DESIGN ANALYSIS
OF A
DRILLED GEOTHERMAL WELL

by

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NOMENCLATURE

A cross-sectional area

Approach ΔT (TP3-T4A) = (TP4-T4B) = (TP5-T4C)

D pipe diameter

d_e equivalent diameter

E total energy

f friction factor

g gravitational constant

h enthalpy

HR heat rate

J 778 Btu/ft.lbf.

K thermal conductivity

Knee ΔT (TP4-T3A) = (TP5-T3B) = (TP6-T3C)

L maximum well depth

$\dot{m}$ mass flow rate

N number of feedwater heaters

p pressure, also power

p_{ht} hydraulic turbine power

p_{st} steam turbine power

q heat flux

Q heat transfer rate

r radial coordinate

T temperature

U overall heat transfer coefficient

NOMENCLATURE (con'd)

V water velocity

z axial location

α gradient of water, also thermal expansion coefficient

η efficiency

ρ density

σ stress

General Subscripts

a stage a

b stage b

bfp boiler feed pump

c stage c

d downer

f friction

g ground

i inner

ins insulation

n-r nonregenerative

o outer

p primary

r riser

reg regenerative

ω water

-Temperature Subscripts

1 well inlet (downer)

2 well bottom

3 well outlet (riser)

INTRODUCTION

The temperature of the earth's crust, with only small variations, increases at a rate of 1.5F per 100 ft of depth. This represents a vast body of heat available to us at 850F, at an average depth of only 50,000 ft, i.e. only slightly deeper than our deepest oil wells. For many applications, this source of heat can and has become an undisputed competitor with many other forms of energy. The major problem is to tap this source of heat with sufficient heat transfer surface, since the earth's crust is essentially a poor thermal conductor, having an average thermal conductivity $(\overline{K})$ = 1.5 B/hr-ft-F.

All geothermal power generation to date has been produced by harnessing natural emissions, or by drilling relatively shallow wells, 10,000 ft or less, to tap existing high pressure water or steam reservoirs (see Table 1). However, due to the limited number and location of these sites, another method of tapping this underground heat is essential in order for geothermal sources to produce a significant fraction of future power requirements.

This new heat tapping method, to be used for producing electrical power, is called drilled geothermal power generation (DGPG). The objective is to drill a 50,000 ft deep well and suspend an insulated coaxial pipe within it as shown in Fig. 1. Cold water will then be pumped down the outer annulus and heated by the earth as it travels over some 375,000 sq ft of heating surface. After reaching the bottom,

TABLE I. WORLD GEOTHERMAL POWER GENERATION AND SOURCE

Field	Fluid	Source Rocks	Installed in 1971 (MWE)	Expected Development
Larderello	Steam	(2)	365	15% possible
New Zealand	Hot water	(1)	192	None planned
Japan				
Matsukawa	Hot water	(1)	20	Perhaps 60 by 1980
Monte Amiata	---	---	25	---
Omikobe	---	---	---	Perhaps 60 by 1980
Shikabe	---	---	---	7
Honshu	---	---	20	120
United States				
The Geysers, Cal.	Steam	(3)	302	110 per year to 1980, - 1,180
Inyo County, Cal.	---	---	---	*
Salton Sea, Cal.	---	---	---	*
Hawaii	---	---	---	*
Eureka County, Nev.	---	---	---	*
Mexico				
Cerro Prieto	Hot water	(4)	75	150 by 1980
Iceland (Namafjall)	Hot water	(3)	3	None known
Northern Taiwan	Hot water	(1)	---	*
USSR				
Pauzhetsh	---	---	6	Perhaps 25 by 1980
Kunashir	---	---	---	Perhaps 13 by 1980

Classification of Source Rocks
- (1) Acid volcanics
- (2) Fractured limestone; dolomite
- (3) Fractured greywacke
- (4) River delta sedements

*Sites with large known potential still in planning stage.

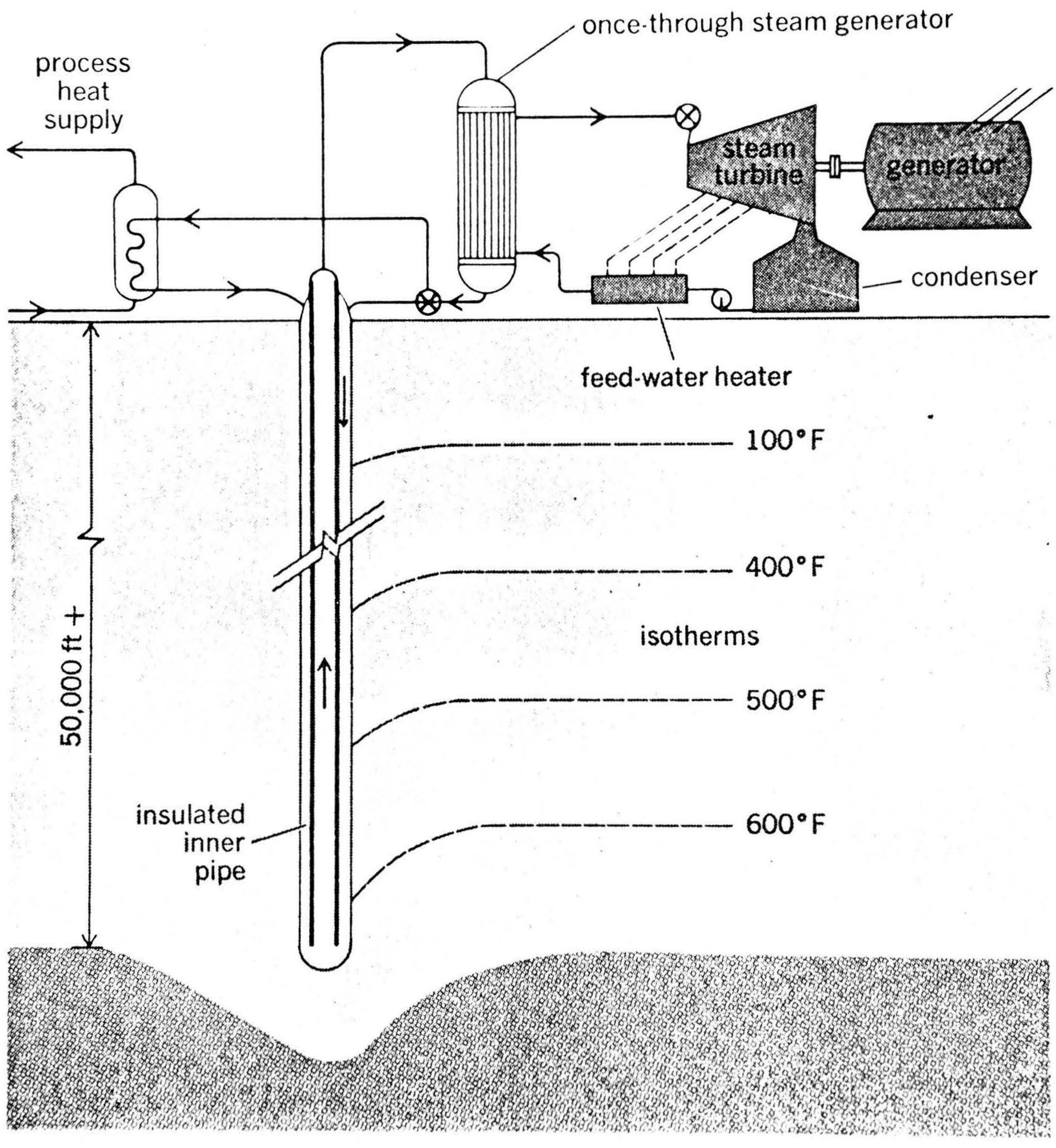


Figure 1. Drilled Geothermal Power Generating System [1]

the hot water will then travel up the inside pipe to the top of the well where it will be converted into electrical energy. It is worth noting that to date wells of this depth have not been drilled and the present goal in drilling has been to obtain oil; however, with the proper incentive wells can be drilled the proper size for geothermal utilization [2].

GOALS AND SCOPE OF RESEARCH EFFORT

In light of the DGPG concept, the goals of the research effort were as follows:

1. Feasible well structure and insulation design for a drilled geothermal well;
2. Determination of the optimum thermodynamic power cycle to be employed at ground level for converting energy of the high temperature water from the well into electrical power;
3. Optimum well operating characteristics for minimum electrical energy cost.

The research design began with a drilled, cased, and concreted 50,000 ft deep well. The design of an insulated center pipe was made in order to allow for the removal of heat from the earth's crust by the circulation of water down the outer annulus and up the inner pipe as shown in Fig. 2. The amount and type of insulation to be used in order to prevent the removal of heat from the hot water in the inner pipe to the cold water in the outer annulus was determined.

Dimensions for the well were calculated so that for a given mass flow there would be an equal average velocity in both the inner pipe and the outer annulus. The maximum length of a free hanging pipe was calculated and from this a proposed structural method for supporting

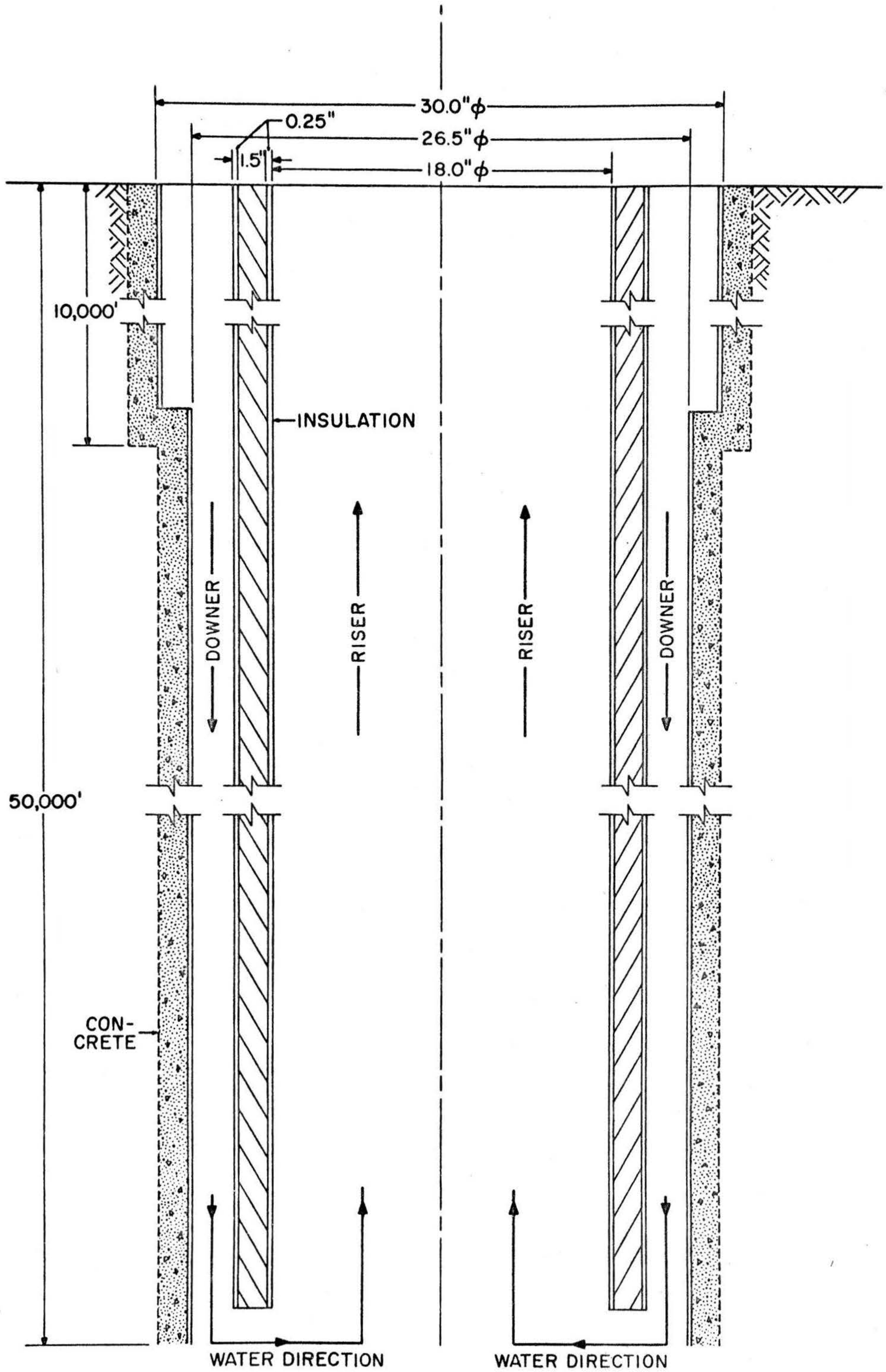


Figure 2. Preliminary Design of the Drilled Geothermal Well

the inner pipe was made. Thermal expansion on the order of 100 ft and pressures ranging from atmospheric to approximately 22,000 psi were also considered. Several designs were disregarded due to these problems, and these designs along with the proposed well design are discussed.

