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15 kWe (NOMINAL) SOLAR THERMAL-ELECTRIC POWER CONVERSION CONCEPT DEFINITION STUDY

Steam Rankine Reciprocator System

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Jay Carter Enterprises

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FOR
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OFFICE OF SOLAR, GEOTHERMAL, ELECTRIC
AND STORAGE SYSTEMS
DIVISION OF CENTRAL SOLAR TECHNOLOGY
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Table of Contents

1 SUMMARY

A study was undertaken to identify steam Rankine power conversion concepts with the potential for significantly improved efficiency at the lower power levels associated with point focusing distributed receiver solar systems. The power conversion subsystem is an engine-generator (E/G) set consisting of a Rankine cycle heat engine operating with steam from a solar receiver and driving a suitable electric generator for power output. The solar collector (dish) and the solar receiver (steam generator) are not within the study scope.

A mathematical model of the Jay Carter Enterprises steam engine concept was developed in order to conduct a parametric analysis covering a broad range of steam inlet pressure and temperature as well as variations in engine configuration and condensing temperature. As a result of this parametric study, a design point at 677°C (1250°F) and 17.2×10^6 N/m^2 (2500 psi) was selected as maximum inlet steam temperature and pressure. A conceptual design of a 3600 rpm, reciprocating expander, sized for a maximum thermal power of 80 kw, was developed. The expander, which is an adaptation of an existing automobile engine has counter-rotating counter-balance shafts to eliminate vibration. The engine can be configured with a single uniflow cylinder for simple cycle operation or with a high pressure cylinder added in a compound reheat cycle.

Power generation is accomplished with a three-phase induction motor directly coupled to the expander and connected electrically to the public utility power grid. Frequency and voltage automatically correspond to line frequency and voltage because of the inherent characteristics of induction motors. The expander, generator, water pump, and control system are dish mounted. The steam condenser, water tank and accessory pumps are ground based.

The maximum heat engine efficiency is estimated to be 28 percent in the simple cycle mode and 33 percent in the compound reheat cycle mode. Dish mounted weight is 297 kg (654 lb) and total weight is 601 kg (1323 lb). Estimated cost is $3,307 or $138 per kw of maximum power output.

The concept is considered to be technically within the state of the art. System efficiency and therefore economic feasibility could be greatly enhanced by incorporating the concept into a total energy system in which both the high quality electrical energy and the exhaust heat are used to satisfy the energy needs of typical users. The efficiency estimated for operation as a total energy system would be about 90 percent.

2 INTRODUCTION

Jay Carter Enterprises, Inc. (JCE) is a research and development organization with specific experience in steam engines for automotive use; and electric power production from wind turbine generators. The background from these project areas is adapted and expanded to meet the requirements of a steam engine-generator design for solar thermal-electric applications.

The scope of this project is a parametric analysis and subsequent design point conceptual design of a steam engine-generator set for operation with a dish type solar collector. The study is one of several parallel efforts on various power conversion concepts for this application. References 1, 2, 3, 4, and 5 are the final reports of the related studies. In each case the study scope involves the engine-generator set only; to the exclusion of the dish mirror or solar heat receiver heat exchanger.

Parametric analyses and conceptual designs were completed for both a simple cycle engine design and a compound-reheat cycle adaptation. The performance estimates of up to 30 percent power conversion efficiency (heat in steam to electric out) are significant in that they represent approximately a 2:1 improvement over the efficiency available from present commercial steam turbine-generator combinations at the similar 20 kWe output size range.

3 PARAMETRIC ANALYSIS

The parametric studies were concerned with determining the effect of inlet steam pressure and temperature on engine of various configurations, sized for a 15 kW generator drive.

3.1 MATHEMATICAL MODEL

The objective of the mathematical model developed for this program was to accurately represent a Rankine cycle steam engine similar to previous JCE steam engines but of a size and configuration suitable for the solar thermal electric application. The model was coded for the Hewlett Packard 2000 computer system to permit rapid and accurate analysis of a wide range of values for various system parameters.

The model was designed to operate in several modes. These include:

1. For a given electrical power output, determine engine configuration, heat engine efficiency, power conversion efficiency and required thermal input. Parameters to be varied include inlet steam pressure, inlet steam temperature, condensing temperature, and expansion ratio.

2. For an engine of a given configuration, determine electric power output, heat engine efficiency, power conversion efficiency, and inlet steam pressure.

   Parameters to be varied include thermal power input, inlet steam temperature and condensing temperature.

3. Similar to mode 1 but fixing thermal power input and determining electrical power output.

The model was designed to perform analyses on simple cycle, compound cycle and compound reheat cycle engines. For all engine types, the model can be operated in the first or third mode, i.e. to determine an engine configuration, and to then automatically operate in the second mode, with that engine configuration, over a series of relative thermal inputs such as Table 3-1 in order to determine an annual average heat engine efficiency and annual average power conversion efficiency.

Table 3-1 Nominal Annual Duty Cycle

Relative Thermal Input Level	Annual Duration Hours
1.00	100
.95	400
.92	400
.90	400
.86	400
.83	400
.76	400
.65	400
.50	400
	3300

The model is programmed to index thru a series of J = 1 to 9. These correspond to the nine different values of relative thermal input [R (J)] given in Table 3-1 with J = 1 being the full power condition with relative thermal input equal to 1. Maximum thermal input (G1), maximum steam inlet pressure (P1), steam inlet temperature (T1), steam condensing temperature (TO), high pressure cyclinder expansion ratio (X1), and low pressure expansion ratio (X2) are specified. A schematic of the model to operate in this manner for a compound reheat cycle engine is shown in Figure 3-1. Symbols are defined in Appendix A. The step-by-step analysis procedure is described in Appendix B.

Two observations can be made about the model. It is largely empirical rather than theoretical. It was developed to reflect the experimental results obtained from previous JCE engines of similar configuration operated over a temperature range of 950-1100^{o}F and pressures up to 2500 psi. It is highly iterative, i.e., many values are required in the analysis prior to the time when it is possible to calculate the value. Initial values are estimated, these are constantly updated as better values are available until all calculated values converge to estimated values.

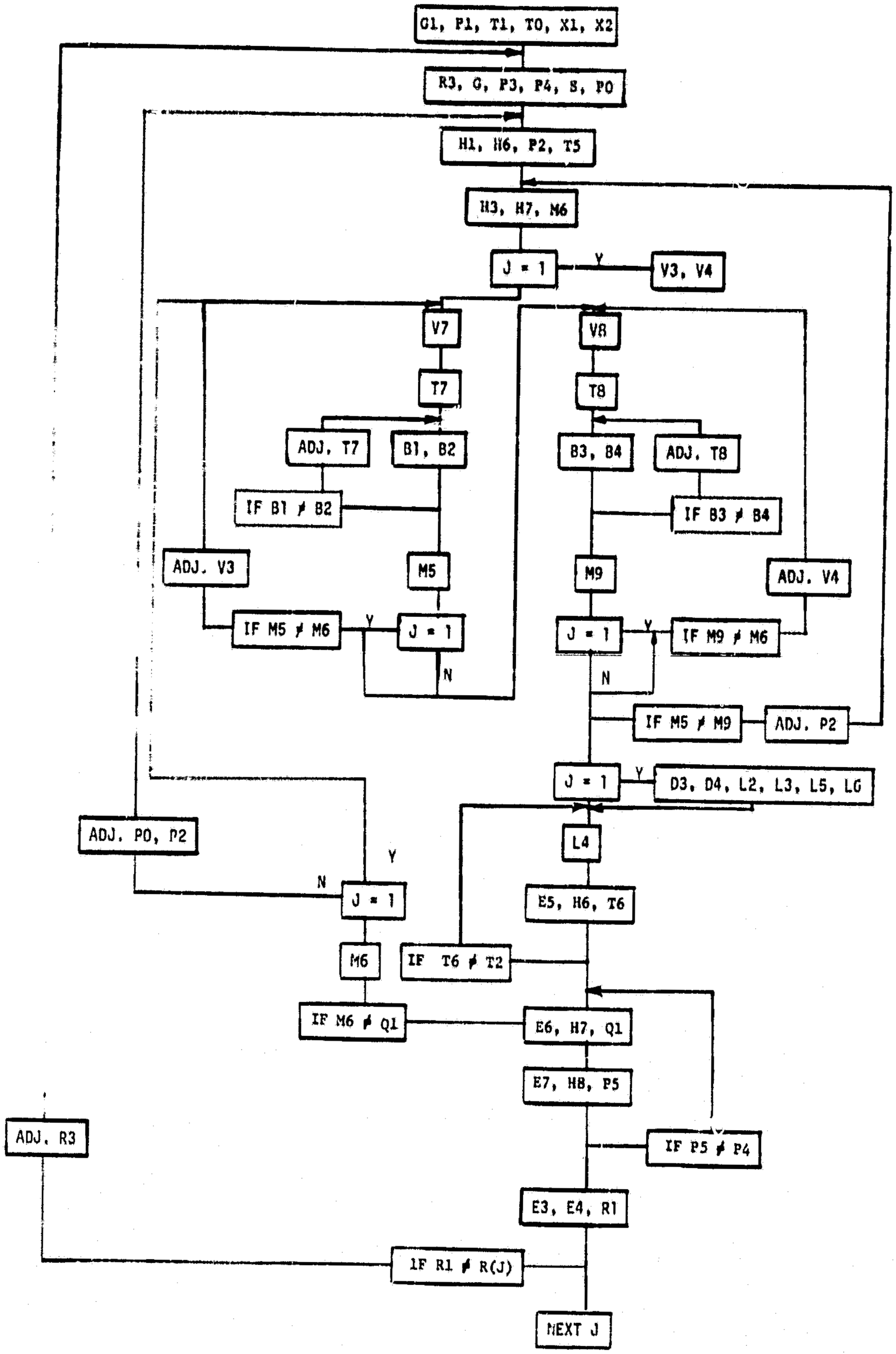


Figure 3-1 Computer Program Flow Diagram

3.2 SIMPLE CYCLE PARAMETRIC RESULTS

For the simple cycle engine, stroke was held fixed at 7.62 cm (3 in) with the number of cylinders being a minimum of two. The number of cylinders was increased as required to limit cylinder diameter to 7.62 cm (3 in) maximum at lower pressures corresponding to increased steam volume flow. The parametric results are shown in Figure 3-2 as a function of cylinder expansion ratio. From these results an initial simple cycle design point of 17.2×10^6 N/m^2 pressure, 677°C initial steam temperature and a cylinder expansion ratio of 16 was recommended. Temperature and pressure recommendation are based on material properties, lubrication requirements, and past design experience in addition to the parametric performance results. Note on the curves 4 and 7 that lowering the condensing temperature has a minimum performance impact on the positive displacement engine.

3.3 COMPOUND-REHEAT CYCLE PARAMETRIC RESULTS

For compound engines, stroke was held fixed at 7.62 cm (3 in) for both the high and low pressure cylinders. A single high pressure cylinder was employed. The number of low pressure cylinders was determined as for the simple cycle engine. The results are shown in Figure 3-3 as a function of low pressure expansion ratio. The temperature and pressure recommendation remain the same as for the simple cycle. The initial recommendation on configuration is for curve number nine (9) at a high pressure expansion ratio of 2.5 and a low pressure expansion ratio of 8.

