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A COMPARISON OF SUBCRITICAL AND SUPERCRITICAL RANKINE CYCLES FOR APPLICATION TO THE GEOPRESSURED GEOTHERMAL RESOURCE

BY

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Abstract

There are several features unique to the geopressure geothermal resource which narrow the range of power cycle alternatives. The purpose of this paper is to describe the thermodynamic and operating restrictions which appear to favor the application of a supercritical Rankine power cycle utilizing propane for the recovery of thermal energy from the geopressure geothermal resource. This power cycle can be integrated into a natural gas recovery scheme that conserves reservoir pressure for brine disposal and produces gas at pipeline pressure.

Introduction

Several factors emerge with regard to the geopressure geothermal resource along the Texas and Louisiana Gulf Coasts that bound the parametric search for a thermal power cycle with hydraulic energy utilization, and a gas recovery scheme. The brine temperature in the geopressure will in general be higher than 395°K (250°F) due to its depth. The temperature will increase predictably with depth in the Wilcox, Frio and Vicksburg sands. In the Tuscaloosa trend, wells over the interval from 18,000 to 22,000 feet will provide brine temperatures on the order of 505°K (450°F). The dominant temperature range to be expected in the Frio and Wilcox sands of interest in the principal geopressure geothermal fairways will fall between 411°K (280°F) and 444°K (340°F). Due to the heating of upper rock formations, the bottomhole fluid temperature will be asymptotically approached with time at the surface as the well is produced. At the 20,000 barrels per day production rate expected, the brine surface flowing temperature will approach bottomhole values within 10°K in a few days. Any acceptable power plant design must operate throughout the expected temperature range if construction standardization is to be achieved. Natural gas recovery will be discussed only insofar as it relates to power plant design considerations.

Thermal Energy Recovery

Rapid cooling of the brine is desirable as a means to inhibit scale formation and corrosion activity. As CO^2 is an integral component of the solution gas, cooling prior to final pressure reduction will retain some CO^2 in solution while slightly reducing the solubility of methane. The dissolution of CO^2 and subsequent formation of carbonate scale is the primary heat exchanger fouling mechanism. Any CO^2 retained in the brine will reduce the load on the amine plant and may provide the margin in a pipeline gas specification to avoid a requirement for sweetening. Brine cooling prior to gas separation reduces the load on the gas dehydration plant and the number of components required for the system. Should brine cooling become an economic necessity, the incremental cost of incorporating a power conversion scheme may make power production competitive. The brine cooling approach temperature is an important factor to consider in cooling tower/ambient heat exchanger selection. Once the brine has been cooled to 340°K (150°F), there appears to be little reason for further cooling for the sake of disposal compatibility. By cooling the brine to 300°K (80°F), a carbonate precipitation problem could occur as the brine is reheated by the natural geothermal gradient in the disposal formation. If silica is present cooling below 340°K (150°F) can facilitate precipitation.

Hydraulic Energy Recovery

Pressure maintenance during brine cooling is necessary to provide sufficient working pressure to control flow through the separator and to enter the disposal well. Hydraulic energy utilization may best begin with conservation of pressure to minimize both gas compression and brine pumping horsepower. The hydraulic energy is the first energy component to be depleted at a constant well flow rate. Pressure Recovery equipment must operate in an isolated manner from the remainder of the production equipment to facilitate its early decommissioning at one site and application at another.

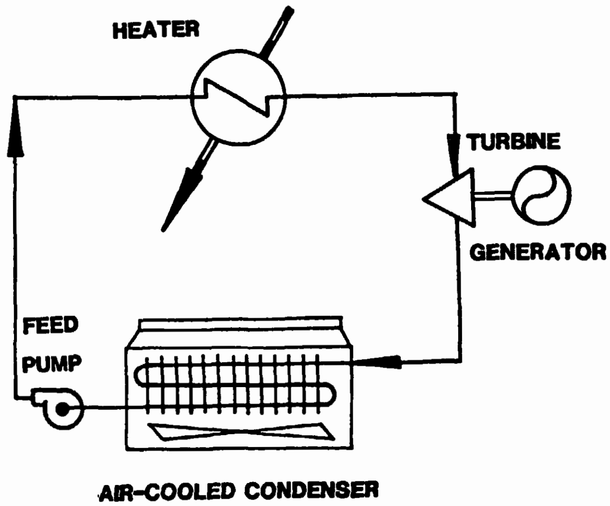
Anticipated wellhead pressures will range from 200 to 600 bar. Many pipelines in the Gulf Coast area operate at/or below 80 bar. There are practical limitations to working pressures that can be economically handled within standard industry practice. As a general rule, an ASME Steam Flange Rating of 40 bar (600 psi) will allow safe working pressures up to 85 bar (1,250 psi) at 422°K (300°F). This rating is readily available in standard process design equipment. A first stage of cooling and separation should be effected above this value to eliminate requirements for gas compression. A subsequent lower pressure stage of gas flashing will require a booster compressor to reinject the gas into the raw gas stream from the first stage separator. The low pressure gas would be the preferred source for plant fuel.

