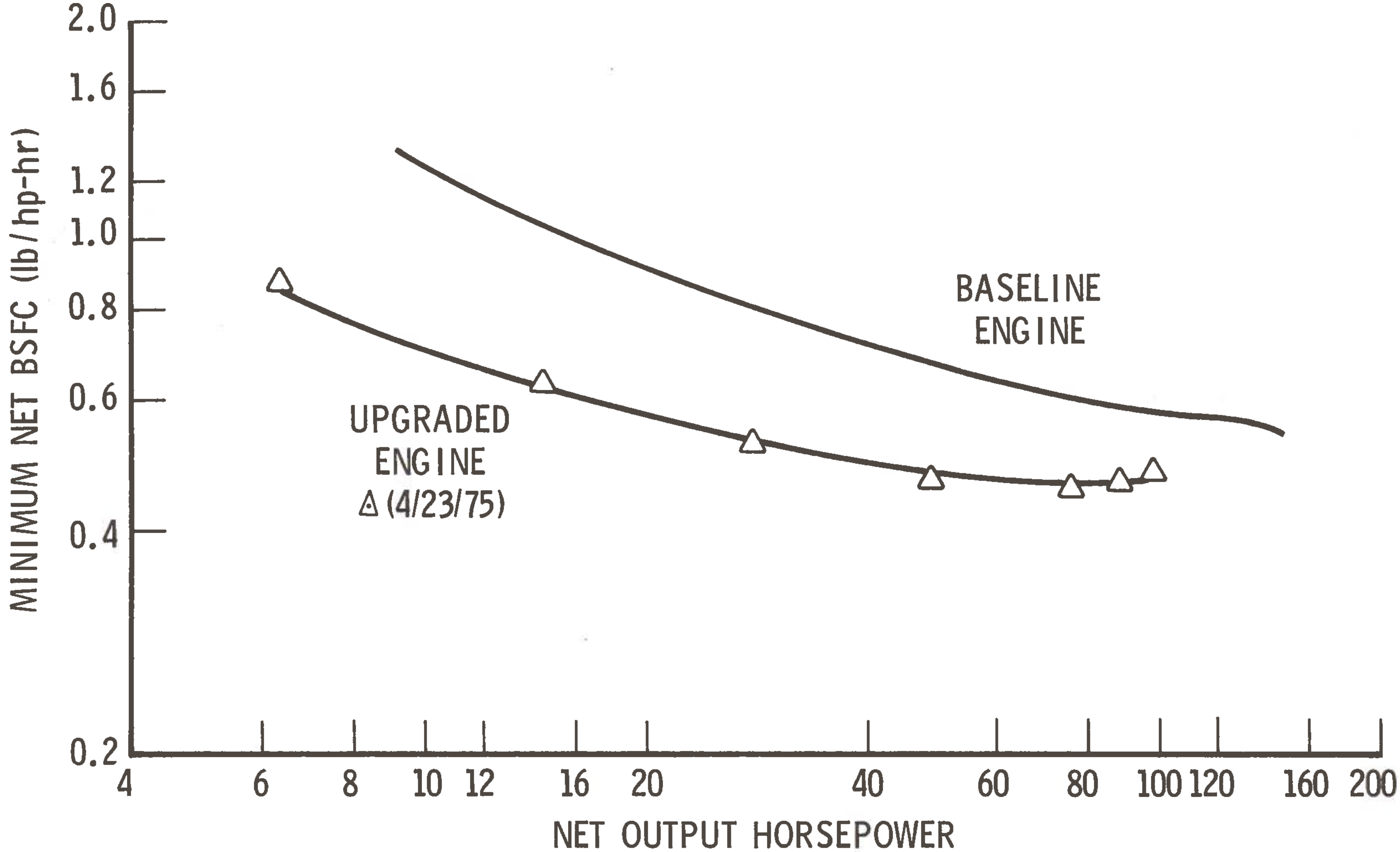


FIGURE 4-21. CHRYSLER GAS TURBINE ENGINE FUEL CONSUMPTION CHARACTERISTICS (Ref. 4-10)

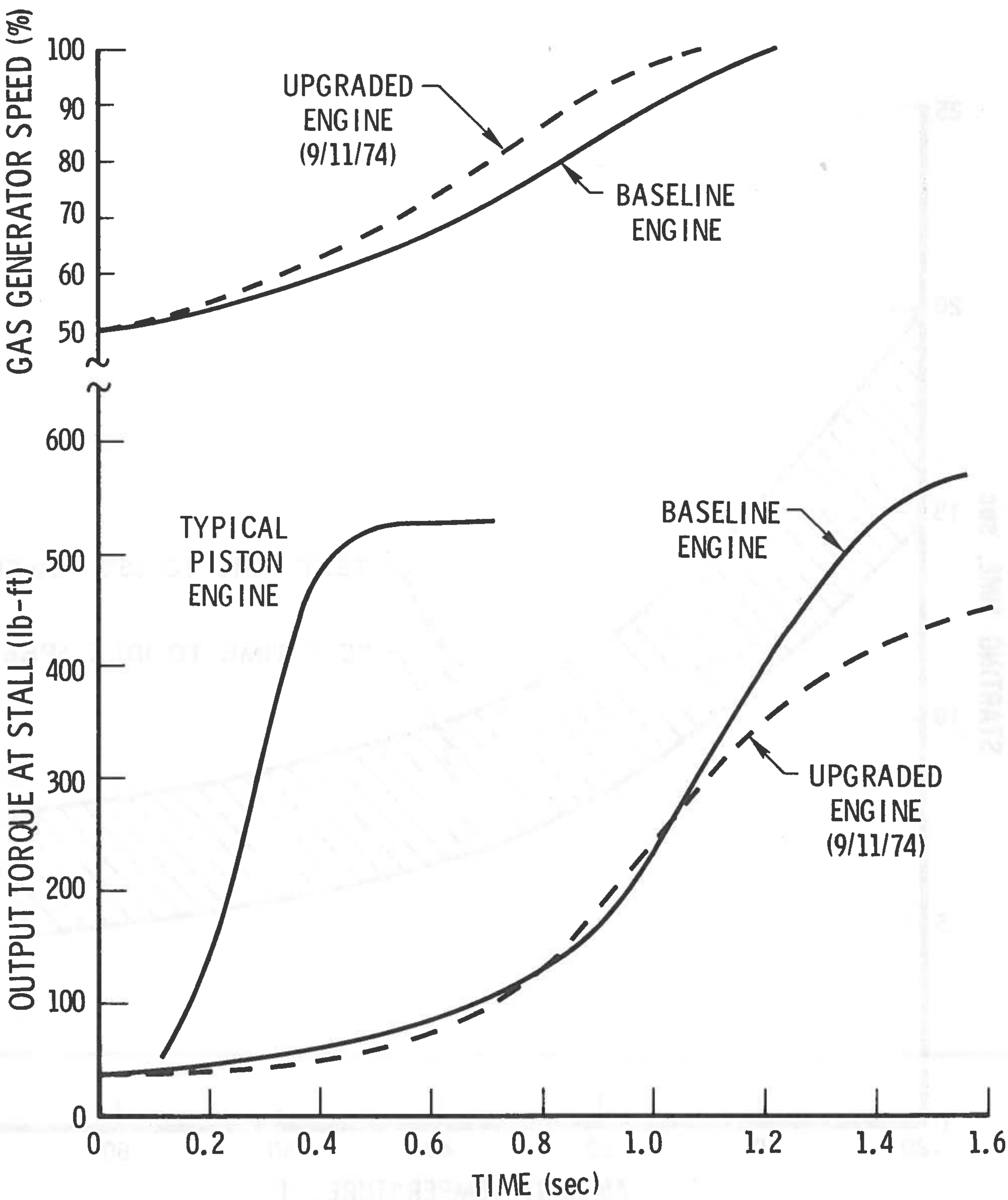


FIGURE 4-22. CHRYSLER GAS TURBINE ENGINE RESPONSE TO POWER DEMAND (Ref. 4-10)

