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Downsizing Assessment of Automotive Stirling Engines

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DOWNSIZING ASSESSMENT OF AUTOMOTIVE STIRLING ENGINES

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SUMMARY

A 67 kW (90 hp) Stirling engine design, sized for use in a 1984 1440 kg (3170 lb) automobile has been serving as the focal point for developing automotive Stirling engine technology under a current DOE/NASA R&D program. Since recent trends are towards lighter vehicles, an assessment was made of the applicability of the Stirling technology being developed for smaller, lower power engines. Using both the Philips scaling laws and a Lewis Research Center (Lewis) Stirling engine performance code, dimensional and performance characteristics were determined for a 26 kW (35 hp) and a 37 kW (50 hp) engine for use in a nominal 907 kg (2000 lb) vehicle. Key engine elements were sized and stressed and mechanical layouts were made to ensure mechanical fit and integrity of the engines. Fuel economy estimates indicated that the Stirling engine would maintain a 30 to 45 percent fuel economy advantage over comparable spark ignition and diesel powered vehicles in the 1984 time period. In order to maintain the performance advantage, particular attention must be paid to the Stirling engine mechanical losses and, although not evaluated in this report, the cold start penalties.

INTRODUCTION

The DOE/NASA Automotive Stirling Engine (ASE) Development Program was initiated in March 1978 to develop Stirling engines for automotive use and to transfer Stirling engine technology to the United States (ref. 1). The original program was aimed at developing technology for an automotive Stirling in the 67 kW (90 hp) range for a nominal 1360 kg (3000 lb) vehicle. Since the current trend is toward smaller and lighter vehicles, the question arose as to whether the technology being developed for the 67 kW (90 hp) reference engine was applicable for lower power engines. The purpose of this effort was to apply the larger engine technology to engines in the 26 to 37 kW (35 to 50 hp) range, determine if there were any major compromises in Stirling performance and, if so, define technology areas that needed advancement to maintain the relative performance advantage (ref. 2) of the Stirling engine over internal combustion engines.

The 26 kW (35 hp) level was initially selected because it represented one of the lowest power automotive engines in recent history (i.e., VW Beetle). The 37 kW (50 hp) level was subsequently selected because it was more representative of the power required for a vehicle of the 907 kg (2000 lb) weight class. It was also the same power level used in a recent study (ref. 3) comparing a downsized Advanced Gas Turbine with comparable Spark Ignition and Diesel powered vehicles. This allowed a comparison of the downsized Stirling engine to other heat engines on a consistent basis.

The approach taken was to (1) take an existing design for a 67 kW (90 hp) Stirling engine and scale it down according to the Philips (ref. 4) and United

Stirling AB(USAB) scaling laws, (2) perform a mechanical layout of the downsized engine considering loads and stresses to determine proper fit and mating of parts, (3) make necessary compromises between design and performance to arrive at a mechanically achievable design, (4) ascertain final engine performance using a Stirling Engine Performance Code (ref. 5), and (5) determine EPA fuel economy and performance for this downsized engine in a nominal 907 kg (2000 lb) vehicle. Although this approach precluded arriving at an optimized engine it was sufficient to determine if major performance degradations would occur.

This report presents the assumptions, approach and procedure used in downsizing, results and sizing for a 37 kW (50 hp) engine (including preliminary results for the 26 kW (35 hp) engine) and, comparisons of fuel economy and performance of the 37 kW (50 hp) Stirling engine against that of internal combustion, diesel and gas turbine engines in a 964 kg (2125 lb) vehicle (ref. 3). Finally an assessment is made of the applicability of the technology being developed at the higher power levels to that of the lower horsepower engines.

APPROACH AND PROCEDURE

Approach

The general approach was to take an existing design for a 67 kW (90 hp) Stirling engine (e.g., ref. 6) and scale it down to the lower power level using scaling laws. This also required scaling of the mechanical and auxiliary power losses to achieve the desired net power output. Once the scaling laws defined the bore, stroke, speed and other details of the engine, its performance was calculated with the Stirling Engine Performance Code (see ref. 5). The calculated power was then compared to the desired power, the scaling factor adjusted, and the process repeated until the desired power was achieved. This usually took one or two iterations. After the engine dimensions were defined, a preliminary mechanical layout of the engine, considering loads and stresses, was made to determine the general fit and mating of parts. In some instances, changes in key dimensions critical to the performance of the engine were required. At this point tradeoffs between combinations of dimensions and the predicted performance were made to allow a mechanical fit of parts which maximized performance. The Stirling Engine Performance Code was then used to formulate a performance map of the engine from which fuel economy and acceleration estimates were made for a 964 kg (2125 lb) inertia weight vehicle using an in-house vehicle performance code.

The baseline engine used for scaling purposes is the 67 kW (90 hp) reference engine design described in reference 6.[1] An overall view of the design and pertinent engine details is given in figure 1 and table 1, respectively. Briefly the engine design develops 67 kW (90 hp) at full power and weighs 190 kg (418 lb). A predicted performance map for the engine is given in figure 2. The expected combined cycle EPA fuel economy with this engine in a projected 1984 vehicle with a test weight of 1440 kg (3170 lb)

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[1]A later version of the reference engine with slightly lower horsepower is described in more detail in reference 2.

(e.g., x-body) is 17.9 km/l (42.1 mpg) using gasoline and 20.6 km/l (48.5 mpg) using diesel fuel. Zero to 60 mph acceleration is estimated to be 15 sec. The same vehicle with a spark ignition engine yields 11.5 km/l (27 mpg) combined fuel economy with gasoline and 15-sec 0 to 60 acceleration.

Scaling Procedure

In scaling the 67 kW (90 hp) reference engine down to lower powers it was assumed the basic engine design remained the same (i.e., 4 cylinder square "U" drive as shown in fig. 1 and refs. 2 and 6) even though this may not be the best mechanical arrangement. The overall scaling process involved several steps. First, the scaling procedure was applied to the reference engine to arrive at a "first try" design for the reduced power level engine. Then the Lewis Stirling Engine Performance Code was used to predict performance for the new engine (the code was first calibrated so that the power predicted for the reference engine agreed with that predicted by USAB). The new design was then fine tuned, by adjusting the scaling factor and thus the engine dimensions, until the Lewis predicted power equaled the desired value.

The scaling procedures were derived from scaling rules used by Philips and United Stirling. One of these procedures, outlined in reference 4, involves scaling of engine speed, in addition to engine dimensions, with the objective of maintaining engine efficiency nearly constant. The other procedure is similar except that engine speed is not scaled.

The two procedures are:

Procedure 1

1. Scale the linear dimensions of the engine proportional to a scaling factor, λ. Therefore, the engine swept volume, V, is scaled proportional to λ^3.

$$V_{new} = \lambda^3 V_{old}$$

2. Maintain engine design speed, N, constant.

$$N_{new} = N_{old}$$

For the ideal and Schmidt Stirling cycles, engine indicated power is proportional to swept volume and speed. Therefore, the ratio of the indicated power of the scaled engine, P_{new}, to that of the reference engine, P_{old}, is:

$$\frac{P_{new}}{P_{old}} = \frac{V_{new}\ N_{new}}{V_{old}\ N_{old}} = \frac{(\lambda^3\ V_{old})\ N_{old}}{V_{old}\ N_{old}} = \lambda^3 \qquad (1)$$

$$\text{or } P_{new} = \lambda^3\ P_{old}$$

These same relationships are approximately true for practical Stirling engines, provided the various losses (pressure drop, appendix gap pumping, adiabatic losses, leakage, etc.) are not too large.

Procedure 2

The second scaling procedure is the same as the first except that engine design speed is now scaled inversely proportional to the scaling factor, λ.

$$\text{That is, } N_{new} = \frac{N_{old}}{\lambda}$$

Scaling of engine speed in this manner keeps the linear speed of the piston the same (since piston stroke is directly proportional to λ). Therefore, when the engine is reduced in size the engine design speed can be increased without increasing those loss components that increase with piston linear speed. Using this second scaling procedure, the indicated power ratio (scaled/reference) is

$$\frac{P_{new}}{P_{old}} = \frac{V_{new}\ N_{new}}{V_{old}\ N_{old}} = \frac{(\lambda^3\ V_{old})(N_{old}/\lambda)}{V_{old}\ N_{old}} = \lambda^2$$

$$\text{or } P_{new} = \lambda^2 P_{old}$$

The 67 kW (90 hp) reference engine was scaled down to 26 kW (35 hp) using procedure 2. A later check showed that higher scaled engine efficiency would have resulted if engine design speed had been held constant. Therefore, in scaling the same reference engine to the 37 kW (50 hp) net power level, the engine design speed was not changed (i.e., procedure 1 was used). A complete set of equations used to scale to the 37 kW level is given in appendix A.

The above scaling equations relate the indicated power of the scaled engine to that of the reference design. However the objective of this study was to scale down the reference design to yield a design with a specified net power. It would, therefore, be more convenient to express the scaling factor as a function of the desired and reference engine net powers. This can be done if the mechanical and auxiliary losses for the scaled engine can be expressed as functions of the scaling factor λ and the reference engine mechanical and auxiliary losses.

It was assumed in this scaling study that the mechanical power losses, P_M, scale proportional to λ^3 for the constant design speed scaling method (procedure 1).

That is:

$$P_{M,new} = \lambda^3\ P_{M,old} \tag{2}$$

It was also assumed that part of the auxiliary power requirement, P_{A1}, scaled proportional to λ^3; the remaining part, P_{A2}, was assumed to be independent of engine size (e.g., alternator power). Therefore, the auxiliary power requirement, P_A, for the scaled engine was assumed to be

$$P_{A,new} = P_{A1,new} + P_{A2} = \lambda^3 P_{A1,old} + P_{A2} \tag{3}$$

Using the relationship between indicated, P_{ind}, and net, P_N, powers,

$$P_{ind} = P_N + P_M + P_A \tag{4}$$

and expressions (1), (2), and (3) above, it can be shown that:

$$\lambda^3 = \frac{P_{N,new} + P_{A2}}{P_{N,old} + P_{A2}} \tag{5}$$

Therefore, the scaling factor, λ, can be calculated directly from the desired net power of the scaled engine, $P_{N,\ new}$, the net power of the reference engine, $P_{N,\ old}$, and that portion of the auxiliary power requirement which is not sensitive to engine size, P_{A2}.

Similarly, for procedure 2 where the engine design speed is inversely proportional to the scaling factor, λ, it can be shown that:

$$\lambda^2 = \frac{P_{N,new} + P_{A2}}{P_{N,old} + P_{A2}} \tag{6}$$

(Appendix B gives the auxiliary power breakdown for the 67 kW (90 hp) engine).

In practice, the scaling factor calculated using equations (5) or (6) yields a first try at the design required to produce the desired net power, $P_{N,\ new}$. A computer simulation of the first try design will, in general, predict a net power which differs (by a few percent) from the desired net power; that is because a practical engine model includes working space losses (pressure drop, etc.) which do not in general scale precisely as λ^3 or λ^2. Therefore, the design will in general need further adjustment until predicted net power is satisfactorily close to the desired value. The most rigorous approach is to adjust λ, and then adjust all engine dimensions accordingly, until a design is found which yields the desired power. (A simpler approach is to adjust one engine parameter, such as bore size, to get the desired power.)

A strict application of the scaling procedure outlined above would require that all engine linear dimensions be scaled proportional to the scale factor, λ. In practice it was found that, for various reasons, some exceptions needed to be made to the rule. For example, the reference engine cooler tube I.D. was already quite small. For this reason and because it was a standard size metric tube, the decision was made to leave the cooler tube I.D. unchanged. Thus, the set of equations needed to scale the various engine dimensions does not adhere precisely to the linear scale factor, λ. Deviations from strict application of the linear scale factor are discussed in appendix A.