The second goal of the research was to optimize the thermodynamic cycle of the surface equipment to be used for the conversion of the hot water arriving at the wellhead into electrical power. The surface equipment included a once through steam generator, a steam turbine, electrical generator, condensor, circulating pumps, and a feed-water heater system as shown in Fig. 1. Two and three stage cycles were considered along with variations in the knee and approach temperature differences. Results of this optimization including graphs of thermal efficiency versus well outlet temperature for a given number of stages, and a given knee and approach temperature difference are presented. The effect of the knee temperature difference, approach temperature difference and the number of stages are also included.

The third and final goal of the research was to calculate the optimum well operating condition. As water is circulated through the well, the temperature of the earth cools down. For a given water mass flow rate, well diameter, gradient of the earth and well depth, it was found that after several hundred hours of operation the well outlet temperature dropped below an acceptable power generating temperature. Therefore the earth's temperature had to be regenerated by having no circulation of water for a certain period of time. This required the use of several wells utilizing the same surface equipment in order to produce continuous electrical power.

In a parallel and supporting study, Fink [3] was primarily concerned with modeling the subterranean heat transfer process, energy transport, and resulting transient earth temperature distribution of the geothermal well. The present author used the pressure and temperature history at the wellhead as determined by Fink to calculate the constantly varying electrical power generation that could be produced by the surface equipment. From this an optimum power generation was determined for a given water mass flow rate, well diameter, gradient of the earth and well depth. Based on this optimum condition, simple economics were used to determine an approximate cost of geothermal electricity.

ANALYSIS

I. DESIGN OF A FEASIBLE 50,000 FT WELL WITH COAXIAL PIPE

Insulating the Central Pipe

The principal objective is to bring the water from the bottom of the well, where the highest temperature is achieved, back to the top in the central riser, without significant temperature loss. Insulation will significantly reduce the heat transfer occurring along the entire 50,000 ft between the rising hot water in the inner pipe and the descending cold water in the outer annulus. Therefore, referring to Fig. 2, the first consideration in the structural design of the well was the amount and type of insulation to be used around the inner pipe.

The derivation of the riser temperature loss ΔT_{23} is as follows:

Assume steady flow of $\dot{m}$ in the riser from a bottom temperature, T_2, to a top temperature, T_3. Applying the First Law to the column:

$$q_{23} = \Delta h_{23} - \Delta z/J \qquad \text{B/lb.} \qquad (1)$$

$$q_{23} = \int_L^0 \pi D_i (U_i/\dot{m})(T_r - T_d)\,dz \qquad \text{B/lb.} \qquad (2)$$

where U_i is the overall heat transfer coefficient, through both water films, both steel liners, and the insulation (k_{ins}), the latter being the dominant resistance.

T_r is the water temperature at level z in the riser

T_d is the corresponding water temperature in the downer.

Substituting $U_i = 2k_{ins}/D_i \ln(r_o/r_i)_{ins}$ (3)

$$T_d = T_1 + \alpha_d z, \text{ with } \alpha_d = (T_2-T_1)/L \quad (4)$$

$$T_r = T_3 + \alpha_r z, \text{ with } \alpha_r = (T_2-T_3)/L \quad (5)$$

equation 2 can be integrated and combined with equation 1 to give:

$$\Delta h_{23} = \frac{L}{J}\left[1 + \frac{\pi k_{ins} J(T_3-T_1)}{\dot{m}\ln(r_o/r_i)_{ins}}\right] \quad (6)$$

The first term, $\frac{L}{J}$, (64.3 B/lb. for a 50,000 ft well) represents the enthalpy drop which is converted to potential energy from well bottom to top, in accordance with Eq. 1. In order to calculate the second term in the bracket, Δh_{23}, the type of insulation had to be chosen.

Many types of insulation were considered, as shown in Table II with thermal conductivities from .00057 to .35 Btu/hr-ft-^oF. The powders, opacified, evacuated and gas-filled had the desired quality of low thermal conductivity. However, it would be impractical to design for pressure as low as 10^{-4} mm Hg in an annular space surrounded by deep well pressures approaching 22,000 psi. Expanded perlite and mica also had low thermal conductivities but the required interspacial

TABLE II. INSULATION PROPERTIES

Descriptive Name	Approx. Density lb/ft^3	Thermal Conductivity $Btu/hr\text{-}ft\text{-}^{o}F$	Interspace Pressure mm Hg
Opacified Powder	7.0	.0015-.0004	10^{-4}
Evacuated Powder	6.0	.00057-.00115	10^{-4}
Gas-Filled Powder	6.0	.001-.004	760
Perlite, Expanded	8.1	.0007	.0001
Mica, Expanded	9.4	.00105	.0001
Powdered MgO		.044	
ZrO_2		.35	
Clear Fused Silica		.338	
J-M Min-K #1301	20.0	.0208 (at 600F)	
Motor Oil	50.0	.07	

pressures again made them impractical to use. Johns-Manville Min-K #1301 insulation, with a thermal conductivity of .0208 Btu/hr-ft-^{o}F appeared to be the most attractive initially. Properties of this insulation are shown in Table III.

Using this insulation and computerizing equation 6, Δh_{23} was calculated for insulation thicknesses of 1.0 and 1.5 in. and inside pipe diameters of 1.5 and 2.0 ft. As shown in Fig. 3, the second term Δh_{23} represents only a fraction of the total heat loss. However, it is also obvious that 1.5 in. of insulation and an inside pipe diameter of 1.5 ft will minimize this fractional loss.

Having set these design characteristics, the initial temperature distribution after the cold water in the riser has been removed, for a mass flow rate of 1.0 x 10^{6}lbm/hr, is shown in Fig 4. As seen later, both the riser and downer temperatures will drop with time as heat is removed from the immediately surrounding earth.

Due to other design problems encountered later it was soon clear that an acceptable insulation must either

(a) have high compressive strength, or

(b) retain its thermal resistance while immersed in high pressure water. Without these requirements being met, the inner and outer steel walls become prohibitively thick and heavy.

After extensive efforts to accommodate conventional dry insulation, using gas pressurization, bellows, etc. (described on page 20), Dr. W. C.

TABLE III. PHYSICAL AND THERMAL PROPERTIES OF JOHNS-MANSVILLE INSULATION, MIN-K #1301 [5]

Physical Property	Min-K 1301**
Maximum service temperature, F^{o}	1300
Nominal density, lbs./cu ft	20
Average transverse strength, psi	75
Compressive strength, psi	
5% compression	94.0
10% compression	200.0
Linear shrinkage, %	1
Thermal conductivity Btu/in/hr/sq ft/F	
200F mean	
300F	0.20
400F	0.21
600F	0.23
800F	0.25
1000F	0.27
Minimum thickness, in.	1/8
Thickness tolerance, in.	±1/32

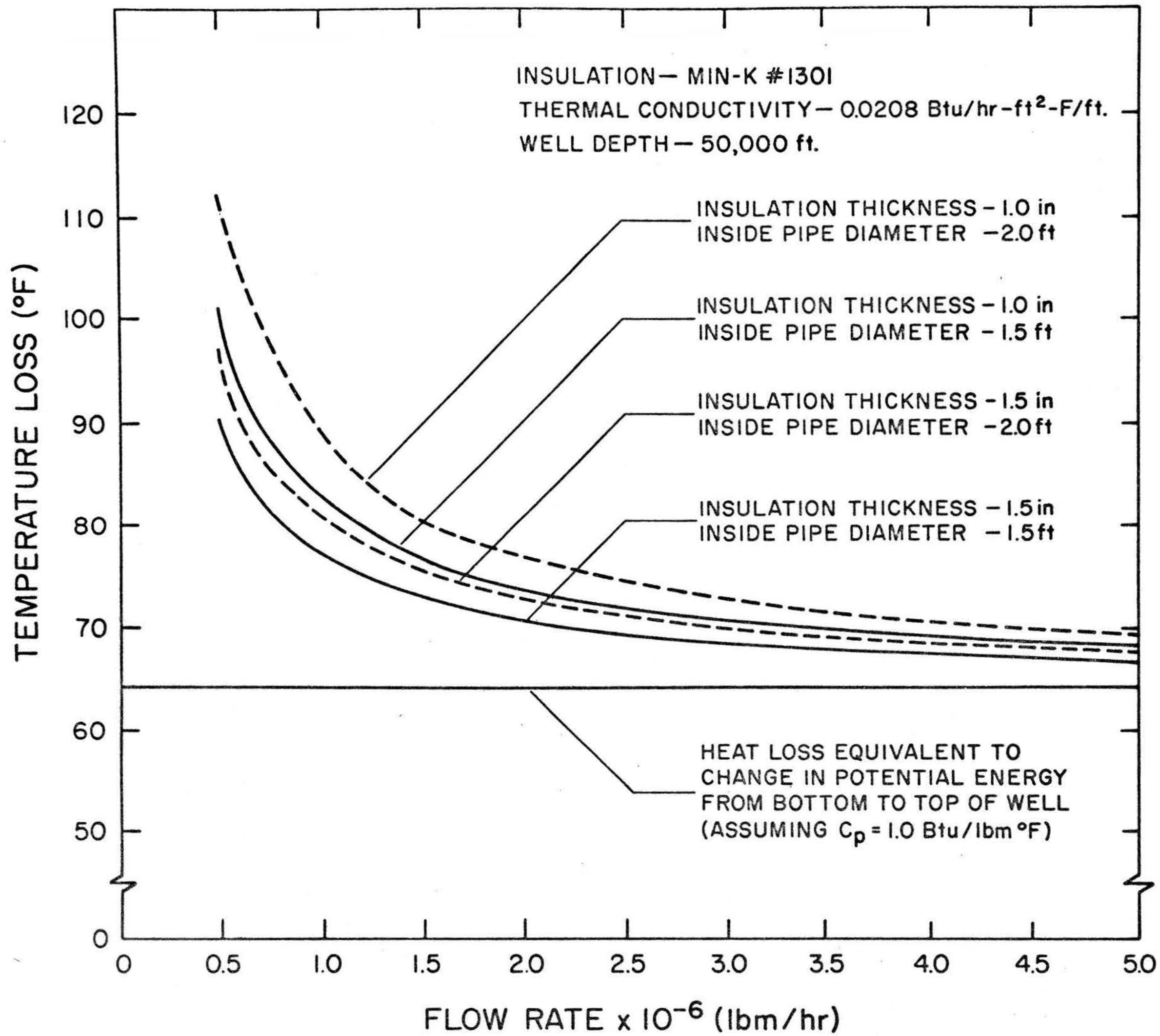


Figure 3. Riser Temperature Loss (ΔT_{23})

