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15 kWe (NOMINAL) SOLAR THERMAL ELECTRIC POWER CONVERSION CONCEPT DEFINITION STUDY — STEAM RANKINE REHEAT RECIPROCATOR SYSTEM

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Foster-Miller Associates, Inc.

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SECTION 1

SUMMARY

An evaluation was made of the potential of a small steam Rankine reheat reciprocating engine to operate at a high level of efficiency, substantially higher than that usually associated with small power units. The specific application for the engine was as a power conversion subsystem for a point-focusing distributed receiver solar thermal electric power system. The scope of the project encompassed the analysis and design of the primary heat engine, and its auxiliaries, and the selection of the alternator; the regenerator, condenser, and water storage tanks were included. Not included were the steam generator/receiver, the collector/mirror, or the structure.

A parametric analysis was conducted resulting in the selection of the design point for the conceptual design: maximum steam temperature and pressure of 973K (1292F) and 12.1 MPa (1750 psia), and a condensing temperature of 373K (212F). For the conceptual design, a two-cylinder/opposed engine layout was selected with an operating speed of 60π rad/s (1800 rpm), directly coupled to a four-pole alternator. The design included an expander-mounted feedwater pump, a novel water/hydraulic valve actuation system, and simple, low cost regenerator operating at modest temperature levels. An important design choice was the elimination of oil from the steam circuit; this was accomplished through the use of carbon/graphite piston rings and brings out the significant advantage of eliminating any oil-related temperature limitation.

The design estimated heat engine efficiency (heat in to steam/shaft power out) is 36 percent and the overall efficiency to electric output is 33 percent. The design layout with the condenser and water tank ground mounted indicates a net weight of 230 kg (500 lb) up at the dish focal point. An evaluation of manufacturability provided a unit manufacturing cost estimate of $1952 in high volume production.

An implementation assessment was made to evaluate the technology status of the engine and its components. The basic engine is very similar to existing internal combustion units, and can be easily made on the same equipment and, in fact, use many of the same components. Two areas which would need modest development are the carbon piston rings and the valve actuation system. The former have precedence which indicated very favorable results - little or no wear after 29 Ms (8000 hr) of operation. The latter will closely follow the technology of diesel engine fuel injector systems and appears to offer no fundamental manufacturing problems.

SECTION 2

INTRODUCTION

The background and major technology basis of this work is the experience of the authors and other Foster-Miller Associates, Inc. (FMA) staff members with the automotive Rankine engine program sponsored by EPA/ERDA in the late 1960's and early 1970's. That program produced a successful engine configuration that was built and tested with 811K (1000F) steam. That engine was used in a car and was designed to be compatible with high volume manufacturing equipment.

The current study for solar applications has the objective of defining the performance potential of a piston-type steam Rankine engine system at the power level and configuration consistent with the heat available from a single parabolic dish mirror concentrator. Accordingly, the study is organized with an initial parametric analysis covering a range of steam conditions and to a lesser extent power level variations. A conceptual design is then completed at a selected steam condition and thermal input size to obtain a definition of the optimum engine configuration and a more precise evaluation of performance. Finally, a review of the conceptual design was made to obtain an evaluation of system interface requirements, estimates of system manufacturing cost and operation/maintenance costs.

This study is one of several parallel efforts on power conversion concepts for parabolic dish mirror solar applications. Ref. 4,5,6,7, and 8 are the final reports of the related studies. In each case the study scope involves the engine-generator only; to the exclusion of the dish mirror or the solar heat receiver heat exchanger.

SECTION 3
PARAMETRIC ANALYSIS

3.1 Background

Since the project scope was for a reheat reciprocator piston steam engine, part of the study examined the older references. As far as reciprocating steam engines are concerned, their peak development occurred during the 1920's and 1930's, and since then very little work, either research or production, has been done on them. Even as late as the early 1940's, reciprocating triple-expansion engines were built in large numbers for installation in Liberty ships, because they offered reasonable efficiency *and* could be built *cheaply* by manufacturing concerns of limited engineering ability. Many are still in service today, indicating another attribute of these engines - longevity (even under probable poor maintenance).

Most of the research on steam engines began to taper off in the 1930's. During the parametric study a review was made of the technical literature going back as far as the 1880's. A noticeable gap occurred from the 1940's until 1970 when the interest in low pollution engines rekindled activity in small reciprocating power systems. In any case, it was observed that the older designs generally used limited temperatures and relatively low pressures; among a few notable exceptions, as early as 1905, were the steam automobiles - even these, however, rarely exceeded 650K (700F) and 5.5 MPa (800 lb/in.^{2}).

Although modern power stations equipped with large steam turbines do employ higher pressures, they generally have steam temperature limits of about 870K (1100F) based on the need for long life boilers using (large quantities of) inexpensive alloys. In the current project, more costly alloys can be considered because of the efficiency gains which can ultimately reduce the total cost of the installation. Turbines have often been considered for small steam systems, say under 100 kW, and while they have the advantages of oil-free operation, simplicity, and small size, they have the disadvantages of poor efficiency, mostly due to leakage, and the requirement of a gearbox when low output speeds are needed.

For the parametric analysis, the ranges of interest of the following parameters were as follows:

1. Net electric power output - 5 to 100 kWe
2. Inlet steam temperatures - 723 to 973K (842 to 1292F)

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3. Inlet steam pressure - 6.89 to 17.2 MPa (1000 to 2500 psia)

4. Condensing temperature - 311 to 422K (100 to 300F).

Initial computer runs studying a range of intermediate (interstage) pressures of 0.62 MPa (90 psia) to 2.6 MPa (370 psia) showed very little difference in efficiency over that range. Therefore, for the parametric analysis, a single pressure of 1.4 MPa (200 psia) was selected for the base study.

3.2 Performance Modeling

When considering the application of a steam engine as the prime mover where a tracking solar collector is used as the heat source, the incentive is great to optimize thermal efficiency. Thermodynamic analysis shows that thermal efficiency is increased as the boiler steam temperature and pressure are raised, the former being more important than the latter. Further analysis shows that multiple staging with reheat significantly increases efficiency. This is consistent with the study scope of a two-stage engine with reheat.

The initial purpose of performance modeling is:

1. Select the piston expander type: uniflow or counterflow (optimize efficiency)

2. Select the design cut-off: inlet valve closing time (optimize efficiency).

For this initial configuration work the first stage inlet temperature/pressure are 973K (1292F), 12.1 MPa (1750 psia); second stage are 973K (1292F), 1.4 MPa (200 psia).

Figure 1 shows idealized PV diagrams for the uniflow and counterflow expander cycles superimposed on one another. The inlet pressure and temperature are the same as are the exhaust pressure and the work. Stroke and clearance volume are also the same. Note that for the uniflow case recompression on the return stroke is complete, that is, when the inlet valve opens the pressure in the cylinder is equal to the inlet pressure. The salient feature of these PV diagrams is that the cut-off is shorter for the counterflow than for the uniflow. This must be true since otherwise the counterflow work (shaded area) would be greater than that of the uniflow.

The shorter cut-off in the counterflow cycle means that less steam has been admitted to the cylinder during the cut-off period in proportion to the respective CO's (see Ref. 1) and to

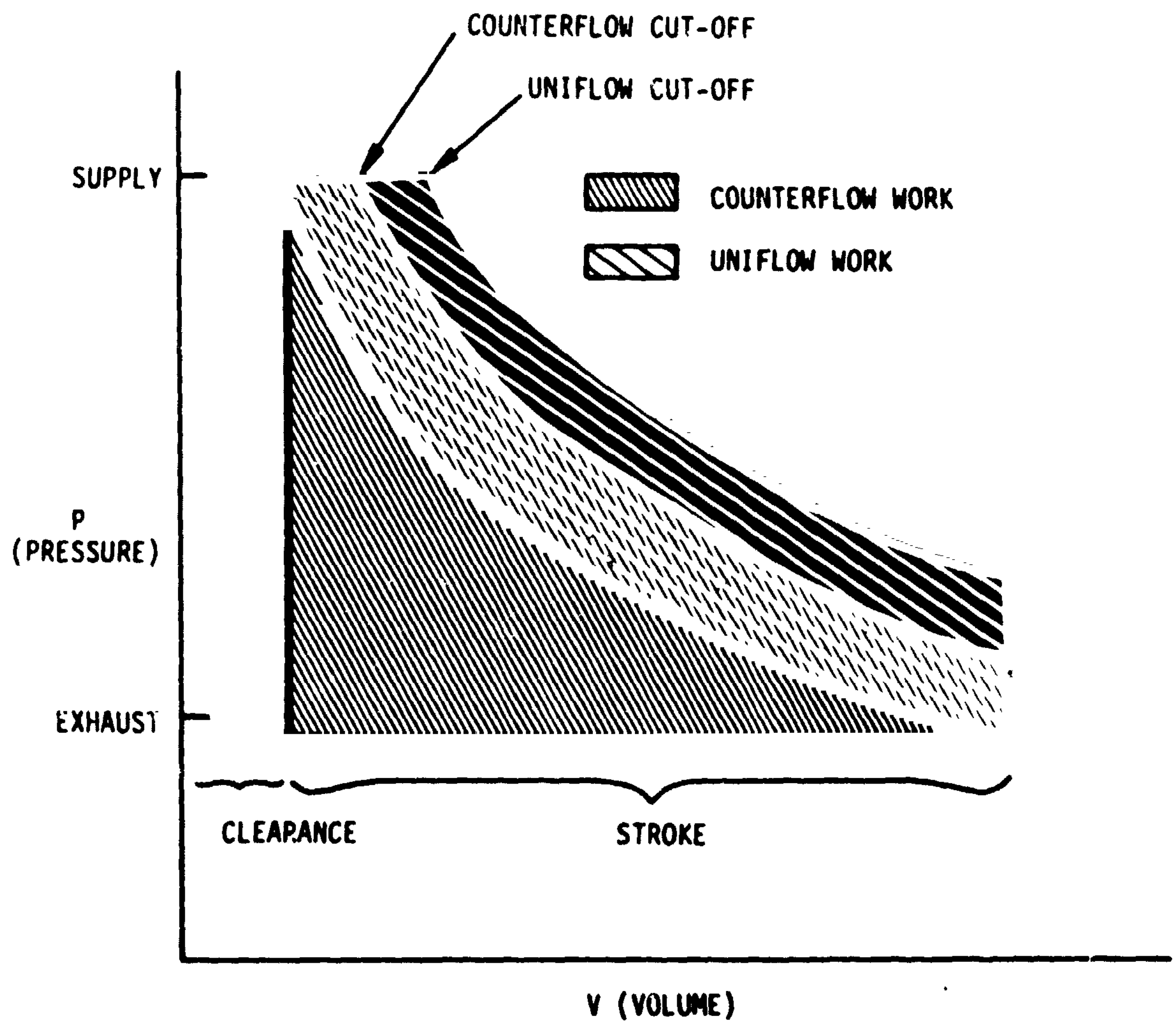


Figure 1. - Idealized PV diagrams, uniflow versus counterflow expanders.

this extent the counterflow is more efficient than the uniflow. Some of the gain, however, is lost because an incremental weight of boiler steam is required to compress the low pressure steam which is in the clearance volume when the inlet valve opens. The increment is proportional to the clearance volume and increases with increasing pressure ratio. This relationship is expressed mathematically by equation A-7 of Ref. 1. See the Glossary for definition of symbols.

$$V_B = \frac{CL}{k}\left(1 - \frac{P_2'}{P_2}\right)$$

For counterflow cycle $P_2' = P_E$; $P_2 = P_s$ and;

$$V_B = \frac{CL}{k}\left(1 - \frac{1}{r_p}\right)$$

So the efficiency of the counterflow cycle increases as the clearance volume decreases and as the pressure ratio decreases. On the other hand the efficiency of the uniflow cycle is essentially independent of both factors. The performance of the counterflow versus the uniflow cycle is graphically depicted in Figure 2. The clearance volume is taken as 5 percent of the displacement, this being readily achievable. Design studies have shown that 3 percent is about the practical limit. The ordinate, η_i, is defined as the indicated work per pound of steam divided by the ideal isentropic work. The abscissa, E_p, is defined as the indicated mean effective pressure divided by the supply pressure. This parameter increases approximately linearly with cut-off.

It is interesting to note that the counterflow characteristic exhibits a peak at intermediate E_p, as opposed to the uniflow. To the right of the peak, η_i drops off due to the fact that the steam pressure at the end of the expansion stroke is increasingly higher than the exhaust pressure, giving rise to more blowdown loss. To the left of the peak the increment of the steam flow required to pressurize the clearance volume is an ever greater percentage of the total steam flow. In addition, the pressure at the end of the expansion stroke drops below the exhaust pressure forming a negative work loop.

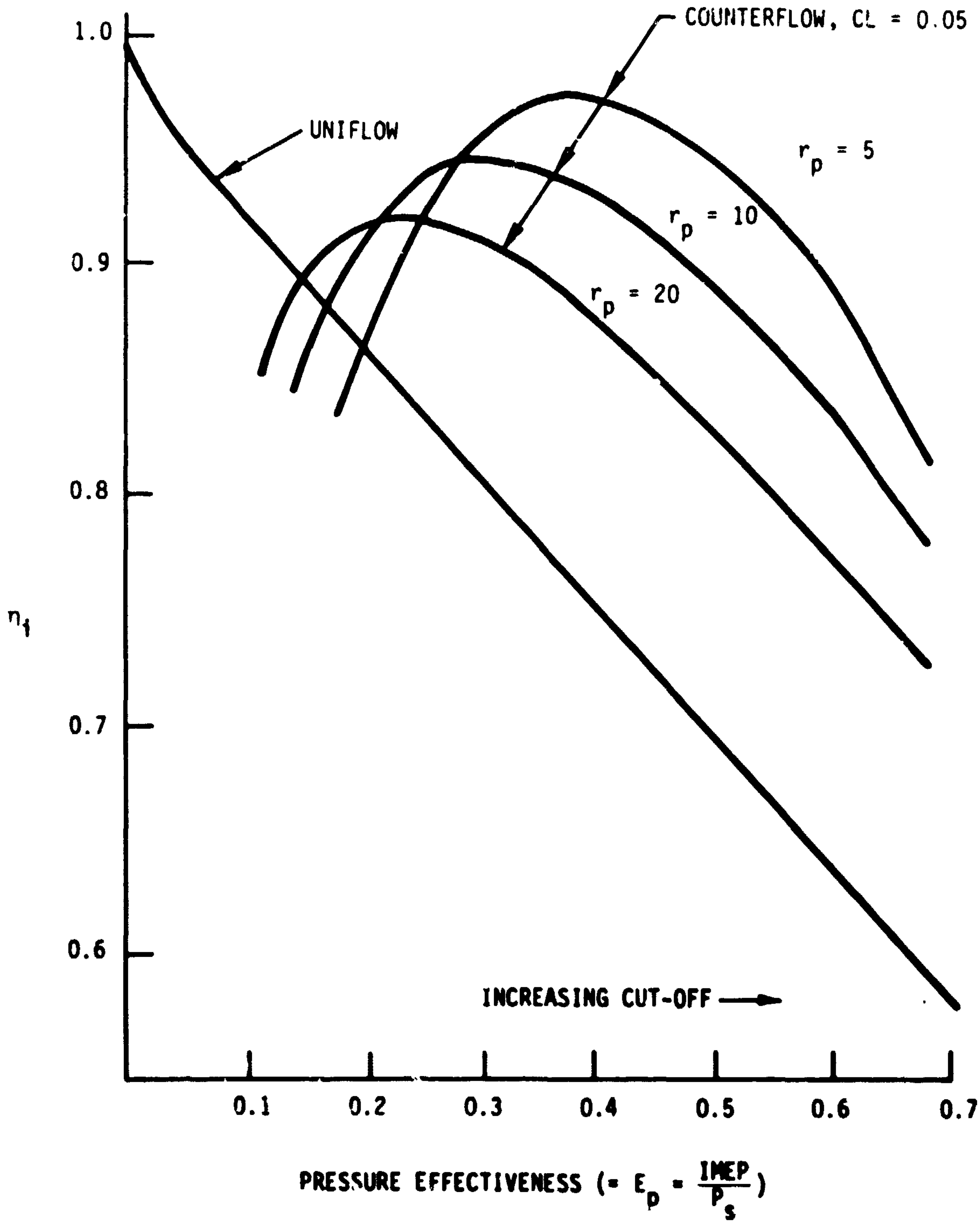


Figure 2 - Expander ef,''ciency characteristic, counterflow versus uniflow.

