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RANKINE CYCLE ENERGY CONVERSION SYSTEM DESIGN CONSIDERATIONS FOR LOW AND INTERMEDIATE TEMPERATURE SENSIBLE HEAT SOURCES

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ABSTRACT

Design considerations are described for energy conversion systems for low and intermediate temperature sensible heat sources such as found in geothermal, waste heat, and solar-thermal applications. It is concluded that the most cost effective designs for the applications studied did not require the most efficient thermodynamic cycle, but that the efficiency of the energy conversion hardware can be a key factor.

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RANKINE CYCLE ENERGY CONVERSION SYSTEM DESIGN CONSIDERATIONS FOR LOW AND INTERMEDIATE TEMPERATURE SENSIBLE HEAT SOURCES

Introduction

Energy extraction and conversion from low or intermediate temperature sensible heat sources present design considerations which are different in many ways from those associated with conventional fossil fuel fired systems, even though the same conversion process, i.e., the Rankine cycle, may be used. Examples of sensible heat sources for energy conversion requiring special consideration are hot liquid geothermal sources, industrial waste heat sources, and solar-thermal collector and storage systems utilizing a sensible heat transfer fluid. Design problems associated with other heat sources such as geothermal steam or thermal energy transferred by heat pipes or other phase change mechanisms will not be considered here.

In the above examples of candidate sensible heat conversion systems, peak heat engine cycle efficiency does not necessarily dominate as it does in conventional power plant applications. In a conventional power plant, we have a sensible heating medium in the form of the combustion products, but these hot gases (greater than 3000°F typically) are so much hotter than the practical peak cycle temperatures (typically 1050°F) that invariably the most economical system design dictates that cycle efficiency be maximized and as much heat as possible be added at the peak cycle temperature. The sensible heat sources cited as examples, however, are generally well below practical equipment temperature limits and thus are not subject to the same constraints as those found in a conventional power plant.

In all cases, the primary criterion for system design should be the most favorable cost per power out ratio considering the total system cost and the power out actually required. Examples of the unique considerations required for different applications utilizing sensible heat from low or intermediate temperature sources are given in the following section.

Examples of System Design

Example 1 -- Low Temperature (145°F) Geothermal Hot Spring Energy Conversion System.

In this application, the economics of generating electricity from a warm spring in Alaska was investigated. The actual power required (~25 kW$_e$) is much below the potential of the warm spring source. The source provides greater than 400 gallons/minute of 145°F water and an essentially unlimited quantity of 45°F water is available for cooling. Thus, even after allowing a factor of 2 to provide for anticipated growth in demand, we are left with a source potential significantly greater than the demand. Under these conditions, heat engine cycle efficiency is of secondary importance and the design constraint is minimum cost for the fixed power requirement. For this particular application, the only real variable in cost was the heat exchangers required since the other hardware was essentially fixed by the chosen power level and availability. The heat engine chosen was a simple Rankine cycle (as shown in Figure 1) operating with R12 as the working fluid. The cycle parameters were then varied and plotted vs. heat exchanger cost as shown in Figures 2 and 3.* The rather surprising result is that the optimum cycle parameters for minimum system cost are significantly different from those yielding maximum cycle efficiency.

In Figure 2, the total heat exchanger cost is seen to decrease with decreasing condenser hotwell temperature due to the improvement in cycle efficiency with the lower sink temperatures, reaching a minimum at about 63°F as shown in Figure 3. Below the 63°F point, the condenser area, and hence cost, increases since area reduction from further improvements in cycle efficiency are more than offset by the larger area required for the condensing temperature to approach the cooling water temperature more closely.

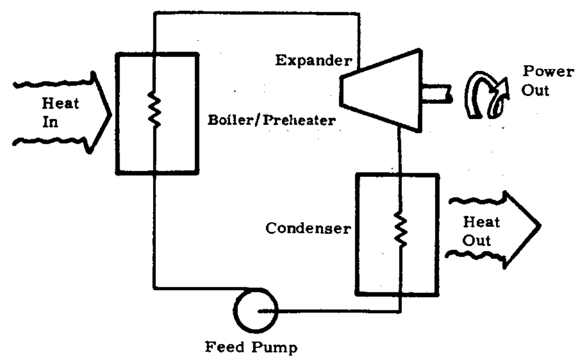

Figure 1. Simple Rankine Cycle System

For each condenser temperature curve in Figure 2, we note that the minimum total heat exchanger cost occurs at a peak cycle temperature of approximately 115°F. Below 115°F, the increase in boiler and preheater area due to the falloff in cycle efficiency from the lower peak cycle temperature dominates over the decreasing area effect of the greater temperature difference between the working fluid and the source water. The opposite occurs above 115°F. The design peak cycle temperature and condensing temperature were choosen to be 120°F and 60°F, respectively, to provide a heat exchanger area safety margin.