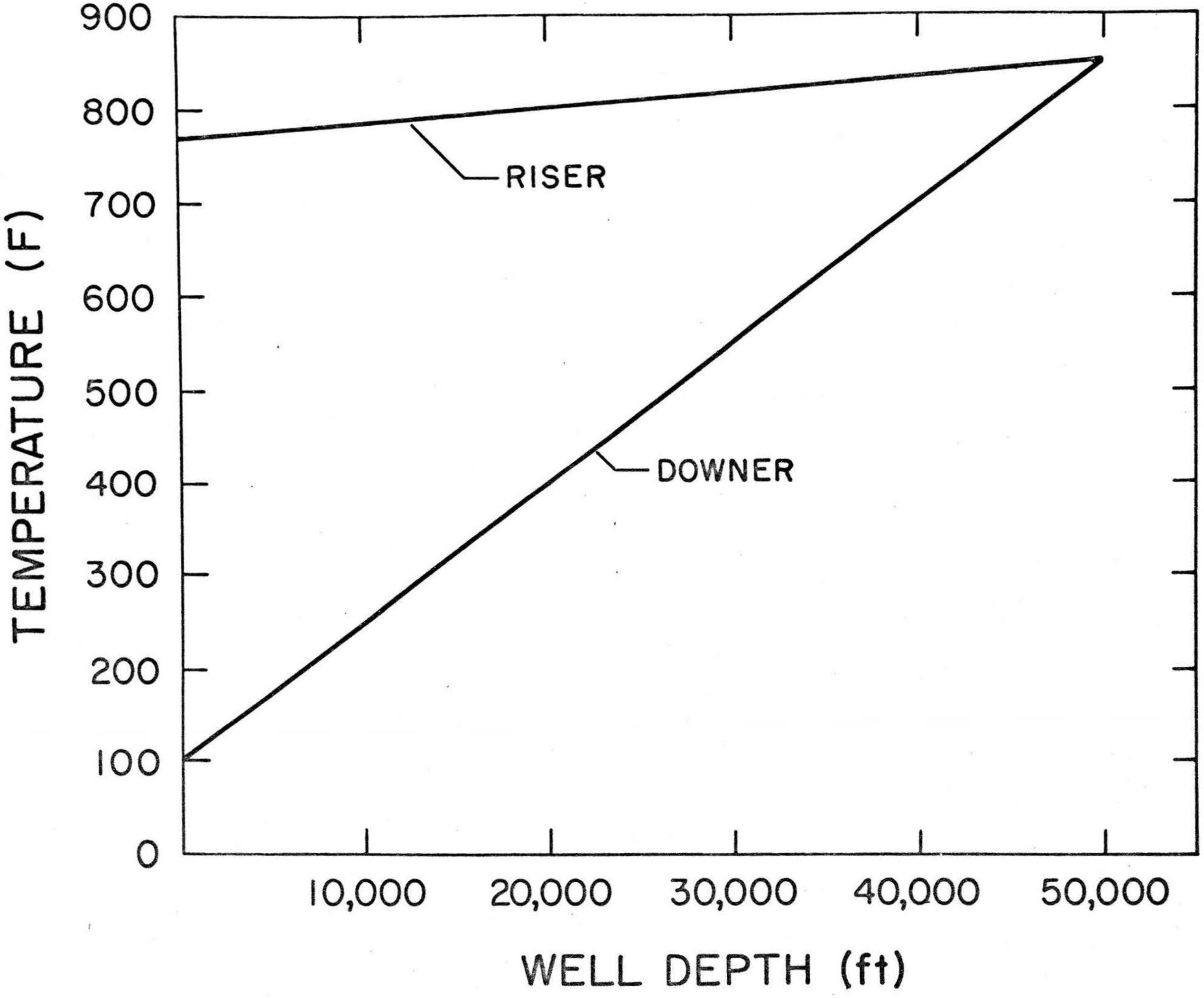


Figure 4. Initial Well Temperature Versus Depth

Thomas [4] suggested the use of common motor oil which has a thermal conductivity of 0.07 Btu/hr-ft-^{o}F. This appears to be the ideal insulation for this design as oil retains its thermal resistance while immersed in high pressure water. Also, due to its density being less than that of water, thermal expansion problems are accommodated easier as shown in the final design, Fig. 5.

Comparing the thermal conductivity of oil to that of J-M Min-K #1301, 5 in. of oil will make the riser temperature loss the same as that for 1.5 in. of J-M Min-K #1301 and the corresponding riser temperature loss will be approximately -70F.

Supporting the Central Pipe

Having determined that the central pipe is to be two coaxial steel liners with a 5 in. oil-filled annulus, the next design problem is how to suspend such a pipe to a depth of 50,000 ft.

If hung from the top of the well, with the oil annulus open at the bottom, the maximum stress at the suspension point of each liner would be $\sigma = L(\rho_{steel} - \rho_{\omega})$. Therefore, a steel having a yield point stress of 80,000 psi could support a pipe only 29,000 ft long.

If the oil annulus were closed at the bottom to reduce the load at the top by the buoyant force, the top stress would be reduced to 73,000 psi, but the buoyant force would be compression of 3,870,000 lb. acting at the bottom, with initial stresses of about 100,000 psi.

These considerations, plus the problems of installation and of differential thermal expansion led to the decision to employ a central

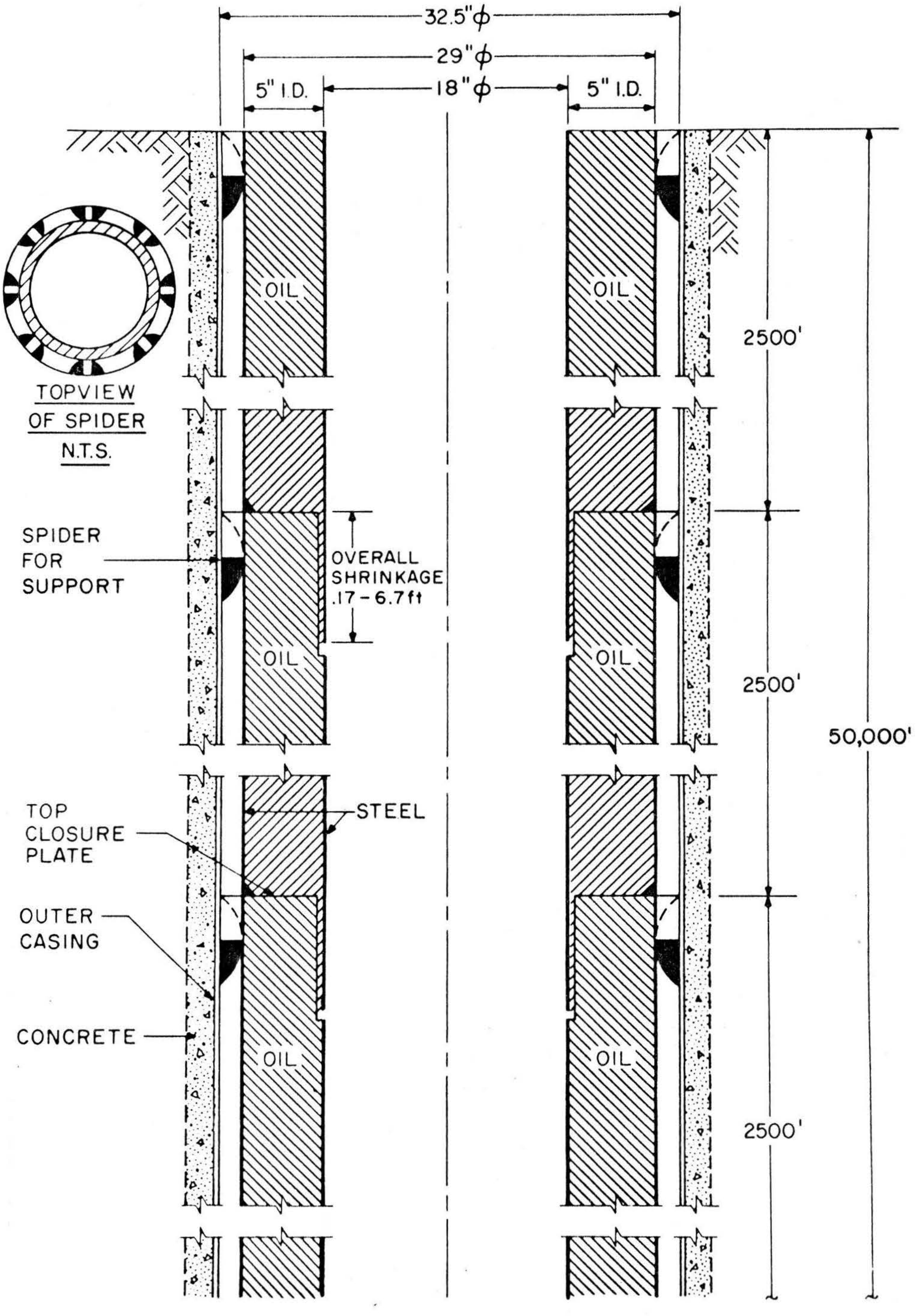


Figure 5. Proposed Well Design

pipe, made of twenty 2,500 ft lengths. Each 2,500 length is suspended by a "spider" to a flange in the outer casing wall, and mates with the previous length above it by a sliding joint on the inner lining, and a weld on the outer, for reasons to be described. This design detail is previously shown in Fig. 5.

Compensation for Differential Thermal Expansion Between the Central Pipe Liners

Perhaps the most difficult problem in devising a feasible well structure, was to accommodate the large differential expansion between the inner and outer liner of the central pipe, as soon as the well flow commences.

Consider a well initially filled with static water at the geothermal temperature gradient. As each successive 2,500 ft section of central pipe is lowered to its assigned depth, its two liners will expand as shown in Fig. 6, a length

$$\Delta L = \alpha_{th} \Delta T S$$

where

α_{th} = coefficient of thermal expansion, in./in.-degF

ΔT = temperature rise at level z, $\Delta T = \alpha_g z$

S = 2,500 ft.

As the well flow is initially established, the inner liner will be heated by the rising water to a value approaching the bottom temperature, T_2, while the outer liner temperature will slowly fall as the downer reduces the geothermal gradient. The result is a maximum

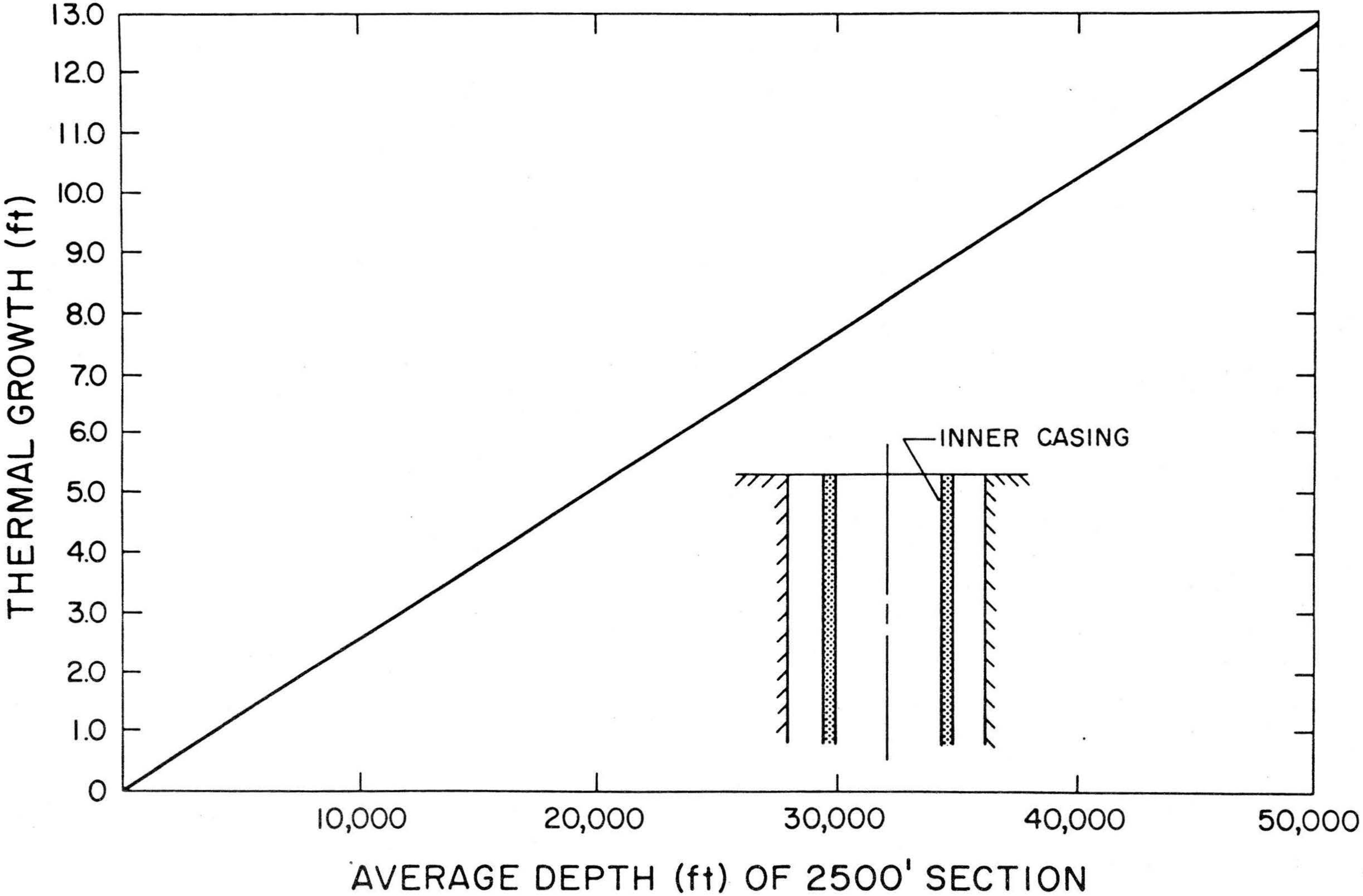


Figure 6. Casing Thermal Growth

differential expansion of 6.09 ft in the top 2,500 ft length, diminishing to about .17 ft in the lowest 2,500 ft length as the well is heated and cooled due to periodic operation. To design for this expansion several alternatives were considered before the idea of a motor oil insulation was thought of:

(a) supporting the pipe by a number of cables imbedded in the insulation between the inner and outer liners and "locking" the two liners to the same expansion by collars. This idea was dropped since it would produce excessive buckling in one liner;

(b) bellows installed on the inner inside pipe which would expand and contract as needed to allow for the thermal expansion of the inner inside pipe.

