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THE VARIABLE PRESSURE SUPERCRITICAL RANKINE
CYCLE FOR INTEGRATED NATURAL GAS AND POWER
PRODUCTION FROM THE GEOPRESSURED GEOTHERMAL
RESOURCE

By
Fred L. Goldsberry

March 1982

Geopressure Projects Office
Houston, Texas

U. S. DEPARTMENT OF ENERGY
Geothermal Energy

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Abstract

The purpose of this paper is to describe a small-scale power plant cycle that utilizes both a variable pressure vaporizer (heater) and a floating pressure (and temperature) air-cooled condenser. Further, it defends this choice on the basis of classical thermodynamics and minimum capital cost by supporting these conclusions with actual comparative examples. The application suggested is for the geopressured geothermal resource. The arguments cited in this application apply to any process (petrochemical, nuclear, etc.) involving waste heat recovery.

Executive Summary

The variable pressure supercritical Rankine cycle is favored for power generation from a liquid phase heat source. The conventional subcritical Rankine cycle is preferred for power generation from a condensing vapor heat source. For the expected brine temperature range of 240°F to 360°F (389°K to 456°K) in the geopressure geothermal resource, the working fluid of choice will be propane. Isobutane is preferred from 360°F to 410°F (456°K to 483°K) and pentane from 410°F to 520°F (483°K to 544°K). Above 450°F, cracking of the process fluid creating non-condensable gases becomes an ever increasing problem with temperature. Conventional flash steam systems will dominate the application to resources at these higher temperatures.

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 gasoline and liquified petroleum gas production plants. These are operated as units of the same physical size contemplated for a single well geopressure geothermal power plant. A power cycle analysis and process layout are presented along with thermodynamic and economic comparisons.

The supercritical propane cycle can produce 37% more power at a 25% lower capital cost per unit output than an equivalent double boiling subcritical isobutane cycle operating from a common resource. Variations in condenser conditions or brine temperature will not vary this relationship appreciably within the design limitations imposed by the geopressure geothermal resource. The complexities introduced by a double boiling cycle are not economically offset by increased generating capacity over the single boiling Rankine cycle.

A plant design configuration and control philosophy are presented. The principal unknowns involved in the operations of a power plant utilizing brine from a geopressure geothermal well are scaling and corrosion control within the brine propane heat exchanger, operating manpower, and periodic preventive maintenance requirements. Confirmation of component performance and reliability will reduce the perceived high risk of long term capital commitments to the production of the resource. Operating experience can be gained to support equitable rule-making by regulatory bodies.

Introduction

The geopressure geothermal resource lies along the Texas and Louisiana Gulf Coast in the Frio, Wilcox, Vicksburg and Tuscaloosa sands. Knowledge of the geopressure has come from thirty years of oil and gas exploration and drilling experience. The geopressure horizon lies generally below the three-thousand meter datum below mean sea level and is comprised of under compacted sand bodies containing brine under lithostatic pressure. The sands are separated by shale breaks which act as sealing barriers to vertical brine flow and as thermal insulators to the flow of geothermal heat. The resulting reservoirs are abnormally pressured and able to lift the fluid contained therein to the surface with a considerable excess of hydraulic energy.

The petroleum industry has been interested in controlling or preventing geopressure brine flows rather than evaluating their potential as an energy resource. The shortage of natural gas in recent years has spurred interest in geopressure exploration because the brine is typically saturated with natural gas in quantities of from thirty to fifty standard cubic feet per barrel ($5.34 - 8.91 \text{ M}^3 \text{ gas/M}^3 \text{ brine}$). The large extent of the resource has prompted various estimates of recoverable gas reserves that could extend the U.S. reserve life index from five to two hundred years. The U.S. Department of Energy through the Geopressure Projects Office is attempting to quantify the extent and nature of the resource while gaining exploration and production experience.

Full utilization of the resource requires that the geothermal heat and geopressure hydraulic energy be recovered. Efforts are being made in hydraulic turbine and thermal power plant design to define the most practical and economically viable means for energy recovery and electric power generation. This paper represents a common reference statement for future process engineering and development tasks in the Geopressure Geothermal Program.

Power Cycle and Gas Processing

Constraints Imposed by the Geopressure Geothermal Resource

There are 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 variable pressure supercritical Rankine power cycle utilizing propane for the recovery of thermal energy from the geopressure geothermal resource. This power cycle is integrated into a natural gas recovery scheme that conserves reservoir pressure for brine disposal and produces gas at pipeline pressure. A power plant concept is presented based upon the present understanding of the geopressure geothermal resource.

