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SECOND LAW ANALYSIS OF ADVANCED POWER GENERATION SYSTEMS USING VARIABLE TEMPERATURE HEAT SOURCES

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ABSTRACT

Many systems produce power using variable temperature (sensible) heat sources. The Heat Cycle Research Program is currently investigating the potential improvements to such power cycles utilizing moderate temperature geothermal resources to produce electrical power. It has been shown that mixtures of saturated hydrocarbons (alkanes) or halogenated hydrocarbons operating with a supercritical Rankine cycle gave improved performance over boiling Rankine cycles with the pure working fluids for typical applications.

Recently, in addition to the supercritical Rankine Cycle, other types of cycles have been proposed for binary geothermal service. This paper explores the limits on efficiency of a feasible plant and discusses the methods used in these advanced concept plants to achieve the maximum possible efficiency. The advanced plants considered appear to be approaching the feasible limit of performance so that the designer must weigh all considerations to find the best plant for a given service. These results would apply to power systems in other services as well as to geothermal power plants.

INTRODUCTION

The Heat Cycle Research Program is currently investigating the potential improvements to power cycles utilizing the moderate temperature geothermal resources to produce electrical power. The technology being considered either improves the performance of the power cycle and reduces the cost of electricity, or it provides a means of utilizing a resource which might otherwise not be used because of institutional or technical barriers. Because of the low quality and high cost of the energy, optimized systems for the generation of electrical power should utilize as much of the energy contained in a unit mass of the fluid as possible. This optimization was confirmed by Demuth and Whitbeck (1982) with a "value analyses" study and by Cassel et al. (1981) with the corresponding "market penetration" study which examined the impact of performance improvements on the cost of electricity and on the future utilization of geothermal produced electrical power if these improvements could be realized.

The Heat Cycle Research Program investigations have specifically examined binary power cycles because for the moderate temperature resources of interest, the binary cycles achieve a higher net brine effectiveness than other types of systems. In these investigations of the binary power cycle, advanced concepts such as supercritical heating, integral countercurrent condensation, appropriate choice of working fluid, and metastable turbine expansions were explored. At typical resource conditions, Demuth (1980) and (1981) found that mixtures of saturated hydrocarbons (alkanes) with these advanced concepts gave improved a 29% performance improvement over proposed plants. Bliem (1987) in subsequent studies showed that the same results were true if halocarbon mixtures (Freons) were used.

With the projected improvements in performance from the concepts identified in these analytical studies (along with internal recuperation studied by Demuth and Kochan (1982)), the program has initiated field investigations to further examine the potential performance gains with these concepts. These results have been experimentally verified by Demuth et al. (1985) and Bliem and Mines (1989). These field investigations, conducted at the Heat Cycle Research Facility currently located in the East Mesa of California's Imperial Valley, examined the validity of the predicted performance improvements through the verification of the assumptions used in the predictions, and the adequacy of the "state-of-the-technology" design methods, as well as fluid transport properties.

Recently, a number of new concepts for power cycles for geothermal use have been introduced and published performance data for these systems is available. For this paper, the operating conditions for the supercritical Rankine cycle have been adjusted for direct comparison of the originator's predicted performance. Kalina and Leibowitz (1989) showed the performance of a Kalina System 12 for geothermal use. Saulson and Rosenblatt (1989) considered a new concept which they call the Low-Temperature Engine System for power production from a flashing well. This system employs a heat-driven heat pump and two separate heat engine cycles. One engine's heat input is from the heat rejected by the other engine and this engine rejects heat to the heat pump at a temperature lower than atmospheric which is pumped to atmospheric by the heat pump.

In the United Kingdom, the U. K. Department of Energy (1988) proposed to use the Trilateral Wet Expansion System for a binary application with a hot-dry rock resource. These new concepts rely on similar considerations to those studied in the Heat Cycle Research Facility (HCRF). One of the primary prerequisites for the achievement of the predicted performance of each of these systems which utilize mixtures as working fluids is that phase changes (boiling and condensation) be carried out close to equilibrium with the phases mixed. This has been studied in the HCRF in detail for the condensation process. In addition, the use of supercritical vaporization allows the generation of vapor without the problems associated with boiling a two-component mixture.

For all such cycles, there is a cycle performance limit for given resource and sink temperatures and practical assumptions relative to rotating equipment efficiencies and heat exchanger approach temperature differences or log mean temperature differences (LMTD's). This paper examines this performance limit and reviews the performance of some of the advanced systems.

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CYCLE PERFORMANCE LIMITATION

Background

A Second-Law-of-Thermodynamics analysis is useful in determining the limitations of performance of power generation systems. Briefly, the second law allows for the definition of available energy, defined by Obert (1960):

> "Available energy is that portion of energy which could be converted into work by ideal processes which reduce the system to a dead state--a state in equilibrium with the earth and its atmosphere."