Mechanical efficiency in Ref. 1 is defined as:

$$\eta_{mech} = 1 - \frac{f_{mep}}{i_{mep}}$$

After considerable study of available experimental and theoretical friction data, the following formulae resulted (see Ref. 2):

$$\eta_{mech} = 0.984 - \frac{4 + 0.013\ P_s}{P_s\ E_p} \quad \text{uniflow}$$

$$\eta_{mech} = 0.984 - \frac{4 + 0.005\ P_s}{P_s\ E_p} \quad \text{counterflow}$$

These expressions are very similar, the only difference being the coefficient of the P_s term in the numerator of the fraction. This reflects the fact that in the uniflow the mechanical parts (piston rings, crankshaft bearing) are loaded during both strokes whereas in the counterflow they are loaded only during the expansion stroke. Another important observation is that the friction model predicts low mechanical efficiency at low IMEP (= P_s E_p). Consequently, the second stage has a lower mechanical efficiency than the first stage. It is also clear that at a given supply pressure mechanical efficiency increases as E_p (or cut-off) increases. This effect together with the fact that η_i decreases with increasing E_p means that the product of η_i and η_{mech} exhibits a maximum.

Figure 3 shows a plot of $\eta_i \times \eta_{mech}$ for the uniflow and counterflow cycles, which is a composite of the two stages. It can be seen that the counterflow cycle has a higher efficiency at a higher E_p than the uniflow. Further, the counterflow curve is relatively flatter yielding better off-design performance.

Overall expander efficiency may be expressed as

$$\eta_{EX} = \eta_{mech} \times \eta_i \times \eta_{HL} \times \eta_{BR}$$

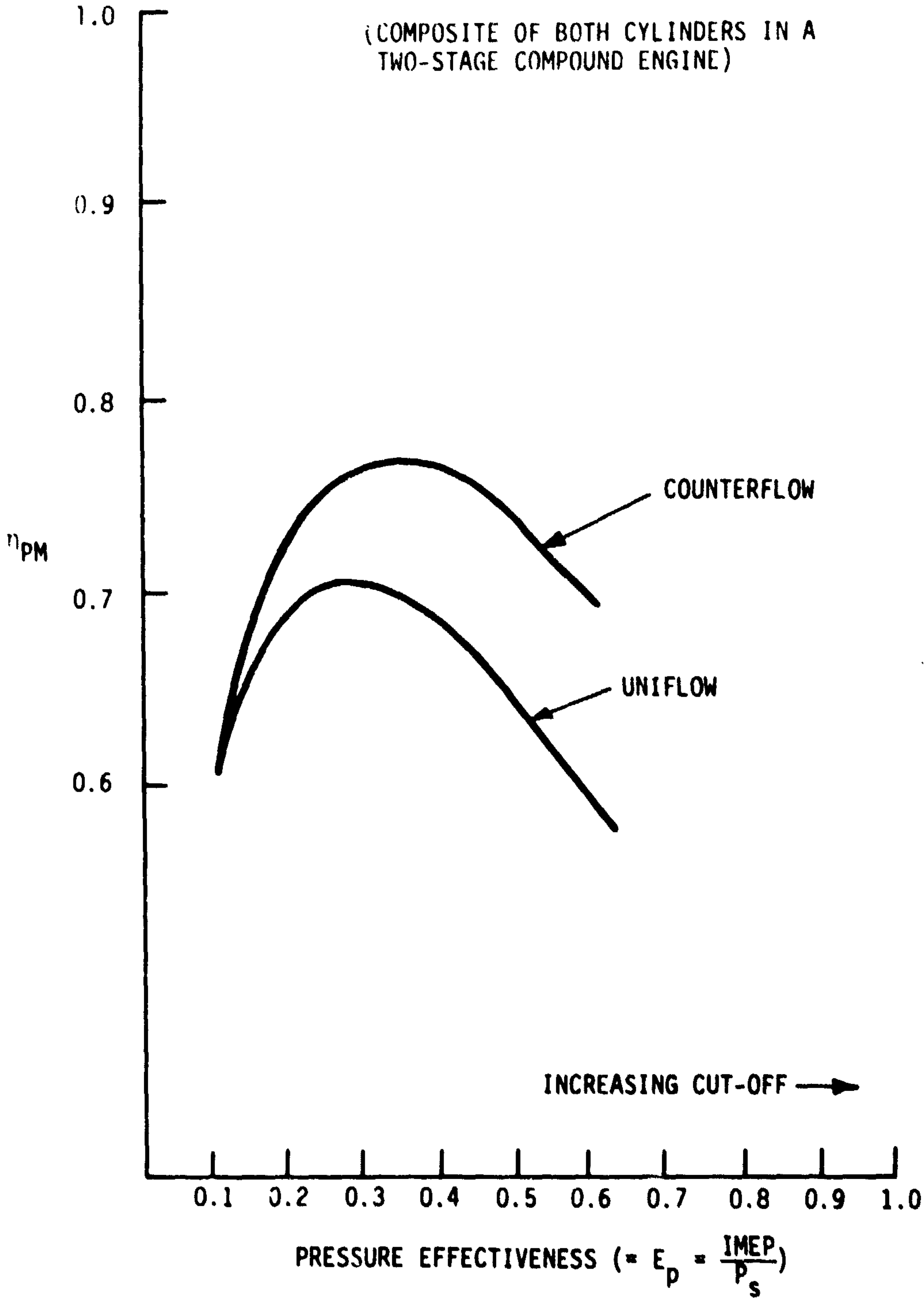


Figure 3. - Expander (prime mover) net efficiency tradeoff.

where η_{HL} is the heat loss efficiency and is a measure of both the cyclical and steady-state heat losses; and η_{BR} is the breathing efficiency measuring the adequacy of the valves. In neither (η_{BR} or η_{HL}) case is there a significant difference between uniflow and counterflow nor is there any substantial dependence upon E_p. Therefore, Figure 3 dictates the counterflow cycle (at least for highly superheated low cycle heat loss stages). It also shows the best E_p from which the displacement and the design cut-off may be chosen.

In view of the fact that the uniflow cycle was one of the latest developments of the steam engine (circa 1920) representing an advancement over the long established counterflow, the question naturally arises: What has changed? To begin with we note that the nature of the uniflow improvement was to reduce the steam flow by minimizing so called "initial condensation." This phenomenon was characterized by very substantial condensation of inlet steam during the cut-off period increasing the mass of boiler steam otherwise required. After cut-off, condensation continued but as the cylinder pressure fell a point was reached where the condensate just formed revaporized.

Thus a heat cycle was completed in which the heat of vaporization was nearly balanced by the heat of condensation and the heat transfer rates were very high, boiling on the one hand and condensing on the other. The relatively low speed of the old engines in the range of 7π rad/s (200 rpm) abetted the process by allowing time for the cylinder wall and head temperatures to vary considerably during the piston cycle.

The uniflow cycle reduced the condensation because the cylinder surface temperature was brought up to boiler temperature by the isentropic compression of the steam left in the cylinder after blowdown. Recompression work thus paid for itself in reduced steam consumption.

It must be noted at this point that inlet steam conditions were only slightly superheated. In fact, saturated steam was a very likely inlet condition. Use of low inlet steam temperature was probably due to the existing state-of-the-art of boiler materials.

Superheat of inlet steam serves to limit initial condensation in two ways. First, if the superheat is high enough and the stage pressure ratio modest, the exhaust temperature is above the saturation temperature of the incoming steam. It is therefore impossible to cool the cylinder surfaces enough to cause initial condensation. Even if the exhaust temperature does drop below the inlet saturation temperature, the amount of initial condensation will necessarily be limited. A T-s diagram showing the two expansions and the saturation temperatures of the inlet steam is shown in Figure 4.

In the second place, the amount of heat transferred to the cylinder surfaces during the condensation process exceeds that transferred away from the cylinder surfaces during vaporization, the excess being on the order of the amount of superheat. Cyclical heat flow is therefore not in balance and the extra heat would have to be rejected from the cylinder to sustain the process. The latter can be easily minimized by careful design.

It is concluded that the counterflow cycle is more efficient than the uniflow in the context of modern boiler materials.

Despite the fact that very large cyclical heat losses have been eliminated, some remains. Utilizing the internal combustion engine cooling data in Ref. 2 as well as actual test results on the automotive steam engine (Ref. 1) gives the following equation for inlet charge cooling.

$$\frac{Q_L}{WC_p} = \Delta T_c = 0.065\ (T_s - T_{ex})\left(\frac{b}{W}\right)^{1/4} + 0.35(T_{sat} - T_{ex})\ \frac{b^2}{W}$$

where

Q_L = inlet charge cooling heat rate

C_p = specific heat of steam

W = steam flow - kg/hr

ΔT_c = inlet charge cooling - K

T_s = supply temperature - K

T_{ex} = exhaust temperature - K

T_{sat} = saturation temperature of supply steam - K_C

b = bore - cm.

For inlet steam conditions specified above, the combined two-stage heat loss efficiency is 96 percent (a 4 percent cooling loss).

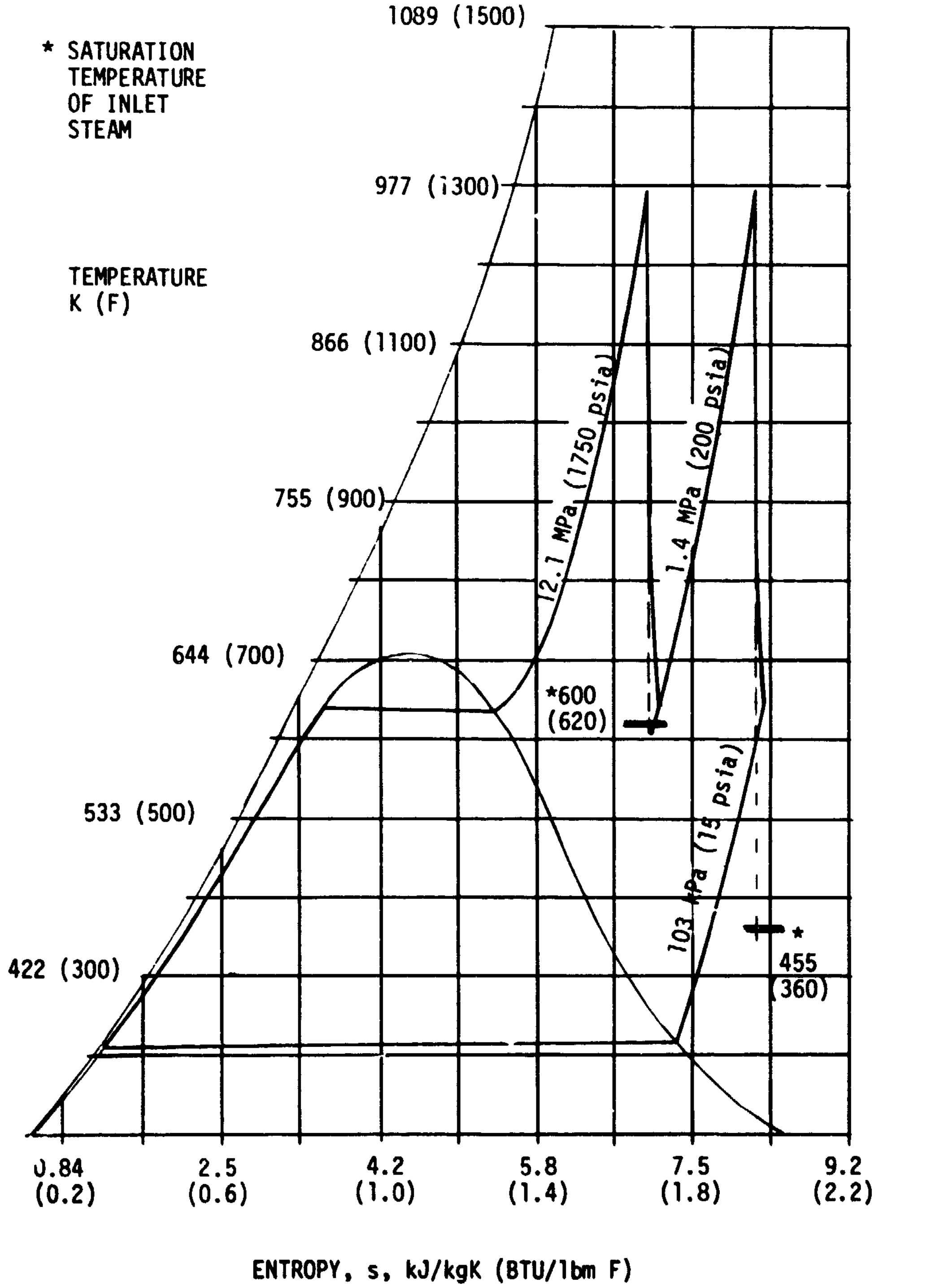


Figure 4. - Temperature-entropy diagram for design point cycle. Exhaust temperature is above inlet steam saturation temperature in both expansion stages to eliminate condensation loss.

A pressure drop of 5 percent was allowed across the inlet and exhaust valves giving a breathing efficiency of 97 percent. The combined expander efficiency considering all losses was estimated at 78 percent.

3.3 Parametric Results

The tabulation of the results of the parametric analysis are shown in Table 1 and are specifically for the 15 kWe electric output level, using the same alternator efficiency of 85 percent in all cases. The speed is 60π rad/s (1800 rpm) and was arrived at after considering valve train requirements. Extensive previous work had shown that the short valve opening times required on a steam engine tended to limit rotative speeds and had a strong dependence on physical size. For the first round of the design, the more conservative approach was to select a slower speed (but one that was compatible with a directly coupled alternator). It is possible to consider a speed of 120π rad/s (3600 rpm) if the cylinder and valve sizes can be such to keep the required acceleration forces within normal limits.

Figures 5 through 10 show the variation of efficiency with condensing temperatures, friction and heat loss characteristics, clearance volume, pressure effectiveness, and percent of load.

Since study of the expected solar input levels, with 15 kWe produced at a relative level of 1.00, showed that the subsystem would spend most of its time at an average level of about 0.80 to 0.85, the design point for best efficiency was selected at 12 kWe. This put full input at 125 percent of the engine design point and half input at 62-1/2 percent of design. As shown in Figure 10, the steam engine exhibits the desirable characteristic of a rather flat curve of efficiency versus percent of load; for the above range from 7.5 to 15 kWe, the change from maximum efficiency is about 3 percent.

The heat engine efficiencies shown in Figure 5 represent the family of optimum engine designs for each cycle. For condenser temperatures above approximately 350K (170F) the engine efficiency falls with increasing condenser temperature as would be expected from the loss in ideal cycle efficiency with decreasing cycle temperature ratio.