In general, where we have "free" sources of heating and cooling fluids such as geothermal and waste heat sources, etc., it would be desirable to perform analyses as represented by Figures 2 and 3 with the addition of power level as a third variable to arrive at the lowest cost per unit output for a given source potential.

*This work performed by Barber-Nichols Engineering, Arvada, Colo., under Contract 02-7895 with Sandia Laboratories.

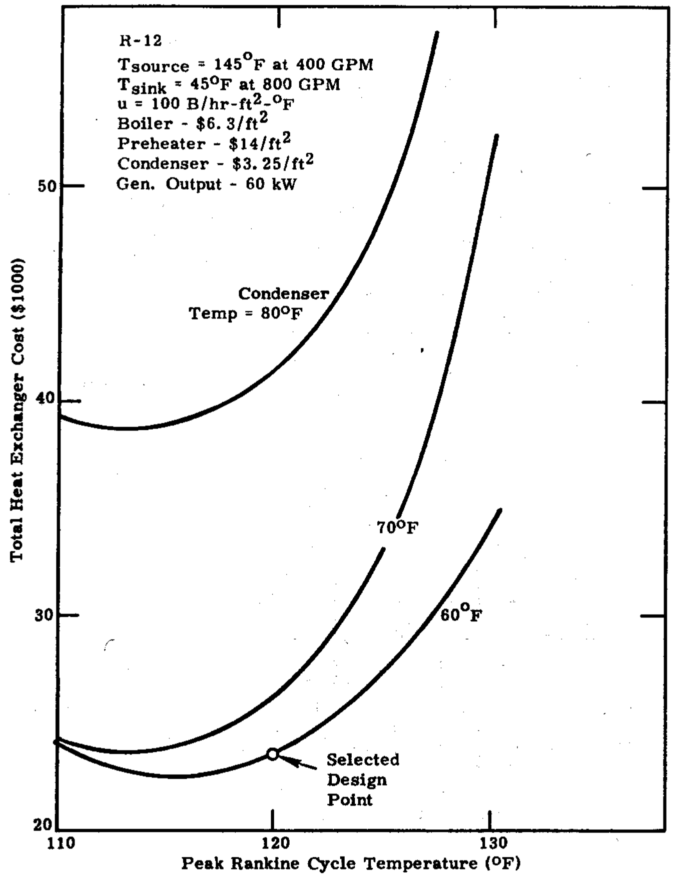

Figure 2. Heat Exchanger Cost for Various Cycle Conditions

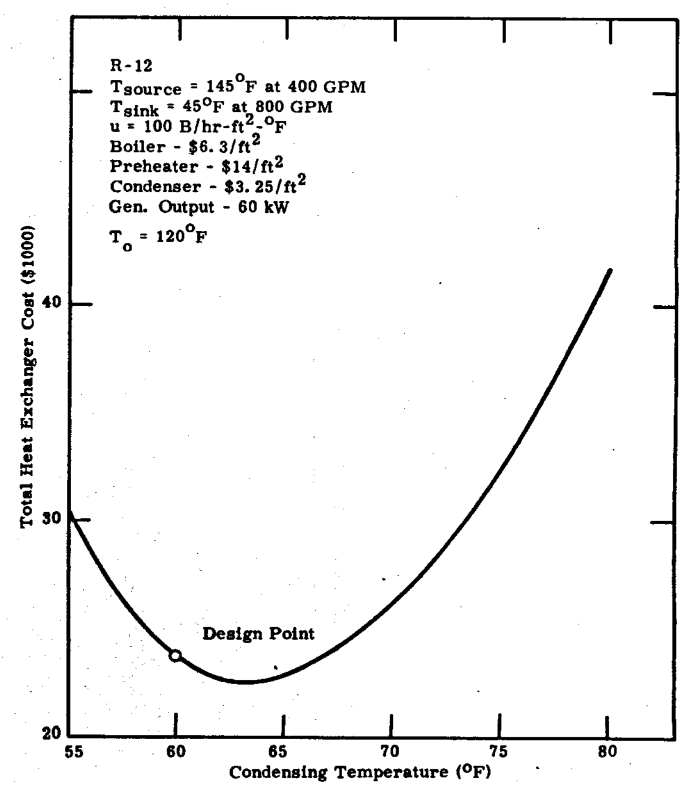

Figure 3. The Effect of Condensing Temperature on Heat Exchanger Cost

Example 2 -- Intermediate Temperature (590°F) Solar Energy Conversion System for Sandia's Solar Total Energy Prototype.

