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The Otto-Engine-Equivalent Vehicle Concept

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Prepared for

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PREFACE

The work described in this report was performed by the
Control and Energy Conversion Division of the Jet Propulsion Laboratory.

ABSTRACT

A vehicle comparison methodology based on the Otto-Engine Equivalent (OEE) vehicle concept is described. As an illustration of this methodology, the concept is used to make projections of the fuel economy potential of passenger cars using various alternative power systems. Sensitivities of OEE vehicle results to assumptions made in the calculational procedure are discussed. Factors considered include engine torque boundary, rear axle ratio, performance criteria, engine transient response, and transmission shift logic.

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SECTION I

INTRODUCTION

The need to improve air quality has led to more stringent Federal emission standards for controlling exhaust emissions from automobiles. Early attempts by automobile manufacturers to meet these emissions standards with conventional engines resulted in significant fuel economy penalties. In addition, the impact of the oil embargo by OPEC nations has helped focus the need for energy conservation in this country, especially in the use of petroleum for transportation. This need to conserve petroleum has led to the passage of Federal fuel economy standards for automobiles.

The possibility of more stringent Federal regulation of vehicle emissions and fuel economy has stimulated interest in developing vehicles with various alternative engines. In making comparisons of vehicles with alternative engines, it is important that these comparisons be made at the vehicle level and that a consistent methodology be followed. A comparison methodology based on the concept of an Otto-Engine-Equivalent (OEE) vehicle was developed during the Automotive Power Systems Evaluation Study (APSES) done by the Jet Propulsion Laboratory and documented in Reference 1.

The Department of Energy (DOE) has sponsored a continuing JPL assessment of automotive technology which is covered in the Automotive Technology Status and Projections (ATSP) work in Reference 2. This work covers the status of automotive technology and makes vehicle-level projections of the fuel economy and emissions potential of various alternative power systems. During the ATSP work, the OEE vehicle concept was further refined based on more detailed weight propagation data. Reference 3 presents the development and application of the OEE procedure for establishing equivalent vehicles.

This report provides a detailed discussion of the OEE vehicle concept and establishes the sensitivity of the predicted fuel economy results to the torque boundary, final drive ratio, performance criteria, transient response, and transmission shift logic.

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SECTION II

CONCEPT DESCRIPTION

Central to the comparison of vehicles with alternative power systems is the OEE vehicle concept. The concept is based on using a vehicle powered by an Otto engine as the reference. An Otto engine is chosen since a large majority of the automotive fleet currently uses this type. The same general concept could be used with a different reference and the results would be equally valid. The important factor is that vehicles with the various power systems be compared on an equivalent basis.

To the individual customer, the OEE vehicle should be indistinguishable in transportation function and driving behavior from the baseline vehicle. In comparison with the baseline vehicle, this concept requires the OEE vehicle to have the same passenger and luggage space, same accessories, same drag coefficient and frontal area, same operating range, and equivalent performance.

The meaning of equivalent performance must be precisely defined. Each alternative engine is sized to provide the same vehicle performance as the baseline vehicle according to some performance criteria. The specific performance criteria must be carefully selected to represent the most important performance factors for the passenger car application. The following four performance criteria have been evaluated during this study: 0-60 mph acceleration time, 10-second acceleration distance, 40-60 mph acceleration time, and a combination of the preceding three criteria.

An OEE vehicle is first synthesized by removing the original power system and installing the alternative power system. Appropriate power system weights are used to account for the vehicle weight change due to the power system weight difference. Vehicle weight propagation effects are included in the calculation procedure to properly account for the influence of power system weight on vehicle design. For example, a larger or smaller engine may cause increases or decreases in the weight of the frame and suspension, wheels and tires, driveline, cooling system, exhaust system, fuel tank, etc. This calculation assumes that each alternative vehicle is designed with the same degree of optimization.

In addition to having different weight characteristics, the various alternative engines have different torque-speed characteristics. This results in significantly different horsepowers in the OEE vehicles. In establishing equivalent performance, the vehicle performance is calculated using a computer simulation program. Vehicle weight propagation effects result from differences in both engine power and engine weight. This procedure to establish OEE vehicles yields the appropriate engine horsepower and vehicle weight for each power system.

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Once the engine horsepower and vehicle weight for each OEE vehicle have been determined, fuel economy is calculated using a vehicle computer simulation program. This program uses steady-state engine map data to predict vehicle fuel economy for the city, highway, and composite driving cycles.

SECTION III

WEIGHT PROPAGATION METHODOLOGY

The procedure followed in establishing an OEE vehicle involves a two-step solution. First, a constant-power-ratio (CPR) solution is obtained for the alternative power system. In this calculation the ratio of installed horsepower to vehicle curb weight is maintained constant for both the baseline and alternative power systems. The second step of the solution involves varying the power ratio in the CPR solution to obtain OEE vehicle performance.

A brief description of the CPR solution will be given first. The power ratio for the baseline vehicle is given by the following equation:

$$R = \frac{P_o}{W_{vo}} \tag{1}$$

where

R = baseline power ratio

P_o = baseline horsepower

W_{vo} = baseline curb weight

The uniform charge (UC) Otto power system is now removed from the vehicle and replaced with an alternative power system having a horsepower (P_o) equal to the baseline vehicle. From a plot of total power system (TPS) weight versus horsepower, the TPS weights are obtained for both the baseline and alternative power systems at a horsepower of P_o. Weight propagation effects result in additional vehicle weight differences for the baseline and alternative power systems. At a constant level (P_o) the alternative vehicle weight is given by:

$$W_{vi} = W_{vo} + \Delta W_{vi} \tag{2}$$

and

$$\Delta W_{vi} = (W_{el} - W_{eo}) Y \tag{3}$$

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where

W_{vi} = alternative vehicle weight at power level P_o.

ΔW_{vi} = difference in vehicle weights for baseline and alternative power systems at power level P_o.

W_{eo} = baseline TPS weight at power level P_o.

W_{el} = TPS weight for alternative power system at power level P_o.

Y = vehicle weight change (lbs) due to TPS weight change (lbs). This is a function of engine type.

In order to match the power ratio of the baseline vehicle, it is necessary to make the following change in engine power for the alternative power system:

$$P_1 = P_o + \Delta P_1 \tag{4}$$

and

$$\Delta P_1 = R\Delta W_{vi} \tag{5}$$

Two weight effects result from this change in power level. There is a change in vehicle weight required due to the change in power level alone. This effect is represented by the following equation:

$$\Delta W_{P1} = \frac{\Delta P_1}{P_o} X \ W_{vi} \tag{6}$$

where

ΔW_{P1} = change in vehicle weight due to the power change ΔP_1.

X = fractional change in vehicle weight due to fractional change in power level.

There is also a change in TPS weight as a result of the change in power. This can be represented by the following equation:

$$\Delta W_{w1} = \Delta P_1 ZY \tag{7}$$

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where

ΔW_{w1} = change in vehicle weight due to the change in TPS weight.

Z = slope of the TPS weight versus horsepower. This is a function of alternative engine type and power.