Thermal Power Generation

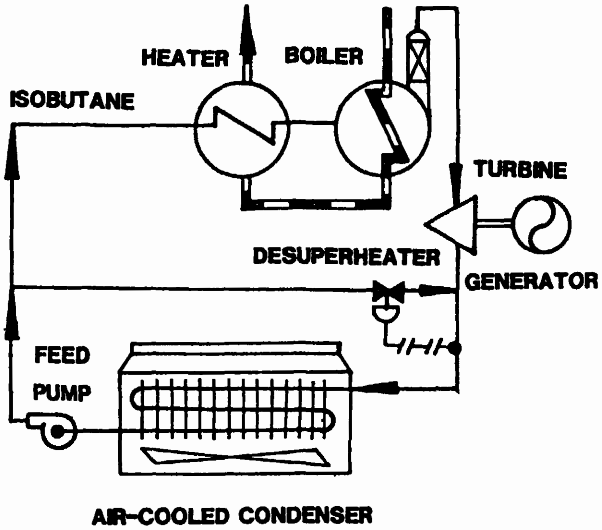
Production well spacings of three kilometers and single disposal well operations for each favor small air-cooled portable power plants over the central station concept. Selection of a binary cycle is tied to oil and gas industry familiarity with gaoline and liquified petroleum gas production plants. These are operated as semiattended units of the same physical size contemplated for a single well geopressure geothermal power plant. The energy produced will be collected in an electric power grid eliminating the need for a hot brine gathering system and a brine disposal system.

Much of the current technical literature cites isobutane as the working fluid of choice in a Rankine cycle because of its characteristic behavior of expanding isentropically from a saturated vapor state into superheat and low pumping requirements. The larger the pumping requirements for a power cycle, the greater the conversion losses of mechanical work within the cycle. This has been the traditional argument against supercritical Rankine power cycles. If, however, the thermodynamics of a supercritical system are sufficiently superior, these difficulties can be overcome. Theoretically, work applied to a power cycle is recoverable less the energy dissipated by inefficiencies in work conversion devices. There are three power cycles that may reasonably be considered for binary power plant operation. Schematic representations for each of these follows so that a component-bycomponent comparison for each power plant may be made.