In this particular application, it is desired to determine the most cost effective cascaded total energy system utilizing a tracking parabolic trough-type collector system, a sensible heat thermal storage system, and a regenerative organic Rankine cycle heat engine as shown in Figures 4 and 5 to provide electrical power and low level heat requirements. See Reference 1 for a more complete description of the system.

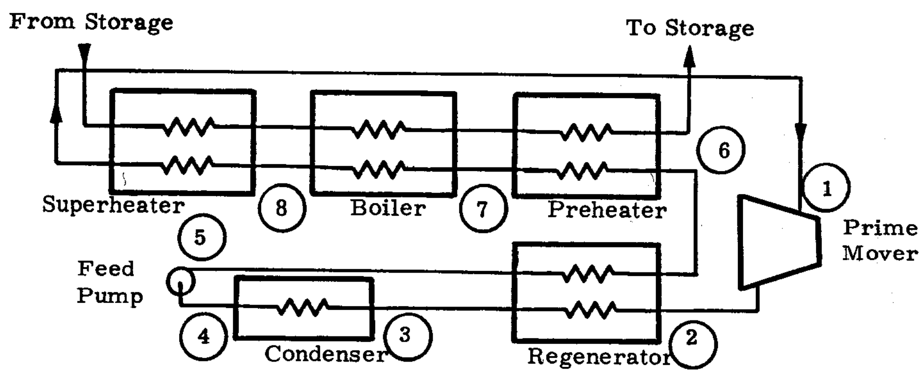

Figure 4. Schematic of a Superheat-Regenerative Rankine Cycle

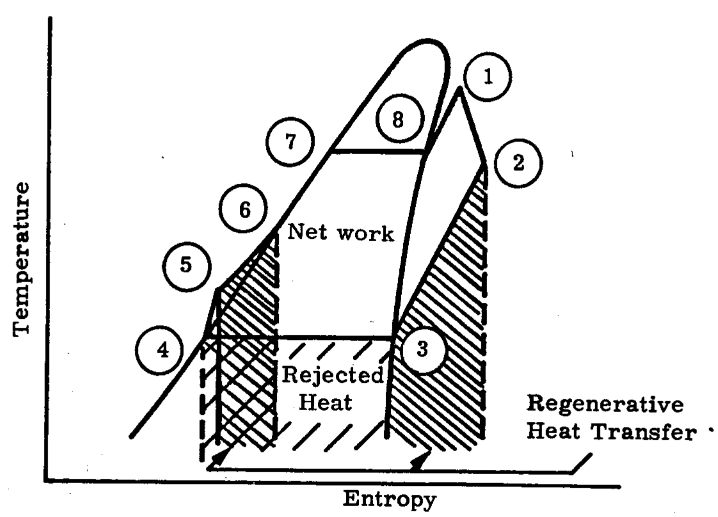

Figure 5. Temperature vs. Entropy Diagram for a "Drying" Type Working Fluid with Superheat-Regenerative Rankine Cycle Superimposed

The heat engine efficiency is important since heat engine thermal requirements directly influence both the number of collectors (generally, the highest cost subsystem) and the amount of heat storage required for a given quantity of equivalent mechanical or electrical energy. The collector efficiency is important in that it determines how many collectors are required to collect the thermal energy required by the heat engine. Unfortunately, the collector and engine subsystems have conflicting requirements for high efficiency, as sketched in Figure 6. The figure indicates that we have engine efficiency increasing with peak cycle temperature or average heat addition temperature while just the opposite is true for the collectors. Thus, the overall system efficiency, which in a simplified analysis can be considered as the product of the engine and collector efficiencies, will exhibit some optimum value for operating temperature. Combined collector-engine system cost will likewise exhibit a similar characteristic. As mentioned earlier, the solar energy will be extracted from the collectors, stored, and be given up to the heat engine using the sensible heat of a heat transfer fluid. Figure 7 indicates how the heat is transferred to the Rankine cycle working fluid by the heat transfer (storage) fluid. A high storage fluid temperature drop (ΔT) requires a smaller mass of sensible heat storage (with correspondingly lower storage cost) and less collector pump work, and it also improves collector efficiency by lowering the return temperature and thus the average temperature of the collectors. However, the engine cycle efficiency will suffer since the average heat addition temperature is lowered by the higher storage fluid ΔT as shown in Figure 8.