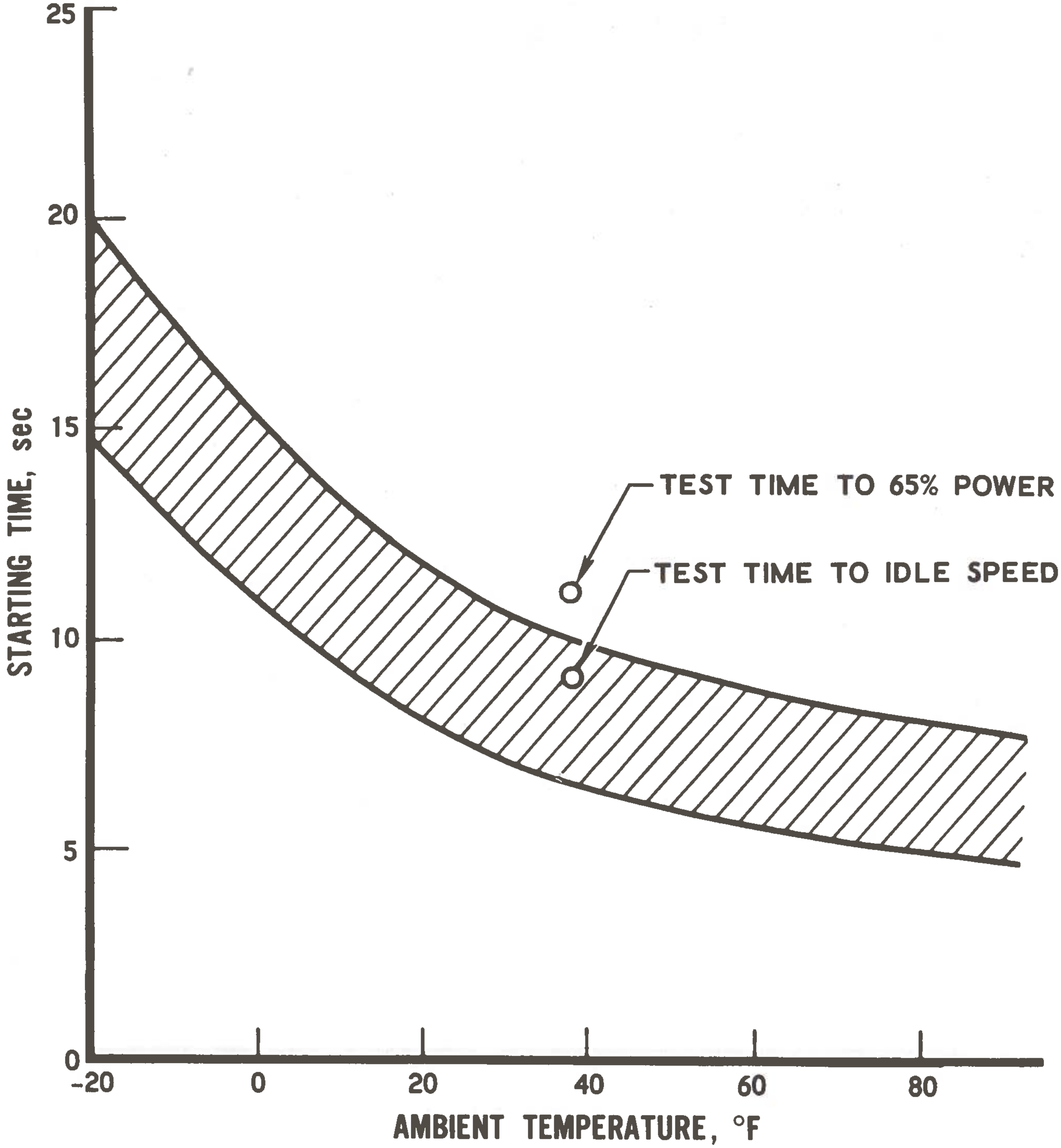
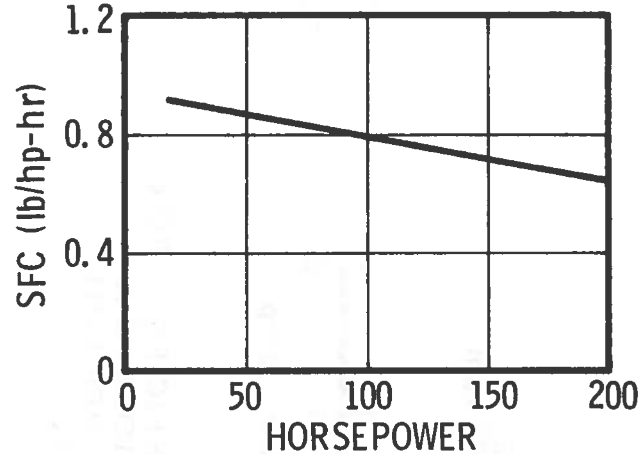
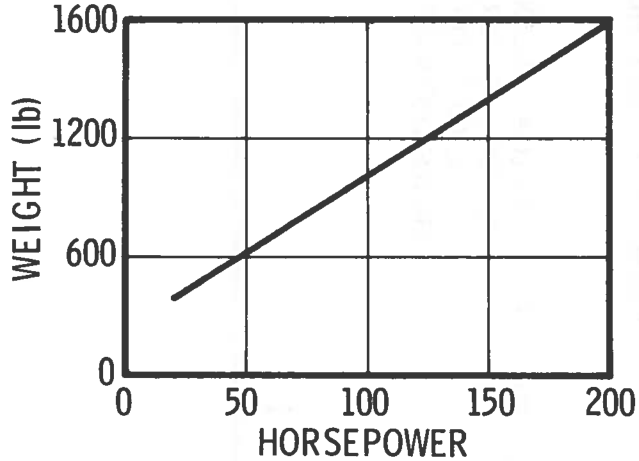


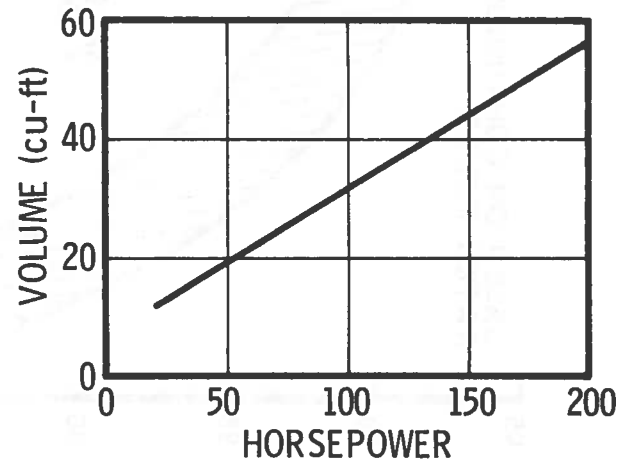
FIGURE 4-23. CHRYSLER GAS TURBINE ENGINE STARTING TIME VERSUS AMBIENT TEMPERATURE (Ref. 4-10)



a. STEAM ENGINE SPECIFIC FUEL CONSUMPTION vs CONTINUOUS HORSEPOWER



b. STEAM ENGINE WEIGHT vs CONTINUOUS HORSEPOWER



c. STEAM ENGINE VOLUME vs CONTINUOUS HORSEPOWER

FIGURE 4-24. STEAM ENGINE CHARACTERISTICS (Ref. 4-3)

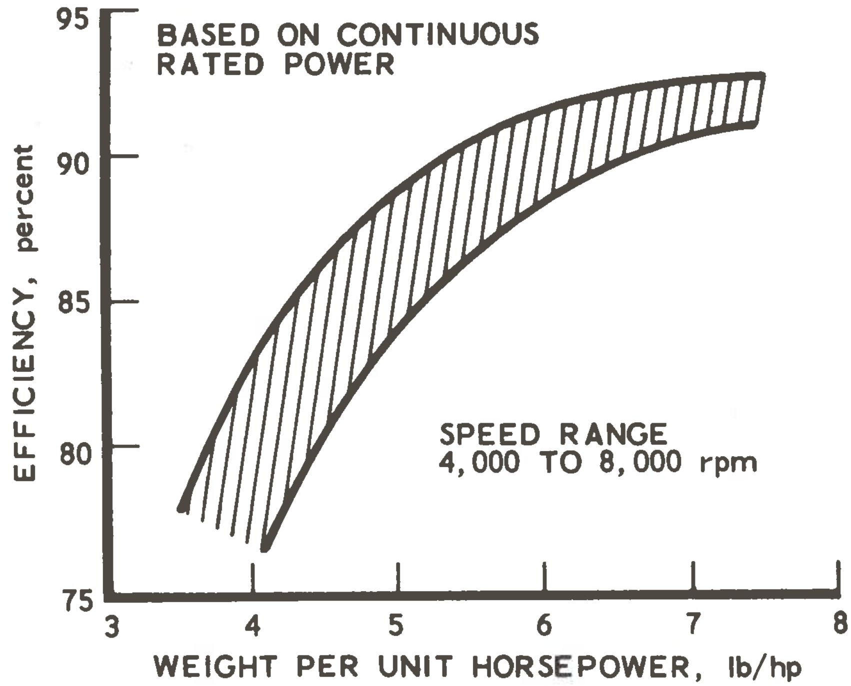


FIGURE 4-25. TYPICAL MAXIMUM EFFICIENCY FOR DIRECT CURRENT MOTORS AS A FUNCTION OF WEIGHT PER UNIT HORSEPOWER (Ref. 4-2)

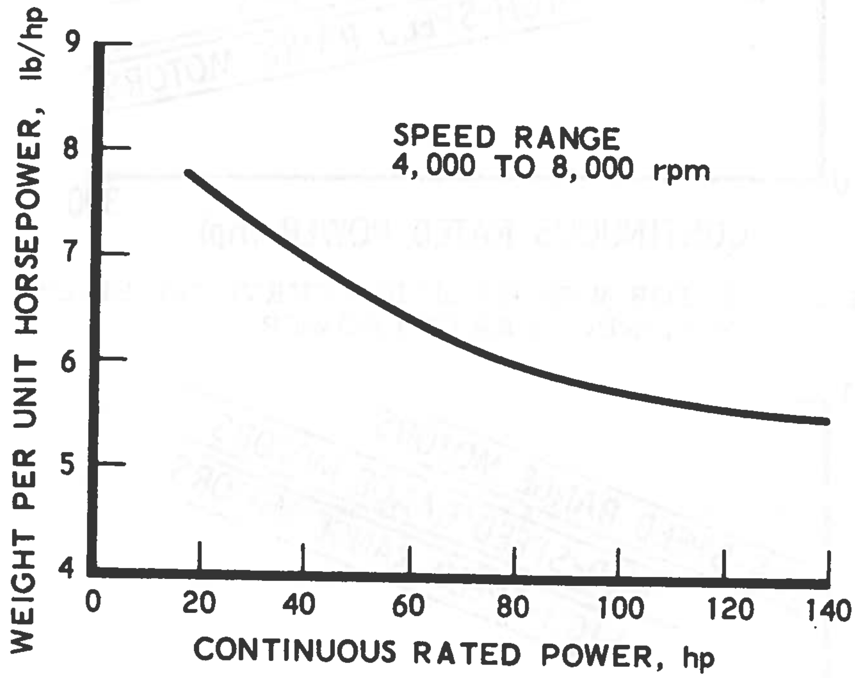


FIGURE 4-26. TYPICAL WEIGHT PER UNIT HORSEPOWER AS A FUNCTION OF RATED POWER FOR DIRECT CURRENT MOTORS INCLUDING FORCED AIR COOLING (Ref. 4-2)

