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Free-Piston Stirling Hydraulic Engine and Drive System for Automobiles

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ABS: The calculated fuel economy for an automotive free piston Stirling hydraulic engine and drive system using a pneumatic accumulator with the fuel economy of both a conventional 1980 spark ignition engine in an X body class vehicle and the estimated fuel economy of a 1984 spark ignition vehicle system are compared. The results show that the free piston Stirling hydraulic system with a two speed transmission has a combined fuel economy nearly twice that of the 1980 spark ignition engine - 21.5 versus 10.9 km/liter (50.7 versus 25.6 mpg) under comparable conditions. The fuel economy improvement over the 1984 spark ignition engine was 81 percent. The fuel economy sensitivity of the Stirling hydraulic system to

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system weight, number of transmission shifts, accumulator pressure ratio and maximum pressure, auxiliary power requirements, braking energy recovery, and varying vehicle performance requirements are considered. An important finding is that a multispeed transmission is not required. The penalty for a single speed versus a two speed transmission is about a 12 percent drop in combined fuel economy to 19.0 km/liter (44.7 mpg). This is still a 60 percent improvement in combined fuel economy over the projected 1984 spark ignition vehicle.

FREE-PISTON STIRLING HYDRAULIC ENGINE AND DRIVE SYSTEM FOR AUTOMOBILES

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SUMMARY

This analytical study compares the computer-simulated fuel economy for a free-piston Stirling hydraulic engine and drive system using a pneumatic accumulator with the fuel economy of a conventional 1980 spark-ignition engine (three speed, automatic transmission) both in the same X-body class vehicle. The free-piston Stirling fuel economy was also compared with the estimated fuel economy of a 1984 spark-ignition vehicle system. The hot-start fuel economy comparisons were made for urban, highway, and combined Federal driving cycles. The free-piston Stirling hydraulic engine and drive system consists of a free-piston Stirling engine operating in an on-off mode and generating hydraulic power directly by means of an integrated hydraulic converter device, a hydraulic accumulator which serves as an energy buffer, a variable-displacement hydraulic motor-pump unit, a hydraulic sump, and an optimal two-speed transmission. The hydraulic accumulator and motor-pump unit also allow the efficient recovery and utilization of the vehicle braking energy.

A baseline system was selected for this study and sensitivities to various system parameters and design variations were investigated. A 272-kg (600-lb) hydraulic system weight penalty was calculated for the baseline system. This was added to the 1361-kg (3000-lb) inertia weight of an X-body vehicle to yield a 1633-kg (3600-lb) inertia weight vehicle. For this 1633-kg (3600-lb) vehicle, the engine was sized at 39 kW (52 hp) with an average efficiency (engine hydraulic output to fuel input) of 43 percent. The engine size was based on maintaining a speed of 88.5 km/hr (55 mph), while climbing a 5-percent grade with no power being withdrawn from the accumulator. The size of the variable-displacement motor-pump was determined by the most severe of the following vehicle performance (acceleration) requirements:

(1) 0 to 96.5 km/hr (0 to 60 mph) in 15 sec
(2) Standstill to 30.5 m (100 ft) in 4.5 sec
(3) 80.5 to 113 km/hr (50 to 70 mph) in 10.5 sec

The most severe of these was traversing 30.5 m (100 ft) in 4.5 sec, which required an 81-cm^3/rev (4.9-in^3/rev) displacement motor-pump for the baseline case. The accumulator was sized to provide the kinetic energy necessary to accelerate the vehicle from standstill to 113 km/hr (70 mph) - 0.068 m^3 (2.41 ft^3) for the 1633-kg (3600-lb) baseline system, assuming an ideal gas.

The results of the study show that the baseline free-piston Stirling hydraulic engine and drive system (with one transmission shift), even though several hundred pounds heavier than the conventional spark-ignition X-body vehicle system, will have a combined fuel economy about 81 percent better than that for a 1984 spark-ignition engine in the X-body class vehicle while giving equal or better performance (21.5 km/liter versus 11.9 km/liter; 50.7 mpg versus 28.0 mpg). This combined fuel economy is nearly twice that of the 1980 spark-ignition engine in an X-body vehicle (21.7 km/liter versus 10.9 km/liter; 50.7 mpg versus 25.6 mpg).

As stated above, the baseline case included a 272-kg (600-lb) weight penalty for the hydraulic system. This was based on use of a spherical steel accumulator and sump but could be reduced to about 193 kg (425 lb) by using a material with a higher stress/density ratio such as an aluminum fiberglass composite or titanium.

The following significant sensitivity results were obtained in this study:

(1) The combined-cycle fuel economy penalty for each 45.4 kg (100 lb) of additional system weight was less than 1.5 percent.

(2) Eliminating the single gearshift utilized in the baseline case reduced the combined driving-cycle fuel economy by only 12 percent.

(3) Decreasing the maximum hydraulic pressure from 34.5 to 27.6 MPa (5000 to 4000 psi) gained about 2 percent in fuel economy, yielded a small 12.2-kg (27-lb) increase in hydraulic system weight, and increased accumulator and sump volumes by 25 percent - 0.0683 to 0.0854 m^3 (2.41 to 3.02 ft^3).

(4) A 23-percent increase in hydraulic motor-pump size reduced fuel economy only 2.4 percent (0.51 km/liter; 1.2 mpg) but decreased acceleration times by 10, 21, and 25 percent, respectively, to standstill to 30.5 m (100 ft) in 4.06 sec, 0 to 96.5 km/hr (60 mph) in 9.6 sec, and 80.5 to 113 km/hr (50 to 70 mph) in 6.4 sec.

(5) Radiation, conduction, and exhaust gas losses from the hot end of the engine during engine-off (standby) conditions had a modest effect on fuel economy. A 50-percent reduction of these losses (from 1.56 to 0.78 kW; 2.09 to 1.05 hp) would increase fuel economy for the baseline case from 21.5 to 22.3 km/liter (50.7 to 52.4 mpg) - about a 3-percent improvement. A 50-percent increase in these losses would reduce the fuel economy to about 20.8 km/liter (49.0 mpg) - a 3-percent decrease.

Even though this study was analytical in nature, the critical variable-displacement motor-pump performance characteristics were based on actual test data of a prototype unit tested with special emphasis on the critical low-speed, low-power operating region.

These results naturally raise a question: Would more conventional engines yield similar results if used with this same hydraulic drive system? This study has not included any detailed evaluation of the hydraulic drive system with other engines. However, it was possible to make some preliminary estimates on the basis of expected engine and hydraulic pump efficiencies. On this basis the following preliminary comparison was made:

Engine drive system	Fuel economy		Estimated improvement over standard spark-ignition engine-powered vehicle with automatic transmission
	km/liter	mpg	
Free-piston Stirling hydraulic engine and drive	21.5	50.7	81
Kinematic Stirling engine and hydraulic drive	20.0	47.0	68
Diesel engine and hydraulic drive	16.2	38.1	36
Spark-ignition engine and hydraulic drive	14.3	33.6	20
1984 spark-ignition-engine-powered vehicle with automatic transmission	11.9	28.0	Standard

Only the kinematic Stirling engine offers fuel economy reasonably close to that of the free-piston Stirling hydraulic engine. However, other factors such as cost, complexity, life, and reliability, in addition to the fuel economy differential, are expected to give the free-piston Stirling hydraulic engine and drive system a substantial advantage over the kinematic Stirling engine and hydraulic drive. It is believed that the results of this study clearly warrant further work on the free-piston Stirling hydraulic engine and drive system for the automobile application.

INTRODUCTION

In recent years, the NASA Lewis Research Center has conducted a number of Stirling engine investigations for a variety of potential applications. These activities have included the Department of Energy's Automotive Stirling Engine Development Project being conducted by the Office of Vehicle and Engine R&D, Technology Development and Analysis Division, and the NASA Stirling Engine Technology effort sponsored by the NASA Conservation and Fossil Energy Systems Branch of the Energy Systems Division. This NASA-funded effort was aimed at broadening the general Stirling engine technology base at Lewis and assessing its applicability to a variety of applications. This study, "Free-Piston Stirling Hydraulic Engine and Drive System for Automobiles," was carried out as part of the NASA-funded effort. It has relied heavily on capabilities and information developed at Lewis under the various DOE and NASA-funded activities. A significant part of the NASA-funded effort has been directed toward the free-piston Stirling engine. The free-piston Stirling is in an even earlier state of development than the kinematic Stirling currently being developed under the automotive program. However, the free-piston Stirling engine inherently provides a high-payoff - high-risk type of advanced heat engine offering the potential for high efficiency, simplicity, and long life.

Other studies (such as ref. 1) have shown the potential fuel economy benefits of hydraulic drive systems for automobiles. However, a key drawback has always been the added complexity of these systems. The free-piston Stirling hydraulic engine and drive system addressed in this study offers a significant degree of simplification in the drive system. It avoids some fundamental problems inherent in kinematic Stirling engine development such as seal life and control complexity; it also offers the potential for even higher efficiency. For these reasons, this study was undertaken to more carefully assess the fuel economy potential of such a free-piston Stirling hydraulic engine and drive system for the automotive application.

Two NASA-funded contractual efforts were carried out to provide needed input for this study. These were (1) a contract to design, fabricate, and test a hydraulic accumulator under adiabatic and isothermal operating conditions (ref. 2), and (2) an investigation to obtain hydraulic motor-pump experimental performance data and to provide appropriate information for scaling weight and performance to the specific system requirements (ref. 3). In addition, the results of two other NASA-funded efforts and a DOE-funded contractual effort were of significant use in this study. These were (1) a study to compare free-piston Stirling engine performance with kinematic Stirling engine performance and to assess the scalability limits of both types of engine configurations (ref. 4), (2) an in-house test and characterization of a 1-kW-output free-piston Stirling engine augmented by a contractual effort to design a hydraulic output modification for the 1-kW (1.34-hp) engine (refs. 5 and 6), and (3) a Department of Energy (DOE)-funded conceptual design of a hydraulic output for a 15-kW (20.1-hp) solar thermal free-piston Stirling engine - the hydraulic output replacing a linear alternator (ref. 7).

This report assesses the potential of the free-piston Stirling hydraulic engine and drive system using a hydraulic accumulator energy buffer to power automobiles or light trucks. The report presents the effects of various system parameters on optimizing the fuel economy. Computer simulations of the urban and highway Federal driving cycle were run and miles-per-gallon results for different combinations of system variables are presented. For comparative purposes, the General Motors 1361-kg (3000-lb) inertia weight X-body car with three-speed automatic transmission was considered as the standard. Also, the free-piston Stirling hydraulic system fuel economy was compared with fuel economy projections for a 1984 spark-ignition engine in an X-body vehicle.

SYSTEM DESCRIPTION

The free-piston Stirling hydraulic engine and drive system is shown schematically in figure 1. The free-piston engine generates hydraulic output directly by means of an integral hydraulic converter to supply hydraulic fluid to the accumulator, thereby compressing the gas in the accumulator. The accumulator serves as an energy buffer, isolating the engine from the vehicle drive system. High-pressure fluid from the accumulator is supplied to the variable-displacement motor-pump on demand by expansion of the gas in the accumulator. The hydraulic motor discharges the hydraulic fluid to the sump, where it is again available to the inlet of the converter. Motor output is transmitted to the vehicle drive shaft through a gearbox, including an optional multispeed transmission. For regenerative braking of the vehicle, the motor-pump is driven as a pump by the vehicle. The pump output is used to recharge the accumulator.