Two designs employing bellows are shown in Figs. 7 and 8. These designs seemed very promising until it was found that a 10 in. high bellows 18 in. in diameter would expand approximately 25% of its length. Therefore, for 122 ft of expansion approximately 600, 10 in. bellows at an estimated cost of $500-600 each would be required. Although $300,000 for bellows alone would be required, cost was not the principal reason for disregarding these designs. The problem was that at the bottom of the well, pressures of approximately 22,000 psi would be experienced and according to the bellows manufacturer [6] a bellows of this size could only stand a 200 psi pressure difference across its walls. The interspace between the inner and outer inside pipes would therefore have had to be pressurized and regulated to within ± 200 psi

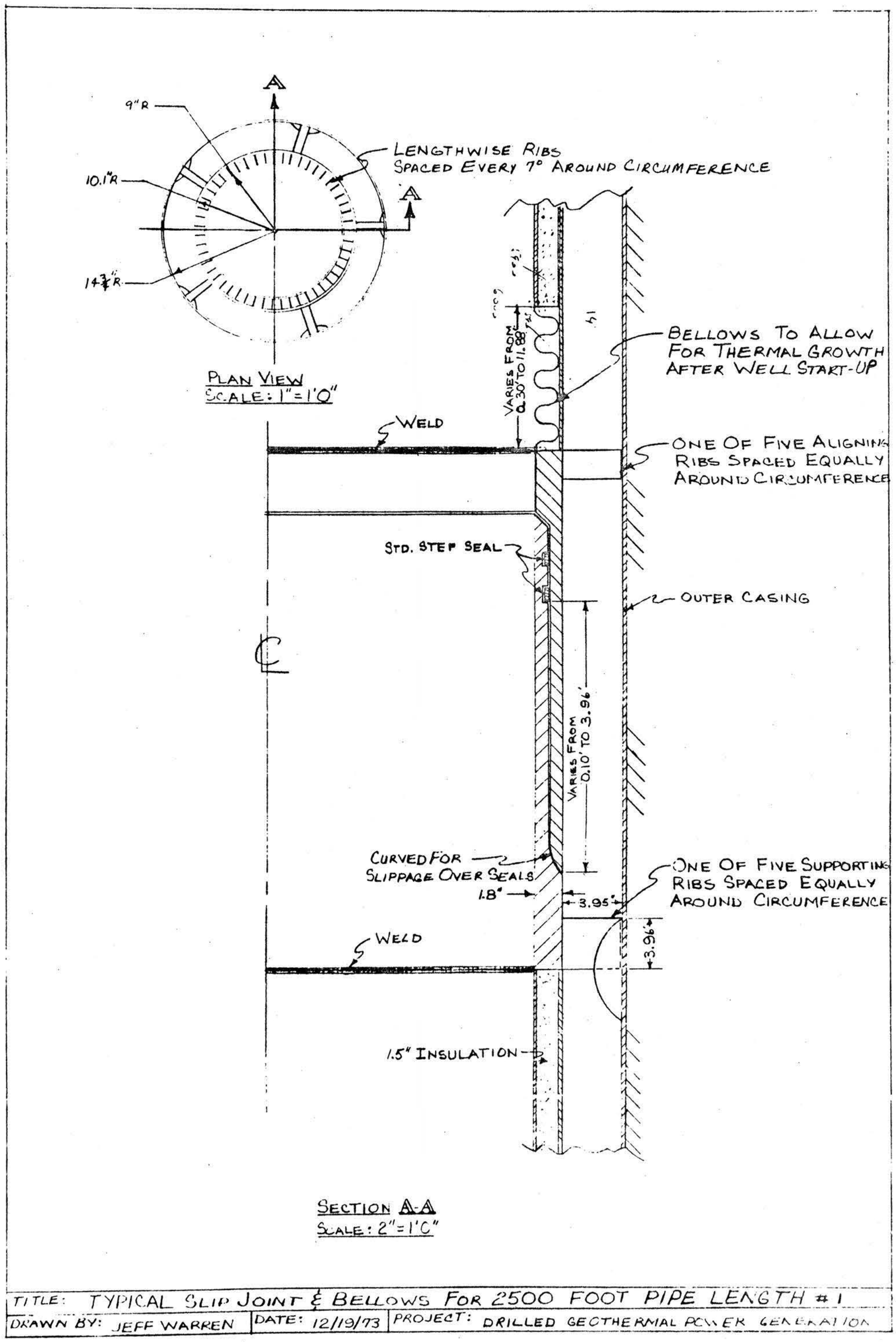


Figure 7. Well Design Using Bellows, Proposal #1

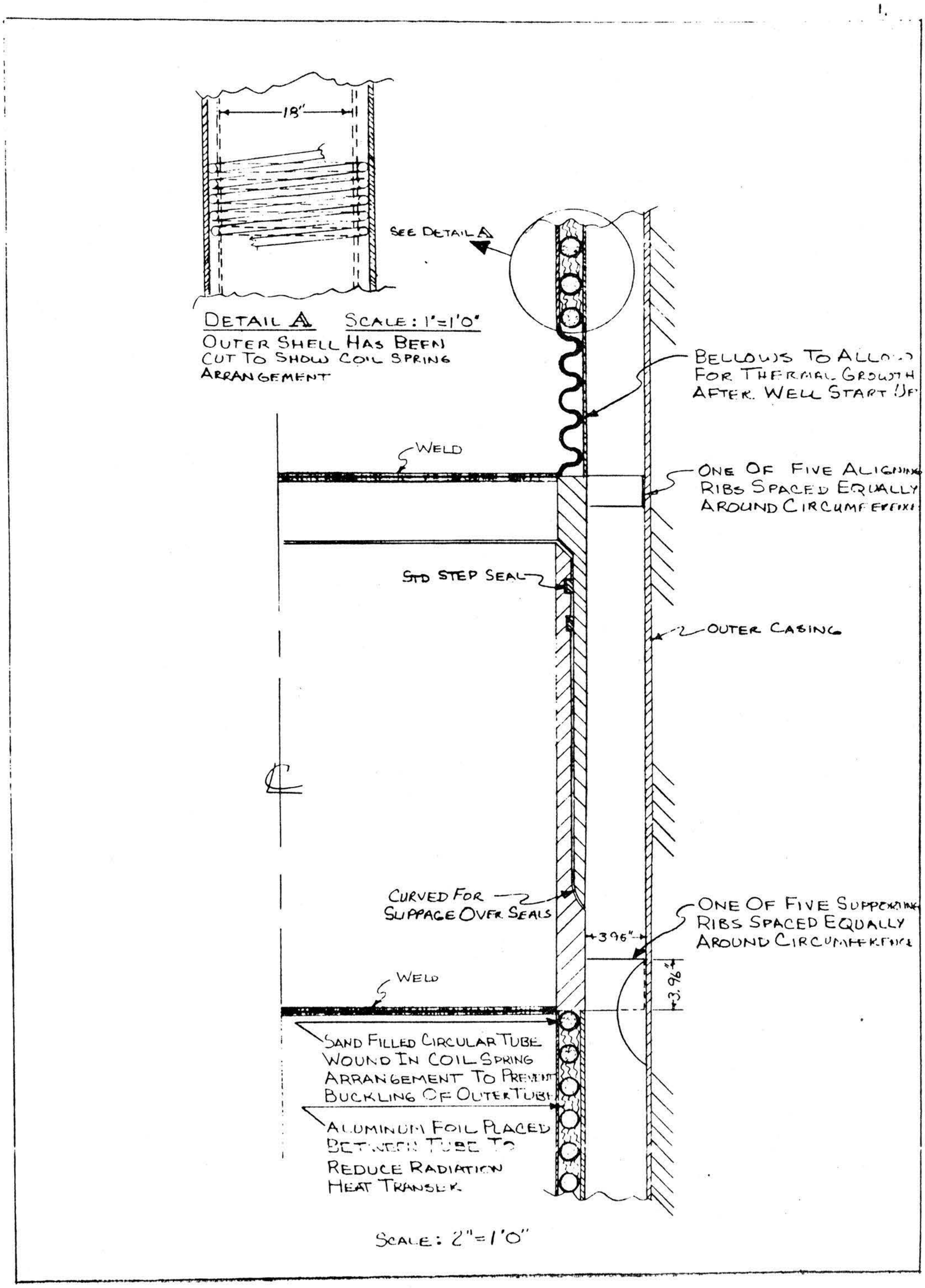


Figure 8. Well Design Using Bellows, Proposal #2

of the well pressure at all times. Pressure equalizing accumulators [7] to be installed in the inner pipe were considered to continuously monitor and correct well pressure changes, but space and pressure limitations ruled them out.

Another solution considered was a series of small pressurizing pipes to the surface, one from each 2,500 ft section and connected to a series of pumps each regulated by pressure sensors in the well. But any pump failure would result in a bellows failure and a plant shutdown.

Selection of Area Ratio, (A_{riser}/A_{downer})

This ratio would normally be chosen to give approximately equal pressure drop in riser and downer, where the frictional pressure drop in each defined as the pressure at the top minus the pressure at the bottom, and is given by:

$$\Delta p_f = \pm \int_0^L f \frac{dz}{d_e} \left(\frac{\rho_\omega V^2}{2g}\right) \qquad \begin{matrix} + \text{ for downer} \\ - \text{ for riser} \end{matrix} \tag{7}$$

The downer, being an annulus and with a lower average temperature would have a smaller value of d_e, a smaller Reynolds Number, and, therefore, would require a lower velocity to keep the two pressure drops equal. However, since the downer is the heat transfer channel, and it is also desirable to maximize the natural circulation "driving head" of the well, it can be shown that equal velocity in downer and riser is close to optimum design [8]. We then have,

$$(A_r/A_d) = (\bar{\rho}V)_d/(\bar{\rho}V)_r = (\bar{\rho}_d/\bar{\rho}_r) \quad (8)$$

For initial operation, with T_3 = 700F, the integrated average density in the downer is about 50 lb./ft^3, and in riser in 33 lb.ft^3, giving $(\bar{\rho}_d/\bar{\rho}_r) = 1.5$. Using this area ratio $(A_r/A_d) = 1.5$ and a previously determined inside pipe diameter of 18 in., 5 in. of oil insulation, and pipe liner of an average thickness of 0.25 in., the outer casing diameter was set at 32.5 in. The outer inside pipe diameter D_o was also set at 29.0 in.

With the decision to go to motor oil as the insulation, thermal expansion is easily accommodated by letting the bottom of the inner pipe of one 2,500 ft section slip over the top of the next 2,500 ft section. Since the oil is lighter than the water no seals are necessary as the oil will always rise to the top of the annular interspace and will always be in pressure equilibrium with the inner water. Also due to the fact that the lower temperature water in the outside downer will always be at a lower pressure than the rising hot water, the outer liner will always be in tension eliminating any buckling problems.

Coaxial Pipe Installation

The final part of the design was to determine a feasible installation for the ten mile long coaxial pipe. As previously stated, the outer well casing with the female spider supports welded every 2,500 ft is assumed to be installed. The final problem therefore, is to assemble and install the inner and outer liner in the water filled well and fill the annular interspace with an oil insulation.

The proposed installation technique consists of six steps shown in Figs. 9a to 9f. In Step 1, 100 ft pipe lengths would be delivered to the site by rail or truck. In Step 2, a specially designed clamp will be used to hold the pipe so that welding of two 100 ft lengths can be accomplished. Note again in Step 3, that the female spider support is already welded to the outer casing. Step 4, is self-explanatory. In Step 5, a remote controlled underwater welder is shown. This would have to be specially designed to stand high pressure and temperature. It is the author's feeling that this type of welder would be supported from the top by cables and would have wheels that would run against the outer or inner casing to help in the aligning process. The top closure plate shown in Fig. 9e is used to support the inner liner. Buoyant forces of the oil insulation, reduce the size of the plate considerably. Finally in Step 6, the hose used to pump in the oil insulation would have an end on it, so designed that it could be inserted as shown and permit easy installation of the oil insulation.