Several factors emerge with regard to the resource which 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 250°F (395°K) due to its depth. The temperature will increase predictably with depth. In the Tuscaloosa trend, wells completed in the interval from 6,000 to 7,000 meters will provide brine temperatures on the order of 450°F (505°K). The dominant temperature range to be expected in the Frio and Wilcox sands of interest in the principal geopressure geothermal fairways will fall between 280°F (411°K) and 350°F (444°K). Due to the heating of upper rock formations, the bottomhole fluid temperature will be asymptotically approached at the surface as the well is produced. At the expected production rate of twenty thousand barrels per day, the brine surface flowing temperature will approach the bottomhole value by 10°F (5.5°K) within a few days. Any acceptable power plant design must operate throughout the anticipated transient temperature range. Furthermore, a single portable plant design that can be used throughout the resource base area is most desirable and appears to be feasible. Rapid cooling of the brine is desirable as a means to inhibit scale formation and corrosion activity. As CO₂ 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 scaling and the load on the gas sweetening plant. The CO_2 retained may provide the margin of safety in a pipeline gas specification to avoid acid gas removal entirely. Brine cooling prior to gas separation reduces the load on the gas dehydration plant.

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 150°F (340°K), there appears to be little reason for further cooling for the sake of disposal compatibility. By cooling the brine to 80°F (300°K), 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 150°F (340°K) can facilitate precipitation. Anticipated wellhead pressures will range from 200 to 600 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 600 psi (40 bar) will allow safe working pressures up to 1,250 psi (85 bar) at 350°F (422°K). These flanges are readily available in standard packaged equipment and piping.

The overall production schematic depicted in Figure I demonstrates the integration of the supercritical power cycle into the natural gas processing system. Gas production will be the primary economic consideration requiring that power production be placed on a non-interference basis. The power production system must be capable of following the production transient of the well. The power plant will be electrically base loaded with its output determined by well production. Geopressured brine wells cannot be economically produced by basing well production upon power plant requirements. Power generation is clearly a second priority in resource production economics.

FIGURE I. GEOPRESSURE GEOTHERMAL ENERGY PRODUCTION

The recovery of hydraulic energy will in many cases not be considered because the bulk of the hydraulic energy will be used to produce the reservoir. In any case, hydraulic energy recovery will be incremental to the installation of the thermal power generation system. At this point in the well test program it appears that only one disposal well per production well will be required and that separator discharge pressure will be sufficient to dispose of all produced fluids. Hence no disposal pumping will be required.

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.

Pressure maintenance during brine cooling is necessary to provide sufficient working pressure to control flow through the separator and to enter the disposal well. Many pipelines in the Gulf Coast areas operate at/or below 80 bar. 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 gas.

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. A paper published by the Gulf Universities Resources Consortium⁽¹⁾ cited the practicality of single well production systems without delving into the nature of the power cycle involved. Their conclusions were based upon reservoir engineering considerations and brine gathering system costs as opposed to electrical system gathering costs.

(1) Reference 8, p 13.

Literature Review

Much has been learned about the nature of the geopressure geothermal resource during the extensive geopressured well testing activities in 1980-81. The Department of Energy's program is quickly moving toward a need to specify, design and test a prototypical power plant that could provide economic answers to the question of the real value of hydraulic energy.

A survey review of both government and private work revealed that actual operating geothermal power plants were of the flash steam, single boiling isobutane or double boiling isobutane design. A review of current literature suggested that those who had performed independent theoretical or paper studies on the problem had generally chosen to proceed with evaluations involving supercritical propane cycles. Ofttimes these studies were aimed at an issue other than optimum power cycle selection. Further investigations revealed that many individuals had independently and often intuitively reached a conclusion supporting a supercritical cycle. But, they typically were not directly involved or advisors to those making procurement decisions.

Two needs immediately arose to support program management decisions for exploration planning. The first was to have the capability for developing real power plant cost data that could be used to place a value upon the geothermal energy component associated with geopressured gas wells. The second was to provide an orderly position document that logically supports the specification of a plant design based upon actual field conditions, thermodynamic fundamentals, realistic equipment performance, and a common cost base for comparison. As a general rule the power plant designer is not directly involved in the exploration and production realities of the "oil patch."

The approach taken in this effort was to set aside previous work and proceed with a systematic analysis of the problem of power plant design on the basis of (1) thermodynamics, (2) process fluid optimization, (3) operating environment, (4) available equipment and (5)

a real world cost comparison. This resulted in a plant design that fits the geopressured geothermal resource. It also allowed categorization of previous studies into concept groupings.

The first grouping in the literature involves the "conventional" approach which was based upon "building what everyone else is building," i.e., flash steam and boiling isobutane cycles. The classic power production study was performed by John Wilson, et al., (Reference 12) in 1975 describing in detail a flash steam and boiling isobutane system for an assumed 375°F resource. The binary boiling isobutane cycle was cited as the preferred approach for the lower end of the resource temperature scale. An EPRI study by Pundyk (Reference 11) presented an analysis of an operating double boiling isobutane plant which is representative of the class of binary cycles that have been reduced to hardware by industry and government. The studies by Blair, et al. (Reference 3) is typical of the resource economic evaluation efforts which have centered upon boiling isobutane cycles.

The second grouping revolves about the conceptual design study performed for the U. S. Energy Research and Development Administration by Brown & Root which as early as 1975 concluded that a supercritical cycle utilizing propane for a central station concept was the preferred way to go. The proposed power cycle included a recuperator that allowed turbine operation completely within the superheat region. The power cycle and working fluid selection processes were described in sparse terms. The bulk of the document is concerned with facility details. Much of the proposed Brown & Root design is devoted to building a conventional large central station with a wet cooling tower served by an extensive brine gathering system.