The total weight increment required as a result of the power change is the sum of these two effects. The adjusted vehicle weight can then be represented by the following:

$$W_{v1} = W_{vi} + \Delta W_{v1} \tag{8}$$

and

$$\Delta W_{v1} = W_{P1} + \Delta W_{w1} \tag{9}$$

This completes the first iteration of the CPR solution. Succeeding iterations are accomplished in a similar way with the resulting weight and power of the vehicle with the alternative power system being given by the following equations:

$$W_{va} = W_{vo} + \Delta W_{vi} + \sum_{j=1}^{N} \Delta W_{vj} \tag{10}$$

and

$$P_a = P_o + \sum_{j=1}^{N} \Delta P_j \tag{11}$$

Normally, three iterations are sufficient for convergence for the systems which have been evaluated. This CPR solution represents an alternative vehicle with the same power ratio as the baseline vehicle.

Alternative vehicles with the same power ratio as the UC Otto baseline vehicle rarely give identical performance. This is due primarily to differences in their torque-speed characteristics. As a result, a second procedure is used to modify the performance level while maintaining the weight propagation methodology.

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The starting point for this procedure is the CPR solution previously outlined. The following equation comes from the CPR solution:

$$\frac{P_a}{W_{va}} = \frac{P_o}{W_{vo}} = R \qquad (12)$$

Now, define the following parameter:

$$I = \frac{R_x}{R} \qquad (13)$$

where

R_x = power ratio which will provide the desired level of performance.

R = power ratio from CPR solution.

The new power ratio can also be written as:

$$R_x = \frac{P_a + \Delta P_x}{W_{va} + \Delta W_{vx}} \qquad (14)$$

where

ΔP_x = difference in power between CPR solution and solution for OEE performance.

ΔW_{vx} = difference in vehicle weight between CPR solution and solution for OEE performance.

Combining Equations 12, 13, and 14 yields:

$$I = \frac{P_a + \Delta P_x}{W_{va} + \Delta W_{vx}} \left(\frac{W_{va}}{P_a}\right) \qquad (15)$$

Since ΔW_{vx} is small compared to W_{va}, a single iteration gives the desired convergence with the following result:

$$\Delta W_{vx} = \Delta P_x \left[\frac{X W_{va}}{P_a} + ZY \right] \tag{16}$$

The development of Equation 16 is analogous to that of Equation 9 in the CPR solution. Combining Equations 15 and 16 and solving for ΔP_x yields:

$$\Delta P_x = \frac{W_{va} (1 - I)}{I \left(\frac{X W_{va}}{P_a} + ZY \right) - \frac{W_{va}}{P_a}} \tag{17}$$

The final engine power and vehicle weight for the alternative vehicle are then given by the following:

$$W_v = W_{va} + \Delta W_{vx} \tag{18}$$

and

$$P = P_a + \Delta P_x \tag{19}$$

SECTION IV

PERFORMANCE CRITERIA

The idea of equivalent vehicle performance is an essential element of the OEE vehicle concept since it is used to establish the engine horsepower and vehicle weight of vehicles with alternative engines. Because there are many measures of vehicle performance, including standing start acceleration, passing maneuvers, distance covered in a certain time, top speed, etc., the meaning of equivalent performance must be precisely defined. With proper horsepower sizing, a vehicle with an alternative engine can be made to match any single baseline vehicle performance criteria. However, it is not normally possible to simultaneously match multiple performance criteria because of the different torque-speed characteristics of the various alternative engines. Thus, when more than one performance criterion is being used, it is necessary to employ a consistent calculational procedure that produces the closest match to the baseline performance.

This study considered four performance criteria: 0-60 mph acceleration time, 10-second acceleration distance, 40-60 mph acceleration time and a combination of the preceding three criteria. Since multiple performance criteria cannot be satisfied simultaneously, the alternative engine was sized to give the minimum root-mean-square (RMS) deviation from the baseline performance for the criteria selected. In all cases the OEE vehicle gives equivalent performance with minor deviations that the average consumer would find indistinguishable from the baseline. The sensitivity of vehicle fuel economy to the performance criteria used in establishing the OEE vehicles is examined in Section IX.

To establish the baseline performance to be used in the OEE vehicle calculations, the UC Otto baseline vehicles were first simulated using the computer performance program. Then all alternative vehicles were equipped with identical manual transmissions and final rear axle ratios, except for the single-shaft (SS) Brayton power system which, because of its unique torque characteristics, requires the use of a continuously variable transmission (CVT). The sensitivity of the fuel economy results to various vehicle parameters (rear axle ratio, shift logic, transmission gears, etc.) is discussed in Section IX.

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SECTION V

POWER SYSTEM WEIGHT CHARACTERISTICS

Weight information for each alternative power system is needed to support the OEE vehicle concept described in the previous section. Data on engine weights were obtained from domestic and foreign automobile manufacturers and government laboratories, as well as from technical publications. Representative weight characteristic curves were established for each engine type using the data base available. The weight data base is discussed in more detail in Reference 2.

Total power system weights for each engine type are given as a function of horsepower in Figure 1. The total power system includes the engine, all auxiliaries, transmission, battery, and cooling system. The curves for conventional engines — UC Otto, naturally-aspirated (NA) diesel, turbocharged (TC) diesel — and advanced conventional engines — rotary UC Otto, reciprocating stratified-charge (SC) Otto, rotary SC Otto — are representative of current weight technology. The curves for the advanced alternative engines — free-turbine (FT) Brayton, single-shaft (SS) Brayton, Stirling — represent projections at a time when those engines have been developed for passenger car use. In the OEE vehicle concept, power system weight plays an important role in determining the projected vehicle fuel economy. The importance of those differences in power system weight will be shown later.