If a system is at a different pressure from the atmosphere, work can be obtained by expansion to atmospheric pressure. If a system is at a different temperature than the surroundings, work may be obtained by transferring heat to a work producing cycle (heat engine). If chemical reactions are possible, for example, if the substance is a hydrocarbon; a reaction which oxidizes the hydrogen to water and the carbon to carbon dioxide or carbonate has the potential to produce work. (In this paper, chemical reaction will not be considered.) Available energy is essentially the same as availability, exergy and essergy.

The available energy, A, of a stream in steady flow is:

$$A = m\,[(h - h_o) - T_o\,(s - s_o)] \qquad (1)$$

where m is the mass flow rate; h, enthalpy ; T, absolute temperature and s, entropy. The subscript o represents the value at the pressure and temperature of the environment, the dead state.

The irreversibility of a process or physical component, I, is the sum of all of the increases and decreases in available energy occurring and can be shown to be equal to:

$$I = - T_o\,\Sigma\, m\, Ds \qquad (2)$$

where T_o is the absolute ambient temperature; m, the mass flow rate and Ds the change in entropy for a given stream. Kalina (1984) has shown that if the cooling water is assumed to be the sink instead of the ambient temperature or ambient wet bulb temperature, an effective ambient temperature can be defined as:

$$T_o = T_e\,(2 - T_e\,[\ln(T_i/T_e)/(T_i-T_e)]) \qquad (3)$$

where T_i and T_e are the inlet and outlet temperatures of the coolant sink in absolute units. For temperatures encountered in normal power production, T_o can be approximated within hundredths of a percent by:

$$T_o = (T_i + T_e)/2 \qquad (4)$$

the mean coolant temperature. Note that, for a process, the outlet available energy is the inlet available energy plus the irreversibility.

For example, for a process transferring heat from a stream at T_h to a stream at T_c in steady flow, the irreversibility will be:

$$dI/dq = (T_o\ DT)/(T_h\ T_c) \qquad (5)$$

where the temperatures are in absolute units and DT is the temperature difference across which the heat is transferred, that is T_h-T_c. An important observation is that if T_h and T_c are significantly greater that T_o, the irreversibility per unit heat transfer at fixed DT is significantly lower than if they are at temperatures near T_o. If T_h and T_c are about 1.5 times T_o (typical of the heat addition process) the irreversibility per unit heat transferred at a fixed T_o and DT will be 44% that of a heat transfer process in which T_h and T_c are approximately equal to T_o such as the heat rejection process. This would imply that closer approaches are advantageous (increase efficiency more) in the heat rejection process than in the heat addition process.

Kalina (1984) shows this relationship by plotting an "exergetic temperature", T_{ex}, instead of actual temperature on heat duty plots for heat exchange, where:

$$T_{ex} = 1 - (T_o/T) \qquad (6)$$

Then,

$$dI/dq = D\ T_{ex} \qquad (7)$$

where D T_{ex} is the difference in the hot and cold exergetic temperatures. For a more complete discussion see a basic engineering thermodynamics text such as Obert (1960).

Methodology

A realistic maximum can be placed on the work produced for a given resource using the ideas discussed in the previous section. If one assumes logical values for heat exchanger pinch-point temperature differences or log-mean temperature differences (LMTD's) and for rotating machinery isentropic efficiencies, the irreversibilities associated with the heat transfer and work processes in a cycle with the optimum match between working fluid and heat source and heat sink can be calculated. The work produced by a given cycle can be determined by subtracting from the available energy in the geofluid source, the irreversibility associated with each of the cycle devices: the heater, the turbine, the condenser (heat rejection) and the pump.

This analysis attempts to be generic in its application. The efficiency produced by this analysis is for the plant only. It does not include parasitic power associated with the heat rejection system, or the geothermal supply and injection system. These systems are separate and the impact of each will vary considerably depending on the particular application. The plant is a separate unit and can be considered separately. The choice of the type of heat rejection system and impact of the supply and injection system are left for a site-specific analysis.

Results

Restricting the analysis to sensible liquid heat sources, a working fluid is postulated which will give close to the minimum irreversibility in each component with a realistic temperature difference in heat exchangers and realistic isentropic efficiency for turbines and pumps. Figure 1 shows the heat addition and heat rejection curves for the ideal working fluid, "unobtainium". For a given pinch point temperature difference, the minimum heat exchanger irreversibility occurs with a constant temperature difference. With a fixed log temperature difference (LMTD) for the exchanger, the minimum irreversibility occurs with a slightly smaller temperature difference on the end of the heat exchanger nearer to T_o. For heater temperatures in the range of this study, for an LMTD of 10 °F, a temperature difference on the cold side of the exchanger of 8 to 9°F (on the hot side, 11 to 12.1°F) gives slightly lower heater irreversibility than a uniform 10°F difference. That increase is between 0.6 and 1.2 % of the heater irreversibility, and is thought to be not worth considering in order to simplify the problem to one of a uniform temperature difference. All enthalpies are referenced to the geofluid mass. Note that the difference in the heat added in the heater and that rejected in the condenser is the cycle net work per unit mass of geofluid.