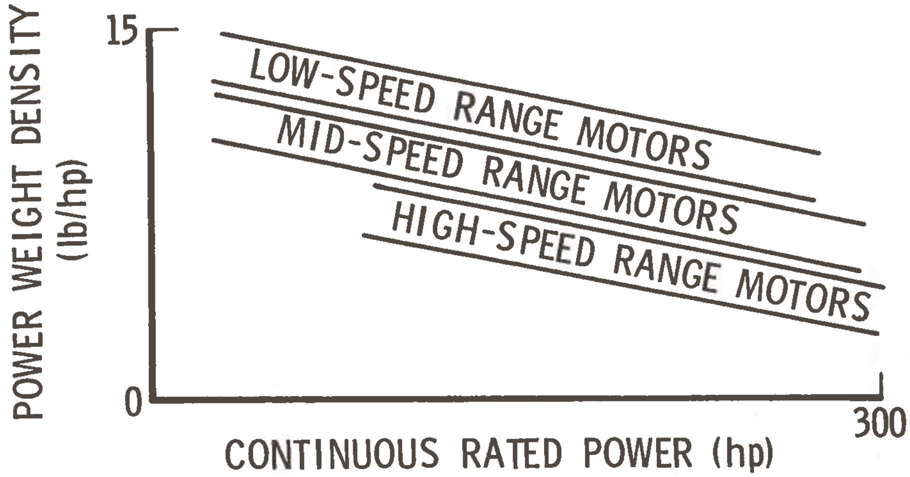


FIGURE 4-27. MOTOR WEIGHT SIZING CURVE BASED ON CONTINUOUS RATED POWER

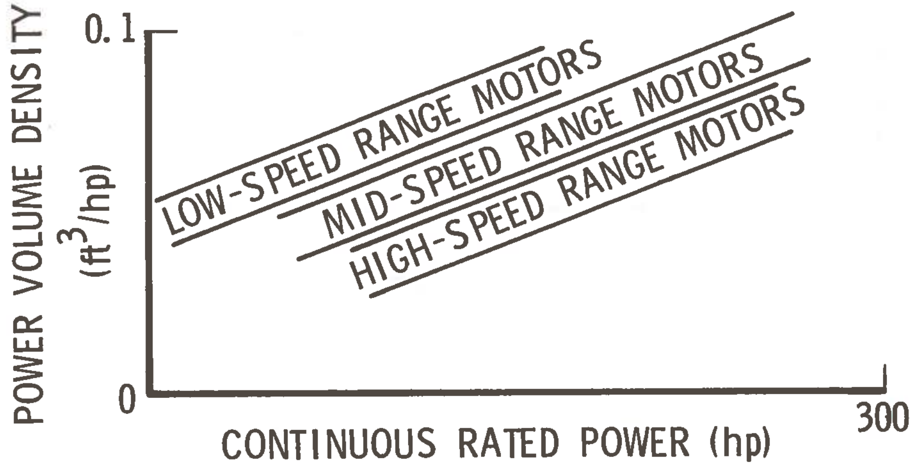


FIGURE 4-28. MOTOR VOLUME SIZING CURVE BASED ON CONTINUOUS RATED POWER

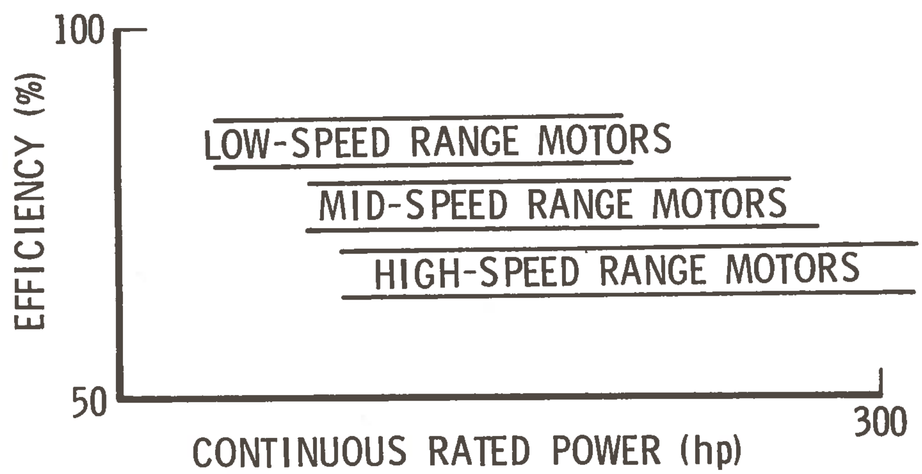


FIGURE 4-29. MOTOR EFFICIENCY AT CONTINUOUS RATED LOAD

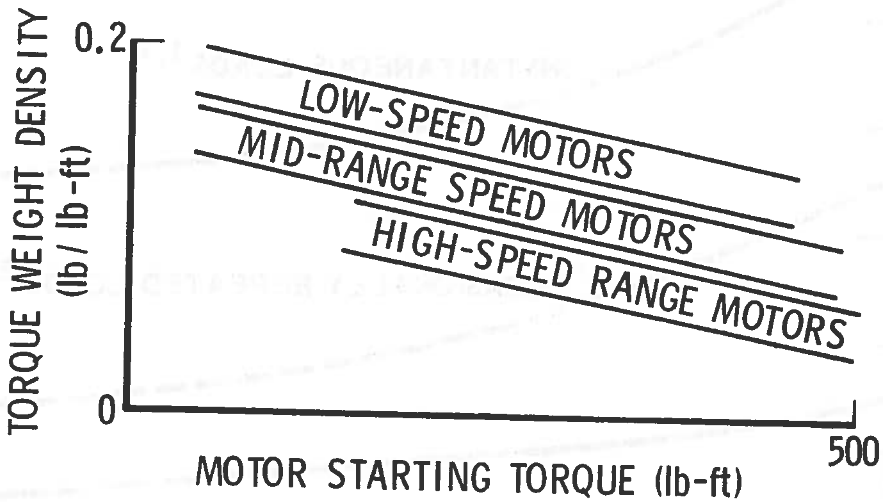


FIGURE 4-30. MOTOR WEIGHT SIZING CURVE BASED ON MOTOR STARTING TORQUE

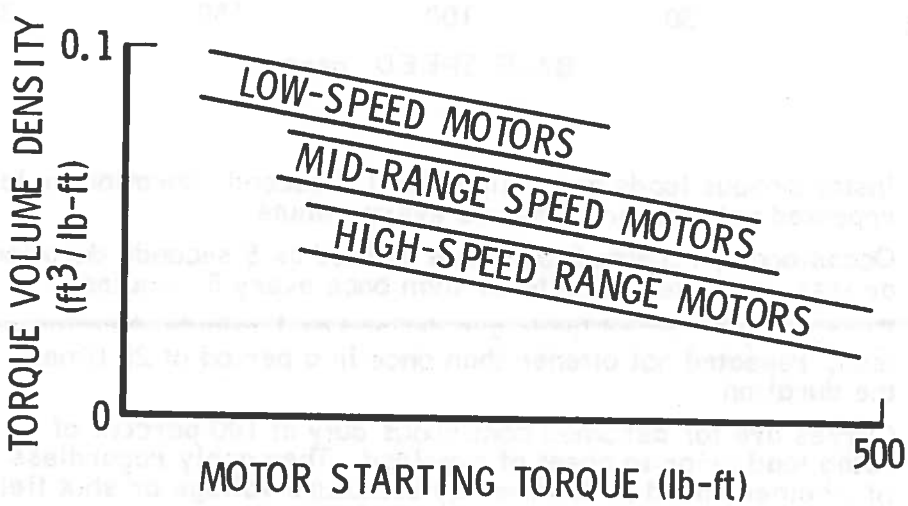
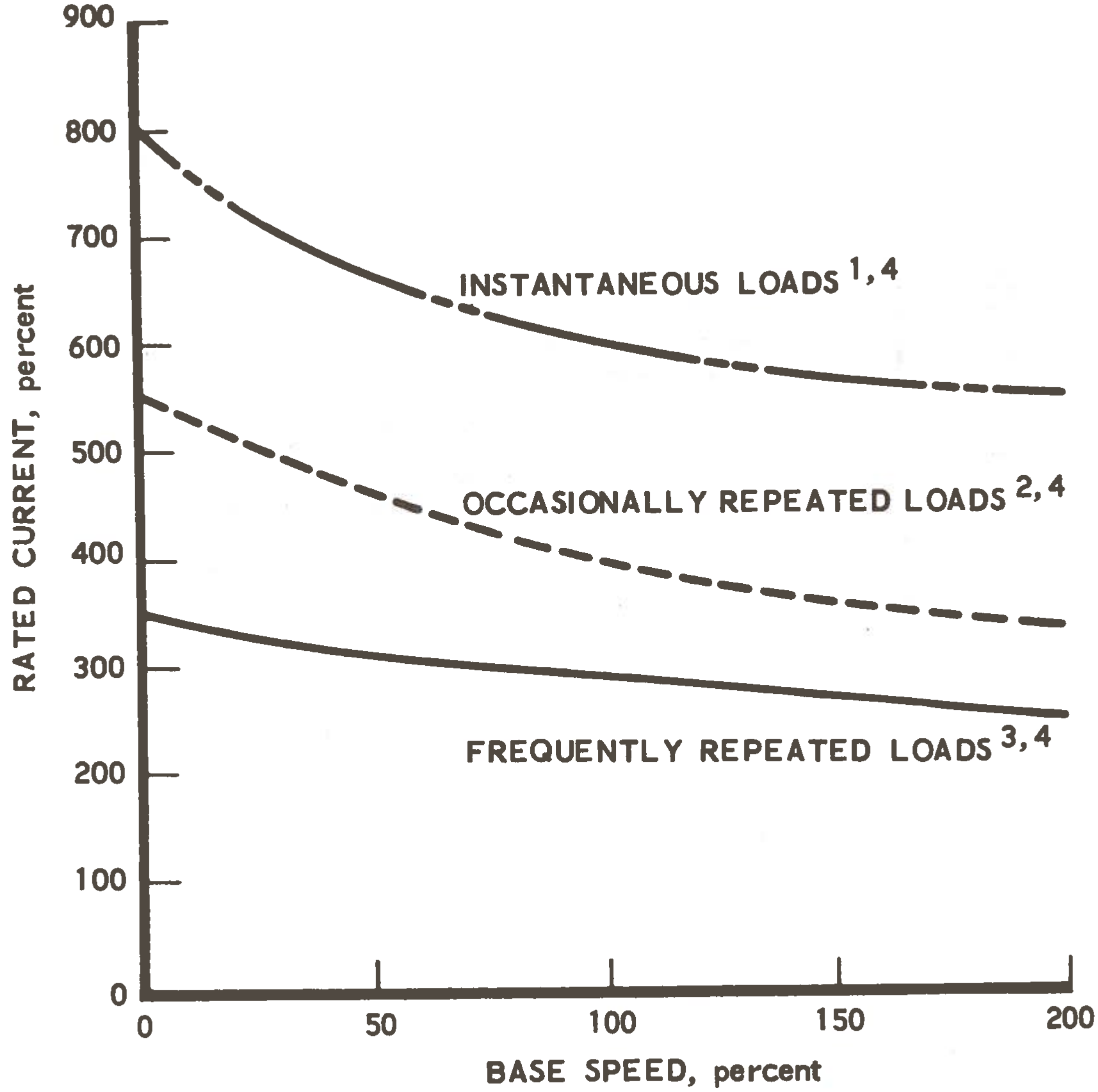


FIGURE 4-31. MOTOR VOLUME SIZING CURVE BASED ON MOTOR STARTING TORQUE



1. Instantaneous loads are defined as 0.5 seconds duration or less, repeated not oftener than once every minute.
2. Occasionally repeated loads are defined as 5 seconds duration or less, repeated not oftener than once every 5 minutes.
3. Frequently repeated loads are defined as 1 minute duration or less, repeated not oftener than once in a period of 20 times the duration.
4. Curves are for assumed continuous duty at 100 percent of rated load prior to onset of overload. They apply regardless of whether speed is obtained by armature voltage or shut field control, and they also apply for regenerating operations.