To summarize, procedure 1 was used, with some deviations from the linear scale factor for certain component dimensions, to arrive at the scaling equations of appendix A. The equations of appendix A were used to scale the reference engine to a nominal 37 kW (50 hp) level. The scaled design was then fine tuned until the Lewis Performance Code predicted the desired 37 kW (50 hp). In a very similar manner, procedure 2 was used to develop a set of scaling equations (not shown) which were used in developing a 26 kW (35 hp) design. In both cases, the mechanical layouts required that additional changes be made to the engine designs.

Stirling Performance Code Calibration

Prior to using the Lewis Stirling Engine Performance Code (ref. 5) to predict downsized engine performance, a comparison of its predictions with that of the USAB engine code was made for the 67 kW (90 hp) reference engine in order to "calibrate" the Lewis code.

The net power predicted by USAB for the reference engine at design is 66.7 kW (89.4 hp). For these same design conditions (15 MPa mean pressure, 4000 rpm engine speed, 820° C heater head temperature and 50° C coolant inlet temperature), the Lewis performance code predicted a net power of 69 kW (92.5 hp). In order to match the net power of the USAB prediction the Lewis code power prediction was adjusted by dropping the heater temperature to 805° C while leaving the other design conditions the same (the heater temperature parameter was chosen as a convenient means for adjusting power level). A comparison of the two codes with the above adjustment is shown in table II. It is seen that the powers now essentially match but the efficiency ratio[2] predicted by the Lewis code is about 7 percent lower than that of the USAB code (the drop in temperature from 820° to 805° C causes less than 1 percent reduction in efficiency). Although not shown, this nominal 7 percent shortfall also occurs at part power conditions (2000 rpm & 5 MPa). For the purposes of this report a heater temperature for the downsized engines of 805° C was used in the Lewis code in order to properly calibrate the power levels. Also, the Lewis predicted efficiency was calibrated by adjusting the fuel flow engine map by 7 percent (to produce a 7 percent efficiency increase) before inputting the engine map to the vehicle performance code (the Lewis prediction in table II includes the power calibration but not the efficiency calibration). These assumptions are necessary since the 67 kW (90 hp) base engine design and performance was determined by USAB code predictions and since a direct comparison of the reference engine and the downsized engine was needed to evaluate any potential major degradations in performance.

Mechanical Layout

The mechanical layouts of the downsized engines were generated using the following major assumptions: (1) the basic engine configuration and operational mode were unchanged from the 67 kW (90 hp) reference engine (ref. 6), (2) the load paths were the same as the reference engine, (3) only static loads were considered in sizing the major components (no transient, thermal or dynamic loading), and (4) only the major load carrying parts of the engine were analyzed (piston, cylinder, tie bolts, connecting rods, crank, crankshaft and crankshaft bearings). The engine dimensions derived from the scaling

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[2]Efficiency ratio is defined as the net engine efficiency divided by the external heat system efficiency. This efficiency ratio is predicted by the Lewis code instead of net efficiency since the external heat system is not modeled.

laws, in combination with the desired working pressure determined the loads on the various elements. These were checked against the loads in an existing engine design (MOD I)[3] and found to be acceptable.

Additional assumptions and constraints used in the cold engine system and drive system were: (1) the piston rod seal housing length was not altered (seal length was considered critical and therefore not altered), (2) the crosshead housing inside diameter (I.D.) was assumed to be the same as the cylinder I.D., (3) the crankshaft cross-sectional area was assumed to be proportioned to the power ratio of the downsized engine over the reference engine, (4) the crankshaft bearing length/diameter ratios were assumed to be identical to the MOD I design, and (5) the coolers used the same tube pattern as the MOD I.

Assumptions and constraints used in defining the hot engine system (external heat system and heater head) were: (1) the preheater plate area was assumed to be proportioned to the ratio of the airflow of the downsized engine to that of the reference engine (i.e., constant efficiency), (2) the air ejectors for the combustion gas recirculation were sized to maintain the same ejector velocity for the downsized engines, (3) the heater tubes were assumed to be the same general shape (convolute) as the reference engine and were adjusted in height to achieve the desired active tube length, (4) the fuel ejectors were similar for both engines although the internal dimensions would be adjusted for the different fuel flow rates, and (5) the combustor reaction volume was assumed to be nominally half of the reaction volume of the MOD I to minimize combustor height (judged to be satisfactory for maintaining an adequate combustion efficiency and to avoid excess wall quenching effects).

Using the above assumptions and the component sizes generated by the scaling laws, a preliminary layout of the downsized engine was made. It was at this point that compromises had to be made to achieve proper mechanical fit of parts without severely affecting the performance of the Stirling cycle. The resulting candidate configurations were checked with the Stirling Performance Code in order to select the combination that least affected performance but met the mechanical constraints.

Determination of Vehicle Performance

With the engine thus defined, an engine performance map was generated using the "calibrated" Lewis Stirling Engine Performance Code. This map was then used with an internal Lewis Vehicle Performance Code to predict both fuel economy and acceleration.

The Vehicle Performance Code is an internal Lewis computer code which calculates automotive vehicle fuel economy for urban, highway and combined driving cycles, as well as the acceleration characteristics under wide-open-throttle conditions. Three sets of input are required: namely, tables

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[3]Whenever insufficient detail was available from the reference engine preliminary design, information from an existing detailed design of the MOD I engine (refs. 7 to 9) was used to help check the assumptions and calculations.

of vehicle component performance data (i.e., accessory loads, transmission losses, and axle efficiency); a list of vehicle and tire constants; and, an engine performance map. Fuel economy calculations output constituent (urban and highway) and combined miles per gallon and transmission efficiency. Acceleration calculations result in a speed-time tabulation through each gear with a summary of specific acceleration times and distances.

No correction is made for cold-start fuel penalties associated with the city driving cycle in the Federal Test Procedure. Thus the actual fuel economies are expected to be somewhat lower than those predicted. For consistency, comparisons with the gas turbine, diesel and spark ignition powered vehicles were all made on the same basis of no cold-start penalty.

For the purposes of this report, the fuel economies were calculated assuming a fuel heating value of 42.771 kJ/g (18,400 Btu/lb) and fuel densities of 0.739 kg/l (6.17 lb/gal) for gasoline, and 0.849 kg/l (7.09 lb/gal) for diesel fuel. All powertrain inertia effects were neglected in fuel economy calculations and dynamometer procedures (see appendix B of ref. 3) were used to determine power requirements. The vehicle acceleration calculations were based on the inertia effects of the vehicle, engine, and wheel assemblies and also included vehicle weight shifts between axles and tire traction limits. Transmission inertia was assumed to be small and was neglected. Transient temperature lags were also not considered.

RESULTS AND DISCUSSION

Two downsized engines were investigated. The first engine was set at an arbitrary power level of 26 kW (35 hp) to examine the effects of a major downsizing. This preliminary effort used simplified assumptions in predicting engine and engine-in-vehicle performance but is included for completeness. A second engine at a power level of 37 kW (50 hp) was more completely investigated since it was representative of the engine power level that would be required for a commuter type vehicle of the nominal 907 kg (2000 lb) weight class. The analytical techniques used for this engine were more rigorous in that an engine performance map was formulated and used with drive train characteristics to enable fuel economy predictions with the Vehicle Performance Code. The 37 kW (50 hp) power level selected was also the same as that used in a recent study (ref. 3) comparing a downsized Advanced Gas Turbine with comparable Spark Ignition and Diesel powered vehicles. This allowed a comparison of the Stirling engine to other heat engines on a consistent basis.

Because of the preliminary nature and simplified approach used, only a brief summary of the 26 kW (35 hp) results is presented. The more detailed discussion and scaling rationale used are deferred until the 37 kW (50 hp) engine is discussed.

26 kW (35 hp) Engine

The power level selected for this engine was arbitrarily set at roughly the lower end of I.C. engine power levels available in the smaller cars (VW Beetle). The intent was to determine if a major downsizing from the 90 hp Reference Engine level to the smaller 35 hp level would require major improvements in technology to maintain the performance advantage of the Stirling over the Internal Combustion engine. The scaling procedure used in this case was the second procedure which scaled speed inversely proportional to the scaling factor, λ. To reach the 26 kW (35 hp) level, the working space linear dimensions of the reference engine were scaled to about 2/3 of their original value and design speed was scaled from 4000 to 6000 rpm. A layout of the initial design led to some changes. For example, it was necessary to decrease the cooler diameter and increase the regenerator diameter to get a satisfactory mating of these components. The engine geometry and parameters finally arrived at for the 26 kW (35 hp) design are summarized in table III.

A cross-section of the resulting downsized engine compared to the 67 kW (90 hp) reference engine is given in figure 3. All major elements have been stressed and checked for fit and mating as discussed in the section on Mechanical Layout. The overall height of the downsized engine is 53 cm (20.9 in) with a diameter of about 40 cm (15.7 in) and a weight of 92 kg (202 lb).

Using the Lewis performance code the engine efficiency for the 26 kW (35 hp) engine was roughly 10 percent less than that of the reference engine at the average operating point. A later check suggested that better efficiency would have resulted if the design speed had not been scaled (as subsequently was assumed for the 37 kW (50 hp) design).

For this preliminary study, an estimate of the fuel economy achievable by the 26 kW (35 hp) Stirling design was made based on earlier fuel economy projections for the reference engine. An equation which can be used to approximate engine-vehicle fuel economy knowing vehicle weight and engine brake power (reference unavailable) is

$$M = \frac{C}{W^{.45} \, P^{.35}} \tag{7}$$

where C = a constant, W = vehicle weight, P = engine net power, and M = fuel economy. Equation (7) can be used to relate fuel economics for vehicles with similar engines but different vehicle weights and engine net power as follows:

$$M_2 = M_1 \left(\frac{W_1}{W_2}\right)^{.45} \left(\frac{P_1}{P_2}\right)^{.35} \tag{8}$$

The effect of changes in engine efficiency, can be accounted for by

$$M_2 = M_1 \left(\frac{W_1}{W_2}\right)^{.45} \left(\frac{P_1}{P_2}\right)^{.35} \frac{2}{1} \tag{9}$$

where = average operating point engine efficiency.

The projected combined driving cycle fuel economy for the 67 kW (90 hp) reference engine in a 1440 kg (3170 lb) vehicle was 17.9 km/l (42.1 mpg) with gasoline. Performance predictions with the Lewis code showed about a 10 percent lower efficiency for the 26 kW (35 hp) engine than for the reference engine. This information can be substituted in equation (9) to predict combined driving cycle fuel economy for the 26 kW (35 hp) engine in a 907 kg (2000 lb) vehicle as follows:

$$M_2 = 17.9 \left(\frac{1440}{907}\right)^{.45} \left(\frac{67}{26}\right)^{.35} \times 0.9 = 27.6 \frac{\text{km}}{\ell} \text{ (65 mpg)}$$

37 kW (50 hp) Engine Design

As discussed previously, the power level of this engine represents that which would be used for future commuter cars of the 907 kg (2000 lb) weight class. The first try at a scaling factor required to yield a design which produces 37.3 kW (50 hp) was calculated via equation (5) of procedure 1 to be:

$$\lambda^3 = \frac{37.3 + 0.63}{67.2 + 0.63} = 0.56 \text{ or } \lambda = 0.824$$

When the engine was scaled using $\lambda^3 = 0.56$ ($\lambda = 0.824$), the predicted net power for the scaled engine was about 1.5 kW (2 hp) greater than desired. The scaled mechanical losses and auxiliary power requirements were: Mechanical losses;

$$P_{M,\text{ new}} = \lambda^3 P_{M,\text{old}} = 0.56(10 \text{ kW}) = 5.6 \text{ kW}$$

Auxiliary Power Requirements;

$$P_{A,\text{new}} = \lambda^3 P_{A,\text{old}} + P_{A2} = 0.56(3.19 \text{ kW}) + 0.63 \text{ kW} = 2.42 \text{ kW}$$

By trial and error it was found that when (1) the engine dimensions were rescaled using $\lambda^3 = 0.54$ ($\lambda = 0.814$), (2) auxiliary requirements were rescaled to 2.35 kW and, (3) mechanical losses were assumed to remain at 5.6 kW, the Lewis code predicted the approximate desired net power of (37.2 kW) 49.9 hp. Rescaling mechanical losses using $\lambda^3 = 0.54$ would have reduced them to 5.40 kW. A design point performance summary for this 37.2 kW (49.9 hp) design is shown in table IV under pre-layout design. At this point in the evaluation, the predicted efficiency ratio of 0.357 was about the same as that predicted by the Lewis code for the reference engine (see table II).