In a recent paper by Baudet, et al., (Reference 2) a supercritical cycle is described for heat recovery from a diesel engine exhaust. The principal items of interest are the proposed use of an air-cooled condenser and a radial inflow turbine for a small scale. The origins of this concept are to be found in gasoline plant technology. It is

also a recuperated cycle operating up to 500°F (530°K). Of passing interest was the 1979 AIChE student contest problem (Reference 5) which was an optimization of a supercritical propane cycle with recuperation.

The objective of this writing is to make a clear case for the supercritical propane cycle over boiling cycles for application to the geopressure geothermal resource in the way that we now understand it will be produced. Further, there is no power cycle that is likely to be superior either thermodynamically, economically or more physically compatible with the geopressured geothermal resource than the supercritical cycle. Working fluids will be examined to demonstrate why either propane and isobutane will be the choice depending upon the resource temperature.

A novel process control philosophy is presented for the first time. This approach will allow one basic plant design to suffice for any foreseen application in the geopressure geothermal resource. It is the variable working pressure concept that allows the elimination of a recuperator and allows positive control of the turbine expansion process down to the saturation point. Component selection is aimed at simplicity, availability, and flexibility. A relative cost comparison is presented for application to other economic studies. The study refines the power cycle outputs into a bus bar kilowatt value. Finally a work output curve is presented that can be used as a basis for estimating the best "real world" energy recovery that can be expected from a geothermal brine resource from 250°F to 360°F. This correlation will allow for a consistent and realistic evaluation of geopressure geothermal brine reserves for total energy content on a unit basis. A short form data sheet has been compiled that can be used to calculate a recoverable energy content per barrel. This will be useful in prioritizing reservoirs for further testing.

Power Cycle Selection

The principal path of approach to a power cycle specification is to first select the cycle, then a working fluid. The process conditions are then optimized for the cycle and the thermodynamic envelope in which it is to operate. The first step in selecting a power cycle is to determine its geometric compatibility with the heat source and sink. The working fluid is selected in accord with the physical constraints imposed by the mechanics of the process.

The temperature-entropy diagram in Figure II depicts the resource and heat constraints. It is useful because the work output of a power cycle is proportional to its area. The more fully a cycle fills the thermodynamic envelope, the more efficient the cycle will be. A power cycle with a quadrangular shape will fit within the grounds imposed by the thermodynamic character of the resource. A temperature difference must be suffered between the resource and the heating portion of the cycle as well as between the cycle heat rejection leg and the ambient temperature heat sink. Both the expansion work output and pump work requirement must suffer some increase in entropy in keeping with the second law of thermodynamics.

The objective of the process designer is to devise a thermodynamic cycle that will encompass the largest area within the resource thermal envelope to maximize work output. For a constant temperature heat source the rectangular shaped Carnot cycle describes the theoretically most efficient means of power generation. The power cycle designer's preference is to utilize heat from any resource in a fashion as close as possible to the ideal. Many techniques have been used to alter the shapes of simple power cycles to approximate a series of Carnot cycles either vertically or horizontally assembled to match the temperature profile of the heat source.

When the heat source is a condensing vapor, the Rankine cycle using a boiling fluid can be disposed to match the thermal characteristic of the resource. Such is the case for steam dominated geothermal systems.

AVAILABLE WORK LOST TO HEAT TRANSFER

AVAILABLE WORK LOST TO POWER CYCLE

FIGURE II. POWER CYCLE MATCHING TO COOLING AND HEATING
CURVES FOR THE HEAT SOURCE AND HEAT SINK

The objective of this effort is to examine the power cycle requirements for efficient exploitation of the geopressure geothermal brine resource and to provide an evaluation and operating description of the power cycle of choice. The preferred power cycle is a supercritical cycle. A supercritical heat exchanger will have a larger surface area but is simpler in mechanical design. More pump work is generally required by the supercritical process due the higher working pressures which in turn require more structural material in piping and heat exchangers. In times past, the increased capital requirements for heavier hardware led to economic compromises based upon cheap fuel in favor of lower pressure processes. The higher heat co-efficients for two-phase systems imply a reduction in heat exchanger surface area and hence construction material. These traditional arguments in favor of low pressure systems are no longer valid as will be demonstrated in the text of this paper. The energy crisis may be described as a major shift in the cost relationship between energy and capital equipment. The power plant designer of the future will find that thermodynamics and economics are more inextricably linked than ever.

In many geothermal applications power cycles have been operated at non-ideal conditions. One must exercise care in comparing thermal efficiencies between competing designs. For example, there are many applications where a small amount of power is generated by utilizing brine heat from 300°F to 250°F (422°K to 388°K). This looks good for the plant specifications but produces only a fraction of the power that otherwise might be available. When the capital investment of the well plus the commercial power grid connection are considered, the economics of the system favor maximization of power generation. In this instance production of all of the available power not just a part of it is the objective. Brine discharge temperatures are limited by heat exchanger design, scale, corrosion, and injection considerations.