The engine efficiency reaches a maximum at about 350K condenser temperature for all other cycle variables while ideal cycle analysis would predict a continuously increasing efficiency with decreasing condenser temperature. The engine concept was restricted to two piston stages with reheat. At the lower

TABLE 1. - PARAMETRIC ANALYSIS TABULATION

Inlet Steam			Mass Flow Rate	Displacement cm^3 (in.^{3})		Condenser Temperature	Feedwater Temperature	Auxiliary Power Required	Efficiencies (%)			Ratio to Ideal Cycle
Temperature K(F)	Pressure MPa (psia) 1st Stage	2nd Stage	g/s (lb/hr)	1st Stage	2nd Stage	K (F)	K (F)	(kW)	Prime Mover	Heat Engine	Overall Power Conversion	
723 (842)	6.9 (1000)	1.4 (200)	21.4 (170)	81.5 (4.97)	1336 (81.5)	311 (101)	367 (201)	0.28	59.1	24.7	21.0	0.614
			20.3 (161)	77.2 (4.71)	1240 (75.7)	322 (120)	362 (193)	0.27	65.8	25.9	22.1	0.671
			19.8 (157)	75.4 (4.60)	1121 (68.4)	339 (150)	368 (203)	0.27	73.5	26.8	22.7	0.775
			20.9 (165)	79.4 (4.85)	719 (43.9)	372 (210)	403 (265)	0.30	84.7	26.7	22.7	0.850
			29.3 (233)	112.2 (6.85)	472 (28.8)	422 (300)	494 (430)	0.40	92.7	21.6	18.3	0.857
	12.1 (1750)		19.7 (156)	52.8 (3.22)	1228 (75.0)	311 (101)	368 (203)	0.46	60.2	26.2	22.3	0.631
			18.7 (148)	50.2 (3.06)	1145 (69.9)	322 (120)	364 (195)	0.44	66.6	27.4	23.3	0.687
			18.3 (145)	49.1 (3.00)	1036 (63.2)	339 (150)	170 (206)	0.43	73.8	28.2	24.0	0.775
			19.3 (152)	51.6 (3.15)	664 (40.5)	372 (210)	405 (270)	0.46	83.7	28.1	23.9	0.846
			25.7 (203)	68.8 (4.20)	413 (25.2)	422 (300)	494 (430)	0.63	91.1	23.8	20.3	0.875
	17.2 (2500)		19.9 (157)	40.3 (2.46)	1239 (75.6)	311 (101)	370 (206)	0.67	59.9	26.0	22.1	0.625
			18.9 (149)	38.3 (2.34)	1154 (70.4)	322 (120)	365 (198)	0.63	66.2	27.2	23.2	0.673
			18.4 (146)	37.4 (2.28)	1042 (63.2)	339 (150)	371 (209)	0.62	73.3	28.1	23.9	0.770
			19.4 (154)	39.3 (2.40)	668 (40.8)	372 (210)	407 (272)	0.67	83.0	27.9	23.7	0.835
			26.0 (206)	52.8 (3.22)	419 (25.5)	422 (300)	496 (433)	0.92	89.8	23.6	20.0	0.849
848 (1067)	6.9 (1000)	1.4 (200)	17.6 (139)	81.5 (4.97)	1297 (79.1)	311 (101)	401 (263)	0.24	61.0	28.3	24.1	0.660
			16.4 (130)	76.9 (4.64)	1185 (72.3)	322 (120)	390 (243)	0.22	68.7	29.9	25.4	0.722
			16.0 (127)	74.0 (4.52)	1069 (65.2)	339 (150)	394 (250)	0.22	76.2	30.8	26.2	0.794
			16.9 (134)	78.1 (4.77)	687 (41.9)	372 (210)	433 (319)	0.24	86.8	30.6	26.0	0.860
			24.0 (190)	111.5 (6.80)	456 (27.8)	422 (300)	539 (511)	0.33	93.1	21.2	21.2	0.859
	12.1 (1750)		15.5 (123)	52.5 (3.20)	1143 (69.8)	311 (101)	391 (244)	0.36	64.8	30.8	26.2	0.691
			15.1 (120)	51.1 (3.12)	1091 (66.6)	322 (120)	390 (243)	0.35	69.7	31.6	26.9	0.731
			14.8 (117)	50.0 (3.05)	989 (60.4)	339 (150)	395 (252)	0.35	76.4	32.4	27.6	0.798
			15.6 (124)	53.5 (3.26)	635 (38.8)	372 (210)	434 (322)	0.39	85.9	32.3	27.5	0.861
			21.7 (172)	73.6 (4.49)	412 (25.1)	422 (300)	542 (516)	0.53	91.2	26.8	22.8	0.851
	17.2 (2500)		15.2 (120)	40.6 (2.48)	1117 (68.1)	311 (101)	392 (247)	0.51	64.7	31.2	26.5	0.690
			14.8 (117)	39.6 (2.42)	1067 (65.1)	322 (120)	392 (245)	0.49	69.4	32.0	27.2	0.731
			14.4 (114)	38.7 (2.36)	965 (58.9)	339 (150)	396 (253)	0.49	76.0	32.9	28.0	0.797
			15.2 (120)	40.7 (2.48)	618 (37.7)	372 (210)	436 (325)	0.52	85.2	32.8	27.8	0.850
			20.6 (163)	55.1 (3.36)	391 (23.8)	422 (300)	540 (513)	0.73	89.7	27.8	23.6	0.853
973 (1292)	6.9 (1000)	1.4 (200)	13.7 (108)	73.9 (4.51)	1161 (70.8)	311 (101)	411 (280)	0.18	67.4	33.3	28.3	0.732
			13.4 (106)	72.8 (4.44)	1117 (68.2)	322 (120)	413 (283)	0.18	72.0	34.0	28.9	0.782
			13.3 (105)	71.9 (4.39)	1020 (62.2)	339 (150)	419 (294)	0.18	78.7	34.8	29.5	0.817
			14.5 (115)	78.5 (4.79)	678 (41.4)	372 (210)	470 (387)	0.20	87.0	33.8	28.7	0.860
			20.5 (162)	111.0 (6.77)	448 (27.3)	422 (300)	587 (596)	0.28	93.1	27.8	23.6	0.858
	12.1 (1750)		12.9 (102)	51.5 (3.14)	1212 (74.0)	311 (101)	422 (299)	0.30	66.7	34.6	29.4	0.732
			12.4 (98)	49.9 (3.05)	1031 (62.9)	322 (120)	416 (288)	0.29	72.5	35.8	30.4	0.790
			12.2 (97)	49.1 (3.00)	940 (57.4)	339 (150)	422 (299)	0.29	78.8	36.5	31.1	0.818
			13.3 (105)	53.3 (3.[illegible]5)	621 (37.9)	372 (210)	473 (391)	0.32	86.3	35.7	30.3	0.858
			18.1 (143)	72.8 (4.44)	396 (24.2)	422 (300)	589 (601)	0.44	91.4	30.2	25.7	0.853
	17.2 (2500)		12.5 (99)	40.2 (2.45)	1057 (64.5)	311 (101)	423 (302)	0.42	66.8	35.3	30.0	0.732
			12.1 (95)	39.2 (2.39)	1003 (61.2)	322 (120)	418 (293)	0.40	72.4	36.4	30.9	0.790
			11.9 (94)	38.6 (2.36)	915 (55.8)	339 (150)	424 (304)	0.40	78.5	37.1	31.6	0.815
			12.9 (102)	41.7 (2.54)	602 (36.7)	372 (210)	475 (396)	0.44	85.7	36.3	30.9	0.854
			17.4 (138)	56.5 (3.45)	380 (23.2)	422 (300)	592 (605)	0.61	90.2	31.0	26.4	0.847

Note: Tabulation for 15 KWe output, 85% alternator efficiency, 60π radians (1800 rpm)

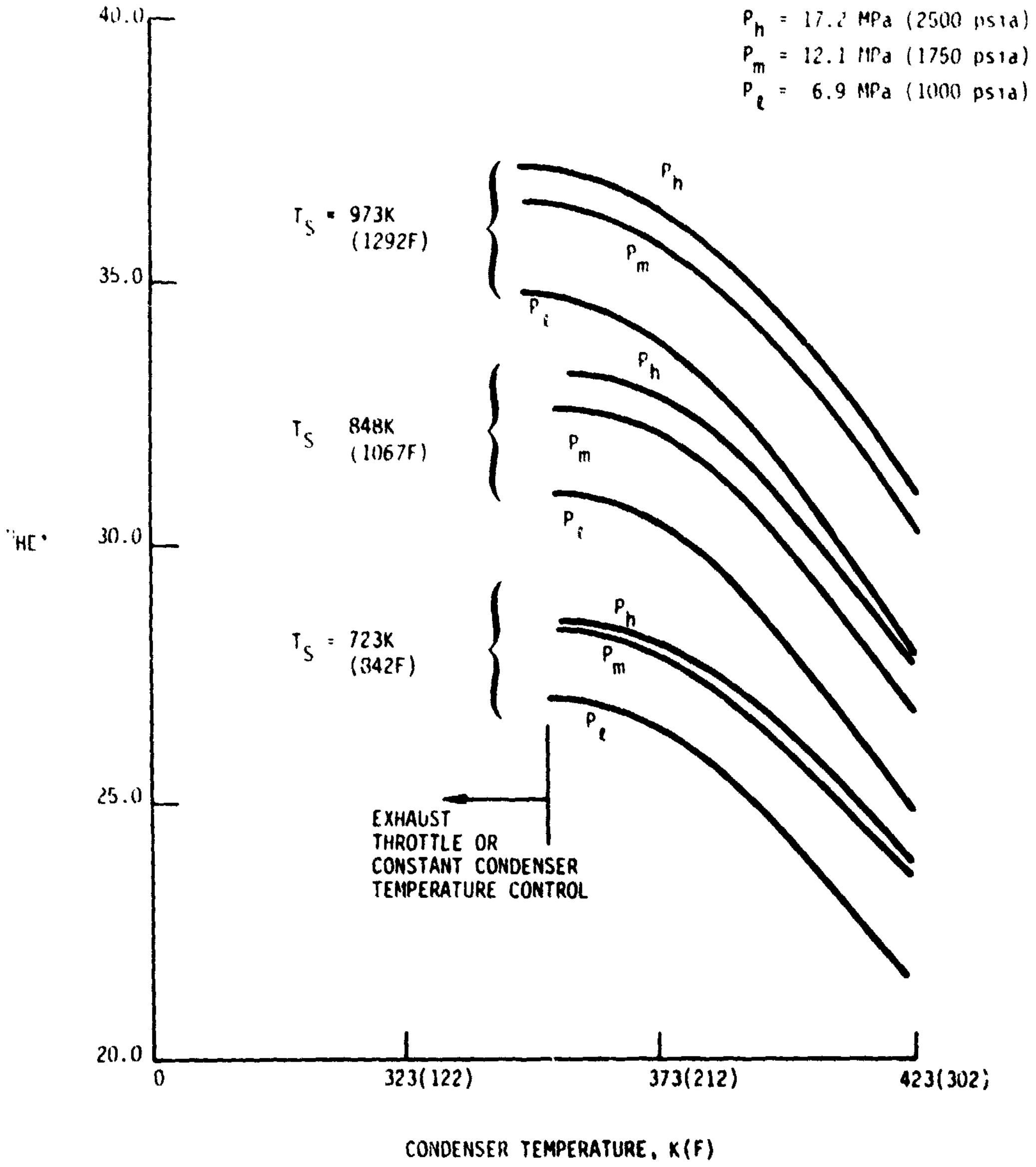


Figure 5. - Heat engine efficiency as a function of condenser temperature for a range of supply temperatures and first-stage inlet pressures. Both expander stages at the same supply temperature.

condenser temperatures, the expander efficiency falls rapidly due to inlet condensation which ultimately overpowers the potential of increasing ideal cycle efficiency.

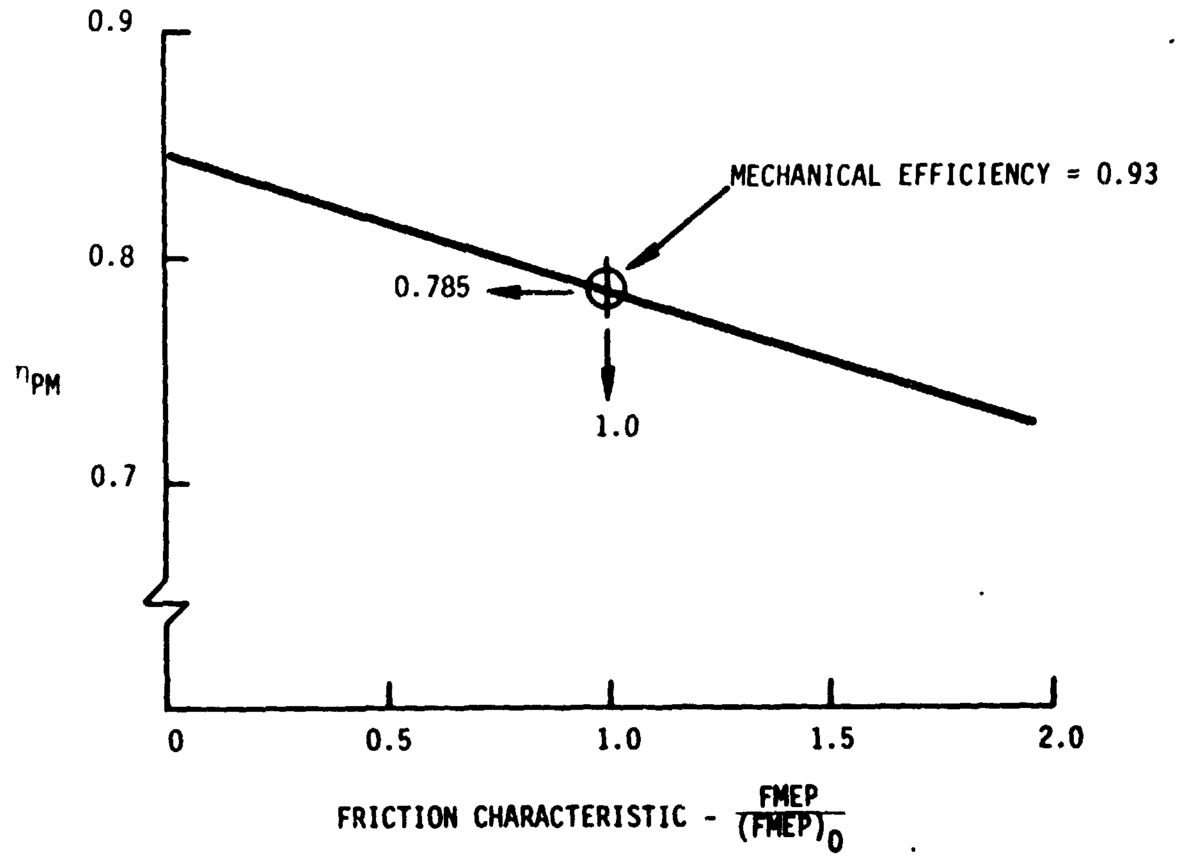
This study concludes that the condenser temperature should be held at about 350K for a two-stage reheat cycle piston expander. This conveniently allows an atmospheric pressure condenser at 373K (212F). If the ultimate in efficiency is desired, a blow-down turbine can be added to the engine and lower condenser pressure used to improve engine efficiency by perhaps two points.

Figure 6 illustrates the influence of expander friction on prime mover (expander) efficiency. The expander efficiency is relatively insensitive to mechanical efficiency as the projected friction is only one-third of the total expander inefficiency. The mechanical efficiency is somewhat more important to engine efficiency as it is a direct multiplier on the output. Diagram, breathing and pressure losses on the other hand are reflected in the expander exhaust energy that can be reintroduced to the cycle through the regenerator.

Figure 7 presents the influence of expander heat loss on the expander efficiency for the conceptual design. The design point loss of 4 percent is typical of the high superheat limited stage temperature ratio designs that produce the highest engine efficiency. The second loss term of the inlet charge cooling equation is zero for these engines as the exhaust temperature is equal to or higher than the inlet steam saturation temperature.

A practical design limitation on expander efficiency is illustrated in Figure 8. The clearance volume (dead space in the cylinder/valve pockets at top dead center) is limited by practical mechanical configurations and allowance for transient thermal expansions. Particularly in the case of the counterflow expander without recompression, this space causes a throttle loss of the initial steam flow to pressurize the cylinder. The loss can be reduced in practice by designing for early exhaust valve closure with a tradeoff in some recompression friction loss.

The tradeoffs in expander design for a range of stage pressure ratio is shown in Figure 9. This shows the ideal diagram efficiency (neglecting breathing, heat transfer and friction) for complete recompression ("uniflow") and counterflow or π (180 deg) exhaust timing engines. The pressure effectiveness term is the fraction of the stage supply pressure delivered as work.