It is further assumed that an "unobtainium" turbine will have an efficiency of 85% and that the pumping work will be small compared to the turbine work. With the approximation that the second law turbine efficiency is approximately equal to the isentropic efficiency, the net work is approximated by applying the turbine efficiency to the available energy in the geofluid minus the irreversibilities in the heat transfer processes.

Figure 2 shows the performance results for this system for different maximum resource temperatures. Each loss or irreversibility is expressed as a fraction of the available energy of the geofluid. (In this case with no restriction on geofluid outlet temperature, it was assumed that the geofluid could be reduced to ambient temperature.) These results are for a uniform temperature difference in both heat exchangers of 10 °F. That is, the pinch point and the log mean temperature difference are each 10 °F. At lower resource temperatures the fraction of the available loss due to heat exchange increases, indicating that smaller pinch points would be justified for lower temperature resources. Note that the condenser irreversibility is larger than the heater irreversibility because the heat exchange fluid temperatures are closer to T_o as explained in a previous section. This would indicate the effectiveness of lower pinch points in the condenser than in the heater. Bliem (1989a) has explored this point from a cost-effectiveness point for the supercritical Rankine cycle. When the irreversibilities are normalized with respect to the heat source decrease in available energy and are subtracted from unity, the second law efficiency (net plant work divided by the available the efficiency varies from about 65 to 75% depending on energy in the geofluid) remains. For this system, the resource temperature for this LMTD and rotating machinery efficiency.

In many cases, the minimum temperature to which the geofluid can be cooled is limited by concerns over deposition of silica. For this case, the minimum geofluid outlet temperature increases as the resource temperature increases. Figures 3 and 4 depict this case with a single heater and single condenser. Here the temperature difference in the heater is not uniform. Therefore, the temperature difference on the hot end of the heater is changed to obtain LMTD's of 10 °F. The net plant second law efficiency with this restriction was between 55 and 60%. The heater irreversibility is increased over the case with no limit. The breaks in the

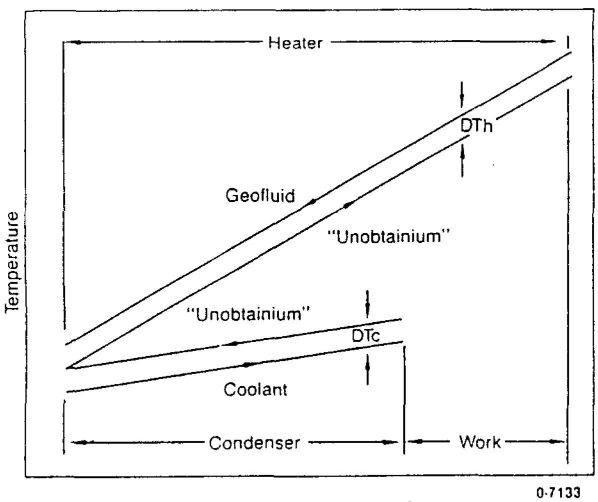

Fig. 1 "Unobtainium" Cycle with Unconstrained Outlet Temperature.

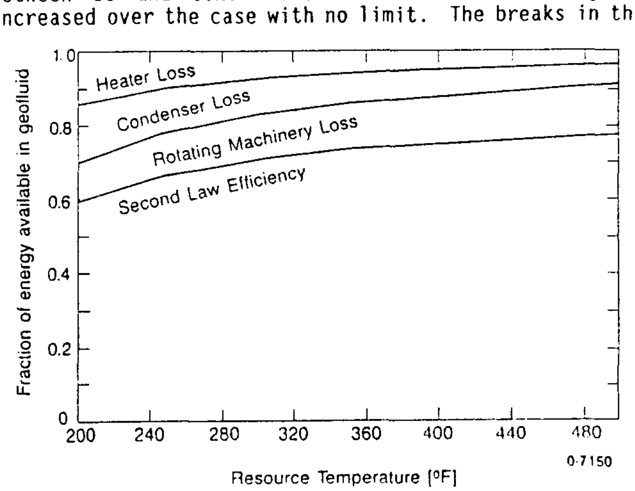

Fig. 2 Efficiency of "Unobtainium" Cycle with Unconstrained Outlet Temperature.

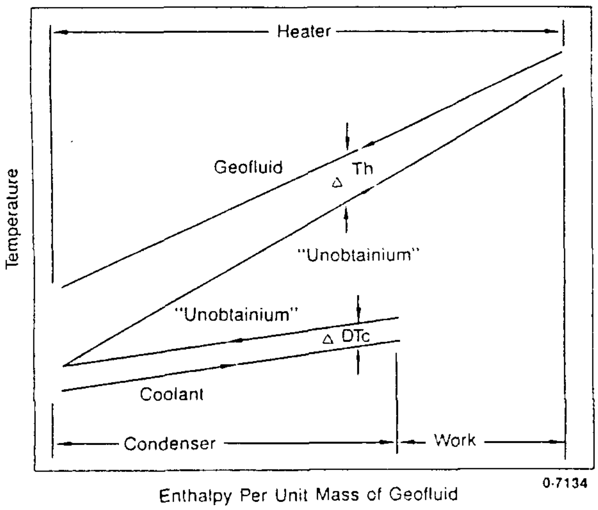

Fig. 3 "Unobtainium" Cycle with Constrained Outlet Temperature.

curves in Figure 4 result from the fact that at low temperatures there is no limit and the limit temperature increases as the resource temperature increases. (At resource temperatures lower that 280 °F, the results are the same as in Figure 2.)