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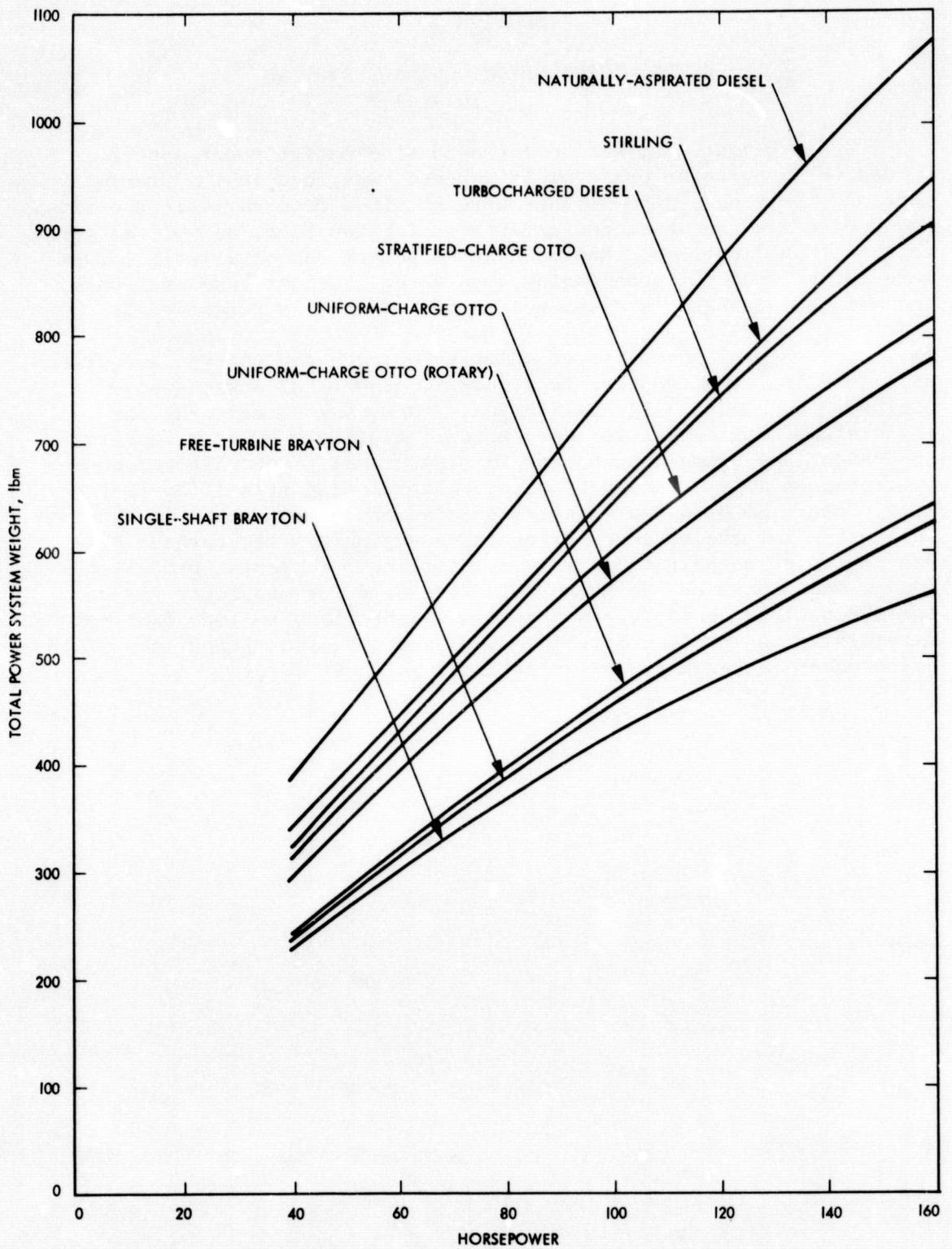
Figure 1. Total Power System Weight vs Horsepower

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SECTION VI

ENGINE TORQUE — SPEED CHARACTERISTICS

It is necessary to establish typical torque-speed characteristics for each alternative engine to support the vehicle performance calculations previously described. The shape of the engine torque-speed characteristic is important in determining the acceleration capability of a vehicle. Using the data base available from automobile manufacturers, government laboratories, and technical publications, characteristic torque curves were developed for each engine type. More detailed information about this data base is included in Reference 2.

Typical torque-speed characteristics for each engine type are given as a function of engine speed in Figure 2. In this plot, engine torque and speed are normalized with respect to the torque and speed at maximum power. For passenger car applications, engines with a high torque ratio at relatively low engine speed are advantageous since they exhibit better acceleration performance. These torque curves indicate that both Brayton and Stirling engines have better torque characteristics than those for conventional engines.

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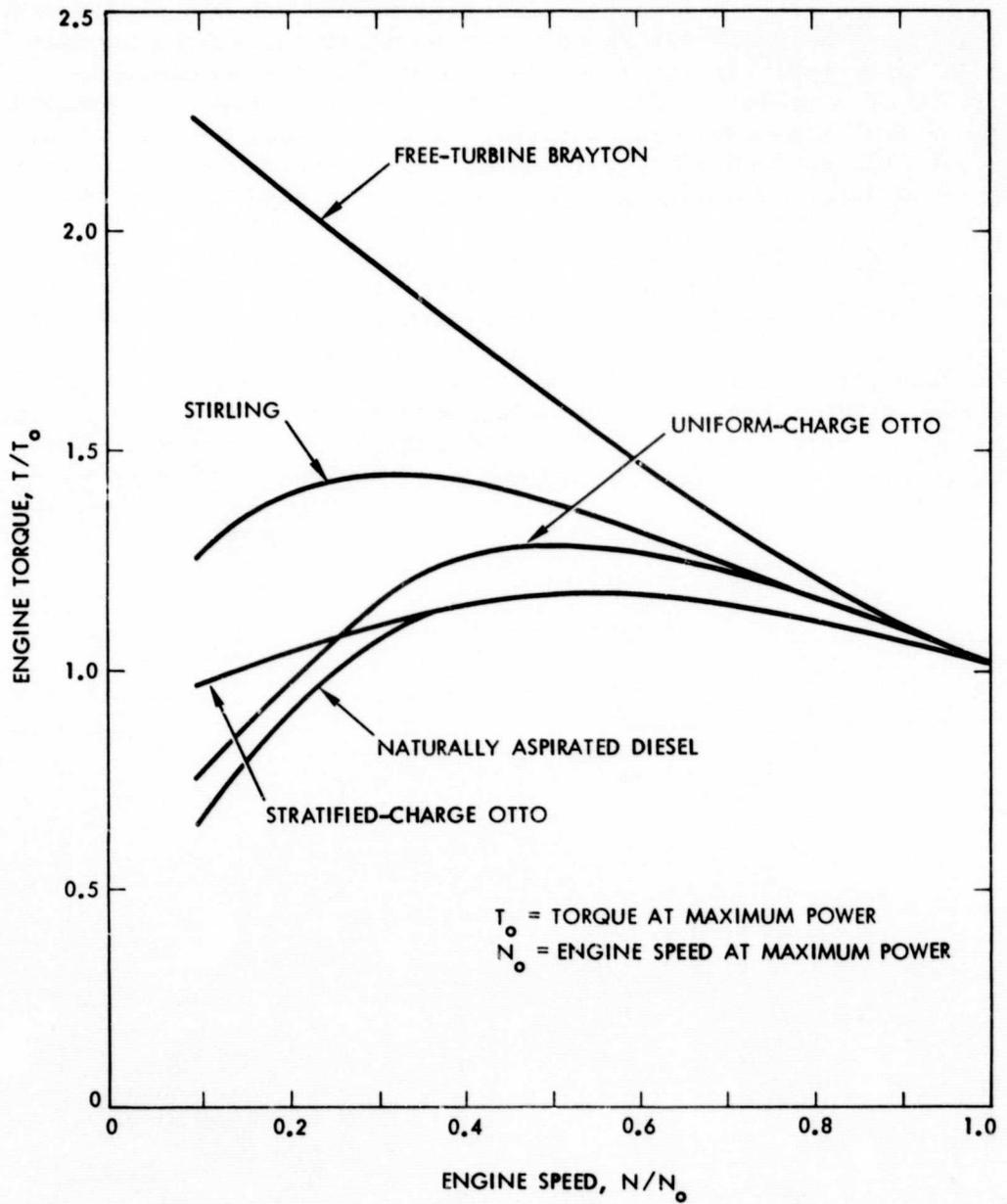
Figure 2. Typical Torque-Speed Characteristics

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SECTION VII

## OEE VEHICLE RESULTS

As an illustration of the use of the OEE vehicle concept, one set of vehicle results will be presented here. The baseline vehicle chosen for this comparison is a full-sized vehicle powered by a UC Otto engine of 120 horsepower and having a curb weight of 3200 lbs. Following the methodology previously discussed, OEE vehicles were established for each alternative power system. The power system weight and engine torque-speed characteristics representative of each engine type were used in these calculations. In establishing equivalent performance, the combined performance criteria were satisfied by minimizing the RMS deviation from baseline performances for the 0-60 mph acceleration time, the 10-second acceleration distance, and the 40-60 mph acceleration time.