SYSTEM REQUIREMENTS

For this study, it was decided to assess the engine-drive system in a conventional family car with full normal performance capabilities. The following performance criteria were selected:

- Standstill to 30.5 m (100 ft) in 4.5 sec
- 0 to 96.5 km/hr (60 mph) in 15 sec
- 80.5 to 113 km/hr (50 to 70 mph) in 10 sec
- Continuous 88.5 km/hr (55 mph) up a 5-percent grade
- Maximum speed of 113 km/hr (70 mph) on a level road

It was further decided that the system should be designed such that these requirements could always be met, regardless of the immediately preceding operating history of the vehicle. This requirement was included to avoid unsafe driving situations such as attempting a high-speed pass maneuver without the normal expected acceleration capability.

A 1980 X-body vehicle was used as the standard vehicle for the study. For the calculations, the X-body vehicle was assumed to have an inertia weight of 1361 kg (3000 lb) including two passengers plus a full tank of fuel. To meet the requirements, a 272-kg (600-lb) weight penalty was calculated for the baseline hydraulic drive vehicle system yielding 1633-kg (3600-lb) inertia weight.

SYSTEM DESIGN APPROACH

To meet the system requirements, a number of design approaches are possible. One approach is to size the engine so that, even with accumulator pressure at the minimum value, the engine could supply the power to meet all of the performance criteria. This approach allows a trade-off of accumulator size against regenerative braking energy recovery but requires a full-size engine. The accumulator would normally be depleted by the time the vehicle reached 56.3 to 64.4 km/hr (35 to 40 mph). Once depleted, the engine must supply all of the power in a load-following operating mode over a major portion of its operating range.

The approach chosen here was to size the accumulator system to store energy equal to the kinetic energy of the vehicle at maximum speed and to size the engine to provide all constant-speed power requirements (including continuous 88.5 km/hr (55 mph) up a 5-percent grade and 113 km/hr (70 mph) on a level road). In this approach, the engine size can be reduced significantly and it operates in an on-off mode only. To assure that full vehicle acceleration capability would always be available, regardless of the preceding vehicle operating history, the accumulator pressure was scheduled with vehicle speed (as shown in fig. 2 for the baseline case). This assures that there is always sufficient stored energy to provide the kinetic energy necessary to accelerate the vehicle to maximum speed (113 km/hr (70 mph) on a level road). The hydraulic motor was then sized to meet the acceleration rate criteria with the scheduled accumulator pressure. To avoid continuous on-off cycling of the engine, a deadband was added to the accumulator schedule so that the engine would cycle off at the top limit of the deadband and cycle on at the bottom limit. For example, for the baseline case shown in figure 2, at a speed of 80 km/hr (50 mph) the engine will turn on when the pressure falls below approximately 20 MPa (2900 psi) and will shut off when the pressure reaches approximately 20.7 MPa (3000 psi). In view of the on-off engine operation, auxiliary power for both the vehicle and engine are provided by means of a separate hydraulic motor operating off the accumulator system. This assures the continuous availability of power for the vehicle accessories such as power brakes and air conditioning. It also provides for independent operation of the engine burner system, which will be required to maintain heater head temperatures at design values while the vehicle is operating but the engine is off. This feature enables the engine to operate on demand, with no degradation of performance.

ENGINE

The engine size was determined, as stated in the preceding section, by road-load requirements, that is, rolling resistance, aerodynamic load, and road grade. The variation of road-load power at the wheels as a function of speed for the 1633-kg (3600-lb) baseline vehicle is shown in figure 3. The engine was sized to maintain the vehicle at least at 113 km/hr (70 mph) on a flat road and at least at 88.5 km/hr (55 mph) up a 5-percent grade. As shown in the figure, 88.5 km/hr (55 mph) up a 5-percent grade was the controlling criterion which then allowed a top speed in excess of 129 km/hr (80 mph).

The required wheel power at 88.5 km/hr (55 mph) on a 5-percent grade was 31 kW (41 hp). The required engine power, based on an overall drive system efficiency (from engine hydraulic output to the wheels) of 79 percent, was 39 kW (52 hp). The 79-percent overall system efficiency consisted of

	Efficiency, percent
Variable-displacement hydraulic motor-pump	88.0
Gearbox	98.0
Differential	96.6
Tires (combined urban and highway)	95.2

At present, there are no known operating free-piston Stirling engines above about 3-kW (4-hp) output power, nor have any design studies been prepared which would apply directly to the requirements of this study. Therefore, it was decided to estimate engine performance by extrapolating from existing results for large (30 to 60 kW; 40.2 to 80.5 hp) kinematic engines and factoring in the results of design studies which did address hydraulic output for free-piston Stirling engines.

It was felt that the best basis for this was to extrapolate indicated efficiencies from results of the DOE-NASA Automotive Stirling Engine Program. The resulting method was to calculate the indicated engine efficiency as equal to 75 percent of Carnot efficiency plus one-half efficiency point for optimizing the engine without any requirement for higher power, high-speed operation. Therefore, using a heater temperature of 762° C (1404° F) (assuming a continuous 15-MPa (2175-psi) charge pressure for 3500 hr of life) and a cooler temperature of 50° C (122° F) with 75 percent of Carnot yielded an indicated efficiency of 51.6 percent. Adding one-half point for optimization gave an indicated free-piston Stirling efficiency of 52.1 percent. The rationale for this procedure is based on references 4, 8, and 9. Reference 8 presents a comparison between the computer-code-predicted and actual peak efficiencies of a Mod I automotive Stirling engine (37.7 percent predicted, 37.4 percent actual). As a result, a high degree of confidence exists in the computer-code-predicted efficiencies. Keep in mind that the Mod I engine incorporates only a portion of the automotive reference engine advancements and still the predicted indicated efficiency of the Mod I engine is 50.1 percent, only one-half percentage point below a value of 50.6 percent obtained by using 75 percent of Carnot (720° C (1328° F) heater, 50° C (122° F) cooler). This reinforces the argument for using 75 percent of Carnot for indicated engine efficiency.

A similar comparison can be made by using the investigation of reference 4. This study assessed comparable kinematic and free-piston Stirling engines at the same power and temperature conditions and found no significant difference in indicated efficiency between the kinematic and free-piston designs. Further, this study suggests that the indicated efficiency can be based on 75 percent of Carnot plus one-half percentage point. The half-point improvement results from reoptimizing the engine at a single low-speed operating point with no requirement for the higher power (as in the automotive engine), high-speed operation. On the basis of these results, the free-piston Stirling engine indicated efficiency for this study was obtained by taking 75 percent of Carnot and adding one-half point for reoptimization, thus, the basis for using 52.1 percent indicated free-piston Stirling engine efficiency.

Overall engine efficiency was then calculated by applying a combustion efficiency of 92.4 percent from reference 9 and a conservative hydraulic converter efficiency of 94 percent from reference 7. This yields an overall engine efficiency of 45.3 percent (ratio of hydraulic power out to fuel in) but does not include auxiliary power requirements, which are accounted for separately. It should be noted that the hydraulic converter efficiency includes the bounce chamber loss, which in reference 7 was minimized by using Freon as the bounce gas. Also note that the reference 7 study indicated the potential for significantly higher converter efficiencies - approaching 99 percent. This was indicated to be achievable by designing and optimizing the engine specifically for hydraulic power output rather than by modifying an existing design of a linear-alternator output engine as was done in reference 10.

To better assess engine operation in the system, a hypothetical engine-performance map was constructed (fig. 4) showing hydraulic power output as a function of engine stroke and hydraulic accumulator-sump pressure differential. Typically, the engine-power output can be represented as a function of stroke raised to a power in the range 1.3 to 1.7. For the purposes of this study, the exponent was chosen as 1.4, consistent with a 1-kW (1.34-hp) output free-piston Stirling engine tested at Lewis. Stable system operation requires that the slope of the system load requirement be steeper than the engine output slope. To provide this characteristic, it was assumed that a null-center-band pump, similar to the design in reference 6, was used in the converter. Although this is a very simplified approach, the resulting map does provide an indication of the range of operating conditions the engine might encounter in the system. The map indicates a stroke range from 70 to 100 percent of full stroke and a power range from 58 to 100 percent of full power for the differential pressure range of 10.3 to 33.1 MPa (1500 to 4800 psi). Examination of available Stirling engine maps, both free-piston and kinematic, indicates that engine efficiency over this power range could vary as much as 10 percent. This 10-percent efficiency range was then applied to the 45.3-percent maximum efficiency previously estimated. This yielded an efficiency range of 40.7 to 45.3 percent and resulted in an average engine efficiency of 43.0 percent, which was then used throughout this study.

One example of the type of free-piston Stirling hydraulic engine believed appropriate to this application is presented in reference 7.

HYDRAULIC SYSTEM

The hydraulic system includes the high-pressure accumulator, the low-pressure sump, and the variable-displacement motor-pump described below. In addition, the system will require some ancillary devices such as over-pressure protection in the form of relief valves, filters, check valves, and a possible shutoff valve. Because flow losses associated with these devices should be minimal, they were not accounted for in this report.

Accumulator

As stated in the section SYSTEM DESIGN APPROACH, the accumulator system (accumulator and sump) was sized to store usable energy equal to the kinetic energy of the vehicle at 113 km/hr (70 mph). Accumulator maximum operating pressure, 34.5 MPa (5000 psi) for the baseline case, was selected on the basis of state-of-the-art hydraulic motor-pump technology. The baseline accumulator pressure ratio was selected at 2.5 (minimum accumulator pressure of 13.8 MPa (2000 psi)) in order to minimize volume and weight while at the same time limiting the differential pressure range over which the motor-pump must operate. Reference 2 indicates that the theoretical optimum accumulator pressure ratio for an isothermalized accumulator would be 2.72.

In the system arrangement addressed in this study, almost all of the energy used by the vehicle must pass through the accumulator in charge-discharge cyclic operation. Thus, it is important that the accumulator cyclic efficiency be as high as possible in order to achieve the best possible vehicle fuel economy. Figure 5 from reference 2 presents accumulator cycle efficiency as a function of the polytropic expansion-compression coefficient and pressure ratio. It is obvious that near isothermal operation or operation at very low pressure ratios is necessary to achieve very high cyclic efficiencies. For purposes of the driving-cycle calculations in this study, the accumulator was assumed to be fully isothermalized. For isothermal operation, the accumulator would be partially filled with a metal foam material to absorb heat during the compression process and to return heat to the gas during the expansion process. For simplicity of calculation, no weight or volume penalty for adding an isothermalizing foam material was included in the basic calculations. These penalties could vary over a very large range and are dependent on the degree of isothermalization required. The potential weight and volume effects of adding isothermalizing material and the degree of isothermalization desired are addressed in the section RESULTS AND DISCUSSION.

Although cylindrical accumulators may be advantageous from an installation and cost consideration, this analysis was limited to calculating weights of spherical accumulators. For simplicity, all of the accumulator calculations were made with the assumption of an ideal gas. For a design stress of 207 MPa (30 000 psi), the volume determined from the stored energy requirement ranged in the study from 0.057 to 0.085 m^3 (2 to 3 ft^3), the baseline system being 0.068 m^3 (2.41 ft^3). Accounting for the compressibility effects of a real gas would increase these values approximately 12 percent. To minimize weight, the accumulator can be fabricated as a composite of a thin aluminum or steel liner with a glass- or Kevlar-wound outer covering. Such a construction would reduce the accumulator weight by one-half as compared with steel. Another advantage of the composite construction is its controlled failure mode with no fragmentation. The sump (low-pressure accumulator) was assumed to be the same size as the high-pressure accumulator and to have the same pressure ratio. Maximum sump pressure for the baseline case was set at 3.45 MPa (500 psi). Composite construction for the sump would save relatively little weight but might still be desirable for safety.