Working Fluid Selection

A power cycle design can be described by constraints peculiar to the geopressure geothermal resource and its production methodology. The heat source will be thermally subcooled brine at 225°F - 410°F (280°K to 480°K) containing large quantities of (20,000 to 18,000 ppm) chloride and carbonate salts. Production rates will range from 5,000 to 40,000 bbl/day (.0092 - .0736 M3/sec). Well spacing will probably be no closer than three kilometers due to sand permeabilities of .01 to .2 Darcies. Brine handling equipment will be designed to operate between 10 and 100 bar for reasons of equipment availability and operating conditions in natural gas pipelines. Separator pressures lower than 10 bar result in flashing of steam and increased production of CO2 without appreciable incremental recovery of natural gas. Maximum natural gas pipeline operating pressures will not exceed 100 bar.

There are several desirable characteristics which should be sought in the selection of a working fluid:

1. Non-corrosive
2. Non-flammable
3. High molecular weight (minimum turbine size)
4. Non-toxic
5. Low cost - availability - storability
6. Compatibility with "off the shelf" equipment
7. High liquid density to minimize pumping costs
8. Low vapor pressure with respect to pressure and temperature operating range considerations.

After a review of the required thermodynamic properties, the selection is reduced to a choice among three basic categories of fluids. Water (steam) was eliminated due to the requirements for a vacuum condenser and the high solids carryover expected from the brine flash system.

Inorganic

Gases

Halocarbons

Hydrocarbons

Ammonia (NH₃)

R-22

Pentane (C₅)

Sulfur Dioxide
(SO₂)

R-504

Isobutane (i-C₄)

R-12

Propane (C₃)

The inorganic gases were eliminated for a variety of reasons. Both are toxic and corrosive. Refrigerant grade anhydrous ammonia is difficult to locate in commercial stocks. Fertilizer grades contain some moisture. Absorption refrigeration grades of ammonia operate with water and thus can contain unacceptable levels of moisture. Sulfur dioxide was popular as a refrigerant in the 1930's but the advent of halocarbons has driven it from the market.

The halocarbons while have good properties and availability, but are expensive (\$2.20 - \$6.90/kg) compared with ammonia (\$.26/kg) and LPG (\$.24/kg). In practical plant applications where flanges and pump seals are expected to leak, working fluid inventory costs can be prohibitive. Environmental concerns with venting also represent difficulties for operation within guidelines.

The widespread use of liquified petroleum gases (hydrocarbons) in process and home heating fuel applications indicates a widespread knowledge of safe handling practice, availability of code stamped equipment, and readily delivered liquified gas supplies.

The selection of hydrocarbon can be made by reviewing specific thermodynamic properties. It is presumed desirable from the standpoint of efficiency that the two-phase region be avoided during expansion in the turbine.

Summary of Critical Properties

		CRITICAL PROPERTIES				REQUIRED TURBINE INLET TEMPERATURE AT 1000 PSIA (68 BAR) FOR A TURBINE EFFICIENCY OF 80% FOR LIQUID FREE EXHAUST	
PARAFFIN	MOLECULAR WEIGHT	TEMPERATURE		PRESSURE		(°F)	(°K)
		(°F)	(°K)	(PSIA)	(BAR)		
Ethane	30	90	305	707	48	150	339
Propane	44	206	370	616	42	270	406
Isobutane	58	274	408	529	36	340	444
N-butane	58	305	425	550	37	370	461
Pentane	72	385	470	488	33	450	506

The similarities in critical temperatures and pressures indicate an apparent translation of the phase envelope with molecular weight and temperature. This characteristic in hydrocarbons lends itself to the custom blending of working fluids to produce optimum thermodynamic properties. Commercial grade propane will include some ethane and butane but will perform much as a pure component. Where supercritical operation is specified the properties of mixtures become less important in heat exchanger design. The condenser may suffer a small degree of non-isothermal behavior but this should not be significant to operations.

Working Fluid Applications
for the Supercritical Rankine Cycle

WORKING FLUID	WORKING FLUID *				BRINE			
	TEMPERATURE RANGE				TEMPERATURE RANGE			
	FROM		TO		FROM		TO	
	°F	°K	°F	°K	°F	°K	°F	°K
Propane	220	378	290	417	240	389	360	456
Isobutane	280	411	350	450	360	456	410	483
Pentane	390	472	460	511	411	483	520	544

* Power cycle operating pressures will vary from the critical pressure of the working fluid to approximately 1200 psia (82 bar).

Upon review of the physical criteria, ethane can be eliminated from consideration because of low condensing temperature limitations. Normal pentane requires a process temperature in excess of that likely to be encountered in the resource. Isobutane would require operation at/or near the upper end of the spectrum. It has desirable pumping properties, vis-a-vis propane (higher density and lower vapor pressure). It is desirable to have the organic working fluid at higher pressure than the brine to facilitate leak detection and prevent contamination of the process stream with brine.