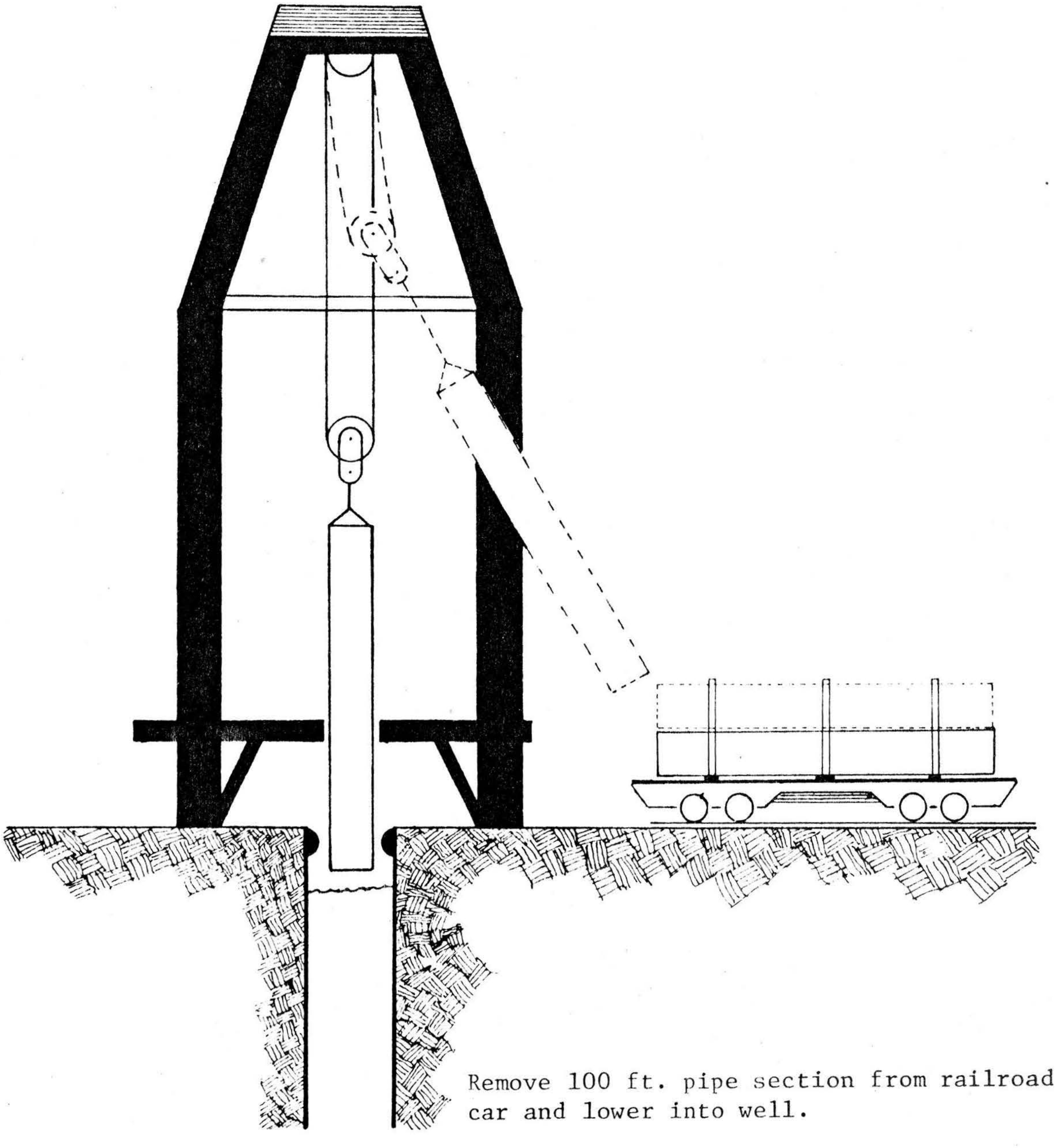
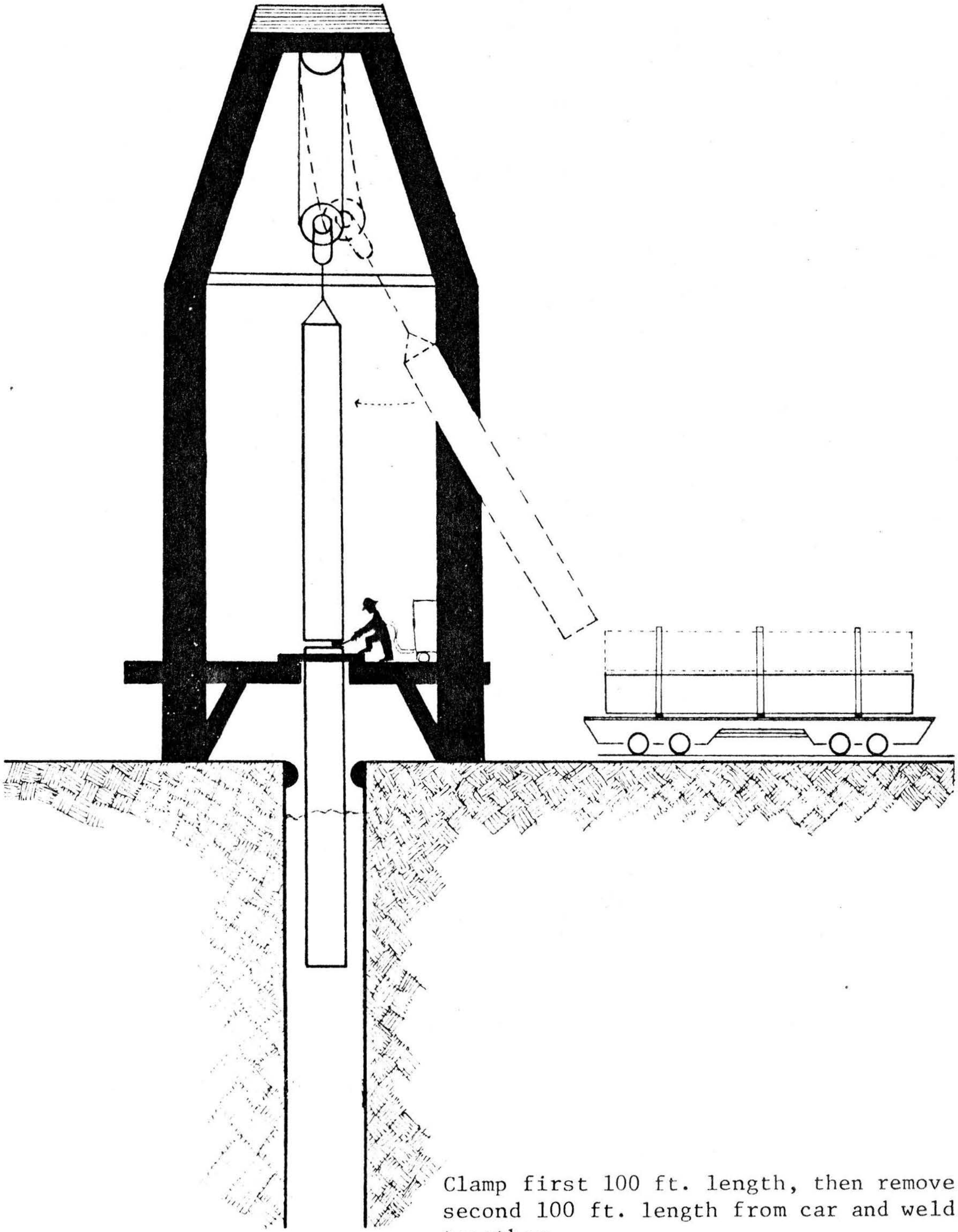


Figure 9a Coaxial Pipe Installation-Step 1



Clamp first 100 ft. length, then remove second 100 ft. length from car and weld together.

Figure 9b Coaxial Pipe Installation-Step 2

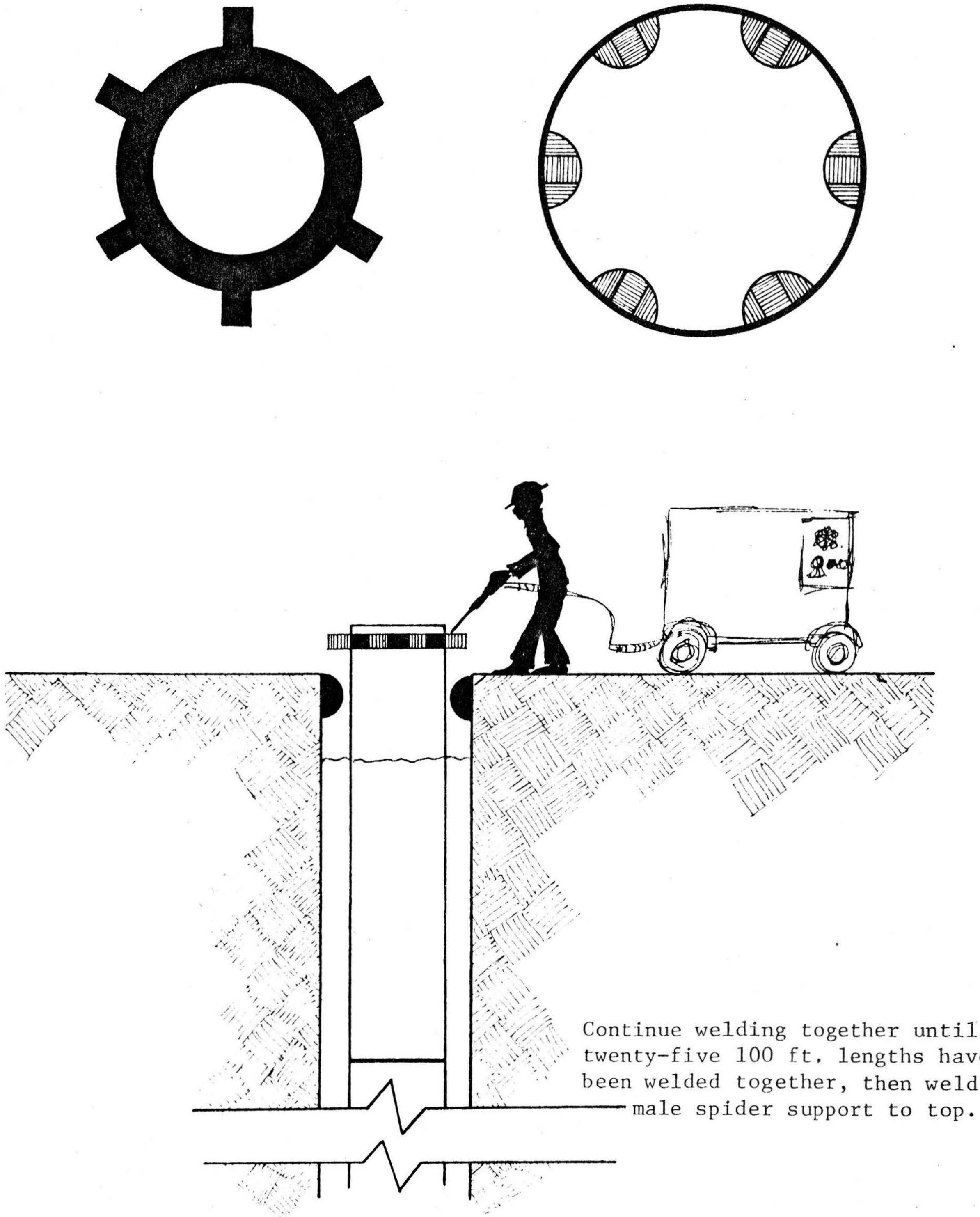
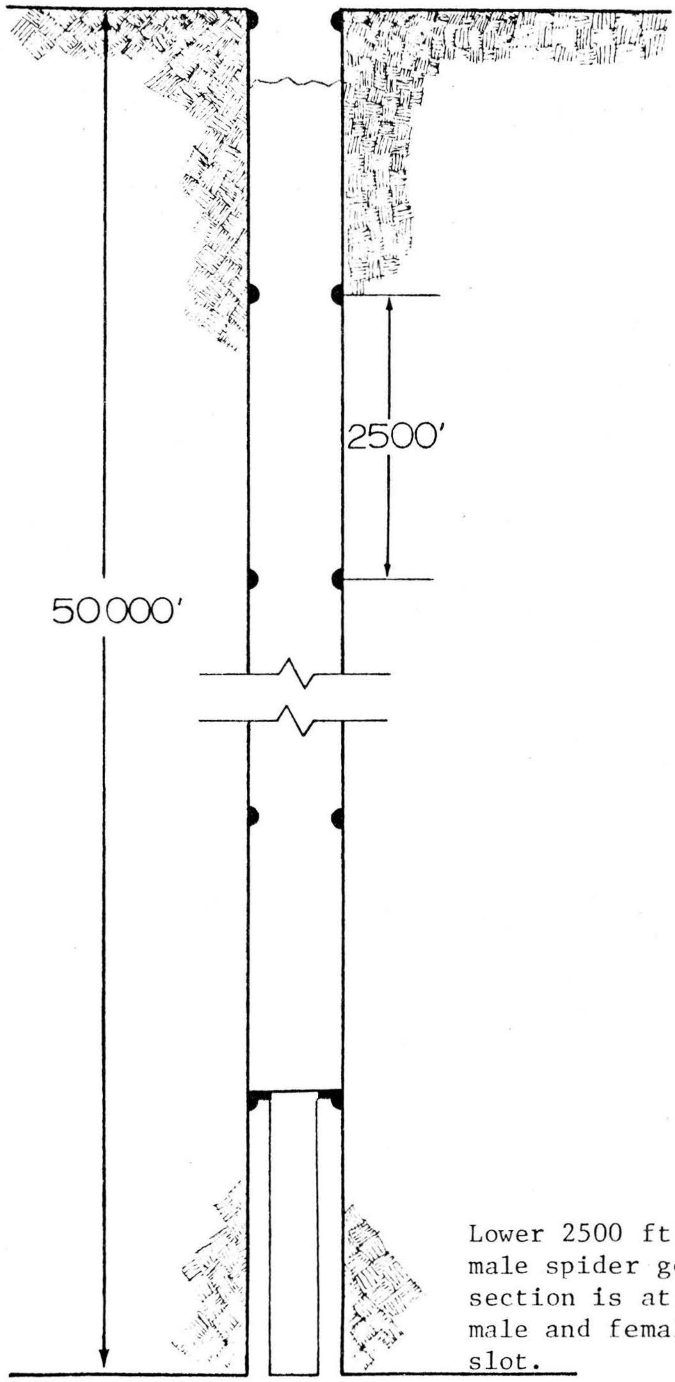
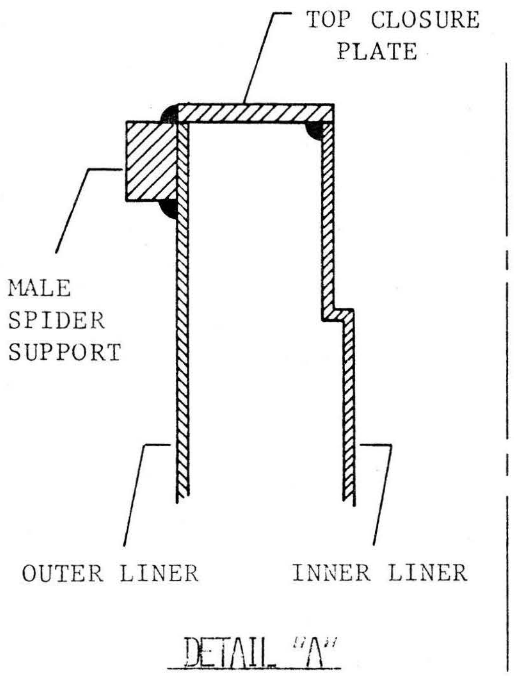


