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SOLAR

SURVEY OF MANUFACTURERS OF HIGH-PERFORMANCE HEAT ENGINES ADAPTABLE TO SOLAR APPLICATIONS

By
W. B. Stine

June 15, 1984

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Jet Propulsion Laboratory
Pasadena, California

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Survey of Manufacturers of High-Performance Heat Engines Adaptable to Solar Applications

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by
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ABSTRACT

This report summarizes the results of an industry survey made during the summer of 1983. The survey was initiated in order to develop an information base on advanced engines that could be used in the solar thermal dish-electric program. Questionnaires inviting responses were sent to 39 companies known to manufacture or integrate externally heated engines. Follow-up telephone communication ensured uniformity of response.

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[1]U. S. Department of Energy and American Society of Engineering Education.

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SECTION I

ENGINE CONSIDERATIONS

There are a number of important reasons why the selection of an engine for solar application is different from engine selection for most other applications. Two important ones are:

(1) High engine thermal efficiency is a primary consideration, overriding most others.

(2) There is an optimum operating temperature for any given engine/collector combination.

A. THERMAL EFFICIENCY

The engine requirement for high thermal efficiency can be understood by considering the formula for calculating the overall cost of building a solar dish power system. This cost per unit power output capability can be expressed as the sum of the collector cost, the engine cost, and the present value of the yearly operating and maintenance (O&M) costs. The collector cost per unit area $(C/A)_c$ includes the cost of the concentrator, receiver, and that portion of the installation cost that is dependent on the total field (aperture) area (land, facilities, etc.).

$$\frac{C_t}{P} = \frac{1000\ (C/A)_c}{I\eta_e\eta_s} + \frac{C_e}{P} + \frac{Y_{O\&M}}{P}\left[\frac{1-(1+r)^{-n}}{r}\right] \qquad (1)$$

where:

C_t = total cost of system, \$

$(C/A)_c$ = cost of collector per unit aperture area, \$/m^2

C_e = cost of engine, \$

I = insolation, W/m^2

n = lifetime of the system, yr

P = power (or electrical) output of the module or plant, kW

r = the annual investment interest rate

$Y_{O\&M}$ = yearly O&M rate, \$/yr

η_s = efficiency of collector

η_e = efficiency of engine

Note that the engine efficiency appears in the collector cost term implying that a smaller collector can be used with an engine of higher efficiency.

As an example, using values typical of currently envisioned parabolic dish solar power systems, the cost of the system per unit power output capability would be on the order of

$$\frac{C_t}{P} = \frac{1000 * 200}{1000 * 0.30 * 0.80} + \frac{5000}{25} + \frac{300}{25}\left[\frac{1 - (1 + 0.12)^{-20}}{0.12}\right]$$

$$= 833 + 200 + 90 = 1123 \text{ \$/kW}$$

Because the cost of the collector is about 75% of the total system cost and is inversely proportional to the engine efficiency, small improvements in engine efficiency will have a significant impact on the total cost of the system. In the case above, for example, improving the engine efficiency from 30 to 31% reduces the cost of the system by 2.4%. At reduced levels of insolation or on a daily or yearly average, this effect is even greater.

B. OPTIMUM OPERATING TEMPERATURE

One way to increase the efficiency of an engine is to increase its maximum operating temperature, T_H. A simple description of the temperature dependence of engine efficiency can be expressed as:

$$\eta_e = K\left[1 - \left(\frac{T_L}{T_H}\right)\right] \qquad (2)$$

where T_L is the temperature at which heat is rejected, and K is an engine-specific constant representing the fraction of Carnot efficiency developed. Although K is assumed to be independent of temperature in this development, it may not necessarily be constant for a particular engine operated at different off-design temperatures. However, when one looks at a wide variety of engines, each operating at its design temperature, the percentage of Carnot efficiency attained does not seem to vary with operating temperature. The current maximum value appears to be in the 60 to 70% range for a well designed engine.

The energy collection efficiency of the collector decreases, due to increasing receiver heat loss, as the receiver temperature, T_r, increases. The energy collection efficiency, η_s, of a solar concentrator may be defined in terms of a receiver heat balance as:

$$\eta_s \equiv \frac{\dot{q}_u}{IA_c} = \phi\rho\tau\alpha - \frac{U_L\ (T_r - T_a)}{CR \cdot I} - \frac{\varepsilon\sigma\ (T_r^4 - T_a^4)}{CR \cdot I} \qquad (3)$$

where:

A_c = area of collector aperture, m^2

A_r = area of receiver aperture, m^2

CR = geometric concentration ratio = A_c/A_r

I = insolation, W/m^2

$\dot{q}_u$ = rate of useful heat added, W

T_r = receiver operating temperature, °K

T_a = ambient temperature, °K

U_L = receiver overall[2] heat loss coefficient, W/m^2°K

α = absorptance (effective) of receiver aperture

ε = emittance (effective) of receiver aperture

ρ = reflectance of concentrator

σ = Stefan-Boltzmann constant, W/m^2°K^4

τ = transmittance of any intermediate cover

ϕ = receiver intercept factor

Neglecting collector parasitics (which are usually 1 or 2% in the usual installation), the overall efficiency of a solar power system is the product of the efficiency of the engine and the efficiency of the solar collector. Because engine operating temperature approximately equals the receiver temperature, a combination of Equations (2) and (3) gives an optimum operating temperature where the system efficiency is maximized. Assuming that the engine can reject heat to ambient temperature ($T_L = T_a$) and it receives heat at receiver temperature ($T_H = T_r$), it can be shown that:

$$\theta_{max}^5 - \frac{3}{4}\theta_{max}^4 + \frac{C_2}{4C_3}\theta_{max}^2 = \frac{C_1 + C_2 + C_3}{4C_3} \tag{4}$$

[2]Convective loss from the cavity in addition to conductive loss away from the cavity.

where the parameters below were defined for simplicity:

$$\theta_{max} = T_{r_{max}}/T_a$$

$$C_1 = \phi\rho\tau\alpha$$

$$C_2 = U_L \, T_a/CR \cdot I$$

$$C_3 = \varepsilon\sigma \, T_a^4/CR \cdot I$$

and the temperatures are in absolute temperature units. Note that the percentage of Carnot efficiency term K from Equation (2) does not appear as a parameter in this expression for optimum operating temperature. This means that the optimum operating temperature for a collector/engine depends primarily on the collector. Achievement of optimum efficiency at this temperature requires the selection of an engine that achieves peak values of K in the vicinity of the optimum operating temperature found from Equation (4). The derivation of this equation is given in Appendix A.

As an example, for the concentrator characteristics given in Table 1-1, the optimum engine operating temperature can be found as a function of concentration ratio. This relationship is displayed in Figure 1-1. Because, for a given intercept factor, ϕ, the cost of a concentrator generally increases with concentration ratio, it can be seen that engines with low operating temperatures are better suited to low concentration ratio collectors. A "bottom-line" cost evaluation of the energy produced often favors these lower cost combinations.

In the design of solar power systems, there is considerable latitude in engine/collector selection because the optimum operating point is not a sharp peak. Figure 1-2 shows the combined collector/engine efficiency for a collector having a concentration ratio of 1000. Although the optimum operating temperature is 780°C, a decrease in output of less than 2% would be experienced if the operating temperature was increased or decreased by 100°C.