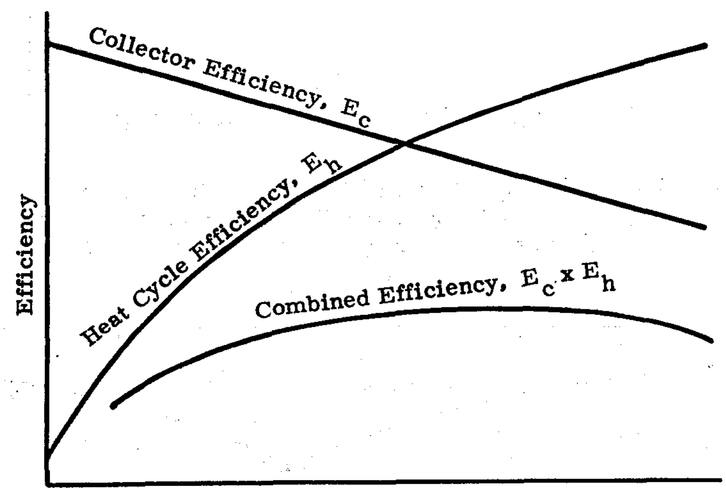

Figure 6. Efficiency vs. Upper Operating Temperature

The system parameters eventually chosen for the Sandia prototype were a design storage temperature ΔT of 115°F which results in approximately 125°F of superheat in the Rankine cycle working fluid (toluene) loop. These values appeared to offer a reasonable compromise between amount of storage capacity and collector field size for experimental purposes.

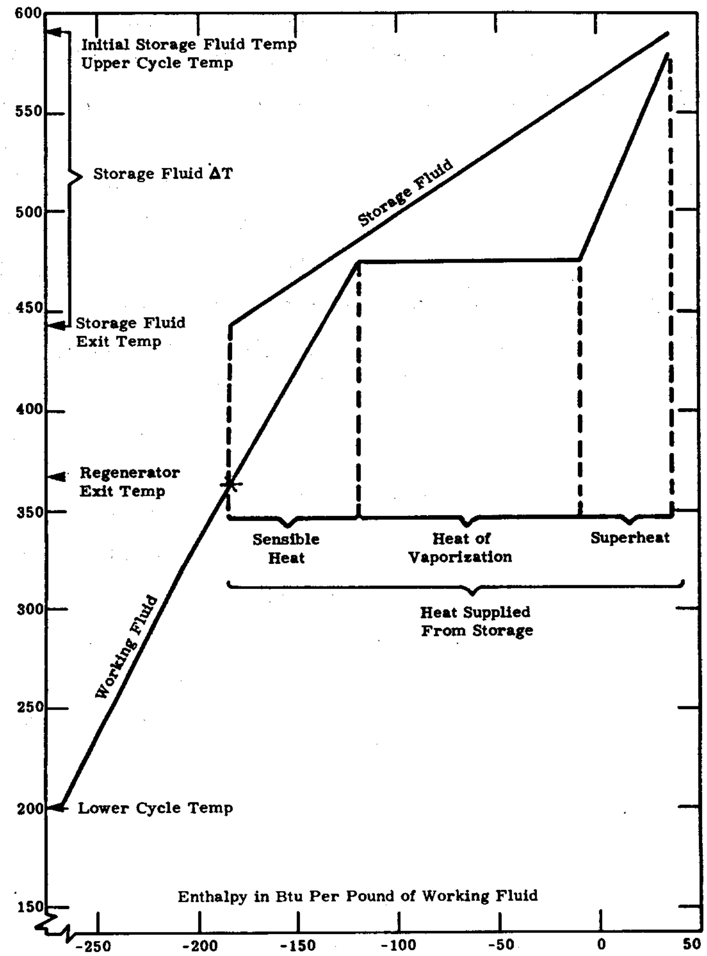

Figure 7. Storage Fluid and Working Fluid Temperature vs. Enthalpy of the Working Fluid for Prototype Solar Total Energy Rankine Cycle System

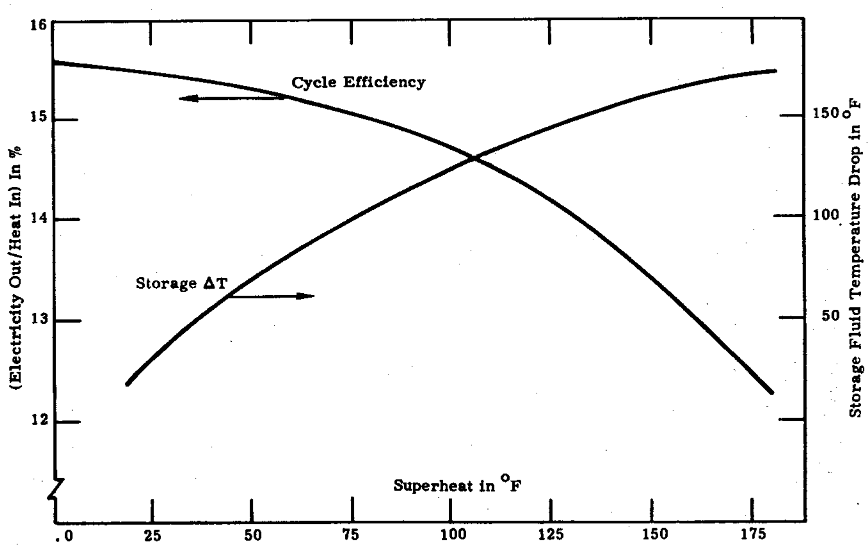

Figure 8. Cycle Efficiency and Storage Fluid Temperature Drop vs. Amount of Superheat. In All Cases, Peak Cycle Temperature is 580°F and Condensing Temperature is 200°F