Hydraulic Motor

The variable-displacement motor-pump selected for this investigation was a Volvo Model V-20. Performance data generated by Volvo for a 178-cm^3/rev (10.9-in^3/rev) displacement unit were acquired under NASA Contract NASW-3299 (ref. 3). These data were scaled down, as recommended in reference 3, to the appropriate smaller motor-pump sizes used in this report. For the baseline system, the operating pressure differential range, defined by accumulator and sump maximum and minimum pressures and the ±0.69-MPa (100-psi) deadband, was 33.8 to 10.3 MPa (4900 to 1500 psi). Motor-pump size was based on meeting the three acceleration criteria:

- Standstill to 30.5 m (100 ft) in 4.5 sec
- 0 to 96.5 km/hr (60 mph) in 15 sec
- 80.5 to 113 km/hr (50 to 70 mph) in 10.5 sec

The first criterion was the most severe and thus determined the variable motor-pump size. For the baseline case, this was a unit with 81-cm^3/rev(4.9-in^3/rev) maximum displacement. A typical efficiency curve for the V-20, plotted using the equations derived from test data, is shown in figure 6.

MECHANICAL DRIVE SYSTEM

The mechanical drive system includes gearbox, drive shaft, differential, axle, and wheels. The maximum speed of the hydraulic motor-pump was set at 4000 rpm. For the vehicle to reach a top speed of 113 km/hr (70 mph), a gear reduction of 4.08 from motor to wheels was required. Consider now the selected baseline case using a two-speed (single shift) transmission. The direct gear ratio of 1.25 corresponds to a motor speed of 1225 rpm at 113 km/hr (70 mph), and the single shift then corresponds to 4000-rpm motor speed at 113 km/hr (70 mph). For a vehicle operating with a single gearshift, the shift efficiency was taken to be 98 percent (ref. 11). The axle efficiency, including the final drive and differential gears, is a function of speed and load but, for this study, was approximated as a constant. Based on reference 11, a value of 0.966 was used for the urban driving cycle and 0.974 was used for the highway driving cycle. The range of axle efficiency values was taken from references 3 and 11. Reference 12 uses a constant tire efficiency of 98 percent. This value was used in this study for the highway fuel economy driving cycle. For the urban driving cycle, a more conservative tire efficiency of 93 percent was used. This lower value represents tire slip due to an increased percentage of operating time in an acceleration, and thus high torque, mode. In determining the overall system efficiency for selecting the free-piston Stirling engine power requirement, a combined (urban-highway) weighted tire efficiency of 95.2 percent was chosen.

AUXILIARY COMPONENT POWER REQUIREMENT FOR STIRLING HYDRAULIC DRIVING CYCLE

The free-piston Stirling hydraulic system provides for a more efficient auxiliary drive system than a conventional internal combustion system. As an example, the alternator load for the free-piston Stirling hydraulic system can be driven by a constant-speed hydraulic motor, thereby minimizing windage losses associated with the variable-speed-driven alternator used in conventional systems. There also are some slight differences between auxiliary components used in the two systems, as can be seen schematically in figures 7 and 8. As one example, the Stirling system uses a combustion air-blower which is not required with the spark-ignition system. The power requirements used for the Stirling hydraulic system components in this study are listed below along with the rationale used to arrive at the values. Keep in mind that the engine is either running at or near full power at high efficiency or is off. Also, when the engine is off, fuel must be supplied to the combustion system to provide for heat conduction and radiation losses; this is necessary because the heater head must be at or near normal operating temperature at all times so that power can be supplied on demand. This is all taken into account in the computer simulation. The combustor heat input requirement necessary to maintain the heater head at operating temperature with the engine off was estimated at 1.56 kW (2.09 hp) as follows: Reference 9 was used to extrapolate the two loss components - (1) heat losses in the heat-generating system and (2) heat conduction losses from the cylinders and regenerators. The heat loss in the generating system (radiation and convection in the heat-generating system plus the exhaust loss) was estimated by extrapolating a plot of indicated power versus heat loss in the generating system to zero indicated power. That is, the engine pistons are not moving. This value turned out to be 0.9 kW (1.21 hp). The conduction losses for low, part, or full load were essentially constant, with a maximum of about 2.4 kW (3.22 hp). However, this value can be reduced slightly on the basis of the difference between the heater head temperature of this study (762° C) and that of reference 9 (820° C). The total from the two components was estimated to be 3.12 kW (4.18 hp). This value was arbitrarily cut in half (to 1.56 kW; 2.09 hp) to reflect two major design changes: (1) the free-piston Stirling engine design power was set at about 60 percent of the reference automotive engine power, and (2) it is a single-cylinder engine inherently more conducive to minimizing conduction losses. Additional insulation and special heat dams could be incorporated to further reduce the conduction losses. The power requirements of the auxiliary components are listed in the following table:

Auxiliary component	Engine off		Engine on	
	Component power consumption			
	kW	hp	kW	hp
Airblower	0.02	0.027	0.52	0.70
Alternator, fuel pump and solenoid valves	.08	.11	.08	.11
Electronics for engine and hydraulic controls	.02	.027	.02	.027
Power steering	.04	.054	.04	.054
Water pump	.07	.094	.07	.094
Total	0.23	0.31	0.73	0.98
Fan[a] (power is function of vehicle speed and power demand is programmed into computer simulation)	0.16 0.21 (maximum at 32.18 km/hr (20 mph))			
Power consumption to maintain heater head at temperature	1.56	2.09	Included in engine efficiency	

[a]Fan is off above 20 mph.

All auxiliary components, including the alternator, are powered by the system hydraulics. The hydraulic motors which drive the auxiliaries were assumed to be 90 percent efficient. (Peak efficiency of the variable motor-pump tested was 96 percent.)

The combustion airblower power consumption of 0.52 kW (0.70 hp) at full power was taken from reference 9. When the engine was not running, the combustion airblower power consumption was estimated to be 0.02 kW (0.027 hp). The estimated fuel flow required to maintain the heater head at operating temperature, as discussed above, was used as the basis for estimating this engine-off combustion airblower power consumption with the assumption that the air-fuel ratio was approximately constant.

The electric power consumption for the alternator, fuel pump, and solenoid-actuated hydraulic control valve was estimated in the following manner: Typical driving-cycle fuel economy analysis runs have consistently used 2 A as the vehicle alternator load. Using a 12-V system requires 24 W. This value was then doubled (48 W) to take into account a 9-W electrically driven fuel pump and associated solenoids for hydraulic control. The constant-speed alternator, hydraulically driven, was assumed to have a 60-percent efficiency, thereby resulting in a (48/0.6) = 80-W power consumption.

The power consumption for electronics for engine and hydraulic controls (less solenoids) was estimated to be 12 W. The automotive Stirling engine reference design of reference 9 used 24 W, but the free-piston hydraulic Stirling of this study required a simpler on-off control system which resulted in the selection of 12 W. Including a 60-percent alternator efficiency yielded a power consumption of 20 W.

The power steering requirement for the automotive Stirling reference design engine of reference 9 was 40 W. This study also used 40 W, which should be conservative since the power could be taken directly from the main hydraulic system.

The mode of operation of the electric fan for radiator cooling was consistent with conventional engine operation. The fan was off at speeds above 32.2 km/hr (20 mph). The maximum power consumption at that speed was about 0.16 kW (0.21 hp) and rapidly dropped to about one-eighth of that at 16.1 km/hr (10 mph). To conserve more power, the fan could have been controlled by temperature rather than by speed.

The water pump power consumption for the automotive reference engine (ref. 9) at low speed was 140 W. The free-piston Stirling power requirement was scaled down to one-half that, or 70 W. This is believed to be conservative since the engine runs only about 22 percent of the time and the engine is considerably smaller than the automotive reference engine.

To demonstrate the sensitivity of the system fuel economy to auxiliary power consumption, cases were run with the auxiliary power (engine off) doubled and cut in half. These results were compared with the baseline fuel economy.

ENGINE AND DRIVE SYSTEM COMPUTER ANALYSIS

The fuel economy of the free-piston hydraulic engine and drive system was calculated over both the urban and highway Federal driving cycles. These driving cycles are shown in figure 9. The objective was to determine fuel economy (in mpg) for a given set of conditions over the driving cycles. Each driving cycle was assumed to start with a fully charged accumulator, and the accumulator was always pressurized to its maximum pressure (full-charge condition) at the end of each driving cycle. This was inherent in the system, for the selected charge pressure - speed schedule, and avoided biasing the fuel economy results because of differences in these stored energy levels. An assessment of cold-start penalties for the projected free-piston engines was beyond the scope of this study. Therefore, all of the fuel economy results calculated are for hot-start conditions, including the comparative internal combustion engine vehicle fuel economy results. A schematic representation of the Lewis conventional model for an automotive Stirling cycle system is shown in figure 7. A schematic layout of the selected hydraulic system is shown in figure 1. To simulate the hydraulic system operation, the Lewis conventional-driving-cycle code was modified as shown schematically in figure 8. For either the urban or highway imposed driving cycle, the energy balances were calculated over each 1-sec time interval. The amount of fuel consumed over this 1-sec period was then calculated and summed over the complete cycle.

Although the modified code was modeled to use the free-piston Stirling engine map, as shown in figure 4, it can be readily modified to accept other types of engine maps. In addition, the code modified for hydraulic output provides subroutine capability for sizing the major components (accumulators, variable-displacement motor-pump, etc.) of the system and for estimating the component weights. The code can also be used in parametric studies to determine sensitivities to various effects such as fuel economy as a function of vehicle weight, thermodynamic process, accumulator pressure and volume, number of gearshifts, and drive efficiency.

Because of the limited scope of this study, no attempt was made to modify the computer code to take into account the running of the engine during vehicle deceleration. This was the only time when the accumulator pressure deviated from the pressure schedule and +0.69-MPa (100-psi) deadband. Instead, the resulting pressure discrepancy was made up during the first operational cycle of the next acceleration period. We do not feel that this compromise significantly affected the fuel economy results presented in this report.

Vehicle system weights were estimated by applying differentials to the 1361-kg (3000-lb) inertia weight X-body spark-ignition engine vehicle used as the standard in the study. It was assumed that the weight of the free-piston Stirling hydraulic engine, including its integral hydraulic converter plus transmission, miscellaneous plumbing, and valves, would be equal to the more powerful conventional spark-ignition engine and integral crank drive plus torque converter and transmission. The hydraulic system weight, consisting of accumulator, sump, motor-pump, and hydraulic fluid, was then the total differential weight to be added to the 1361-kg (3000-lb) standard vehicle inertia weight.

It should be noted that vehicle inertia weight, including an allowance for the hydraulic system weight penalty, is an input to the calculation. A specific hydraulic system weight is then calculated in the analysis and becomes an output. If this calculated hydraulic system weight does not reasonably match the original allowance, it may be necessary to iterate on hydraulic system weight or to apply a correction based on a weight sensitivity analysis.