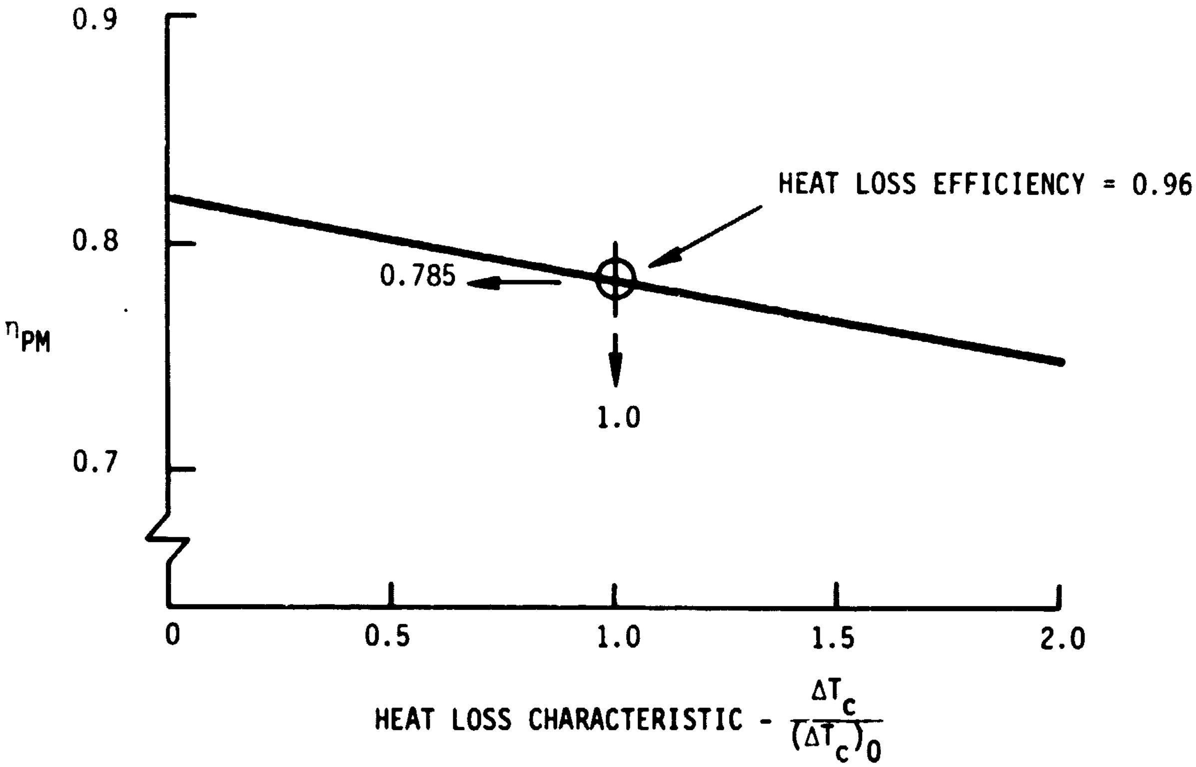
$(FMEP)_0 \quad = \quad 0.005\ Ps + (0.11 + 6.9 \times 10^{-5}\ IMEP)S$

(Derived from IC Engine Experiments)
Ps - Supply Pressure - kPa
IMEP - Indicated Mean Effective Pressure - kPa
S - Piston Speed - m/min

For Design Piston Speed = 240 m/min

$(FMEP)_0 \quad = \quad 0.005\ Ps + 27 + 0.016\ IMEP$

Figure 6. - Influence of friction characteristic on prime mover efficiency.



$$\frac{Q_L}{WC_p} = \Delta T_c = 0.065\,(T_s - T_{ex})\left(\frac{b}{W}\right)^{1/4} + 0.35\,(T_{sat} - T_{ex})\,\frac{b^2}{W}$$

ΔT_c = Inlet Charge Cooling - K

T_s = Supply Temperature - K

T_{ex} = Exhaust Temperature - K

T_{sat} = Saturation Temperature of Supply Steam - K

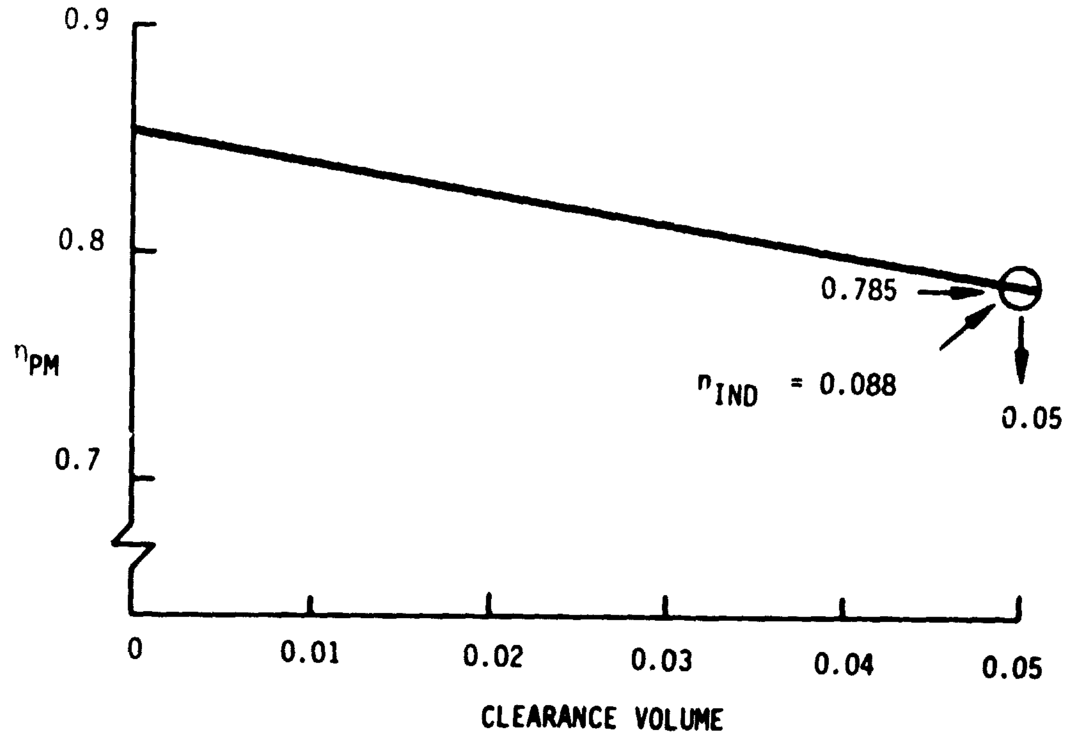
b = Bore - cm

W = Steam Flow - kg/hr

C_p = Specific Heat of Steam

Q_L = Inlet Charge Cooling Heat Rate

Figure 7. - Influence of heat loss characteristic on prime mover efficiency.



$$\eta_i = \frac{CO + \frac{CL+CO}{k-1}\left[1 - \left(\frac{CL+CO}{1+CL+CO}\right)^{k-1}\right] - \frac{1}{r_p}}{CO + \frac{CL}{k}\left(1 - \frac{1}{r_p}\right)} \quad \frac{k-1}{k}\left(1 - r_p^{\frac{1-k}{k}}\right)^{-1}$$

CO = CUT-OFF VOLUME FRACTION OF DISPLACEMENT

CL = CLEARANCE VOLUME FRACTION OF DISPLACEMENT

r_p = PRESSURE RATIO

k = RATIO OF SPECIFIC HEATS = $\frac{C_p}{C_v}$

Figure 8. - Influence of clearance volume on prime mover efficiency.

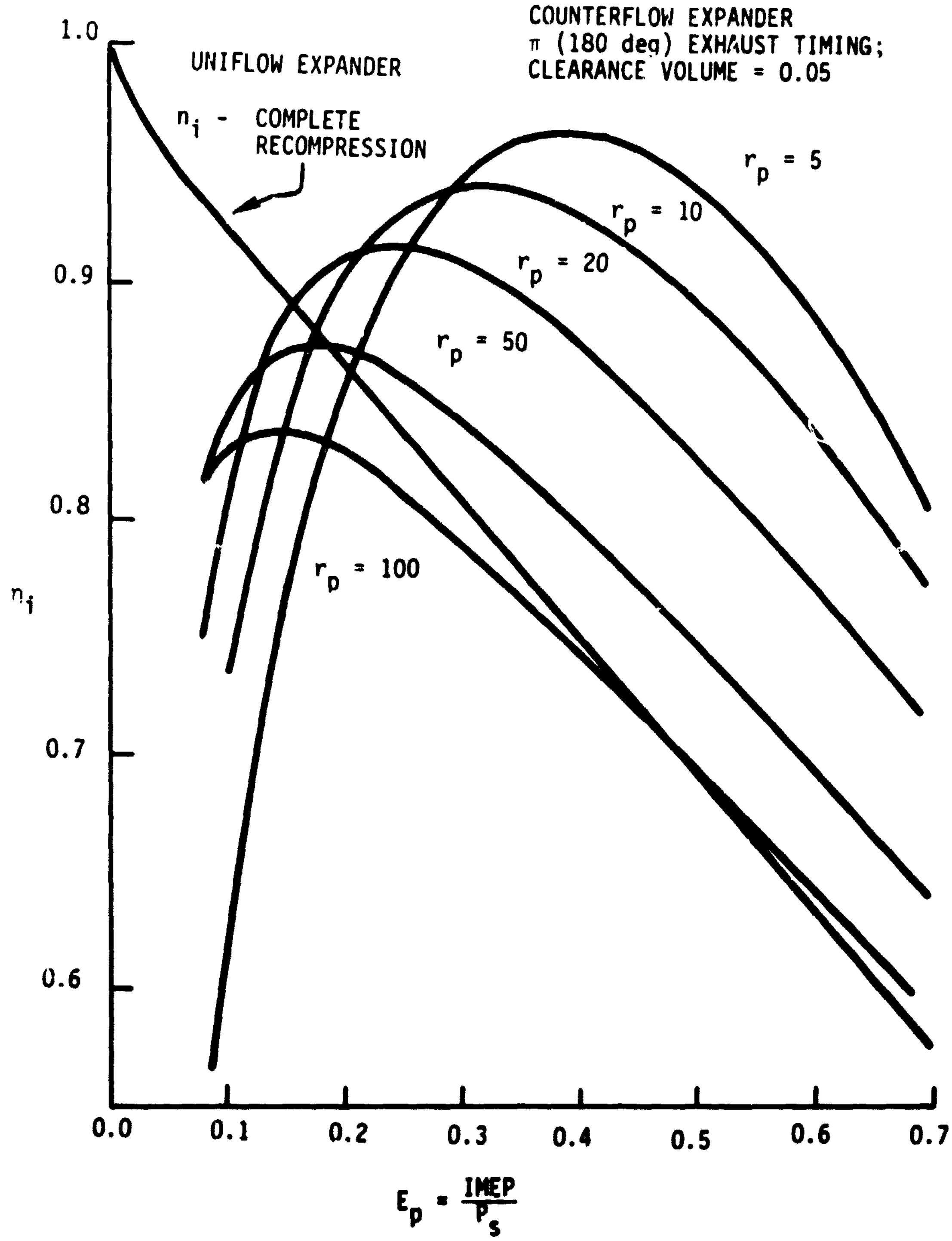


Figure 9. - Efficiency versus pressure effectiveness.

The complete recompression expander is ideally unaffected by the pressure ratio available as the recompression stroke prevents inlet blowdown. As the power is increased (by increasing cut-off), the expansion ratio departs from the ideal and efficiency decreases steadily with power.

The counterflow expander with a fixed clearance suffers an inlet blowdown loss proportional to the available pressure ratio. This blowdown loss can be reduced to a small efficiency penalty if the pressure effectiveness (E_p) is high. Increasing E_p to reduce the percentage loss due to blowdown is traded off against lower expansion ratio. Figure 9 shows that the reduced expansion ratio at higher E_p reduces the efficiency of higher pressure ratio (r_p) stages significantly. For lower r_p stages, the ideal expansion ratios can be achieved at very high E_p and with a very low blowdown penalty. The compound engines of this study resulted in individual stage pressure ratios around 10 to 1. The efficiency versus E_p is relatively flat around the maximums at low stage pressure ratio so that power modulation by changing cut-off (Ep) can yield high efficiency over a substantial power range.

The expander efficiency versus load at constant speed is shown in Figure 10. This curve illustrates the relatively constant efficiency of the expander; and hence, an engine over a 2 to 1 power range. The design load was selected to maximize annualized efficiency with a small penalty at maximum power, 15 kW in this example.

A parametric analysis of design point power level indicates relative insensitivity to speed and power level. Three approaches to engine design for a range of design point powers were investigated. For a given output speed, capacity can be changed by changing the cylinder size. For a range of output speeds, capacity can be changed by scaling an "optimum" design at the best piston speed (scale factor proportional to speed). Power at any speed or scale can also be increased by adding pairs of cylinders.

The modeling techniques for engine performance are derived from engine data ranging from approximately 3 to 30 kW per cylinder. The data base did not reveal certain loss trends expected in very small engines. At very small size, conduction, radiation, ring leakage and practical mechanizations should become more dominant and reduce performance expectations below that indicated in the model used. Consequently, the following projections should be considered to be increasingly optimistic at power levels below perhaps 10 kW.

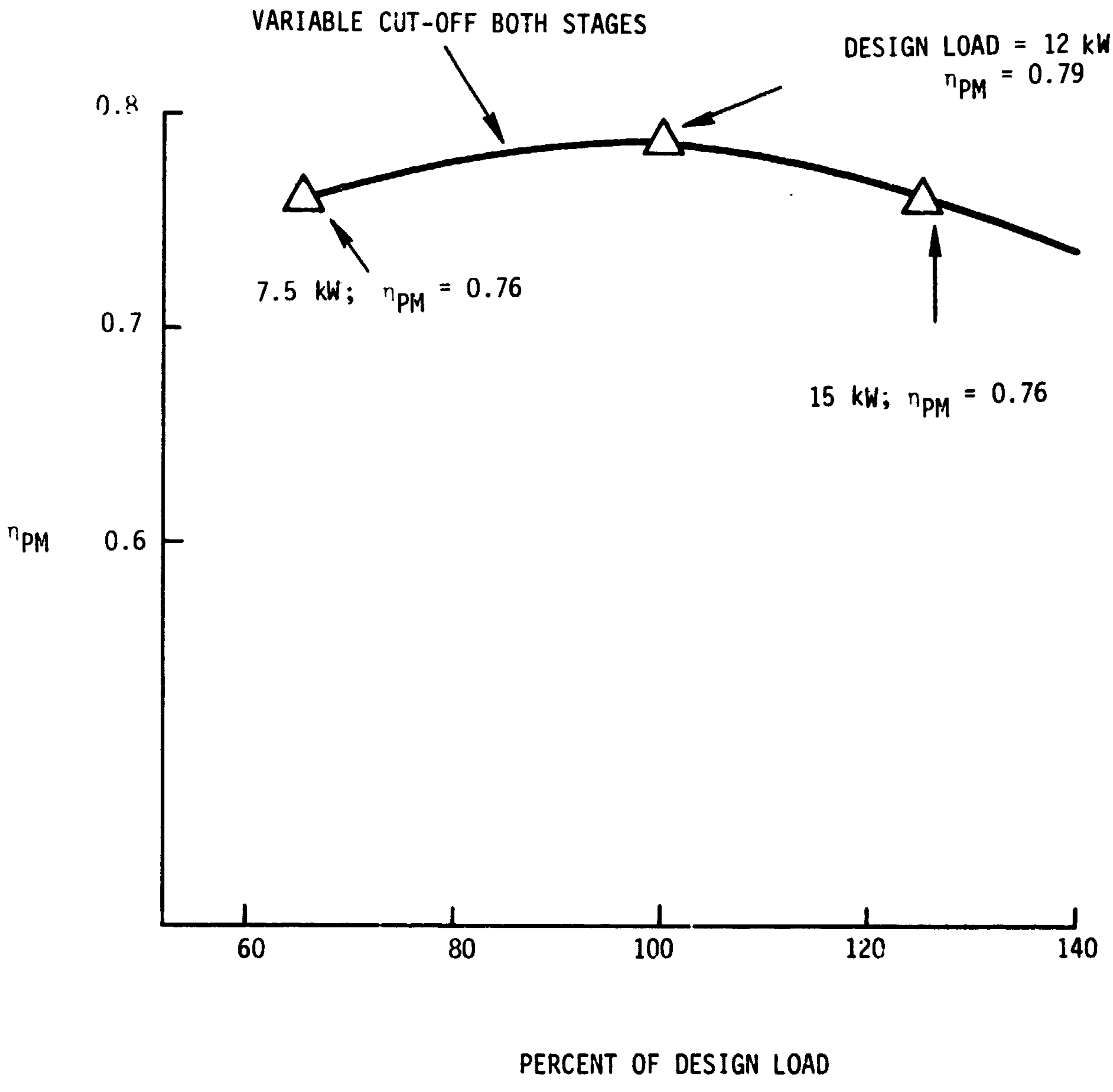


Figure 10. - Effect of load on prime mover efficiency.

Table ^ presents the power and efficiency predictions for constant speed engines 60π rad/s (1800 rpm) designed for different maximum power levels. The reference design is for the 26 kW conceptual design completed for this program. Two scaled engines were analyzed for performance. The two scaled designs are dimensionally exactly similar to the reference design, one is two-thirds the size and one is one and a half times the reference size. The resultant designs thus have power levels proportional to the cube of the scale factor and piston speeds directly proportional to the scale factor. Power per square inch of piston area is proportional to piston speed while the brake mean effective pressure (BMEP) is the same for all designs. Power and weight scaling implies that the specific weight (kg/kW) will also be constant for the three engine-generator packages.

The efficiency of the expander and engine is predicted to be slightly higher for the smallest engine and somewhat lower for the largest engine. For the highly superheated and limited pressure ratio cycle used, the heat loss efficiency is very high so that changes in specific power contribute very small changes in heat loss. Expander friction is the only loss predicted to change significantly with size at constant speed. The efficiency projections are thus related to piston speed with the largest engine at the highest piston speed being the least efficient.