Propane appears to fit within the range of thermodynamic constraints for the power cycle most effectively. The property exhibited by isobutane of expanding isentropically into the superheat range from a saturated vapor state would be a decided advantage in a boiling/condensing heat exchange as might be encountered in a steam dominated resource. Superheated propane expands isentropically into the phase envelope offering an advantage in equipment simplicity and control flexibility. A single basic plant design may function with reasonable efficiency throughout the range of anticipated thermal conditions. Working fluid optimization by mixing propane with other hydrocarbons can be effected later in the program as conditions warrant. Hydrocarbon mixtures can be optimized by considering average molecular weight as controlling factor for physical characteristics. Note the relationship of pure component molecular weight to critical properties in the summary. The condensing of hydrocarbon mixtures can be slightly nonisothermal but this should not significantly affect operations.

Demonstration of the effectiveness of adjusting fluid properties to match the resource temperature by component mixing will be a logical step in the refinement of the supercritical design concept.

Supercritical Propane and Subcritical Isobutane Cycle Comparison

Much of the current technical literature cites boiling isobutane as the cycle and working fluid of choice because of its generally low pumping horsepower requirements and its characteristic behavior of expanding isentropically from a saturated vapor state into superheat. The larger the pumping requirements for a power cycle, the greater the conversion losses of mechanical work within the cycle. If, however, the thermodynamics of one 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.

An example is shown in Figure III that compares 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 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) to allow the working fluids to share a common enthalpy scale. This unique set of conditions allows objective comparisons of thermodynamic properties for a common heat source. 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. The mass flow rate in the high pressure cycle is 66% of the total flow through the condenser. This rate is compatible with maintaining the 10°F (5.55°K) temperature approach in the heater. The later economic comparison for capital construction costs were based upon a 15 MMBTU/hr heat source. This allowed each power cycle to be based upon a nominal 1,000 ton air-cooled refrigeration packaged condenser for cost estimating purposes.

FIGURE III. COMMON UNIT MASS - THERMODYNAMIC COMPARISON

FOR SUBCRITICAL AND SUPERCRITICAL 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 derived brine.
2. The equipment to execute the supercritical cycle 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 isobutane system are eliminated.
3. The condenser for the subcritical isobutane system will have a larger capacity requirement than the comparable supercritical propane system and a desuperheater with controls.

SUPERCRITICAL CYCLE

SUBCRITICAL CYCLE

DOUBLE BOILING CYCLE

FIGURE IV. POWER CYCLE COMPONENT COMPARISON

The equipment required to execute each of the cycles is graphically depicted in Figure IV. The attractiveness of the supercritical cycle from an operational simplicity standpoint is apparent.

In all cases the energy consumption of the plant auxiliaries is approximately the same. The supercritical cycle is superior to the double boiling cycle. The double boiling cycle requires more heat exchangers of nearly the same total area as the single boiling cycle. The double inlet turbine will be more expensive to purchase and more complicated to operate. The construction requirements for a double boiling power plant will probably not be in concert with the single well production scheme.

The process heat exchanger for brine to propane in the supercritical cycle will be a single section unit. The isobutane subcritical exchanger will have a comparable single-phase section to the propane cycle and in addition an evaporator section with mist eliminator. The cost trade becomes more heat transfer surface in a single unit or a smaller surface divided between two units with their attendant controls. Mechanical simplicity favors the propane cycle. Ranges for overall heat transfer coefficients indicate the total surface area requirements for both 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.

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⁽²⁾ of INEL cited an 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 can be accounted for in slightly different condenser and boiler conditions. The relative relationships of performance remain consistent with this study showing the supercritical cycle rated as approximately 20% superior to double boiling subcritical Rankine cycle.

POWER PLANT ENERGY BALANCE

POWER CYCLE:	SUPERCRITICAL PROPANE	SUBCRITICAL SINGLE BOILING ISOBUTANE	SUBCRITICAL DOUBLE BOILING ISOBUTANE
	(KW)	(KW)	(KW)
GENERATOR OUTPUT:	591	359	415
PUMP POWER:	(117)	(29)	(45)
CONDENSER FAN POWER:	(47)	(50)	(50)
AUXILIARY POWER:	(30)	(30)	(30)
NET POWER:	397	250	290
THERMAL EFFICIENCY:	9.0%	5.6%	6.6%

BRINE HEAT SOURCE: 4396.2 KWT(15,000,000 BTU/HR.)

CARNOT THERMAL EFFICIENCY: 17.6%

TURBINE EFFICIENCY: 80%

GENERATOR & WINDAGE LOSSES: 10%

ELECTRIC MOTOR EFFICIENCY: 92%

POSITIVE DISPLACEMENT FEED PUMP MECHANICAL EFFICIENCY: 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 thermodynamics design practice. 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 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 is more attractive than the base investment. 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 a long-term operation. 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 will not change the relative merits of the conclusions reached herein.