3.4 EFFECT OF ENGINE POWER LEVEL ON PERFORMANCE

A limited number of parametric calculations were made to evaluate the effect of engine power level on performance. The results are shown in Figure 3-4. Note that the heat engine efficiency does not account for the impact of the electric generator which will also exhibit a general trend of reducing efficiency at smaller sizes.

	Pressure $N/m^2 \times 10^6$	Temperature ^oC
1	10.34	538
2	17.24	538
3	10.34	677
4	17.24	677
5	10.34	816
6	17.24	816
7	17.24	677 (38°C CONDENSING)

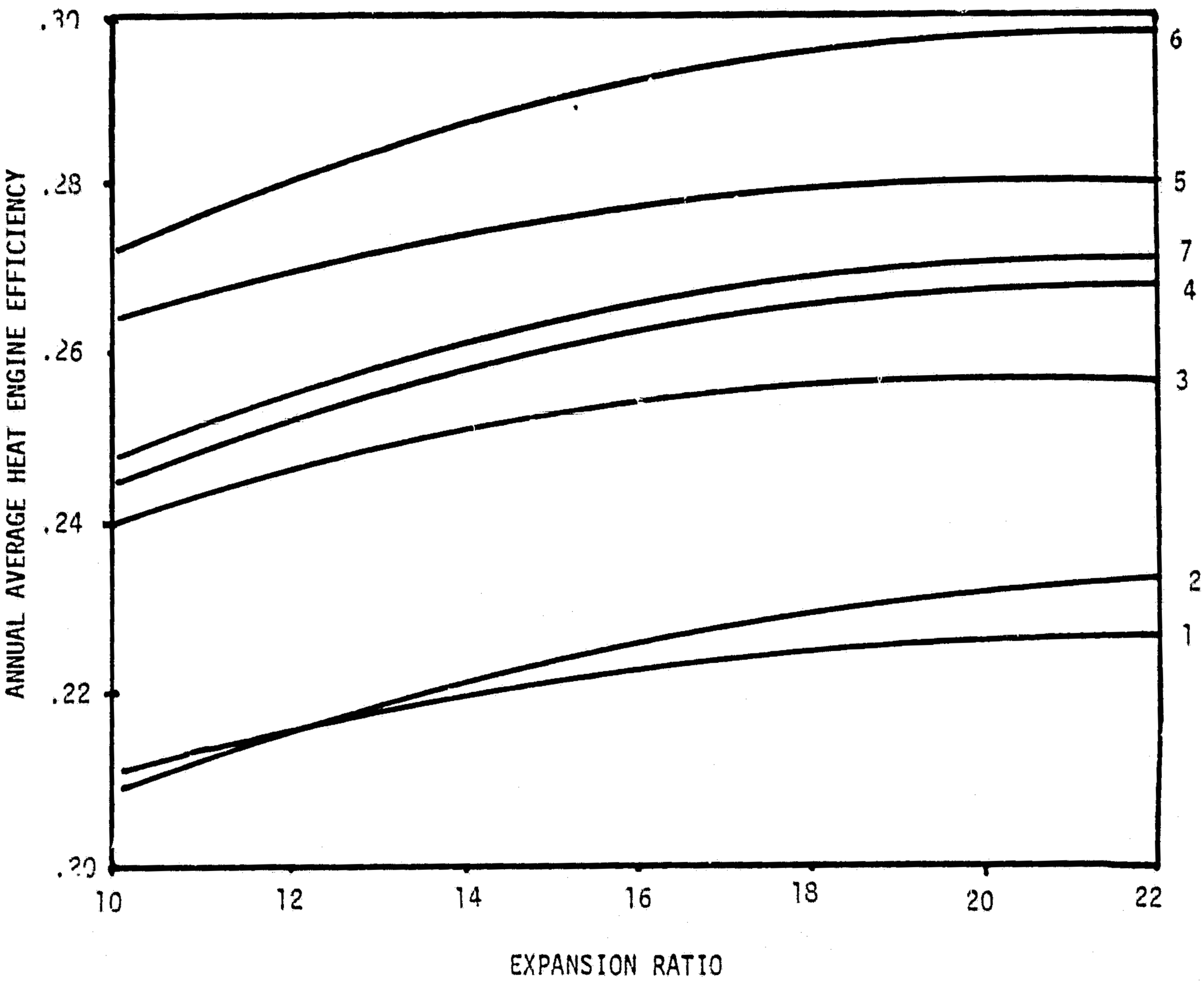


FIGURE 3-2 Simple Cycle Rankine Engine Parametric Study
15 kWe, 100°C Condensing Temperature

	Pressure $N/m^2 \times 10^6$	Temperature °C	H.P. ER
1	10.34	538	1.5
2	17.24	538	1.5
3	10.34	677	1.5
4	17.24	677	1.5
5	10.34	538	1.5
6	17.24	538	2.0
7	17.24	538	2.0
8	17.24	677	2.0
9	17.24	538	2.5
10	17.24	677	2.5
	17.24	677	3.0

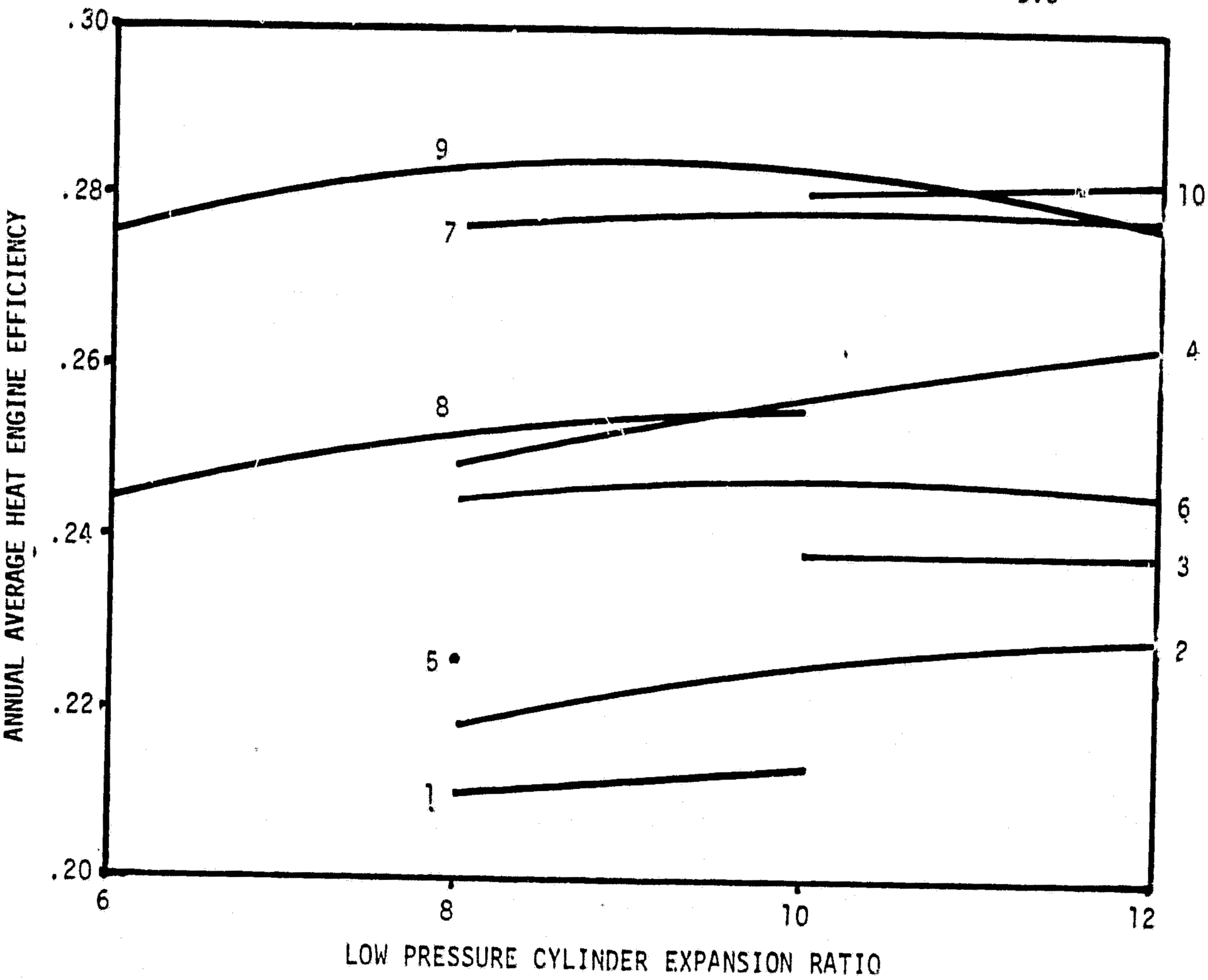


FIGURE 3-3 Reheat Cycle Rankine Engine Parametric Study
15 kWe, 100°C Condensing Temperature

	Type	Pressure $N/m^2 \times 10^6$	Temperature ^oC
1	Simple	13.79	566
2	Simple	17.24	677
3	Reheat	17.24	677