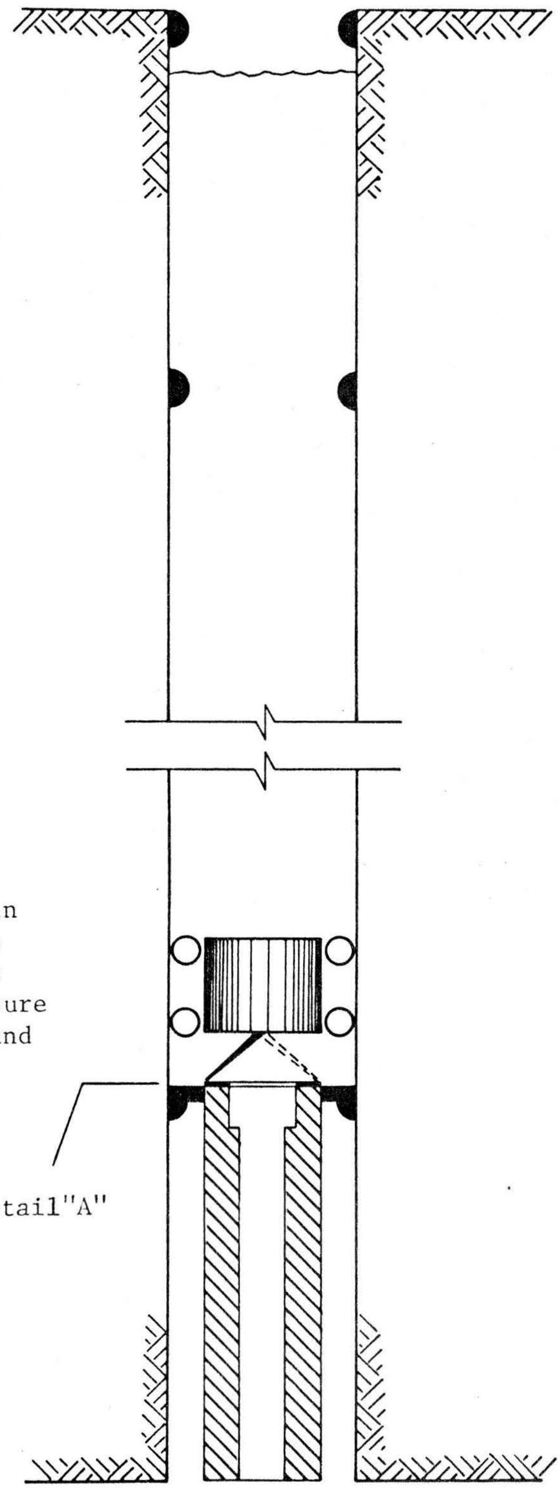
Figure 9c Coaxial Pipe Installation-Step 3



Lower 2500 ft. section into well by letting male spider go between female spider. When section is at bottom of well, turn so that male and female spider align and drop into slot.

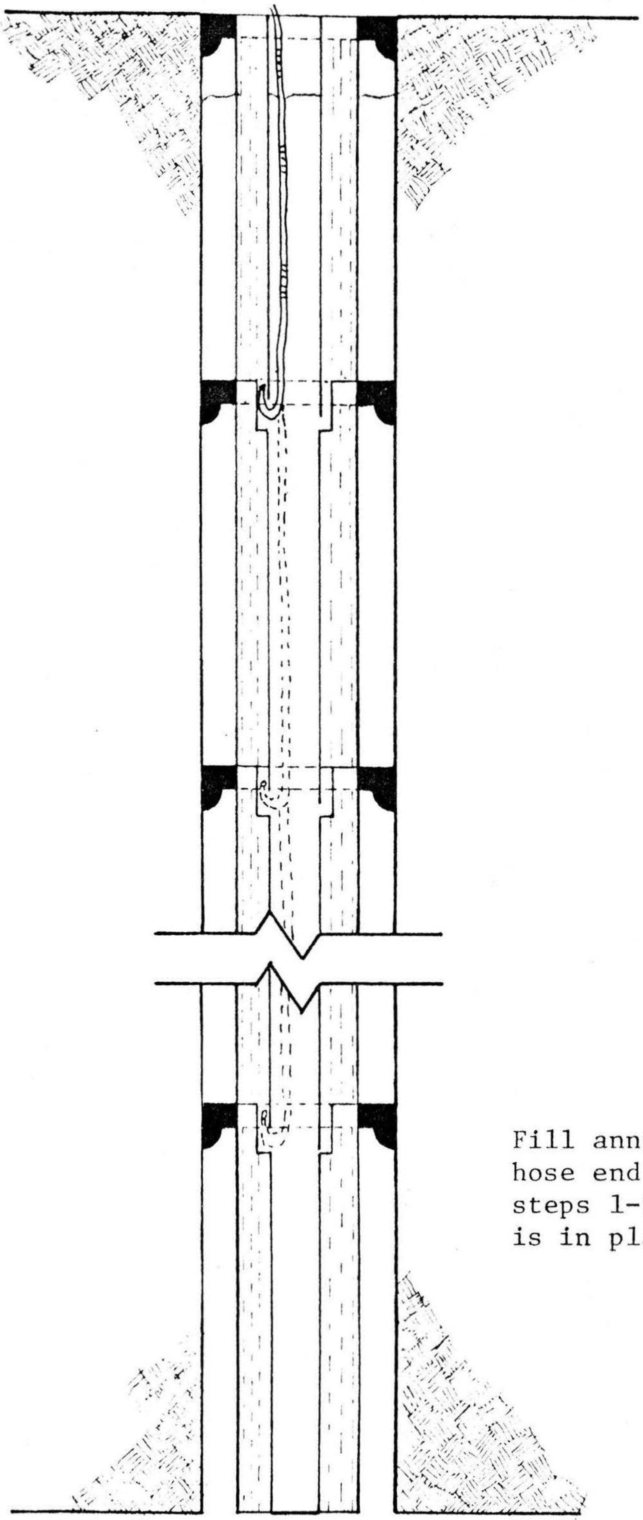
Figure 9d Coaxial Pipe Installation-Step 4





Assemble and lower inner liner in 100 ft. lengths similar to steps 1 and 2 until 2500 ft. length is constructed. Then weld top closure plate, lower to bottom of well and weld into place.

Figure 9e Coaxial Pipe Installation-Step 5



Fill annulus with oil using special hose end as shown, then repeat steps 1-5 until entire coaxial pipe is in place.

Figure 9f Coaxial Pipe Installation-Step 6

II. OPTIMUM THERMODYNAMIC CYCLE ANALYSIS OF THE ABOVEGROUND POWER GENERATION SYSTEM

Optimization of the aboveground powerplant cycle for the conversion of the hot water from the well into electrical power, was the second goal of the research. As previously shown in Fig. 1, the surface equipment included a once-through steam generator, condensor, circulating pumps and a feed-water heater system. A schematic diagram of this equipment is shown in Fig. 10 and a graph of temperature versus the percentage of heat transferred throughout the system is given in Fig. 11 for a three stage cycle.

Using the characteristic regenerative correction factors shown in Fig. 12, the feedwater temperature entering each section and the regenerative heat rate for each stage in Btu/Kwhr were determined. Using these factors the total thermal efficiency of the system was determined as follows:

$$\eta = \frac{\Sigma \text{ Power (steam turb,+hydr turb.-pumps)}}{\text{Heat Supplied (to steam generator)}} \tag{1}$$

$$= \frac{P_a + P_b + P_c + P_{ht} - P_{bfp}}{Q_g} \tag{2}$$

We have

$$Q_g = \dot{m}_p(h_{p3}-h_{p6}) = Q_a+Q_b+Q_c$$

$$= \dot{m}_a(h_{4a}-h_{2a}) + \dot{m}_b(h_{4b}-h_{2b}) + \dot{m}_c(h_{4c}-h_{2c}) \tag{3}$$