The results of the OEE vehicle calculations for full-sized vehicles are given in Figure 3. A constant power/weight line is shown passing through the baseline UC Otto point to aid in making comparisons. Vehicles with conventional engines (NA diesel, TC diesel) and advanced conventional engines (rotary UC Otto, SC Otto, rotary SC Otto) all require more horsepower than vehicles with the baseline (UC Otto) engine to achieve equivalent performance. Vehicles with advanced engines (FT Brayton, SS Brayton, Stirling) show definite advantages by
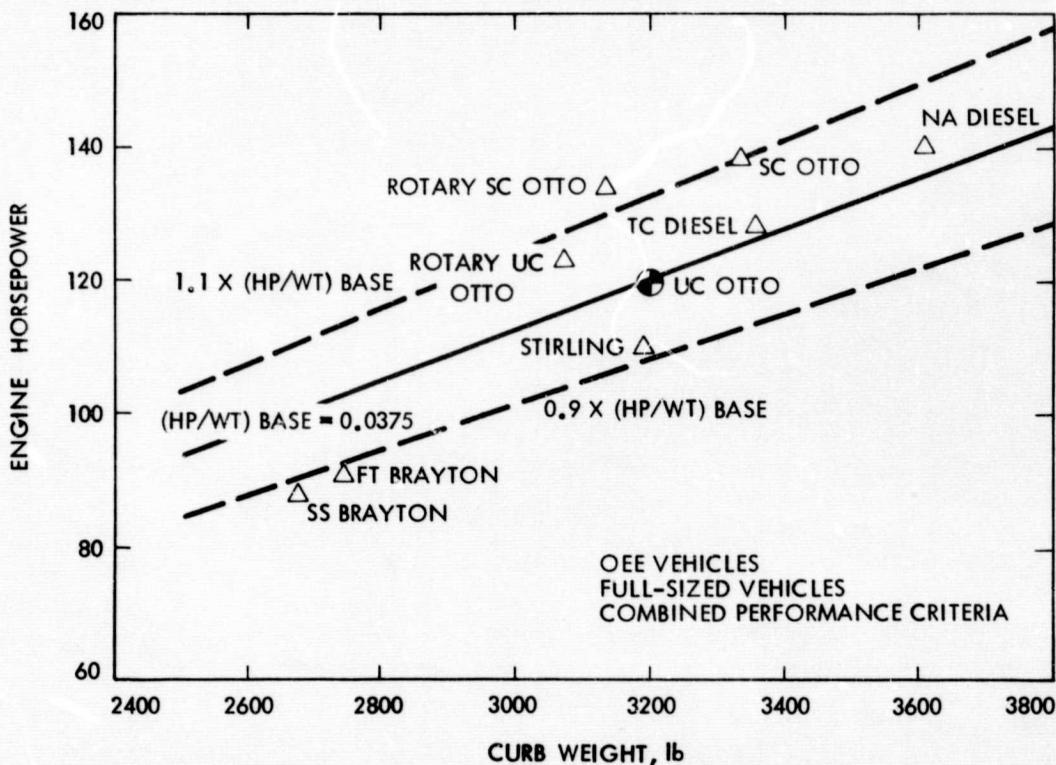
Figure 3. OEE Vehicle Results for Full-Sized Vehicles

requiring less horsepower than the baseline vehicle for equivalent performance. Vehicles with FT Brayton and SS Brayton engines show definite weight advantages, being considerably lighter than the baseline vehicle. Vehicles with rotary engines also show some weight advantage, but not as much as the Brayton vehicles. Vehicles with NA diesel, TC diesel, and SC Otto engines are heavier than the baseline vehicle. Vehicles with advanced alternative engines require a lower horsepower/weight than the baseline vehicle for equivalent performance. Additional OEE vehicle comparisons can be found in Reference 2.

VEHICLE FUEL ECONOMY

Using the OEE vehicle results from Section VII and representative fuel consumption maps (Reference 2) for each engine type, vehicle fuel economies were calculated using the vehicle computer simulation program. The fuel economy results for full-sized vehicles are shown in Figure 4 for the composite driving cycle. Fuel economy results are all expressed as gasoline equivalent mileages. The emissions constraints considered for these calculations were the Federal legislated emissions standards (0.4 g/mi HC, 3.4 g/mi CO, 1.0 g/mi NO_x). The conventional engines (UC Otto, NA diesel, TC diesel) and some of the advanced conventional engines (rotary UC Otto, SC Otto, rotary SC Otto) have one bar (cross-hatched) to represent 1978 values and another bar (open) to represent the projected 1985 values for these engines. The 1978 base for fuel economy is chosen to be the best fuel economy shown for vehicles with conventional engines.