FIGURE 4-32. OVERLOAD CAPABILITY OF COMPENSATED DIRECT CURRENT MOTORS (Ref. 4-14)

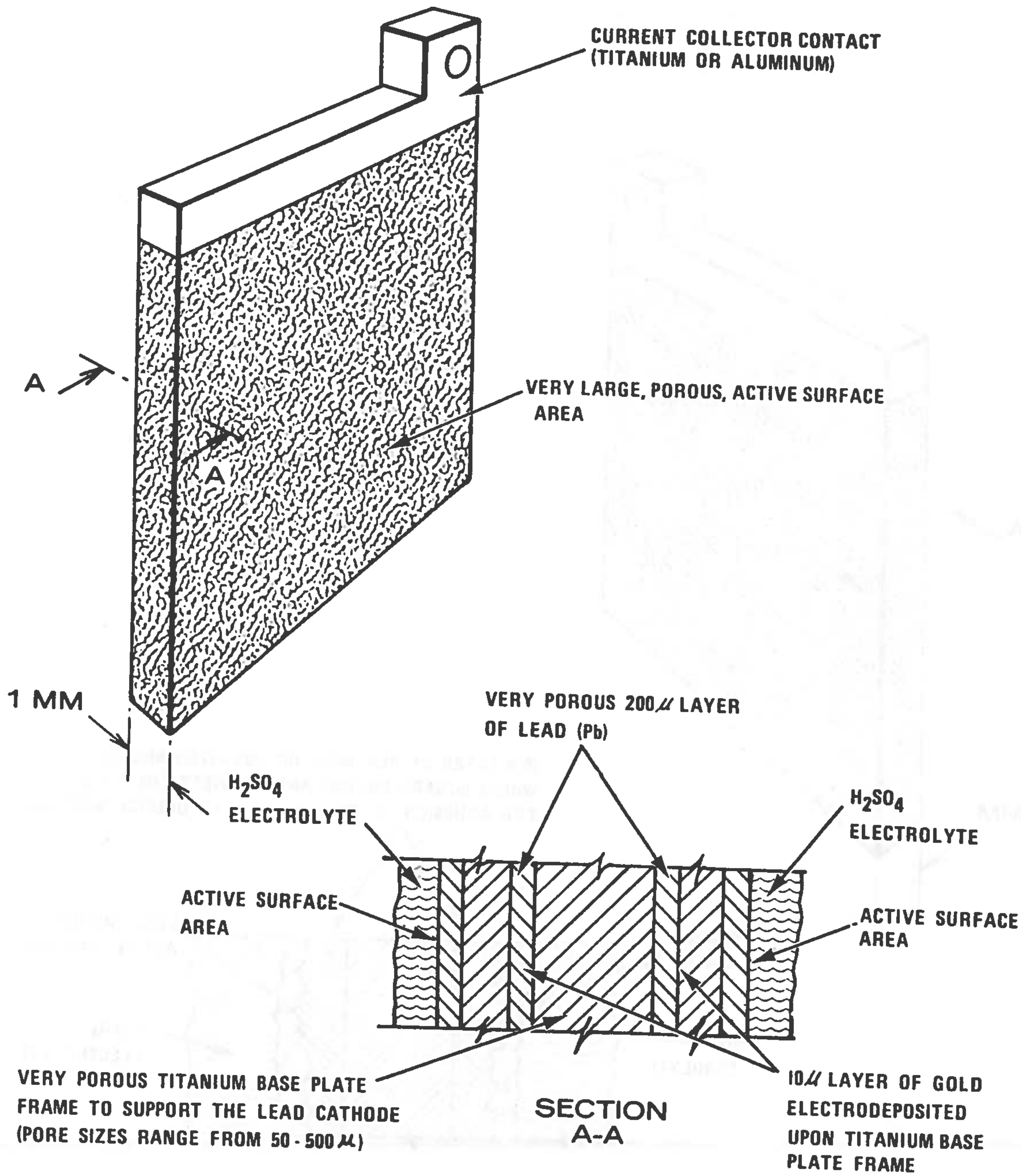


FIGURE 4-33. SECTIONAL DIAGRAM OF LIGHTWEIGHT LEAD-ACID ANODE BY RHEINESCHWESFAELISCHER ELECTRIZITATS WERKE (RWE) OF WEST GERMANY (Ref. 4-18)

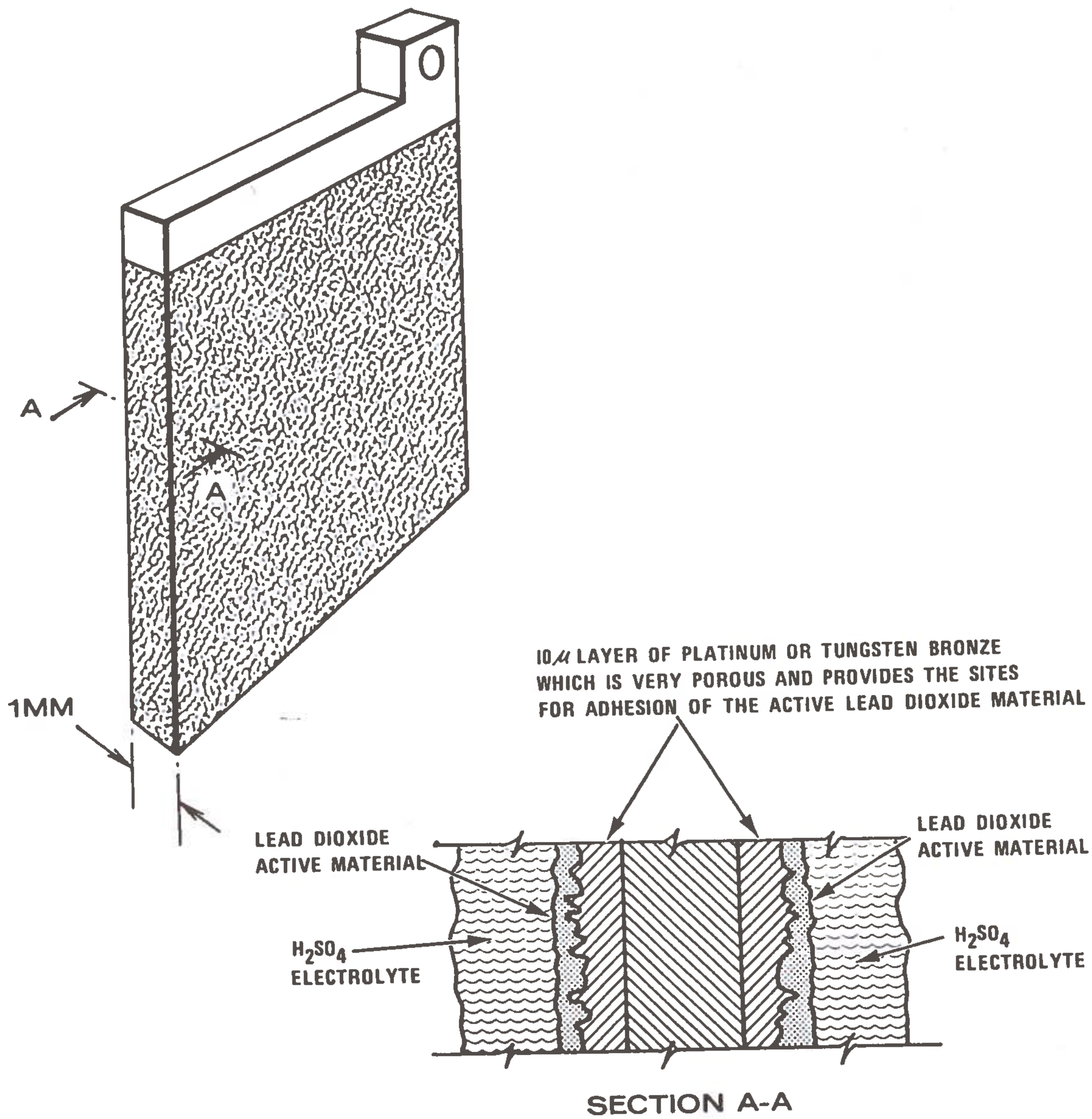


FIGURE 4-34. SECTIONAL DIAGRAM OF LEAD-ACID LIGHTWEIGHT CATHODE BY RWE OF WEST GERMANY (Ref. 4-18)

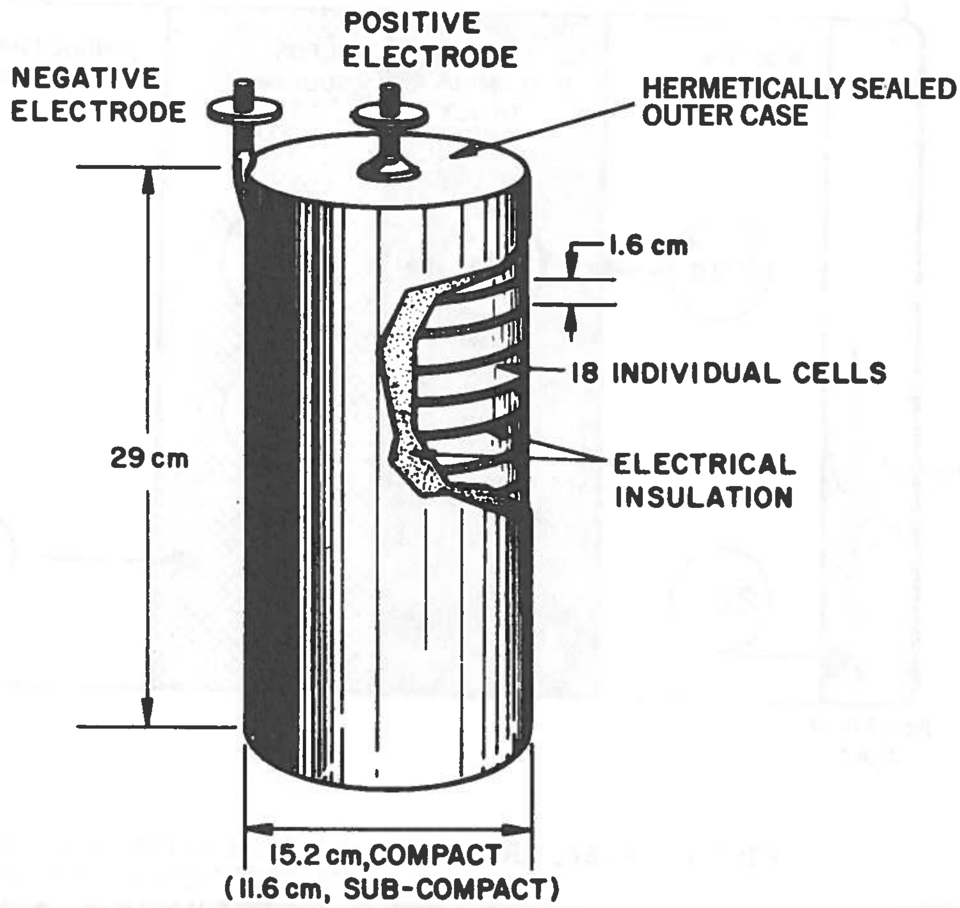


FIGURE 4-35. CONSTRUCTION OF PROTOTYPE LITHIUM-SULFUR TRACTION BATTERY BY ARGONNE NATIONAL LABORATORIES (Ref. 4-18)