Example 3 -- Large Scale Solar-Thermal Energy Conversion System Utilizing Intermediate (600°F) Peak Temperatures

In this example, several cycles and working fluids were examined to determine which system would be the most cost effective based on the cost of collectors and storage. As in example 2, the number and cost of collectors goes up with decreasing cycle efficiency and the cost of sensible heat storage goes up with decreasing ΔT. The system parameters for this example were chosen to be compatible with those being studied for several applications including the central power plant size cited. The results of the analyses are summarized in Table I. T-S and T-H plots for the ideal cycles are shown in Figures 9 through 16.

TABLE I

Summary of Cycle Impact on Conversion Efficiency and Sensible Heat Source ΔT

Cycle Number	Working Fluid	Cycle Temperatures High/Low in °F	Cycle Efficiency[1] Ideal/Non-Ideal in %	Sensible Heating Source ΔT (Ideal) in °F	Comments
1	Water	590/90	35/29	185	Superheat cycle with two reheats. Maximum pressure is 600 psia.
2	Toluene	600/100	37/30	190	Superheat cycle with regeneration. Maximum pressure is 200 psia.
3	Fluorinol-50[2]	600/100	23/18	290	Superheat cycle without regeneration. Maximum pressure is 900 psia.
4	Trifluoroethanol	600/100	33/28	340	Supercritical cycle with regeneration. Maximum pressure is 800 psia.

Notes: 1. Non-ideal cycle calculations assume the following component efficiencies which might be typical of a 300 MW_e plant:

Generator Efficiency - 0.98
Turbine & Feed Pump Efficiencies - 0.85
Regenerator (where used) Efficiency - 0.90
Pressure Drop Losses and Parasitic Power Losses Not Considered

2. Registered trademark of the Halocarbon Products Corporation

Cycles 1 and 2 in Table I indicate that superheat cycles using water and toluene have roughly equivalent efficiencies and result in approximately the same temperature drop in the heating medium at a given efficiency level. Increasing the amount of superheat in both cycles decreases cycle efficiency and increases the heat source temperature drop and conversely for a decrease in the level of superheat.

Cycle 3 was an attempt to find a suitable fluid and cycle which would not require either a reheater or a regenerator (to minimize costs) and yet be supercritical or nearly so to get a large ΔT in the heating source. Fluorinal-50* is a 50 mole percent trifluoroethanol - 50 mole percent water

*Registered trademark of the Halocarbon Products Corp.

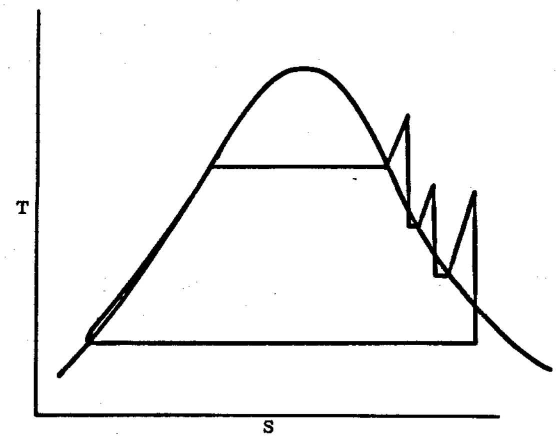

Figure 9. Temperature vs. Entropy for Cycle 1 of Example 3

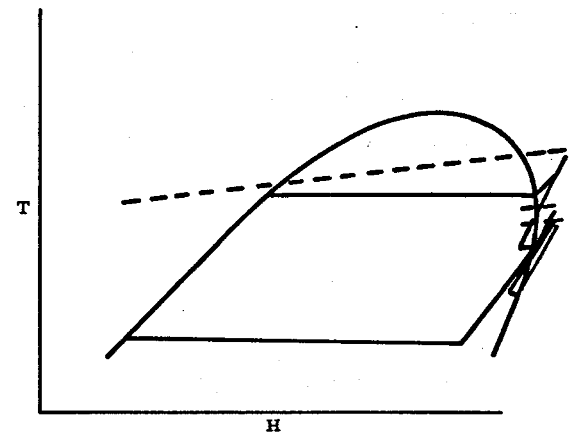

Figure 10. Temperature vs. Enthalpy for Cycle 1 of Example 3. Dotted Lines Are for the Heating (Source) Fluid