TABLE 2. - ENGINE PERFORMANCE AS A FUNCTION OF POWER LEVEL AT CONSTANT OUTPUT SPEED 60π rad/s (1800 rpm) FOR TWO-CYLINDER COMPOUND EXPANDERS

Relative Performance	Engine Relative Dimensions		
Scale factor	2/3	1	1-1/2
Maximum power (kW)	7.7	26	87.7
Engine efficiency (%)	37.3	36.6	35.3
Relative engine efficiency	1.019	1	0.965
Expander mechanical efficiency (%)	97.0	96.2	95.1

Scaling the engines at constant piston speed, that is, dimensions proportional to revolutions per minute, results in equal mechanical efficiency for all sizes. Again, the heat loss efficiency is predicted to change insignificantly over the range of sizes investigated. The engine efficiency is thus constant over the range of powers shown in Table 3. Note again that other loss mechanisms should be anticipated in the smaller engines that would reduce the efficiencies with progressively smaller engines below roughly 10 kW. Tables 2 and 3 illustrate that the optimum power level for a pair of cylinders changes rapidly with changes in desired output speed. The specific weight changes gradually with speed.

Changing maximum power level by adding cylinder pairs is expected to be the most attractive approach if several engine sizes are desired. A single cylinder pair designed for the minimum desired size can be doubled or tripled, etc., by adding identical hardware to a range of crankcase designs. Constant output speed is compatible with a wide range of alternator powers at a given speed as the alternator design is a similar process wherein the number of wire turns, rather than wire size, are usually proportional to power. This approach results in the lowest weight at any given power level and also tends to reduce cost through parts commonality.

TABLE 3. - ENGINE PERFORMANCE AS A FUNCTION OF POWER LEVEL AT CONSTANT PISTON SPEED 4.1 m/s (800 ft/min) FOR TWO-CYLINDER COMPOUND EXPANDERS

Output Speed	Scale Factor	Power (kW)	Relative Specific Weight (kg/kW)/(kg/kW) 60π rad/s (1800 rpm)
120π rad/s (3600 rpm)	1/2	6.5	1/2
60π rad/s (1800 rpm)	1	26	1
40π rad/s (1200 rpm)	1-1/2	58.5	1-1/2
30π rad/s (900 rpm)	2	104	2

SECTION 4

CONCEPTUAL DESIGN

4.1 Design Point Selection

A review of the results of the parametric analysis was made to select steam conditions and condensing temperature for the conceptual design. The results indicated, in general, that the higher the initial and reheat steam temperature, the higher the engine efficiency. The limiting factor (aside from material cost) is material capability and the capability of the engine cylinder lubrication system. Previous experience at FMA has proven the capability of steam engines with oil lubricated rings for operation with 811K (1000F). It is expected, however, that higher steam temperatures can be utilized in the counterflow design if the piston rings are located well down on the cylinder or if some nonlubricated design concept such as carbon rings is adapted. Accordingly, the maximum temperature in the parametric range 973K (1292F) was selected for both the initial and reheat steam conditions.

The parametric analysis indicated that engine efficiency is relatively insensitive to the initial steam pressure. There are, however, practical design limitations involving piston ring sealing capability against the very high pressure. Accordingly, the steam pressure was chosen intermediate in the parametric range at 12.1 MPa (1750 psia). As discussed previously, the engine performance is almost totally insensitive to the intermediate or reheat pressure in the 1.4 MPa (200 psia) range.

The selection of condenser temperature (pressure) for a reciprocating piston engine is different than that for a turbine system. For the turbine, system performance typically will increase as condenser temperature is decreased. This is not true, however, for the reciprocator as illustrated in Figure 5. As condenser temperature decreases beyond a certain value, the theoretical gain is overshadowed by thermal losses in the form of initial condensation losses as inlet steam is exposed to the piston returning from the condenser end of the stroke. For the conceptual design, an atmospheric condenser was chosen with a condensing temperature of 373K (212F).

Figure 11 is a flow diagram that illustrates the design point conditions. Note that a 5 percent pressure loss is assumed at the inlet and exhaust valves of each cylinder. The intermediate pressure was changed from 1.4 MPa (200 psia) as used in the parametric analysis to 1.1 MPa (161 psia) (after the reheater) to give equal pressure ratios in both stages of 9.43 to 1. The steam flow rate is determined by the thermal input

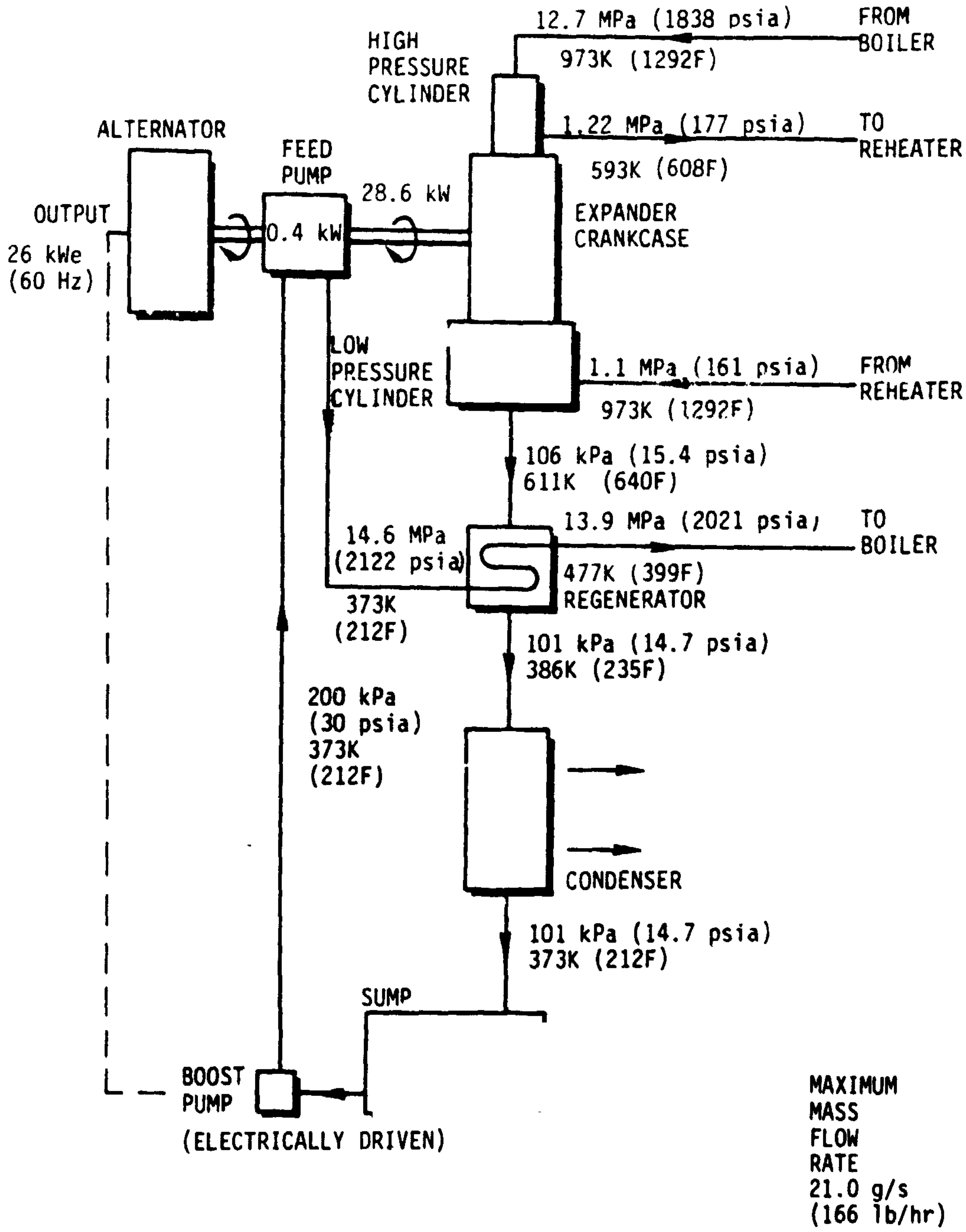


Figure 11. - Rankine cycle flow diagram
(at maximum power).

available at the solar receiver and the specific enthalpy rise (kW/g) required in the receiver to produce the 973K (1292F) outlet steam temperature desired. For the conceptual design, NASA specified the receiver maximum thermal power at 80 kW resulting in a steam flow rate of 21.0 g/s (166 lb/hr). The system's maximum electrical output then is a result of the specified thermal input operating through the engine and generator efficiencies.

4.2 Expander Design

The purpose of the expander is to drive the alternator. In order to keep the overall system as simple as possible, a direct-drive, no gearbox design was desirable. And to use relatively straightforward and easily available equipment, the rotative speed therefore had to be restricted to 120π rad/s or 60π rad/s (3600 or 1800 rpm). An alternator designed for 40π rad/s (1200 rpm) rotative speed could be used; however, the weight (and size) of the system increases with decreasing speed. The choice now remained between the first two speeds, and this depended heavily on the choice of valve gear. Excellent results have been obtained in the past with poppet valves as far as life, sealing capability, low cost, and lack of requirement for lubrication are concerned. But due to the relatively short (compared to I.C. engine practice) valve events which are usually specified for modern steam engines, the higher speed would be difficult and expensive to implement. 60π rad/s (1800 rpm) offers a good middle ground upon which to base a reciprocating steam engine design; the higher speed could be accommodated using a rotary valve system or equivalent, where high speed is not a mechanical problem. Breathing is a problem at high speeds, and rotary valves suffer from leakage (and wear and the requirement for lubrication if seals are used). In addition, in order to keep piston speeds within reasonable limits [in the range of 4 m/s (800 ft/min)] multiple cylinders for the second stage might have to be used to avoid very over-square cylinder dimensions.

Table 4 lists the design specifications of the 1800 rpm expander-generator power conversion system concept. The generator is actually in induction motor driven by the expander to operate as an induction generator. The generator (motor) is a 30 hp commercial motor available from Gould, Inc., Electric Motor Division as part of their E-plus line of high efficiency motors. The expander itself is a two-cylinder design consisting of a single high pressure (stage I) and a single low pressure (stage II) reheat cylinder. The design point is picked on the basis of the expected solar duty cycle as discussed in subsection 3.3. Figure 12 is a pictorial of the expander illustrating the compact size of the high pressure and (relatively) high speed engine.

TABLE 4. - SOLAR RANKINE STEAM ENGINE SPECIFICATIONS

Two cylinder, opposed, single throw crank
Single acting with crossheads
Compound expansion with reheat
Inlet temperatures 973K (1292F)
Atmospheric pressure condensing
Poppet valves, feedwater pressure actuated
Counterflow: 3% clearance volume
Carbon piston rings (no oil in steam)
Speed: 60π rad/s (1800 rpm) nominal - actual ≈1840 rpm
Stroke: 68 mm (2.67 in.)
Piston speed: 4.1 m/s (800 ft/min)

	Stage I	Stage II
Inlet pressure, MPa (lb/in.^{2})	12 (1750)	1.1 (153)
Bore, mm (in.)	43 (1.71)	149 (5.86)
Displacement, cm^3 (in.^{3})	100 (6.12)	1179 (72.0)
Maximum piston thrust, kN (lb)	17.0 (3816,	17.4 (3916)

Performance	Design Point	Maximum	Minimum
Electric output, kWe	21	26	13
Cut-off (%)	18.0	23.0	11.2
Flow rate, g/s (lb/hr)	16.8 (136)	21.0 (166)	11.1 (87.9)
Stage I MEP, MPa (lb/in.^{2})	4.2 (602)	5.1 (737)	2.7 (389)
Stage II MEP, kPa (lb/in.^{2})	363 (52.6)	444 (64.4)	234 (33.9)
IkW (IHP)	25.3 (33.9)	31.0 (41.5)	16.3 (21.9)
Expander efficiency (%)	87.9	87.4	82.8
Engine efficiency (%)	35.9	35.9	34.1
Alternator efficiency	92.1	91.6	90.8
Net electrical efficiency (%)	33.0	33.0	31.4

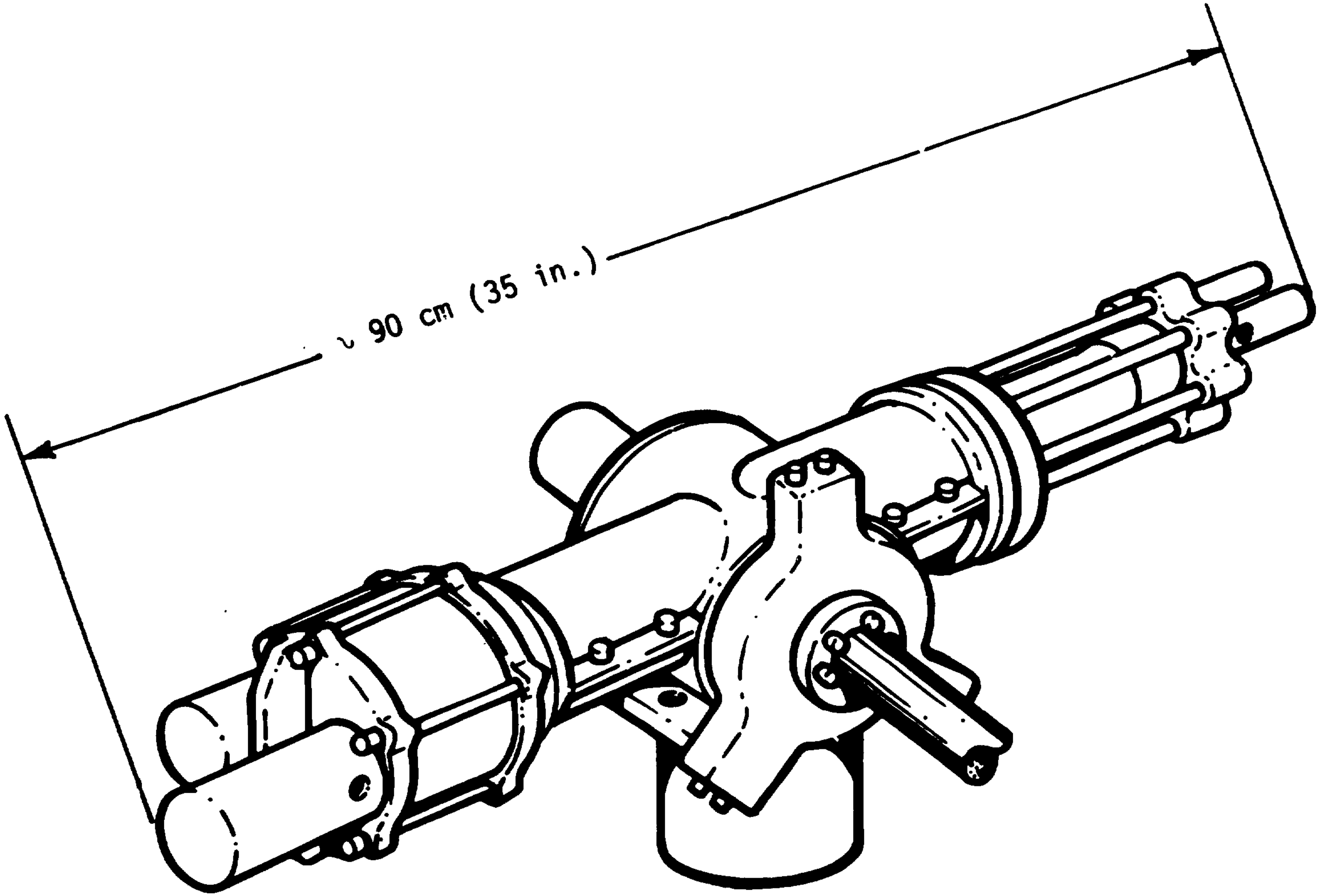


Figure 12. - Compact steam engine design exterior.

Figure 13 is a section view of the expander cylinders and crankcase. The layout and construction of the crankcase is nearly identical to internal combustion (I.C.) engine practice. Because steam pressure and piston area determine the piston and crank loads, the design stresses can be determined by the designer unlike a diesel or gasoline engine where cylinder pressures are required at certain levels to allow the cycle to function. The design stress levels in a steam engine would be similar to those of a diesel. Steam engine shock loads are lower than in I.C. engines due to the absence of diesel knock or gasoline engine detonation. This would allow low cost components (the crankshaft and connecting rods can be made of cast iron).

The currently available trimetal bearings, used in I.C. engines, are perfectly satisfactory for the steam engine. In fact, even less expensive bearings could be selected because the contaminants associated with I.C. engines would not be present. The lubrication environment can be likened to a gearbox - there are no combustion gas blow-by, no acids, no particulates. Because there are no piston rings on the crosshead, there should be a minimum accumulation of wear particles; all told, it is likely that crankcase lubricant could last for several years.