The detailed calculations made in scaling the dimensions according to the scaling factor $\lambda^3 = 0.54$ ($\lambda = 0.814$) are shown in appendix C. The equations used in making the calculations of appendix C are given in appendix A.

The resulting engine dimensions for this pre-layout engine are summarized and compared with the reference engine dimensions in table V. Using these dimensions a preliminary mechanical layout of the engine, considering loads and stresses, was made to determine the general fit-of-parts. The major difficulty found as a result of the layout was that the cooler diameter was too large relative to the regenerator diameter for sound mechanical design. The compromise solution to the problem was to reduce the cooler diameter and increase the regenerator diameter. The cooler diameter was reduced by maintaining the same cooler tube dimensions and spacing, but reducing the number of cooler tubes from 326 to 283 per cylinder. The regenerator matrix diameter was increased from 54.5 mm (2.147 in) to 57.5 mm (2.265 in) and the matrix length was not changed.

The Lewis code was then used to predict performance for the revised design. The results are shown in table IV (after-layout design). It is seen that the modified cooler and regenerator designs reduced net power by 1.3 kW (1.8 hp) and the efficiency ratio from 0.357 to 0.353 (about 1 percent).

At this point, the predicted net power for the new design of 35.9 kW (48.1 hp) was below the desired value of 37.0 kW (50.0 hp). The cylinder bore size was therefore increased to bring the power back to the desired level. Increasing the bore size from 52.9 mm (2.083 in) to 53.8 mm (2.118 in) resulted in a design with a predicted power of 37.8 kW (50.7 hp); the performance for this design is summarized in table IV (Final Design). It is seen that the increase in bore size resulted in a slight decrease in indicated efficiency but a slight increase in the efficiency ratio. The increase resulted because the mechanical and auxiliary losses were not changed for the slight increase in engine power. The predicted efficiency ratio for the downsized engine of 0.355 is about the same as the 0.356 predicted for the reference engine. The details of the engine geometry for this downsized engine are defined in table V under "Scaled engine - Final design, scaled mechanical losses."

The mechanical loss of 5.6 kW for the downsized engine was arrived at by scaling the 10.0 kW reference engine mechanical loss proportional to λ^3. This implies that design mechanical losses are proportional to the design power level. A more conservative estimate, based on mechanical losses expected in existing engines resulted in a calculated mechanical loss of 9.74 kW. Figure 4 shows these estimates along with other data on similarly designed Stirling engines as a function of indicated power. It can be seen that the mechanical losses of 5.6 kW estimated for the downsized engine as well as those shown for the reference engine (dotted line) are considerably lower than current experience indicates. The estimate of 9.74 kW for the downsized engine is nearer what would be expected from today's technology. (P40, (40 kW) and MOD I (53.6 kW) Stirling engines).

If downsized engine mechanical losses are assumed to be 9.74 kW (13.06 hp), auxiliary losses are assumed to remain at 2.35 kW (3.15 hp), and the bore size is increased further to 55.8 mm (2.195 in), then the predicted net power for the resulting design is 37.9 kW (50.9 hp). The performance summary for this design is shown in table IV and the engine geometry is defined in table V. Table IV shows that the conservative mechanical loss estimate of 9.74 kW (13.06 hp) results in an efficiency ratio of 0.322 or down about 10 percent from the 0.356 predicted for the reference engine. It appears that particular attention must be paid to the engine mechanical losses in order to achieve the performance desired for the downsized engine (or reference engine). However, it also appears that if the 10 kW mechanical loss of the reference engine is achievable then the 5.6 kW predicted for the downsized engine appears reasonable.

A cross-section of the downsized engine is given in figure 5. The overall height of the 37 kW (50 hp) engine is 61.5 cm (24.2 in) with a diameter of about 42.7 cm (16.8 in). The weight of the basic engine is estimated to be 119 kg (261 lb) compared to the estimated weight of 190 kg (418 lb) for the 67 kW (90 hp) reference engine. Figure 6 compares the overall sizes of both engines.

37 kW (50 hp) Engine Performance

Using the design parameters arrived at by the scaling methods discussed (table V - After layout, scaled mech. losses) a map of the engine net power was generated by the Stirling Engine Performance Code. This map, shown in figure 7, accounts for both mechanical and auxiliary losses. The off-design mechanical losses were generated using the following equation:

$$P_M = P_{M,D} \frac{N}{N_D} \frac{(\overline{P} + 5\text{MPa})}{20\text{MPa}} \tag{10}$$

where: P = Mean pressure, MPa
N = Engine speed (design value, N_D = 4000 rpm)
P_M = Mechanical power loss (design value, $P_{M,D}$ = 5.6 kW)

The off-design auxiliary power requirements that were used are shown in figure 8. The variation of the power requirements with mean working space pressure was due to the combustion blower power requirement. Linear interpolation was used for mean pressure between 15 and 3 MPa. The 3 MPa curve was assumed to be the minimum requirement.

The engine efficiency ratio calculated for several operating points is shown in tabular form in table VI as a function of engine speed and mean pressure. As mentioned earlier, engine efficiency ratio did not include the losses of the external heat system. Figure 9 gives the external heat system efficiency as a function of the fraction of design fuel flow rate. The resulting fuel flow rate for each operating point can be calculated from the following equation:

$$\dot{m}_g = C \frac{P_{NET}}{\left(\frac{\eta_{net}}{\eta_{ext}}\right) \eta_{ext} H_v} \tag{11}$$

where: $\dot{m}_g$ = fuel flow, gm/sec
H_v = fuel heating value, J/gm
P_{NET} = brake power, kW
C = conversion factor = 1 kJ/sec-kW
$\frac{\eta_{net}}{\eta_{ext}}$ = engine efficiency ratio
η_{ext} = external heat system efficiency

The heating value used for gasoline was H_v = 42.711 kJ/gm (18400 Btu/lbm). In this case, the fuel flows calculated by the above equation were adjusted (divided by 1.07). This adjustment calibrates the Lewis predicted efficiencies (increases then by 7 percent) with the USAB predicted efficiencies for the reference engine. The engine design fuel flow for gasoline is found by direct substitution of the design performance parameters and division by the calibration factor to be:

$$\dot{m}_g = 1 \frac{\text{kJ}}{\text{sec-kW}} \frac{37.9 \text{ kW}}{(0.355)(0.903)\ 42.771 \frac{\text{kJ}}{\text{gm}} (1.07)} = 2.58 \frac{\text{gm}}{\text{sec}} \left(20.5 \frac{\text{lbm}}{\text{sec}}\right)$$

For each of the off-design points, an iterative procedure is required to determine fuel flow since

$$\eta_{ext} = f(\dot{m}_g)$$

The resulting calibrated fuel flow map is shown in figure 10.

Tabular forms of the engine maps of figures 7 and 10 constitute the engine description that was required for the Vehicle Performance Code. Engine performance in a more familiar format is also shown in figure 11 where the specific fuel consumption is plotted against engine power for various speeds.

Fuel Economy Comparisons

In order to determine whether the downsized Stirling maintained a significant performance advantage over other equivalent automotive engines, fuel economies were compared for a specific vehicle utilizing several different power plants. The work of reference 3, which compares fuel economies for a 964 kg (2125 lb) inertia weight vehicle powered by 37 kW (50 hp) Advanced Gas Turbine, Spark Ignition and Diesel engines, was extended by Klann to include the downsized Stirling engine of this report. A 1981 Dodge Colt with improved aerodynamics (C_D = 0.39 to better represent future vehicles) was assumed as the baseline. For an equivalent Diesel-powered vehicle a 1980 Volkswagon diesel engine was scaled up (slightly) to the 37 kW (50 hp) level. Other details of the engines, powertrain characteristics and vehicle characteristics can be found in reference 3. It should be noted that all fuel economies and vehicle accelerations were made on a consistent basis using the Vehicle Performance Code described earlier in this report. Cold start penalty and engine weight was ignored for all engines so all fuel economy projections will be a slightly optimistic. However, the relative comparisons between engines should be valid.

Fuel economy and acceleration characteristics for the 37 kW (50 hp) Stirling were projected for gasoline in a 4 speed manual transmission vehicle and for diesel fuel in a 5 speed transmission vehicle. The sensitivities of these characteristics to drive axle ratio were investigated and are summarized in figures 12 and 13 (both fuel economy plots are for gasoline). The particular Colt vehicle chosen for simulation had a 4 speed manual transmission and a final drive ratio of 3.47 when used with the baseline spark ignition engine. The sensitivity results shown in figure 12, led to a change in the axle ratio from 3.47 to 2.9 for use with the downsized Stirling. This change in axle ratio increased projected fuel economy for the Stirling from 25.9 km/l (61 mpg) to 27 km/l (63.6 mpg). The chosen Diesel-powered vehicle had a

5 speed transmission and a final drive axle ratio of 3.90 when used with the baseline diesel engine. The sensitivity results of figure 13 led to a change in this axle ratio from 3.90 to 3.33 for the Stirling engine.

Comparisons of the Stirling engine fuel economies with those projected for the baseline spark ignition and diesel engines are shown in figure 14. Results are shown for the Federal Urban, Highway, and Combined driving cycles. The fuel economy for the gasoline fueled Stirling over the combined cycle was projected to be about 27 km/l (64 mpg) or 42 percent higher than the spark ignition's 19 km/l (45 mpg). The fuel economy for the diesel fueled Stirling over the combined cycle was projected to be 31 km/l (74 mpg) or 45 percent higher than the diesel engine's 22 km/l (51 mpg).

A vehicle using a metal belt continuously variable transmission (CVT) and gasoline fuel was also used to compare projected fuel economies for the Stirling, gas turbine and baseline spark ignition engines. The results are shown in figure 15. The Stirling combined cycle fuel economy was projected to be 28 km/l (66 mpg) or 10 percent higher than the gas turbine's 26 km/l (60 mpg) and 38 percent higher than the spark ignition's 20 km/l (48 mpg). The same figure shows that for diesel fuel, the combined cycle fuel economy for the Stirling was 32 km/l (76 mpg) or 33 percent better than the diesel engine's 24 km/l (57 mpg).

A summary of wide open throttle acceleration characteristics for the various engines with both the CVT transmissions and the manual transmissions are shown in figures 16 and 17 respectively. In general, the Stirling's acceleration characteristics appear adequate as compared to the other power plants.

To evaluate the effect of the more conservative mechanical loss estimate on the fuel economy of the downsized Stirling, the net engine efficiencies were compared at the design operating point (15 MPa, 4000 rpm) and the average operating point (AOP). The Vehicle Performance Code was used to determine that the average operating point over the combined driving cycle was at 4.63 MPa and 1523 rpm. Table VII shows that the change in net efficiency at the average operating point caused by the increase in mechanical losses reduced the combined driving cycle fuel economy from 27.0 km/l (63.5) mpg to 24.8 km/l (58.3 mpg). Thus, when compared to the spark ignition engines' 19 km/l (45 mpg), the downsized Stirling shows about a 30 percent advantage with the higher estimate of mechanical losses.

Although the AOP efficiency change is the more appropriate value to use in making the above corrections, table VII shows that it wouldn't make much difference, in this case, whether the design point or AOP efficiency change was used. This also justifies the adjustment made in the fuel flow map to calibrate the Lewis predicted efficiency with USAB predicted efficiency at the design operating point instead of at the AOP.