Table 1-1. Nominal Collector Case Parameters

Parameter	Nominal Value
I	1000 W/m^2
T_a	298/°K
U_L	60 W/m^2°K (open cavity)
$\phi\rho\tau\alpha$	0.90
ε	0.90

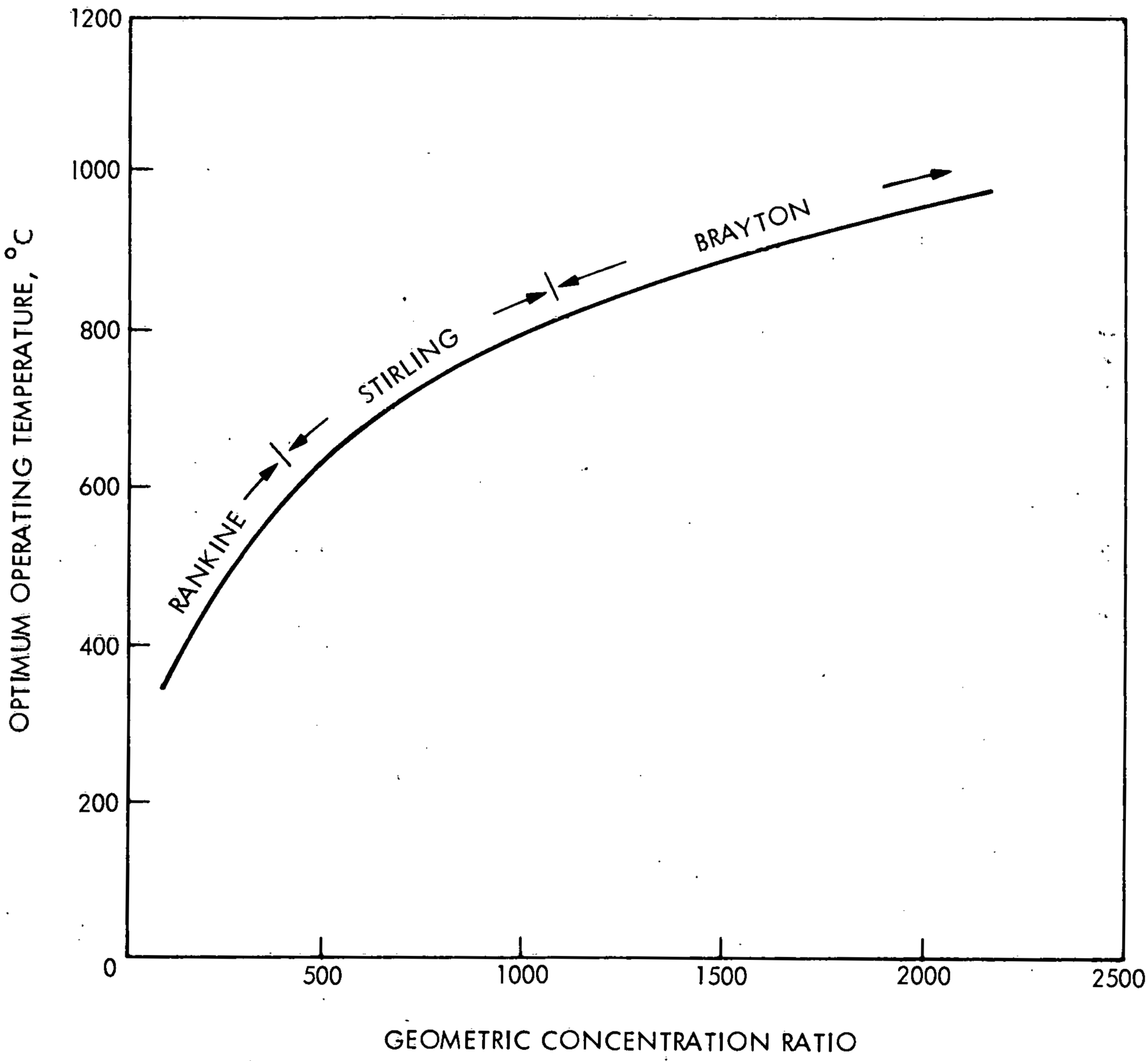

Figure 1-1. Optimum Operating Temperature for the Collector Defined in Table 1-1 Operating in Combination with an Engine

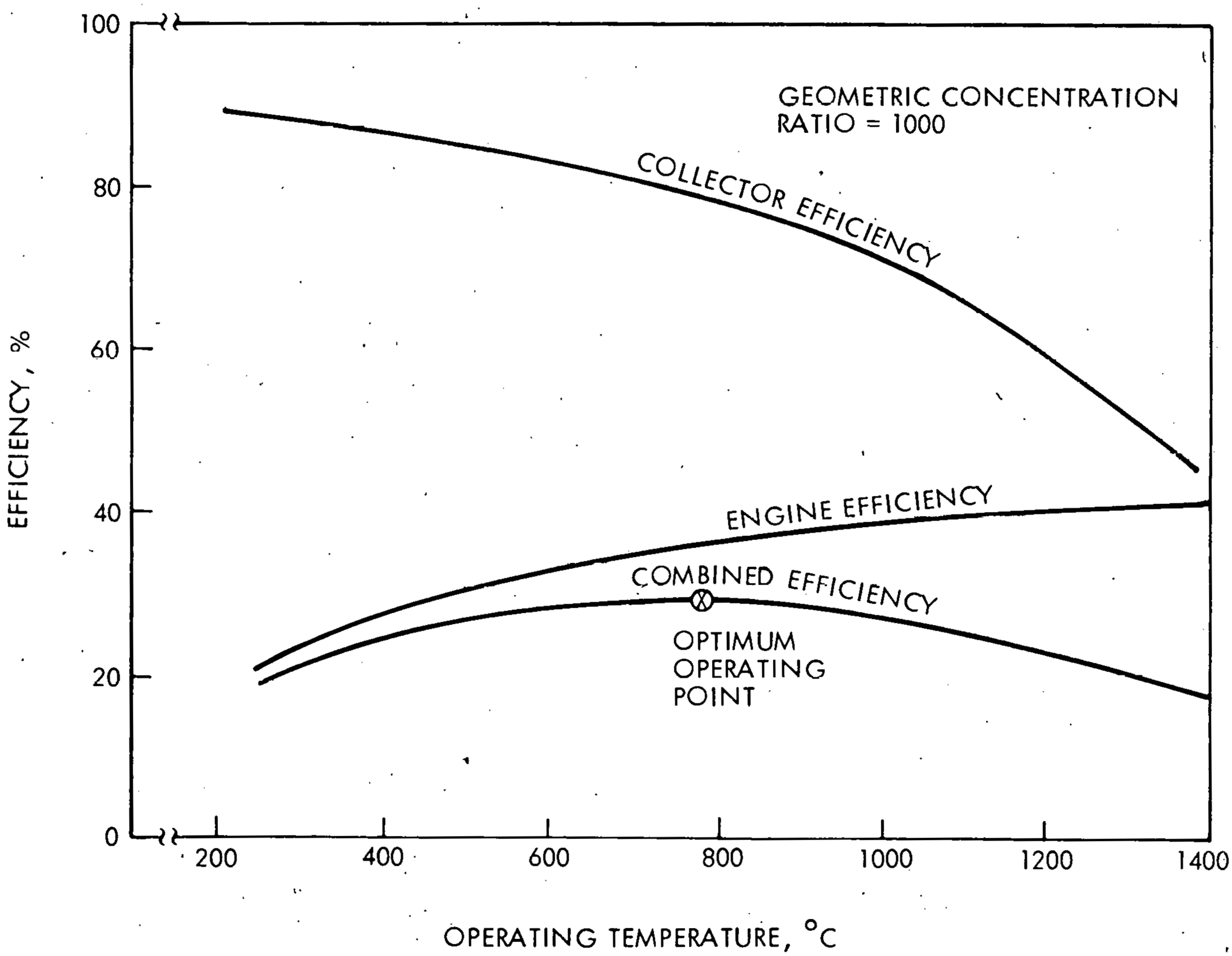

Figure 1-2. Collector and Engine Efficiency Variation with Operating Temperature for a Collector Having a Geometric Concentration Ratio of 1000

C. OTHER CONSIDERATIONS

In addition to these two primary considerations there are other factors specific to solar applications that must be considered when selecting an engine for solar application. One of the most important is that the engine must be designed for external heat addition to the working fluid. This heating usually takes place in a solar receiver. This is more easily done with Rankine-, Brayton-, and Stirling-cycle engines than with Otto- or Diesel-cycle engines.