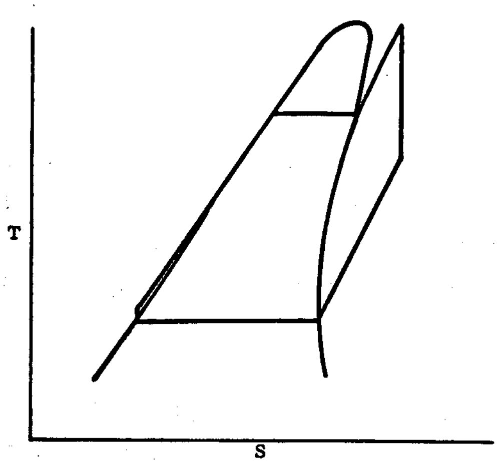

Figure 11. Temperature vs. Entropy for Cycle 2 of Example 3

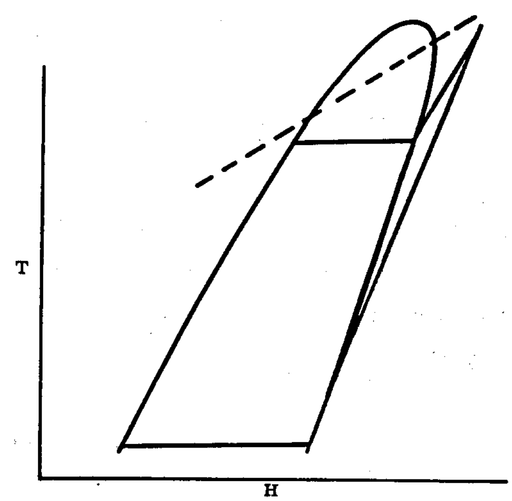

Figure 12. Temperature vs. Enthalpy for Cycle 2 of Example 3. Dotted Line is for the Heating (Source) Fluid

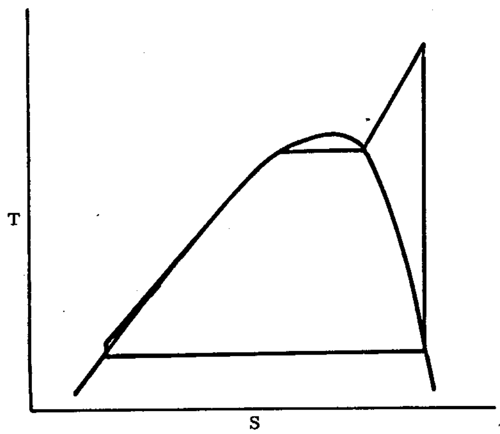

Figure 13. Temperature vs. Entropy for Cycle 3 of Example 3.

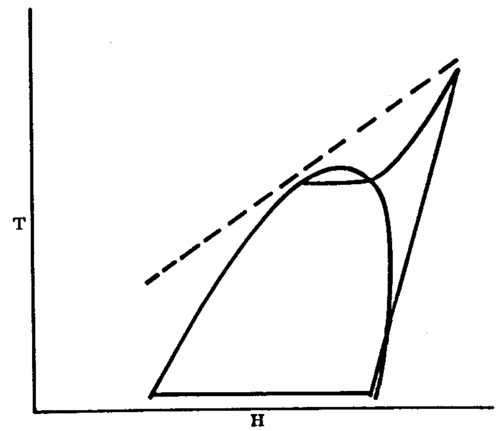

Figure 14. Temperature vs. Enthalpy for Cycle 3 of Example 3. Dotted Line is for the Heating (Source) Fluid.

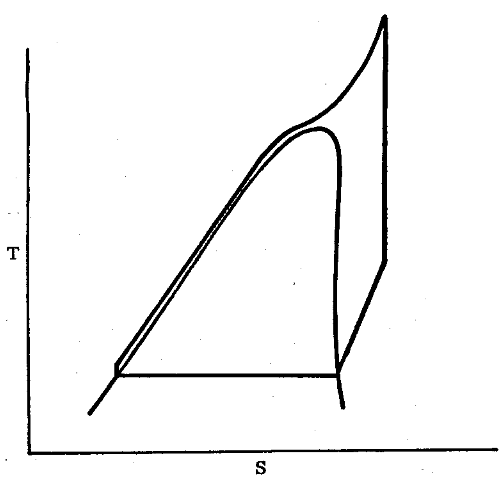

Figure 15. Temperature vs. Entropy for Cycle 4 of Example 3.