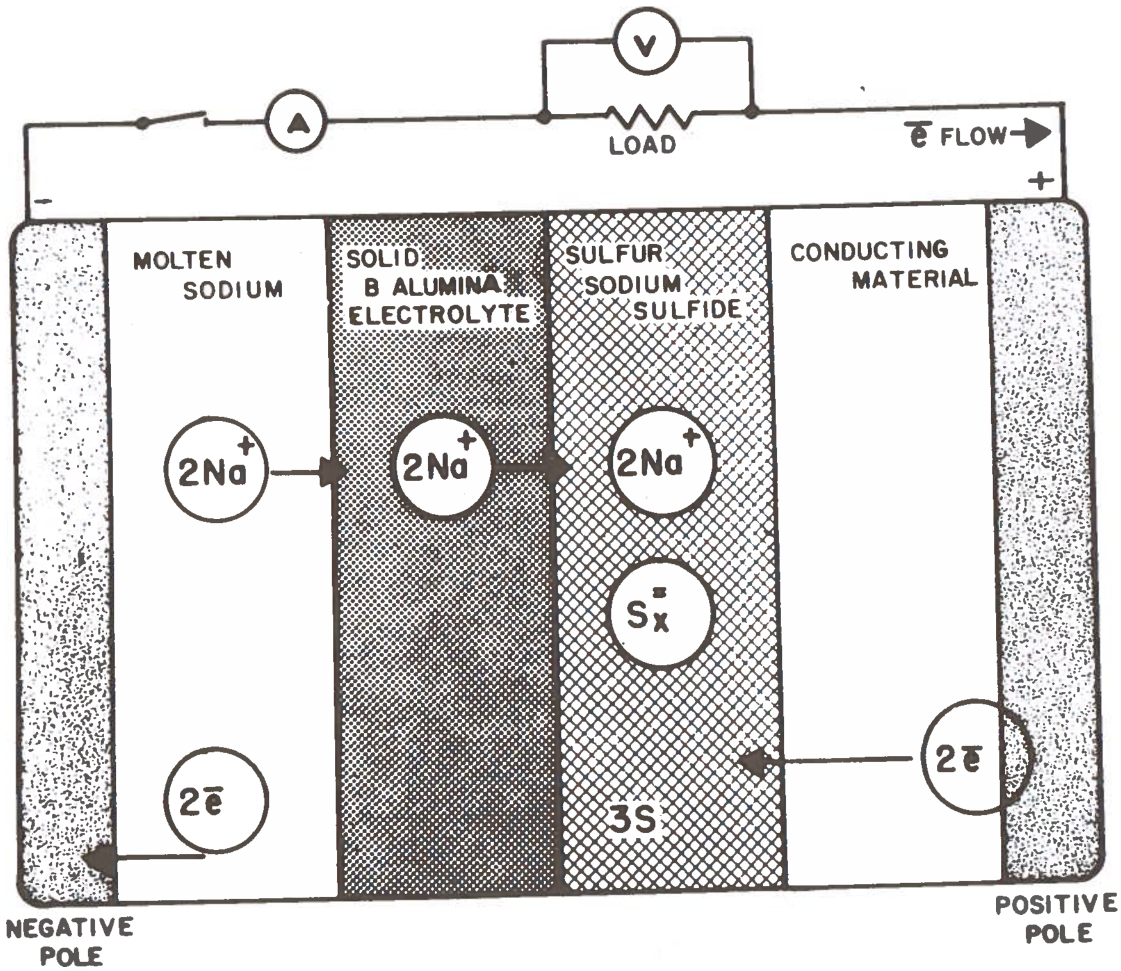


FIGURE 4-36. REACTION MECHANISM FOR THE SODIUM-SULFUR HIGH-ENERGY TRACTION BATTERY (Ref. 4-18)

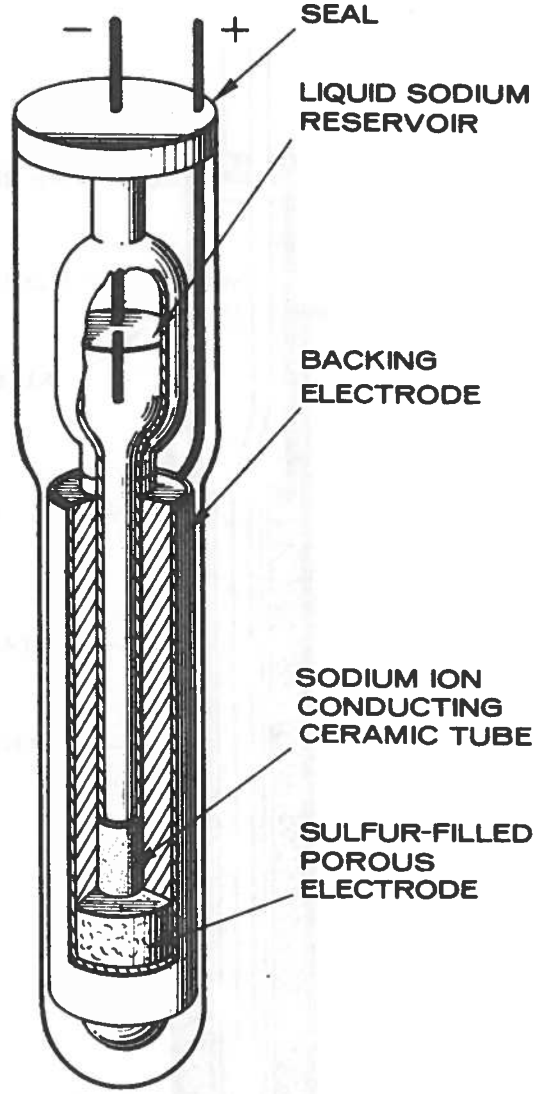


FIGURE 4-37. ADVANCED SODIUM-SULFUR CELL DEVELOPED BY FORD MOTOR COMPANY (Ref. 4-18)

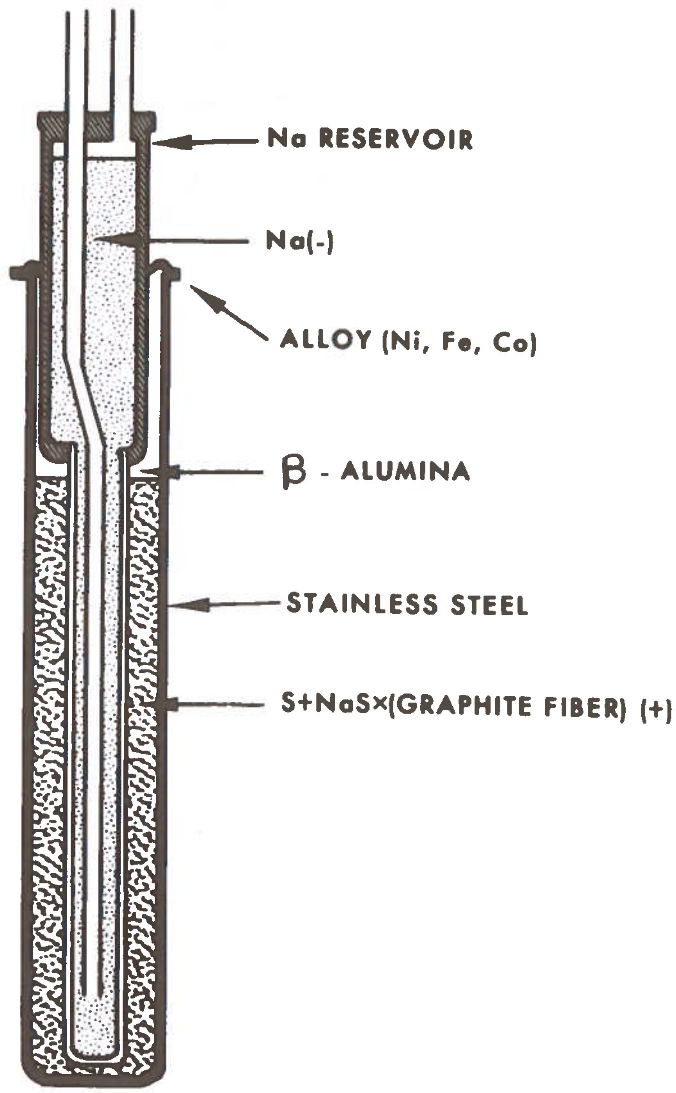


FIGURE 4-38. SECTIONAL DIAGRAM OF ADVANCED SODIUM-SULFUR BATTERY BY YUASA BATTERY CO., LTD. (Ref. 4-18)