The crossheads would be aluminum and rise in a crosshead guide (sleeve) of steel or steel with a coating. Since small I.C. engines virtually never use separate crossheads, this might be considered an additional component and expense; however, crossheads are used in air compressor designs and are quite well worth the extra effort when long life and performance are concerned. The essence of a crosshead is to allow side thrusts associated with crank and con-rod layouts to be taken care of in the oil-bath region of the crankcase, and not in the working cylinder.

The wrist pin and bearing would follow previous work at FMA which employed so-called palm-end bearings. This design used production machinery to form the bearing holding groove in the crosshead and used the same bearing insert as a production diesel engine; this method of maximizing bearing loading area is successfully being used now on truck diesel engines (Detroit Diesel among several).

Because of the modest conditions in the crankcase, it is envisioned that the case be cast in aluminum alloy and split through the plane of the shaft and crosshead guides. This would allow easy moulding techniques and two-axis line boring; all internal areas are accessible, and during assembly components are

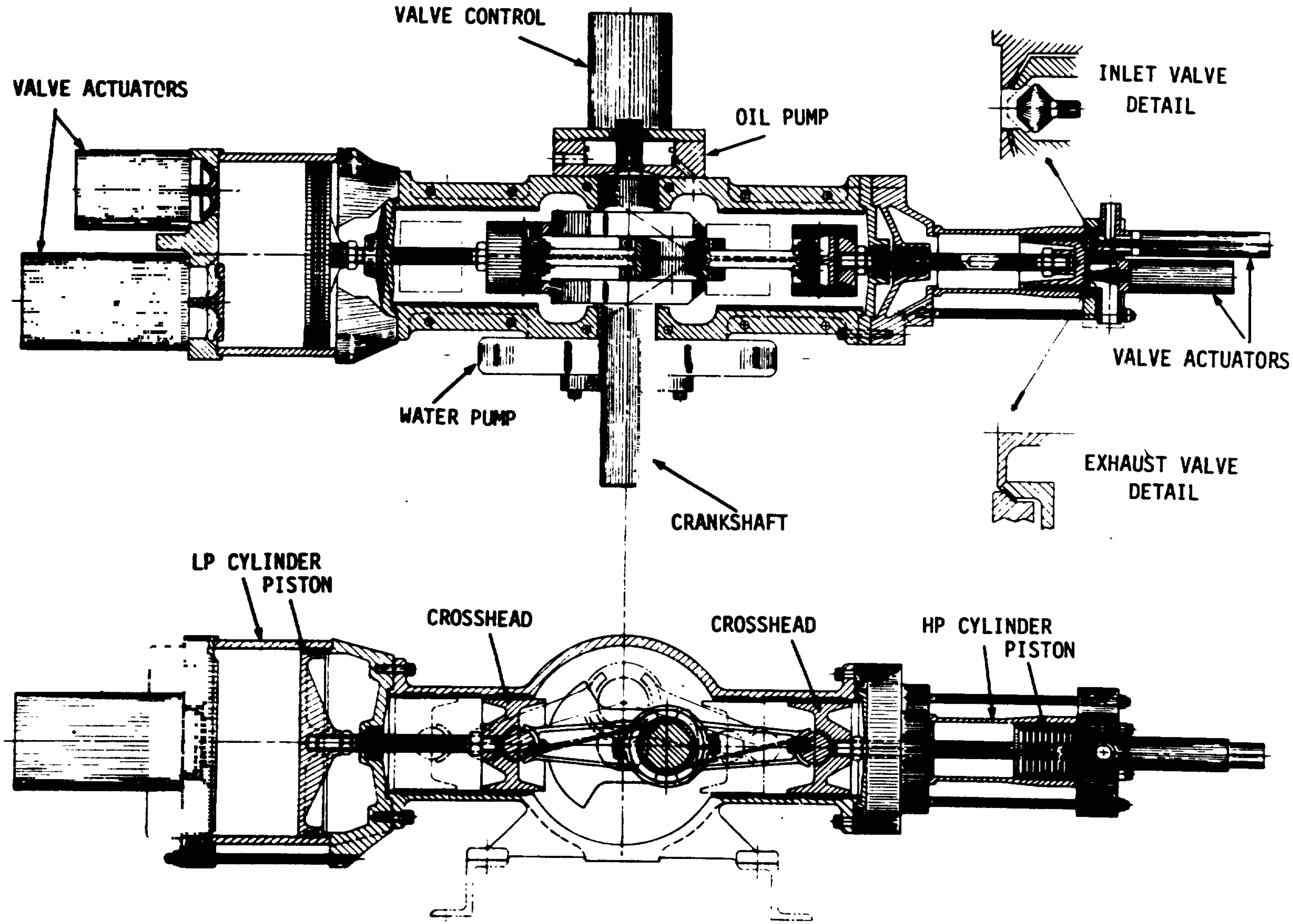


Figure 13. - Expander - section views.

conveniently laid in place. As the two crankcase halves are bolted together, the crosshead sleeves are clamped in position; no special tools or presses are required to change sleeves (or bearings).

The piston rod seals are held by simple disc shaped aluminum covers which form the end caps for the oil lubricated compartments. From the center axis to these points, all materials operate at moderate temperatures (the crankcase operates at atmospheric pressure) which can be controlled by an oil cooler if it proves necessary. Nearly all heat in the crankcase is due to conduction from the engine cylinders. In order to have best engine performance, that is, least heat lost to the crankcase, the case should be allowed to operate at the higher temperature levels consistent with long oil life. Since heat conduction is proportional to material conductivity, dimension, and temperature difference, the first two can be minimized by design, and the last by the selection of crankcase operating temperature level.

Many of the parts of the cylinders and heads can be made of Meehanite, a high grade cast iron with controlled Pearlite and graphite formation.

The pistons may require mechanical access to the ring grooves to allow insertion of piston rings for fitting as in the case of cast iron rings, or be of two or three piece design. They will be operating at an average temperature between supply steam and exhaust - easily within the range of Meehanite.

A major feature of the expander design is elimination of oil lubrication in the cylinder/ring area. There are two important reasons to eliminate the need for cylinder lubricating oil, the first being the need to prevent oil fouling and decomposition in the steam reheater, the second being the desirability of keeping oil out of the condenser and water tank. The benefits of oil-free operation are a substantial reduction in maintenance requirements, cleaning and refilling; the complexities and need of oil separation equipment are obviated. It is projected that the Rankine system under consideration can run for many months without attendance, perhaps even years. A major future benefit of operation without oil is the elimination of steam temperature restrictions due to the oil itself. As far as the piston, cylinder and cylinder head are concerned, if these components are actively cooled, the steam temperature could approach those found in I.C. engines - several thousand degrees peak. The temperature limit would now be dictated by steam generator materials and technology.

The design technique for eliminating the need for oil lubrication in the cylinders is the use of carbon/graphite piston rings. The carbon/graphite material itself has an inherent lubricating capability and can provide long life and low friction where used against a polished cylinder wall. Previous testing on an engine without uniflow exhaust ports showed excellent results for low wear and long life (Ref. 3). It is projected by several suppliers that carbon rings can handle the higher temperatures of the current design, and their higher cost will be offset by improved efficiency and elimination of auxiliary equipment.

A second concept that was considered to control steam leakage by the piston involved the equivalent of labyrinth seals. The effectiveness of this design would depend on how accurately the piston to cylinder clearance could be made *and* controlled through the operating temperature range of parts. These problems could possibly be solved with the use of low-expansion materials, geometric design (tapered cylinder bore, for example) and cylinder wall/piston cooling. For the present, the more likely approach to control leakage appears to be the adaption of carbon piston rings. A concept reserved for future consideration is to combine the labyrinth system with carbon rings - in effect leaky carbon rings. This concept is not possible in I.C. engines because leakage here would burn the rings. The benefits of such a design could be a relaxation of the requirement for close tolerances, a reduction in the number of rings required and lower friction.

The valves would be made from I.C. engine valve materials but used, in effect, in reversed positions. I.C. inlet valve materials would suit steam engine exhaust valve conditions, and that for I.C. exhaust would serve for steam inlet conditions. The operating environments for valves in steam engines is much less severe than in I.C. engines; a leak in the latter would result in a burned valve and self destruction while a leak under steam will not increase the valve temperature above normal *at all*.

The inlet valve manifold cartridges (allowing easy replacement and thermal isolation of inlet steam from the cylinder head) would be made of stainless steel or perhaps *Incoloy* 800H, since they are subject to full steam temperature and continuous supply pressure. Stellite valve seats would be provided in similar fashion to I.C. practice.

The outlet valve manifolds are similarly replaceable cartridges, but operate in much more modest temperatures and can be made of Meehanite.

The cylinder heads and liners could be made of Meehanite, because like the pistons, they operate at an intermediate temperature range between steam supply and exhaust, again, within the application range of this cast iron.

FMA has devised a hydraulic (feedwater) valve operator scheme which can follow diesel engine injector practice in design and construction and offers the following advantages:

1. Fast and variable inlet valve closure (cut-off)
2. Reduction of valve train inertia
3. Elimination of valve stem leakage
4. Automatic system pressure regulation.

One of the severe problems in steam engine design is the control of steam leakage by the valve stems or operating rods. In the past, this has generally meant the use of asbestos/graphite packing systems which have proven successful. Now with the requirements for high efficiency, next to no maintenance, and the desireability for higher pressures to reduce system size and weight, a virtually leak-proof system is required.

In the FMA design, high pressure feedwater is the valve actuating medium, with pressure applied to direct-acting pistons attached to the valve stems. The clearance around the pistons is selected to allow enough continuous water flow to keep the piston end below saturation temperature to prevent vapor lock. The average pressure level is adjusted to nearly balance the supply steam pressure, hence the leakage of steam to the piston actuator, or vice-versa, is at a minimum. A relatively long valve stem is used, moving within a matching guide with close fit, to provide thermal isolation for the actuating piston. Because of the practically zero flow through the close fit, the temperature gradient along the stem and its guide are almost identical as are then their growths due to thermal expansion. So not only is there virtually no leakage, but also there is provided an automatic means to control what in I.C. practice would be called valve clearance. This is required to ensure both complete closing of the valve and limited valve/seal closing contact speeds.

The FMA system also provides a minimum valve train inertia due to the direct coupling of the piston and the valve stem. The inertia of the actuation medium can be maintained at a low level by the selection of the connecting piping sizes. There is no inertia associated with the rotary distribution/control valve.

The rotary valve itself controls the opening and closing (cut-off) of the valves by varying the axial position of a moveable sleeve in its housing, exposing more or less flow area through slots in the valve drive shaft. In effect, it is a distributor of feedwater pressure to the valve actuating pistons. Both act together to provide a valve motion highly desirable in steam engines: one of variable duration but of constant lift.

Automatic system pressure control is provided by allowing increasing feedwater pressure to bias the moveable sleeve in a direction to allow longer cut-off and therefore more steam consumption (and more power) until the supply pressure drops again. In a solar electric system with power grid connection, the engine will automatically adjust power level in response to a change in solar input. The expander will maintain boiler pressure with the cut-off system while the pump flow will respond to maintain boiler temperature.

The feed pump is a three cylinder, single acting ram configuration as shown in Figure 14 and is directly mounted on the drive-shaft face of the expander crankcase. At the cost of adding to the number of components albeit identical, the three cylinder arrangement gives several desirable traits: the flow pulsations are minimized; the loading on the drive eccentric is reduced; there is redundancy in the event of a solenoid failure; and three cylinders are available for solenoid valve control, giving a somewhat modulated water flow (rather than on/off in the single cylinder case). The same basic design has been used by FMA for many years and has given excellent service.

4.3 System Configuration

The basic configuration of the system is shown in Figure 15. The design calls for the engine and alternator to be mounted adjacent to the receiver, out in front of the collector. In order to keep cantilevered weight to the minimum, the condensing system and water storage tank are placed at and below ground level. One advantage there is that space is not restricted in the vicinity of the condenser and a large natural draft stack may be used to provide cooling air flow. Another advantage is that a large water storage tank can be used which allows long periods between makeup water addition, if necessary, and the same tank acts as an underground freeze protection system for virtually unlimited periods.

The dish mounted Rankine cycle engine package has relatively few restrictive requirements as to position or orientation. There is no need, for example, to have the expander cylinder head(s) immediately adjacent to the receiver. Both high temperature steam supply lines can be run some distance from the steam generator, but do require insulation. In the layout of the power subsystem at the receiver, it was considered more important to keep the center of gravity as close as possible to the dish hinge point to reduce the cantilevered load than to keep piping short. Due to the significantly larger component weight of the alternator, it was placed next to the receiver as shown in Figure 16 which resulted in 0.6m (2 ft) longer steam and water lines between the

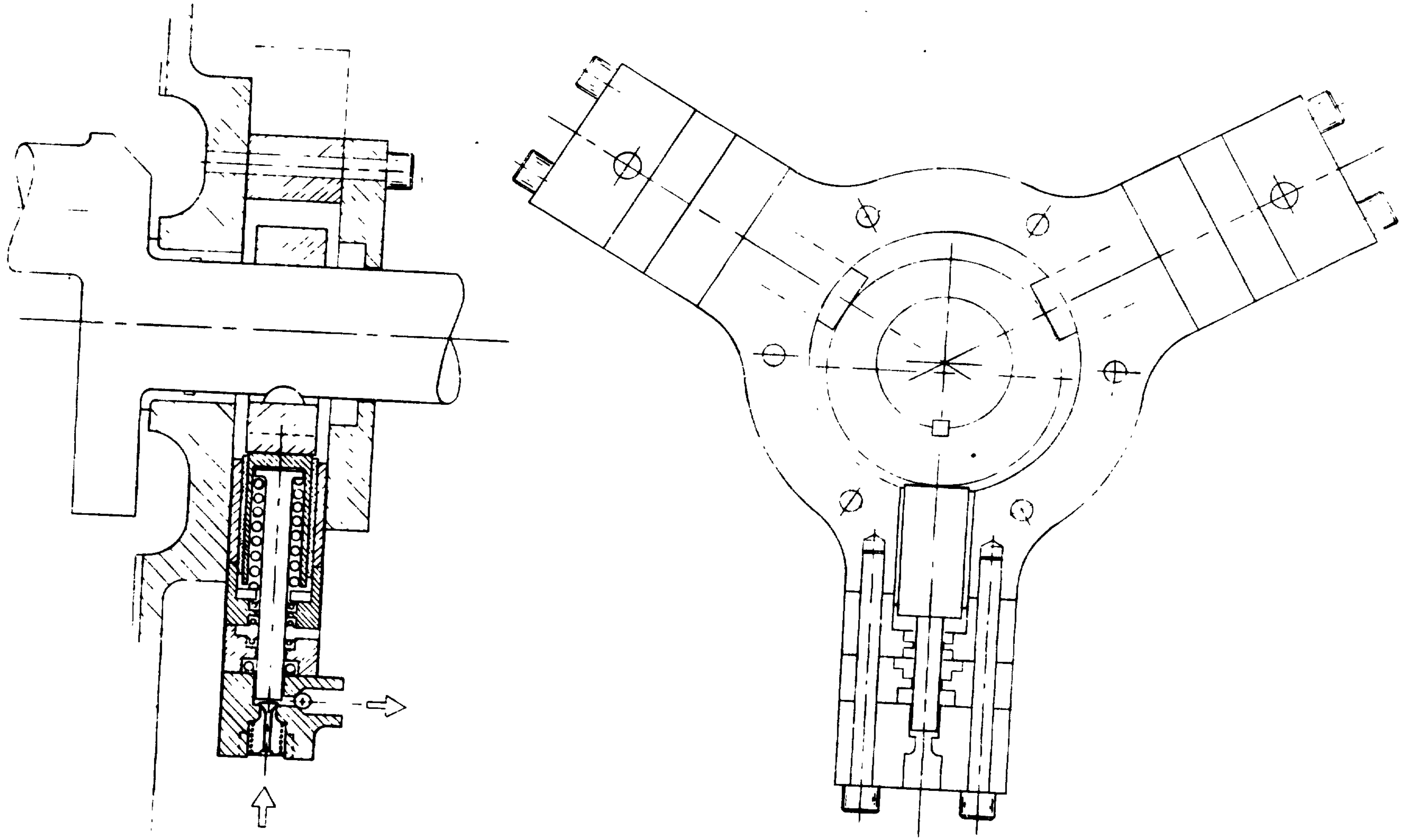


Figure 14. - Feedwater pump.