In the case of a restricted minimum geofluid temperature, adding internal recuperation allows the heater irreversibility to be decreased substantially (although there is an added irreversibility associated with the recuperative heat transfer.) These results have been shown by Demuth and Kochan (1982) for a Rankine cycle using turbine exhaust to recuperatively heat the working fluid. Figure 5 shows a schematic diagram of such a system. Another recuperative scheme used in utility steam plants is feedwater heating with steam bled from intermediate turbine stages. Demuth and Kochan (1982) considered this method of recuperation but found in general results were no better than with the turbine exhaust recuperation which is somewhat less complex.

Figure 6 shows the temperature enthalpy (heat duty) diagram for an "unobtainium" cycle with internal recuperation. The hot "unobtainium" in the recuperator would come from the expansion process say as hot turbine exhaust or a small turbine bleed stream. This heat is

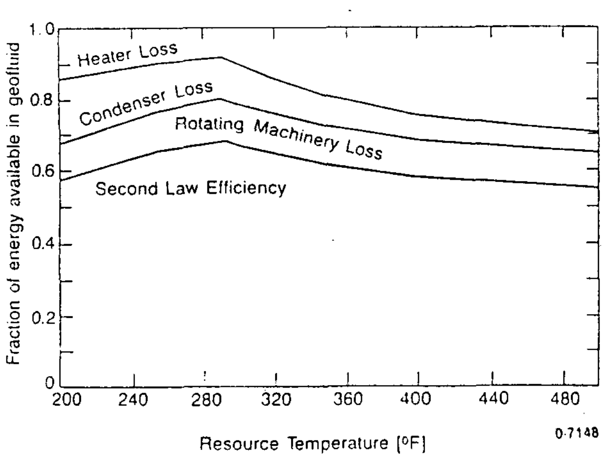

Fig. 4 Efficiency of "Unobtainium" Cycle with Constrained Outlet Temperature.

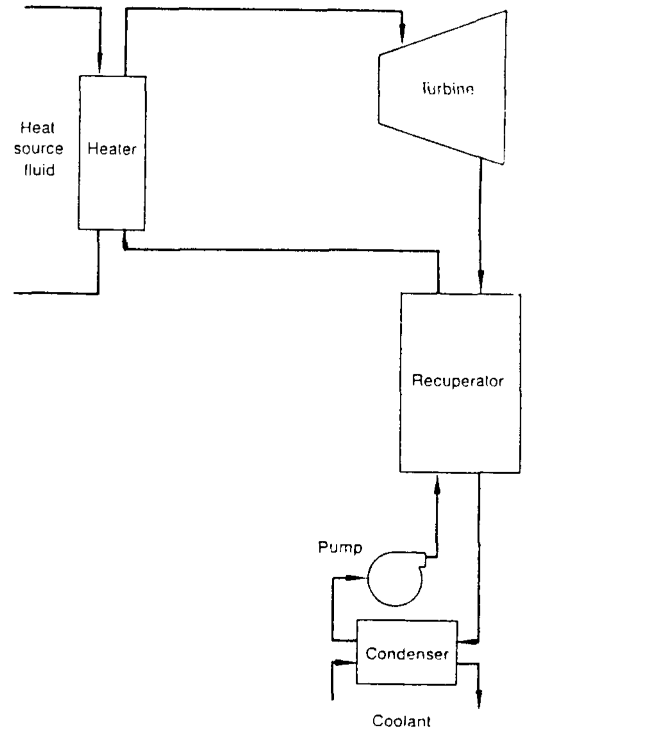

Fig. 5 Recuperated Rankine Cycle.

used to perform enough preheating of the working fluid to give an ideal match of heating and cooling curves in the heater. The match in the recuperator may not be as good. For this analysis, the temperature difference at the hot end of the recuperator is adjusted to maintain a 10 °F LMTD.

Figure 7 shows the performance results for the recuperated cycle with a geofluid outlet temperature limit. Here, again the LMTD for each exchanger is 10 °F and the rotating machinery isentropic efficiency is 85%. Note that there is an added loss for this system, the recuperator irreversibility. However, Demuth and Kochan (1982) showed that the heater and condenser irreversibilities are decreased giving approximately the same second law plant efficiency as for the case with no outlet temperature restriction. The recuperator takes

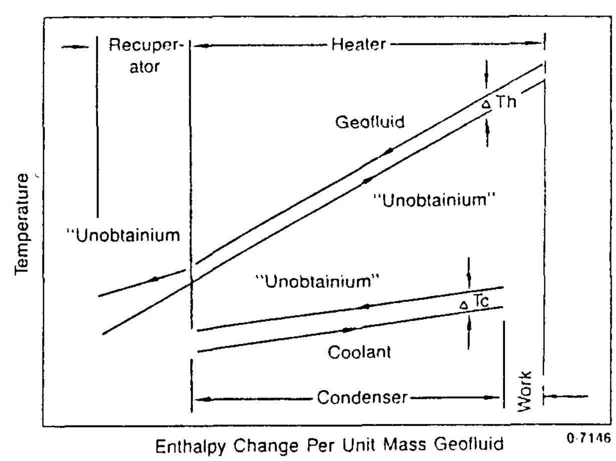

Fig. 6 Recuperated "Unobtainium" Cycle with Constrained Outlet Temperature.