Because insolation varies during the operating period, the engine control system must be able to respond to variable heat input while maintaining a constant engine speed. For many proposed applications, both the heat in and the load will vary while a synchronous alternator maintains a constant engine speed. Special control system modifications are required to handle these variations for some engine designs.

The rated power output of engines currently of interest to the solar dish program is between 10 and 100 kW. This range is based on the size of the solar power dishes currently envisioned, i.e., between 5 and 20 m in diameter. The mass and volume of the engine should be small enough to keep the cost of engine supporting structure at a minimum and to minimize blockage of incoming solar radiation.

Finally, an engine used in solar power applications must have a long service life and low O&M costs. Engines used in solar applications should have lifetimes approximately ten times longer than automotive engines; the latter are typically designed for lifetimes of about 5000 hours with periodic maintenance every 500 hours.

SECTION II

SURVEY

In order to assess the development status of engines in the 10-to-100-kW power output range, survey letters were sent to 39 companies. These companies were chosen because of their known past involvement in designing externally heated engines in the appropriate power and efficiency range. The list of addressees given in Appendix B was compiled after discussions with JPL personnel involved with engine procurement. The individual noted as the contact was the person who responded to the survey. The engine data report and cover letter are included as Appendix C. The survey requests technical information on the engine cycle in addition to information on maintenance, production, and modification of the engine and on the manufacturer's capabilities. Because information was being sought on developmental projects, it was decided to emphasize responses to the technical data outlined on the first page of the attached data report form.

In the cover letter, it was noted that the information provided would only be used internally by the Jet Propulsion Laboratory (JPL) in order to protect those companies responding with proprietary data. However, as responses were received, it became apparent that most of the information was for general distribution. Because of the transition in management of the parabolic dish program from JPL to Sandia National Laboratories-Albuquerque, each respondent was asked if their data could be made generally available; affirmative replies were received in all cases.

SECTION III

RESPONSE SUMMARY

Most of the 39 companies which received a survey form have responded either with pertinent engine data or an indication that they are not involved in any appropriate projects. No response was assumed to mean either that no appropriate projects existed within the company or that information on engine development projects could not be made available. This was confirmed by telephone in most cases; a follow-up letter was sent to companies located outside the United States.

Brief descriptions of the engine development projects found in this survey are given below. Table 3-1 summarizes the findings in a format enabling comparison among engines. The two most important parameters for solar applications, cycle efficiency and peak operating temperature, are presented graphically in Figure 3-1.

A. BRAYTON CYCLES

1. Allison

The Allison Gas Turbine Operations Division of General Motors in Indianapolis, Indiana, is developing a small, high efficiency gas turbine, known as the AGT 100, for automotive applications. This program parallels the Garrett Turbine development of the AGT 101. The engine consists of a single-stage centrifugal compressor and a radial turbine with a rotary ceramic regenerator. At a design turbine inlet temperature of 1288°C (2350°F) the engine produces 75 kW of power at an efficiency of 43%. The second of these engines is currently being built.

2. Garrett AiResearch

The Garrett AiResearch Corporation in Torrance, California, produces a subatmospheric Brayton-cycle engine that has been converted for operation on a solar concentrator. The cycle is regenerative, with heat addition occurring at atmospheric pressure and heat rejection at approximately one-half an atmosphere. It is a closed cycle using air as the working fluid. When operating with a turbine inlet temperature of 871°C (1600°F), the engine will produce 11 kW of electricity at 27% thermal efficiency. These engines are also being proposed as part of a gas-fired heat pump system, and six have been built and tested to date.

3. Garrett Turbine

Garrett Turbine Engine Company in Phoenix, Arizona, is developing a solar version of their ceramic-component automotive Brayton-cycle engine (the AGT 101). It incorporates a centrifugal compressor and turbine and a ceramic rotary regenerator. Operating with a turbine inlet temperature of 1371°C (2500°F), the engine will produce 75 kW of shaft power at an efficiency of 47%.

A prototype engine using metallic rather than ceramic parts has been built and tested. The ceramic components are currently undergoing development testing.

4. Microturbo

Microturbo S. A. of Toulouse, France, manufactures a line of small gas turbine engines used for ground and aircraft auxiliary power units. A typical engine, the recuperated Gevandan 9, Model 2R, incorporates a centrifugal compressor, a radial turbine, and a cross-flow regenerator. Operating with a turbine inlet temperature of 750°C (1382°F), the engine will produce 80 kW of shaft output at an efficiency of 18%. Many of these engines have been made and incorporated into military applications.

5. Solar International (Turbomach)

Solar International of San Diego, California, builds a line of small gas turbine engines used for auxiliary power units and other power applications. Two of their engines, the Gemini and the Titan, are in the power range appropriate for solar dish applications. Both use centrifugal compressors and radial turbines and do not incorporate regeneration. The Gemini will produce a maximum power of 21 kW(shaft) operating at an efficiency of 8.7%. The Titan produces 67 kW of power at a similar thermal efficiency. Many of these engines have been built and are in service throughout the world.

B. RANKINE CYCLES

1. Barber-Nichols

The Barber-Nichols Engineering Company of Arvada, Colorado, has built an organic Rankine-cycle engine that has been tested on a solar concentrator. The engine consists of a centrifugal pump, a turbine, and a permanent magnet alternator on a single shaft. The working fluid is toluene, which reaches a maximum temperature of 400°C (752°F) when producing 20 kW of electricity at a thermal efficiency of 23%. Regeneration is included in the cycle, and heat is rejected to the atmosphere by a fan-cooled condenser.

2. Bertin et Cie

Bertin et Cie in Plaisir, France, has developed an organic Rankine cycle which uses Fluorinert as the working fluid. Operating at a maximum temperature of 250°C (482°F), the engine produces 50 kW of electrical power. Although no cycle efficiency information was given, the turbine operates at 112 Hz (6720 rev/min) with a turbine efficiency of 78%. This engine is in the prototype development stage for use in solar or waste heat recovery applications.