RESULTS AND DISCUSSION

Fuel economy projections were calculated over the Federal urban and highway driving cycles and for the combined driving cycle. These are presented below for the baseline system and for several variations aimed at assessing sensitivity to various system and component design parameters. Included in these results are fuel economy sensitivity as a function of vehicle weight, number of gearshifts, accumulator pressure and pressure ratio, tire efficiency, engine-off heat losses, and auxiliary load variation. In addition, runs were made to assess the benefit of recovering braking energy as well as to determine the increase in fuel economy resulting from lowering the performance requirements. The baseline system design parameters used in the study are presented in table I, along with the hydraulic system weights and projected fuel economies.

To make a meaningful comparison with the Stirling free-piston hydraulic baseline system, the fuel economy of a 1980 spark-ignition Phoenix with three-speed automatic transmission was computer calculated by using the same driving cycle and EPA Clayton dynamometer computer program. The hot-start results for the free-piston Stirling hydraulic system and the spark-ignition engine are shown in figure 10. The Stirling free-piston hydraulic system, though several hundred pounds heavier, showed almost a two-to-one improvement in combined fuel economy (21.5 versus 10.9 km/liter) (50.7 versus 25.6 mpg). The combined projected fuel economy for a 1984 spark-ignition engine also is shown in figure 10. The Stirling free-piston hydraulic system combined fuel economy showed an improvement of more than 80 percent over the projected values for the 1984 spark-ignition engine. Congress, as stated in reference 13, has mandated that the fleet of automobiles built for 1985 must achieve 11.7 km/liter (27.5 mpg) as measured by the dynamometer tests administered by the Environmental Protection Agency. These results conservatively surpass that mandate by 84 percent. Table II shows the effect of vehicle weight on fuel economy. The baseline case is compared with two cases above and two cases below the baseline vehicle weight in 90.7-kg (200-lb) increments. Also shown in the table are the associated accumulator size and hydraulic system weight. As an approximation, the combined fuel economy increased about 1.5 percent for a 45.4-kg (100-lb) weight reduction, and the corollary held for a weight increase.

For baseline fuel economy calculations, the total vehicle weight was taken to be 1633 kg (3600 lb), including a 272-kg (600-lb) hydraulic system weight allowance. The baseline hydraulic system weight was then computer calculated at 274 kg (604 lb); the component breakdown is shown in figure 10. No attempt was made to correct the fuel economy values for the 1.8-kg (4-lb) difference between the assumed and calculated hydraulic system weights. This difference had an insignificant effect on the fuel economy. However, the use of composite materials for the accumulator and sump would reduce the hydraulic system weight by approximately 77 kg (170 lb), yielding a 2.4-percent improvement in fuel economy to 22.1 km/liter (51.9 mpg).

Table III shows the effect on fuel economy of the number of transmission speeds and selected gear ratios. One surprising finding of the study was that the system could achieve the performance requirements with a single-speed (no shift) transmission. It was apparent that a two-speed transmission made a significant improvement in fuel economy of about 12 percent, compared with a single-speed transmission, for the combined driving cycle. An additional shift (three speeds) only increased the fuel economy about 2.4 percent more. Thus, there was not much of an incentive to go beyond a two-speed transmission. As a result, the baseline case was chosen with a two-speed transmission. Figure 11 shows fuel sensitivity to gear ratio for one-, two-, and three-speed transmissions. These calculations were all made for the same vehicle inertia weight of 1633 kg (3600 lb). Some small reduction in the fuel economy of the two and three-speed transmissions would result if the appropriate weight differentials were included in the analysis.

As stated previously, all of the calculations in this study were based on an ideal gas and assumed the thermodynamic process to be isothermal in both the accumulator and sump. This isothermalization can be achieved by filling the units with a high-surface-area material that has high porosity and the appropriate thermodynamic properties. Obviously, this requires larger total accumulator and sump volumes to compensate for the isothermalizing material volume in the accumulator and sump. This correction was not made in the computer calculations since the degree of isothermalization required was not defined. However, the results of the computer analysis provided additional information on the operation of the accumulator system. This allowed further assessment of the degree of isothermalization required and the associated efficiency and weight penalty. Even though an accumulator pressure ratio of 2.5 was chosen for the baseline case, the pressure schedule versus speed, coupled with a +0.69-MPa (100-psi) deadband, resulted in a computer-calculated maximum pressure ratio over the combined driving cycle of 1.89. More importantly, the average pressure swing was only about 1.15. Figure 5 shows the effect of various degrees of isothermalization (varying fill factor) on accumulator cyclic efficiency. Even with zero fill factor (pure adiabatic) the cyclic efficiency at the average pressure ratio (1.15) was about 96 percent. At a fill factor of 10 percent, this increased to about 98 percent; and at a fill factor of 25 percent, to about 99 percent. These efficiencies, along with an estimate of the weight and fuel economy penalties associated with the addition of the isothermalization material, are tabulated as follows:

	Fill factor, percent		
	0	10	25
Accumulator cyclic efficiency, percent	96	98	99
Isothermalization penalty[a]:			
Weight, kg (lb)	--	67 (30.4)	197 (89.3)
Fuel economy, percent	--	-0.9	-2.6
Net fuel economy effect, percent	-4	-2.9	-3.6

[a]Including accumulator containment.

This first-order assessment indicates the optimum fill factor to be of the order of 10 percent. However, this yields a fuel economy improvement of only about 1 percent, which may not be worth the added cost, increased accumulator volume, and complexity. Use of composite materials would have only a small effect on these results since only a third of the weight penalty (isothermalization material plus accumulator containment) comes from the increased containment weight. Isothermalization of the sump, which handles only 10 percent of the energy handled by the accumulator, would have substantially less effect on fuel economy and thus was not evaluated further.

As previously discussed, the tire efficiency was taken to be 93 percent for urban driving and 98 percent for highway driving. If, now, 98-percent tire efficiency were assumed for both the urban and highway driving cycles, the combined fuel economy would increase to 22.2 km/liter (52.22 mpg), an increase of about 3.0 percent over the baseline case. A theoretical 100-percent tire efficiency results in a combined fuel economy improvement of 5.0 percent. These results are presented in table IV.

Another interesting finding of the study was the fuel economy sensitivity to accumulator pressure level and pressure ratio. For a given maximum accumulator pressure there was a very small effect on fuel economy when the accumulator pressure ratio was changed from 3 to 2. Referring to table V, for accumulator maximum pressures of 31.0 MPa (4500 psi) and 27.6 MPa (4000 psi), the fuel economy was relatively constant for each maximum pressure level when the pressure ratio was changed from 3 to 2. However, as the maximum pressure was reduced from 34.5 MPa to 27.6 MPa (5000 psi to 4000 psi) at a constant pressure ratio (2.5), the combined fuel economy increased from 21.5 km/liter to 22.0 km/liter (50.67 mpg to 51.88 mpg), a 2-percent increase. This apparently resulted from reducing motor-pump operation in the low-efficiency, high-pressure, low-speed operating region. The lower maximum pressure accumulator required a larger volume, almost 25 percent larger, which was reflected in the hydraulic system weight. The fuel economy adjustments (< 0.13 km/liter; 0.3 mpg) for these small weight increases were not included in the table as all of these calculations assumed the 1633-kg (3600-lb) baseline vehicle inertia weight.

In the mode of operation selected for this study, the engine cycled on and off to maintain accumulator pressure in accordance with the design pressure schedule and +0.69 (100 psi), -0 MPa (0 psi) deadband. Examination of the computer printouts for the baseline case showed that the engine was on about 23 percent of the time during the urban driving cycle. The engine cycled on about 230 times during the 1372-sec urban cycle for an average on-time of only 1.4 sec. During the 765-sec highway driving cycle, the engine was on 39 percent of the time. The engine cycled on about 180 times during the highway cycle for an average on-time of only about 1.7 sec. This very short average on-time might be expected to make startup and shutdown inefficiencies significant even though the heater head was maintained at operating temperature all the time. Although beyond the scope of this study, increasing the control deadband for the accumulator pressure schedule should be investigated to reduce the cycling and to increase the average on-time periods.

Discussed in the section AUXILIARY COMPONENT POWER REQUIREMENTS is a list of the fixed auxiliary power requirements during a driving cycle when the engine was off. These auxiliaries include airblower, alternator, fuel pump, solenoid valves, electronics for control, power steering, and water pump. These fixed power losses totaled 0.23 kW (0.31 hp). Runs were made with this value doubled and then halved. The results are tabulated in table VI. With a value of 0.46 kW (0.62 hp), the combined fuel economy dropped 3 percent, from 21.53 km/liter to 20.85 km/liter (50.67 mpg to 49.05 mpg). Reducing the fixed auxiliary to 0.115 kW (0.154 hp) increased the combined fuel economy about 2 percent, to 21.9 km/liter (51.56 mpg). Results of this sensitivity comparison showed that the effect of these auxiliary power requirements on the combined fuel economy was modest.

As discussed previously, fuel still had to be supplied when the engine was off (in the standby mode). This fuel was necessary to maintain the engine at or near normal operating temperature at all times so that power could be supplied on demand. The energy loss took into account hot-end engine losses due to radiation, conduction, and exhaust gases. Table VII shows the sensitivity of fuel economy to these losses. Modest changes in combined fuel economy (+3.0 and -3.5 percent) resulted from a 50-percent decrease and a 50-percent increase, respectively, in the baseline hot-end engine standby loss. Although it is desirable to minimize these losses, the fuel economy sensitivity to them was modest.

To assess the benefit accrued from recovering the braking energy, the baseline case was compared with a case run without braking energy recovery. As expected, the difference in highway fuel economy with or without braking energy recovery was very small. On the other hand, a 16.1-percent reduction in fuel economy occurred during the urban cycle, yielding a 12-percent reduction for the combined cycle, when braking energy recovery was eliminated. These results are tabulated in table VIII. For extended braking operation (e.g., down a long hill), once the accumulator has been fully charged, the pump output power could be dissipated through a bypass loop consisting of a control valve to burn off the pressure and a heat exchanger to reject the heat either to ambient air or to the engine coolant loop.

As a final comparison, the effects of relaxing the performance requirements were assessed. In this system, since motor size directly controls the acceleration torque available, one larger and two smaller variable-displacement motor-pumps were investigated. Table IX presents the fuel economy results and acceleration times associated with each performance criterion, along with the required time for each criterion. The baseline system used a hydraulic motor-pump with 81-cm^3/rev (4.49-in^3/rev) maximum displacement. As shown in table VIII, the standstill to 30.5-m (100-ft) acceleration time was met and the 0- to 96.5-km/hr (60-mph) and 80.5- to 113-km/hr (50- to 70-mph) required acceleration rates were exceeded with the baseline system. Going to the extreme and decreasing the motor-pump displacement by about 26 percent (60 cm^3/rev; 3.7 in^3/rev) had little effect on the highway fuel economy and increased the urban fuel economy about 3 percent. The combined fuel economy improvement was then only about 1.6 percent. Again, these results were all based on a vehicle inertia weight of 1633 kg (3600 lb). Adding in the small weight-benefit effect of the 60-cm^3/rev (3.7-in^3/rev) motor-pump (assuming motor-pump weight was proportional to displacement) would have increased this to only about 2.2 percent. This modest fuel economy gain was made at the expense of substantial increases in the acceleration times, from 18 to 52 percent longer than with the baseline unit. On the other hand, increasing motor-pump size to 100 cm^3/rev (6.1 in^3/rev) yielded a relatively hot-performing vehicle with a 0 to 96.5-km/hr (60-mph) time of 9.6 sec at the cost of a fuel economy penalty of 2.4 percent. Again, accounting for the small weight penalty for the larger motor would have increased the fuel economy penalty to only about 3 percent. When this relative insensitivity of fuel economy is considered,cost, packaging, and the marketability of a hot-performing vehicle may be the more significant factors in considering any trade-offs in motor-pump size with acceleration capability.