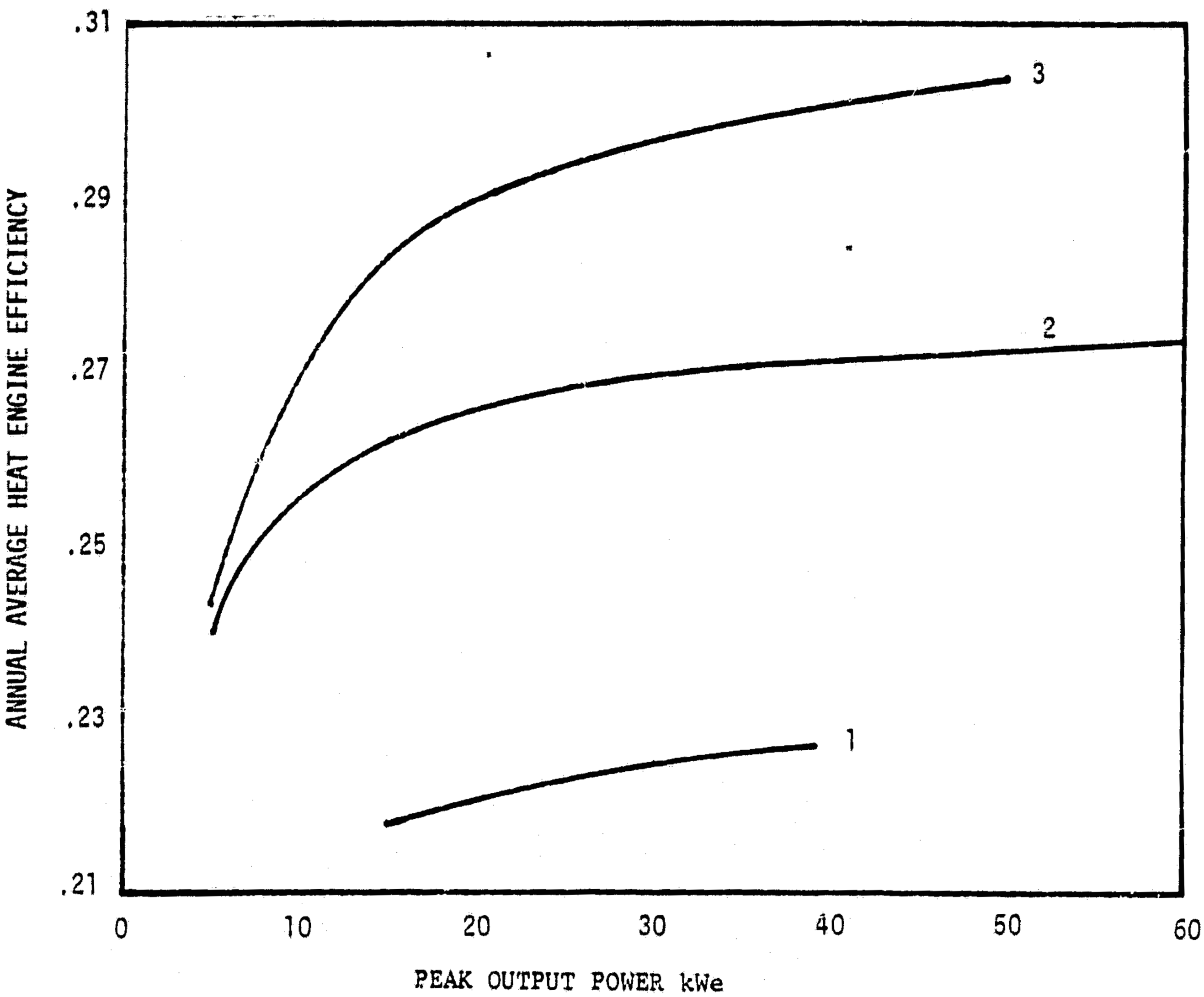


FIGURE 3-4 PEAK OUTPUT POWER vs EFFICIENCY
100^oC CONDENSING TEMPERATURE

4 CONCEPTUAL DESIGN

The JCE recommendation of steam inlet pressure of 17.2×10^6 N/m^2 and temperature of 677°C was accepted by NASA for use as the concept design point. NASA then directed that the system be sized for a 80 kW thermal input (relative input level 1.00 in Table 3-1) with the kW electrical output dependent on the conversion system efficiency. The 80 kW is defined as heat added to the engine working fluid in the steam generator (solar) receiver).

4.1 SIMPLE CYCLE CONCEPT

The simple cycle consists of an expander, water pump and electrical generator which are dish mounted and a heat rejection system which is ground based. Figure 4-1 is an engineering drawing of the dish mounted components.

4.1.1 EXPANDER

Further studies involving expander configuration continued after the selection of a system design point. A method of counter balancing a single cylinder engine was developed permitting the consideration of a single cylinder configuration. Additionally studies were conducted on the effect of varying both the stroke and the diameter of the cylinder to obtain maximum efficiency. Figure 4-2 gives results at various expansion ratios for a single cylinder, simple cycle engine with a square displacement, i.e., stroke equal to diameter. At the concept design point, this configuration has an advantage over multiple cylinder configurations (Figure 3-2) because of reduced friction and thermal losses.

The expander concept design therefore is a single cylinder uniflow simple cycle engine which operates at a nominal speed of 3600 rpm. At the selected expansion ratio of 17.1, cylinder diameter and stroke are both 7.62 cm (3 in) for a displacement of 346 cm^3 (21.2 in^3). Engine operation is the same as previously employed in the very reliable automobile engine. Steam enters the steam chest and passes into the clearance volume as the impulse valve is actuated by the rising piston. The descending piston uncovers exhaust ports allowing the steam to exhaust and pass to the condenser.

The cylinder has a screw-on head made from a 316L stainless steel casting. The head contains the steam inlet fitting, steam chest, and an impulse steam admission valve assembly consisting of carbide valve and seat and a Rene' 41 valve return spring. The stellite actuator is screwed into the top of the piston. The admission valve is capable of delivering the required steam flow with short cut-off without lubrication.

The piston is two-piece with a 4130 steel upper section and a ductile iron lower section. Piston ring lubrication is accomplished by injecting oil directly into the region of ring travel. In this region, steam temperature is sufficiently reduced so that required lubrication is obtained without oil deterioration. The 8620 steel connecting rod is joined to the piston with a needle bearing and to the crankshaft with a sleeve bearing.

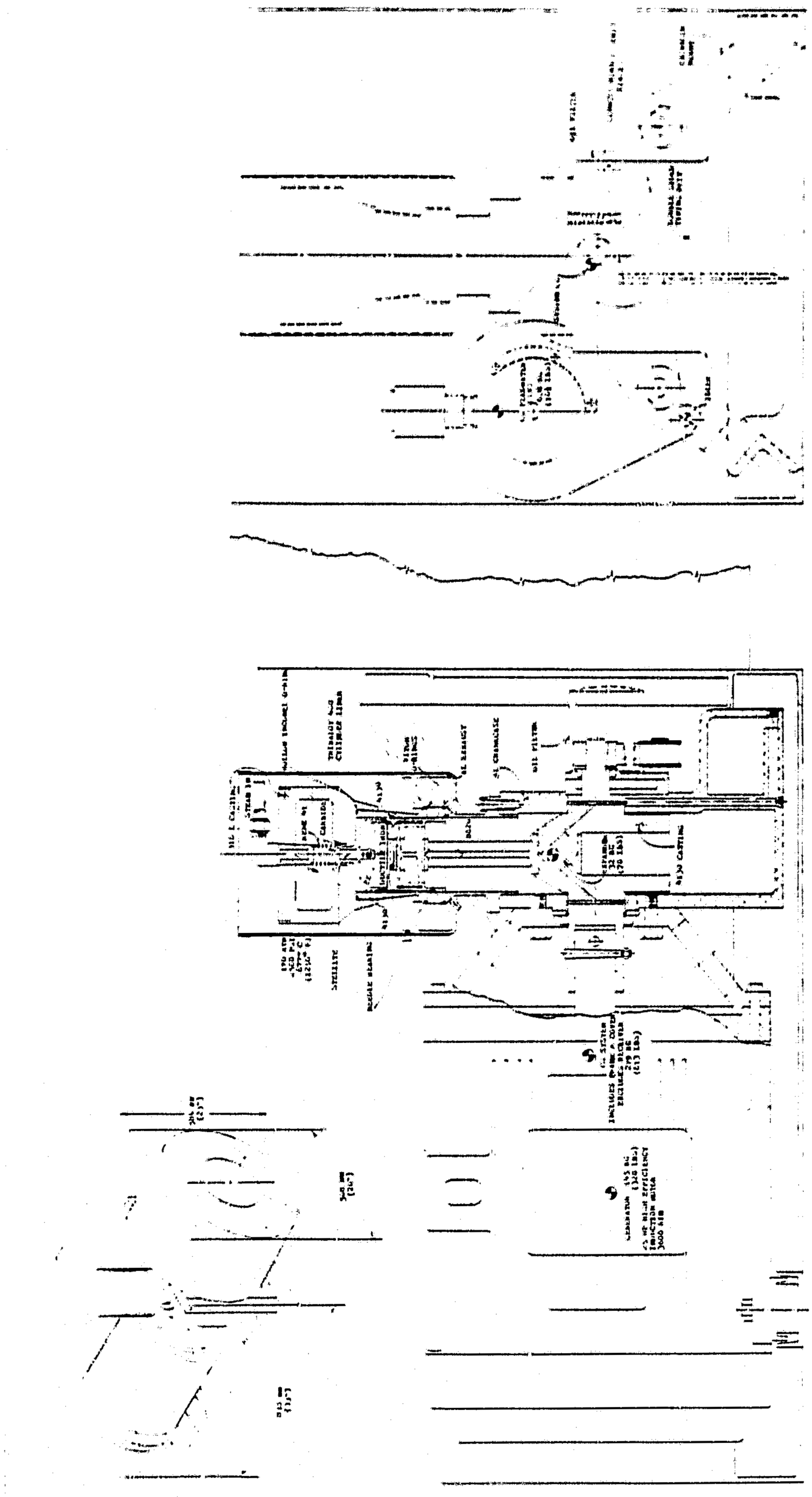


FIGURE 4-1 SIMPLE CYCLE CONCEPT—LAYOUT OF DISH MOUNTED COMPONENTS

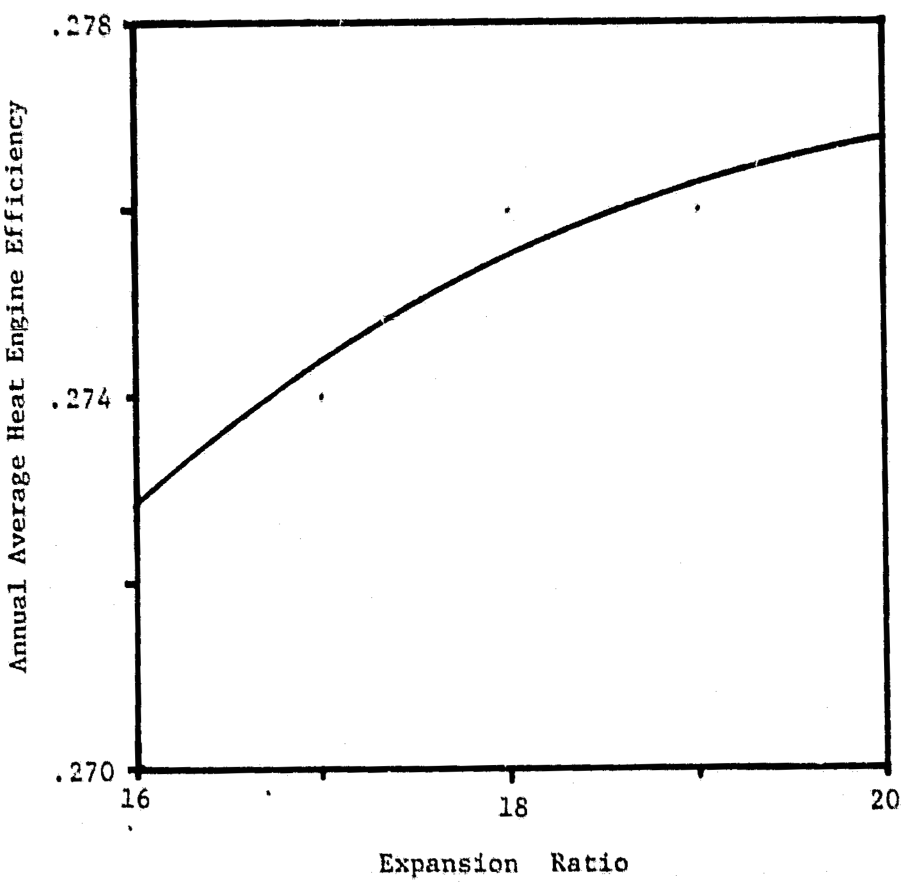


FIGURE 4-2 SIMPLE CYCLE SINGLE CYLINDER EXPANDER STUDY
80 kW THERMAL INPUT; 100°C CONDENSING TEMPERATURE
PRESSURE = 17.24×10^6 N/m^2; TEMPERATURE = 677°C

The Tenzaloy aluminum crankcase contains a reservoir of oil for lubrication of the crankshaft connecting rod bearing. Mounted externally to the crankcase and belt driven by the crankshaft are two counter-rotating counter-balance shafts. These allow the reciprocating mass of the piston and connecting rod to be perfectly counterbalanced so that the expander will be vibration free.

4.1.2 WATER PUMP

The pump for the solar thermal electric application is identical to the unit used on the JCE automobile engine. It is mounted to the crankcase and is belt driven by the crankshaft. Pump efficiency is estimated to be 90 percent.