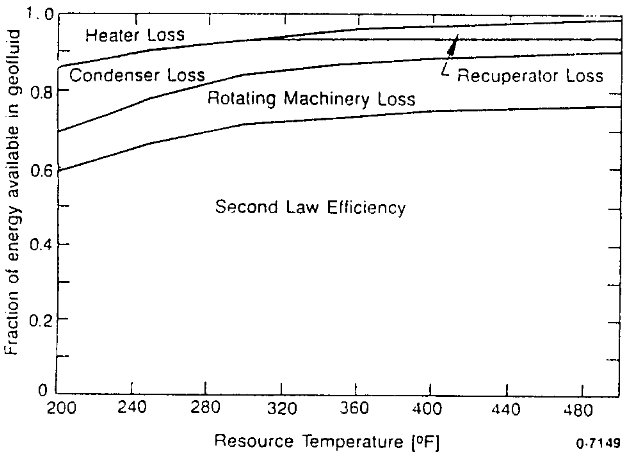

Fig. 7 Efficiency of Recuperated "Unobtainium" Cycle with Constrained Outlet Temperature.

some of the heating duty from the heater and some of the heat rejection duty from the condenser. (Compare the heater and condenser irreversibilities in Figure 2 with the heater, condenser and recuperator irreversibility in Figure 7.) Generally, the second law efficiency varies from 65 to 75% depending on the resource temperature.

ADVANCED SYSTEM PERFORMANCE

How close do the latest advanced cycles approach the maximum plant efficiency defined in the previous section? First, the systems with limited geofluid outlet temperature will be considered. The reason for this is that most of the systems which have been optimized for operation under this constraint. Then, the Trilateral cycle and the Heat Cycle concept of turbine expansion "through-the-dome" (See Demuth (1983).) will be considered for cases in which there is no constraint on outlet temperature.

Constrained outlet temperature

In many geothermal applications, the outlet temperature of the source fluid is limited to some minimum value by silica precipitation considerations. Three actually proposed systems subject to such a constraint are considered to illustrate the methods which use real fluids to approach the behavior of "unobtainium". A supercritical Rankine cycle with exhaust gas recuperation and a Kalina System 12 are discussed in some detail and the Polythermal Technologies Low Temperature Energy System is also discussed, but not to the same degree because there is less in the literature concerning specific state-point data.State points for the supercritical Rankine cycle and the Kalina System 12 are given in Reference 13 where the two systems are compared under the same assumptions. Kalina and Leibowitz (1989) give a more detail on individual processes in the Kalina System and the supercritical Rankine cycle illustrated here is discussed in earlier work from the Heat Cycle Research program (See Demuth (1981) and Demuth and Kochan (1982)).

The path pursued in the Heat Cycle program has been to use a Rankine cycle as depicted in Figure 5 and approximating the near linear heating curve of working fluid by operating at pressures above the critical point. The use of a mixture allows for critical pressures and temperatures to be matched with the given resource maximum temperature and remain at moderate pressures (for example below the rating for 600 psi flanges).

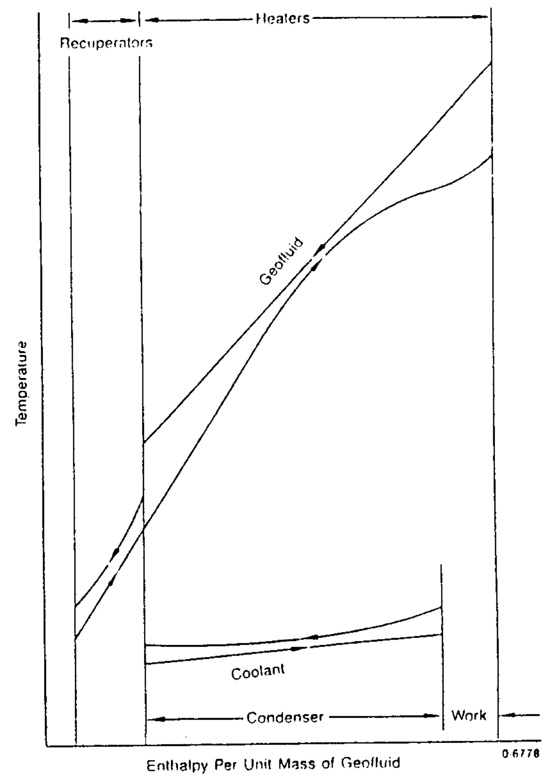

Fig. 8 Temperature-Enthalpy Diagram for Supercritical Rankine Cycle.