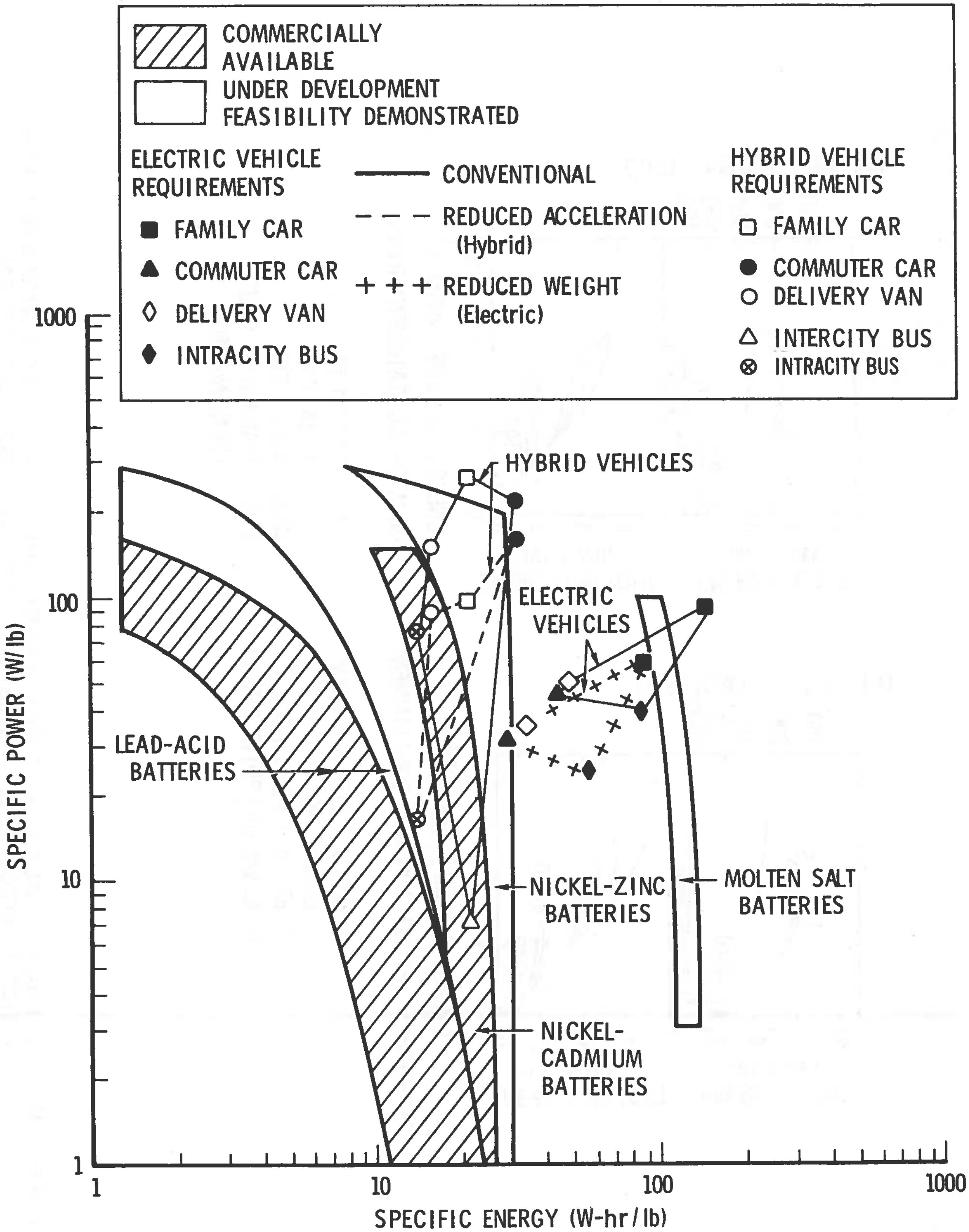


FIGURE 4-39. COMPARISON OF BATTERY PERFORMANCE CAPABILITIES AND VEHICLE REQUIREMENTS

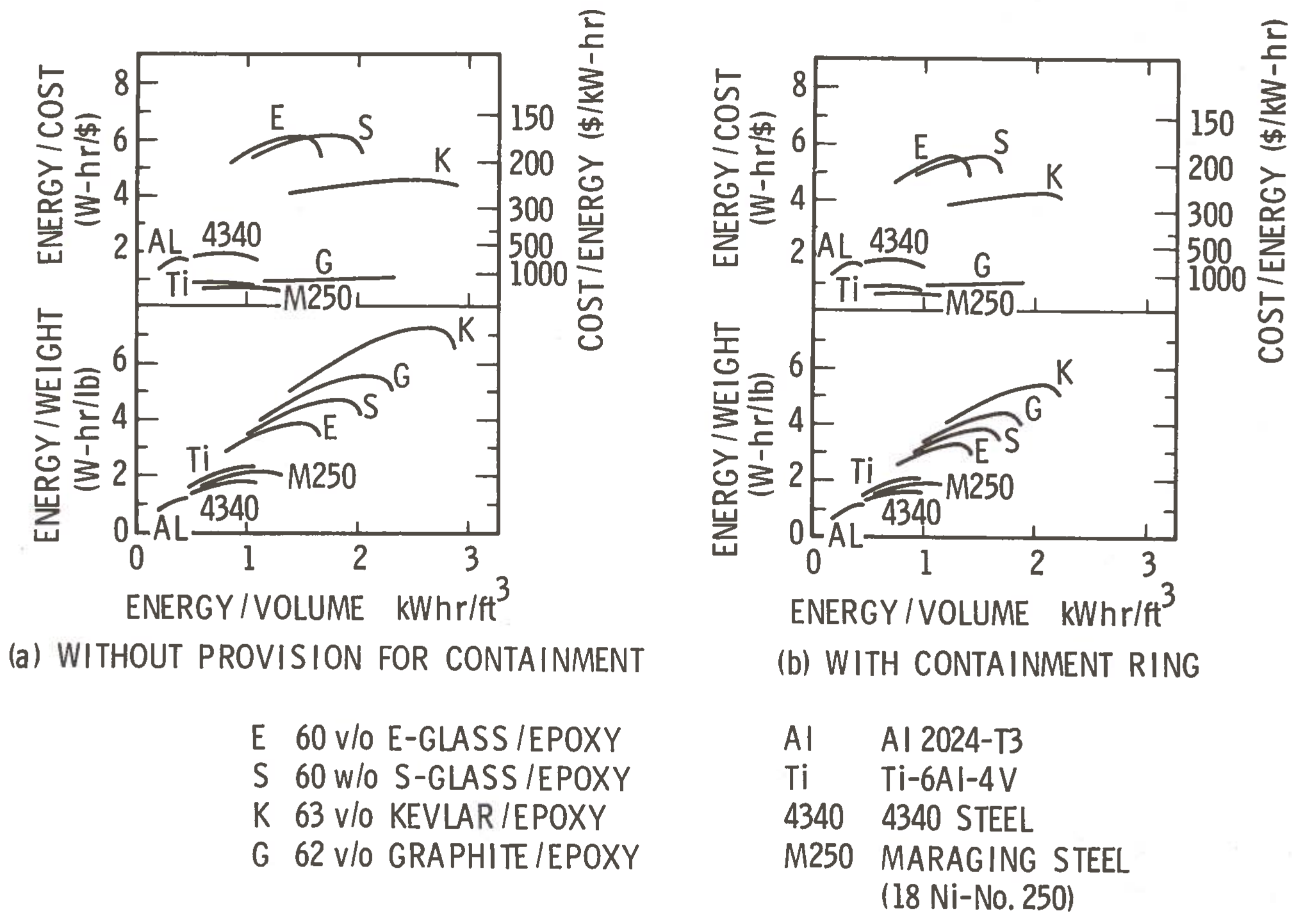


FIGURE 4-40. ESTIMATED FLYWHEEL SUBSYSTEM SPECIFIC ENERGY AND COST FOR VARIOUS ROTOR MATERIALS AND FOR LIFE (OR INSPECTION INTERVAL) OF 10^5 CYCLES (Ref. 4-39)

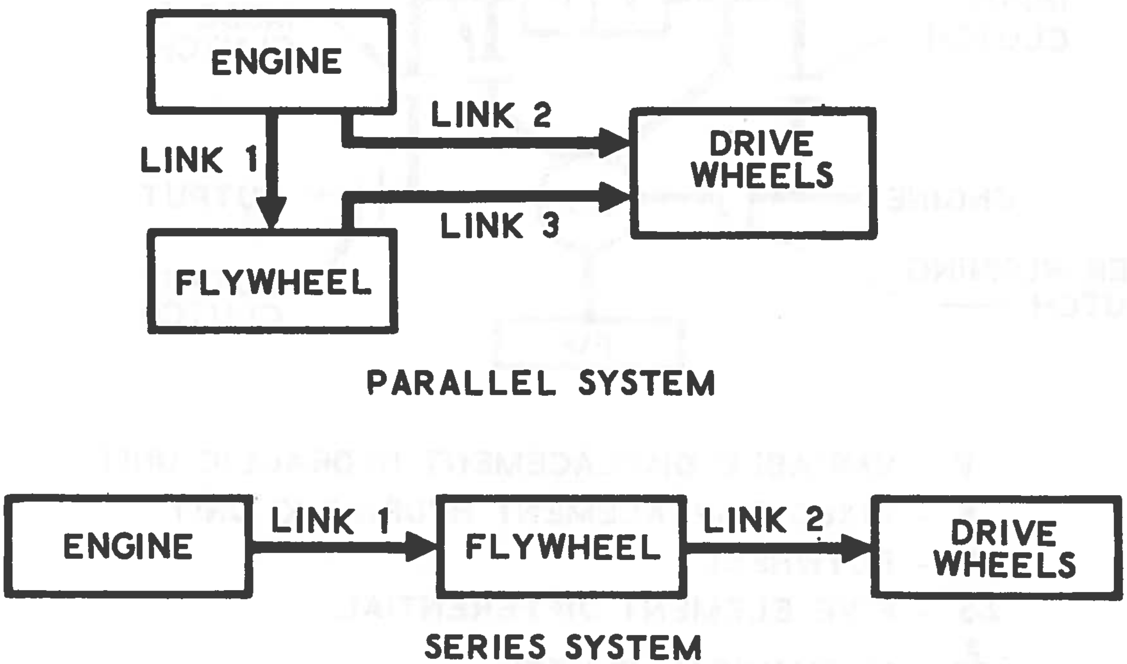
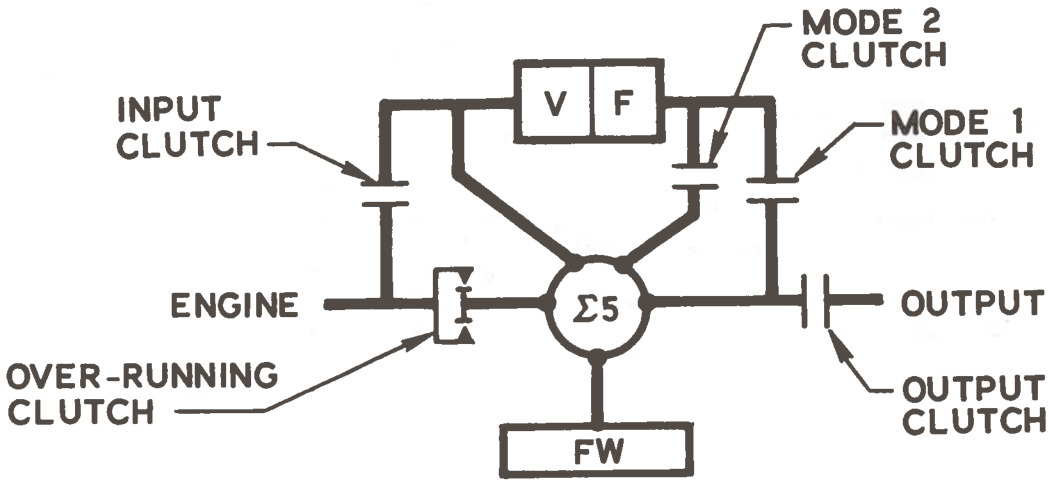


FIGURE 4-41. PARALLEL AND SERIES CONFIGURATIONS FOR ENERGY FLOW IN HYBRID VEHICLES (Ref. 4-40)



V - VARIABLE DISPLACEMENT HYDRAULIC UNIT
F - FIXED DISPLACEMENT HYDRAULIC UNIT
FW - FLYWHEEL
$\Sigma 5$ - FIVE ELEMENT DIFFERENTIAL
- MECHANICAL CLUTCH

FIGURE 4-42. SUNDSTRAND ALTERNATE 8C TRANSMISSION (Ref. 4-40)