4.1.3 ELECTRIC GENERATOR

The electric generator for the Carter concept is a three phase, 220 volt, 3600 rpm, high efficiency, squirrel cage induction motor. The motor-generator will be directly coupled to the expander crankshaft and will serve as the expander flywheel and starter. The motor is briefly tied to the utility line at start-up and is reconnected to the utility line when the expander reaches 3600 rpm. The expander then tends to drive the induction motor faster than synchronous speed, causing the motor to start generating, automatically feeding power back into the power grid. Frequency of the generated power is that of the excitation, i.e. line frequency. Output voltage is stabilized by line voltage and will be slightly higher than line voltage. Expander speed is limited by the power it generates, i.e. the higher above synchronous speed the generator is driven, the higher the electric power that is generated.

Generator efficiency is shown in Figure 4-3.

4.1.4 HEAT REJECTION SYSTEM

The heat rejection system consists of the condenser, water tank, and associated pumps. The condenser system consists of a lead and brass condenser, condenser fan, a fiberglass stack and associated framing. Exhaust steam is transported from the dish mounted expander through flexible steam lines to the condenser. Air is induced to flow through the condenser by the stack. Figure 4-4 is a layout of the ground based condenser stack and water tank.

Pressure differential across the condenser is equivalent to the difference in density between the ambient air and the warm air in the stack multiplied by the height of the stack. Assuming that the condensing capacity of condensers of equivalent design and thickness follow the relationship

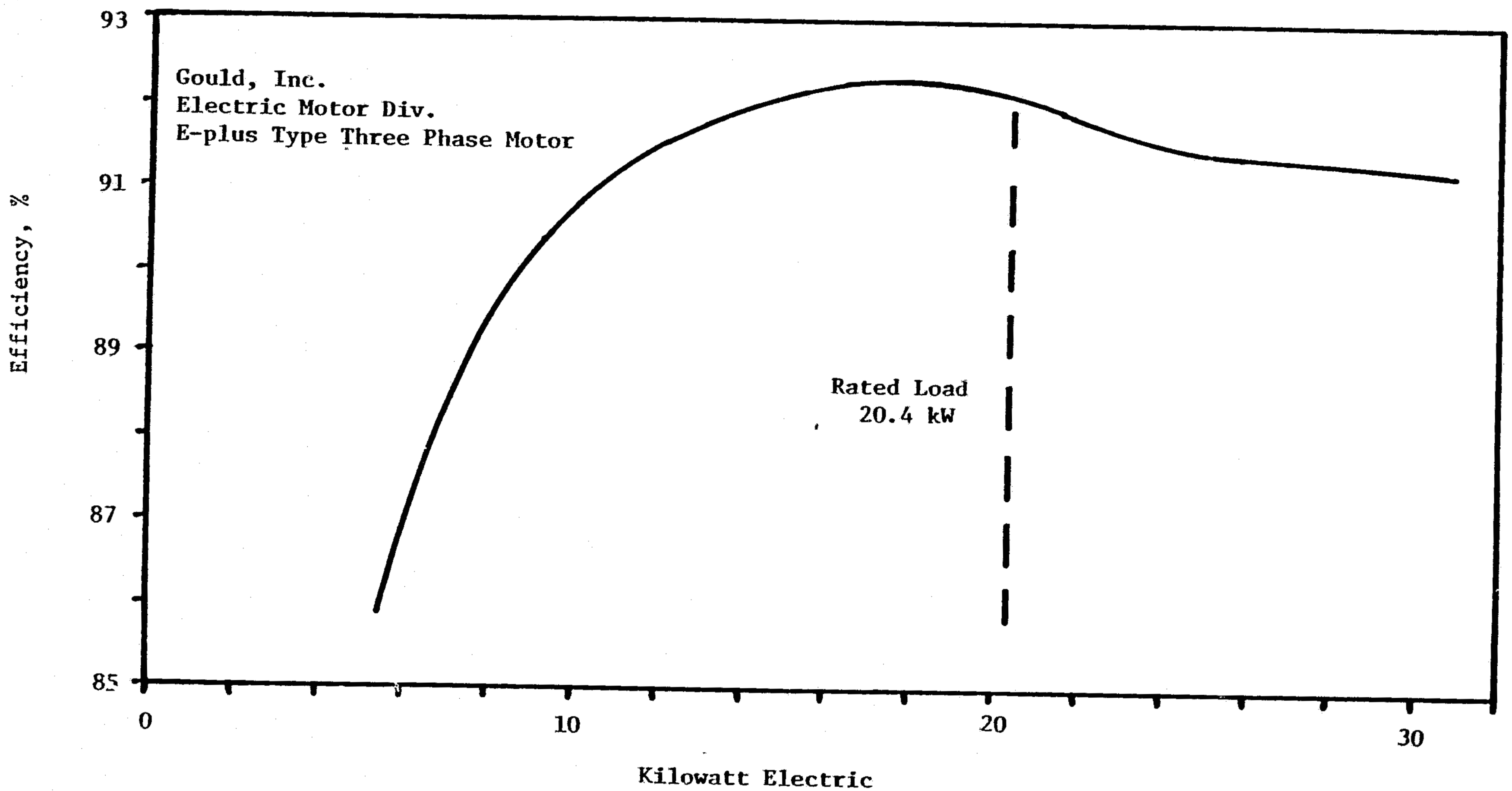


FIGURE 4-3 Electric Generator Efficiency

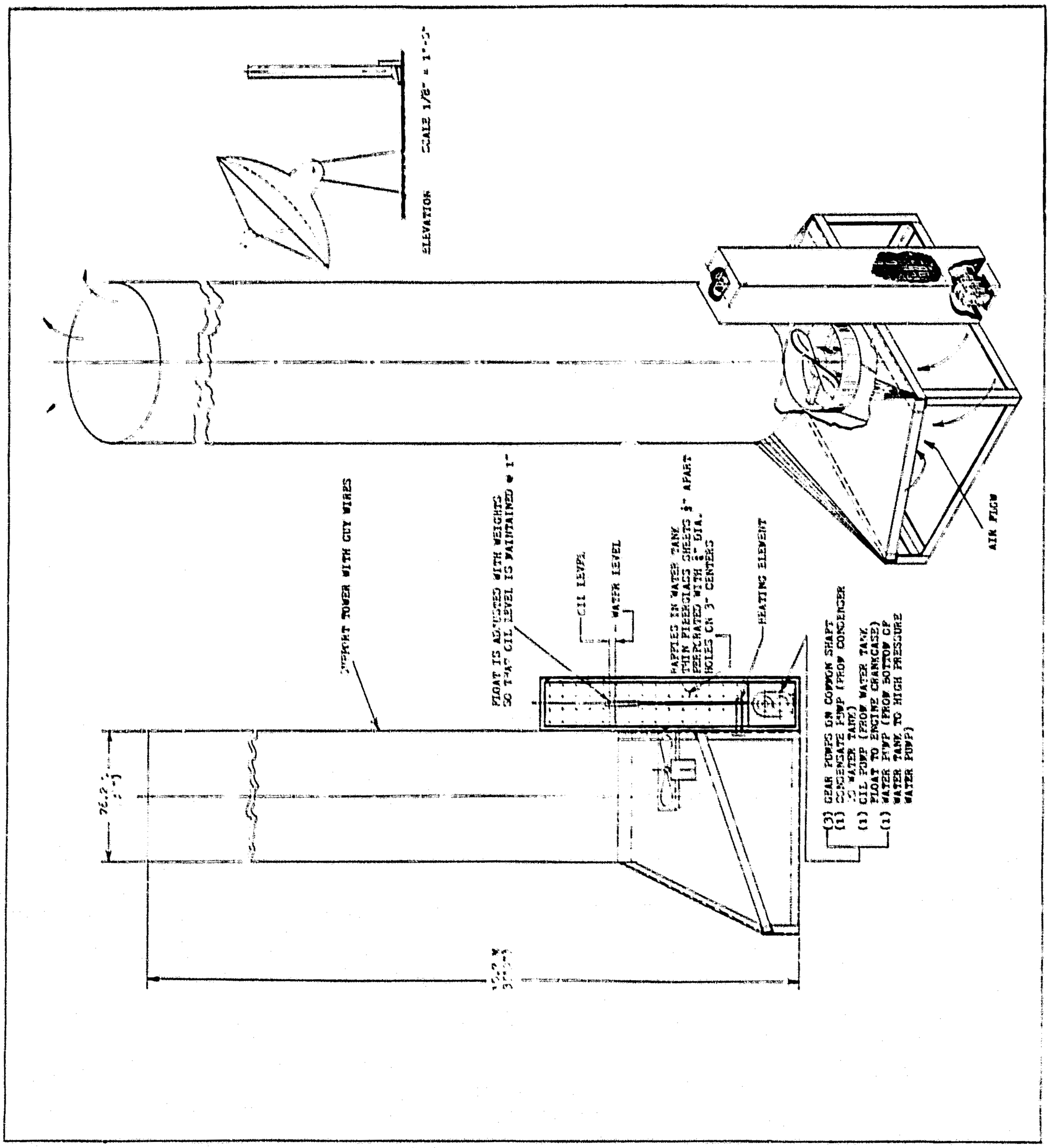


FIGURE 4-4 LAYOUT OF HEAT REJECTION SYSTEM

$$\frac{C^1}{C} = \frac{A^1}{A}\sqrt{\frac{\Delta P^1}{\Delta P}}$$

where C = condensing capacity

A = condenser area

Δ P = pressure drop across condenser

The capacity of the condenser can be estimated by comparing it to the JCE automobile condenser. The temperature difference across the condenser is expected to be 30°C or more, the stack is about 10 meters high and the condenser has an area of 1.39 m^2.

Density of air of 30 % relative humidity
@ 35°C (95°F) = 1.105 kg/m^3 (.069 lb/ft^3)
@ 65°C (149°F) = 1.025 kg/m^3 (.064 lb/ft^3)

ΔP^1 = (1.105 - 1.025) x 10 = .8 kg/m^2 = 7.89 N/m^2
ΔP (automobile) = 186.75 N/m^2 (.75 in. water)
A^1 = 1.39 m^2 (15 ft^2)
A (automobile) = .65m^2 (7 ft^2)
C (automobile) = 5.49 x 10^8 J/k (520,000 BTU/h)

$$\therefore C^1 = 5.49 \times 10^8 \times \frac{1.39}{.65} \times \sqrt{\frac{7.89}{186.75}} = 2.41 \times 10^8 \text{ J/h}$$

The peak heat load from the system will be

(H7 - H8) x Q1 = (.031 - .007) x 10^8 J/kg x 79.45 kg/h
= 1.91 x 10^8 J/h

Therefore, the condenser fan will not be required to operate under normal conditions.

The condensate is drawn from the bottom of the condenser and pumped into an adjacent fiberglass water tank. The tank is fitted with vertical perforated fiberglass sheets which act as collectors for the lubricating oil which is mixed with the water condensate. The oil rises to the surface of the water where it is drawn off by a pump and transferred to the expander crankcase. Water in the tank is pumped to dish mounted high pressure pump as required.

The three ground based gear pumps, i.e., condensate, oil lift, and water lift are driven by an electrically driven motor. The power requirement of this motor is minimal and is largely due to gear friction since little work is actually being done by the pumps. The total power requirement for accessories which consist of the condenser fan and the auxiliary pumps was conservatively estimated in the analysis to be 1/8 x G x R3/15 kWe. The maximum value of this relationship is .2 kWe at a thermal input of 80 kW.