1 UNIFORM-CHARGE OTTO	7 FREE-TURBINE BRAYTON (METAL)
2 NATURALLY-ASPIRATED DIESEL	8 FREE-TURBINE BRAYTON (CERAMIC)
3 TURBOCHARGED DIESEL	9 SINGLE-SHAFT BRAYTON (METAL)
4 ROTARY UNIFORM-CHARGE OTTO	10 SINGLE-SHAFT BRAYTON (CERAMIC)
5 STRATIFIED-CHARGE OTTO	11 STIRLING (METAL)
6 ROTARY STRATIFIED-CHARGE OTTO	12 STIRLING (CERAMIC)
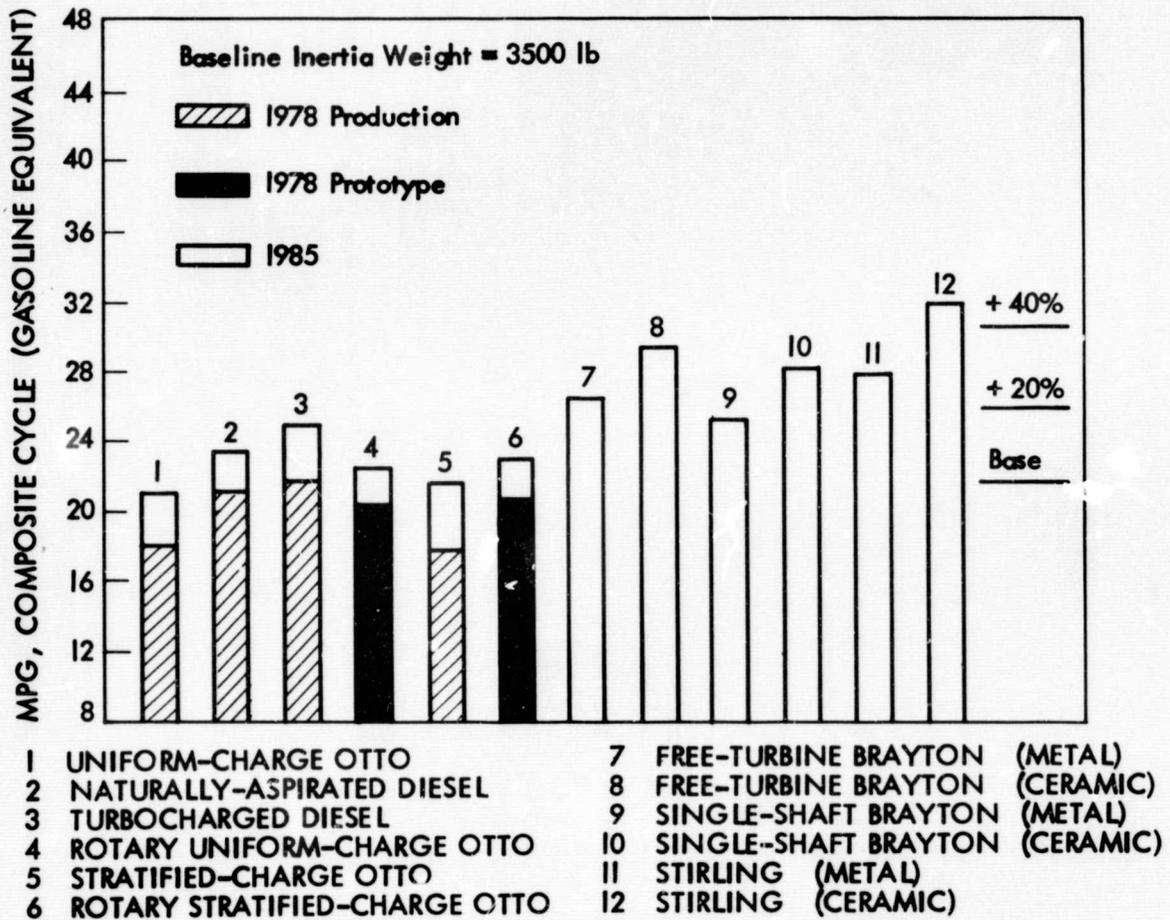
Figure 4. Composite Fuel Economy Results for
Full-Sized Vehicles

On the composite driving cycle, vehicles with advanced alternative engines shown significantly better fuel economy than vehicles with conventional engines when measured relative to the 1978 base. Relative to this base, advanced (ceramic) Brayton engines show 30% better fuel economy while advanced (ceramic) Stirling engines show a 40% fuel economy advantage. Relative to the projected 1985 base, these fuel economy advantages of vehicles with Brayton and Stirling engines are reduced by about 10%. The plot also shows that vehicles with SC Otto and rotary engines are projected to have fuel economies between those for the UC Otto and TC diesel engines. Additional fuel economy comparisons can be found in Reference 2.

SECTION IX

SENSITIVITY STUDIES

Studies have been made to determine the sensitivity of the OEE vehicle results and the fuel economy projections to the assumptions made in the calculational procedure. Factors considered include engine torque boundary, rear axle ratio, performance criteria, engine transient response, and transmission shift logic.

A. ENGINE TORQUE BOUNDARY EFFECTS

In sizing an alternative engine to provide equivalent vehicle performance, the shape of the engine torque-speed characteristic has a significant effect. To determine the sensitivity of the OEE vehicle results to changes in the torque boundary, the small SC Otto and TC diesel vehicle cases were rerun using the torque boundary for the UC Otto engine. Equivalent performance was established using the combined performance criteria. The horsepower and vehicle weight results of these sensitivity runs are given in Table 1. The power ratio parameter (I) is defined as the power ratio required for OEE performance divided by the power ratio for the baseline UC Otto vehicle. Note the decreases in power ratio which occur for both the SC Otto and TC diesel when their torque boundaries are replaced with the torque boundary for the UC Otto engine. This indicates that both engines suffer in the OEE vehicle concept from having less desirable torque-speed characteristics and higher engine weights than the UC Otto engine.

The effect of torque boundary on vehicle fuel economy is shown in Table 2 for the same vehicles given in Table 1. The torque boundary has a significant (about 10%) effect on the projected fuel economy of the SC Otto engine.

Table 1. Torque Boundary Sensitivity Results for Horsepower and Vehicle Weight

Engine Type	Torque Boundary	Power Ratio Parameter (I)	Engine Horsepower	Vehicle Curb Weight (lbs)
UC Otto	UC Otto	1.00	60	1750
SC Otto	SC Otto	1.11	72	1888
	UC Otto	1.00	62	1805
TC Diesel	TC Diesel	1.04	67	1878
	UC Otto	1.00	65	1850

Table 2. Torque Boundary Sensitivity Results for Fuel Economy

Engine Type	Torque Boundary	MPG-City		MPG-Highway		MPG-Composite	
		Diesel	Gas Eqvt.	Diesel	Gas Eqvt.	Diesel	Gas Eqvt.
UC Otto	UC Otto		29.5		41.1		33.8
SC Otto	SC Otto		27.0		40.5		31.8
	UC Otto		30.3		42.0		34.7
TC Diesel	TC Diesel	38.0	33.5	56.4	49.8	44.6	39.4
	UC Otto	38.3	33.8	56.5	49.9	44.8	39.5

B. REAR AXLE RATIO EFFECTS

The effect of the final drive ratio on vehicle fuel economy was examined by varying the rear axle ratio of the small SC Otto vehicle case by +15%. The results of these calculations are given in Table 3. The rear axle ratio is seen to have little effect on the predicted composite fuel economy when the engines are sized to provide OEE vehicle performance. Equivalent performance was again based on satisfying the combined performance criteria.

Table 3. Effect of Rear Axle Ratio on Fuel Economy of OEE Vehicles

Engine Type	Rear Axle Ratio	Power Ratio Parameter (I)	Engine Horsepower	Vehicle Curb Weight (lbs)	Gasoline (Eqvt.)
UC Otto	3.7	1.00	60	1750	33.8
SC Otto	3.15 (-15%)	1.135	74	1980	32.4
SC Otto	3.7 (std.)	1.11	72	1888	31.8
SC Otto	4.26 (+15%)	1.06	67	1850	31.7

C.   PERFORMANCE CRITERIA EFFECTS

As previously mentioned, the four performance criteria considered in this study were the 0-60 mph acceleration time, the 10-second acceleration distance, the 40-60 mph acceleration time, and a combination of the preceding three criteria. The choice of performance criteria directly affects the horsepower sizing of alternative engines for equivalent vehicle performance. The OEE vehicle horsepower and weight results for the FT Brayton are shown in Table 4 for the four performance criteria.