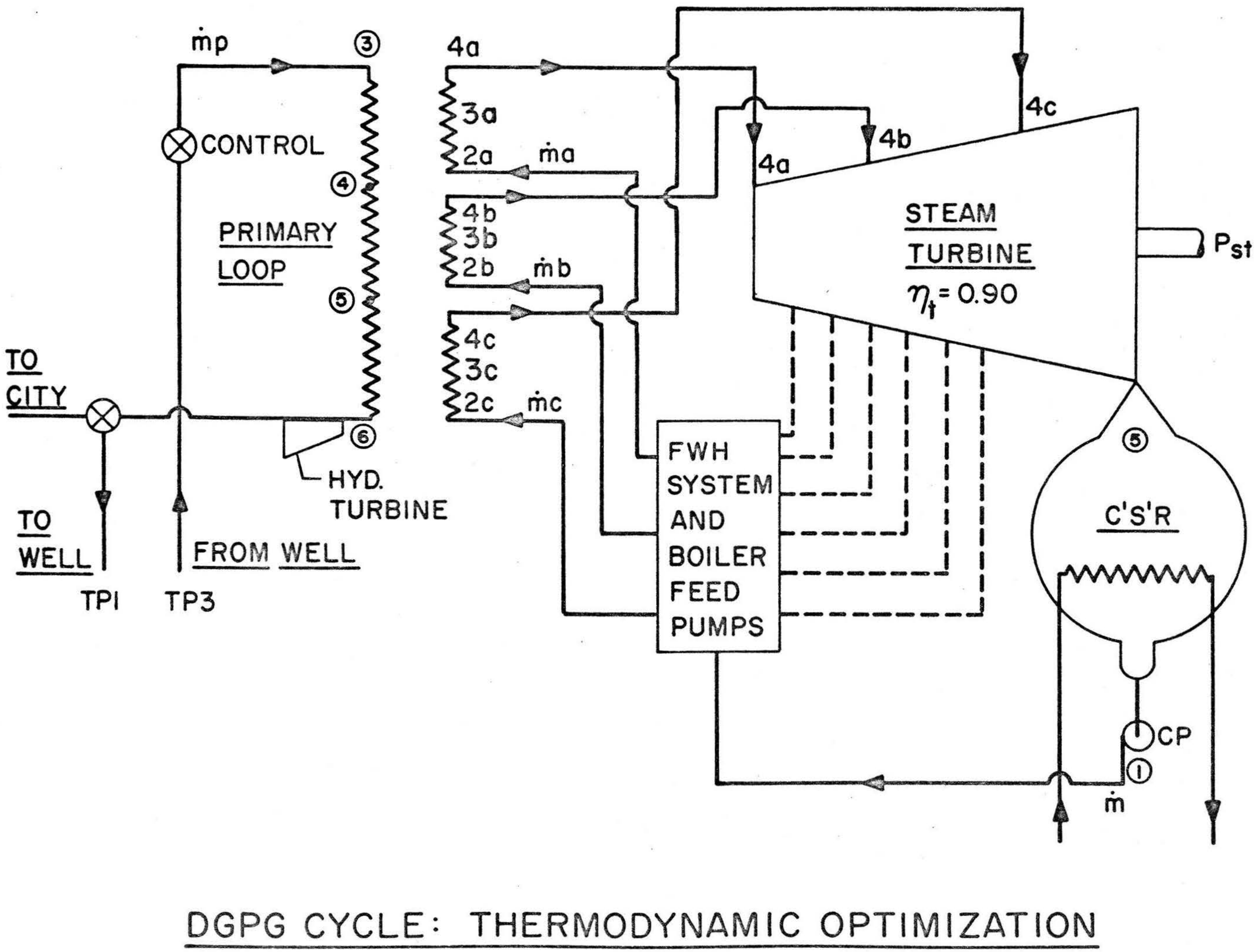


Figure 10. Schematic Diagram of the Power Generation System

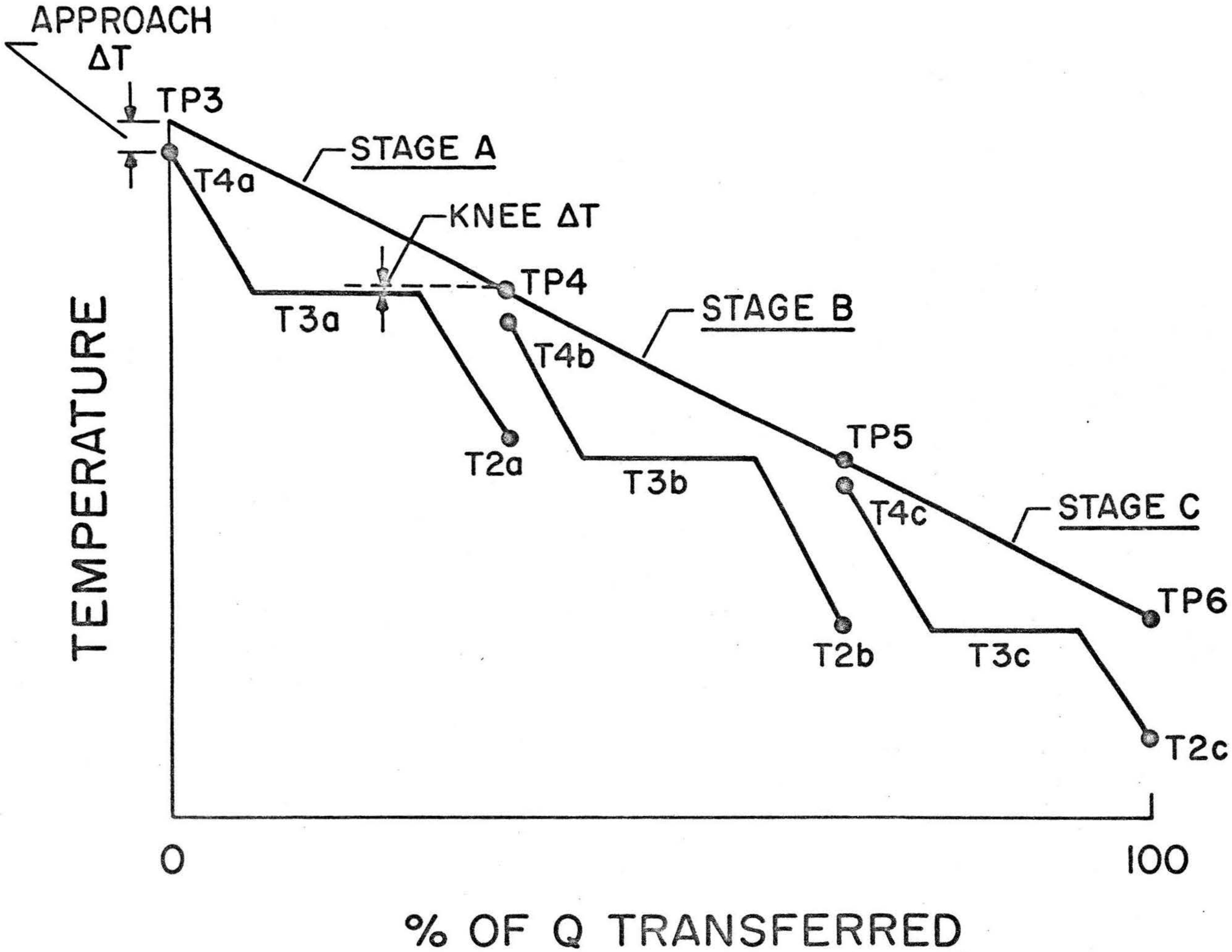


Figure 11. Temperature Versus Percent of Heat Transferred

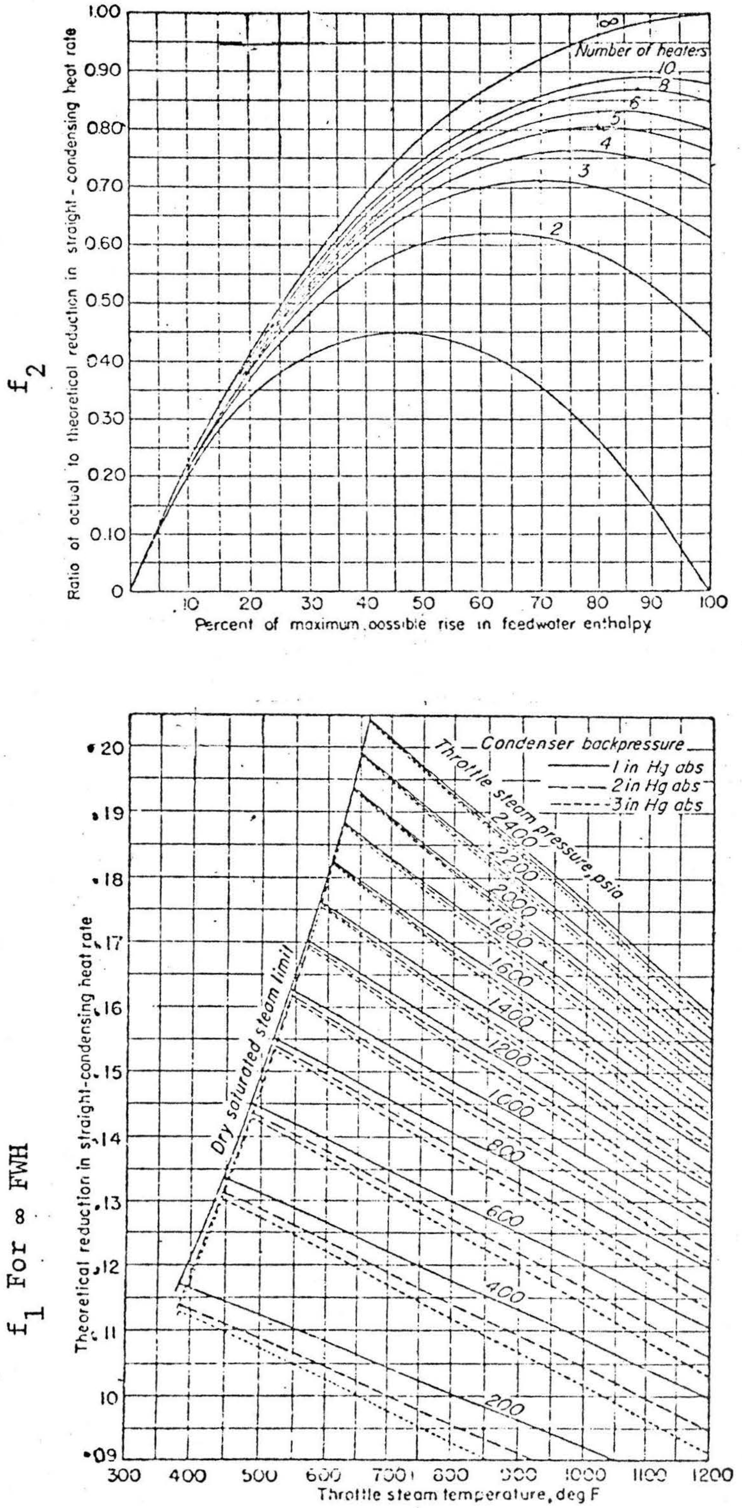


Figure 12. Correction Factors for Regenerative Heat Rate [9]

where the subscripts identify the points in Fig. 11.

For each stage of the steam generator:

$$\begin{aligned}
\dot{m}_p(h_{p3}-h_{p4}) &= \dot{m}_a(h_{4a}-h_{2a}) \\
\dot{m}_p(h_{p4}-h_{p5}) &= \dot{m}_b(h_{4b}-h_{2b}) \qquad (4) \\
\dot{m}_p(h_{p5}-h_{p6}) &= \dot{m}_c(h_{4c}-h_{2c})
\end{aligned}$$

To determine the feedwater entry conditions:

$\dfrac{(h_{2a}-h_1)}{h_{3a}-h_1}$ is set by the peak of f_2 curve, (Fig. 12), for the chosen number of feedwater heaters.

To determine the steam turbine power in a regenerative cycle:

$$P = Q/(HR)_{reg} = [Q_a/(HR)_a]_{reg} + [Q_b/(HR)_b]_{reg} + [Q_c/(HR)_c]_{reg} \qquad (5)$$

Also

$$[HR_a]_{reg} = [HR_a]_{n-r}(1-f_1f_2)_a = \dot{m}_a(h_{4a}-h_{2a})3413/P_a \qquad (6)$$

and

$$[HR_a]_{n-r} = 3413(h_{4a}-h_1)/(h_{4a}-h_5) \qquad (7)$$

so that

$$P_a = \dot{m}_a\{(h_{4a}-h_{2a})(h_{4a}-h_5)/[(1-f_1f_2)_a(h_{4a}-h_1)]\} \qquad (8)$$

putting

$$\phi_a = \{(h_{4a}-h_{2a})(h_{4a}-h_5)/[(1-f_1f_2)_a(h_{4a}-h_1)]\}; \tag{9}$$

combining (1), (2), (3), (8), (9):

$$\eta = \frac{\frac{\dot{m}_a}{\dot{m}_p}\cdot\phi_a + \frac{\dot{m}_b}{\dot{m}_p}\cdot\phi_b + \frac{\dot{m}_c}{\dot{m}_p}\cdot\phi_c}{(h_{p3}-h_{p6})} \tag{10}$$

and TP4,TP5 are established by:

$$(TP3-TP4) = (TP4-TP5) = (TP5-TP6) = \frac{1}{3}(TP3-TP6) \tag{11}$$

The above equations were computerized and an optimization was made using the conditions shown in Table IV. An approach ΔT or temperature difference, was defined as (TP3-T4A) = (TP4-T4B) = (TP5-T4C) and was varied in the optimization from 10, 20, 30F. A knee ΔT or temperature difference was also defined as (TP4-T3A) = (TP5-T3B) = (TP6-T3C) and was varied in the optimization from +10, 0, -10F. The heat exchanger entry temperature (TP3) and exit temperature (TP6) were also varied as shown in Table IV. Computer outputs for the optimization are given in Appendix A and graphical results are given in Figs. 13-17.

In Fig. 13 the effect of wellhead temperature (TP3) versus the thermal efficiency of the surface equipment is shown. As seen here an increase in the wellhead temperature yields an increase in the thermal efficiency. Fig. 14 shows that as the knee temperature difference becomes smaller the thermal efficiency increases. An increase in the

TABLE IV. OPERATING CONDITIONS INVESTIGATED FOR THERMODYNAMIC CYCLE OPTIMIZATION*

Primary Loop	Units		
Mass Flow Rate ($\dot{m}_p$)	lbm/hr	1.0×10^6	constant
Heat Exchanger Entry Temperature (TP3)	F	200F→600F every 50F 600F→900F every 100F	
Heat Exchanger Exit Temperature (TP6)	F	200, 300	
Well Entry Temperature (TP1)	F	100	
Pressure Entering Heat Exchanger (PP3)	PSIA	4,000	

Secondary Loop		
Condensor Pressure (P5)	PSIA	0.7
Temperature Leaving Condensor (T_1)	F	90
Approach Temperature Difference	F	10, 20, 30
"Knee" Temperature Difference	F	+10, 0, -10

Efficiencies		
Steam Turbine		.90
Hydraulic Turbine		.90

Feedwater Heating System			
3 Stage Cycle	NA	NB	NC
At TP6 = 200F	8	4	1
At TP6 = 300F	8	5	3
2 Stage Cycle			
Number of Heaters per Stage			
At TP6 = 200F	8	4	-
At TP6 = 300F	8	5	-

*All combinations of variables shown were investigated.