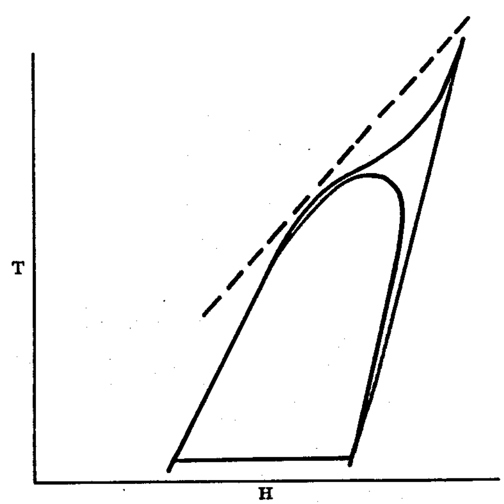

Figure 16. Temperature vs. Enthalpy for Cycle 4 of Example 3. Dotted Line is for the Heating (Source) Fluid.

(85 - 15 percent by weight) mixture which yields a saturated vapor after an isentropic prime mover expansion under the conditions shown. Although the fluid gave rather poor efficiencies compared to the other fluids, there may be other fluids of a similar nature which would yield more promising results.

Cycle 4 utilizes 100 percent trifluoroethanol with a critical temperature of 440°F. As can be seen in the table, this cycle yields a very attractive combination of good cycle efficiency and a large temperature drop. Trifluoroethanol has several other good points such as having a moderate critical pressure (715 psia), a low freezing point (-49°F), and being nonflammable. Some of its disadvantages are a temperature stability limit of 625°F and a relatively high cost of approximately $5/lb.

If we look at the energy extracted from any sensible heating medium, we find that the energy is proportional to the ΔT experienced by the heating medium as noted previously. For this example, the maximum available ΔT from the sensible heat source is 500°F, the same as the spread between the high and low cycle temperatures, which is reasonable as a basis of comparison since in the ideal case we could heat the Rankine cycle working fluid to the highest heating fluid source temperature and cool the heating fluid to the minimum working fluid temperature. On this basis, we note that Cycle 4 has an extraction efficiency of $0.28 \times \frac{340}{500} = 0.19$ compared with $0.29 \times \frac{185}{500} = 0.11$ for Cycle 1. Thus, Cycle 4 could extract nearly twice the energy from a given sensible heat source, which might be a very important factor as in certain geothermal applications where maximum power from a limited source is required or in a solar system requiring a large thermal storage.

Example 4 -- Low Temperature (190°F) Solar Energy Conversion System.

For this study, we have the same major subsystems (i.e., collectors, sensible heat storage, Rankine engine) as in examples 2 and 3 except that the collectors are of the flat plate type operating at a peak temperature of 190°F and the shaft power is to be used for irrigation pumping.

For both cycles, the combined irrigation pump and gearbox efficiency was assumed to be 0.7, the prime mover efficiency was assumed to be 0.7, the regenerator efficiency was assumed to be 0.8, and the boiler feed pump efficiency was also assumed to be 0.8. The peak cycle temperature was assumed to be 180°F, and the condenser hotwell temperature was assumed to be 60°F in both cases. The peak collector efficiencies were taken from a manufacturer's data, using 90°F for the average ambient temperature and 200 Btu/hr ft^2 for the average insolation. The results of the calculations are given in Table II.

From Table II, it can be seen that Cycles 1 and 2 have the same overall "system" efficiency; however, Cycle 2 would require a smaller heat storage volume and less collector pump work by a factor of 2/7. Thus, on the basis of collector and storage costs, (usually the highest cost items) Cycle 2 would provide the most cost effective system. However, a detailed cost analysis including the heat engine costs would be necessary for final determination of the most economical system.

TABLE II

Comparison of Performance Parameters for Two Low Temperature Solar Energy Conversion Systems

Cycle Number	Working Fluid	Power Cycle Efficiency Ep	Sensible Heat Transfer Fluid ΔT	Average Collector Temperature	Collector Efficiency Ec	System Efficiency Ep x Ec
1	R11	0.068	$20^{\circ}F$	$180^{\circ}F$	0.48	0.031
2	R13B1	0.054	$70^{\circ}F$	$155^{\circ}F$	0.57	0.031

Discussion and Conclusions

The common element in all the above examples is that the most cost effective system design did not require the most efficient thermodynamic cycle. However, this does not imply that the heat engine conversion hardware such as the turbine and generator (or irrigation pump in example 4) should not be as efficient as possible.