Note that the FT Brayton requires the lowest horsepower to meet the second performance criteria (10-second distance) and the highest horsepower to meet the third performance criteria (40-60 mph time). In all cases, the horsepower for the FT Brayton is less than that of the UC Otto baseline. This is somewhat explained by the shape of the torque-speed characteristics of the engines. The FT Brayton gains a significant advantage in the second performance criteria by having much better low-end torque than the UC Otto baseline. This advantage is somewhat reduced in meeting the third performance criteria since the torque curve for the FT Brayton approaches that of the UC Otto at the high end.

Since it is evident that using only one criterion can discriminate against some engines, it is essential that performance criteria be carefully chosen to yield a fair comparison of vehicles with alternative engines.

D.   ENGINE TRANSIENT RESPONSE EFFECTS

Engine transient response can have an effect on the sizing of alternative engines for equivalent vehicle performance. If acceleration time is measured relative to the time when the accelerator pedal

Table 4. Effect of Performance Criteria on OEE-Vehicle Horsepower and Weight

Engine Type	Performance Criteria	Power Ratio Parameter (I)	Engine Horsepower	Vehicle Curb Weight (lbs)
UC Otto	all	1.0	60	1750
FT Brayton	0-60 mph time	0.918	48.8	1546
FT Brayton	10-sec distance	0.874	45.5	1517
FT Brayton	40-60 mph time	0.924	49.2	1550
FT Brayton	RMS combined	0.920	48.9	1548

is depressed, any delayed response of the engines would be included in the acceleration time. To examine this effect, calculations were made for the SS Brayton engine using response delay times of 1.0 and 1.5 seconds and matching the performance of the UC Otto baseline. The results of these calculations are given in Table 5 for both small and full-sized vehicles.

When engine transient response is neglected, the vehicle power/weight required for the SS Brayton engine is much less than that for the UC Otto baseline. This advantage is reduced significantly when the SS Brayton engine is assumed to have a response delay time of 1.5 seconds. Even when the transient response is considered, the engine horsepower and vehicle weight for the SS Brayton are substantially less than those for the UC Otto baseline. This is due to the significant difference in projected power system weights for the two engines as shown in Figure 1.

E. TRANSMISSION SHIFT LOGIC EFFECTS

In performing vehicle simulation calculations for meeting performance criteria or simulating city or highway driving, the method

Table 5. Effect of Transient Response on Sizing of SS Brayton for OEE Performance (0-60 mph Acceleration Time)

Vehicle Size	Engine Type	Response Delay Time (sec)	Power Ratio Parameter (I)	Engine Horsepower	Vehicle Curb Wt (lbs)
Small	UC Otto	0.0	1.0	60	1750
Small	SS Brayton	0.0	0.86	43	1454
Small	SS Brayton	1.0	0.91	47	1490
Small	SS Brayton	1.5	0.94	49	1511
Full-Sized	UC Otto	0.0	1.0	120	3200
Full-Sized	SS Brayton	0.0	0.86	86	2658
Full-Sized	SS Brayton	1.0	0.93	95	2712
Full-Sized	SS Brayton	1.5	0.98	100	2743

used in determining when to shift gears in the transmission can be a significant factor affecting the results. The sensitivity of the results to changes in shifting logic was examined by comparing cases using two basic shifting methods.

The first method is based on shifting gears at specified vehicle velocities. The base velocity shift points were taken from the values which are used in the JPL chassis dynamometer facility. These shift points are somewhat arbitrary since they are not specified in the Federal Test Procedure (FTP). Results were obtained for two other cases which used shift points 20% greater or less than the base shift points. Table 6 shows the velocity shift points used in this evaluation.

The second method is based on shifting gears in a way that is dependent on the vehicle power requirements. The computer program logic selects the highest possible gear which permits the engine to meet the necessary power requirements of the next velocity point on the driving cycle. This method is more easily understood by referring to Figure 5. Two lines, the Downshift Limit Line and the Upshift Limit Line, divide the graph into three regions (A, B, and C). Five constants (C_{vo}, C_{v1}, C_{v2}, V_o, V_1) are needed to describe these two lines. Values of the five constants for the three cases studies are given in Table 7.

Table 6. Velocity Shift Points (MPH)

Shift Points	Gear Shift			
	1-2	2-3	3-4	4-5
Base	17.0	28.0	36.0	41.0
+20 %	20.4	33.6	43.2	49.2
-20%	13.6	22.4	28.8	32.8

Table 7. Power Shift Constants

| Shift Points	Line Constants				
	V_o	V_1	C_{vo}	C_{v1}	C_{v2}
Base	18.0	73.0	12.86	45.63	83.33
+20 %	16.7	80.3	11.93	50.19	83.33
-20 %	19.8	65.7	14.14	41.06	83.33
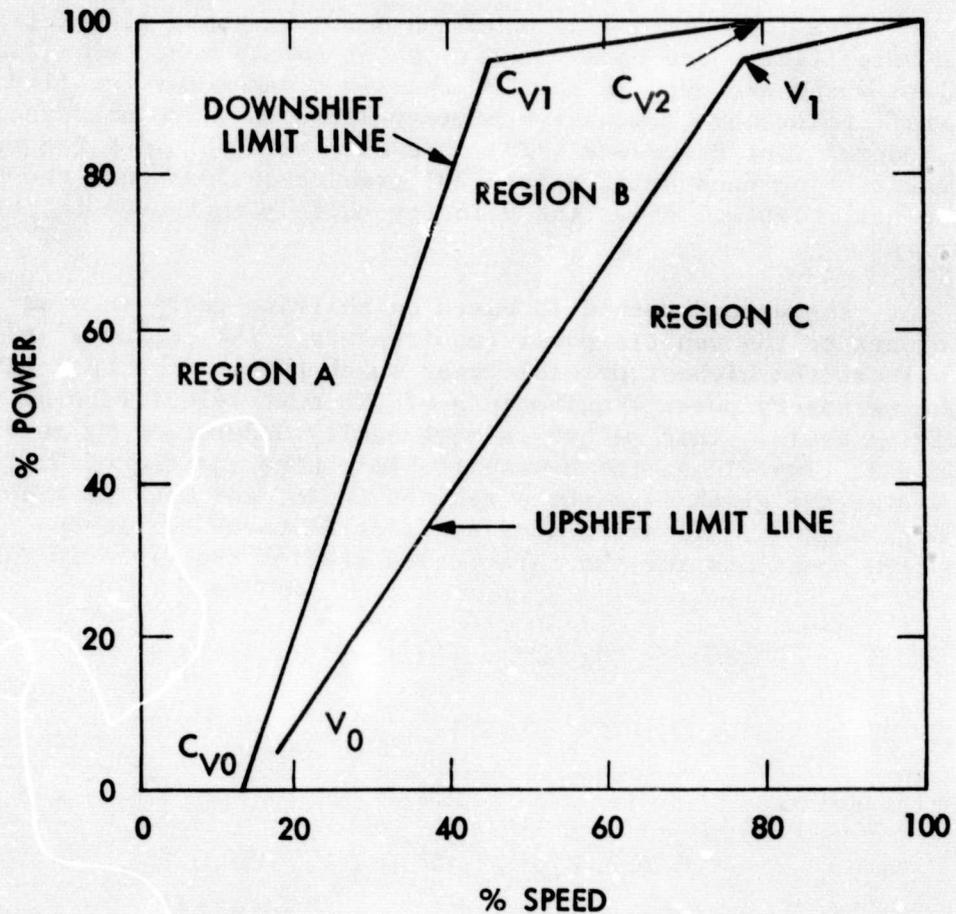
Figure 5. Power Shift Logic

The computer program logic makes the following checks to determine whether or not it should shift using the proper shift method.