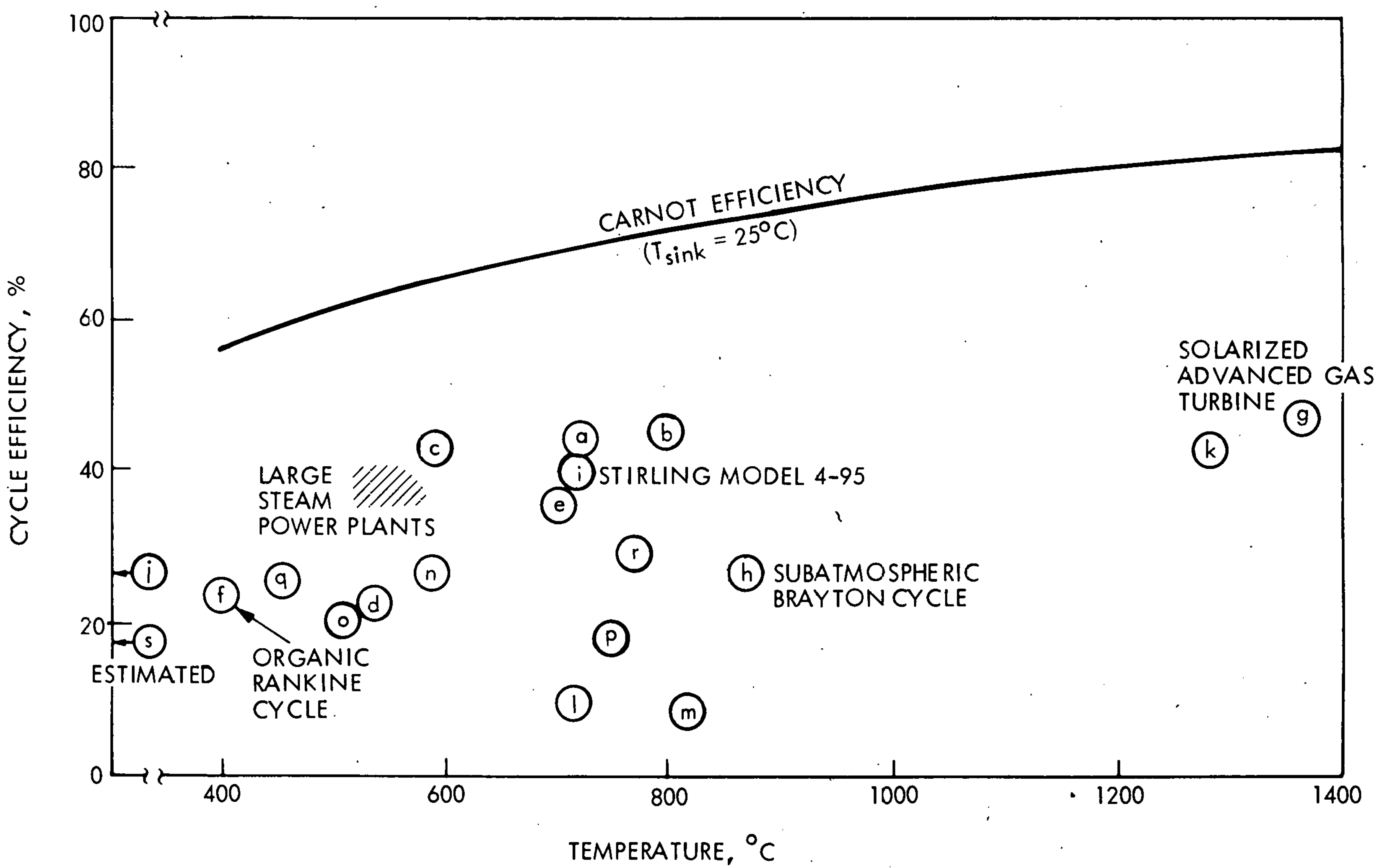

Figure 3-1. Comparison of Maximum Cycle Temperature and Engine Efficiency for Engines Noted in Table 3-1. (Circled letters correspond to those used for listing engine manufacturers in the table.)

Table 3-1. Summary of

	Company	Engine Designation	Cycle	Open/ Closed	Mechanics	Alternator	Output, kW	Efficiency Goal, %	Speed, Hz	T_{source}, °C	T_{sink}, °C
a)	Sunpower, Inc.	1982 D-2	Stirling	C	Free piston	Linear	10[a]	44	50	710	35
b)	Stirling Thermal Motors	STM4-120RH	Stirling	C	Variable angle swash plate	Opt.	40	48 (45 @ max)	46.7	800	45
c)	Bertin et Cie		Stirling	C	Free piston/elect. coupled displacer	Linear	3	43	50	590	70
d)	Foster-Miller	Simple Cycle	Steam Rankine	C	Reciprocating/crank	Opt.	22.4	22	30	538	100
e)	Foster-Miller	Compound Reheat	Steam Rankine	C	Reciprocating comp./reheat water actuated valves	Opt.	28.2	36	30	700	100
f)	Barber-Nichols Engineering Co.	ORC	Organic Rankine	C	Turbomachine	High-speed integral	20[a]	23	1000	400	50
g)	Garrett Turbine Engine Co.	SAGT Ceramic	Brayton	O	Turbomachine/ ceramic rotor	Gearbox/ belt	75	47	1450	1371	27
h)	AiResearch	SABC	Brayton	C	Turbomachine	High-speed mag coupled	11[a]	27	1183	871	16
i)	United Stirling AB	4-95	Stirling	C	Reciprocating/crank	Direct drive	24.4[a]	40	30	720	50
j)	United Tech Res. Center	Solar Chiller Drive Cycle	Rankine	C	Turbomachine	Opt.	18.7	24	717	149	45
k)	Allison	AGT 100	Brayton	O	Turbomachine	Opt.	75	43	1132	1288	16
l)	Solar International	Gemini	Brayton	O	Turbomachine	Opt.	21	9	1565	715	15
m)	Solar International	Titan	Brayton	O	Turbomachine	Opt.	67	9	1021	813	21
n)	Dutcher Industries	-	Steam Rankine	C	Reciprocating	Opt.	40	27	33.3	593	88
o)	Thermo Electron	Bottoming Cycle	Organic Rankine	C	Turbomachine	Opt.	35	21	617	516	27
p)	Microturbo (France)	Gevaudan -9 Mod. 2R	Brayton	O	Turbomachine	Opt.	80	18	838	750	25
q)	Sundstrand	ORC Proposal	Rankine	C	Turbomachine	High-speed integral	22[a]	27	667	427	27
r)	Mechanical Technology, Inc.	3 kWe FPSE	Stirling	C	Free piston	Linear	3[a]	24	60	760	52
s)	Bertin et Cie	50 kWe ORC	Rankine	C	Turbomachine	Opt.	50[a]	78 (turb)	112	250	24

[a]kW(electric)

Legend:

ORC = organic Rankine cycle
SAGT = solarized advanced gas turbine
SABC = subatmospheric Brayton cycle
AGT = automotive advanced gas turbine
FPSE = free-piston Stirling engine
Opt. = optional

