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A Reciprocating Solar Heated Engine Utilizing Direct Absorption
by Small Particles

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ABSTRACT

A new type of reciprocating solar engine utilizing small particles to absorb concentrated sunlight directly within the cylinders is described. The engine operates by drawing an air particle mixture into the cylinder, compressing the mixture, opening an optical valve to allow concentrated sunlight to enter through a window in the top of the cylinder head, absorbing the solar flux with the particles, and converting the heat trapped by the air-particle mixture into mechanical energy with the downward stroke of piston. It differs from other gas driven heat engines using solar energy in three main respects. First, the radiant flux is deposited directly in the working fluid inside the cylinder; second, the heat is directed to the appropriate cylinder by controlling the solar flux by an optical system; third, the gas is heated during a significant portion of the compression stroke. The thermodynamic efficiency of the engine is calculated using an analytical model and is compared to several other engine cycles of interest.

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Introduction

Solar energy is being considered as a serious alternative to fossil fuels as an energy source to operate heat engines. A significant international program is underway to develop large scale solar thermal power plants capable of providing electricity to large segments of the population. A parallel effort exists to develop small systems capable of delivering on the order of 10 Kw of mechanical power per module by use of individual point focus collectors operating small Stirling or Brayton cycle engines.

A simple, reciprocating solar engine (SOLGIN) that uses direct absorption of sunlight to produce mechanical work is described here as an alternative to the small power systems mentioned above. The engine uses a suspension of ultrafine carbon particles within the cylinder to absorb concentrated solar radiation and heat the gas. The cycle begins when an air particle mixture is drawn into the cylinder and compressed. At some point during the compression stroke an optical valve directs concentrated sunlight through a transparent cylinder head into the gas particle mixture. The particles absorb the sunlight, and heat the gas. In the expansion stroke, the heated gas drives the piston and oxidizes the particles. The use of optical valving and an open cycle design leads to a new thermodynamic cycle that the authors have been unable to find described elsewhere in the literature.

The use of concentrated sunlight to operate a Stirling cycle engine was discussed by Daniels (1) and a design was proposed by Finkelstein and Daniels in 1960. A working model was described in 1962 (2). While these engines also use a transparent window to allow the sunlight to enter the cylinder, they differ in several important respects from the design discussed here. Because they operate with a Stirling cycle, heat is added to the engine continuously and not valved. The Stirling engine uses a closed cycle design so that the gas charge is sealed and the heat must be removed by an external cooling system. In the earlier design

the sunlight is focused onto a fine mesh screen absorber that is attached to the top of the displacer piston.

In current solar Stirling engine designs (3), concentrated sunlight is directed onto a metal or ceramic receiver that must transfer the heat to the working gas by conduction. The heated gas expands against a piston doing mechanical work. In addition to the working piston there is a transfer or displacer piston that moves the gas to and from the heating chamber. The expanded gas is passed through a recuperator to recover heat for the next charge.

There are several complicating factors that must be overcome to achieve a viable solar Stirling engine. These include the following: (1) the heat exchanger that transfers the solar energy to the gas, (2) the relative motion of the displacer piston and the working piston, (in free piston designs), (3) sealing the gas charge, and (4) recuperating heat from the expanded gas and supplying it to the next charge.

The SOLGIN concept has several advantages over Stirling engines. Because the particles are so small they do not heat significantly above gas temperature, thereby eliminating the need for a receiver that must survive very high temperatures. There is no displacer, every piston is a working piston. The engine operates with air in an open cycle, so that there is no need to permanently seal the gas charge. The working gas is expelled each cycle, reducing the cooling requirements. High engine efficiency is achieved without the need of a recuperator.

In the following sections the small particle heat exchanger is described briefly and the operation of the engine including the optical valving system is discussed. The next section contains an analysis of the thermodynamic efficiency of the engine and compares it to the familiar Otto and Diesel cycles. The solar collector section contains a discussion of the realistic constraints on the system and provides some examples of typical operating parameters.