(1)   If the operating point falls within Region C, upshift.

(2)   If the operating point falls within Region A, downshift.

(3)   If the operating point falls within Region B, don't shift.

The effect of the shift logic on vehicle performance was examined by evaluating the full-sized UC Otto vehicle (120 hp) for three performance criteria.   The results of this evaluation are given in Table 8.

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Table 8. Effect of Shift Logic on Vehicle Performance

Performance Criteria	Shift Logic		
	Power Shift	Velocity Shift	Δ (%)
0-60 mph	11.63 sec	17.85 sec	53.5
10-sec distance	489.4 ft	423.8 ft	-13.4
40-60 mph	5.85 sec	10.26 sec	75.4

A time history graph for these full-throttle acceleration calculations is given in Figure 6. This plot gives engine rpm and transmission gear as a function of time. It is evident that the velocity shifting method is giving poorer performance because it is upshifting at the specified shift points and is therefore not utilizing the full performance potential of the engine. The power shift method was used in all performance matching procedures.

The second area examined was the influence of the shifting logic on the fuel economy over the driving cycles. Figure 7 shows the variation in fuel economy for a small SC Otto vehicle using the six different shifting strategies which have been examined. Notice the pronounced difference between the results for the two basic methods with the velocity shift case yielding a 26% lower composite mileage than the power shift case. It is also evident that the power shift method is relatively uninfluenced by the change in constants, whereas the composite mileage for the velocity shift method varies by as much as 24%, depending on the shift points used.

To further aid in understanding the large variation in mileage, a section of the urban driving cycle (from 1372 to 1877 seconds) is plotted in Figure 8. From this plot, it is clear that the power shift method selects as high a gear as possible (consistent with the required power) that results in a lower engine speed and higher mileage.
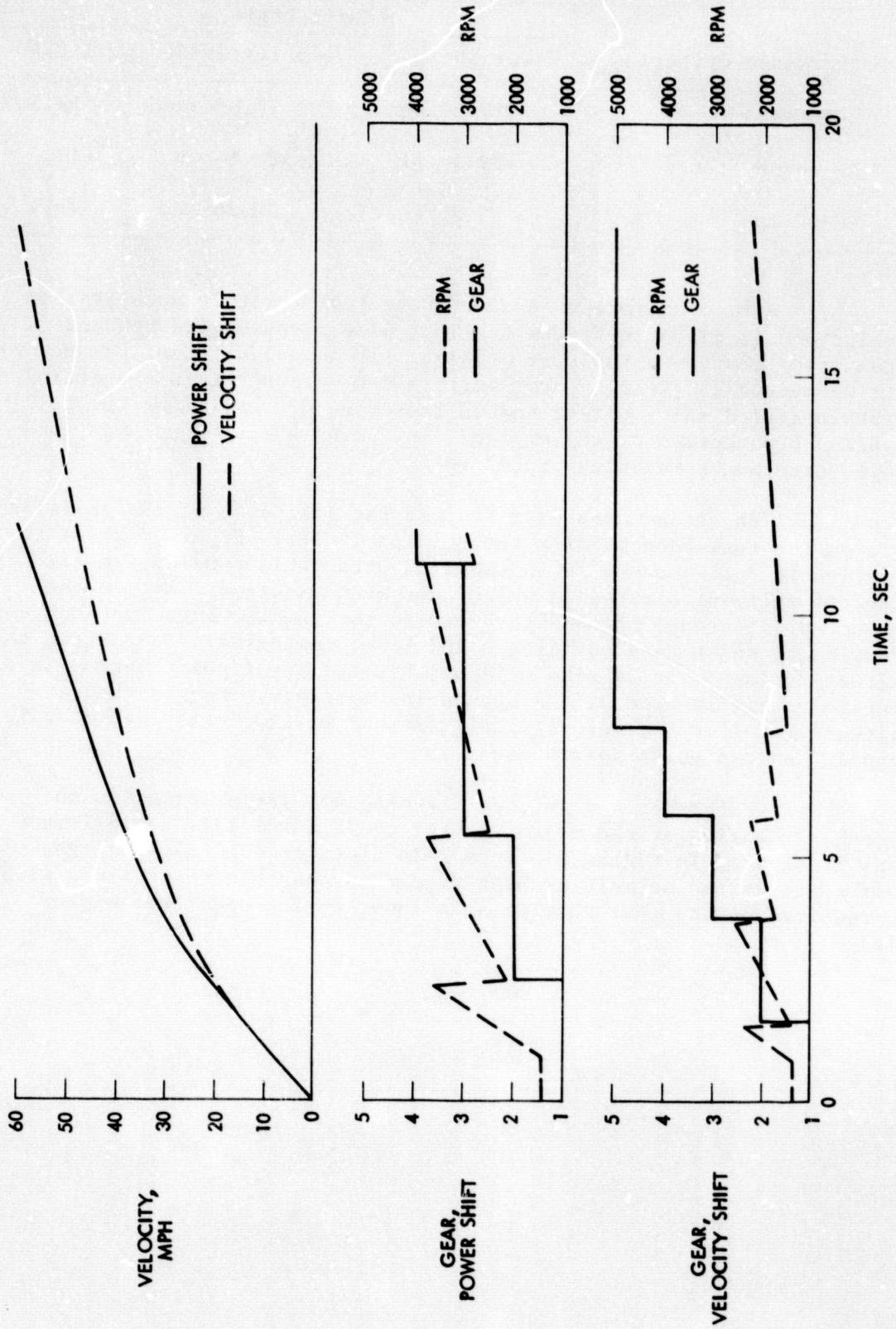
Figure 6. Transmission Shifting Strategies for Full-Sized UC Otto Vehicle for Wide-Open Throttle Acceleration Maneuver