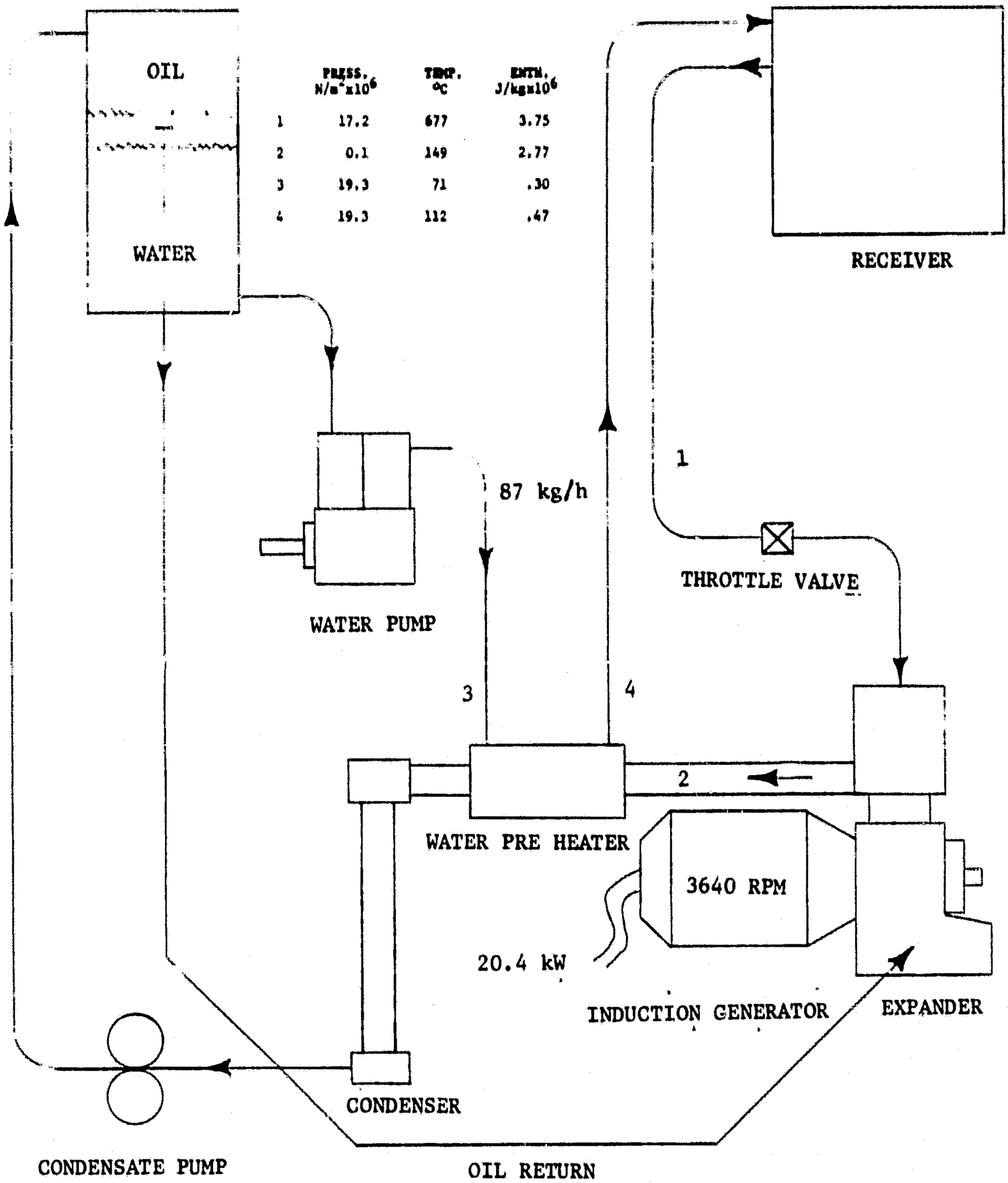
4.1.5 SYSTEM PERFORMANCE

Figure 4-5 is a flow diagram of the simple cycle system conceptual design. Table 4-1 lists system parameters and efficiencies at various values of relative thermal input (RTI). Thermal input is 80 kW at RTI of 1. The summation of annual power production and annual average efficiencies are derived from the nominal annual duty cycle given in Table 3-1. Inlet steam temperature is 677°C (1250°F) and condensing temperature is 100°C.

Table 4-1 Simple Cycle System Performance

REL. THERMAL INPUT RTI	OUTPUT POWER G kW	STEAM PRESS P0 $N/m^2 \times 10^6$	WATER FLOW RATE Q1 kg/h	FEEDWATER TEMP. T9 °C	ACCESSORY POWER P3 kW	WATER PUMP POWER P4 kW	GENERATOR EFF E1	EXPANDER EFF. E2	HEAT ENGINE EFF. E3	POWER CONV. EFF. E4
1.00	20.4	17.235	87.17	112	.171	.566	.921	.923	.279	.257
.95	19.5	16.759	83.08	112	.156	.514	.922	.919	.279	.257
.92	18.9	16.380	80.81	113	.149	.492	.922	.917	.278	.257
.90	18.5	16.118	79.00	113	.142	.477	.923	.915	.278	.257
.86	17.6	15.539	74.91	114	.127	.440	.923	.911	.278	.257
.83	17.1	15.250	73.09	114	.119	.410	.922	.908	.278	.256
.76	15.6	14.533	66.74	116	.104	.358	.921	.900	.277	.255
.65	13.1	13.181	57.20	119	.075	.283	.917	.884	.274	.251
.50	9.6	11.327	43.58	124	.037	.186	.906	.849	.267	.242

Annual power production	53072 kW-h
Annual average heat engine efficiency	.276
Annual average power production efficiency	.254



	PRESS. $N/m^2 \times 10^6$	TEMP. °C	ENTH. $J/kg \times 10^6$
1	17.2	677	3.75
2	0.1	149	2.77
3	19.3	71	.30
4	19.3	112	.47

FIGURE 4-5 SIMPLE CYCLE FLOW DIAGRAM - 80 kW INPUT

4.1.6 SYSTEM WEIGHT

The estimated weight of various system components are listed in Table 4-2. The total weight of dish mounted components is 278.3 kg while ground based components weigh 304.2 kg.

Table 4-3 Estimated Simple Cycle System Weight

Dish Mounted Components	Wt. kg	
Generator	145.3	
Expander	31.2	
Water pump	5.0	
Feed water heater	4.5	
Controls	3.2	
Throttle valve	3.2	
Steel frame	31.1	
Sheet metal cover	40.0	
Misc. (belts, pulleys, safety valve, brackets, lines, etc.)	15.0	
	278.3	
Ground Based Components		
Condenser	40.9	
Fiberglass water tank	27.2	
Fiberglass stack	136.2	
Accessory pump and motor	9.1	
Frame	50.0	
Miscellaneous	40.8	
	304.2	
Total		582.5

4.2 COMPOUND-REHEAT CYCLE CONCEPT

The compound-reheat cycle concept is similar to the simple cycle concept, the major difference being that a second cylinder is added to the expander. Figure 4-6 is a layout of the two cylinder compound-reheat concept.

4.2.1 EXPANDER

The expander is a two cylinder, compound-reheat cycle reciprocating engine which operates norminally at 3600 rpm. The counter-flow high pressure cylinder has a diameter and stroke of 4.23 cm (1.67 in) and a displacement of 59.40 cm^3 (3.67 in^3) while the low pressure cylinder has a diameter of 12.70 cm (5 in) and a stroke of 7.62 cm (3 in) with a displacement of 965.28 cm^3 (58.91 in^3). The two cylinder configuration results from studies of one low pressure cylinder vs two low pressure cylinders similar to the friction and heat loss studies that resulted in a single cylinder

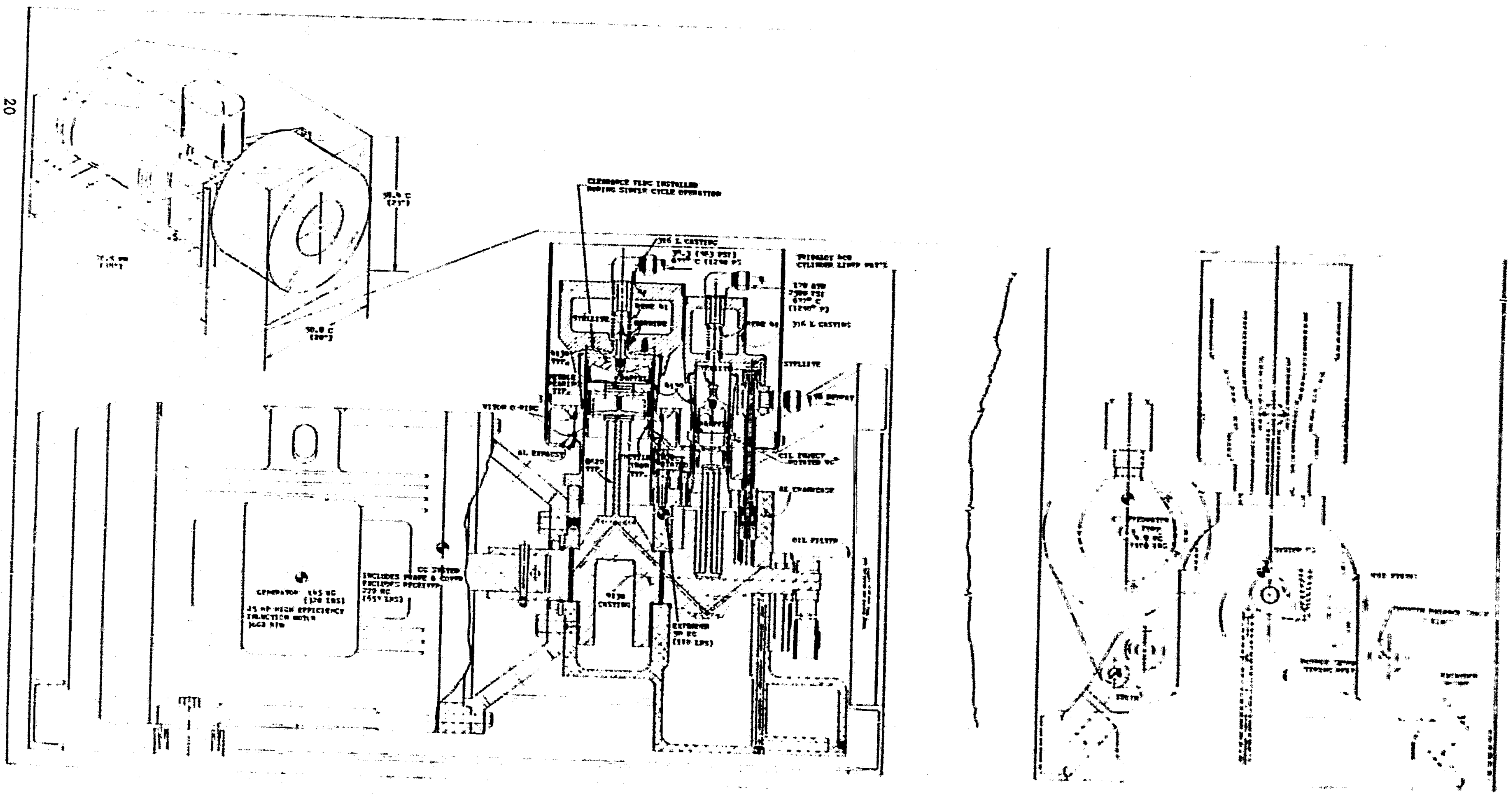


FIGURE 4-6 COMPOUND REHEAT CYCLE CONCEPT-LAYOUT OF DISH MOUNTED COMPONENTS

configuration for the simple cycle. The study results are then subject to practical limits on minimum diameter by the pressure impulse considered tolerable and on stroke by the piston velocity.