The question arises as to how much of the 81-percent improvement in fuel economy as compared with the projected conventional 1984 spark-ignition vehicle could be achieved by using a more conventional engine with the same type of energy-buffered hydraulic drive system used in this study. A detailed evaluation of this question was beyond the scope of this study. However, preliminary estimates were made by simply ratioing the fuel economy results of the study on the basis of expected engine and hydraulic pump efficiencies. The results are presented in table X. This approach ignored a number of specific differences such as auxiliary power requirements, idling or engine-off losses, and weight differences. However, it is believed that these differences are unlikely to affect the following conclusions: (1) the free-piston Stirling hydraulic engine and drive system offers fuel economy improvements significantly greater than would be achievable with either a diesel or spark-ignition engine and (2) the kinematic Stirling offers a fuel economy improvement approaching that of the free-piston Stirling engine. However, factors such as cost, complexity, life, and reliability, in addition to the fuel economy differential, give the free-piston Stirling hydraulic engine and drive system a substantial advantage over the kinematic Stirling system.

CONCLUDING REMARKS

Results of this analytical study demonstrate that a 39-kW (52-hp) free-piston Stirling - hydraulic drive automotive system with a pneumatic accumulator can meet and surpass system performance requirements while showing fuel economy at least 81 percent better than that of a projected 1984 spark-ignition engine and about twice that of a 1980 spark-ignition engine, both in X-body class vehicles.

The fuel economy results were sufficiently encouraging that the following simplifying assumptions used in the study should not have significantly altered the projected fuel savings:

(1) The accumulator and sump were isothermalized and used ideal gases.

(2) The total weight differential between the standard spark-ignition vehicle and the free-piston Stirling hydraulic powered vehicle is represented by the accumulator, sump, hydraulic oil, and motor-pump.

(3) The indicated efficiency of the free-piston Stirling engine is equal to that of kinematic Stirling engines for the same heat addition and rejection temperatures and, for different temperatures, can be scaled directly as a percentage of Carnot efficiency.

Although a detailed evaluation has not been made, preliminary estimates also indicate that the free-piston Stirling engine should be significantly better than more conventional alternative engines (spark-ignition, diesel, and kinematic Stirling) with the same type of hydraulic drive system.

No cost projections were made in this study. The hydraulic system arrangement used in this study was not optimized. Other arrangements and combinations of hydraulic components should be further investigated. The study results indicated that there is little incentive to go beyond a two-speed (single shift) transmission; surprisingly the performance requirements could be met with a single-speed (no shift) transmission with only a 12-percent fuel economy penalty for the combined driving cycle.

Although isothermal operation of the accumulator and sump was assumed in the detailed analysis, it was found that the average cycle pressure ratio over the driving cycle was very low. Thus, penalties associated with adiabatic accumulator and sump operation would be expected to be very small, and the addition of an isothermalizing material to the accumulator may be unwarranted. Another important finding of the study was the relatively modest fuel economy sensitivity to engine heat losses during engine-off (standby) condition and to engine and vehicle auxiliary component power requirements such as combustion airblower, alternator, and power steering.

Although this study was analytical in nature, the critical variable-displacement motor-pump performance characteristics were based on actual test data of a prototype unit tested with special emphasis on the critical low-speed, low-power operating region. Considerable work has been conducted with electric and hybrid vehicles using flywheels as energy buffers but little has been done with the free-piston Stirling hydraulic system with an energy buffer. We hope that this report will encourage further work on energy-buffered Stirling hydraulic automotive systems.

Free-piston Stirling engine technology is still in its infancy. Kinematic Stirling engines have been built up to a few hundred horsepower in size, and a major NASA-DOE effort is currently under way in developing one for automotive application at about 60 kW (80 hp). However, the largest free-piston Stirling engine built to date is about 3 kW (4 hp) and the only hydraulic output free-piston Stirling engine under development is a heart pump engine at about 8-W (0.01-hp) output power.

The critical technology need for this engine drive system, then, is to design, build, and test a free-piston Stirling hydraulic engine in the 30- to 40-kW (40- to 54-hp) output range in order to develop and validate this technology in the appropriate size range. The initial logical technical step in this process would be to conduct a preliminary engine and system design effort to generate a specific engine design, to refine the fuel economy and performance projections of this study, and to assess system costs and packaging. Such an engine design generated for automotive application could be of an appropriate size and could be applied almost directly for solar electric power generation, thus significantly reducing development costs for the two applications.

Congress has mandated that the fleet of automobiles built for 1985 must achieve 11.7 km/liter (27.5 mpg) as measured by dynamometer tests administered by the Environmental Protection Agency. This report presents a system concept that conservatively would surpass that mandate by 84 percent.

REFERENCES

1. Beachley, N. H.; and Otis, D. R.: A Study of Accumulator Passenger Cars Based on the Ifield Hydrostatic Pump/Motor Unit. UCRL-15390, Lawrence Livermore Laboratory, University of California, Aug. 1981.

2. Rifkin, William D.: Free-Piston Stirling Engine Accumulator Buffered Hydrostatic Vehicle Drive, Final Report. Energy Research and Generation, Inc., NASA Contract NAS3-21483, 1979.
3. Magi, Mort; et al.: On Variable Hydrostatic Transmission for Road Vehicles, Powered by Supply of Fluid at Constant Pressure. (Rept.-301-7663, Volvo Flygmotor A. B.; NASA Contract NASW-3299.) NASA CR-165246, 1981.
4. Vatsky, Arthur; Chen, H. S.; and Dineen, John: A Conceptual Study of the Potential for Automotive-Derived and Free-Piston Stirling Engines in 30-400 Kilowatt Stationary Power Applications. (MTI 81TR38, Mechanical Technology Inc.; NASA Contract NAS3-21291.) NASA CR-165274, 1982.
5. Schreiber, Jeff: Testing and Performance Characteristics of a 1 kW Free-Piston Stirling Engine. NASA TM-82999, 1983.
6. Toscano, W. M.; Harvey, A. C.; and Lee, K.: Design of Hydraulic Output Stirling Engine, Foster-Miller, Inc., NASA Contract NAS3-22230.
7. White, M. H.: Conceptual Design and Cost Analysis of Hydraulic Output Unit for 15 kW Free-Piston Stirling Engine. NASA CR-165543, 1982.
8. Dowdy, Mac: Mod I Stirling Engine System Performance. (MTI-81 ASE 224PR17, Mechanical Technology Inc., 1981.)
9. Automotive Stirling Reference Engine Design Report. (MTI-81-ASE-164DR-2, Mechanical Technology Inc.; NASA Contract DEN3-32.) NASA CR-165381, 1981.
10. Dochat, George R.; et al.: Design Study of a 15 kW Free-Piston Stirling Engine - Linear Alternator for Dispersed Solar Electric Power Systems. (MTI-79TR47, Mechanical Technology Inc.; NASA Contract DEN3-56.) NASA CR-159587, 1979.
11. Blumberg, Paul: Powertrain Simulation: A Tool for the Design and Evaluation of Engine Control Strategies in Vehicles. SAE paper 760158, Feb. 1976.
12. Johnson, R. A.: Conceptual Design Study of an Improved Gas Turbine (IGT) Powertrain. (DDA-EDR-9719, Detroit Diesel Allison; NASA Contract DEN3-28.) NASA CR-159604, 1979.
13. Gray, Charles L., Jr.; and von Hippel, Frank: The Fuel Economy of Light Vehicles. Sci. Am., vol. 244, no. 5, May 1981, pp. 48-59.
14. Marshall, W. F.; and Stamper, K. R.: Engine Performance Test of the 1975 Chrysler-Nissan Model CN633 Diesel Engine. DOT-TSC-OST-75-44, Energy Research and Development Administration, 1975.
15. Performance Characteristics of Automotive Engines in the U.S. Report No. 7 - Third Series, BERC/OP-78/34, Department of Energy, Bartlesville Energy Technology Center, July 1978.

TABLE I. - STIRLING HYDRAULIC BASELINE INPUT AND OUTPUT PARAMETERS

(a) Input conditions

Accumulator and sump	Isothermal
Available engine power, kW (hp)	39 (52)
Average engine efficiency, percent	43
Maximum displacement of scaled Volvo V-20 hydraulic motor-pump, cm^3/rev (in^3/rev)	81 (4.94)
Maximum speed of hydraulic motor-pump, at 113 km/hr (70 mph), rpm	4000
Total vehicle inertia weight, kg (lb)	1633 (3600)
Accumulator maximum pressure, MPa (psi)	34.5 (5000)
Accumulator minimum pressure, MPa (psi)	13.8 (2000)
Sump maximum pressure, MPa (psi)	3.4 (500)
Accumulator and sump size, m^3 (ft^3) each	0.0683 (2.41)
Motor-pump shaft speed over wheel speed	
Direct connected	1.25
Single shift	4.08
Gear direct drive efficiency, percent:	
Urban	96.6
Highway	97.4
Shift efficiency, percent	98
Tire efficiency, percent:	
Urban	93
Highway	98
Drag coefficient	0.417
Vehicle frontal area, m^2 (ft^2)	1.98 (21.34)
Effective wheel radius, m (in.)	0.305 (12)

(b) Output conditions

Total hydraulic system weight, kg (lb)	274 (604)
Accumulator weight, kg (lb)	145 (320)
Sump weight, kg (lb)	10 (22)
Oil weight, kg (lb)	58 (128)
Volvo motor-pump weight, kg (lb)	61 (134)
Vehicle weight less hydraulic system weight, kg (lb)	1359 (2996)
Clayton dynamometer EPA standard fuel economy, km/liter (mpg):	
Highway	24.03 (56.54)
Urban	19.85 (46.70)
Combined	21.54 (50.67)

TABLE II. - FUEL ECONOMY AND ACCUMULATOR SIZE AT VARIOUS VEHICLE INERTIA WEIGHTS

Vehicle weight with hydraulic		Vehicle weight without hydraulic		Hydraulic system weight		Accumulator volume		Highway		Urban		Combined	
								Fuel economy					
kg	lb	kg	lb	kg	lb	m^3	ft^3	km/liter	mpg	km/liter	mpg	km/liter	mpg
1451	3200	1201	2648	250	552	0.0607	2.145	25.39	59.71	21.19	49.85	22.89	53.85
1542	3400	1280	2822	262	578	.0645	2.279	24.70	58.10	20.51	48.24	22.21	52.23
[a]1633	[a]3600	[a]1359	[a]2996	[a]274	[a]604	[a] .0683	[a]2.414	[a]24.04	[a]56.54	[a]19.85	[a]46.70	[a]21.54	[a]50.67
1724	3800	1438	3170	286	630	.0721	2.550	23.42	55.08	19.24	45.26	20.92	49.21
1814	4000	1517	3344	298	656	.0759	2.682	22.81	53.65	18.69	43.97	20.35	47.86

[a]Baseline.