Heat Engine Data

Working Fluid	Flow Rate, kg/h	P_{max}, MPa	P_{min}, MPa	P_{avg}, MPa	Energy Source	Cooling	Regen?	Dimensions, cm	Total Mass, kg	No. Units Built	Contact/Phone
Helium	N/A	-	_	2	Solar-"flat collector"	Air	Yes	44 dia x 83	150	1 Prototype	William T. Beal (614) 594-2221
Helium	N/A	-	_	11	Sodium heat pipe	Water	Yes	30 dia x 64	75	Final design phase	R. J. Meijer (313) 995-1755
Helium	N/A	-	_	5.5	Gas/solar	Water	Woven screen	?	?	Test model	Michel Dancette
Water	136	6.9	0.1	-	Oil/gas	Boiling condenser	None	?	100	18	Roger Demler (617) 890-3200
Water	76	12.8	0.1	-	Lab boiler	Boiling condenser	Yes	?	?	1	Roger Demler (617) 890-3200
Toluene	544	5.9	0.012	-	Solar	Air condenser	Fire tube	110 dia x 150	390	1	Bob Olander (303) 421-8111
Air	866	0.5	0.1	-	Solar	Ambient air in	Ceramic rotary	56 dia x 81	190	Component testing	Dan Kreiner (602) 231-7090
Air	376	0.1	0.045	-	Solar	Ambient air in	Tube sheet	88 dia x 190	364	6	George McDonald (213) 512-4519
Helium or H_2	N/A	-	-	15	Solar	Water	Wire mesh	118 dia x 207	350	25	Worth Percival (703) 549-7174
R-11	1620	2.2	0.1	-	Hot water	Air	Yes	45 dia x 61	34	2	Gorken Melikian (203) 727-7554
Air	1224	2.2	0.2	-	Diesel fuel	Ambient air in	Ceramic	58 dia x 56	136	2	Harold E. Helms (317) 242-5355
Air	1147	0.36	0.1	-	JP-4	Ambient air in	No	25 x 33 x 56	26	Many	Bill Owen (619) 238-5754
Air	4915	0.38	0.1	-	JP-4	Ambient air in	No	43 x 41 x 76	41	Many	Bill Owen (619) 238-5754
Water	94	10.3	0.07	-	Diesel No. 2	Air condenser	Yes	63 x 30 x 69	264	1	Ted J. Smith (619) 578-5502
Trifluoro-ethanol/water	786	4.8	0.1	-	Exh. gas	Air	Yes	91 x 152 x 91	228	3	Mike Kaplow (617) 890-8700
Air	3060	0.33	0.1	-	JP-4	Ambient air in	Yes	41 dia x 77	55	Many	Herb J. Sagendorf (516) 567-3780
Toluene	817	4.5	0.009	-	Solar	Air	Yes	81 dia x 120	296	Component testing	Doug Lacey (815) 226-7991
Helium	N/A	7	5	6	Nat. gas	Water	Yes	96 x 61 x 66	135	Prototypes	George Dochat (518) 785-2242
Fluorinert	5256	1.1	0.007	-	Ht. trans. fluid	Water	?	?	?	Prototype	A. Verneau (3) 056 25 00

3. Dutcher Industries

Dutcher Industries of San Diego, California, has developed a small steam engine as part of the California and U.S. automotive steam engine development program. Distilled water is the working fluid for this closed double-expansion reheat/regenerative cycle. With steam at 10.3 MPa (1494 psia) and 593°C (1099°F), the engine is designed to produce 40 kW shaft output at an efficiency of 27%. One prototype had been tested, with a second engine having been partially fabricated, before program funding stopped.

4. Foster-Miller

Foster-Miller Associates of Waltham, Massachusetts, is developing two reciprocating steam Rankine-cycle engines. One, a single-piston, simple cycle produces 22.4 kW of power at an efficiency of 22% with superheated steam at 538°C/6.9 MPa (1000°F/1000 psia). A compound reheat version produces 28.2 kW at an efficiency of 36% when supplied with superheated steam at 700°C/12.8 MPa (1292°F/1856 psia). This second engine incorporates reheat between the high-pressure and the low-pressure cylinders, a regenerator, and water-actuated valves. Eighteen of the single-cylinder engines have been built and tested, and one compound reheat cycle engine is currently undergoing test.

5. S.P.S.

S.P.S., Inc., of Miami, Florida, produces a line of organic Rankine cycles in the power range of 10 to 400 kW. Freon working fluids are used, with power being produced by a rotary screw expander. These engines operate from hot water or steam at 66°C (151°F) or higher as long as the cooling water is 55°C (100°F) cooler. For a source temperature of 100°C (212°F), the efficiency of the cycle is 15%. A number of these units have been produced and sold to foreign countries. Because of the low operating temperature, these units are too large and inefficient to be considered for parabolic dish applications and, therefore, are not included in Table 3-1.

6. Sundstrand

Sundstrand Energy Systems of Rockford, Illinois, proposed an organic Rankine cycle to be developed for the solar dish program from their previous organic Rankine-cycle design experience. The proposed engine combined an axial-flow turbine, a high-speed alternator, and Pitot pump as a single rotating unit. Using toluene as a working fluid, the engine was designed to produce 22 kW of electricity at a design efficiency of 27% when operating at 427°C (800°F). The design included regeneration with an air-cooled condenser surrounding the unit.

7. Thermo Electron

Thermo Electron Corporation of Waltham, Massachusetts, is developing a Rankine cycle for waste heat recovery applications. The engine consists of an axial turbine expander along with boiler, pump, and condenser. The working fluid is a mixture of trifluoroethanol and distilled water that when heated to 516°C, will produce a 35 kW shaft power with an efficiency goal of 25%. Three of these engines have been built and tested.

8. United Technologies Research Center

The United Technologies Research Center, located in East Hartford, Connecticut, has developed a small Rankine power cycle that utilizes solar-produced hot water to drive the compressor of a solar chiller. Decoupled, it could be used to drive a generator at the focus of a parabolic dish. The engine incorporates a centrifugal compressor and a radial turbine with a heat exchange boiler. The working fluid is R-11, which reaches a maximum temperature of 149°C (300°F) when producing 18.7 kW (25 hp) of shaft power at 24% thermal efficiency. The cycle incorporates a regenerator, and the condenser is air cooled.

C. STIRLING CYCLES

1. Bertin et Cie

Bertin et Cie, located in Plaisir, France, is developing a 3-kW test model of a free-piston Stirling engine with a linear electric generator on the power piston and a linear motor on the displacer for control of the phase angle. Although this size is outside the range of this study, information is included because of the engine's scale-up potential. The engine is a hermetically sealed unit opeating at 50 Hz, with helium at 3.7 ~~to~~ 5.5 MPa (537 to 798 psia) as the working fluid. The design operating temperature is 590°C (1094°F) with a sink temperature of 70°C (158°F).

2. Mechanical Technology, Incorporated

Mechanical Technology, Incorporated (M.T.I.), located in Latham, New York, has under development a 3-kW(electric) free-piston Stirling engine for small power applications. Although this particular engine is too small to fit solar application requirements, it is included because of its development potential. The engine incorporates a linear alternator and oscillates at 60 Hz. Operating at a high temperature of 760°C (1400°F), the engine has a thermal efficiency of 24% using helium at 6 MPa (870 psia) as the working fluid. Currently, there are prototype engines of this design operating with a natural gas burner as the heat source.

3. Stirling Thermal Motors

Stirling Thermal Motors of Ann Arbor, Michigan, under licensing agreements with Phillips (Sweden), is developing a 40-kW kinematic Stirling engine. The engine is a four-cylinder, double-acting configuration incorporating a variable angle swash plate drive. The working fluid is helium at 11 MPa (1595 psia), which is heated to 800°C (1472°F) by a sodium heat pipe transfer unit. The predicted efficiency at this operating condition is 48%. The final design of the prototype engine has been completed and the major castings made. Fabrication and testing of the prototype of this engine will be completed when funding is obtained.

4. Sunpower

Sunpower, Inc., in Athens, Ohio, is developing a free-piston Stirling engine connected to a linear alternator. The engine has an electrical output of 10 kW with an efficiency goal of 44%. The working fluid is helium at 2 MPa (290 psia) with a maximum cycle temperature of 710°C (1310°F). One prototype engine of this size has been built and tested.

5. United Stirling

United Stirling of Malmo, Sweden, produces a four-cylinder, crankshaft-drive reciprocating Stirling engine, which has been tested on a solar concentrator. The working fluid can be either helium or hydrogen with power output controlled by gas pressure. At a mean pressure of 15 MPa (2175 psia), the engine will generate 24.4 kW of electricity at an efficiency of 40% when heated to 720°C (1328°F). Twenty-five of these engines have been produced and are undergoing testing for various applications. A 55- to 60-kW version of this engine is under development for a Messerschmitt-Boelkow-Blohm dish system in Saudi Arabia.