COMMON BASIS POWER PLANT COST ESTIMATES

CAPITAL EQUIPMENT ITEM	SUPERCRITICAL PROPANE CYCLE	SUBCRITICAL SINGLE BOILING ISOBUTANE CYCLE	SUBCRITICAL DOUBLE BOILING ISOBUTANE CYCLE
	(\$K)	(\$K)	(\$K)
A/C CONDENSER	170	182	174
.L HEAT EXCH.	190	83	95
BOILER	—	69	90
FEED PUMP	70	16	26
TURBINE-GEN.	236	195	250
SKID FAB.	120	120	135
FIELD ERECTION	60	60	60
TOTAL	846	725	830
UNIT COST (\$/KW)	2,131	2,900	2,862

The Variable Pressure Supercritical Rankine Cycle

A more detailed specification of the supercritical propane cycle is needed to define the mechanical components that will be required for execution. A geopressure geothermal well will operate in a transient mode throughout its life. The formulation of control set points must be designed to enable the power plant to take full advantage of the thermal energy available in any given well. A single line flow diagram is supplied in Figure V to describe the proposed power conversion concept.

There are three features that are important to small-scale plant economics. The first is that operation be unattended. The second is that off the shelf/shop packaged components be used wherever possible. Finally plant portability and operating flexibility are important in producing a depletable resource.

FIGURE V. SUPERCRITICAL POWER CYCLE WITH CONTROL SCHEME

To match resource requirements the plant design will incorporate a modular concept. Individual modules should have large turndown ratio capability and be tolerant of a wide range of operating conditions.

The control system is based upon a simple cascade philosophy to meet these requirements. The flow rate of the well is controlled by the choke setting. The brine production will flow through the shell side of the heat exchanger to the separator. Separator back pressure is regulated by a back pressure control valve in the gas line. Separator drum level is controlled by a dump valve on the brine line coming from the vessel. Propane circulation will be regulated by a temperature controller that senses propane temperature to the inlet of the expander turbine to actuate a propane feed pump bypass valve. The energy production rate is to be power limited (base loaded). The high pressure side of the propane process is regulated by the turbine throttle functioning as a back pressure valve. The back pressure setting is allowed to vary with temperature according to the curve developed later in Figure VI. The low pressure side is controlled by condenser temperature which can be allowed to float with ambient conditions.

Turbine speed will be controlled by the generator acting as a brake utilizing the commercial power grid as the speed control mechanism. As such, the power plant will function as a baseload system automatically tracking well production rates and temperature conditions. A decline in well production rate will result in a reduction of maximum propane temperature which opens the pump bypass valve matching propane and brine circulating rates. In the event of heavy scaling, a temperature reduction on the propane outlet will cause the propane circulation rate to be reduced giving rise to a high brine discharge temperature. The brine discharge temperature will be alarmed to the emergency shutdown system. As the propane flow rate diminishes, the turbine throttle will close in an attempt to maintain the variable temperature dependent cycle operating back pressure.

The turbine exhaust will be condensed in the free floating temperature air-fin cooler to take maximum advantage of ambient conditions. A thermostatically controlled multispeed motor will match condenser fan requirements to ambient air conditions. This allows the turbine to operate through a narrow back pressure range while allowing increased overall thermal efficiency of the plant on cool days through reduction of fan motor power requirements. The propane is then fed to the low pressure propane accumulator. The propane feed pump will operate between the low pressure accumulator tank and a high pressure nitrogen charged damper/accumulator. The condenser will be a conventional air-cooled heat exchanger. These are available in standard units complete with accumulators, inlet headers, fans and controls.

A hydrocarbon turboexpander driving an induction motor wired as a generator is currently available from several sources in units ranging from a few hundred kW up to the 15,000 kW range. The propane feed pump can be a multi-stage centrifugal or plunger pump with appropriate bypass valving.

To summarize the power cycle control system:

1. The propane circulation rate is controlled by the feed pump bypass valve as it senses propane temperature to the turbine.
2. The variable propane pressure at the turbine inlet is controlled by the throttle control nozzles acting to regulate back pressure.

An additional controller may be placed in the system to measure brine inlet temperature and to automatically set the maximum propane temperature and pressure points to compensate for startup transients and production well decline. Brine flow rate is automatically tracked by the propane circulation rate. Heat exchanger scaling/fouling or loss of the propane coolant will be detected by the high brine temperature alarm.

The Mollier diagram for propane provides a graphic illustration of the supercritical Rankine cycle and the constraints placed upon the system by the aforementioned considerations for temperature, pressure, and leakage. Point 1 on Figure VI begins with a saturated liquid propane stream being pumped from condenser pressure (150-300 psia (10.2 - 20.9 bar) corresponding to 80-140°F (300-333°K)) to a pressure above the critical value. The high pressure liquid is then heated from ambient temperature at Point 2 to a superheated vapor at Point 3. The vapor is then fed to a turbine where it expands to produce work and is discharged to the condenser as a near saturated vapor at Point 4. Condensing occurs in the air-fin cooler in the process leg from Point 4 to Point 1.

The power cycle operates with a floating condenser to match the ambient air temperature. The advantage is a significant increase to plant efficiency. The disadvantage lies in operating the turbine at off-design conditions which can reduce plant capacity. The radial inflow single stage turbine is more flexible in this regard than the more conventional multi-stage machines associated with conventional steam generation plants. Figure VII shows the small degradation of turbine efficiency with lower condenser temperature. The turbine output increases with reduced back pressure resulting in a net improvement in cycle efficiency.