The Heat Exchanger

An important aspect distinguishing the SOLGIN from other solar powered engines is its use of small particles as a heat exchanger. Small particles are suspended in a gas and are irradiated with solar energy. The particles absorb the solar energy and because of their very large surface area and small size, they quickly release this energy as heat to the surrounding gas. In order to maximize the effectiveness of the small particles as solar receivers, the optical properties of the particles must be such that they have a high absorptance in the solar spectrum. The particles should be extremely small, less than 0.1 micrometer in diameter. In particles this size the entire volume of the particle acts as the absorber. Because it is an open cycle, particles must be generated continuously during the cycle. Therefore, an efficient and convenient method for producing particles must be available. Furthermore, the particles and their oxides should be environmentally benign since they will be exhausted at the end of each cycle.

Carbon appears to be the ideal composition for the particles. Methods are available for producing the extremely small carbon particles; the optical properties of carbon are excellent for this application; and a number of forms of carbon exist which have different oxidation rates as a function of the temperature so that if the correct form of carbon is selected, the particles will oxidize during the heating process and be exhausted as carbon dioxide gas.

The effectiveness of a gas-particle mixture as a solar absorbing media has been discussed elsewhere (4,5). Of particular interest are (1) the mass of carbon particles necessary to produce a given absorption as a function of particle size, and (2) the optical efficiency of the particles as solar collectors. The mass loading per unit surface area necessary to produce an extinction of $1/e^2$ on the way from the front to the back of the chamber is only 14 micrograms per square centimeter. This mass of particles is very small compared to the mass of a combustion fuel to run the engine without sunlight (for the same power output). For particles less than 0.05 micrometers in diameter the collection efficiency can exceed 90% without the use of a cavity (6).

Description of the Engine

A complete cycle of the solar engine consists of four parts. (1) Near the bottom of the upward stroke of the piston, with no incoming solar flux, ambient air that has been injected with a small mass of fine particles enters the cylinder and is compressed. (2) At some point in the compression stroke the optical valve directs sunlight into the air-particle mixture. The small particles absorb the radiation and quickly release the heat to the surrounding gas. During the expansion stroke the particles begin to oxidize, turning into CO_2 . (3) At some point during the expansion stroke, the solar flux is diverted to the other cylinder and the heated air expands adiabatically against the piston producing mechanical work. (4) Near the bottom of the downward stroke of the piston the air-oxide mixture is exhausted, completing the cycle. The complete utilization of the solar resource requires the use of two or more cylinders. More than two cylinders may also be employed, with each utilizing solar radiation for an appropriate fraction of the total cycle time.

A schematic design for a two cylinder engine is shown in Figure 1. Each cylinder has a solar "window" which allows solar flux to enter. The window may be manufactured from one of several types of high silica glasses, and made appropriately thick to withstand the internal pressure. The "window" may be shaped like a lens to provide a more uniformly illuminated space inside the cylinder. The shape is convex inward to insure that it is in compression from the internal pressure in the chamber. The light from the solar concentrator is directed to the appropriate cylinder by the use of an optical "valve". There are many possible designs for this "valve". One particular design illustrated in Figure 2 utilizes a rotating element composed of conically shaped refractive elements. The design has the advantage of low losses (the surfaces can be treated or coated to significantly reduce the reflectivity) and the two light paths are symmetrical. Figure 3a and 3b illustrate the light path for two different positions of the valve. Note that light is always being directed at one or the other piston, never anywhere else.

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Figure 1. Cross section of a SOLGIN showing the major components.

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Figure 2. Illustration of an optical valve composed of two cone sections. If the valve is made from transparent material, it deflects the light by refraction. The valve may also operate by making the top surface reflecting.

Other optical valves may be designed using elements with the same shape as the top of the valve illustrated in Figure 2, but with reflective upper surfaces. In this case the light is reflected into the engine. The design can be expanded to four or more cylinders with additional light valves or by utilizing a different optical arrangement.