Another point of interest is that the engine efficiency ratio at design for the 37 kW (50 hp) Stirling is essentially the same as that for the reference engine (0.355 vs 0.356). Using the approximate equation discussed earlier (eq. (9)) the fuel economy for the downsized engine is estimated to be:

$$M_2 = M_1 \left(\frac{W_1}{W_2}\right)^{.45} \left(\frac{P_1}{P_2}\right)^{.35} \left(\frac{\eta_2}{\eta_1}\right) = 17.9 \left(\frac{1440}{964}\right)^{.45} \left(\frac{67}{37}\right)^{.35} \frac{0.355}{0.356} = 26.3 \frac{km}{\ell} \text{ (62 mpg)}$$

which is reasonably close to the 27 km/l (64 mpg) calculated using the engine maps and the Vehicle Performance Code. This calculation assumes the external heat system efficiencies for the two cases are the same.

Downsizing Assessment

In general, it appears that the Stirling engine can be downsized from the 66.7 kW (90 hp) power level to the 37 kW (50 hp) power level or lower without major changes in the design and with little effect on engine efficiency. The only significant changes encountered were in trade-offs between the regenerator and cooler sizes and these did not cause major degradations in performance. The method by which the designs were arrived at, that is, scaling from a known higher power design, precluded optimization of the design and hence some gains in performance might be realized if an engine were specifically designed for the lower power levels.

Two areas that warrant special attention are mechanical losses and cold start penalties. As discussed previously, mechanical losses similar to what occurs in existing dual-crank engines significantly degrades the Stirling engine performance. Although a relative performance advantage of 30 percent over the spark ignition engine powered vehicles still appears feasible, significant penalties are incurred by the higher mechanical losses. Thus the technology currently being addressed in the area of mechanical losses (bearings, seals, etc.) must be actively pursued. Future designs should reconsider single shaft designs (V-drive) or other drives with the potential of achieving lower mechanical losses.

Although the cold start penalty has not been addressed in this report (all engines in fuel economy comparisons ignored cold start penalties), it is anticipated that the relative losses due to cold start will increase as the engine size decreases (surface area/volume increases with decreasing size). Further, it is expected that the Stirling will suffer more due to the relative mass of hot parts in the engine (heater head and external heat system). Because of this it is felt that a portion of the performance advantage relative to the spark ignition engine will disappear as the size decreases.

CONCLUDING REMARKS

The purpose of this effort was to determine whether downsizing the Automotive Stirling from a power level of 66.7 kW (90 hp) to 37 kW (50 hp) or lower, would require major new technologies to retain the relative projected performance advantage of the Stirling over the internal combustion engine. In general, it is concluded that the technologies being addressed in the Automotive Stirling Engine Program are adequate. There are two areas (common to all heat engines) however, where close attention to detail must be paid or the high performance potential of the Stirling will be degraded. These are mechanical losses and cold start penalty.

Although the techniques used to arrive at the downsized design were approximate, they were adequate enough to determine if major problems would be encountered in the smaller sized engines. The only area encountered, during the mechanical layouts of the smaller engines, requiring design compromises was that of the regenerator-cooler arrangement. The penalties paid in design trade-offs however were minimal.

Fuel economy comparisons with equivalent 37 kW (50 hp) Spark Ignition, Diesel and Gas Turbine engines in a 964 kg (2125 lb) vehicle indicated that the Stirling had a 38 to 42 percent fuel economy advantage over the I.C. engine using gasoline as a fuel (higher if diesel fuel is used). A 33 to 45 percent advantage is expected over the Diesel engine depending on whether standard transmissions or CVT's are used. Comparisons with the Gas Turbine with a CVT indicated that the Stirling should have about 10 percent better fuel economy. Combined EPA fuel economy estimates for the Stirling in the 964 kg (2125 lb) vehicle ranged between 27 km/l (64 mpg) (31.5 km/l or 74 mpg with diesel fuel) for manual transmissions and 28 km/l (66 mpg) (32 km/l or 76 mpg with diesel) for CVT's. Even assuming no further improvement in mechanical losses over that estimated for the MOD I engine, the Stirling still retains about a 30 percent performance advantage over an equivalent I.C. engine.

APPENDIX A

EQUATIONS USED TO SCALE DOWN THE WORKING SPACE DIMENSIONS OF THE REFERENCE ENGINE DESIGN TO THOSE OF A NOMINAL 37 kW (50 HP) ENGINE

SWEPT VOLUME SCALING

Cylinder diameter, D_c:

$$D_c = \lambda D_{c_R}$$

where the subscript R denotes a reference engine dimension and λ is the linear dimension scale factor.

Piston stroke, S:

$$S = \lambda S_R$$

This yields the desired relationship between old and new swept volumes:

$$V = \frac{\pi D_c^2}{4} S = \frac{\pi \left(\lambda D_{c_R}\right)^2}{4} \lambda S_R = \lambda^3 V_R$$

HEATER SCALING

(1) Scale heater tube dead volume according to λ^3:

$$\therefore \frac{V}{V_R} = \frac{\frac{\pi D^2}{4} L N_t}{\frac{\pi D_R^2}{4} L_R N_{tR}} = \frac{D^2 L N_t}{D_R^2 L_R N_{tR}} = \lambda^3 \tag{1}$$

where

D, D_R heater tube inside diameters
L, L_R tube lengths
N_t, N_{tR} number of tubes

(2) Scale heater tube outside heat transfer area proportional to λ^2 (maintain tube wall thickness and ratio of effective heat transfer length to tube length the same):

$$\therefore \frac{(D + 2W_R)\, L N_t}{(D_R + 2W_R)\, L_R N_{tR}} = \lambda^2 \tag{2}$$

where

W_R reference engine tube wall thickness

(3) Scale the spacing between heater tubes according to $\lambda^{1/3}$ (see ref. 5):

$$\therefore S = \lambda^{1/3} S_R \tag{3}$$

where S, S_R is the spacing between heater tubes.

(4) Scale the average diameter of the heater tube array, and therefore, the average circumference around the array proportional to λ.

$$\therefore \frac{N_t(D + 2W_R + S)}{N_{tR}(D_R + 2W_R + S_R)} = \lambda$$

(N_t is the number of tubes per cyclinder. The circular heater tube array consists of tubes for four cylinders.)

$$\therefore \frac{N_t}{N_{tR}} = \lambda\left(\frac{D_R + 2W_R + S_R}{D + 2W_R + S}\right) \tag{4}$$

Equations (1) and (2) are used to solve for the new heater tube diameter as follows:

$$\frac{\text{Eq. 1}}{\text{Eq. 2}} \Rightarrow \frac{D^2}{D_R^{\,2}} = \lambda\,\frac{D + 2W_R}{D_R + 2W_R}$$

$$\text{or } (D_R + 2W_R)D^2 - \lambda D_R^{\,2} D - 2\lambda D_R^{\,2} W_R = 0$$

Solving the quadratic equation for D yields:

$$D = \frac{\lambda D_R^{\,2} + \sqrt{\lambda^2 D_R^{\,4} + 8\lambda D_R^{\,2} W_R(D_R + 2W_R)}}{2(D_R + 2W_R)} \tag{5}$$

When equation (3) is substituted into equation (4), a new value for the number of heater tubes, N_t can be calculated according to:

$$N_t = \lambda N_{tR}\,\frac{D_R + 2W_R + S_R}{D + 2W_R + \lambda^{1/3} S_R}$$

When the resulting value is rounded off to a whole number and/or adjusted to an even number, equation (2) can be used to calculate a new tube length:

$$L = \lambda^2 L_R \frac{N_{tR}}{N_t}\frac{(D_R + 2W_R)}{(D + 2W_R)}$$

REGENERATOR SCALING

The regenerator linear dimensions were scaled strictly proportional to λ:

$$\therefore\ D = \lambda D_R;\ L = \lambda L_R$$

The matrix porosity was maintained the same.

COOLER SCALING

Cooler assumptions were (1) maintain the cooler I.D. and O.D. the same, and (2) scale the inside heat transfer area according to λ^2. Therefore,

$$\frac{A_{HT}}{A_{HT_R}} = \frac{LN_t}{L_R N_{tR}} = \lambda^2 \tag{8}$$

Also assume

$$\frac{N_t}{N_{tR}} = \lambda \Rightarrow \frac{L}{L_R} = \lambda$$

These assumptions also require that the cooler dead volume scale according to λ^2 since

$$\frac{V}{V_R} = \frac{L\ N_t}{L_R\ N_{tR}} = \lambda^2$$

CONNECTING DUCT DEAD VOLUME SCALING

The connecting duct dead volumes were scaled according to λ^3 except for the regenerator - cooler connecting duct dead volume which was left unchanged because it was already quite small.

APPENDIX B

DEFINITION OF AUXILIARY POWER REQUIREMENTS CONSIDERED IN THE STUDY

One part of the auxiliary power requirement, P_{A1}, was assumed to scale proportional to λ^3. P_{A1} was assumed to be the sum of combustion blower, P_{BL}, water pump, P_{WP}, and compressor, P_{COMP}, power requirements. The remaining part of the auxiliary power requirement, P_{A2}, was assumed to be independent of engine size. P_{A2} was assumed to be the sum of the alternator, P_{ALT}, and the belts, P_{BELTS}, power requirements.

Therefore, the scaling assumption made for the auxiliary power requirements was:

$$P_{A,new} = \lambda^3 P_{A1,old} + P_{A2}$$

$$= \lambda^3 (P_{BL} + P_{WP} + P_{COMP})_{old} + P_{ALT} + P_{BELTS}$$

The reference engine values for the auxiliary power requirements at 15MPa mean working space pressure and 4000 rpm engine speed were:

Combustor blower, P_{BL}, kW (hp)	1.85(2.48)
Water Pump, P_{WP}, kW (hp)	.64(.86)
Hydrogen compressor, P_{COMP}, kW (hp)	.70(.94)
Alternator, P_{ALT}, kW (hp)	.33(.44)
Belts, P_{BELTS}, kW (hp)	.30(.40)
Total, kW (hp)	3.82(5.12)

$$\therefore P_{A1_{old}} = (P_{BL} + P_{WP} + P_{COMP})_{old} = 3.19 \text{ kW } (4.28 \text{ hp})$$

$$P_{A2} = P_{ALT} + P_{BELTS} = 0.63 \text{ kW } (0.85 \text{ hp})$$

APPENDIX C

SCALING CALCULATIONS

Scaling procedure 1 resulted in an engine scaling factor of $\lambda^3 = 0.56$ ($\lambda = 0.824$) to scale the reference engine to the 37 kW (50 hp) level. When the scaling was carried out using this factor, the Lewis computer code predicted brake power for the scaled engine about 1.5 kW (2 hp) too high. By trial and error it was found that $\lambda^3 = 0.54$ ($\lambda = 0.814$) yielded an engine design with a predicted power of about 37 kW (50 hp). The following engine scaling calculations were carried out using $\lambda^3 = 0.54$ ($\lambda = 0.814$).