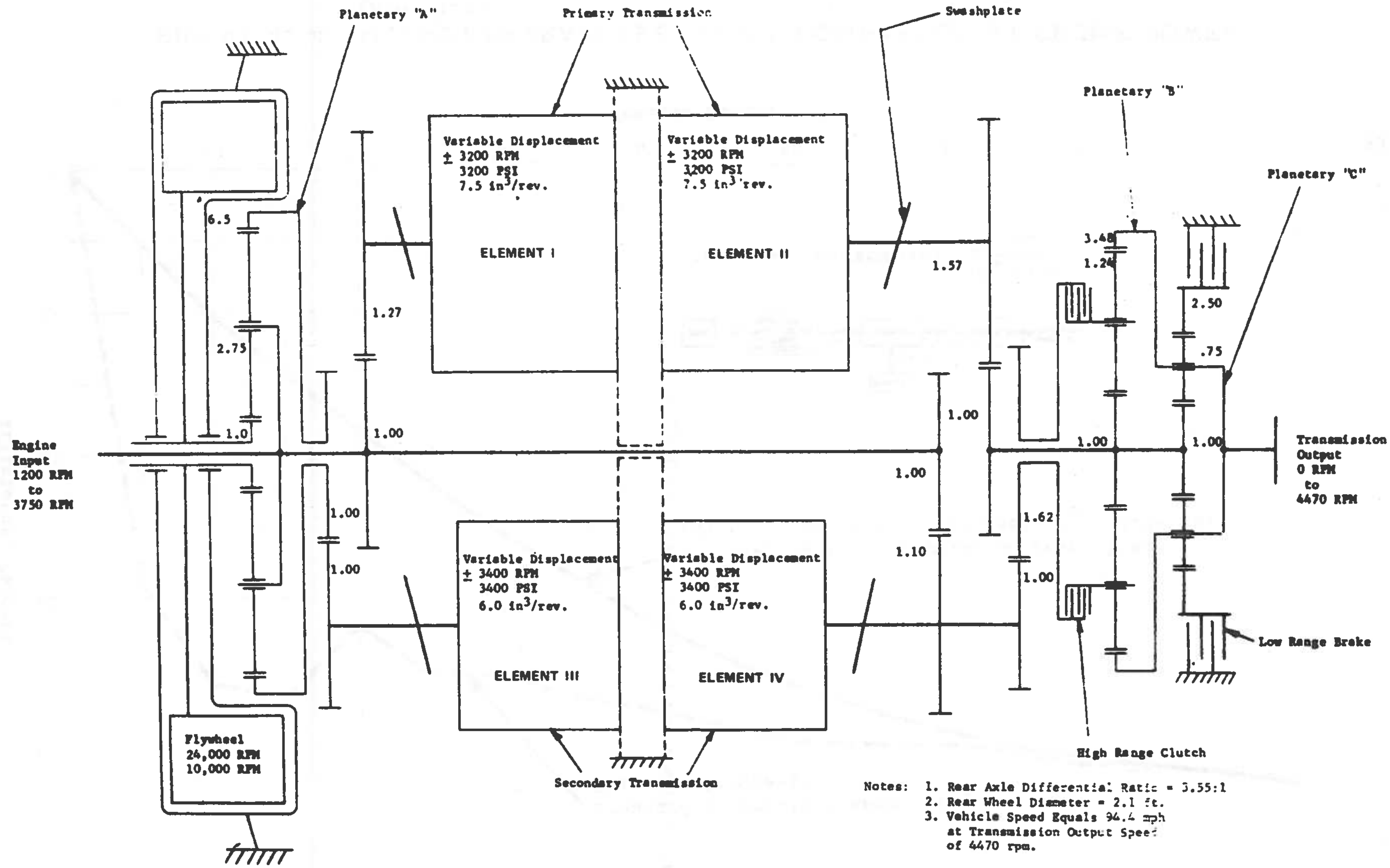


FIGURE 4-43. SCHEMATIC OF THE MTI RECOMMENDED TRANSMISSION DESIGN (Ref. 4-41)

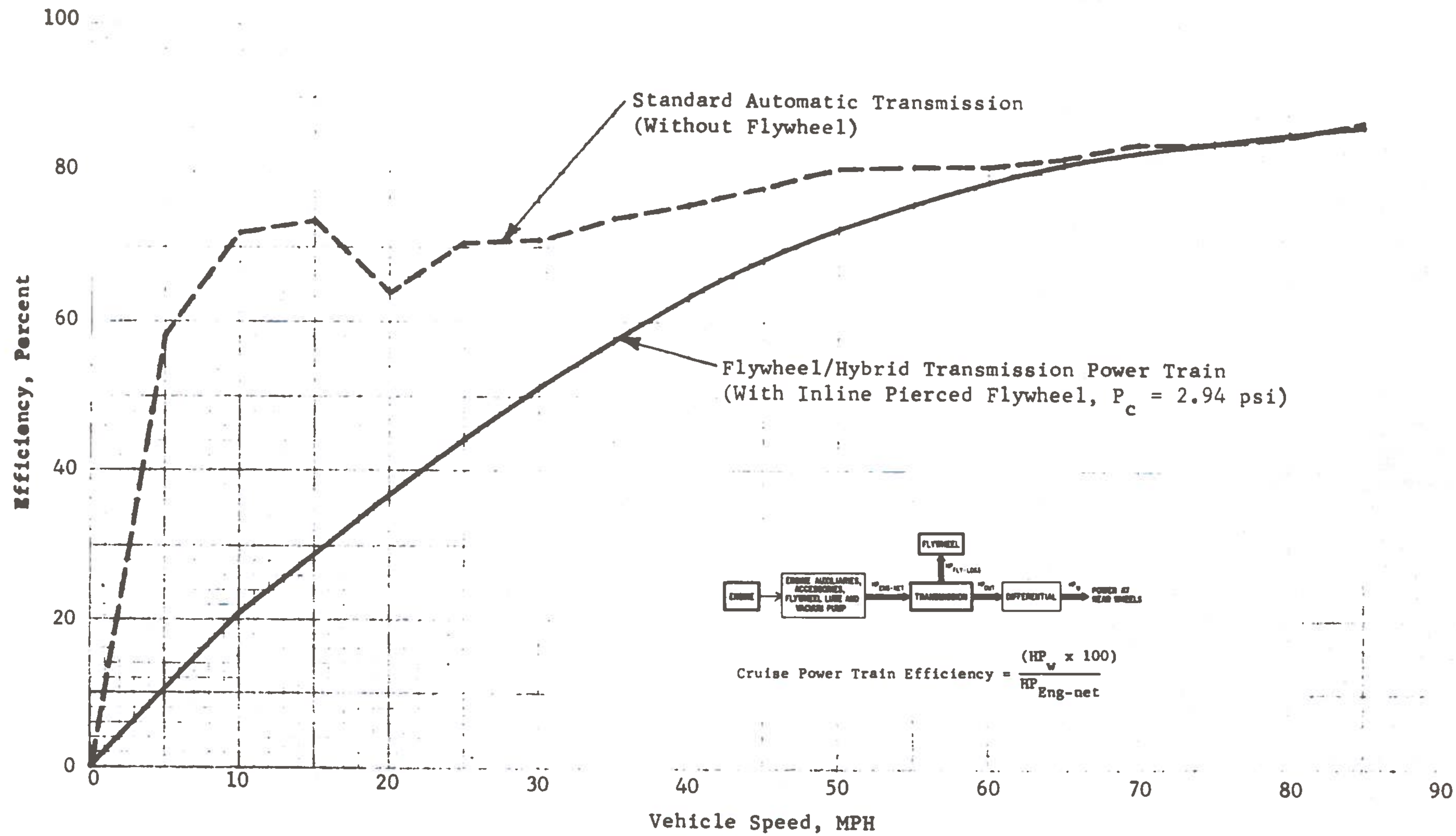


FIGURE 4-44. MTI POWERTRAIN EFFICIENCY COMPARISON AT CRUISE POWER (Ref. 4-41)

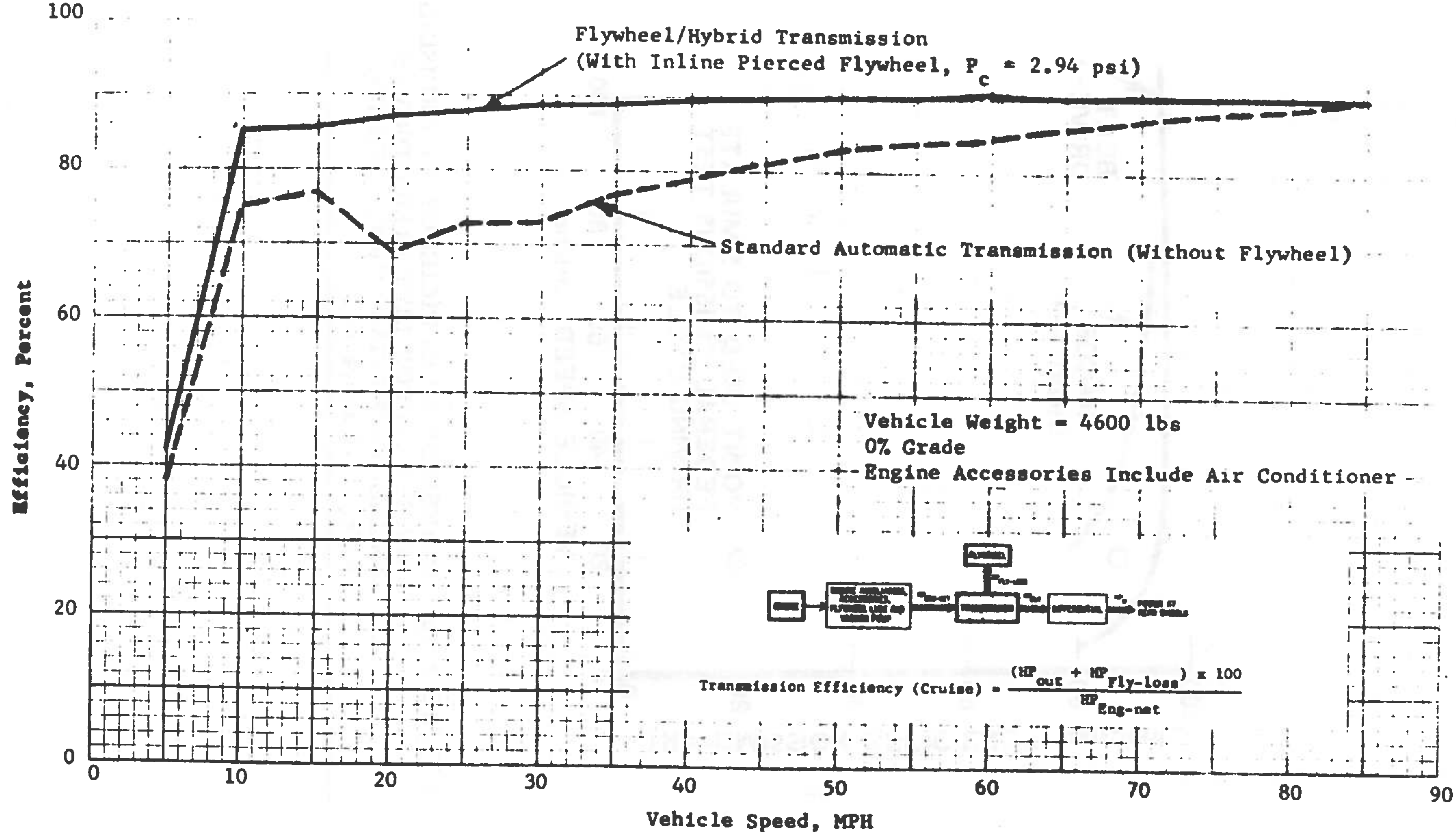


FIGURE 4-45. MTI COMPARISON OF TRANSMISSION EFFICIENCIES AT CRUISE POWER (Ref. 4-41)