The piston moves in simple harmonic motion throughout the cycle, and heat is added at a constant rate during that time. The phase of the cycle at which the valve is opened will be considered in the next section. This method of adding heat produces a not-so-familiar thermodynamic cycle indicated by the solid line in Figure 4. The figure is not drawn to scale since the shape of the two segments of path 2 depends on the rate at which heat is added to the gas, and the rate of change of the volume during the time heat is added. In order to compare this thermodynamic cycle with more familiar cycles, consider two other ways in which heat may be added: (1) the gas is heated during such a short time when the piston is near top dead center such that the volume may be considered constant, (2) the piston is allowed to move as heat is added, keeping the gas at constant pressure. These two methods produce the familiar "Otto" and Diesel" cycles generally associated with internal combustion engines. These two cycles are illustrated with dotted lines in Figure 4.

Thermal Efficiency of the Engine

It is desirable to calculate the efficiency of the engine for different periods and phases of heat injection. The thermodynamic question can be phrased as follows: given a constant rate of heat input for a fraction of the cycle equal to the inverse of the number of cylinders, what is the optimum timing for the heat injection and what is the corresponding efficiency? In this section we will address this question for the case of a two-cylinder SOLGIN.

The thermal efficiency of the engine is defined as the ratio of the net mechanical work done by the engine to the heat supplied by solar radiation to the engine during one cycle. The work done by the engine is given by the area enclosed by the paths shown on a pressure-volume graph

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Figure 3. Optical path for two different positions of the optical valve.

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Figure 4. Approximate p-V diagrams comparing the Diesel and Otto cycles with a possible Solgin cycle in which neither pressure nor volume is held constant during the addition of heat. When drawn to scale for the same heat input the efficiency of each cycle is proportional to the area enclosed.

in Figure 5 and depends upon the point in the compression stroke at which the addition of heat is begun. For a two-cylinder engine, the addition of heat may begin at any volume in the compression stroke. To find an expression for the efficiency as a function of this volume, assume that the addition of heat begins at a increment of time δ after the mid-point of the compression stroke when the volume has been reduced by an increment, x . Once an expression for the efficiency as a function of x has been obtained, the value of x for which the efficiency is a maximum can be determined.

The determination of the work and thus the efficiency of the cycle when heat is added while neither volume nor pressure is held constant requires knowledge of the equation of path 2 of Figure 5. To simplify the derivation, it is convenient to divide path 2 into two segments. The first segment begins at the end of the adiabatic compression of the air to a volume, $V_2 - x$, where V_2 is the volume at the midpoint of the compression or expansion stroke, and x is the volume change. The optical valve is opened and the air-particle mixture absorbs heat at a constant rate, H . The volume decreases to its minimum value, V_3 , and maximum pressure, p_3 . The second segment begins as the volume begins to increase and ends as the volume reaches $V_2 + x$, and the solar heating stopped. In order to derive an equation for segment 1 and segment 2, write the first law of thermodynamics and the ideal gas law in differential form assuming that the state variables are functions of the time, t . From the first Law:

$$nC_v \frac{dT}{dt} = \frac{dQ}{dt} - p \frac{dV}{dt}, \quad (1)$$

and from the ideal Gas Law:

$$p \frac{dV}{dt} + V \frac{dp}{dt} = nR \frac{dT}{dt}. \quad (2)$$

Eliminate the derivative, dT/dt , between the two equations to obtain the differential equation:

$$\frac{dp}{dt} + \left\{ \frac{k}{V} \frac{dV}{dt} \right\} p = \frac{RH}{VC_v}, \quad (3)$$

where the rate at which heat is absorbed by the gas, dQ/dt , is replaced by the constant, H , and k is the ratio of the heat capacity at constant pressure to the heat capacity at constant volume. For air, k is about 1.4. The general solution of this differential equation can be written:

$$pV^k = \frac{RH}{C_v} \int V^{k-1} dt + C. \quad (4)$$

Since this equation is used in calculating the work done, it is convenient to evaluate the constant of integration, C , at the point p_1, V_1 for segment 1 and p_3, V_3 for segment 2. A reasonably good approximation for the integral in the above equation that yields an upper bound for the efficiency can be obtained by assuming the function, V^{k-1} is a linear function of the time with a decreasing slope, m_1 , for segment 1, and an increasing slope, m_2 , for segment 2, where

$$m_1 = (V_3^{k-1} - \{V_2-x\}^{k-1})/(\delta t_1) \quad (5a)$$

$$m_2 = (\{V_2+x\}^{k-1} - V_3^{k-1})/(\delta t_2) \quad (5b)$$

and δt_1 is the time interval that heat is allowed to enter the gas during segment 1, and δt_2 is the time interval that heat is allowed to enter the gas during segment 2. For this assumption, the equation for segment 1 can be written:

$$pV^k = \frac{RH\delta t_1}{2C_v} \frac{V^{2k-2} - (V_2-x)^{2k-2}}{V_3^{k-1} - (V_2-x)^{k-1}} + p_1 V_1^k \quad (6)$$

and the equation for segment 2 can be written:

$$pV^k = - \frac{RH\delta t_2}{2C_v} \frac{V^{2k-2} - V_3^{2k-2}}{V_3^{k-1} - (V_2+x)^{k-1}} + p_3 V_3^k \quad (7)$$

where the total heat added to the gas, Q , is equal to $H(\delta t_1 + \delta t_2)/N$. The work done over the cycle is given by:

$$W = \int_{v_1(ad)}^{v_2-x} p dV + \int_{v_2-x(seg\ 1)}^{v_3} p dV + \int_{v_3(seg\ 2)}^{v_2+x} p dV + \int_{v_2+x(ad)}^{v_1} p dV. \quad (8)$$

After carrying out the integrations and simplifying the results, the efficiency can be written as:

$$\eta = 1 - \left[\frac{v_2+x}{v_1} \right]^{k-1} + \frac{1}{2} \left[1 - \left[\frac{v_3}{v_2+x} \right]^{k-1} \right] + \frac{1}{4} \left[1 - \frac{N}{n} \sin^{-1} \left(\frac{y}{8} \right) \right] \left[1 - \left[\frac{v_2-x}{v_2+x} \right]^{k-1} \right], \quad (9)$$

where the inverse sine functions are a result of the change of variable from a time interval to a volume interval, x , as shown in Figure 6, N is the number of cylinders, and it was convenient to define a dimensionless variable, $y = 8x/(v_2 - v_3)$. The timing of the heat injection is given by y , and is chosen in equal increments from 0 to 8. When $y = 0$, the addition of heat begins at the mid-point of the compression stroke (a phase angle of 90 degrees) and when $y = 8$, the addition of heat begins at the maximum compression (a phase angle of 180 degrees).

Actual values of the efficiency of the two-cylinder engine as a function of y for several different compression ratios are shown in Table I. Values calculated from Eq. 9 are displayed graphically in Figure 7. As can be seen, the efficiency is a strong function of both the timing and the compression ratio. The time delay (or phase angle) which produces the maximum efficiency appears to be very nearly independent of the compression ratio, however. Although it is evident from the graphs of Figure 7, a careful analysis of the equation for the efficiency as a function of y confirms that it has a maximum at $y = 0$ for a two-cylinder engine for each compression ratio. The function is not changing rapidly in this region so that pin-pointing the y -value any closer is not critical. However, it is clearly important to begin the addition of heat

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Figure 5. The p-V diagram for a two-cylinder engine. The optical valve directs the solar flux into the cylinder beginning at a volume of V_2-x in the compression stroke and continuing until the volume has reached V_2+x on the expansion stroke.

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Figure 6. The volume of one cylinder as a function of time. The addition of heat begins at $T/4+\delta$, where T is the period of the cycle.