SECTION IV

CONCLUSIONS

From a survey of 39 engine manufacturers, a list was compiled of 19 engines (in the 10-to-100-kW range using either Brayton, Rankine, or Stirling cycles) that employ external heat addition and, therefore, are potentially applicable for parabolic dish-electric modules. Many companies responding to the survey are not listed in Section III because their engines were not applicable for solar power generation or their interest in development of these engines had ceased.

It was observed that, because of the dominance of Otto- and Diesel-cycle engines involving internal combustion/heat addition, few applicable external-heat-addition engines exist in the size range considered. Many manufacturers of Brayton and Rankine engines indicated that the size of their engine line was larger than 100 kW power output. Several Stirling engine manufacturers, on the other hand, are developing engines smaller than 10 kW.

From discussions with many of the manufacturers, it was found that during the 1970's, there was considerable interest in developing new engines in the 10-to-100-kW range both for automotive applications and solar energy conversion. Several companies, who had initiated small engine development programs during that time, have dropped further development due to reduced federal funding for solar and automotive engine technology development.

It appears from the survey that the technology exists to produce external-heat-addition engines of appropriate size with thermal efficiencies of over 40%. Developmental problem areas seem to be materials and sealing. High efficiency Brayton-cycle engines must operate at 1000 to 1400°C, where most materials lose their strength. High efficiency kinematic Stirling-cycle engines can attain an efficiency of 40% at temperatures in the range of 700 to 800°C, with sealing and component thermal cycling proving to be the significant problem areas. Rankine cycles operate at even lower temperatures with lower efficiencies; with this engine cycle, the goal is to design efficient, multi-stage expansion turbines for the power output associated with solar applications.

All of the engines listed in Table 3-1 are of a developmental nature having little, if any, operating experience. Because O&M costs will be an important factor in determining the cost effectiveness of parablic dish solar thermal power systems (Section I), it is necessary to gain operating experience that will provide this cost data. Furthermore, the possibilities of exploring design modifications to reduce O&M costs for solar applications can probably be more economically addressed in the development stage. Thus, it appears that the parabolic dish program can benefit from close interaction with the different engine development programs identified in this survey.

It is concluded that high efficiency engines can be developed for solar parabolic dish applications. Prototypes and limited production models are currently in operation. Because one specific engine or engine type does not emerge from this survey as being dominant for solar applications, it appears that additional developmental effort should be expended on a range of different engine options. The availability of different engine options will permit greater flexibility in matching dish systems to application requirements and in providing competitive sources.

APPENDIX A

DERIVATION OF EQUATION (4)

Overall efficiency of a solar power system is:

$$\eta = \eta_e \ \eta_s$$

Combining Equations (2) and (3) and letting $T_H = T_r$ and $T_L = T_a$:

$$\eta = K\left(1 - \frac{T_a}{T_r}\right)\left[\phi\rho\tau\alpha - \frac{U_L\ (T_r - T_a)}{CR\cdot I} - \frac{\varepsilon\sigma(T_r{}^4 - T_a{}^4)}{CR\cdot I}\right]$$

Define:

$$\theta = \frac{T_r}{T_a}$$

$$C_1 = \phi\rho\tau\alpha$$

$$C_2 = \frac{U_L\ T_a}{CR\cdot I}$$

$$C_3 = \frac{\varepsilon\sigma T_a{}^4}{CR\cdot I}$$

then,

$$\eta = K\left(1 - \frac{1}{\theta}\right)\ [C_1 - C_2\,(\theta - 1) - C_3\,(\theta^4 - 1)]$$

Taking the derivative with respect to θ :

$$\frac{d\eta}{d\theta} = K\left(1 - \frac{1}{\theta}\right)[-C_2 - 4C_3\theta^3] + K\left(\frac{1}{\theta^2}\right)[C_1 - C_2\ (1 - \theta\ ) - C_3(\theta^4 - 1)]$$

and to find the inflection point (optimum η with respect to θ) and calling this temperature θ_{max};

$$0 = 4C_3\theta^3_{max} + 3C\theta^2_{max} - C_2 + \frac{C_1 + C_2 + C_3}{\theta^2_{max}}$$

Simplifying, we get:

$$\theta^5_{max} - \frac{3}{4}\ \theta^4_{max} + \frac{C_2}{4C_3}\ \theta^2_{max} = \frac{C_1 + C_2 + C_3}{4C_3}$$

which is Equation (4).

APPENDIX B

ADDRESS LIST

Manufacturer	Contact
AFI Energy Systems 110 S. Orange Ave. Livingston, NJ 07039	Robert H. Sawyer (201) 533-2091
Arthur D. Little, Inc. 20 Acorn Park Cambridge, MA 02140	Peter Teagan (617) 864-5770
Barber-Nichols Engineering Co. 6325 W. 55th St. Arvada, CO 80002	Bob Olander (303) 421-8111
Detroit Diesel Allison Div. General Motors Corp. 13400 W. Outer Dr. Detroit, MI 48228	Gene Helms (317) 242-5355
Dornier System GmbH 7997 Immenstaad West Germany	Dr. Lippmann (07545) 8-3440 U.S. Rep. (213) 681-3491
Dutcher Industries 7564 Trade St. San Diego, CA 92121	Ted Smith (619) 578-5502
Energy Research and Generation Lowell & 57th St. Oakland, CA 94608	Dr. Benson (415) 658-9785
Energy Technology, Inc. 4914 E. 154th St. Cleveland, OH 44128	John Martin (216) 587-0555
Foster-Miller Assoc., Inc. 350 Second Ave. Waltham, MA 02154	Roger L. Demler (710) 324-1468
Garrett AiResearch Mfg. Co. 9851 Sepulveda Blvd. Los Angeles, CA 90009	
Garrett Turbine Engine Co. 111 S. 34th St. Phoenix, AZ 85010	Ray Rockey (602) 231-2679
General Electric Co. Aircraft Engine Group One Neuman Way Cincinnatti, OH 45215	M. C. Hamsworth

Manufacturer	Contact
General Electric Co. Advanced Energy Programs 501 Allendale Road King of Prussia, PA 19406	Robert Tharpe (215) 962-5665
Hamilton Standard Div. United Technologies Corp. Bradley International Airport Windsor Locks, CT 06096	Don Phillips (203) 623-1621
Jay Carter Enterprises Route 1 Box 405A Burkburnett, TX 76354	Jay Carter, Sr. (817) 569-2238
Kawasaki Heavy Industries Ltd. Seagram Bldg, Room 3309 375 Park Ave. New York, NY 10022	Mr. Ono (212) 759-4950
Lucas-France 11 Rue Lord-Byron 75008 Paris France	
M.A.N. Werk, Augsburg Stadtbachstrasse 1 8900 Augsburg West Germany	i.V. Siemer and i.A. Jenke
Mechanical Technology, Inc. Energy Systems Division 968 Albany-Shaker Rd. Lathan, NY 12110	Bruce Goldwater (518) 785-2211
Messerschmitt-Boelkow-Blohm GmbH Space Division Postfach 801169 8000 Munchen 80 West Germany	Helmut Hopmann (089) 6000 5521
Microturbo Chem. dů Pont-de-Rupe BP 2089 31089 Toulouse Cedex France	Herb Sagendorf (U.S. Rep) (516) 567-3780
North American Turbine Corp. P.O. Box 40510 Houston, TX 77040	Mr. Fairbanks (713) 466-6200
Ormat Systems, Inc. 98 South Street Hopkinton, MA 01748	Nicholas Christopher (617) 653-6300