SWEPT VOLUME SCALING

Cylinder diameter, D_c:

$$D_c = \lambda D_{cR} = (0.814)(65\text{mm}) = 52.9\text{mm} \quad (2.083 \text{ in})$$

Piston stroke, S:

$$S = \lambda S_R = (0.814)(34\text{mm}) = 27.7\text{mm} \quad (1.090 \text{ in})$$

These yield the following relationship between swept volumes:

$$V = \frac{\pi D_c^{\,2}}{4} S = \frac{\pi(\lambda D_{cR})^2}{4}(\lambda S_R) = \lambda^3 V_R = (0.54)(112.8 \text{ cm}^3) = 60.9 \text{ cm}^3 \quad (3.71 \text{ in}^3)$$

HEATER SCALING

Tube inside diameter:

$$D = \frac{\lambda D_R^{\,2} + \sqrt{\lambda^2 D_R^{\,4} + 8\lambda W_R D_R^{\,2}(D_R + 2W_R)}}{2(D_R + 2W_R)}$$

$$= \frac{(0.814)(2.75\text{mm})^2 + \sqrt{(0.814)^2(2.75\text{mm})^2 + 8(0.814)(0.69\text{mm})(2.75\text{mm})^2(2.75 + 2 \times 69\text{mm})}}{2(2.75 + 2 \times 0.69\text{mm})}$$

$$= 2.36\text{mm} \; (0.0930 \text{ in})$$

Tube outside diameter:

$$\text{Tube O.D.} = D + 2W_R = 2.36 + 2 \times 0.69 = 3.74\text{mm} \; (0.1472 \text{ in})$$

Number of heater tubes, N_t:

$$N_t = \lambda N_{tR} \frac{D_R + 2W_R + S_R}{D + 2W_R + \lambda^{1/3} S_R} = (0.814)(24)\frac{(2.75 + 2 \times 0.69 + 1.55 \text{ mm})}{(2.36 + 2 \times 0.69 + .934 \times 1.55\text{mm})} = 21.4 \sim 22$$

Heater tube length, L:

$$L = \lambda^2 L_R \frac{N_{tR}}{N_t} \frac{(D_R + 2W_R)}{(D + 2W_R)} = (0.814)^2 (276\text{mm}) \left(\frac{24}{22}\right) \left(\frac{2.75 + 2 \times 0.69\text{mm}}{2.36 + 2 \times 0.69\text{mm}}\right) = 220.3\text{mm (8.67 in)}$$

Effective heater tube length, L_e:

$$L_e = \frac{220.3}{276} \times 241\text{mm} = 192.4\text{mm (7.57 in)}$$

REGENERATOR SCALING

Regenerator Inside Diameter:

$$D = \lambda D_R = (0.814)(67\text{mm}) = 54.5\text{mm (2.147 in)}$$

Matrix Length, L:

$$L = \lambda L_R = (0.814)(50.9\text{mm}) = 41.4\text{mm (1.631 in)}$$

COOLER SCALING

$$\frac{L\ N_t}{L_R N_{tR}} = \lambda^2 \qquad (1)$$

Assume

$$\frac{N_t}{N_{tR}} = \lambda = 0.814$$

Therefore,

$$N_t = \lambda N_{tR} = (0.814)(400) = 325.6 \sim 326$$

Then, from equation (1),

$$\frac{L}{L_R} = \lambda = 0.814$$

$$L = \lambda L_R = (0.814)(68\text{mm}) = 55.3\text{mm (2.176 in)}$$

Effective cooler tube length:

$$L_e = \lambda L_{e_R} = (0.814)(56\text{mm}) = 45.5\text{mm (1.793 in)}$$

CONNECTING DUCT SCALING

Volume:

Use

$$V_{new} = \lambda V_R$$

Compression space cooler:

$$\lambda^3 V_R = (0.54)(46.8 \text{ cm}^3) = 24.3 \text{ cm}^3 \ (1.542 \text{ in}^3)$$

Cooler-regenerator:

Leave unchanged at 1.8 cm^3 (0.1098 in^3).

Regenerator-heater:

$$\lambda^3 V_R = (0.54)(24 \text{ cm}^3) = 13.0 \text{ cm}^3 \ (0.791 \text{ in}^3)$$

Heater - expansion space:

$$\lambda^3 V_R = (0.54)(16.8 \text{ cm}^3) = 9.07 \text{ cm}^3 \ (0.554 \text{ in}^3)$$

CYLINDER AND REGENERATOR HOUSING WALL THICKNESS

These were originally scaled proportional to λ but were adjusted during the engine layout.

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TABLE I. - REFERENCE ENGINE DETAILS (REF. 6)

Overall dimensions:		
Height, cm		695
Diameter of preheater, cm		510
Height from output shaft, cm		480
Operational data:		
Heater tube outside wall temperature, °C		820
Coolant top tank temperature, °C		50
Max. cycle mean pressure, MPa		15
Full load engine speed, rpm		4000
Working gas		Hydrogen
Performance:	Max. power	Max. efficiency
Pressure, MPa	15	15
Speed, rpm,	4000	1180
Indicated power, kW	80.4	29.0
Mechanical losses, kW	10.0	2.6
Auxiliaries, kW	3.8	.7
Net power, kW	66.7	25.7
External heat system efficiency, percent	90.3	92.4
Net efficiency, percent	34.5	43.3

TABLE II. - POWER CALIBRATION OF LEWIS STIRLING ENGINE CODE AGAINST USAB CODE FOR THE 67 kW (90 hp) REFERENCE ENGINE (REF. 6)

Computer code	Indicated power, kW (hp)	Indicated efficiency, percent	Friction loss, kW (hp)	Auxiliary loss, kW (hp)	Net power, kW (hp)	Efficiency ratio, η_{net}/η_{ext}
USAB	80.41 (107.8)	46.1	10.0 (13.4)	3.8 (5.1)	66.7 (89.4)	0.382
Lewis[a]	81.0 (108.6)	42.9	10.0 (13.4)	3.8 (5.1)	67.2 (90.1)	.356

[a]Heater temperature lowered from 820° C used by USAB to 805° C to nominally match USAB's net power. All other operating conditions and engine dimensions were identical. With a 820° C heater temperature the Lewis code predicted 69.0 kW net power and 0.362 efficiency ratio.

TABLE III. - ENGINE GEOMETRY COMPARISON --- 26 kW (35 hp) SCALED ENGINES AND 67 kW (90 hp) REFERENCE ENGINES

Parameters	Reference engine (Ref. 6)	Scaled 26 kW (35 hp) engine
Drive mechanism, cylinders:		
Piston diameter, mm (in)	65 (2.559)	46.5 (1.832)
Displacer rod diameter, mm (in)	13 (.512)	8.7 (.342)
Displacer dome height, mm (in)	120 (4.72)	80 (3.15)
Displacer-wall GAP, mm (in)	0.4 (.01575)	0.4 (.01575)
Cylinder wall thickness, mm (in):		
hot end	4.0 (.157)	2.54 (.10)
cold end	3.5 (.138)	1.5 (.06)
Crank radius, mm (in)	17 (.6693)	11.3 (.4468)
Stroke, mm (in)	34 (1.339)	22.7 (.894)
Connecting rod length, mm (in)	95 (3.740)	63.4 (2.497)
Swept volume, cm^3 (in^3)	112.8 (6.88)	38.5 (2.36)
Regenerator:		
Units per cycle	1	1
Matrix diameter, mm (in)	67 (2.638)	53 (2.087)
Matrix length, mm (in)	50.9 (2.004)	34 (1.338)
Wire diameter, μm (in)	50 (1.969×10^{-3})	50 (1.969×10^{-3})
Fill factor, percent	.327	.400
Porosity, percent	.673	.600
Housing wall thickness, mm (in):		
hot end	7 (.276)	4.57 (.180)
cold end	3 (.118)	2.30 (.090)
Cooler:		
Units per cycle	1	1
Tubes per cycle	400	236
Tube inside diameter, mm (in)	1 (.03937)	1 (.03937)
Tube outside diameter, mm (in)	1.7 (.06693)	1.7 (.06693)
Tube length, mm (in)	68 (2.677)	45.2 (1.780)
Effective tube length, mm (in)	56 (2.205)	37.2 (1.466)
Estimated effective coolant cross-sectional flow area for total flow per cycle, cm^2 (in^2)	32.3 (5.0)	14.4 (2.228)
Heater:		
Tubes per cycle	24	20
Tube inside diameter, mm (in)	2.75 (.1083)	2.05 (.0806)
Tube outside diameter, mm (in)	4.13 (.1626)	3.43 (.1349)
Tube length, mm (in)	276 (10.87)	178.1 (7.01)
Effective tube length, mm (in)	241 (9.488)	155.4 (6.12)
Connecting Duct Volumes:		
Compression space-cooler, cm^3 (in^3)	46.8 (2.856)	18.9 (1.153)
Cooler-regenerator, cm^3 (in^3)	1.8 (.1098)	1.8 (.1098)
Regenerator-heater, cm^3 (in^3)	24 (1.465)	9.72 (0.593)
Heater-expansion space, cm^3 (in^3)	16.8 (1.025)	6.80 (0.415)

TABLE IV. - PERFORMANCE SUMMARY FOR 37 kW (50 hp) SCALED ENGINE
(DESIGN OPERATION POINT AT 15 MPa AND 4000 RPM)

Design description and changes	Indicated power, kW (hp)	Indicated efficiency, percent	Mechanical losses, kW (hp)	Auxiliary losses, kW (hp)	Net power, kW (hp)	Efficiency ratio, η_{net}/η_{ext}
Pre-layout	45.2 (60.6)	43.4	5.6 (7.51)	2.35 (3.15)	37.2 (49.9)	0.357
After layout: Increased regenerator I.D. from 54.5 to 57.5 mm; decreased cooler diam. by reducing tubes from 326 to 283	43.8 (58.8)	43.1	5.6 (7.51)	2.35 (3.15)	35.9 (48.1)	.353
Final design: Regenerator and cooler same as after layout; increased bore size from 52.9 to 53.8 mm	45.8 (61.4)	42.9	5.6 (7.51)	2.35 (3.15)	37.8 (50.7)	.355
Alternate design using current experience for mechanical losses: Regenerator and cooler same as after layout; further increased bore size to 55.8 mm	50.0 (67.1)	42.4	9.74 (13.1)	2.35 (3.15)	37.9 (50.9)	.322

TABLE V. - ENGINE GEOMETRY COMPARISONS OF 67 kW (90 hp) REFERENCE ENGINE AND VARIOUS 37 kW (50 hp) SCALED ENGINES

Parameters	Ref. engine (Ref. 6)	Scaled engine		
		Pre-layout	Final design, scaled mech. losses	Alternate design, conservative mech. losses
Drive Mechanism, Cylinders:				
Piston diameter, mm (in)	65 (2.559)	52.9 (2.083)	53.8 (2.118)	55.8 (2.195)
Displacer rod diameter, mm (in)	13 (.512)	10.6 (.417)	10.6 (.417)	10.6 (.417)
Displacer dome height, mm (in)	120 (4.72)	97.5 (3.84)	97.5 (3.84)	97.5 (3.84)
Displacer-wall GAP, mm (in)	.4 (.01575)	.4 (.01575)	.4 (.01575)	.4 (.01575)
Cylinder wall thickness,				
mm (in): hot end	4.0 (.157)	3.25 (.128)	3.80 (.148)	3.80 (.148)
cold end	3.5 (.138)	2.84 (.112)	3.25 (.128)	3.25 (.128)
Crank radius, mm (in)	17 (.6693)	13.8 (.545)	13.8 (.545)	13.8 (.545)
Stroke, mm (in)	34 (1.339)	27.7 (1.090)	27.7 (1.090)	27.7 (1.090)
Connecting rod length, mm (in)	95 (3.740)	77.3 (3.044)	77.3 (3.044)	77.3 (3.044)
Swept volume, cm^3 (in^3)	112.8 (6.88)	60.8 (3.71)	62.9 (3.84)	67.7 (4.13)
Regenerator:				
Units per cycle	1	1	1	1
Matrix diameter, mm (in)	67 (2.638)	54.5 (2.147)	57.5 (2.264)	57.5 (2.264)
Matrix length, mm (in)	50.9 (2.004)	41.4 (1.631)	41.4 (1.631)	41.4 (1.631)
Wire diameter, μm (in) ($\mu m = 10^{-6}$ m)	50 (1.969×10^{-3})	50 (1.969×10^{-3})	50 (1.969×10^{-3})	50 (1.969×10^{-3})
Fill factor, percent	.327	.327	.327	.327
Porosity, percent	.673	.673	.673	.673
Housing wall thickness, mm (in):				
hot end	7 (.276)	5.7 (.225)	5.8 (.228)	5.8 (.228)
cold end	3 (.118)	2.4 (.096)	2.5 (.098)	2.5 (.098)
Cooler:				
Units per cycle	1	1	1	1
Tubes per cycle	400	326	283	283
Tube inside diameter, mm (in)	1 (.03937)	1 (.03937)	1 (.03937)	1 (.03937)
Tube outside diameter, mm (in)	1.7 (.06693)	1.7 (.06693)	1.7 (.06693)	1.7 (.06693)
Tube length, mm (in)	68 (2.677)	55.3 (2.176)	55.3 (2.176)	55.3 (2.176)
Effective tube length, mm (in)	56 (2.205)	45.5 (1.793)	45.5 (1.793)	45.5 (1.793)
Estimated effective coolant cross-sectional flow area for total flow per cycle, cm^2 (in^2)	32.3 (5.0)	17.4 (2.70)	17.4 (2.70)	17.4 (2.70)
Heater:				
Tubes per cycle	24	22	22	22
Tube inside diameter, mm (in)	2.75 (.1083)	2.36 (.0930)	2.36 (.0930)	2.36 (.0930)
Tube outside diameter, mm (in)	4.13 (.1626)	3.74 (.1473)	3.74 (.1473)	3.74 (.1473)
Tube length, mm (in)	276 (10.87)	220.2 (8.67)	220.2 (8.67)	220.2 (8.67)
Effective tube length, mm (in)	241 (9.488)	192.3 (7.57)	192.3 (7.57)	192.3 (7.57)
Connecting Duct Volumes:				
Compression space-cooler, cm^3 (in^3)	46.8 (2.856)	25.3 (1.542)	25.3 (1.542)	25.3 (1.542)
Cooler-regenerator, cm^3 (in^3)	1.8 (.1098)	1.8 (.1098)	1.8 (.1098)	1.8 (.1098)
Regenerator-heater, cm^3 (in^3)	24 (1.465)	13.0 (.791)	13.0 (.791)	13.0 (.791)
Heater-expansion space, cm^3 (in^3)	16.8 (1.025)	9.1 (.554)	9.1 (.554)	9.1 (.554)