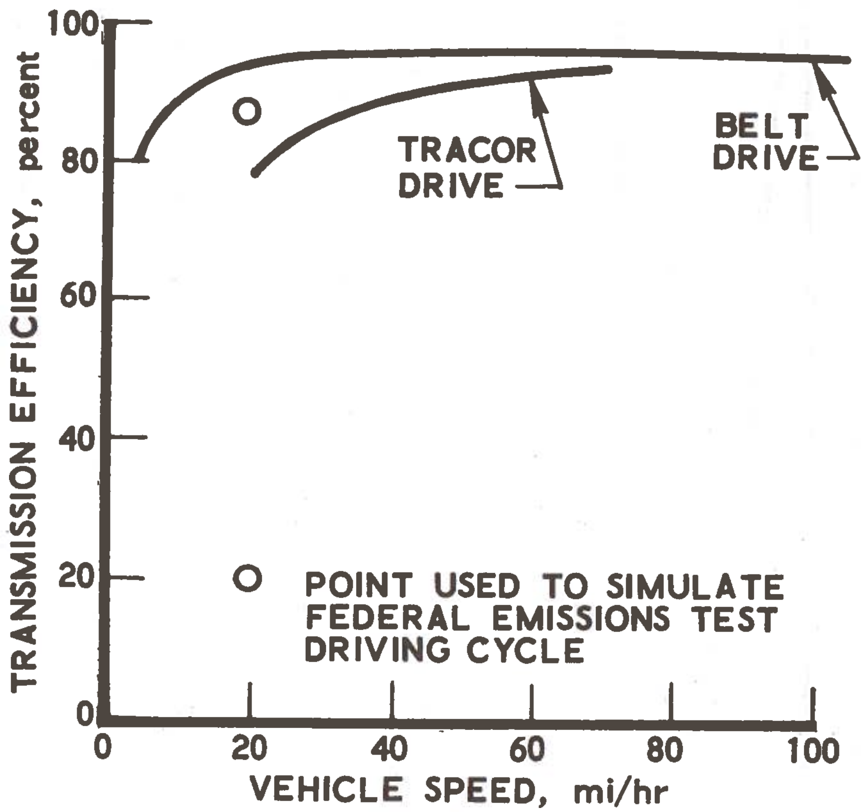


FIGURE 4-46. TRANSMISSION EFFICIENCY AT CRUISING CONDITIONS FOR DIFFERENT DRIVES - SINGLE-SHAFT GAS TURBINE ENGINE (Refs. 4-42 and 4-44)

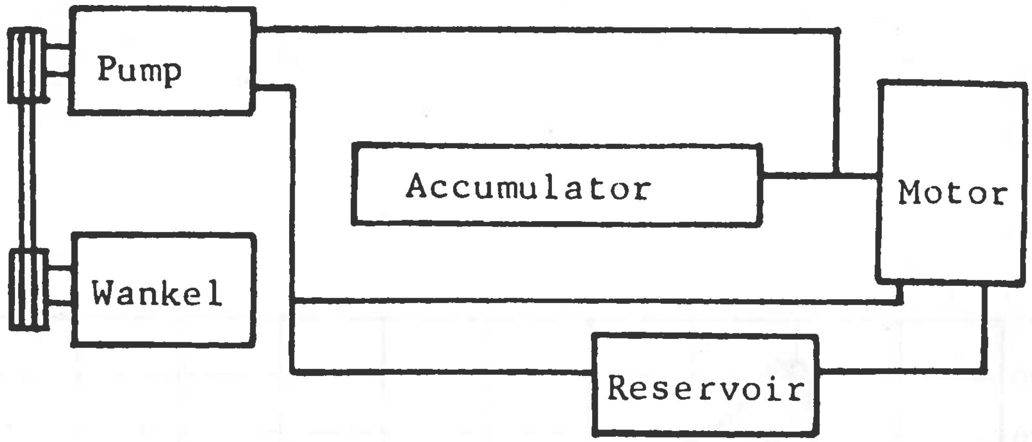


FIGURE 4-47. HYBRID HYDRAULIC POWERPLANT (Ref. 4-48)

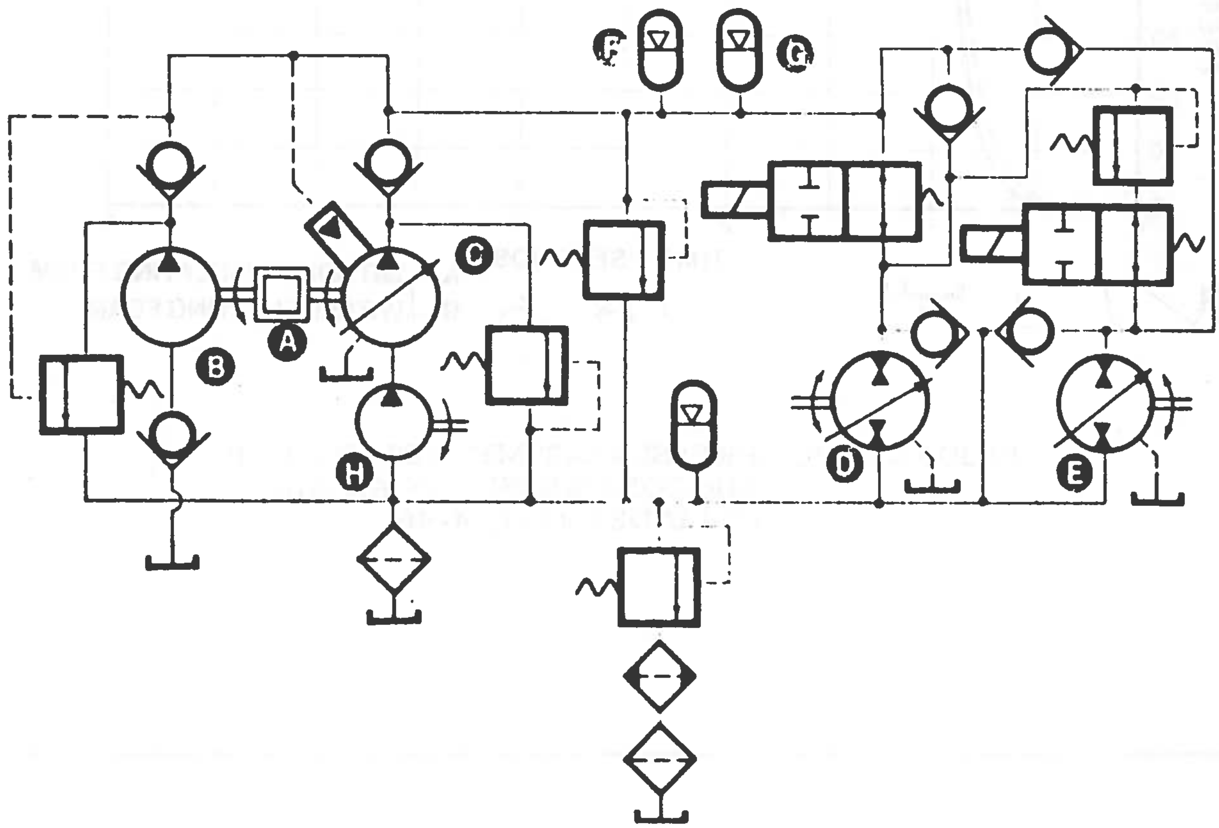


FIGURE 4-48. ACCUMULATOR-CHARGED HYDRAULIC CIRCUIT FOR PROPOSED INTERMEDIATE SPEED, START-AND-GO VEHICLE (Ref. 4-45)

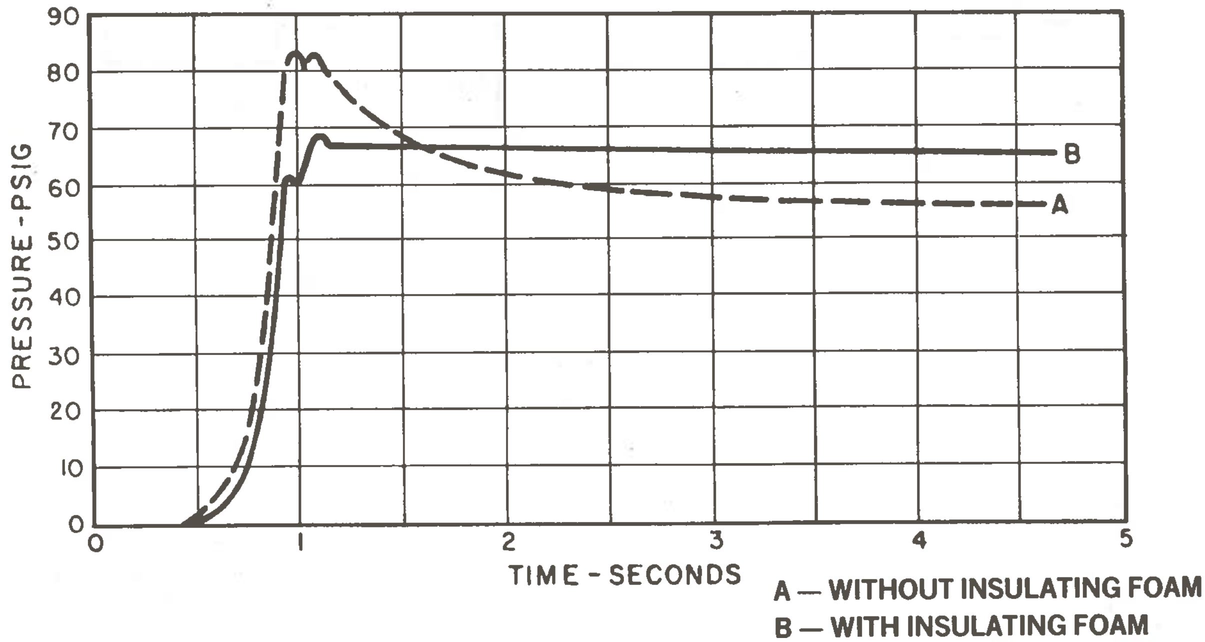


FIGURE 4-49. PRESSURE-TIME HISTORY FOR AIR COMPRESSED IN AN AIR CYLINDER (Ref. 4-46)