TABLE III. - FUEL ECONOMY AT VARIOUS SPEED RATIOS

[1633-kg (3600-lb) inertia weight vehicle; isothermal accumulator.]

Motor-pump shaft speed over wheel speed			Highway		Urban		Combined	
Direct	Shift 1	Shift 2	Fuel economy					
			km/liter	mpg	km/liter	mpg	km/liter	mpg
2 shifts (3 speed)								
1.50	2.00	4.08	24.03	56.53	20.33	47.82	21.84	51.39
1.25			24.26	57.08	20.43	48.05	21.99	51.73
1.00			24.36	57.31	20.48	48.18	22.06	51.90
.75			24.17	56.82	20.44	48.09	21.96	51.68
.50			23.58	55.46	20.27	47.68	21.63	50.89
Single shift (2 speed)								
2.50	4.08	0	22.72	53.46	19.39	45.62	20.76	48.84
2.00			23.33	54.88	19.65	46.24	21.15	49.77
1.50			23.90	56.23	19.80	46.58	21.46	50.48
1.25			24.03	56.54	19.85	46.70	21.54	50.67
[a]1.00	[a]4.08	[a]0	[a]23.88	[a]56.19	[a]19.76	[a]46.49	[a]21.43	[a]50.41
.75	4.08	0	23.07	54.28	19.50	45.89	20.96	49.32
.50	4.08	0	21.49	50.56	19.00	44.71	20.05	47.17
No shift (1 speed)								
4.08	0	0	20.67	48.61	17.82	41.93	19.00	44.69

[a]Baseline.

TABLE IV. - BASELINE FUEL ECONOMY AT VARIOUS TIRE EFFICIENCIES

[1633-kg (3600-lb) inertia weight vehicle; isothermal accumulator.]

Tire efficiency, percent	Highway		Urban		Combined	
	Fuel economy					
	km/liter	mpg	km/liter	mpg	km/liter	mpg
93 Urban, 98 highway (baseline)	24.04	56.54	19.85	46.70	21.54	50.67
98	24.04	56.54	20.89	49.15	22.20	52.22
100	24.51	57.65	21.29	50.07	22.63	53.23

TABLE V. - FUEL ECONOMY AT VARIOUS MAXIMUM ACCUMULATOR PRESSURES AND PRESSURE RATIOS

[1633-kg (3600-lb) inertia weight vehicle.]

Accumulator pressure ratio	Accumulator pressure				Accumulator volume		Hydraulic system weight		Highway		Urban		Combined	
	Fuel economy													
	MPa		psi		m^3	ft^3	kg	lb	km/liter	mpg	km/liter	mpg	km/liter	mpg
	max.	min.	max.	min.										
	Maximum hydraulic pressure, 34.5 MPa (5000 psi)													
[a]2.5	[a]34.5	[a]13.8	[a]5000	[a]2000	[a]0.0683	[a]2.414	[a]274	[a]604	[a]24.04	[a]56.54	[a]19.85	[a]46.70	[a]21.54	[a]50.67
	Maximum hydraulic pressure, 31.0 MPa (4500 psi)													
3.0	31.0	10.3	4500	1500	0.0773	2.729	285	628	24.04	56.55	20.24	47.62	21.79	51.26
2.5	31.0	12.4	4500	1800	.0759	2.682	279	616	24.09	56.67	20.21	47.53	21.79	51.25
2.0	31.0	15.5	4500	2250	.0790	2.791	286	631	24.08	56.64	20.21	47.53	21.78	51.24
	Maximum hydraulic pressure, 27.6 MPa (4000 psi)													
3.0	27.6	9.2	4000	1333	0.0869	3.070	292	644	23.93	56.28	20.64	48.55	22.00	51.75
2.5	27.6	11	4000	1600	.0854	3.017	286	631	24.07	56.61	20.65	48.56	22.06	51.88
2.0	27.6	13.8	4000	2000	.0889	3.140	293	646	24.15	56.80	20.63	48.52	22.08	51.93

[a]Baseline.

TABLE VI. - INFLUENCE OF AUXILIARY POWER REQUIREMENTS ON BASELINE FUEL ECONOMY

Baseline system with -	Highway		Urban		Combined	
	Fuel economy					
	km/liter	mpg	km/liter	mpg	km/liter	mpg
0.460-kW fixed auxiliaries	23.54	55.36	19.07	44.86	20.85	49.05
0.230-kW fixed auxiliaries (baseline)	24.04	56.54	19.85	46.70	21.54	50.67
0.115-kW fixed auxiliaries	24.29	57.14	20.30	47.75	21.92	51.56

TABLE VII. - INFLUENCE OF STANDBY HEAT LOSSES ON BASELINE SYSTEM FUEL ECONOMY

Baseline system with -	Highway		Urban		Combined	
	Fuel economy					
	km/liter	mpg	km/liter	mpg	km/liter	mpg
50-Percent increase in heat loss, 2.34 kW	23.63	55.59	19.02	44.75	20.85	49.05
Baseline heat loss, 1.56 kW	24.04	56.54	19.85	46.70	21.54	50.67
50-Percent decrease in heat loss, 0.78 kW	24.46	57.53	20.76	48.84	22.28	52.40

TABLE VIII. - INFLUENCE OF BRAKING ENERGY RECOVERY ON BASELINE SYSTEM FUEL ECONOMY

	Highway		Urban		Combined	
	Fuel economy					
	km/liter	mpg	km/liter	mpg	km/liter	mpg
Baseline system	24.04	56.54	19.85	46.70	21.54	50.67
Baseline system without braking energy recovery	22.80	53.63	16.65	39.17	18.95	44.58

TABLE IX. - EFFECT OF VEHICLE PERFORMANCE ON FUEL ECONOMY

[1633-kg (3600-lb) inertia weight vehicle; isothermal accumulator.]

Hydraulic motor size		0 to 30.5 m (0 to 100 ft)	0 to 96.5 km/hr (0 to 60 mph)	80.5 to 112.6 km/hr (50 to 70 mph)	Highway		Urban		Combined	
					Fuel economy					
cm^3/rev	in^3/rev	Acceleration time, sec			km/liter	mpg	km/liter	mpg	km/liter	mpg
100	6.10	4.06	9.60	6.40	23.76	55.88	19.22	45.20	21.03	49.46
[a]81	[a]4.94	[a]4.50	[a]12.10	[a]8.50	[a]24.04	[a]56.54	[a]19.05	[a]46.70	[a]21.54	[a]50.67
70	4.27	4.85	14.30	10.30	24.08	56.64	20.14	47.36	21.74	51.13
60	3.66	5.30	17.10	12.90	24.04	56.54	20.41	48.01	21.90	51.50
		[b]4.50	[b]15.00	[b]10.50						

[a]Baseline.
[b]Required performance.

TABLE X. - PRELIMINARY COMPARISON OF ALTERNATIVE ENGINES WITH ENERGY-BUFFERED HYDRAULIC DRIVES

Engine and drive system	Engine efficiency, percent	Pump efficiency, percent	Combined fuel economy		Estimated improvement in combined fuel economy, percent
			km/liter	mpg	
Free-piston Stirling hydraulic engine and drive	46	94	21.5	50.7	81
Kinematic Stirling engine and hydraulic drive	42	95	20.0	47.0	68
Diesel engine[a] and hydraulic drive	34	95	16.2	38.1	36
Spark-ignition engine[b] and hydraulic drive	30	95	14.3	33.6	20
1984 spark-ignition engine with automatic transmission	--	--	11.9	28.0	(c)

[a]1975 Chrysler-Nissan model CN633.
[b]1978 Ford 98 CID engine.
[c]Standard.

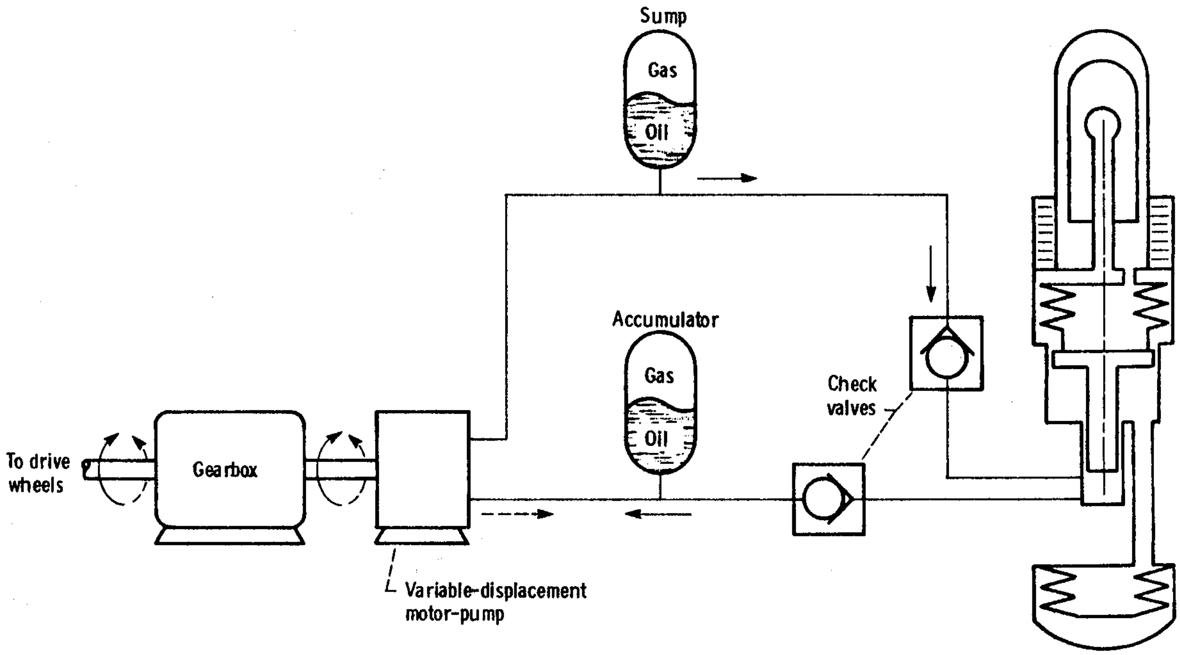


Figure 1. - Free-piston Stirling hydraulic engine and drive system.

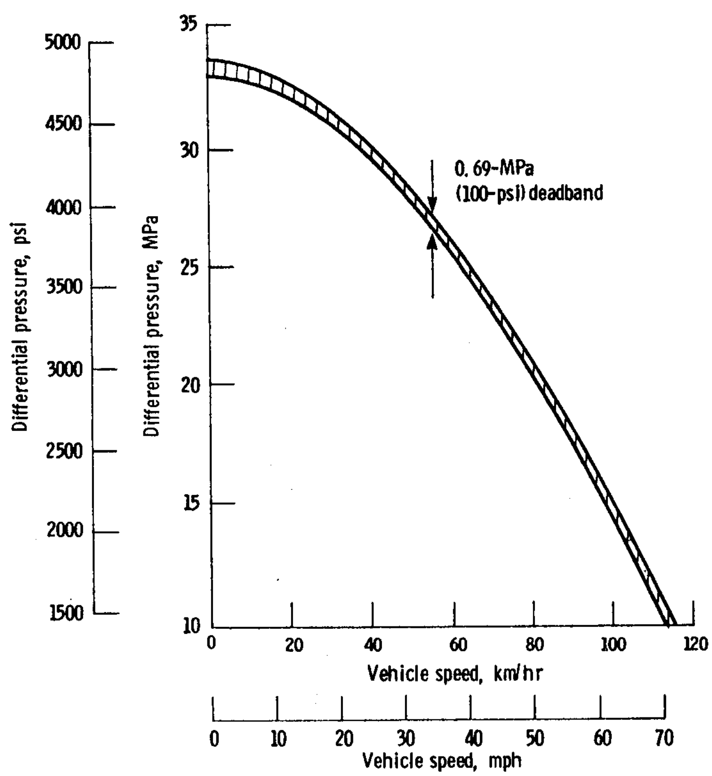


Figure 2. - Effect of vehicle speed on differential pressure (accumulator minus sump) for baseline vehicle.