TABLE VI. - ENGINE EFFICIENCY RATIO, η_{net}/η_{ext} FOR 37 kW (50 hp) STIRLING

Mean pressure, MPa	Engine speed, rpm					
	500	1000	2000	3000	3500	4000
15	0.369	0.396	0.394	0.376	0.367	0.355
11	.327	.385	.391	.378	.369	.358
7	.306	.355	.371	.363	.355	.346
3	.177	.258	.289	.280	.271	.259
1		.029	.060			

TABLE VII. - EFFECT OF MECHANICAL LOSSES ON FUEL ECONOMY OF DOWNSIZED STIRLING (37 kW (50 hp) ENGINE IN 964 kg (2000 lb) VEHICLE)

Type of mechanical losses	Conditions	Mechanical losses, kW (hp)	Efficiency ratio, η_{net}/η_{ext}	External heat system efficiency, percent	Net engine efficiency, percent		Combined fuel economy, km/l (mpg)
					Lewis code	Lewis code "calibrated"	
Advanced technology	Full load: 15 MPa, 4000 rpm (see table IV)	5.6 (7.51)	0.355	90.4	32.1	34.3	[a]27.01 (63.55)
	Average oper. point: 4.63 MPa, 1523 rpm	1.03 (1.38)	.334	91.3	30.5	32.6	[a]27.01 (63.55)
Current engine experience	Full load (see table IV)	9.74 (13.1)	.322	90.2	29.0	31	[b]24.40 (57.41)
	Average operating point	1.79 (2.40)	.306	91.5	28.0	30.0	[c]24.80 (58.34)

[a]Calculated using vehicle driving cycle.

[a]Assumed proportional to change in net efficiency at full load (i.e., $27.01 \times \frac{31}{34.3} = 24.40$).

[b]Assumed proportional to change in net efficiency at average operating point (i.e., $27.01 \times \frac{30}{32.6} = 24.80$).

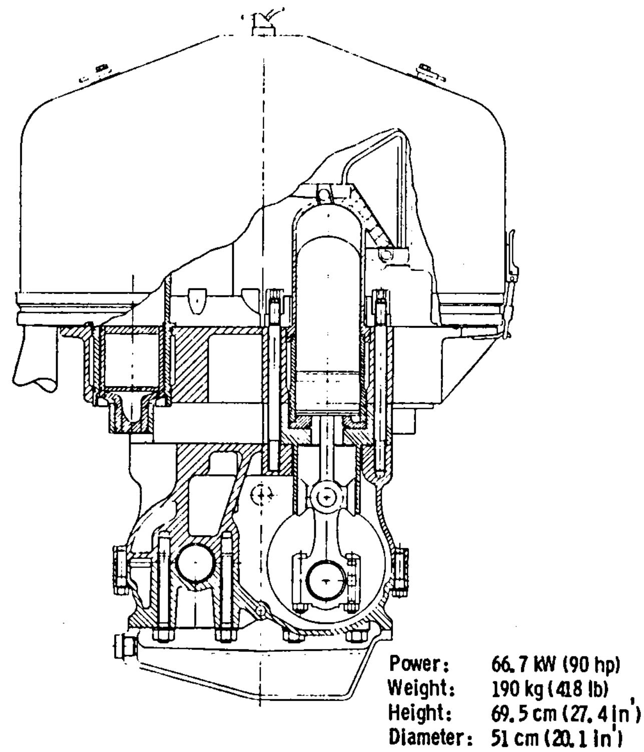

Figure 1. - Overall view of reference Stirling engine.

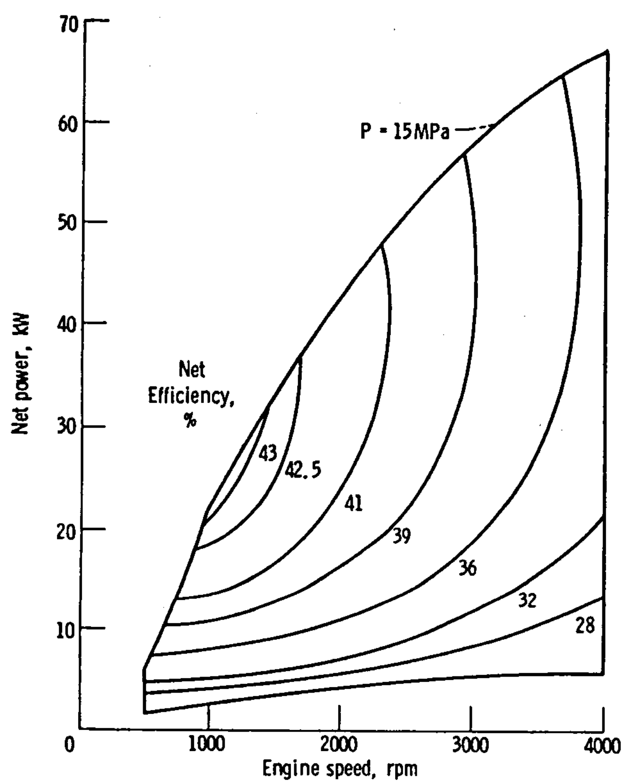

Figure 2. - Performance map of 67 kW (90 hp) reference engine.

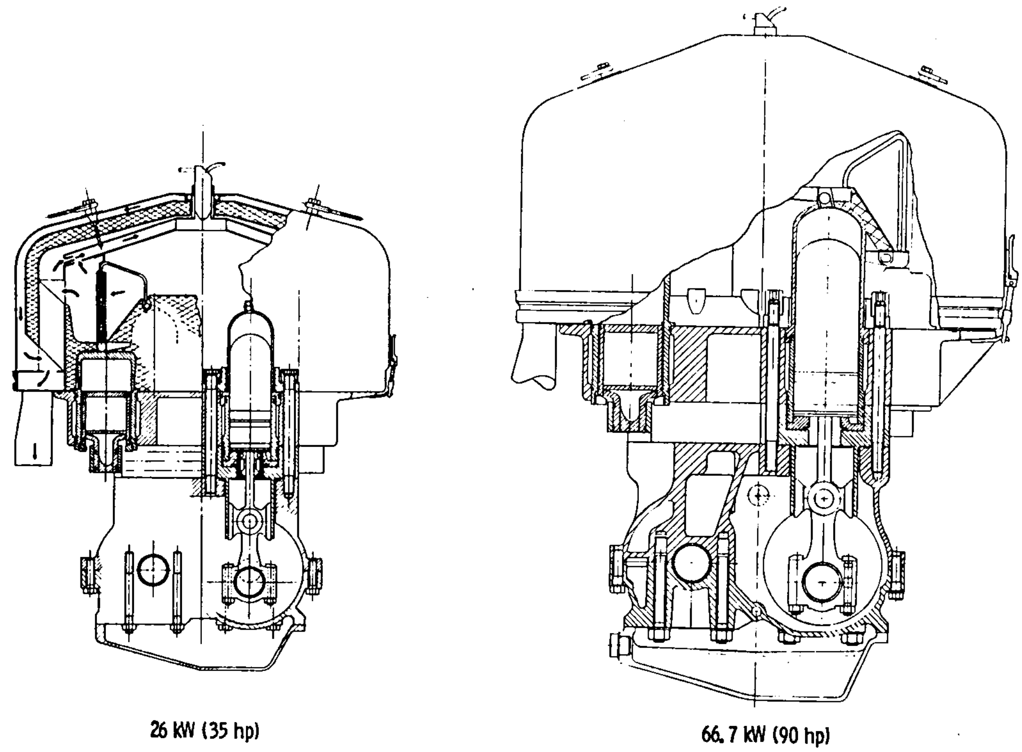

Figure 3. - Size comparison of downsized 26 kW (35 hp) with 66.7 kW (90 hp) reference engine.

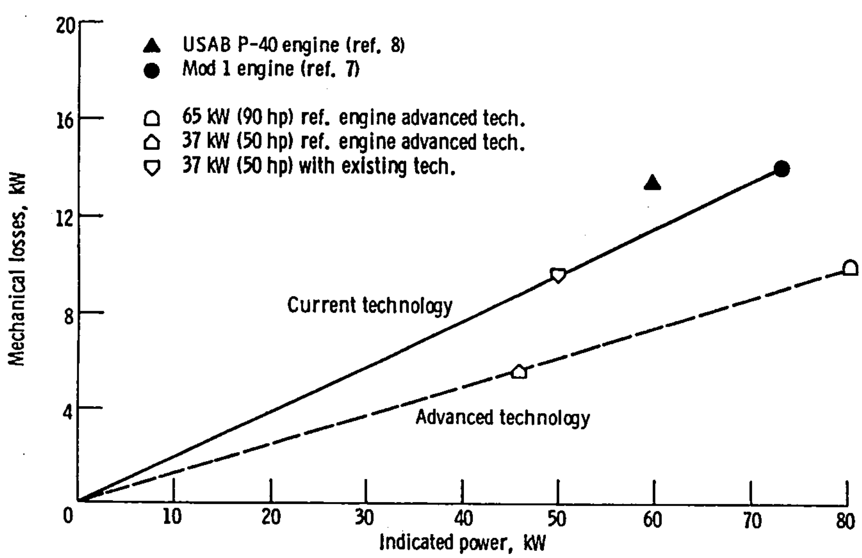

Figure 4. - Mechanical losses of dual-crankshaft Stirling engines.

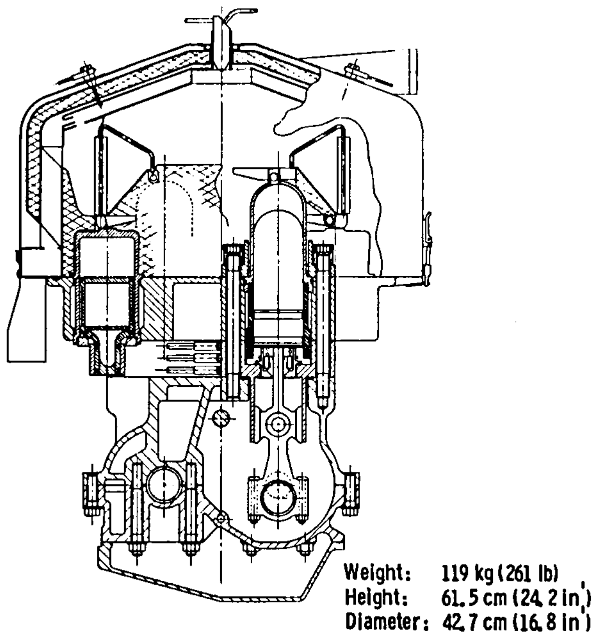

Figure 5. - Cross-section of 37 kW (50 hp) downsized Stirling engine.