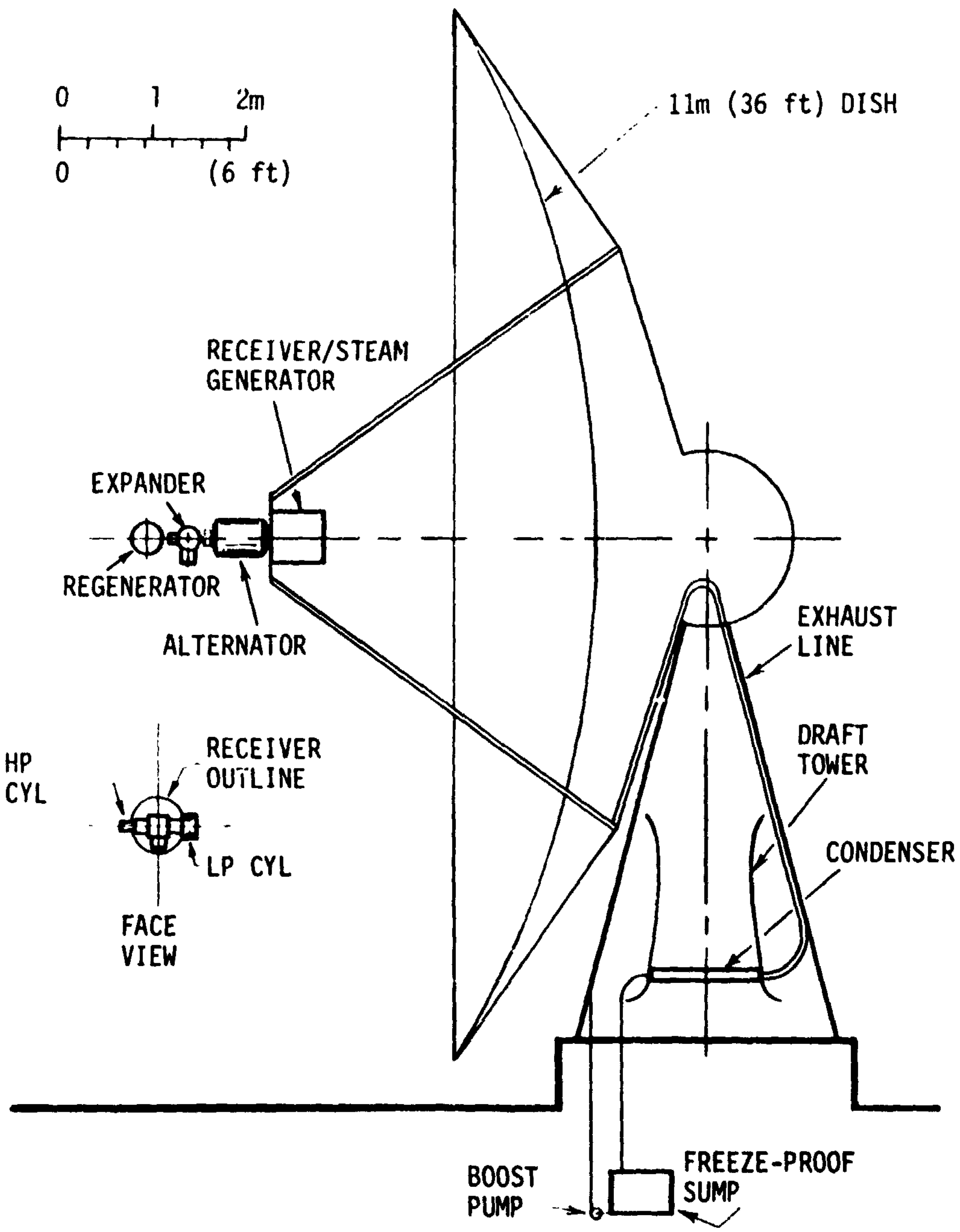
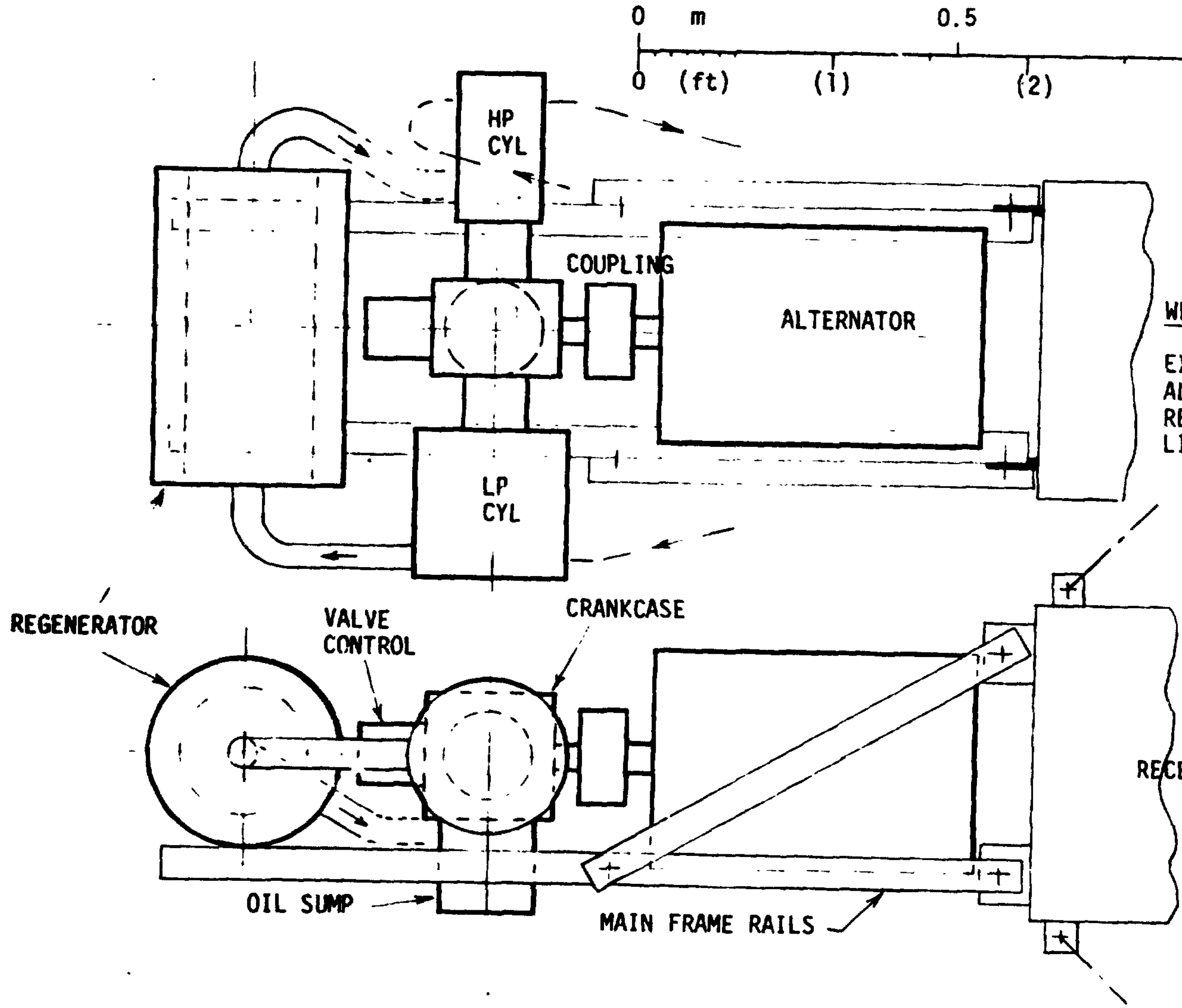


Figure 15. - FMA solar Rankine engine/alternator with collector/receiver.



WEIGHT ESTIMATE	kg	lb
EXPANDER	50	110
ALTERNATOR	160	350
REGENERATOR	5	10
LINES AND FRAME	15	30
TOTAL	230	[illegible]00

Figure 16. - Solar Rankine engine - mounting on receiver.

expander and the steam generator. Position does remain a matter of choice, and could be changed.

The expander itself can operate in any orientation, but it is desirable to arrange the crankcase to be self-draining to the oil sump. Unless already so provided, the alternator would require a thrust bearing to carry the weight of the armature when it is operating with its axis nearly vertical.

The present design for the expander shows the simplest possible layout for the crankshaft, and in this configuration, an unbalanced inertia force does exist. In the development of this conceptual design to the next stage of the prototype design, the exact weights of the reciprocating parts can be evaluated and a close estimate of the unbalanced forces can be made. In any event, a completely balanced expander can be had by including a balancing shaft as is presently done on some I.C. engines. This would add only a small increase to the cost of the unit, the shaft being a noncritical casting and the required drive gearing being quite straightforward. The result would be a smooth running expander with no vibration to transmit to the rest of the tower system.

The regenerator which is also part of the dish mounted package can be of very simple construction as shown in Figure 17. The high pressure feedwater is contained within a helically coiled stainless steel thin wall tube. The tube is in turn wrapped with aluminum fins which are bonded to the tube. This particular commercially available finned tubing has been used in many past configurations as part of steam systems with excellent results. The coiled finned tubing is fitted in the annulus formed between two sheet metal cans whose only purpose is to force the exhaust steam through the finned tube matrix. The center can is a plug, and the outer one has virtually no pressure requirement. The modest inlet temperature of the exhaust steam allow inexpensive fiberglass insulation to be used.

Once the exhaust steam has passed through the regenerator, it now travels to the condenser on the ground. Its pressure is almost atmospheric and any heat losses in transit, in fact, reduce the load on the condenser; the tubing involved can therefore be very lightweight and uninsulated, and a simple low cost hose can provide the flexible connection at the dish hinge point.

The condenser itself can be fabricated, according to the least expensive production notions. It can follow present automotive practice and be made from copper, aluminum, or even galvanized steel; its core would be thin to allow use of a natural draft stack; the core and headers have no pressure requirement.

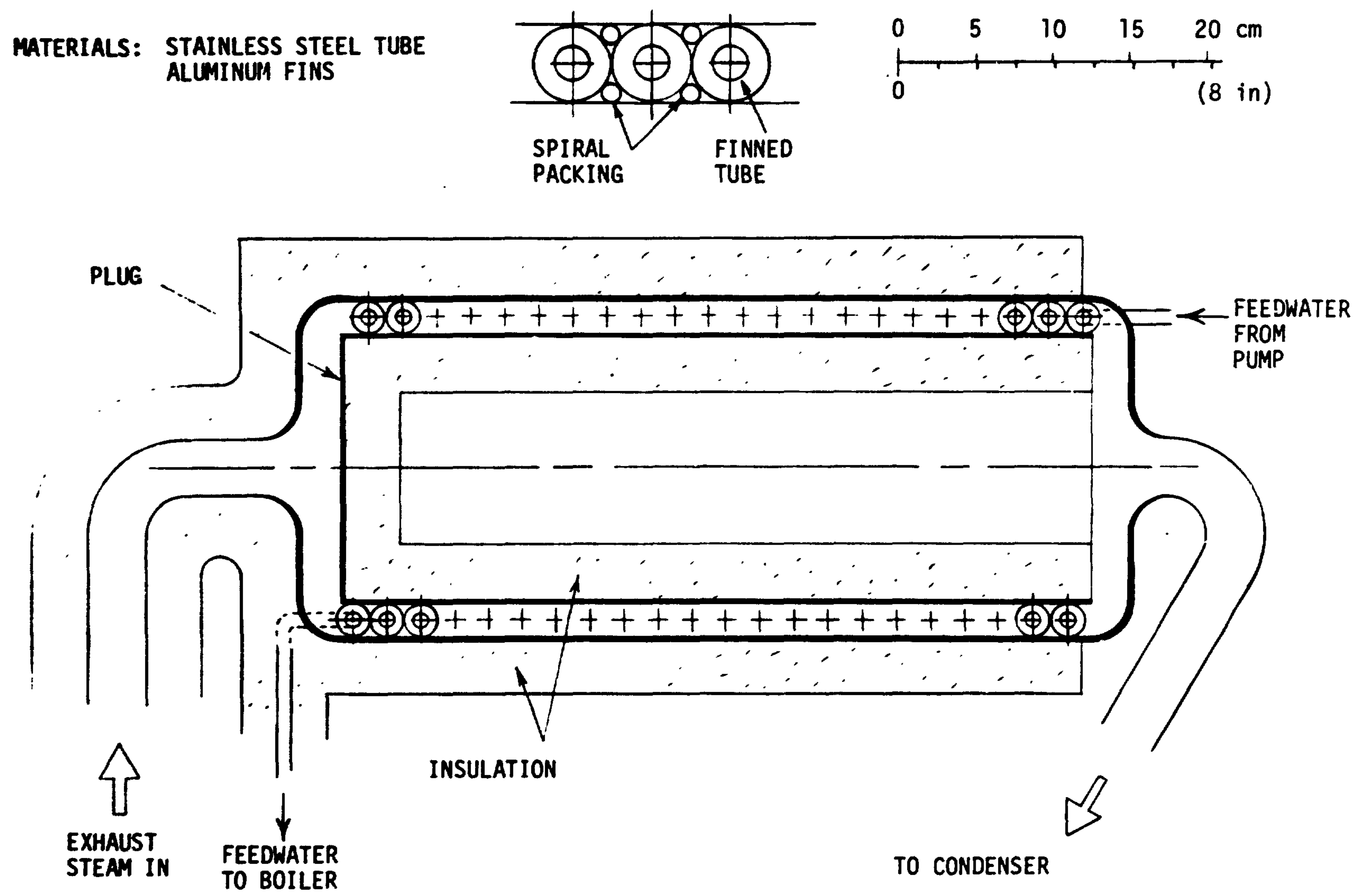


Figure 17. - Regenerator.

The draft stack could be provided by merely fitting the dish tower with fiberglass panels.

The condenser drains by gravity to an underground tank. It is seen then that the condensing system is a passive device; as its loading is reduced, the air/steam interface simply moves further into the condenser core. A simple device for maintaining hot condensate (or sump water) temperature would be a damper system operated by a bimetal spring sensing rejection air temperature. Keeping the sump just below the boiling point also provides deaeration of the feedwater.

The underground tank/sump can be made large as practical to give a long time period between makeup water addition (to replace water lost through leaks and evaporation). The tank benefits from the added insulation of the surrounding earth, and would never freeze if below the frost line. It provides a source for the start up of a system that could have its above-ground components well below freezing. Previous test experience has shown that the FMA feed pump, after being cold soaked overnight at 244K (-20F), having access to 283K (50F) water was able to draw a suction, prime itself, and begin pumping at pressure without having the water freeze within.

4.4 System Operation

Refer to the flow diagram of Figure 11 and the overall layout of Figure 15 as a guide to the discussion of system operation.

Upon the signal that the solar input level is high enough to begin operation, an electrically driven boost pump draws water from the below-ground freeze-protected sump tank and delivers it through the check valves of the main feed pump to the steam generator. During this period, the dish has been brought into position to focus the sun's rays on the receiver. When sensors detect that steam is being generated and has reached a prescribed pressure or temperature, the main alternator is run as a motor to turn over the expander. After a few turns the expander will be self-sustaining and begin to pump more water to bring the system pressure up to operating level, and begin to produce electric power.

The control of the entire subsystem is based on regulating the temperature of the outlet steam. Temperature sensors in the steam generating coils control solenoids on the feed pump inlet valves which provide changes in the water flow. Each solenoid is designed to hold its companion inlet valve open for a variable period of time which has the effect of unloading that particular pump cylinder; therefore, water is not raised in pressure, only to be bypassed with the associated energy waste. The solenoid is cycled on and off; as the steam temperature rises, the off time proportion is increased with a resulting increase in water flow rate. With the increased flow, the system pressure rises which immediately causes the expander valve controller to go to longer cutoff, tending to bring the pressure back to the operating level. The expander is now swallowing more steam and the power output is increased.

When the engine/generator set is coupled to the local power grid system, the grid provides a built-in speed control. The induction generator operates at a speed slightly higher than synchronous for 4-pole, 60 Hz electric machines, in the range of 193 rad/s (1840 rpm) depending on load, in order to deliver electric power. If the solar thermal input increases, the engine power increases and its output torque (primarily) and speed (some) increase. In the case of induction generators, the slip speed increases; for synchronous generators, the slip angle increases. In both cases, the output voltage increases as necessary to allow the increased current (the primary variable) to be delivered to the grid. Thus, the grid acts like a stiff damper for the engine/generator set. At zero electric output, the set would "idle" at synchronous speed. The generator thus provides a speed control on the engine allowing only a slight change in response to a corresponding change in engine power available. It should be noted that the electric power output is controlled by the grid reference at 60 Hz regardless of the speed. Also, the unit can operate with only 60 Hz reference and need not be connected to a full grid.