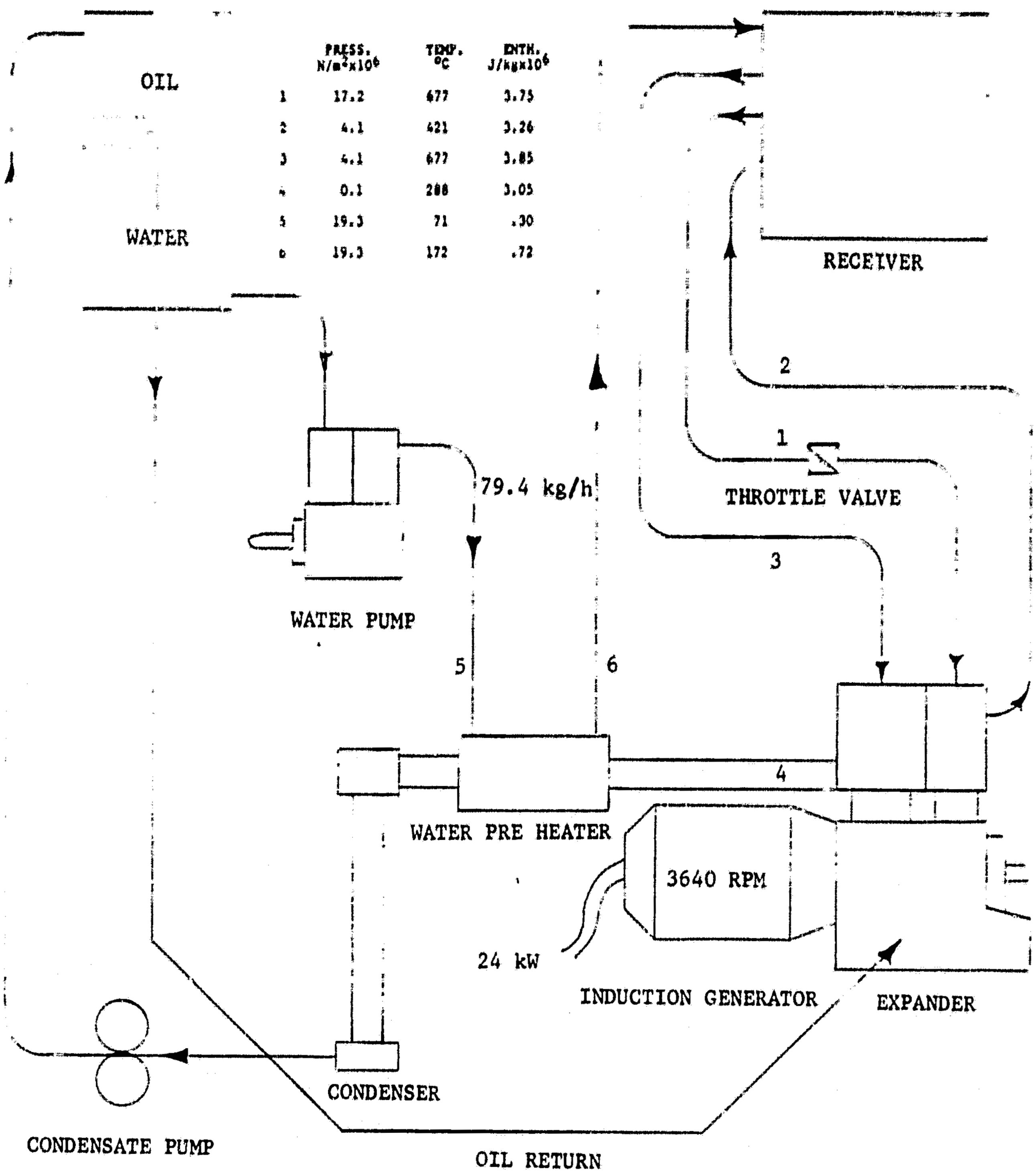
Steam enters the expander through the H.P. inlet fitting into the H.P. steam chest. It passes through the H.P. admission valve when the piston is near top dead center. Steam is expanded 3.25 times its volume as the H.P. piston descends. The crankshaft has a cam lobe to lift the H.P. cylinder exhaust valve. Steam is driven out of the H.P. exhaust valve opening near the top of the H.P. cylinder as the H.P. piston rises. The steam is circulated back through the thermal receiver where it is reheated to the original steam inlet temperature. The reheated steam passes to the L.P. steam chest. After entering through the L.P. admission valve, the steam is again expanded to 8.60 times its volume. It then exhausts out the bottom of the L.P. cylinder after the L.P. piston uncovers exhaust ports.

4.2.2 SYSTEM PERFORMANCE

Figure 4-7 is a flow diagram of the compound-reheat cycle system conceptual design. Table 4-2 lists system parameters and efficiencies at various values of relative thermal input (RTI). Thermal input is 80 kW at RTI of 1. The summation of annual power production and annual average efficiencies are derived from the nominal annual duty cycle given in Table 3-1. Inlet steam temperature is 677°C (1250°F) and condensing temperature is 100°C.

4.2.3 SYSTEM WEIGHT

The only variation in system weight from that given for the simple cycle concept is in expander weight which would increase to 50 kg. This would increase dish mounted weight to 296.5 kg and total system weight to 600.7 kg.



	PRESS. $N/m^2 \times 10^6$	TEMP. °C	ENTH. $J/kg \times 10^6$
1	17.2	677	3.75
2	4.1	421	3.26
3	4.1	677	3.85
4	0.1	288	3.05
5	19.3	71	.30
6	19.3	172	.72

FIGURE 4-7 COMPOUND REHEAT CYCLE FLOW DIAGRAM-80 kW INPUT

Table 4-2 Reheat Cycle System Performance

REL. THERMAL INPUT RTI	OUTPUT POWER G kW	STEAM PRESS P0 $N/m^2 \times 10^6$	WATER FLOW RATE Q1 kg/h	FEEDWATER TEMP. T9 °C	ACCESSORY POWER P3 kW	WATER PUMP POWER P4 kW	H.P. EXPANDER EFF E5	L.P. EXPANDER EFF. E6	HEAT ENGINE EFF. E3	POWER CONV. EFF. E4
1.00	23.88	17.235	79.45	172	.201	.507	.93	.82	.33	.30
.95	22.53	16.566	75.36	173	.179	.507	.92	.81	.32	.30
.92	21.77	16.304	73.55	174	.171	.507	.92	.80	.32	.30
.90	21.25	15.973	71.73	174	.164	.417	.92	.80	.32	.30
.86	20.22	15.546	69.01	176	.149	.417	.91	.79	.32	.29
.83	19.37	15.146	66.28	177	.134	.417	.91	.78	.32	.29
.76	17.50	14.222	60.84	180	.112	.313	.90	.76	.31	.29
.65	14.50	12.912	52.66	186	.082	.313	.89	.72	.30	.28
.50	10.19	10.851	41.31	199	.045	.164	.86	.64	.28	.26

Annual power production	61367 kW-h
Annual average heat engine efficiency	.313
Annual average power conversion efficiency	.288

5 POWER CONVERSION SUBSYSTEM OPERATION

The following is a discussion of subsystem operation under varying conditions. A schematic of the control system is given in Figure 5-1.

5.1 START-UP

As the temperature in the receiver starts to increase, low pressure hot water and steam flow into the expander steam chest and from there into the exhaust manifold through a temperature controlled blowdown solnoid valve. When the temperature of the steam flowing through the blowdown valve reaches 150°C, the valve will close allowing steam pressure to build up in the receiver steam tubes and expander steam chest. An expander control which monitors both steam pressure and rpm turns the starter-generator on for one second when the steam pressure is greater than 2.75×10^6 N/m^2 and the rpm is less than 100. When the rpm is equal to or greater than line frequency (approximately 3600 rpm) the generator is connected to the utility line. Since the generator will also act as a motor if the rpm drops below 3600 rpm, the generator is disconnected from the utility line if the rpm drops below line frequency. Maximum rated output of the generator is reached at approximate 3700 rpm while still maintaining the same frequency as the utility line.

If the utility line loses power, the generator will stop generating unloading the expander and allowing the expander to go into an overspeed (rpm greater than 3700). If an overspeed should occur for any reason, a centrifugally operated throttle valve will close and hold the rpm between 3700 and 4500 rpm. Under these conditions, steam pressure will rise to 18.6×10^6 N/m^2, and open the safety valve which is tied into the steam line upstream of the throttle valve allowing the steam to flow through some finned tubing located in the condenser stack before going into the condenser. Here the steam temperature would drop to a temperature that could be safely handled by the condenser. Since as much as 65 percent of the heat absorbed by the receiver ends up in the condenser and since the condenser with the stack can condense this steam without the need of a condenser fan, this additional heat added to the condensing system could also be handled. This additional heat is in the form of high temperature steam which would increase the air temperature in the stack downwind of the main condenser, resulting in an increased natural draft. During this period of overspeed, a battery operated servo would continue to cause the collector to track the sun. If after 30 minutes, utility power had not come back on, then the collector servo would move the receiver out of the sun and allow the receiver to cool down. In this way, if utility power is off for less than 30 minutes, the solar generator will be ready to produce power as soon as the utility lines come back on.

5.2 SHUT-DOWN

As heat flow to the receiver is reduced, water flow is also reduced so that the steam temperature will remain near its design point. As the water flow is reduced so will the steam pressure drop, so that at some