Figure 8 show a temperature-enthalpy plot for the heat exchange processes of the cycle. This representation is used rather than the exergetic temperature-enthalpy because it is more familiar to designers. If the exergetic temperature had been used, the area between curves would be directly proportional to the heat transfer irreversibility. In this diagram, as discussed in the background section, a temperature difference at the hot end of the heater represents approximately half the irreversibility of the same temperature difference in the condenser because of their relation to T_o. Note that the largest temperature differences are near the hot and cold ends of the heater. The enthalpy change of the condenser is lined up with the heater because the difference in heat transferred is the net work for the cycle. The temperature difference in the condenser is practically constant. This is a result of using a mixture for a working fluid and achieving integral condensation in countercurrent flow. The recuperator, which heat the working fluid by using turbine exhaust is relatively small and the temperature difference is nearly constant. The primary inefficiency is the large temperature difference near the cold end of the heater. Another advanced system operating under the same constraints is the Kalina System 12. Figure 9 shows a schematic diagram of this system. This system was unveiled in January 1989 and Exergy, Inc. has, since that time, developed a newer system which with higher performance and more complexity which might be desirable for a large installation. The Kalina system uses a mixture of water and ammonia for a working fluid. Because the working fluid becomes wet as it expands, a reheat stage and second turbine are required. Figure 10 shows a temperature-enthalpy diagram for this system. Note that because the superheater and reheater heat the working fluid through a similar temperature range,

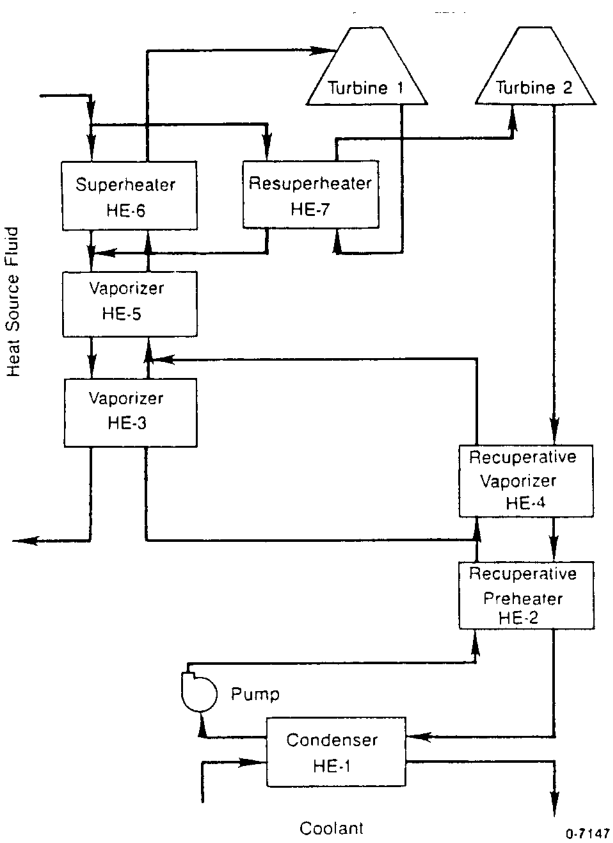

Fig. 9 Kalina System 12.

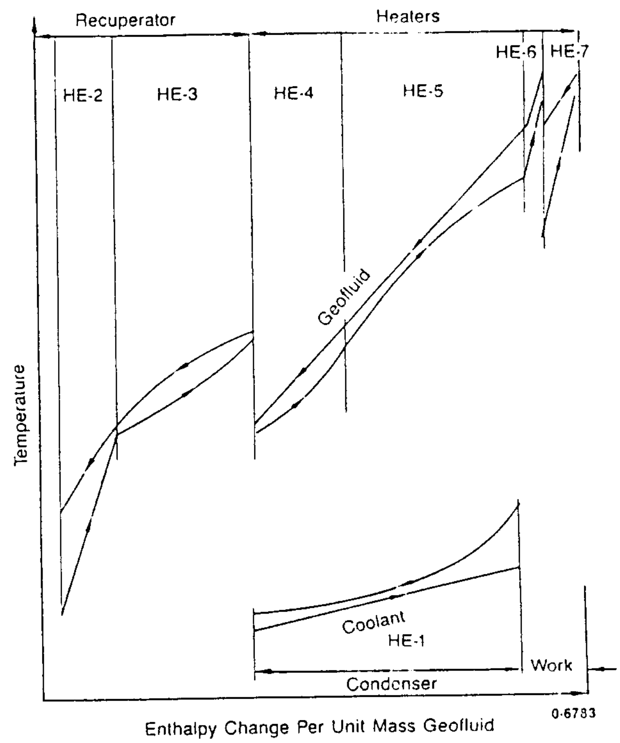

Fig. 10 Temperature-Enthalpy Diagram for Kalina System 12.

irreversibilities are minimized by splitting the geofluid flow between these units. (This is the rightmost part of the figure. Note that the individual heat exchangers are identified on both figures.) The initial vaporization of the working fluid is split between the geofluid in HE-3 and turbine exhaust in HE-5. (The discontinuity in slope of the heating curve of the working fluid is a result of the flow split.) This is necessary to maintain small temperature differences in HE-5 while avoiding a temperature pinch in HE-3. Again the condenser is lined up with the external heater train and the net work is shown as the difference between the heat transfer in and out.