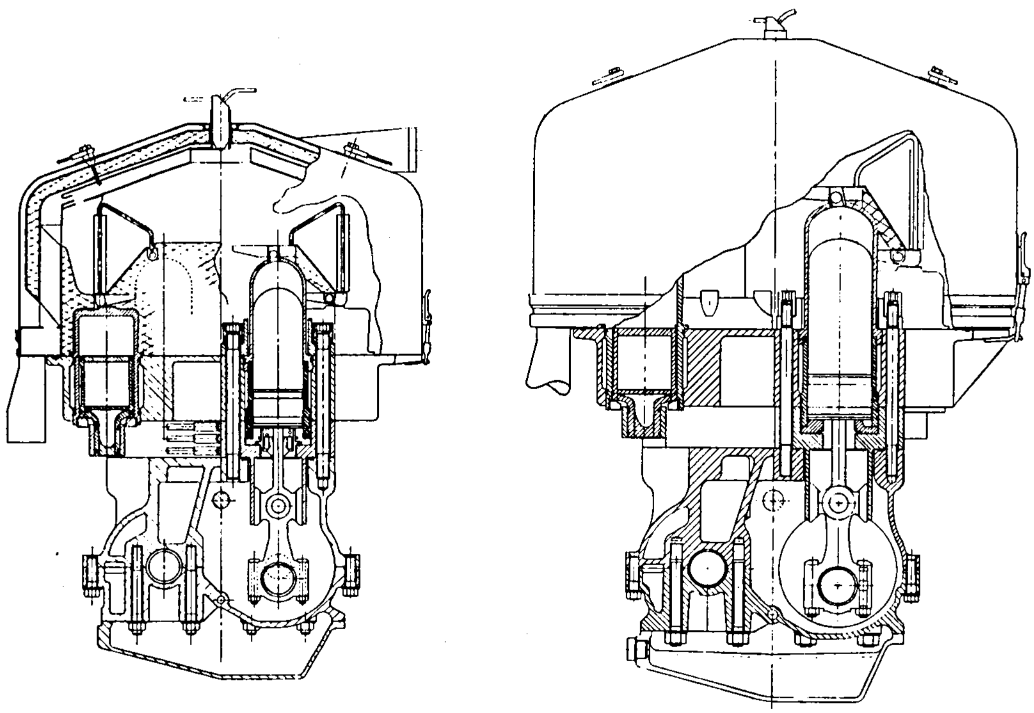

Figure 6. - Size comparison of downsized 37 kW (50 hp) with 66.7 kW (90 hp) reference engine.

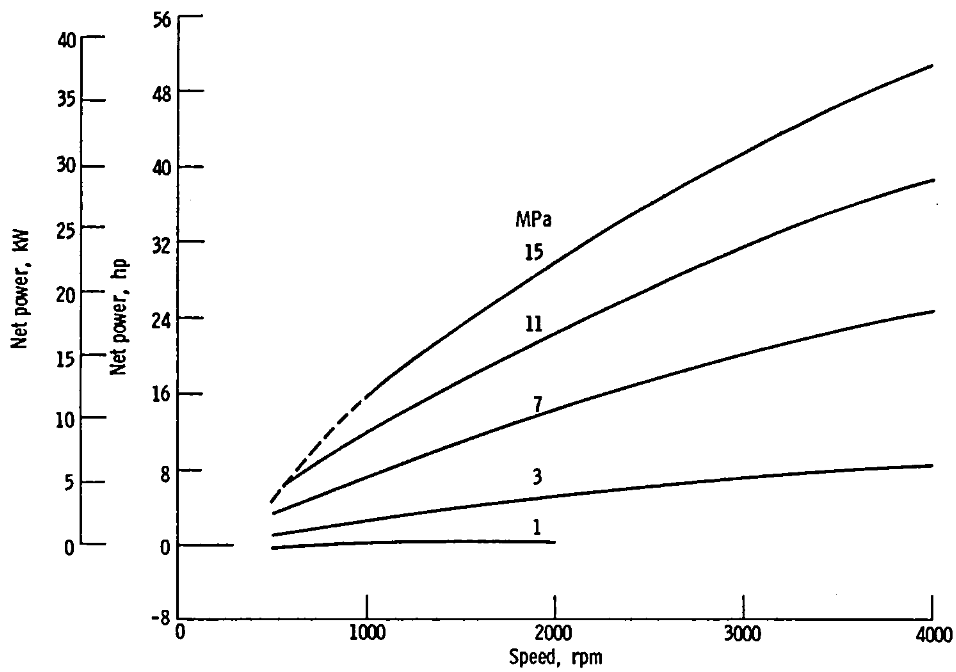

Figure 7. - 37 kW (50 hp) engine net power as a function of engine speed and mean working space pressure.

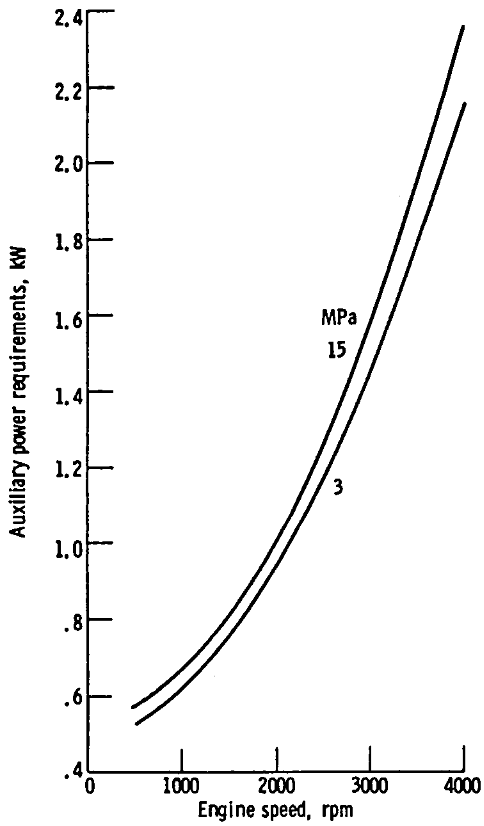

Figure 8. - Auxiliary power requirements as a function of engine speed and mean pressure for the 37 kW (50 hp) engine.

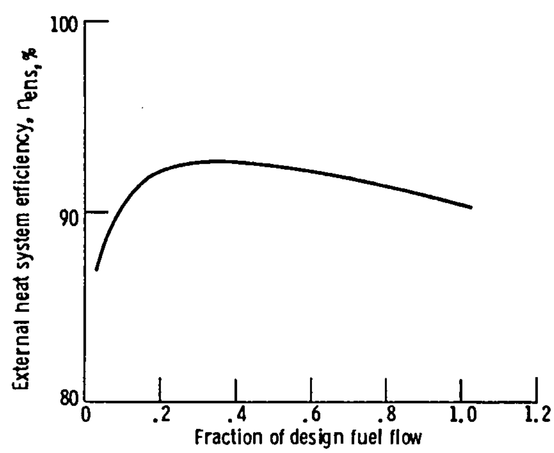

Figure 9. - External heat system efficiency as a function of fraction of design fuel flow assumed for 37 kW (50 hp) engine and reference engine.

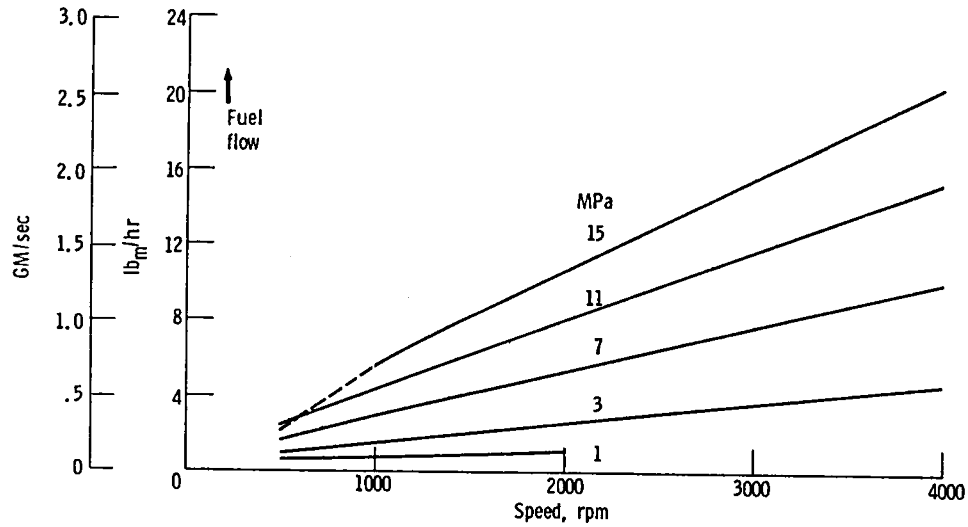

Figure 10. - 37 kW (50 hp) engine fuel flow as a function of engine speed and mean working space pressure.

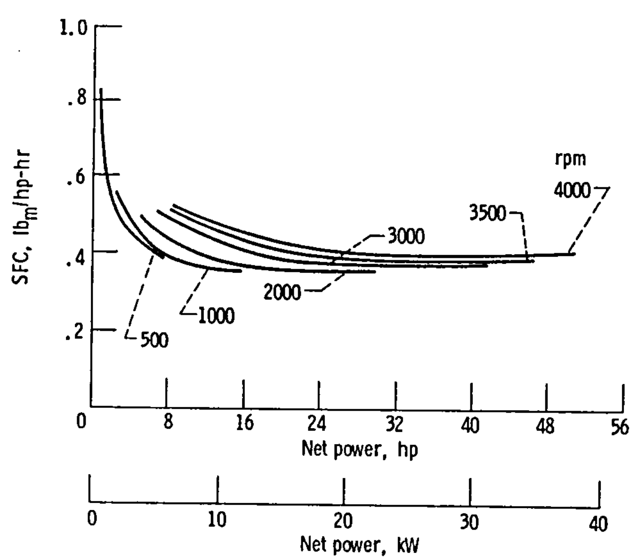

Figure 11. - Predicted specific fuel consumption (SFC) for downsized 37 kW (50 hp) engine as a function of net power and engine speed (includes mechanical and auxiliary losses).

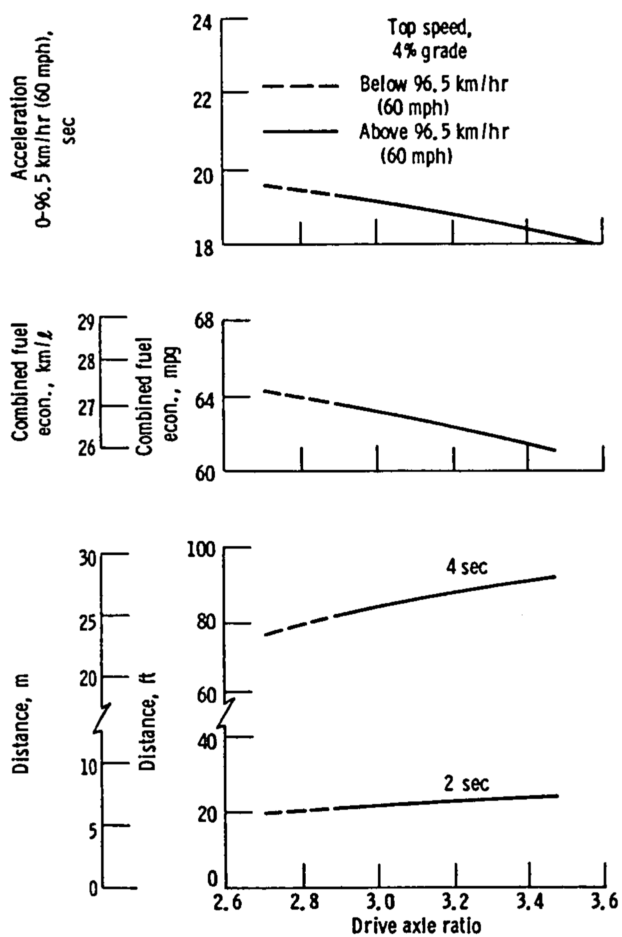

Figure 12. - Effect of axle ratio on acceleration and fuel economy for downsized stirling with 4-speed transmission in 964 kg (2125 lb) vehicle.