The expander/water pump can respond quickly to thermal input changes as measured by steam generator thermocouples; the expander likewise responds quickly to the resulting pressure changes. Thus no special or additional components are required to interface with the receiver for transient control. Again, the solenoid valve control system provides a built-in receiver cooling safety system - extra water can be pumped to cool the steam generator with an overpressure safety dumping the extra steam. Likewise in the worst-case event of expander stoppage, the overpressure trip would vent the steam lines and the boost pump could continue to supply water during the time the dish is moved out of focus.

SECTION 5

IMPLEMENTATION ASSESSMENT

5.1 Development Status

The basic design of this expander is the same as present high production I.C. engines, even departing from older steam engines' use of slide valves to use I.C. engine poppet valves. All of the parts of the engine/generator system, save the expander piston seals and perhaps certain components of the valve system, represent nearly standard production parts which involve no development. It is intended, for example, that eventual production follow many practices of the manufacturer of small gasoline engines; specifically a die-cast aluminum crankcase is appropriate for the expander as is a cast-iron crankshaft. Existing oil pump units, oil seals, water seals are suitable.

The primary development area remains that of the piston seals and it is an area that has already shown success. Development is needed not so much to show if carbon piston rings will work, but to find out if their performance can be improved. The same is true of the labyrinth piston seal concept, and detailed study can show if their combined use can be advantageous.

The secondary area of development is design of the water-hydraulic valve gear. There is no inherent problem to be overcome; rather, this hydraulic system offers a unique solution to several combined problems (valve stem leakage, variable duration valve opening, automatic pressure regulation) and the design problem, as such, is material selection and component configuration, not the development of a material with heretofor unavailable characteristics.

In both of the above cases, since the "problems" are well known and defined, and the solutions do not depend on the discovery of a certain material by a certain time, the likelihood of the success of this expander is very high and the time to be ready for production is probably the shortest of any competing system.

In terms of producibility, practically all the parts of the power system are in production in some form. Although the carbon piston rings do represent components that would require new production techniques for an engine manufacturer, they are in production by carbon fabricators. It is concluded that no part represents something that would delay or otherwise impede the introduction of the power subsystem into general use.

The prime mover should prove as durable as the previous generations of steam engines. The contamination-free nature of the crankcase, coupled with the oil-free operation of the working fluid zones should provide long periods between engine checks. There are no finite-lived components such as spark plugs or points which must be changed at intervals as in I.C. engines.

By design, all parts that are subject to wear are individually replaceable pieces. For example, the valves themselves are supplied as cartridges which include the valve seat. Replacement can be as simple as changing a spark plug - the cylinder head need not be removed for valve seat grinding. The cylinders themselves are sleeves which are changeable without special tools. It is conceivable that an entire engine overhaul could be done on site within a few hours.

Future efficiency improvements can come through increases in steam temperature and in reduction of mechanical losses. In a future generation, the higher temperatures could be handled by actually water cooling the cylinders and heads regeneratively; the main problem is less in the expander and more in the solar-steam generator. If the high steam temperatures can be provided, the expander can be modified to handle them. Reduction in mechanical losses can come from development of reduced friction components in the expander; these improvements would follow naturally with the maturation of the engine system.

5.2 Cost Analysis

5.2.1 Expander Cost. - The cost of a steam expander is on the same order as a diesel engine. Most of the parts can be manufactured on the same equipment; indeed, the "soft" loading of a steam engine would allow some parts to follow the light and cheap practice of lawn mower gasoline engines (the crankshaft could be cast iron). Only a small amount of high temperature alloy would be required for parts of the valve system; the temperature of the cylinder head itself is within the capability of Meehanite, a high grade cast iron.

The following cost estimates have been broken down by production quantity into three groups: small quantities for experimental use (less than 10 units), low quantities (up to 2000 units per year), and high production quantities. These costs consider *only* the expander, with cut-off control, oil filter and feedwater pump, as described in the parts list of Table 5.

1. Small quantities - $20,000 per unit (selling price)
2. Low quantities - $2,000 per unit (factory cost)
3. High quantities - $730 per unit (factory cost).

TABLE 5. - EXPANDER PARTS LIST

1. Crankcase
2. Crankcase bolts (8 short)
3. Crankcase bolts (4 long)
4. Crankshaft
5. Main bearings (2 pair)
6. Shaft oil seal
7. Shaft oil seal holder
8. Shaft oil seal bolts (4)
9. Oil pump unit
10. Oil pump housing
11. Oil pump housing cover
12. Oil pump "O" ring
13. Oil pump shaft oil seal
14. Oil pump bolts (4)
15. Big end bearings (1 pair)
16. Connecting rods (2)
17. Retaining rings (2 pair)
18. Retaining rings bolts (4)
19. Wrist pins (2)
20. Wrist pins bolts (4)
21. Wrist pins bearings (2)
22. Crossheads (2)
23. Crossheads sleeves (2)
24. HP rod seal holder
25. HP rod seals (2)
26. HP steam seal holder
27. HP steam seal holder bolts (6)
28. HP steam seals (2)
29 HP steam seals retaining ring
30. LP rod seal holder
31. LP rod seals (2)
32. LP rod seal bolts (6)
33. HP piston rod
34. LP piston rod
35. Piston rod locknuts (4)
36. HP piston
37. HP piston lock ring
38. HP carbon piston rings (9)
39. HP carbon piston rings carriers (9)
40. LP piston
41. LP carbon piston rings (2)
42. LP carbon piston rings carriers (2)
43. HP cylinder
44. LP cylinder
45. HP cylinder head
46. HP cylinder head studs (6)
47. HP cylinder head nuts (6)
48. LP cylinder head
49. LP cylinder head studs (6)
50. LP cylinder head nuts (6)

TABLE 5. - EXPANDER PARTS LIST (CONTINUED)

51.	HP inlet valve
52.	HP exhaust valve
53.	HP inlet cartridge
54.	HP exhaust cartridge
55.	HP cartridge retainer (4)
56.	HP cartridge bolts (4)
57.	HP inlet guide
58.	HP inlet guide bushing
59.	HP exhaust guide bushing
60.	LP inlet valve
61.	LP exhaust valve
62.	LP inlet cartridge
63.	LP exhaust cartridge
64.	LP cartridge retainers (4)
65.	LP cartridge bolts (4)
66.	LP inlet guide
67.	LP inlet guide bushing
68.	LP exhaust guide bushing
69.	HP inlet valve actuator piston
70.	HP inlet valve actuator cylinder
71.	HP inlet valve actuator stem
72.	HP inlet valve actuator cover
73.	HP exhaust valve actuator piston
74.	HP exhaust valve actuator cylinder
75.	HP exhaust valve actuator stem
76.	HP exhaust valve actuator cover
77.	LP inlet valve actuator piston
78.	LP inlet valve actuator cylinder
79.	LP inlet valve actuator stem
80.	LP inlet valve actuator cover
81.	LP exhaust valve actuator piston
82.	LP exhaust valve actuator cylinder
83.	LP exhaust valve actuator stem
84.	LP exhaust valve actuator cover
85.	Valve controller housing
86.	Valve controller cover
87.	Valve controller shaft
88.	Valve controller spider key
89.	Moving sleeve
90.	Guide pins (3)
91.	Springs (3)
92.	Water seals (3)
93.	Water seals carriers (3)
94.	Water seals retainer
95.	Distributing pipes (4 different)
96.	Fittings (8)
97.	Water pump body
98.	Water pump bolts (6)
99.	W. P. eccentric

TABLE 5. - EXPANDER PARTS LIST (CONTINUED)

100.	W.P. eccentric key
101.	W.P. tappets (3)
102.	W.P. tappets bushings (3)
103.	W.P. rams (3)
104.	W.P. rams seals (9)
105.	W.P. rams springs (3)
106.	W.P. spring retainers (3)
107.	W.P. heads (3)
108.	Inlet valves
109.	Delivery valves (3)
110.	HP seals (3)
111.	Oil seal holders (3)
112.	Water seal holders (3)
113.	Fittings (6)
114.	Bolts (6)
115.	Oil sump
116.	Oil sump pickup
117.	Oil attachment
118.	Oil suction tube

For the pricing of the small quantities of experimental hardware, we have taken an hourly rate of \$36 which is a representative value that is all inclusive - overhead, fees, mark-ups, cost of money, etc. It includes rates primarily for skilled labor - model makers, technicians and engineers. For the production hardware, the rates include only the factory overhead. The figures for the production hardware represent only factory costs and should be appropriately marked up for sale, depending upon the particular manufacturer's financial policies and practices.

5.2.2 Total System Direct Cost. - The contract called for a direct cost estimate excluding overhead, capitalization of equipment, selling costs and profit. A make or buy assumption was still required as a large number of parts and components would be purchased complete from traditional engine component suppliers.

Table 6 lists the estimated purchases and direct shop costs for a small factory dedicated to assembling the solar engines. The primary shop work would be the complete manufacturing of the expander. Even within the expander, a substantial cost fraction would be purchased parts and rough castings.

Table 6. DIRECT COSTS

Direct Costs: 26 kW Solar Rankine Electrical Power Subsystems	Purchased Complete Components	Direct Shop Cost
Expander (with cut-off control, oil filter, and feedwater pump)		365
Alternator (with line power cranking)	430	
Regenerator 13m x 2.2 cm diam (42 ft x 7/8 in.) fin-tube	80	10
Condenser (radiator core)	30	
Boost Pump (2 speed electric)	20	
Steam Overpressure Relief (with switch)	20	
Shaft Coupling		10
Water Demineralizer Cartridge	10	
Water Tank (buried concrete sump) (could be integral with tower foundation)	15	
Draft Stack (fiberglass) (can be part of tower structure)	20	
Pipes and Frame		
Exhaust "Down Spout" (atmospheric pressure)	5	
Boost Pump to Feedpump (cold)	10	
Feedpump to Regenerator (cold)	5	
Regenerator to H.P. boiler (warm)	10	
Boiler to H.P. Cylinder (hot)	10	
H.P. Exhaust to Reheater (warm)	10	
Reheater to L.P. Cylinder (hot)	15	
L.P. Cylinder to Regenerator (warm)	5	
Insulation on High Temperature Lines	5	
Engine Frame (on dish assembly)		20
Control System		
Computer (start & sync. logic, water flow loop on boiler temperature, expander includes pressure control, emergency shutdown and cool logic and fault signals)	100	50
Thermocouples (boiler, regenerator, condenser)	80	
Water Sump Low Level Switch	1	
Speed Pickup	5	
Oil Pressure Switch	1	
Feedpump Flow Control Solenoids	15	
Feedpump Emergency Bypass Valve	10	
Bimetal Condenser Shutter System		10
Relays and Wiring	20	
Panel with Fault Lights		20
TOTALS	932	485

Estimation of the engine selling price while not requirement of the contract can be approached starting with the two cost elements in Table 6. Additional assumptions required include factory overhead, G&A/sales rates, capital investment and amortization policy, production rate, and profit margin.

5.2.3 Operating and Maintenance Costs. - Steam engines have always been equated with durability, long life, and ease of maintenance. As mentioned before, the application of load is nonviolent; there are no combustion contaminants in the working fluid nor entrainment of dust. Crankcase lubricant operates under conditions equivalent to a gearbox, again it is in effect isolated and can run for years without an oil change. The only area of uncertainty is the piston seal and two schemes have potential, precedence, and no requirement for cylinder lubrication: carbon piston rings (having been used on an engine for over 29 Ms (8000 hr) with no "marked wear". Ref. 3.

Engine maintenance should essentially consist of periodic checks of the oil level and checks for steam leaks. Degradation of the oil should be very slow, loss of oil should be minimal. We estimate initial oil changes once every 1000 hr, probably increasing to 3300 hr as experience is gained with the installation.

A check for steam leaks should be made approximately 100 hr after initial installation since leaks are likely to show fairly early. Periodic visual inspections, once every 500 hr, should be sufficient after that.

Engine overhaul should consist of a replacement of all seals, piston rings, intake valves and gaskets. We anticipate a minimum time between overhauls of 6600 hr.

Average operation 3300 hr/year of which the duty cycle is

1. 100 hr at full power
2. 400 hr at 95 percent power
3. 400 hr at 92 percent power
4. 400 hr at 90 percent power
5. 400 hr at 86 percent power
6. 400 hr at 83 percent power
7. 400 hr at 76 percent power
8. 400 hr at 65 percent power
9. 400 hr at 50 percent power.

Based on this schedule, oil changes would occur once a year, and overhauls once every two years.

Degradation of performance will occur during the period between overhauls, mainly due to leakage of seals. The degree of degradation will be determined during endurance testing.

The cost of maintenance (oil change and tightening of joints) is estimated at: 3 hr at $20/hr or $60/year.

Cost of overhaul is estimated at:

1.	Labor 8 × $20/hr		$160
2.	Material		140
			$300/overhaul
		or	$150/year

SECTION 6

CONCLUSION

The concept definition study of this steam Rankine reheat reciprocator system has brought forth the following conclusions. A high efficiency and low cost power system can be a practical near-term reality with the adoption of a compound expansion steam engine. This system offers an expander which closely follows conventional gasoline or diesel engine design and manufacturing procedures, requiring virtually no change in current production formats. The system uses an off-the-shelf electric generator, and requires no electric power conditioning, or phase or frequency control system. It is quiet. The expander is compact and low weight, and it is conceivable that an aluminum-framed generator can further reduce the system weight.

The expander has been designed with ease of maintenance in mind; it is simple and quick to fix due to careful consideration of the few critical areas (the prime example being the replaceable valve cartridges). No special equipment is needed to take apart or reassemble the engine. No special gas or fluid charging systems are needed since no special gases or fluids are used. The working medium is water, the most universally available liquid.

The expander makes use of readily available and low cost materials such as aluminum and cast iron. Only a small amount of high grade alloys are needed (in the inlet valve cartridges). While the proposed carbon piston rings will go through a grade selection process, no special materials need be sought or developed. There are no stringent requirements for seals - loss of working fluid is unimportant; in the shut-down condition, there is no pressure anywhere in the system.

There are only modest requirements for filters: since the system is closed, no large low pressure drop filters are needed as in the case of the open-cycle gas turbine. Oil filtration may be passive or perhaps not required at all due to gearbox like operating conditions of the crankcase. The water in the system can be self-filtering - once the storage tank (water pump) is filled with demineralized water, there is ample time during shut-down periods for any impurities (primarily wear particles) to settle. Dissolved gases are naturally driven off by maintaining the water sump near condensate temperature, approximately 360K (190F).

Summary of highlights:

1. 31 to 33 percent overall efficiency (≡ electric power out over heat supplied) maintained from full load down to half load.

2. Low weight at receiver: expander only ≃ 50 kg (100 lb); package weight including expander, alternator, regenerator and water pump ≃ 230 kg (500 lb).
3. Low cost: factory cost ≃ $1952 or ≃ $90/kW for complete system.
4. No oil in steam; potential for handling higher inlet temperatures as steam generator technology advances.
5. Ground mounted condenser, natural draft cooling.
6. Below ground water tank for freeze protection.
7. Minimum development required.
8. Compatible with thermal storage and fuel combustion augmentation.

SECTION 7

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SECTION 8

GLOSSARY

Expander Symbols and Terminology

Dimensional Characteristics:

CO = Cut-off. Fraction of stroke that intake valve is open.

CL = Clearance Volume. Fraction of piston displacement clearance at top of stroke.

b = Bore. Cylinder bore.

s = Piston Speed. Stroke × 2 × rotating speed.

Performance Characteristics (Expander):

r_p = Stage Pressure Ratio. Supply divided by exhaust pressure.

E_p = Pressure Effectiveness. Fraction of supply pressure delivered as mean piston effective pressure (indicated, brake, etc.).

η = Efficiency. For expander this is based on fraction of ideal isentropic work available over r_p.

η_i = Indicated Efficiency. Ideal efficiency of expander P × V diagram without heat, pressure and friction losses.

η_{BR} = Breathing Efficiency. Correction to indicated efficiency for intake and exhaust pressure losses.

η_{HL} = Heat Loss Efficiency. Correction to indicated efficiency for inlet steam cooling and expander heat rejection.

η_{mech} = Mechanical Efficiency. Fraction of indicated work delivered to output shaft after friction losses.

$\eta_{PM} = \eta_{EX}$ = Prime Mover or Expander Efficiency. Fraction of isentropic work available from r_p delivered to output shaft ($\eta_{EX} = \eta_i \times \eta_{BR} \times \eta_{HL} \times \eta_{mech}$).

IMEP = Indicated Mean Effective Pressure

BMEP = Brake Mean Effective Pressure.