For illustration, the performance curves shown in Figure 17 represent regenerative Rankine cycle calculations which have identical input, with the exception of the assumed prime mover efficiencies. We note that for a solar system with a specified peak cycle temperature (from Figure 17), the system with the higher prime mover efficiency requires fewer collectors, less high temperature thermal storage (if this method of storage is used), and less regenerator and condenser capacity for a given output. The differences are summarized for a peak cycle temperature of $550^{\circ}F$ in Table III. If the system efficiency is specified (i.e., the electricity to usable heat ratio) such as in a cascaded total energy system, we find that the peak cycle temperatures are significantly lower for the system with the high efficiency prime mover. Referring to the curves for Cycles A and C in Figure 17, we see that if the desired electricity to heat ratio (cycle efficiency) is 15 percent, the peak cycle temperature could be reduced by approximately $110^{\circ}F$, and if the ratio is specified to be 20 percent, we find the peak temperature lowered by almost $200^{\circ}F$. These potential reductions in peak temperatures can significantly lower the thermal losses (and hence raise the efficiency) and/or reduce the insulation required for the entire system including collectors, high temperature storage (if used), piping, etc. In addition, the lower peak temperatures yield more versatility in all areas of system design, particularly in the materials and methods of construction required. The net result of utilizing a higher efficiency prime mover can be lower overall system cost and/or higher performance, regardless of the heat conversion cycle used (the example was arbitrarily chosen). In Sandia's studies to date with solar-to-thermal-to-electrical energy conversion systems, the cost of the heat conversion subsystem (which includes the prime mover) by itself does not appear to be a dominant factor in the cost of the total system, but the performance of this subsystem has a profound influence on the total system cost.

In summary, each application requires careful study from a total systems viewpoint to determine the most cost effective system.

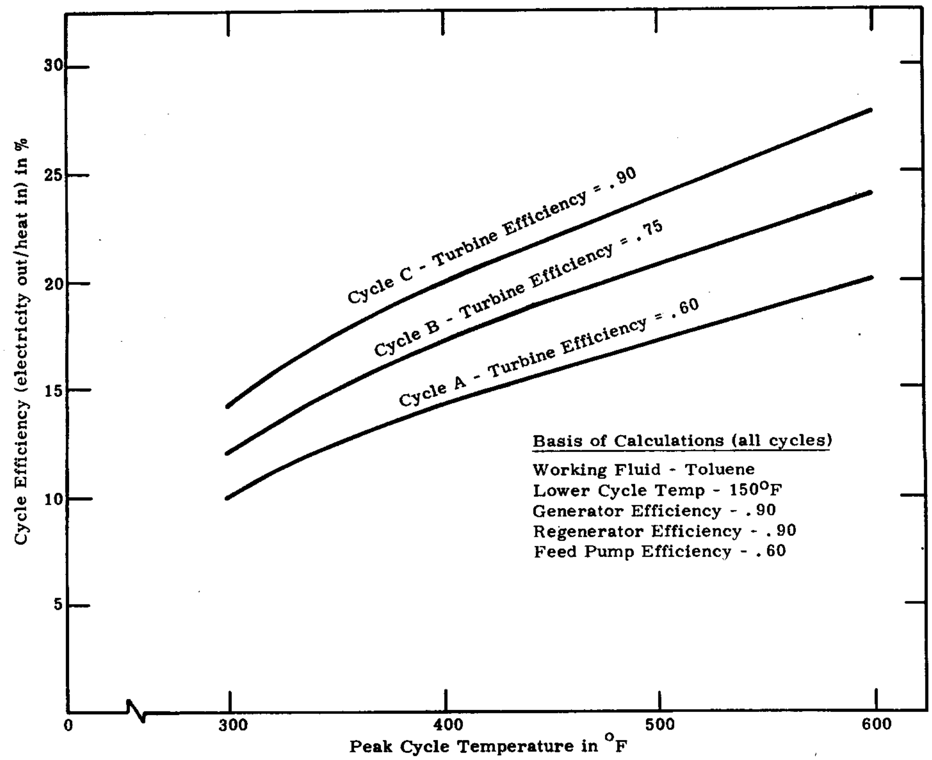

Figure 17. Calculated Cycle Efficiency vs. Peak Cycle Temperature for Cycles Which Are Identical with the Exception of Turbine Efficiency

TABLE III

Comparison of System Parameters for the Rankine Cycle Systems Represented in Figure 17. Peak Cycle Temperature is 550°F in Each Case

Parameter	Cycle A $E_T = .6$	Cycle B $E_T = .75$	Cycle C $E_T = .9$
Input Heat Rate (Btu/Hr/100 kW_e)	1,849,000	1,535,000	1,329,000
Regenerator Heat Rate (Btu/Hr/100 kW_e)	695,000	470,900	324,200
Condenser Heat Rate (Btu/Hr/100 kW_e)	1,468,000	1,154,000	948,200
Mass Flow Rate (lb/Hr/100 kW_e)	8,430	6,670	5,520
Pump Work (hp/100 kW_e)	8.08	6.39	5.29

Reference

1. L. Torkelson and George Treadwell, editors, Solar Total Energy Program Semiannual Report, SAND76-0205, Sandia Laboratories, Albuquerque, NM, March 1976, printed June 1976.

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