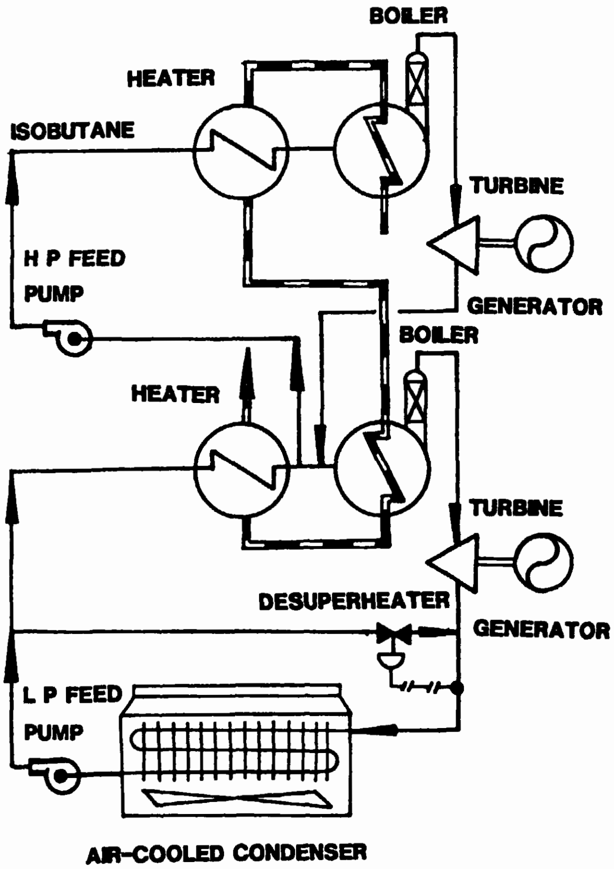
SUPERCRITICAL CYCLE



SUBCRITICAL CYCLE



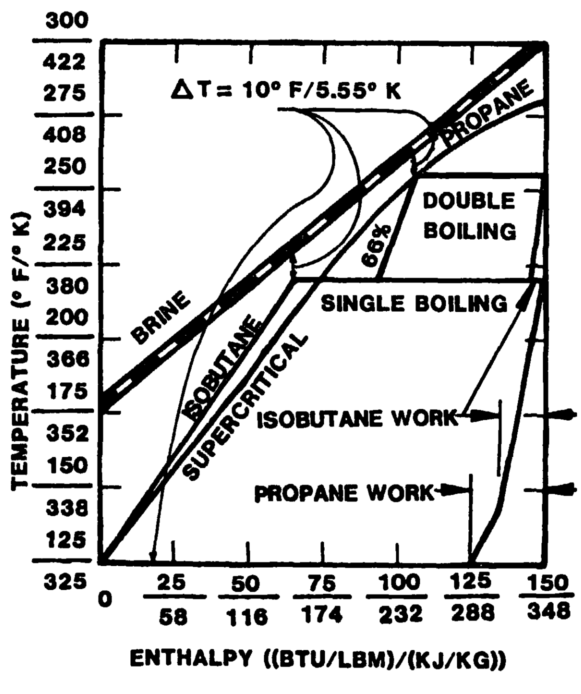
DOUBLE BOILING CYCLE



An example was selected to compare the supercritical propane and subcritical isobutane cycles on unit mass-for-unit mass basis. A common heat source was selected in the form of a brine stream cooling uniformly from 300°F to 175°F (422°K to 353°K). A closest temperature approach of 10°F (5.55°K) for both cycles was chosen. For convenience, the enthalpy change for each cycle was fixed at 150 BTU/lbm (348.9 KJ/KG) for each material and the working pressure was allowed to vary. A common condenser temperature of 125°F (325°K) was selected. The working pressure for propane was set at 1,000 psia (68 bar) while that for isobutane was computed to be 312.6 psia (21.3 bar). This unique set of conditions allows for an objective comparison of thermodynamic performance to a common heat source for the selected cycles and working fluids. Two versions of the subcritical Rankine cycle for isobutane were considered. A simple single boiling pressure cycle is used as the base case while the more efficient double boiling cycle is shown as well. The mass flow rate in the high pressure cycle is 66% that of the total flow through the condenser to maintain the 10°F (5.55°K) temperature approach in the heater.

COMMON UNIT MASS-THERMODYNAMIC COMPARISON

FOR ISOBUTANE/SUBCRITICAL AND PROPANE/SUPERCRITICAL RANKINE POWER CYCLES



It is helpful to summarize the relative merits of each power cycle:

Subcritical Isobutane Cycle Advantages

1. Isobutane cycle operates at lower pressure resulting in lower equipment costs.

2. Inefficiencies in converting shaft work to pump work favor the isobutane cycle, due to its lower pumping requirements.

3. The log mean temperature difference for the isobutane cycle is slightly greater for the same temperature approach differential -- boiling heat transfer offers advantages in reducing required heat transfer surface area.

Supercritical Propane Cycle Advantages

1. The supercritical propane cycle offers a better theoretical efficiency for utilizing heat from a "sensible heat" thermal resource such as geopressure brine.

2. The equipment to execute the supercritical propane cycle is simpler in that the primary heat exchanger is an extended process feed heater (economizer). The two-phase heat exchanger, the mist extractor, and the control system required in the subcritical system are eliminated.

3. The condenser for the subcritical isobutane system will have both a larger capacity requirement than a comparable supercritical propane system and a desuperheater.

4. The double boiling cycle includes more complex controls and components for a small increase in efficiency.

The application of comparable power plant component efficiencies to the power cycles do little to change the relative merits of the power cycles.

The shaft power required to drive the feed pump may be transferred within the cycle by use of an electric drive train with efficiencies as follows:

	EFFICIENCIES
Generator windage and bearings	92 %
Electric cables and motor	92 %
Pump	95 %
Efficiency product	80 %

For purposes of internally generating feed pump energy, the net penalty to the cycles can be calculated.

PUMP WORK COMPARISON FOR EACH CYCLE

	SUPERCRITICAL PROPANE BTU/LBM / KJ/KG	SUBCRITICAL SINGLE BOILING BTU/LBM / KJ/KG	SUBCRITICAL DOUBLE BOILING BTU/LBM / KJ/KG
Pump Work (80% Eff.)	6.1 / 14.2	1.5 / 3.5	2.3 / 5.3
Expander Work (80% Eff.)	22.4 / 52.1	13.6 / 31.6	15.7 / 36.5
Net Shaft Work	16.3 / 37.9	12.1 / 28.1	13.4 / 31.2

The energy consumption of the plant auxiliaries is approximately the same. The supercritical cycle is superior to the double boiling cycle. It requires fewer heat exchangers with a total area comparable to the double boiling cycle. The double inlet turbine will be more expensive on a unit capacity basis for the double boiling cycle. A possible alternate high pressure turbine-driven feed pump arrangement would be more costly and present the same control difficulties. The operation of a small complicated system will probably not be in concert with the single well production scheme concept.