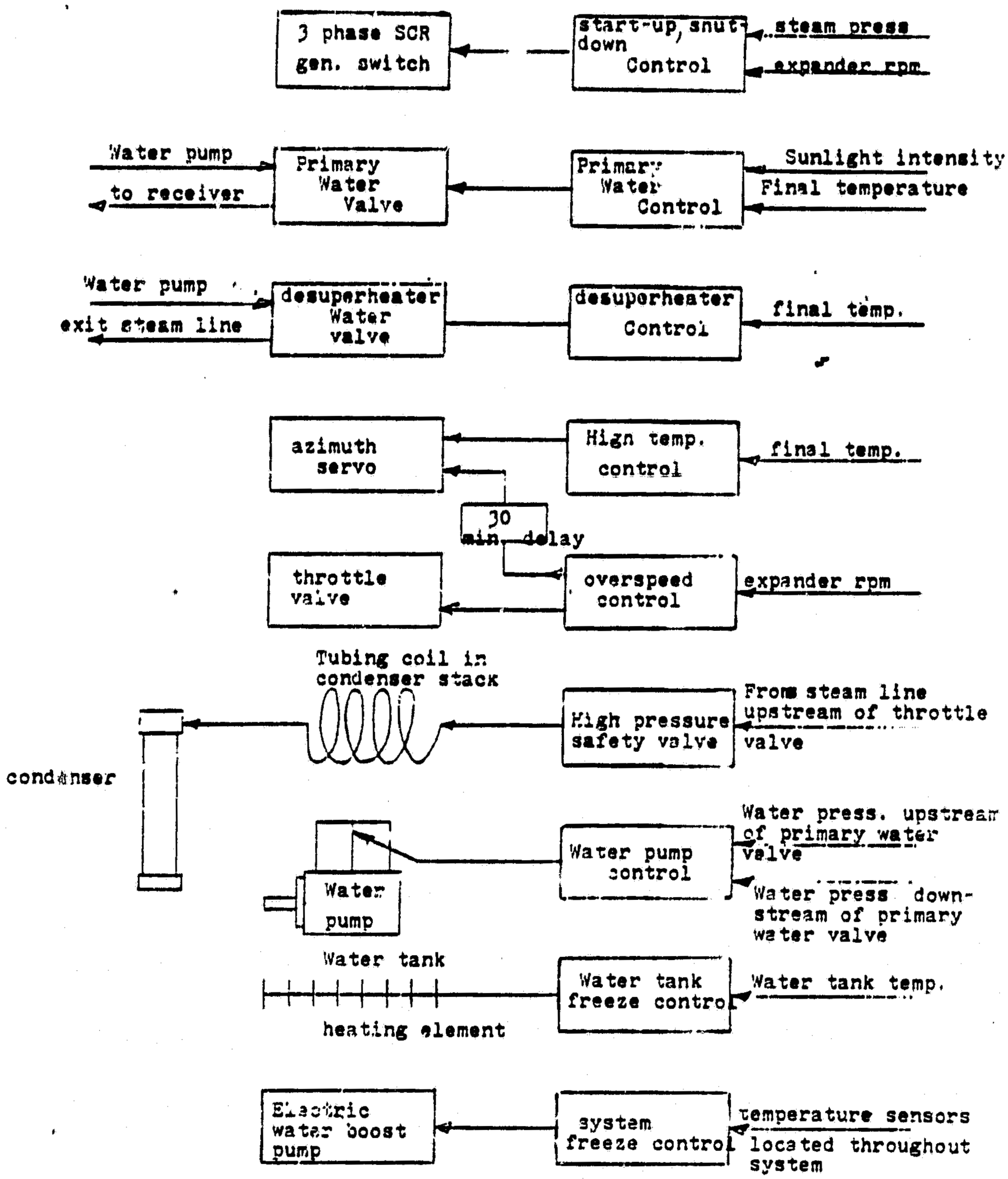


Figure 5-1 Control System Schematic

pressure the expander rpm will drop below 3600 rpm and will be disconnected from the utility line. The pressure and rpm will continue to drop until at some pressure (approximately 2.75×10^6 N/m^2) the expander will no longer run. At this time steam flow through the receiver and expander stops. Then as the pressure and temperature continue to drop, a special accumulator initially charged to approximately $.7 \times 10^6$ N/m^2 starts to unload a given quantity of water into the receiver, priming it for the next startup.

5.3 RAPID TRANSIENTS

There will be transients in heat flow to the receiver due to the passage of clouds between the sun and the collector. These transients may be rapid and are in addition to normal transients due to change in position of the sun relative to the collector. Control of temperature of the steam exiting the receiver during these rapid transients will require a rapid response in control of feedwater flowing thru the receiver. A system similar to the temperature control system used in the JCE automobile engine will be required. This will consist of four temperature feedbacks to the control unit and five water injection nozzles in the receiver.

The location of the temperature pick-ups for information feedback and the location and flow rate of the water injections are given in Table 5-1. The control system is expected to provide temperature control of $\pm$ 15°C during transients going from 5 percent to 100 percent of full power in 10 seconds.

Table 5-1 Location of Temperature Feedback and Water Injection

Feedback	Location	Temperature °C Off	Temperature °C On	Injection	Location	Full Flow
1	.33L	343	538	1	0 (start)	100% Primary Water
				2	.25L	33% " "
2	.67L	343	538	3	.4L	33% " "
3	.9L	566	649	4	.75L	20% " "
4	L(exit)	671	682	5	.95L	10% " "

5.4 ORIENTATION DURING SOLAR TRACKING

The system tracking method has been defined as a two-axis elevation-asthmuth type in which a reference point on the system which is on the bottom of the dish in the a.m. is also at the bottom in the p.m.

The expander is designed to be mounted to the dish with the crankcase sump mounted toward the dish. At start-up, the cylinders are vertical with the crankcase on the bottom. At noon the cylinders will be horizontal but the sump will be on the bottom. At shut-down, the orientation is the same as at start-up. Therefore crankcase oil and any condensate collecting in the oil are always toward the bottom of the expander and can be pumped out of the expander.

5.5 FREEZE PROTECTION

Distilled water should be used in the system, therefore a mechanical-electrical freeze protection system must be employed. The system would require an insulated water tank, insulated water and steam lines between the engine and ground base condenser and water tank, an electric thermostat control-led water tank heater, a transfer water pump and thermocouples, located at points throughout the system that would likely freeze first.

A description of how the system works is as follows: a thermostatic controlled electric heater holds the water tank above 4°C. A separate thermostat which monitors temperatures at several points throughout the system turns on the transfer water pump whenever any temperature drops to 2°C. The transfer pump moves 4°C water from the water tank through the water line feeding the feedwater pump, through the check valves on the feedwater pump, through the receiver, through the expander cylinder heads, through the temperature operated blowdown valve into the exhaust manifold and then back into the water tank by way of the condensate pump. A small orfice at the bottom of a "U" trap located in the exhaust manifold catches the water and allows it to go straight to the condensate pump, bypassing the condenser. These pumps operating off one motor, circulate water through the system and keep it from freezing.

Should anything fail to operate as described, the system could freeze up. Damage to the system can be eliminated by the use of a flexible water tank, expansion bellows located at key points in the tubing and controls, and high pressure tubing capable of forcing the ice to extrute through the tubing. An operator assisted thaw out period would be required in the event of a freezeup.

6 IMPLEMENTATION ASSESSMENT

6.1 SYSTEM DEVELOPMENT STATUS

The proposed JCE steam Rankine power-electrical conversion system (simple cycle) requires no new technology development for mass production. The simple cycle expander is similar to a 2 cycle I.C. engine in that the exhaust ports are located in the cylinder and uncovered by the piston. Pistons, rings, bearings, crankshaft and crankcase are similar to a small diesel engine. The induction generator is nothing more than a mass produced 25 hp induction motor. The inlet valve is made from carbide and should present no problem for mass production, as many intricate parts are now mass produced out of carbide. The controls which require thermocouples, solnoid valves, pressure transducers, rpm indicators, and electronic printed circuit boards are all presently being mass produced.

The expander conceptual design is specifically based on the simple cycle JCE automobile engine. The automobile engine has evolved through three generations of design and has numerous hours of successful operation at a design temperature of 566°C. The technology barrier to increased steam temperatures is the oil lubrication system and control of oil temperatures to within acceptable limits. This is accomplished by locating the piston rings well down on the piston.

A simple cycle development program would require three operating temperature increments beginning at 538°C and progressing to 677°C. The estimated system performance at each step is given in Table 6-1.

Table 6-1 Simple Cycle Development Schedule

CYCLE Cycle	STEAM TEMP. T1 °C	STEAM PRESS P1 $N/m^2 x10^6$	MAX HEAT ENG. EFF. Peak E3	MAX POWER CONV. EFF. Peak E4	AVG HEAT ENG. EFF. Avg. E3	AVG POWER CONV. EFF. Avg. E4
Simple	538	15.99	.237	.218	.235	.216
Simple	607	16.81	.258	.238	.256	.236
Simple	677	17.235	.279	.257	.276	.254

It is anticipated that a compound reheat cycle expander development would take place in six discreet steps. In the first three steps, the expander would operate on a simple cycle, i.e., only the low pressure cylinder would be active. The H.P. cylinder is deactivated by removing the lift rod from the exhaust valve and capping off the steam inlet fitting. The expansion ratio of the L.P. cylinder is increased by installing a plug to decrease its clearance volume. In this case, high pressure steam is admitted to the L.P. cylinder where it expands to 30 times its volume as the piston descends. As before, steam exhausted by the L.P. cylinder passes to the condenser. The capability for initial operation in the simple cycle mode will allow perfection of the balancing system required for a single cylinder engine and would allow for a progression from 538°C, the operating temperature of the existing automobile expander, to 677°C, the target operating temperature. After completion of the simple cycle development, the expander would then be changed to the reheat cycle configuration and the three temperature steps repeated. The estimated system performance at each of the six development steps is given in Table 6.2.

Table 6-2 Reheat Cycle Development Schedule

CYCLE Cycle	STEAM TEMP T1 °C	STEAM PRESS P1 $N/m^2 x10^6$	MAX.HEAT ENG. EFF. Peak E3	MAX.POWER CONV. EFF. Peak E4	AVG. HEAT ENG. EFF. Avg. E3	AVG. POWER CONV. EFF. Avg. E4
Simple	538	15.215	.248	.228	.240	.221
Simple	607	16.497	.267	.248	.258	.238
Simple	677	17.235	.286	.264	.277	.255
Reheat	538	16.539	.290	.260	.277	.255
Reheat	607	17.125	.310	.280	.296	.272
Reheat	677	17.235	.330	.300	.313	.280

6.2 DURABILITY AND MAINTENANCE

The proposed steam system should be very durable. Because it is a closed system, oil can be injected into the piston ring belt greatly increasing piston ring and cylinder wall life. The oil that is used for ring and cylinder lubrication and that passes out the exhaust ports is separated from the water. Since the exhaust steam temperatures and mean piston and ring temperatures are below the oxidation temperature of the special synthetic oil used, and since there are no acids formed as a result of combustion as in an I.C. engine, oil should not have to be changed but once every 2 years. The expander could run 6 months or a year without adding oil. Any oil that may pass through the receiver turns into a graphic like substance which settles out in the water tank. It does not remain in the receiver tubes nor does it damage the system in any way.

Main bearings, rod bearings, and wrist pin bearing life should be at least as good as stationary diesels, which have run continuously over two years. Stress levels in the high temperature parts are equal or less than boiler standards for 100,000 hour life. The potential is possible for the system to have a 10 year life between overhauls with many components lasting 20 to 30 years. Experience and running time are needed before projection can be made as to which components might fail first, what components are overdesigned and could be reduced, and as to new and better designs.

Average annual operating and maintenance cost should run about 5 percent of the unit costs per year with major overhauls costing 50 percent of the units costs. These costs estimates are approximately the same as that allowed for stationary diesel engines.

6.3 MANUFACTURING COST

Preliminary estimates of power conversion sybsystem unit manufacturing cost in 1978 dollars can be made using cost per pound analysis. Using graphs and curves developed showing the cost per pound of various manufactured items similar in complexity, material costs and quantity, an estimate can be made for the production costs at rates of 100, 1,000, 10,000 and 100,000 units annually. This is given in Table 6-3. An earlier analysis indicated the complete unit weight including ground base components for the simple cycle @ 582 kg (1283 lb) and for the reheat cycle at 601 kg (1323 lb).

Table 6-3 Cost Estimates of Power Conversion Subsystem

	Simple	Reheat
100 units @ $10/lb	$12,830	$13,230
1,000 units @ $5/lb (approximate cost/lb for Gasoline and diesel generators)	6,415	6,615
10,000 units @ $3.5/lb (approximate cost/lb for small tractors)	4,490.5	4,630.5
100,000 units @ $2.5/lb (approximate cost/lb for standard automobiles	3,207.5	3,307.5

7 DISCUSSION OF RESULTS

The program being reported here has arrived at the concept definition of a steam Rankine power conversion subsystem which, when supplied with 80 kW of solar thermal energy, can produce high quality electrical energy with 30 percent efficiency at an estimated cost of $3,307. This is an estimated cost of $138 per kW of maximum power output. It is not known if this represents an economically feasible system or not. The cost of a complete system must be considered and compared to the total cost of producing electricity by other means.