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Figure 7. The per cent efficiency of the two-cylinder engine as a function of the delay of heat input. The variable y divides the volume between V_2 and V_3 into 8 increments. When $y = 0$, the heat is added beginning at V_2 and when $y = 8$, the heat is added beginning at V_3 . (r is defined as the compression ratio V_1/V_3)

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Figure 8. Parabolic solar collector with secondary reflecting surface. Secondary reflecting surface should be placed so that light beam focuses near center of compression chamber.

well before maximum compression. The addition of heat beginning at a phase angle of 90-95 degrees should produce an efficiency reasonably close to the maximum.

It can be shown that the expression given in Equation 9 gives efficiencies slightly above the exact value. Numerical integrations were carried out to better determine the efficiencies for several cases. The results of the numerical integration for the case $\gamma = 0$ and compression ratio of 8 yielded an efficiency of 41.2% or 2% lower than the value obtained from Equation 9. The actual efficiencies are shown in Table I and are generally 1.0% to 3.0% lower than those given by Equation 9.

In order to see what engine parameters correspond to the calculated efficiencies, some operating conditions must be specified. For example, assume that the engine consists of two cylinders, design considerations dictate a maximum allowable pressure in the cylinder of about 20 atmospheres, the available heat and particle selection produces a maximum temperature of 1000°K , and the concentrating collector provides 200 Joules of heat to the air per cycle. After an initial pressure, volume and temperature are specified, eg. 1 atmosphere, 1 liter and 300°K , respectively, the thermodynamic parameters can be determined by calculating the successive states around the loop. For this example the maximum possible compression ratio is about 8:1. A larger compression ratio increases the efficiency of the cycle, but the maximum pressure would then exceed the limit.

Further analysis indicates that the efficiencies can be significantly increased by increasing the number of cylinders, all else being equal, and the efficiency of an eight-cylinder engine is in excess of 58% for a compression ratio of 8:1. The frictional losses increase with the number of cylinders, but if a practical system of optical valving can be devised, the engine would have a excellent over-all efficiency.

For comparison, the efficiencies of both the "Otto" and the "Diesel" cycle can be calculated from the same conditions and assumptions as above. Their efficiencies are well known and can be found in most textbooks on thermodynamics or heat engines. For the "Otto" cycle the efficiency is given by:

$$1 - \frac{T_1}{T_2} = 1 - \left[\frac{V_2}{V_1} \right]^{k-1}$$

where the temperature ratio is the ratio of the initial temperature of the gas to the temperature after the adiabatic compression. For the "Diesel" cycle, the efficiency is given by:

$$1 - \frac{(T_4 - T_1)}{k(T_3 - T_2)},$$

and the subscripts 1, 2, 3, and 4 refer to the initial temperature, the temperature after the adiabatic compression, the highest temperature of the cycle, and the exhaust temperature respectively. For a maximum allowable pressure of 20 atm and heat input of 200 J, the efficiencies of the "Otto" and "Diesel" cycles are 54% and 56%, respectively. However, actual efficiencies attained by internal combustion engines are much smaller than those predicted, primarily due to the variable specific heats, dissociation, and heat loss. For example, an Otto cycle burning iso-octane and a Diesel cycle burning liquid dodecane, both using fuel/air mixtures chosen for maximum economy, have actual efficiencies of about 40% (7). For the SOLGIN cycle, the mass of particles introduced into the chamber is very small compared to the mass of a combustion fuel, and furthermore, the operating temperature is lower. Thus, the reduction in efficiency of the SOLGIN due to the introduction of fuel and due to heat loss should be much less than for the internal combustion engines. It is conceivable that under actual operating conditions, the Otto, Diesel, and SOLGIN cycles have very nearly equal efficiencies.

Because of the differences in operating characteristics between SOLGIN and the Stirling or Brayton cycle, it is less easy to compare these engines for equivalent operating parameters. For instance, the Brayton cycle may incorporate regeneration, reheat or intercooling systems to increase the efficiency of ideal cycles. The Stirling engine generally uses high static pressure to obtain good efficiency, giving much higher pressures than in SOLGIN. In both the Stirling and Brayton cycles the source and sink temperatures may be specified to calculate the

efficiency. If the engines are compared by setting the maximum temperatures occurring in the cycle to the same value, SOLGIN will generally have a comparable efficiency.