Comparing Figures 8 and 10 show that there is little difference in heat exchange irreversibility between the two systems. Temperature differences in the heating processes are similar for the two systems. In the System 12, more of the heating duty is done recuperatively, to avoid temperature pinch at the cold end of the heater. It is interesting to note that per unit of work produced approximately 25% more heat is transferred in the System 12 than in the supercritical Rankine system. This does not automatically indicate a larger heat exchanger area because the heat transfer coefficients may be different enough to offset the increased heat load for recuperation.

The other system which was designed for a heat source with a restricted outlet temperature, described in Saulson and Rosenblatt (1989), is the Low Temperature Engine System (LTES). This system consists of three separate subsystems: receiving primary heat from the heat source is a heat-driven heat pump (heat amplifier) which produces a sink below ambient temperature, a

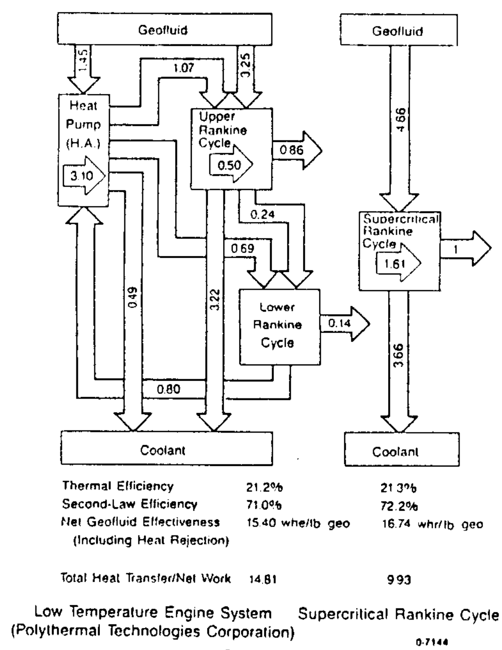

Fig.11 Low Temperature Energy System and Supercritical Rankine Cycle.

Rankine cycle heat engine which also receives heat from the heat source as well as from the heat pump and rejects heat to ambient and to a second heat engine, and a second Rankine cycle which receives heat around ambient temperature and rejects heat to the heat pump below ambient. The heat pump proposed is an ammonia-water absorption system. Figure 11 shows this system schematically along with a supercritical Rankine cycle for the same service. The arrows within the boxes represent internal recuperation within the individual cycles. The numbers represent energy flows for a net work output of one energy unit. Notice again, that there is a large amount of recuperative heat transfer for the LTES compared with the Rankine cycle.

Unconstrained Outlet Temperature

Some applications have no constraint on the heat source outlet temperature. The Trilateral cycle is proposed for a hot dry rock power plant by the United Kingdom, Department of Energy (1988). Figure 12 shows the cycle on a temperature-entropy diagram. The cycle consists of heating a liquid to its boiling point (somewhat below its critical point) and then expanding it

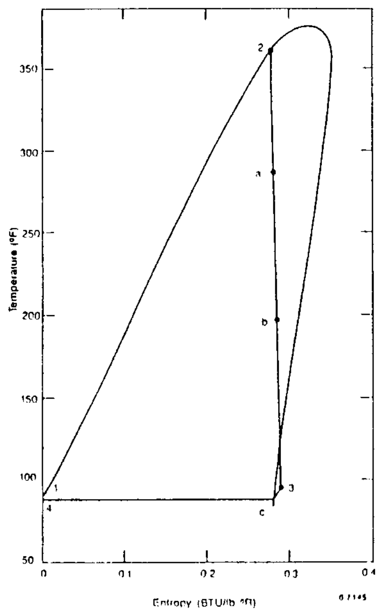

Fig. 12 The Trilateral Cycle.

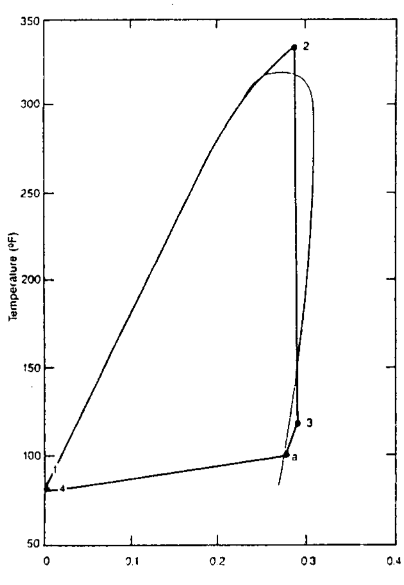

Fig. 13 The Supercritical Rankine with Expansion "Through-the-Vapor-Dome".

through the two-phase region to vapor at the condenser pressure. This is possible with fluids with retrograde saturated vapor lines (that is, saturated vapor lines in which the entropy decreases as the temperature decreases) such as butane and the heavier hydrocarbons.