Manufacturer	Contact
Pratt & Whitney Aircraft Group United Technologies Corp. 400 Main St. East Hartford, CT 06108	Mr. Mendleson (203) 565-4321
Rolls Royce Ltd. Sales Engineering Division PO Box 31 Darby DE28BJ England	J. A. J. Rees Derby (0332) 42424 Ext. 157
SOFRETES Mengin Z.I. G'Amilly BP 163 45200 Montargis France	
S.P.S. Inc. P.O. Box 380006 Miami, FL 33138	Victor S. Warminger (305) 754-776
Societe Bertin et Cie BP 3 78373 Plaisir Cedex France	P. Tarbos M. Dancette
Societe E.C.A. 17 Av. du Chateau 92190 Mendon France	
Solar Turbines International P.O. Box 80966 San Diego, CA 92138	Bill Owen (619) 238-5754
Stirling Power Systems Corp. 7101 Jackson Rd. Ann Arbor, MI 48103	Lennart Johansson (313) 665-6767
Stirling Thermal Motors 2841 Boardwalk Ann Arbor, MI 48104	R. J. Meijer (313) 995-1755
Sundstrand Energy Systems 4747 Harrison Ave. Rockford, IL 61101	Mr. Bates (815) 226-6000
Sunpower, Inc. 6 Byard St. Athens, OH 45701	William Beale (614) 594-2221

Manufacturer	Contact
Thermo Electron Corp. 101 First Ave. Waltham, MA 02154	Mike Koplow (617) 890-8700
United Stirling, Inc. 211 The Strand Alexandria, VA 22314	Worth Percival (703) 549-7174
United Stirling (Sweden) AB & Co. Box 856 S20180, Malmo Sweden	
United Turbine (Sweden) AB M. Grangesbgsg 18 Malmo Sweden	
Williams Research Corp. 2280 W. Maple Rd. Walled Lake, MI 48088	William Bower (313) 624-5200
Alpha United, Inc. P. O. Box 847 El Segundo, CA 90245	Don Elbert (213) 322-9570
Sanders Associates 95 Canal Street Nashua, NH 03060	Bear Davis (603) 885-4321
Pratt & Whitney Canada	Colin B. Wrong (514) 647-7584
United Technologies Research Center Silver Lane East Hartford, CT 06108	Gorken Melikian (203) 727-7000
Ricardo Consulting Engineers BridgeWorks Shoreham-by-Sea West Sussex BN45FG England	

APPENDIX C

DATA REPORT AND COVER LETTER

JET PROPULSION LABORATORY California Institute of Technology • 4800 Oak Grove Drive, Pasadena, California 91109

July 18, 1983

Refer to 341-WBS:gg

Attention: Chief Engineer's Office

Dear Sirs,

Jet Propulsion Laboratory has an interest in low-cost, high-performance heat engines with power outputs in the 10 to 100 kW range for application to parabolic dish solar power systems. We are surveying engine manufacturers to ascertain whether there are advanced heat engines on the horizon which would enhance the economic viability of the production of electricity from solar energy.

In a typical application, the engine is placed at the focus of a parabolic dish solar energy concentrator. Concentrated solar flux heats the engine's working fluid, operating the engine and driving an alternator. Low-temperature heat is typically rejected at the engine to eliminate the necessity of fluid transport to the ground.

The parameters which govern the design of dish power systems are, in their order of importance,

a) thermal conversion efficiency,

b) operation and maintenance cost,

c) capital cost of unit.

The application source temperature can be as high as needed with materials capability being the limiting factor.

The current technology with which we are working has the parameters listed in Table 1. It is the purpose of this study to identify developed or partially developed engines in the 10 to 100 kW output range that are suitable for parabolic

Telephone (213) 354-4321 Twx 910-588-3269 Twx 910-588-3294 Telex 67-5429

dish power systems. In addition, it is important to ascertain the probable timing of commercialization and the probability of sales for other than solar applications since the initial solar market is not envisioned to be large.

Attached is a questionnaire indicating the information needed on any advanced engine you believe may be applicable to the solar dish program. I will contact you by telephone in about a week to discuss candidate engines and will transcribe the appropriate data at that time. If you prefer, you may fill out the attached questionnaire ahead of time and return it to me at the following address:

Dr. William (Bill) Stine
Jet Propulsion Laboratory
Mail Stop 507/228
4800 Oak Grove Drive
Pasadena, CA 91109

Realizing the sensitivity of the type of data which we are requesting, the questionnaire will be marked for JPL internal use only. The information is to be used only for JPL studies and will not be disclosed to outside parties.

Will you please reply indicating the appropriate person to contact for this information.

Sincerely,

William B. Stine, Ph.D.

Attachment

cc: AFI Energy Systems
Arthur D. Little
Barber-Nichols Engineering
Detroit Diesel Allison
Dornier Systems GmbH
Dutcher Industries
Energy Research & Generation
Energy Technology Inc.
Foster-Miller Assoc.
Garrett AiResearch Mfg.
Garrett Turbine Engine
GE - Valley Forge
Hamilton Standard
Jay Carter Enterprises
Kawasaki Heavy Ind.
Lucas-France
M.A.N. Werk, Augsburg
Mechanical Technology, Inc.
M.B.B. GmbH
Microturbo
North American Turbine
Ormat Systems
Pratt & Whitney
Rolls Royce Ltd
SOFRETES Mengin
SPS Inc.
Societe Bertin et Cie
Societe E.C.A.
Solar Turbines International
Stirling Power Systems
Stirling Thermal Motors
Sundstrand Energy Systems
Sunpower
Thermo Electron
United Stirling
United Turbine
Williams Research

TABLE 1

PERFORMANCE AND COST PARAMETERS OF CURRENTLY AVAILABLE ENGINES APPLICABLE

TO PARABOLIC DISH SOLAR POWER SYSTEMS

Engine Type	Engine Brake Efficiency	Maximum Temperature	Fraction of Carnot	Type*	Capital Cost**	O&M Costs
Rankine - organic	23%	400°C	.41	T		
Brayton	32%	815°C	.44	T	Approx. $200 per Kw	Turbine Machinery $300/eng/yr
Brayton - ceramic	41%	1149°C	.52	T		
Brayton - subatmospheric	27%	871°C	.37	T		Kinematic Machinery $860/eng/yr
Stirling	40%	720°C	.57	K		

*T - Turbomachinery   K - Kinematic

**Based on 100,000 cumulative, 25,000 units/year (1981 dollars)

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POWER CONVERSION UNIT PERTINENT DATA REPORT

Note: This report is to be completed by the appropriate JPL employee during and after telecon or personal interview with each manufacturer's representative(s).

Report Prepared By: ______________________________ Section _____

Engine Thermodynamic Cycle

Closed	________	Rankine	___
		Stirling	___
		Brayton	___
Open	________	Other	___

Engine Mechanical Arrangement

Rotating

Compressor		Expander	
Centrifugal		Turbine	___
Axial	___	Vane	___
Vane	___	Lobe	___
Lobe	___	Screw	___
Other	___	Other	___

Reciprocating

Crank	___
Swash Plate	___
Free Piston	___
Other	___

Manufacturer

Name ______________________________

Location ______________________________

Telephone No. ______________________________

Person(s) Contacted ______________________________

Engine Performance

Design Specifications

Shaft Power Output	: ________	kW	or	________	HP
Shaft speed	: ________	rpm	or	________	Hz
Energy Source(s) for Current Engine	: ______________________________				
Source Temperature	: ________	°F	or	________	°C
Sink Temperature	: ________	°F	or	________	°C
Thermodynamic Medium	: ______________________________				
Flow rate of medium	: ________	lbm/s	or	________	kg/s
Mean Effective Operating Pressure	: ________	lbf/in^2	or	________	MPa
Maximum Cycle Pressure	: ________	lbf/in^2	or	________	MPa
Minimum Cycle Pressure	: ________	lbf/in^2	or	________	MPa