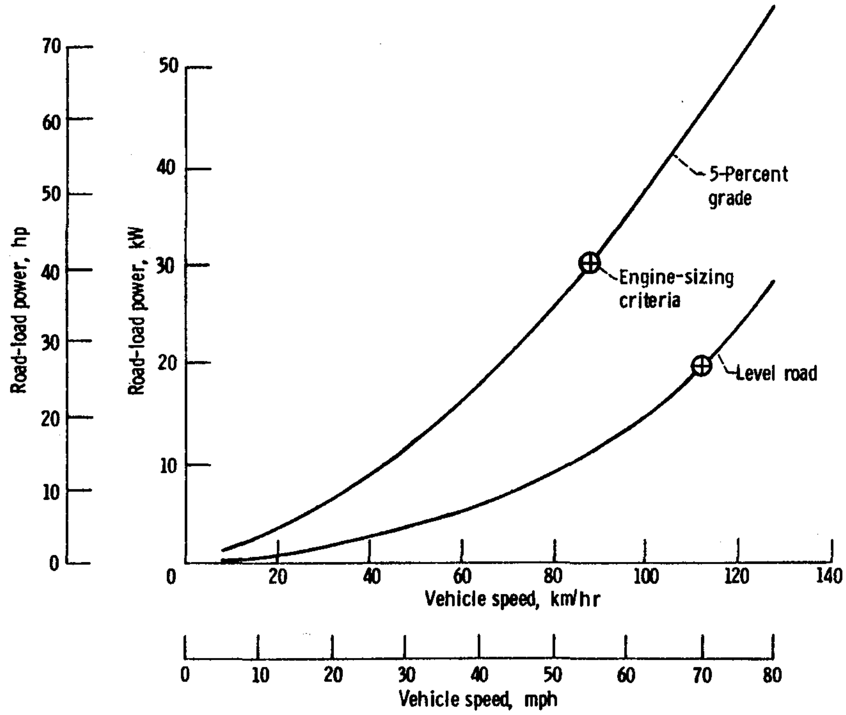


Figure 3. - Automotive road-load power requirements at the wheels for a 1633-kg, 3600-lb vehicle.

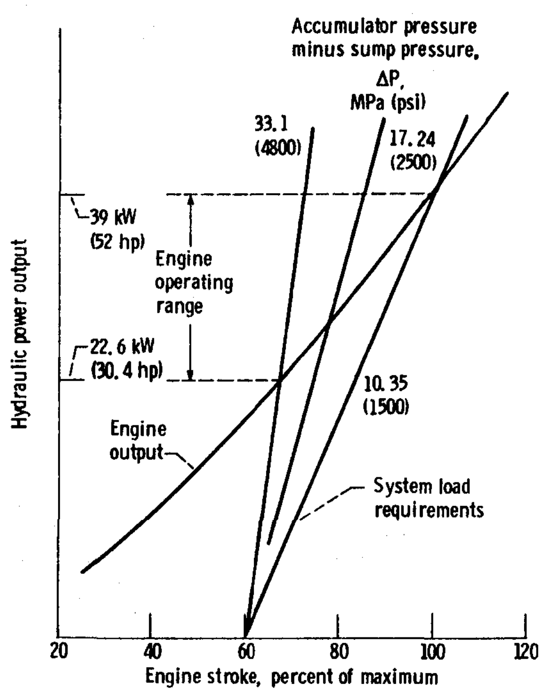


Figure 4. - Operating map of Stirling free-piston engine with hydraulic output. (Average engine efficiency, 43 percent.)

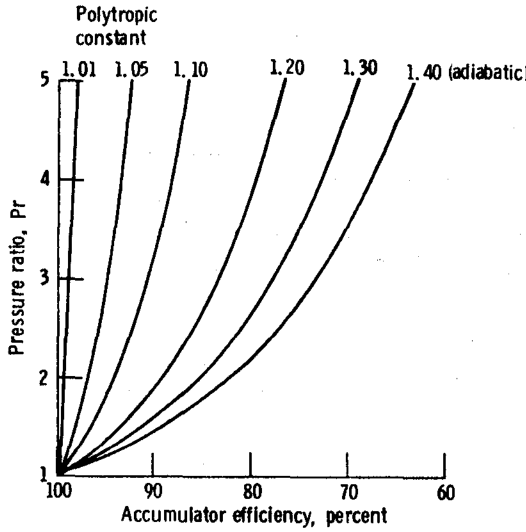


Figure 5. - Accumulator efficiency as a function of pressure ratio and polytropic expansion/compression constant. (From ref. 2.)

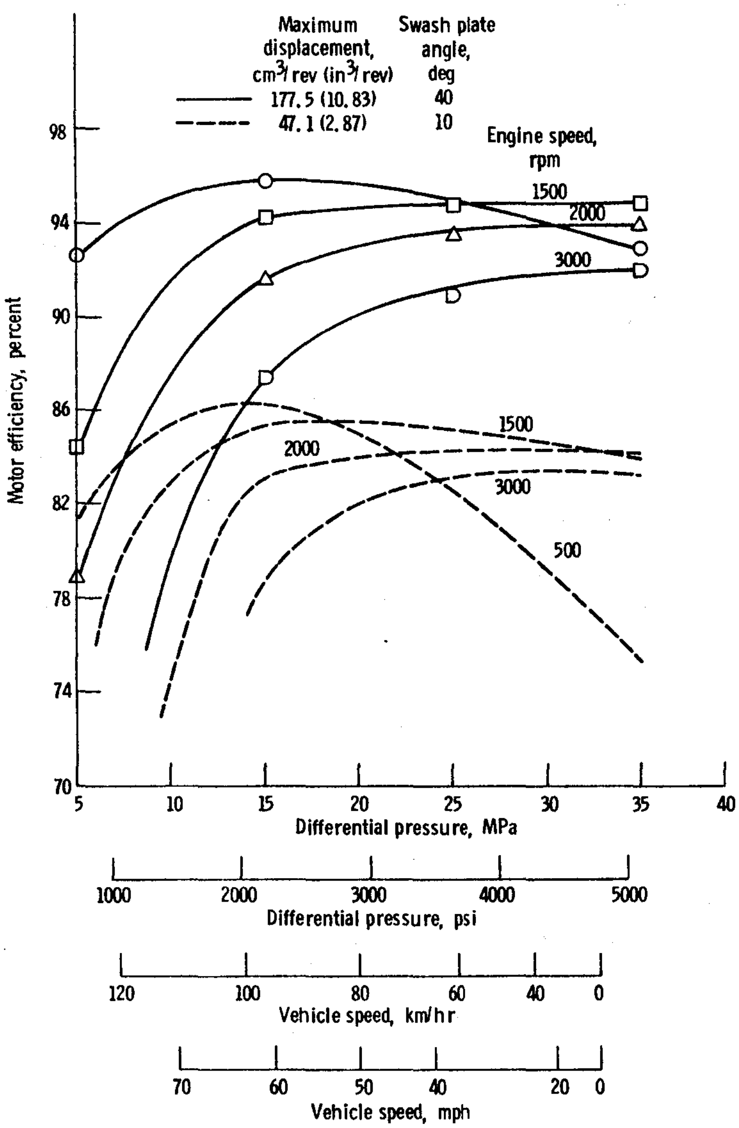


Figure 6. - Volvo V-20 variable-displacement motor effici[illegible] function of

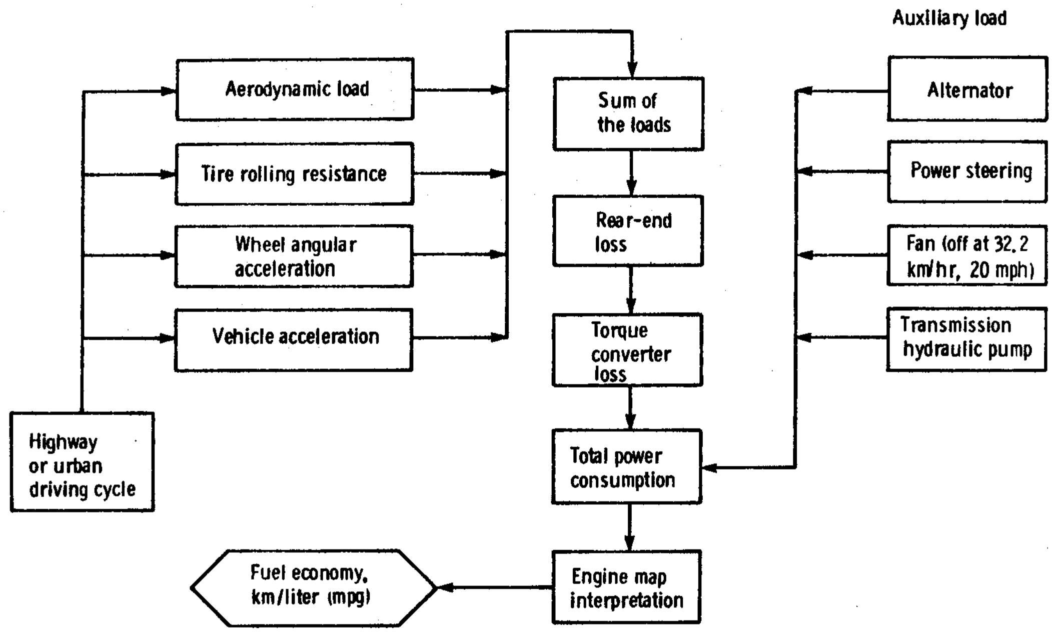


Figure 7. - Schematic representation of conventional model for automotive driving-cycle system.