An alternative scheme was proposed by Demuth (1983) which involves the use of a standard supercritical Rankine cycle in which the fluid is heated past the critical temperature and then expanded through the two-phase region to essentially saturated vapor at the turbine outlet state. Figure 13 shows this cycle which also relies on a working fluid with a retrograde saturated vapor line. Demuth concluded that the vapor expanded through the turbine would stay in a metastable state and that condensation would not occur. This fact is being investigated with nozzle tests at the Heat Cycle Research Facility.

Figure 14 shows the temperature enthalpy diagram for the two cycles. Note that the Trilateral cycle has lower irreversibilities at the higher temperatures while the Supercritical Rankine cycle has lower irreversibilities at the lower temperatures and during heat rejection. The latter also takes more heat from the source fluid. The result is a 3% improvement in performance for the Supercritical Rankine cycle over the Trilateral cycle.

The primary result of this study is that there is a practical limit to the plant performance. In addition, it is shown that many of the advanced systems are approaching this limit. Each system has advantages and disadvantages. The power plant designer/operator must weigh these advantages and disadvantages and in combination with economic and site-specific constraints, and select the power system which best meets the requirements for that application

CONCLUSIONS

Figure 15 summarizes the second law efficiencies for these systems along with estimates from References 7 and 8 for flash steam plants and for a plant with the Heber Binary technology at a slightly higher resource temperature. The theoretical curves are for heater LMTD's between 12 and 16 °F; condenser LMTD's between 8 and 10 °F and recuperator LMTD's 12 to 13 °F. Within these ranges, the results do not change noticeably. This

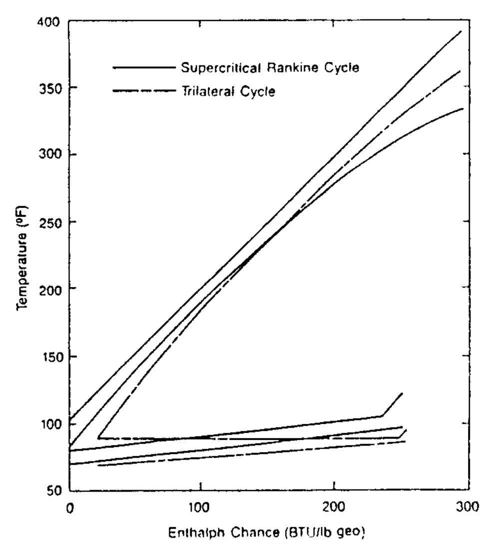

Fig. 14 Comparison between Cycles with No Minimum Temperature Limit.

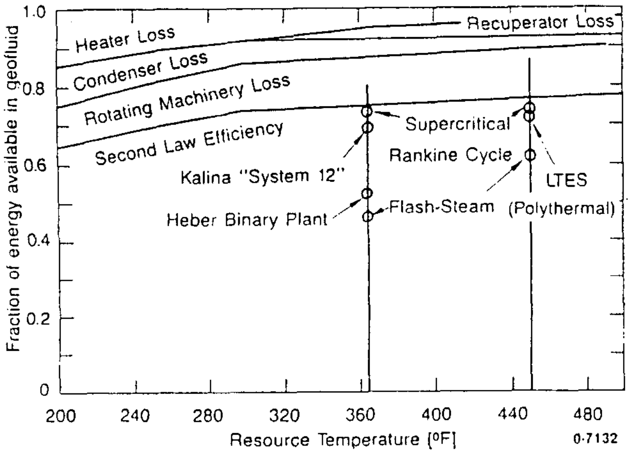

Fig. 15 Results for Advanced Systems.

is the range of the LMTD's for the systems pictured in Figures 8 and 10. These results indicate that all of these advanced technologies have the potential to significantly improve on current technology and each has the potential to approach the reasonable expectation of maximum second law plant efficiency.

Summarizing the results of this paper:

With LMTD's around 10°F (close to a practical limit) and turbine efficiencies of 85%, net plant second law efficiencies of 65 to 75% are possible.

Increasing these efficiencies are expected to be difficult without large increases in heat exchanger size or improved turbine efficiencies.

Small approach temperature differences and LMTD's in the heat rejection heat exchange process reduce irreversibilities more than small approach temperature differences in the heating process. Similarly, in the heating process, smaller temperature differences on the cold end of the unit will reduce irreversibilities more than small temperature differences on the hot end.

The advanced system considered here appear to be approaching this limit and investigation for a particular application may favor one system over another. The design engineer must decide on the most cost-effective alternative.

This study does not include effects of heat source delivery systems or heat rejection systems. The geofluid supply and injection systems should be similar for all binary alternatives, however the heat rejection systems may be different because of the optimum cooling water temperature rise (or air temperature rise for dry cooling). These effects must be addressed separately.

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