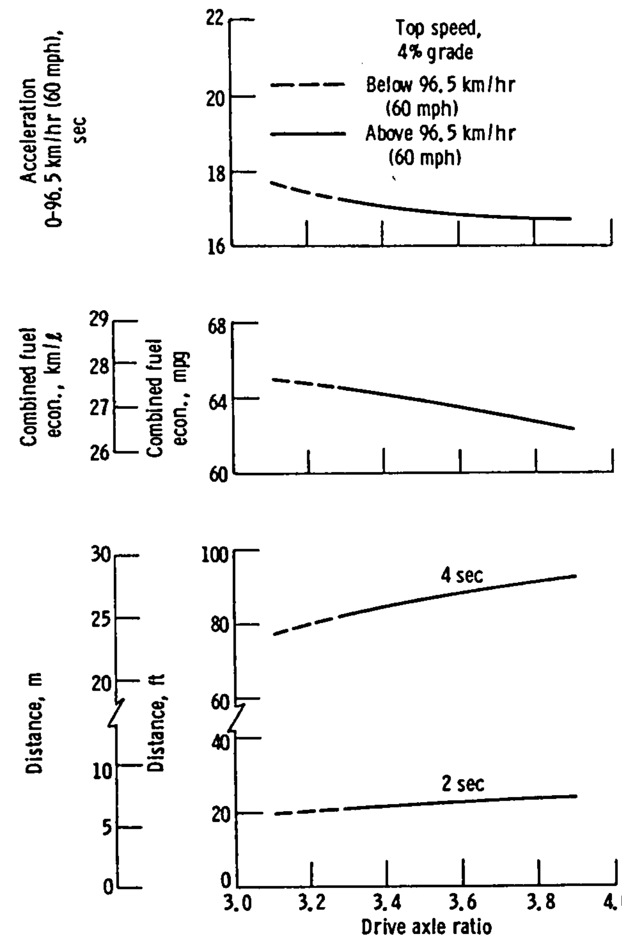

Figure 13. - Effect of axle ratio on acceleration and fuel economy for downsized Stirling with 5-speed transmission in 964 kg (2125 lb) vehicle.

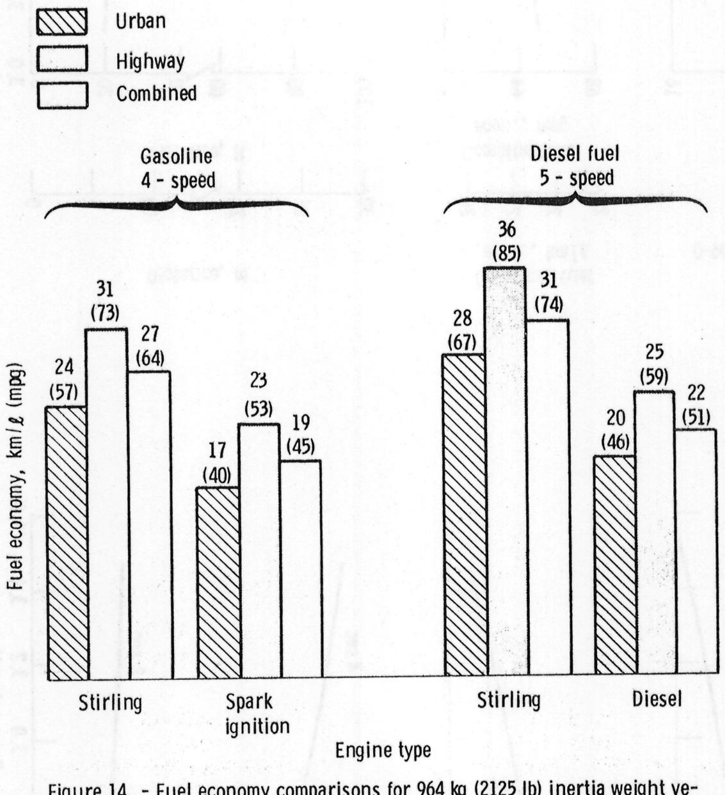

Figure 14. - Fuel economy comparisons for 964 kg (2125 lb) inertia weight vehicles with 37 kW (50 hp) engines and manual transmissions - vehicle C_D = 0.39.

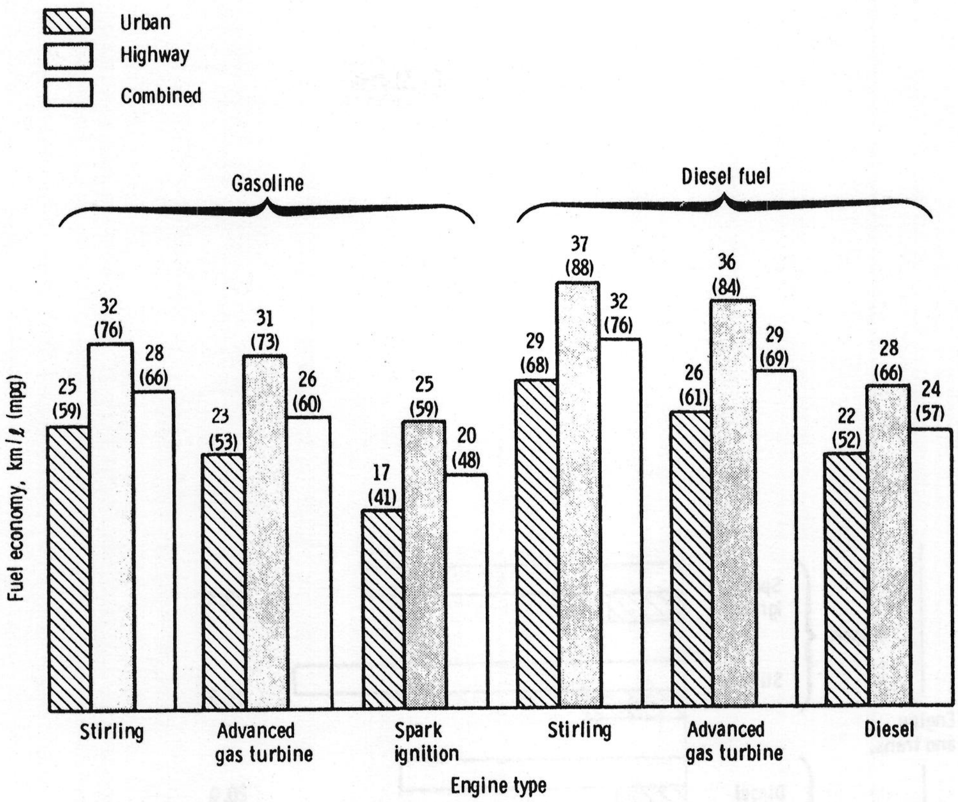

Figure 15. - Fuel economy comparisons for 964 kg (2125 lb) inertia weight vehicles with 37 kW (50 hp) engines and metal belt CVT's - vehicle C_D = 0.39.

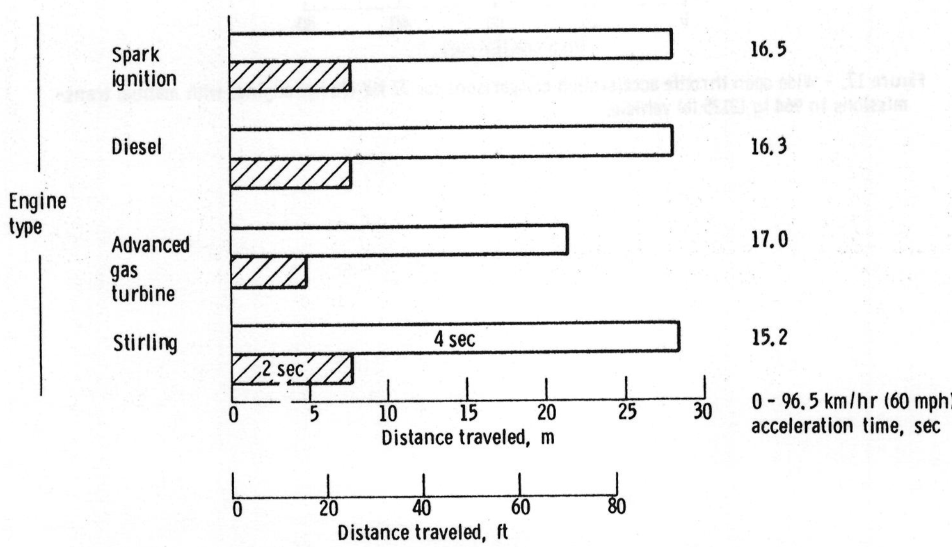

Figure 16. - Wide-open throttle acceleration comparisons for 37 kW (50 hp) engines with metal-belt CVT's in 964 kg (2125 lb) vehicle.

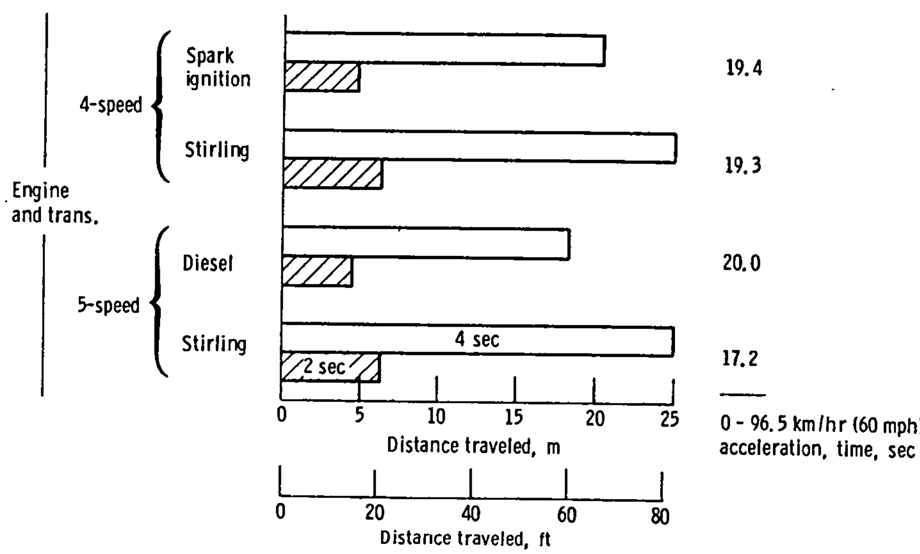

Figure 17. - Wide open throttle acceleration comparisons for 37 kW (50 hp) engines with manual transmissions in 964 kg (2125 lb) vehicle.

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16. Abstract

A 67 kW (90 hp) Stirling engine design, sized for use in a 1984 1440 kg (3170 lb) automobile has been serving as the focal point for developing automotive Stirling engine technology under a current DOE/NASA R&D program. Since recent trends are towards lighter vehicles, an assessment was made of the applicability of the Stirling technology being developed for smaller, lower power engines. Using both the Philips scaling laws and a Lewis Research Center (Lewis) Stirling engine performance code, dimensional and performance characteristics were determined for a 26 kW (35 hp) and a 37 kW (50 hp) engine for use in a nominal 907 kg (2000 lb) vehicle. Key engine elements were sized and stressed and mechanical layouts were made to ensure mechanical fit and integrity of the engines. Fuel economy estimates indicated that the Stirling engine would maintain a 30 to 45 percent fuel economy advantage over comparable spark ignition and diesel powered vehicles in the 1984 time period. In order to maintain the performance advantage, particular attention must be paid to the Stirling engine mechanical losses and, although not evaluated in this report, the cold start penalties.

17. Key Words (Suggested by Author(s)) Stirling engine; Heat engine; Automotive propulsion	18. Distribution Statement Unclassified - unlimited; STAR Category 85; DOE Category UC-96

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