The process heat exchanger for brine to propane will be a single section unit. The isobutane exchanger will have a comparable economizer section to the propane cycle and in addition an evaporator section with mist eliminator. The cost trade is between more heat transfer surface in a single unit or a smaller surface divided between two units with their duplicative controls. Ranges for overall heat transfer coefficients indicate the total surface area requirements for the cycles are comparable. The principal resistance to heat flow in the process heater will be on the brine side. In the condenser, it will be on the air side. Mechanical simplicity favors the supercritical cycle.

The calculated net improvement in cycle performance of the double boiling cycle over the single boiling cycle is 11%. For the supercritical propane cycle over the single boiling isobutane, the improvement was 35%. Calculations by O. J. Demuth [1] of INEL cited in 18% advantage for double boiling over single boiling and 42% higher for a 90% propane, 10% isopentane supercritical cycle over a single boiling isobutane cycle. The Demuth calculations were performed for a 280°F resource with cooler condenser conditions. The differences between the calculations in this paper and the INEL study can be accounted for by the slightly different condenser and boiler conditions. The relative relationships of performance remain consistent with the supercritical cycles rated as approximately 20% superior to double boiling subcritical Rankine cycles.

A more complete plant energy balance can be developed for a hypothetical power plant utilizing a 4,396 kW_t (15×10^6 BTU/hr) heat source based upon the preceding example but including parasitic loads.

PLANT ENERGY BALANCE

	SUPERCRITICAL PROPANE (kW)	SUBCRITICAL SINGLE BOILING ISOBUTANE (kW)	SUBCRITICAL DOUBLE BOILING ISOBUTANE (kW)
Generator Output	591	359	415
Pump Power Requirement	117	29	45
Condenser Fan Power	47	50	50
Auxiliaries	30	30	30
POWER OUTPUT	397	250	290
THERMAL EFFICIENCY (%)	9.0	5.6	6.6

Brine Heat Source = 4396 kW_t = 15×10^6 BTU/hr

Carnot Cycle Thermal Efficiency - 17.56%

Assumed Parameters:
(Turbine Efficiency = 80%)
(Generator & Windage = 90%)
(Electric Motor Efficiency = 92%)
(Positive Displacement Pump = 95%)

A common basis economic comparison is helpful to further illustrate the following underlying point. Economic performance is directly linked to thermodynamic performance. The chief arguments used to support the selection of low vapor pressure fluids has been (1) lower pumping costs (both capital and fuel) and (2) lower costs attributed to low pressure piping. These statements are true but not overriding with regard to economics or good thermodynamic design practice.

[1] Reference 2. pp 9-11.

The following cost analysis compares the cycles on both a common energy balance basis and a common unit cost basis for capital equipment. Due to the fact that there are economies of scale for the propane feed pump and turbine generator set, some "real world adjustments" to the cost comparison have been made. The unit cost for a two-stage turbine will add to the unit price for the double boiling cycle. The plant envisioned will be composed of shop fabricated process skids with minimum field assembly involved.

PLANT COST ESTIMATE

CAPITAL EQUIPMENT ITEM	SUPERCRITICAL PROPANE ($K)	SUBCRITICAL SINGLE BOILING ISOBUTAN ($K)	SUBCRITICAL DOUBLE BOILING ISOBUTANE ($K)
AC Condenser	170	182	174
Brine/ Liquid Hydrocarbon Exchanger	190	83	95
Boiler (Brine on tube side with mist eliminator)	-	69	90
Feed Pump	70	16	26
Turbine Generator	236	195	250
Skid Fabrication	120	120	135
Field Erection	60	60	60
TOTAL PLANT COSTS	846	725	830
PLANT UNIT CAPACITY COST ($ kW)	2,131	2,900	2,862

The supercritical propane cycle will provide lower unit capitalization charges and a greater net cash flow to the operator. The total capital required is greater for the supercritical propane cycle for a specified resource. The incremental investment for supercritical over the subcritical plant is very attractive. The cost of the power system is incidental to the overall cost of drilling the wells.

For purposes of this analysis the decision to add a low temperature power system is viewed as an incremental investment. It is anticipated that the first decision to install generation capacity will be made to meet onsite power requirements for long-term gas production operations. Cycle optimization through parametric economic calculations based upon a detailed engineering design are beyond the scope of this paper. However, the capital cost per unit output is the dominant economic factor and further optimization studies will not change the relative merits of the conclusions reached herein.

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