FIGURE VI. TURBINE INLET TEMPERATURE FOR VARIABLE PRESSURE SUPERCRITICAL CYCLE

FIGURE VII. FLOATING CONDENSER IMPACT UPON
 EXPANSION EFFICIENCY OF RADIAL INFLOW TURBINE

The power cycle control point conditions will be configured to operate at maximum efficiency. A heat exchanger designed for use in a single-phase hydrocarbon heating regime will operate through a wide range above the critical pressure. The expansion of high temperature and pressure vapor into the two-phase region can cause turbine blade erosion and reduced expander efficiency if allowed to exceed a few percent. Selection of the appropriate initiation point for the expansion process is critical to the design of the plant.

The traditional means for designating the initiation point for the expansion line is to follow the process backward from the desired turbine exhaust condition, using the approximate performance that can be achieved with the turbine. As there is a desire to operate the turbine with a variable condenser pressure, an acceptable operating range must be established by analysis for the type of turbine to be used. A turbine of radial inflow design is normally used in hydrocarbon service. Condensate in the exhaust stream can be tolerated at levels less than five percent. A specific speed of .6 has been selected to yield an expander efficiency slightly above 80 percent. Two expansion lines are plotted from the condenser conditions to intercept the isobars. These will become the control set points for pressure and temperature for optimum cycle operation.

The next consideration is the cooling and heating curves for brine and propane. The operating pressure for the power plant is a function of maximum achievable propane temperature. FIGURE VI depicts the family of pressure curves on the temperature - enthalpy diagram that will be selected.

While the slope of the brine cooling curve is assumed constant due to a constant specific heat for brine, the curvature of the supercritical propane heating curve is pronounced. As the pressure of the propane is increased, the curvature is reduced allowing better differential temperature matching between the fluids in the heat exchanger. This in turn provides more complete utilization of the heat in the brine.

A close temperature approach at the hot end, while thermodynamically desirable, results in hotter brine discharge temperatures. A close temperature approach at the brine discharge end will result in insufficiently heated propane. An ever increasing pressure will cause the propane heating curve to flatten approaching a straight line. Aside from the apparent mechanical limitations on handling such pressures there occurs a diminishing return between the incremental power generated at higher pressures and the propane pump work required to feed the heat exchanger. Applying reasonable expander and pump efficiencies to the calculations results in the conclusion that the incremental increase in power output is small between 1,200 psia (82 bar) and 2,000 psia (136 bar) and possibly negative above 2,000 psia (136 bar) due to pump-expander irreversibilities. Consequently, working pressures above 1,250 psia (85 bar) [600 psia (41 bar) ASME flanges] were considered to be impractical.

Geopressured Geothermal Energy Calculations

The conversion of thermal energy into power has been the subject of numerous theoretical economic analyses and considerable discussion. A practical demonstration of how work output can be greatly increased with a comparatively small increase in temperature can be illustrated with the temperature - enthalpy diagram for the supercritical propane cycle. A series of cycle work output calculations were made assuming a 154 psia - 85°F (10.5 bar - 302°K) condenser condition beginning with a flowing brine temperature of 250°F (394°K). The cycle operating pressure will be set at the minimum point of 700 psia (47.6 bar) with a 10°F (5.5°K) closest temperature approach and a maximum propane temperature of 237°F (387°K). The brine discharge temperature is constrained by the temperature approach to 160°F (344°K). The net work output of the cycle was divided by the brine flow rate.

A series of calculations are made by raising the brine temperature 10°F (5.55°K) adjusting the brine discharge temperature to allow a 10°F (5.55°K) heat exchanger approach temperature differential and computing the work output per unit of brine flow. The cycle work output values were then normalized with respect to the 300°F (422°K) brine temperature. The 300°F (422°K) brine case represents the lowest temperature case for the maximum operating propane pressure 1,200 psi (81.6 bar); also the 300°F (422°K) temperature is a good mid-range value for the resource. Figure VIII demonstrates the behavior of this normalized work function ($\bar{W}$) with respect to brine temperature. There is a dramatic rise in relative work output above 300°F (422°K). This is the result of a lower brine discharge temperature because the thermal pinch point is shifted toward the cooler end of the heat exchanger. The dimensionless function ($\bar{W}$) demonstrates the characteristics of the propane cycle with resource temperature using achievable performance specifications for each piece of equipment.

The objective of this analysis is to demonstrate that the variable pressure supercritical propane cycle is the design of choice between 250°F (394°K) and 340°F (444°K) where the bulk of the geopressure

FIGURE VIII. DIMENSIONLESS WORK OUTPUT FUNCTION FOR THE
VARIABLE PRESSURE SUPERCRITICAL PROPANE CYCLE

ESTIMATED GEOPRESSURED GEOTHERMAL BRINE RECOVERABLE ENERGY CONTENT CALCULATION

NATURAL GAS ENERGY

KW-HR/BBL

$$.0001 \times \frac{\text{GWR}}{\text{HEAT RATE - 10,000 BTU/KW-HR}} \times \frac{\text{HHV}}{\text{HEAT RATE - 10,000 BTU/KW-HR}} = \underline{\hspace{2cm}}$$