Table I

Efficiencies of a two cylinder SOLGIN determined by exact numerical integration as a function of the phase of heat injection, y , and compression ratio, r .

y	0	1	2	3	4	5	6	7	8
r=6	.380	.379	.375	.369	.360	.347	.330	.301	.223
r=8	.412	.411	.407	.400	.391	.376	.358	.327	.241
r=10	.433	.432	.428	.421	.411	.392	.377	.344	.253
r=12	.449	.447	.444	.436	.426	.410	.391	.357	.261

The Solar Collector

An important consideration when designing the solar engine is the availability of concentrated solar flux from the collector. For a small engine such as discussed here, the best choice for a collector is probably a parabolic dish because it provides the high concentration of solar flux necessary to operate the engine efficiently. One manufacturer (8) advertises a collector 6 meters in diameter which supplies over 20 kW of power in a 6 centimeter circle at the focus of the parabolic dish. An engine may be mounted at the focus and move with the collector as it tracks the sun.

As an alternative, the engine can be mounted near the vertex of the parabola rather than at the focus. A secondary reflecting surface is employed to direct the focussed beam from the primary surface back toward the vertex of the parabola as shown in Figure 8. There is a loss of solar power due to the reflectivity of the secondary surface and by shading of the primary surface by the secondary surface, but for some engine designs there may be an advantage in not moving the engine with the collector. A heavier engine with a large number of cylinders is a reasonable design option for this configuration.

Whatever the design of the collector, the amount of flux available and the area over which it is directed should be determined. The size of the engine's solar window should be large enough to transmit as much as possible of this flux to the compressed gas in the cylinder. When the available solar power into the cylinder is known accounting for any losses in the optical valving and through the solar window, the total heat, Q , available to the compressed gas can be calculated from:

$$Q = \text{Solar power} \times (\text{engine period/number of cylinders}).$$

Once the heat, Q , is known, a reasonable approximation for the increase in temperature of the gas produced by this addition of heat is given by:

$$\Delta(T) = Q/nC,$$

where n is the number of moles of gas in the cylinder and C is its molar heat capacity. The value of C depends upon the conditions under which heat is added to the gas and can be determined by experiment for the conditions of operation of the engine. The heat capacity of an ideal gas is approximately 21 Joules/mole-degree K for heat added at constant volume and approximately 30 Joules/mole-degree K for heat added at constant pressure. Since under these conditions air behaves approximately as an ideal gas, the experimental values of the heat capacity will probably not differ greatly from those of an ideal gas.

As an example, assume that the available solar power into a cylinder is 16 kW after accounting for losses in the secondary reflecting surface, in the optical valving, and in the optical receiver. If an eight-cylinder engine is operated at a speed of 1200 rpm, the heat into the gas during one cycle is 100 Joules per cylinder. If the molar heat capacity of the gas is 25 J/mole degree K, a temperature increase of 400 degrees K can be achieved by using about 1/100 of a mole of gas per cylinder. At standard conditions, 1/100 of a mole of air occupies a volume of 224 cc. The cylinder displacement should then be approximately 224 cc if a temperature of 400 degrees K is desirable. If a higher temperature difference is preferable, then the engine may be operated at a slower speed and/or the initial operating volume of the air may be reduced.

Summary

A new type of reciprocating piston engine using concentrated solar radiation as an energy source and utilizing small particles as the heat exchanger is presented. Carbon particles with diameters of one-tenth of a micrometer or less are suggested as the ideal particles for the heat exchanger because of their favorable optical, chemical and physical properties. The thermal efficiency of the engine depends upon the operating conditions, but for the alternatives discussed here, the ideal efficiencies for two cylinder engines exceeds 50%, and for a larger number of cylinders is in excess of 60%.

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