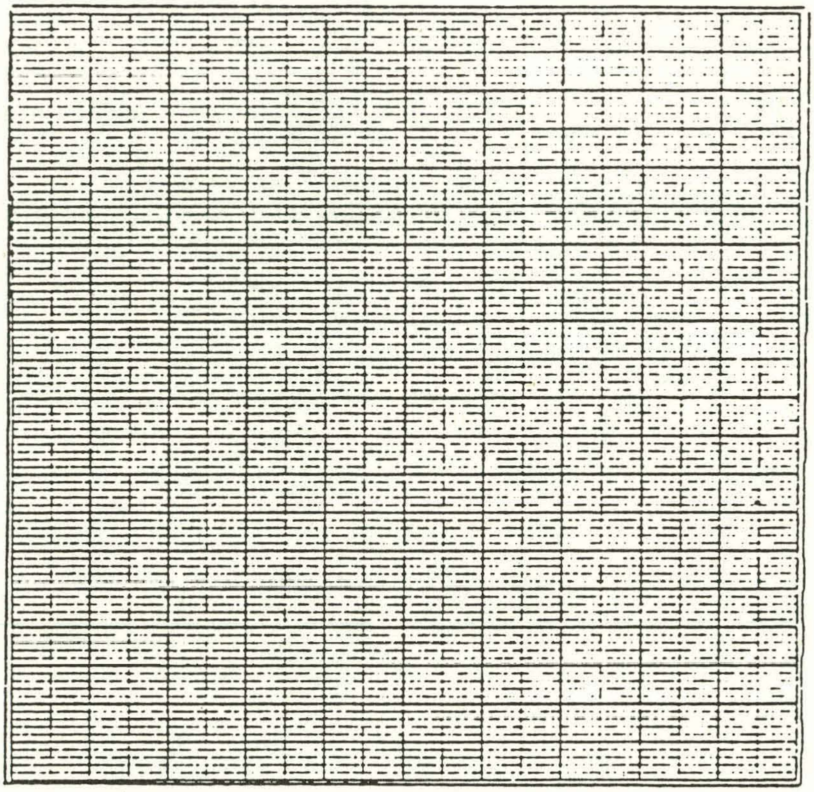
THERMODYNAMIC CYCLE
WITH STATE POINTS
DEFINED

Notes:

Present experimental results wherever possible.

Define coordinates.

Define state points as precisely as possible.



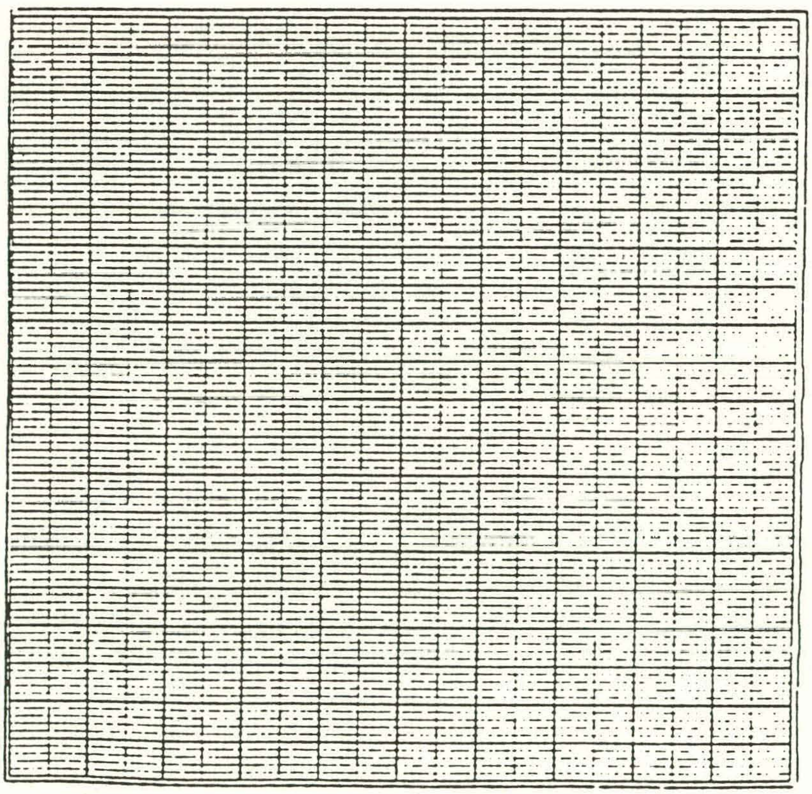
PART-LOAD
CHARACTERISTICS

Notes:

Present experimental results wherever possible.

Specify form of input power, i.e., whether full or direct thermal.

Specify form of output power, i.e., shaft or electric (AC or DC and voltage).



Input Power

Engine Maintenance Requirements

Mean Time Between Failure (MTBF)

For components failing most frequently

Component	Hours
1. ______	______
2. ______	______
3. ______	______
4. ______	______
5. ______	______

For the complete engine - - - - - - - - - - - - - - - - - ______

Routine Maintenance Requirements (RMR)

Frequency	______ 1/y
Labor	______ m.h./y
Parts	______ $/y
Facilities (Chargeable to RMR)	______ $/y
Service Equipment (Chargeable to RMR)	______ $/y

Minor Overhaul Requirements (MnOR)

Frequency	______ 1/y
Labor	______ m.h./y
Parts	______ $/y
Facilities (Chargeable to MnOR)	______ $/y
Service Equipment (Chargeable to MnOR)	______ $/y

Major Overhaul Requirements (MjOR)

Frequency	______ 1/y
Labor	______ m.h./y
Parts	______ $/y
Facilities (Chargeable to MjOR)	______ $/y
Service Equipment (Chargeable to MjOR)	______ $/y

Production Potential

	Production Rate Rate (Units/y)	Unit Value at Stated Production Rate Cost ($/Unit)	Price ($/Unit)
Current	__________	__________	__________
1985	__________	__________	__________
1990	__________	__________	__________

Producibility

Current (Short discussion, including states of component and complete unit development, prototype testing, number of units in the field, and commercial availability)

Potential for Cost Reduction (Short Discussion)

Sources of Funding

Current ______________________________

Scheduled ______________________________

Potential ______________________________

Adaptability to Solar Systems

Modifications Required for Solarization

Cost for Required Modifications: ____

____

Estimated or Actual? E A (Circle One)

Collector Requirements

Required Engine Thermal Input at Rated Power ______________ kW

Required Receiver Operating Temperature __________°F or ____________ °C

Power Conditioning Requirements

Performance and Maintenance Requirements for the Solarized Engine as Compared with Those for the Engine in Its Current Form

Capabilities of the Manufacturer

Capitalization ______________________________ $

Sales Volume ______________________________ $/y

Current Products

Product	Percent of Total Sales
1. ________________	________________
2. ________________	________________
3. ________________	________________

Number of Employees ________________

Factory Floor Space ________________ ft^2

Note: Refer to the Dunn and Bradstreet and Thomas registers in compiling the above requested data.

Fraction of personnel currently dedicated to the PCU ________________

Fraction of facilities currently dedicated to the PCU ________________

Related past accomplishments of the manufacturer:

____

____

____

____

____

____

Selling Price of the PCU:

Lot Size	Selling Price
1	________ $/Unit
10	________
100	________
1,000	________
10,000	________

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SURVEY OF MANUFACTURERS OF HIGH-PERFORMANCE HEAT ENGINES ADAPTABLE TO SOLAR APPLICATIONS