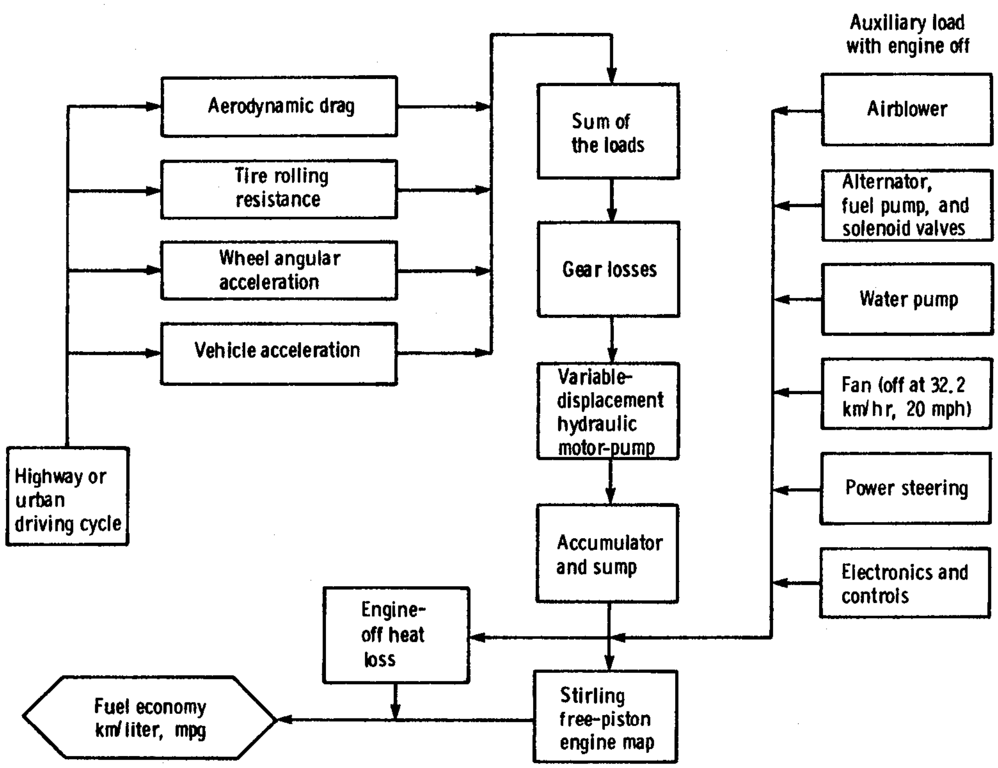
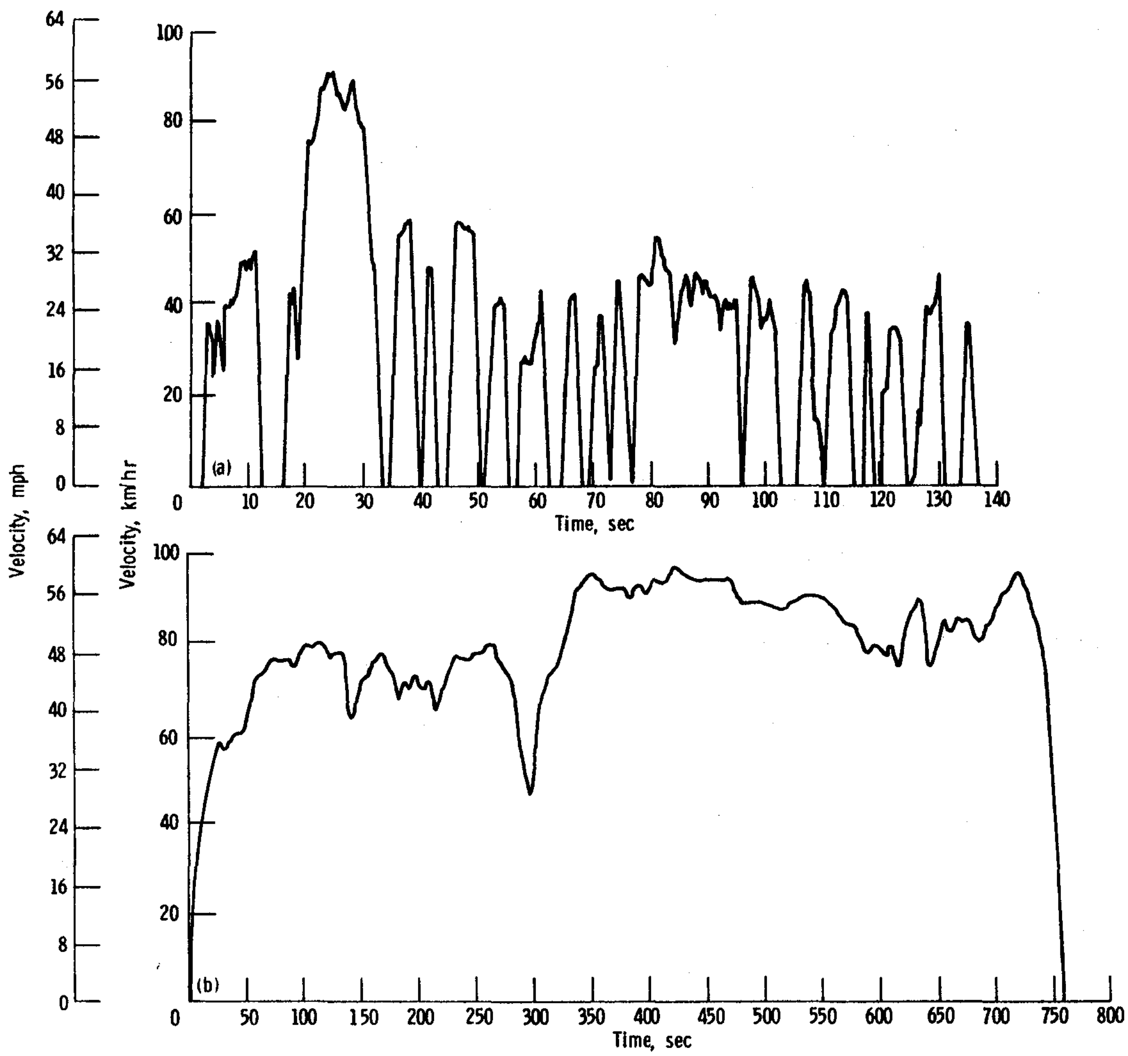


Figure 8. - Computer input code for Stirling free-piston hydraulic system.



(a) Urban Federal driving cycle.
(b) Highway Federal driving cycle.

Figure 9. - Relationship between velocity and time for highway and urban Federal driving cycles.

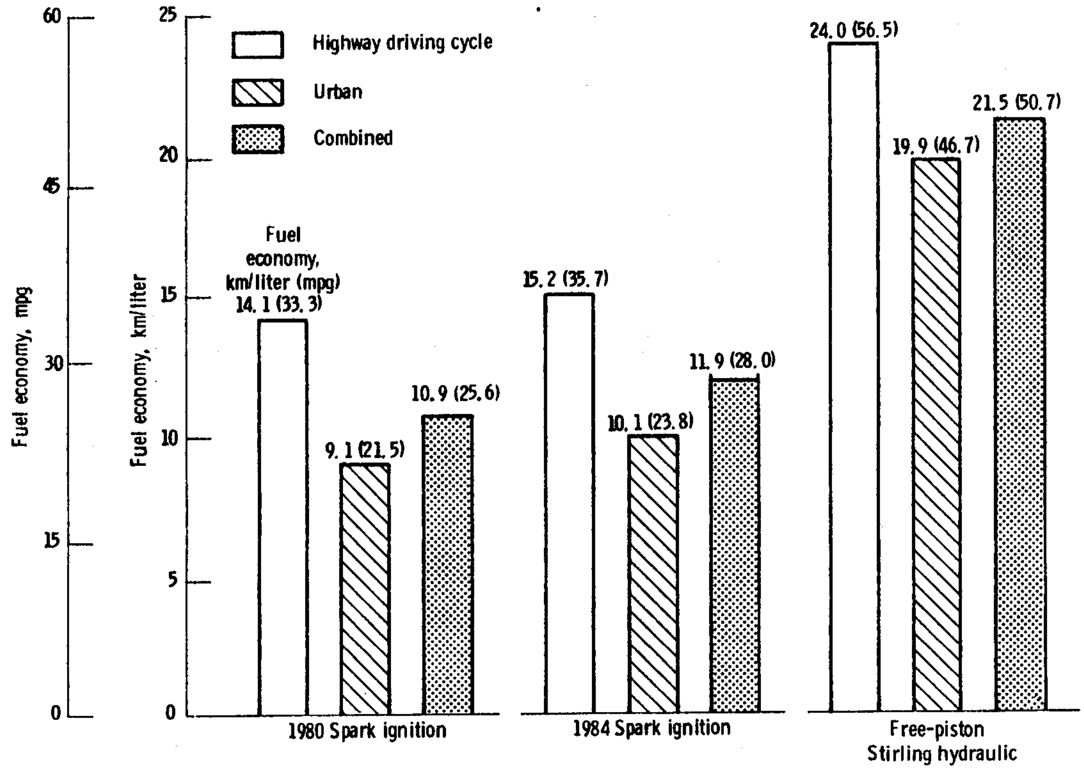


Figure 10. - Hot-start fuel economy of free-piston Stirling hydraulic system and spark-ignition engines in X-body vehicles.

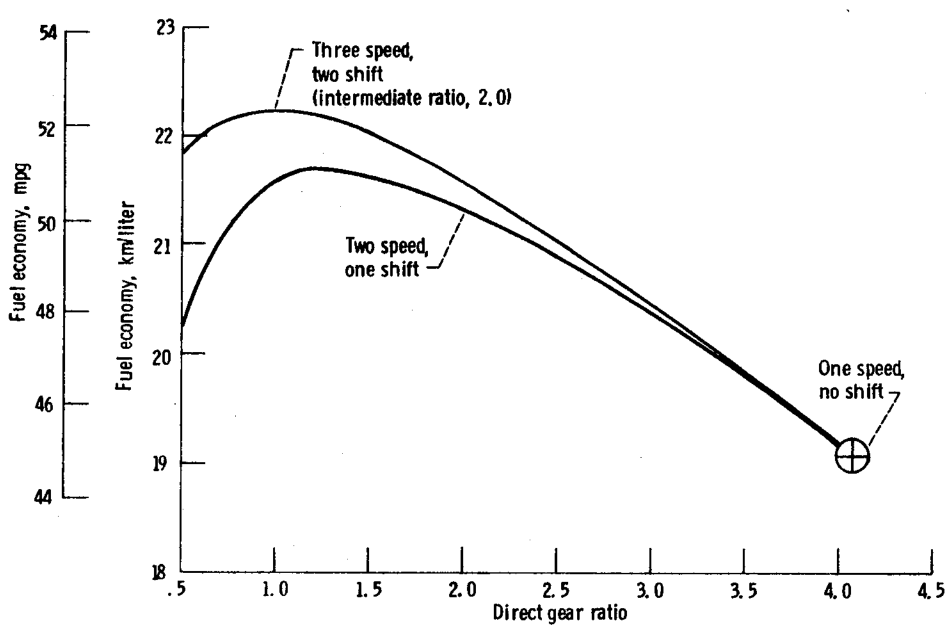


Figure 11. - Effect of final gear ratio on fuel economy of baseline vehicle for combined cycle.

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15. Supplementary Notes

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16. Abstract

This analytical study compares the calculated fuel economy for an automotive free-piston Stirling hydraulic engine and drive system using a pneumatic accumulator with the fuel economy of both a conventional 1980 spark-ignition engine in an X-body class vehicle and the estimated fuel economy of a 1984 spark-ignition vehicle system. The results of the study show that the free-piston Stirling hydraulic system with a two-speed transmission has a combined fuel economy nearly twice that of the 1980 spark-ignition engine - 21.5 versus 10.9 km/liter (50.7 versus 25.6 mpg) under comparable conditions. The fuel economy improvement over the 1984 spark-ignition engine was 81 percent. The study also includes the fuel economy sensitivity of the Stirling hydraulic system to system weight, number of transmission shifts, accumulator pressure ratio and maximum pressure, auxiliary power requirements, braking energy recovery, and varying vehicle performance requirements. An important finding of the study is that a multispeed transmission is not required. The penalty for a single-speed versus a two-speed transmission is about a 12-percent drop in combined fuel economy to 19.0 km/liter (44.7 mpg). This is still a 60-percent improvement in combined fuel economy over the projected 1984 spark-ignition vehicle.

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