What is known is that the steam Rankine system produces about 30 percent high quality energy in the form of the electric output and rejects about 60 percent in the condenser and the remaining 10 percent in the generator and other external losses. The heat rejected in the condenser represents primarily the steam heat of condensation at a constant temperature of 100°C. The equipment required to reject this heat is considerable. In fact, over half the estimated cost of the power conversion subsystem is for equipment to extract this heat from steam and transfer it to the atmosphere. It would appear that the economic feasibility of the system would be greatly enhanced if the condenser heat could be utilized in some form of total energy system.

The small, distributed solar systems may have unique advantages for use in the total energy mode. It is assumed that the solar systems would be located in close proximity to energy users such as houses, business, etc. If these users can utilize the 100°C reject heat for water heating, space heating, and/or air conditioning the real efficiency would be quite high. If the 60 percent reject energy should become entirely useful the realized efficiency of the solar conversion would be 90 percent. Even for partial utilization efficiencies of 60 percent and greater can be envisioned.

8 SUMMARY OF RESULTS

A study was undertaken to define a concept of a nominal 15 kWe steam Rankine power conversion sybsystem for a point focusing distributed solar thermal-electric system.

A mathematical model of the Jay Carter Enterprises steam engine concept was developed in order to conduct a parametric analysis covering a broad range of steam inlet pressure and temperature as well as variations in engine configuration and condensing temperature. As a result of the parametric study, a design point at 677°C (1250°F) and 17.2×10^6 N/m^2 (2500 psi) was selected as maximum inlet steam temperature and pressure with a maximum solar thermal power of 80 kW.

Conceptual designs were developed for a simple cycle and a compound reheat cycle engine concept. The simple cycle concept employs a single cylinder simple cycle uniflow expander. The reheat concept employs a 3600 rpm, compound- reheat cycle reciprocating expander containing a single high pressure cylinder and a single low pressure cylinder with the steam being reheated to the maximum inlet temperature after expansion by the high pressure cylinder. Both expanders have counter-rotating counter-balance shafts to perfectly balance the expander and eliminate vibration. Except for the difference in expanders the two concepts are identical. Power generation is accomplished with a three-phase induction motor directly coupled to the expander and connected electrically to the public utility power grid. Frequency and voltage automatically correspond to line frequency and voltage because of the inherent characteristics of induction motors. The expander, generator, water pump and control system are dish mounted, i.e., connected to the solar receiver.

The steam condenser, water tank, and accessory pumps are ground based. A summary of estimated efficiency, weight, and cost is given in Table 8-1.

The reheat cycle concept has a maximum heat engine efficiency of 33 percent and a maximum power conversion efficiency of 30 percent as compared to the simple cycle efficiencies of 28 percent and 26 percent. The simple cycle power conversion subsystem is considered to be well within the state of the art. The reheat cycle concept will require more development effort and incurs a slightly higher risk than does the simple cycle because the reheat cycle expander is a greater departure from the current JCE automobile expander which is the basis of both expander concepts.

Table 8-1 Summary of Results

Cycle	Max. Heat Engine Eff. Percent	Max. Power Conversion Eff. Percent	Avg. Heat Engine Eff. Percent	Avg. Power Conversion Percent	Wt kg	Cost Dollars
Simple	.279	.257	.276	.254	582	$3,207
Reheat	.330	.300	.313	.280	601	$3,307

APPENDIX A

Definition of Symbols

C6 - Specific volume in H.P. cyl before inlet valve opens

C7 - Specific volume in H.P. cyl after inlet valve opens

C8 - Specific volume in L.P. cyl before inlet valve opens

C9 - Specific volume in L.P. cyl after inlet valve opens

D3 - Diameter of high pressure cylinder

D4 - Diameter of low pressure cylinder

E1 - Generator efficiency

E2 - Expander efficiency

E3 - Heat engine efficiency

E4 - Power conversion efficiency

E5 - H.P. expander efficiency

E6 - L.P. expander efficiency

E7 - Ideal cycle efficiency

G - Electric power

G1 - Peak thermal power

H1 - H.P. inlet enthalpy

H2 - Ideal H.P. exhaust enthalpy

H3 - L.P. inlet enthalpy

H4 - Ideal L.P. exhaust enthalpy

H6 - Actual H.P. exhaust enthalpy

H7 - Actual L.P. exhaust enthalpy

H8 - Feedwater enthalpy

H9 - H.P. clearance volume enthalpy after valve opens

APPENDIX A - Definition of Symbols (cont'd)

L2 - H.P. friction loss

L3 - H.P. thermal loss

L4 - Short circuit loss

L5 - L.P. friction loss

L6 - L.P. thermal loss

M3 - Mass in H.P. cylinder before valve opens

M4 - Mass in H.P. cylinder after valve opens

M5 - Mass flow rate calculated for H.P. cylinder

M6 - Estimated mass flow rate

M9 - Mass flow rate calculated for L.P. cylinder

P1 - Maximum steam inlet pressure

P2 - H.P. cylinder exhaust pressure

P3 - Accessory power

P4 - Estimated water pump power

P5 - Calculated water pump power

P0 - Steam inlet pressure

Q1 - Calculated water flow rate

R(J) Specified relative thermal input

R1 - Calculated relative thermal input

R2 - Ratio H.P. power/total power

R3 - Ratio output power/maximum output power

S - Gross shaft power

S1 - Net shaft power

S2 - Maximum gross shaft power

S3 - H.P. power

S4 - L.P. power

APPENDIX A - Definition of Symbols (cont'd)

T1 - Steam inlet temperature

T2 - H.P. ideal exhaust temperature

T5 - Estimated H.P. exhaust temperature

T6 - Calculated H.P. exhaust temperature

T7 - Temperature in H.P. cylinder after valve opens

T8 - Temperature in L.P. cylinder after inlet valve opens

T9 - Temperature feedwater

T0 - Condensing temperature

U1 - Internal energy in H.P. cylinder before inlet valve opens

U2 - Internal energy in H.P. cylinder after inlet valve opens

U3 - Internal energy in L.P. cylinder before inlet valve opens

U4 - Internal energy in L.P. cylinder after inlet valve opens

V3 - H.P. cylinder displacement

V4 - L.P. cylinder displacement

V7 - H.P. cylinder clearance and cut-off volume

V8 - L.P. cylinder clearance and cut-off volume

X1 - H.P. cylinder expansion ratio

X2 - L.P. cylinder expansion ratio

APPENDIX B

Mathematical Model Step-by-Step Procedure

1. Electric power output (G) = G1 x E4 x R3; kWe
   where E4 = power conversion efficiency, initially estimated to be .25

   $$R3 = \frac{G - \text{Part Power } (J \neq 1)}{G - \text{Full Power } (J = 1)}$$ initially estimated to be $R\,(J)^{1.2}$

2. Net shaft power (S1) = G/E1; kW
   where E1 = generator efficiency from Figure 4-3

3. Gross shaft power (S) = S1 + P3 + P4; kW
   where P3 = accessory power $\cong$ 1/8 x G x R3/15; kW
   P4 = water pump power, initially estimated to be 3/8 kW

4. Inlet steam pressure (PO) = S/S2 x (P1 - 1.03 x 10^6) + 1.03 x 10^6; N/m^2
   where S2 = S - Full Power (J = 1)

5. From tables - high pressure cylinder inlet enthalpy (H1), Exhaust enthalpy (H6), exhaust temperature (T5) and Exhaust pressure (P2). H6 and T5 are initially estimated to be equal to ideal exhaust enthalpy (H2) and temperature (T2)

6. Similarly - low pressure cylinder inlet enthalpy (H3), and exhaust enthalpy (H7). H7 is initially estimated to be equal to ideal exhaust enthalpy (H4).

7. R2 = (H1 - H2) x E5/ [(H1 - H2) x E5 + (H3 - H4) x E6]
   where high pressure cylinder efficiency (E5) and low pressure cylinder efficiency (E6) are initially estimated to be .9 and .8 respectively.

8. High pressure cylinder power (S3) = S x R2
   Low pressure cylinder power (S4) = S - S3

9. Low pressure cylinder volume (V4) = S4 x X2 x 11.93 x 10^6/(P2-2.76x10^5);cm^3

10. High pressure cylinder volume (V3) = .45 V4/X2; cm^3

11. High pressure cylinder clearance & cutoff volume (V7) = V3 x .9/X1 cm^3

12. From tables - specific volume of steam before valve opens (C6) m^3/kg

13. Steam mass in clearnace volume before value opens (M3) = V7/ C6 x 10^{-6} kg

14. Steam internal energy before valve opens (U1) = H6 - P2 x C6 J/kg

15. $B1 = M3 \times (H1 - U1)$

16. Temperature in clearance volume after valve opens (T7) initially assumed to be T1 + 50

17. From tables - enthalpy (H9) and specific volume of steam (C7) after valve opens

18. Steam mass in clearance volume after valve opens (M4) = V7/C7

19. Steam internal energy after valve opens (U2) = H9 - .872 x P0 x C7

20. $B2 = M4 \times (H1 - U2)$

21. Adjust T7 until B1 = B2

22. Mass flow rate thru high pressure cylinder (M5) = (M4 - M3) x 21600; kg/h

23. Adjust V3 until M5 = system mass flow rate (M6)
    M6 is initially estimated as M6 = S x 3600 x 1.15/(H1 - H2 + H3 - H4)

24. Similarly determine mass flow rate of low pressure cylinder (M9)

25. Adjust V4 until M9 = M6

26. Calculate diameter of high and low pressure cylinders (D3 & D4) depending on number of cylinders and stroke/diameter relationship.

27. If $J \neq 1$, Skip 23, 25 & 26 and adjust P2 until M5 = M9

28. Determine engine losses

    High pressure cylinder friction loss (L2)
    High pressure cylinder thermal loss (L3)
    Thermal short circuit between cylinders (L4)
    Low pressure cylinder friction loss (L5)
    Low pressure cylinder thermal loss (L6)

29. High pressure cylinder efficiency (E5) = S3/ (S3 + L2 + L3 + L4)

30. High pressure exhaust enthalpy (H6) = H1 - (H1 - H2) x S3/(S3 + L3 + L4)

31. From tables - high pressure cylinder exhaust temperature (T6)

32. If $T6 \neq T5$, update T5, recalculate L4

33. Low pressure cylinder efficiency (E6) = S4/ (S4 + L5 +L6)

34. Low pressure exhaust enthalpy (H7) = H3 - (H3 - H4) x S4/ (S4 +L6)

35. Water flow rate (Q1) = (S1 + P3 + P4 + L2 + L5) x 3600/ (H1 - H6 + H3 - H7)

36. If Q1 $\neq$ M6, update M6
If J = 1, return to equation 11
If J $\neq$ 1, adjust P0 & P2 and return to equation 5

37. Ideal cycle efficiency (E7) = (H1 - H2 + H3 - H4)/(H1 - H2 + H3 - .9 x (H4 -970))

38. Feed water enthalph (H8) = (H7 - 970) x .9

39. Water pump power (P5) = Q1 x P0 / (2.95 x 10^9); kW

40. If P4 $\neq$ P5, update P4 and return to equation 35

41. Heat engine efficiency (E3) = S1 x 3.6 x 10^6/((H1 - H8 - H6 + H3) x Q1)

42. Power conversion efficiency (E4) = E3 x E1

43. Relative thermal input (R1) = G/ (G1 x E4)
if R1 $\neq$ R(J), adjust R3 and return to equation 1.

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