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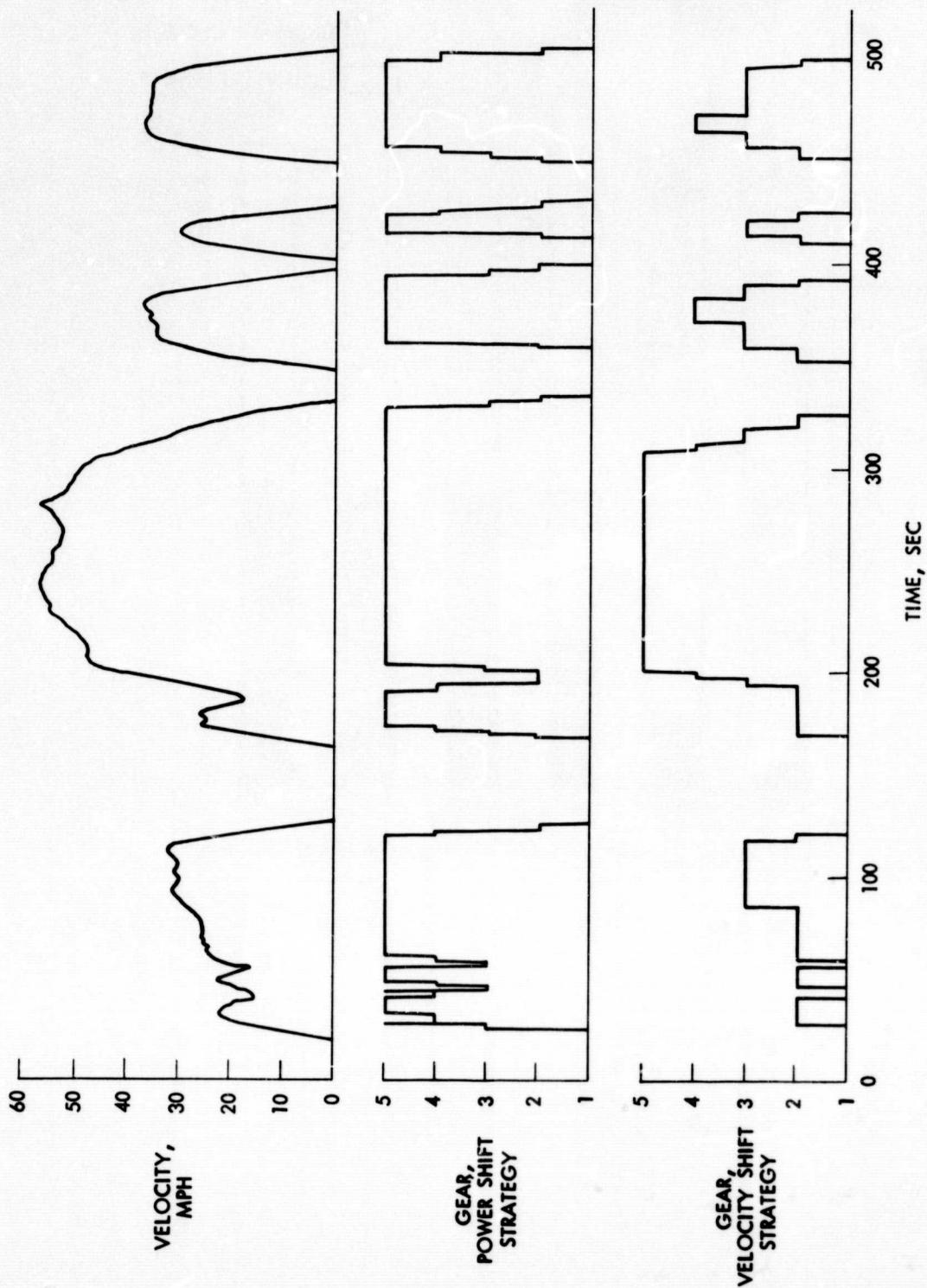
Figure 7. Transmission Shifting Strategies for Small SC Otto Vehicle
Over Portion of Urban Driving Cycle (1373-1877 sec)

29
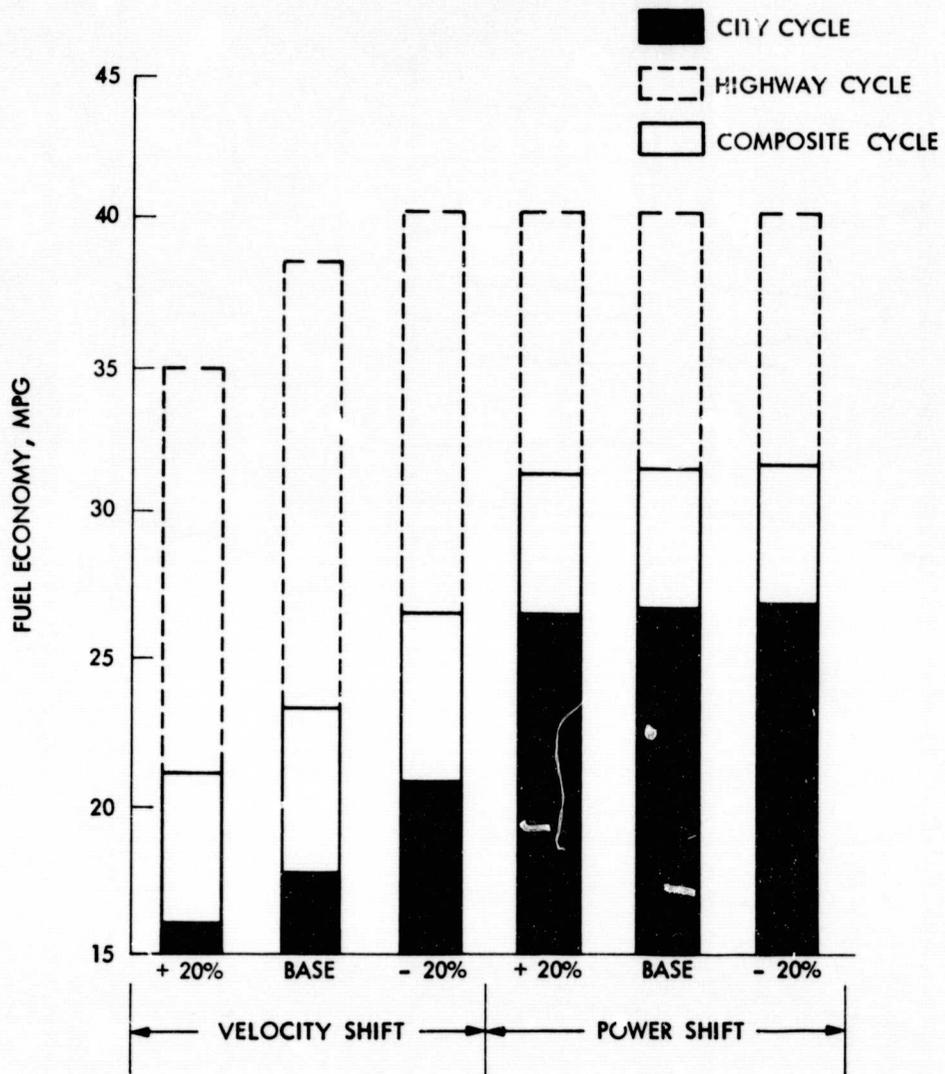
Figure 8. Sensitivity of Fuel Economy to Changes in Transmission
Shift Logic for Small SC Otto Vehicle

SECTION X

SUMMARY

This report has outlined a methodology that can be used in making comparisons of alternative power systems on an equivalent basis. Results of the sensitivity studies clearly indicate the need to use representative data for the engine and transmission characteristics of the power systems being compared. Further evaluation of engine transient response and transmission shift logic seems justified based on the sensitivity studies.

Although the Otto-Engine-Equivalent vehicle concept has been applied to passenger cars in this report, the general method is valid for other applications which have different constraints.

REFERENCES

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