THERMAL ENERGY

$$.0089 \times \frac{\gamma}{\text{THERMAL EFFICIENCY - 9\%}} \times \frac{C_P}{\text{THERMAL EFFICIENCY - 9\%}} \times \frac{\bar{W}}{\text{THERMAL EFFICIENCY - 9\%}} \times \left(\frac{T}{\text{THERMAL EFFICIENCY - 9\%}} - 175.0 \right) = \underline{\hspace{2cm}}$$

HYDRAULIC ENERGY

$$.000238 \times \frac{\Delta P}{\text{EXPANDER EFFICIENCY - 85\% GENERATOR EFFICIENCY - 92\%}} = \underline{\hspace{2cm}}$$

TOTAL ENERGY RECOVERABLE

=

GWR - GAS RECOVERED FROM SEPARATOR (SCF/BBL)

HHV - HIGHER HEATING VALUE (BTU/SCF)

γ - BRINE SPECIFIC GRAVITY (DIMENSIONLESS)

C_P - SPECIFIC HEAT (BTU/LBM-°F)

$\bar{W}$ - DIMENSIONLESS WORK FUNCTION FOR VARIABLE
PRESSURE SUPERCRITICAL PROPANE CYCLE

T - FLOWING WELLHEAD TEMPERATURE (° F)

ΔP - FLOWING DIFFERENTIAL PRESSURE ACROSS CHOKE (PSIG)

geothermal resource lies. The supercritical propane cycle plant as envisioned in this writing can function above 340°F (444°K) but it is believed that a shift to isobutane as the working fluid at this point will have some advantages.

The implications of the dimensionless relative work output function to the resource base energy estimates are substantial. The use of electrical output rather than a direct heat application was selected as a basis for evaluation because the power based examples is universally applicable. The power plant described herein can be standardized for use throughout the geopressure geothermal resource area. The value for electricity is easily determined and directly relatable to natural gas values for purposes of prospect economic evaluation. A sample calculation form is included following Figure VIII that can be utilized to evaluate a brine resource.

It is believed that this multi-pressure process design approach is unique to the geopressure geothermal resource. It allows the movement of portable plants from one well to the other. It allows for precise control of saturation in the turbine exhaust. The avoidance of custom designed fixed power plants in favor of the portable - skid mounted - mass produced standard design plant will result in significant cost reductions for new plants and maintenance. A higher salvage or residual value will result for the plant at the time of reservoir depletion.

Prototype Power Plant Experiment

The first geopressure geothermal power plant experiments will be designed to provide realistic operating parameter data for economic studies. The data will be necessary to institutionally pave the way for fair and equitable regulatory decisions. Small baseloaded power generation facilities have not traditionally been desirable from the utility company standpoint. Energy production income must offset the expense to an operating organization of maintaining standby fossil fuel "peaking" capacity.

This plant concept lends itself to low design risk scale up. The turboexpander-generator sizing constraints are well understood. The air-cooled floating temperature condenser can be paralleled with comparable units to increase capacity without altering system characteristics. Process pumping requirements can be met by conventional hydrocarbon pumps of various sizes. The supercritical propane brine heat exchanger represents the principal design uncertainty. The understanding of brine-side fouling factors and both brine and propane heat transfer coefficients is limited. Much of the work in hydrocarbon systems has been with isobutane for application in steam dominated geothermal systems. Geopressured geothermal brines have not been tested in the field for thermodynamic properties as extensively as have hydrothermal brines from the western United States.

It can be anticipated that a portable plant of less than one megowatt output can be built on three skids. The first skid will include an electric drive feed pump, and air-cooled condenser package, turbine generator package, and

controls. The second skid will be a portable transformer and substation or load to dissipate produced power. The third skid will support a shell and tube heat exchanger and gas separator. The exchanger skid will most likely require modification as experimentation progresses and will be subject to the harshest operating conditions in both scaling and corrosion, and pressure induced mechanical loads.

For purposes of early demonstration and experimentation, it is recommended that a plant be built around a 1,000 ton (4.4 mWt) packaged air-cooled heat exchanger. The electrical load will probably be resistance heater bank.. Startup power can be supplied by an auxiliary diesel generator. All plant drives will be electric motors. Controls are to be pneumatic and will be energized by two electric drive air compressors that are supplemented with a high pressure nitrogen cylinder as the emergency backup system.

A reduced scope brine cooling loop is planned for the initial testing. It will be an air-cooled propane condenser with a shell and tube process heat exchanger. This system can be used for dual purpose of gaining heat exchanger operations experience while cooling the brine prior to injection for scale and corrosion control in the disposal well.

It is anticipated that the plant will be used on a variety of test wells. The flexibility of the control system will be all important to successful evaluation of heat exchangers on multi-well tests. The large turndown ratio will allow time dependent heat exchanger fouling data to be collected and the effectiveness in scale inhibition techniques to be measured.

Finally, the plant should be designed and constructed by an experienced petrochemical plant A/E firm with packaged plant experience. Structural capacity should include hoist supports for ready maintenance of key equipment items. This plant will be torn down and rebuilt numerous times making serviceability an overriding factor in the skid layout. A small incremental expenditure for ease of maintenance can be repaid many times over in downtime costs.

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