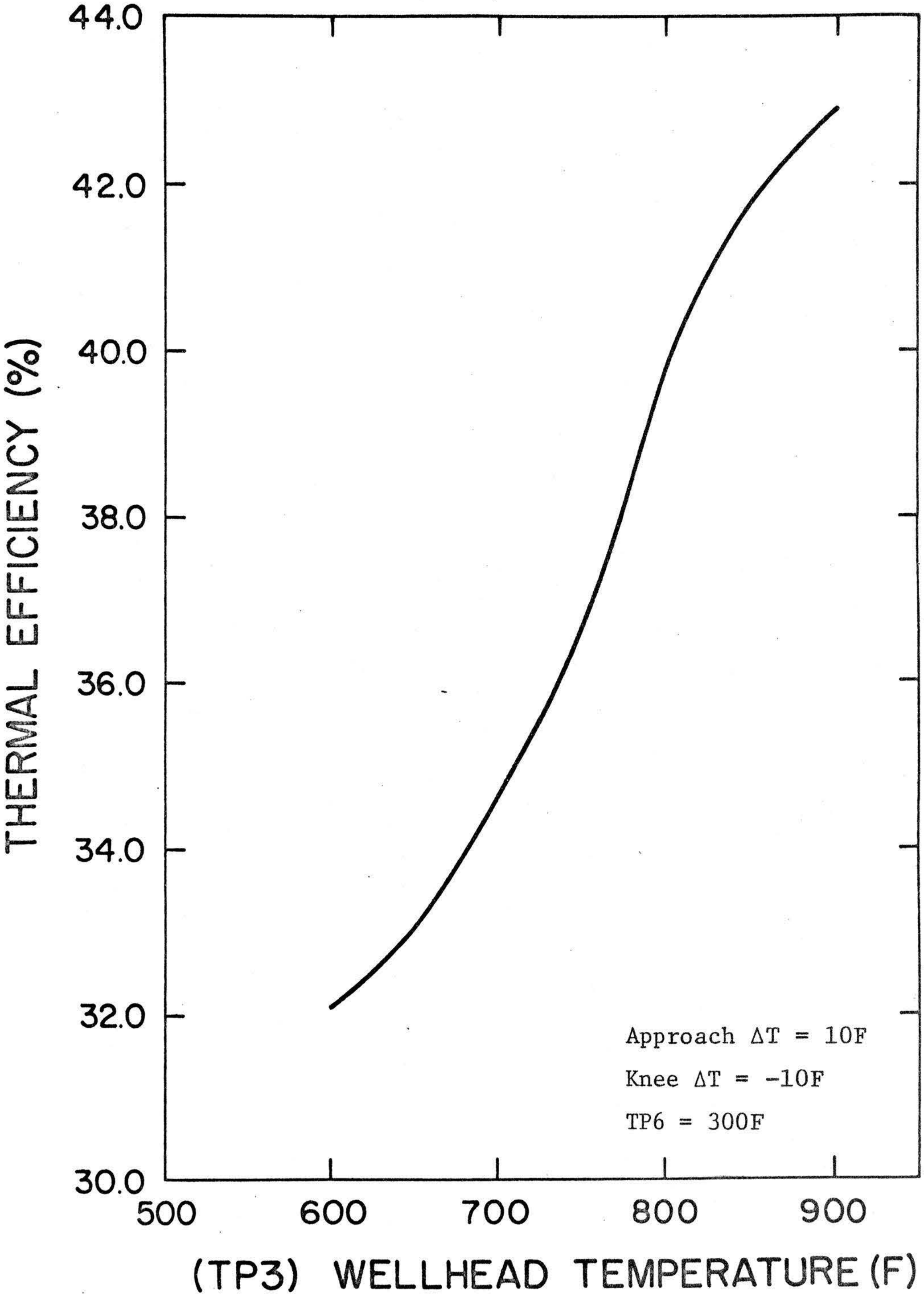


Figure 13. Thermal Efficiency Versus Wellhead Temperature for a Three Stage Cycle

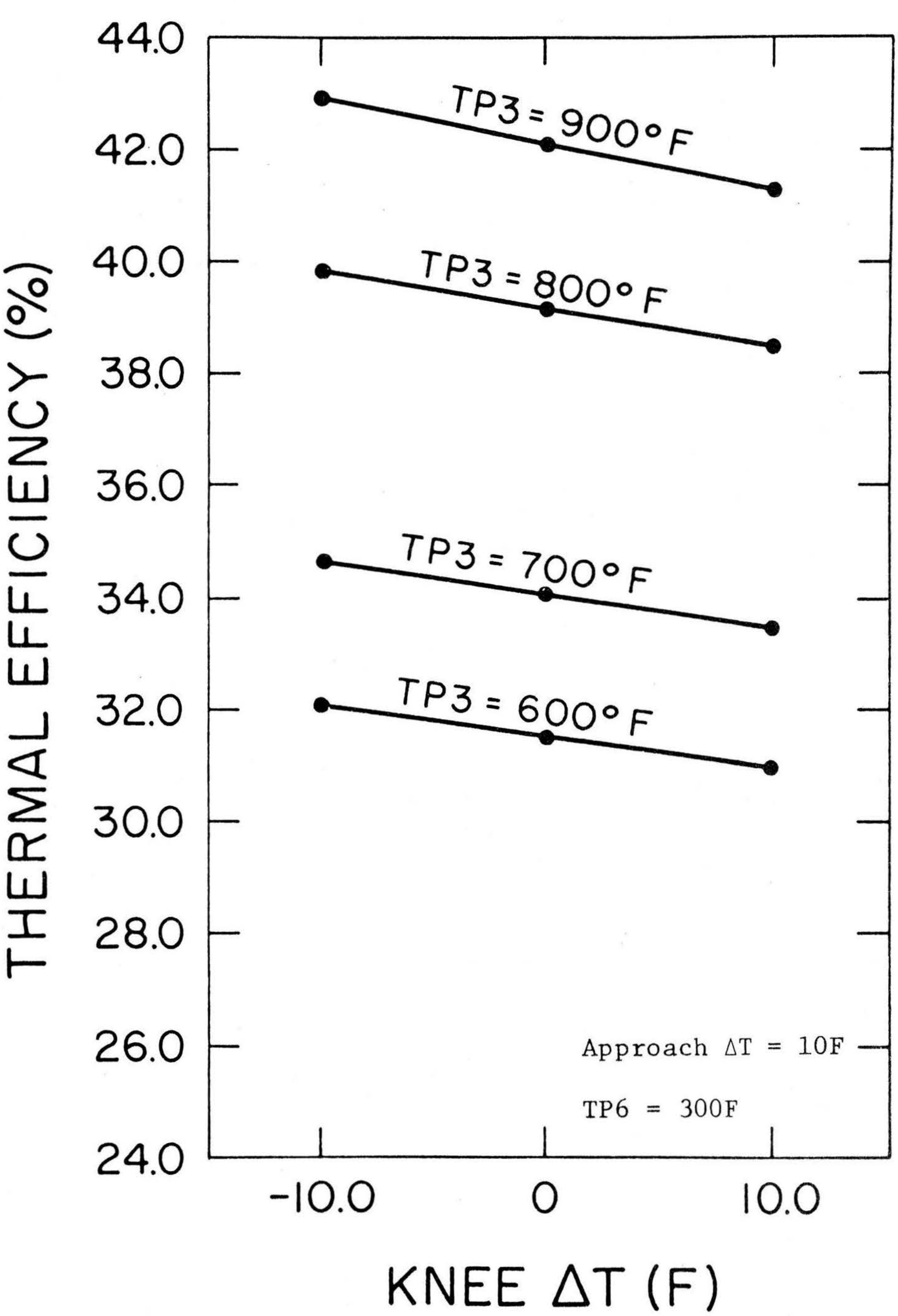


Figure 14. Thermal Efficiency Versus Knee ΔT for a Three Stage Cycle

approach temperature had a very small decreasing effect on the efficiency and consequently no graph of this is shown. Fig. 15 illustrates that as the heat exchanger exit temperature is increased there is an appreciable increase in the thermal efficiency of the system. Fig. 16 shows the effect of a two and three stage cycle on the thermal efficiency of the system. As seen from this graph, for every wellhead temperature a three stage cycle has the higher efficiency.

Having completed this optimization, good engineering practice and economics were used to determine the design conditions to be used in optimizing the well operating conditions. The variables to be considered were again the approach ΔT, knee ΔT, number of stages, and TP6, the heat exchanger exit temperature. Due to Fink's results [3], which indicated a wellhead temperature TP3, approaching 300F and lower after any length of running time, the heat exchanger exit temperature, TP6, was set at 200F. Since the length and cost of the heat exchanger approaches infinity as the approach ΔT approaches 0.0F, an approach ΔT was arbitrarily set at 20.0F to reduce costs. The defined knee ΔT was set at 0.0F since the actual knee temperature difference was approximately 10.0F and a three stage cycle was chosen over two stages as it was felt that this would appreciably increase the efficiency more than the cost. Having chosen these designs characteristics for the surface equipment, Fig. 17 was produced and these thermal efficiencies were used to determine the optimum operating conditions for the least expensive geothermal power in the third and final goal of the research.

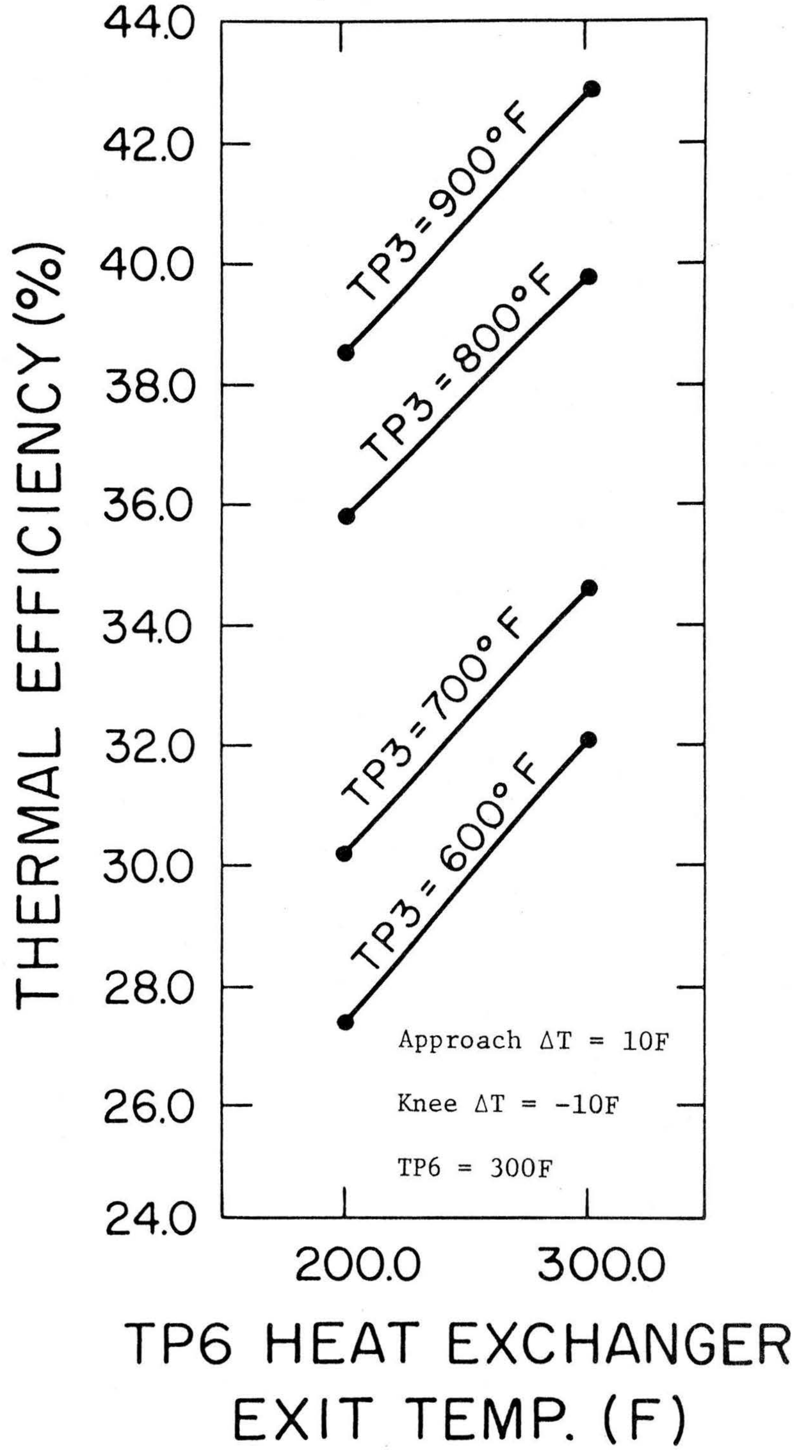


Figure 15. Thermal Efficiency Versus Heat Exchanger Exit Temperature for a Three Stage Cycle

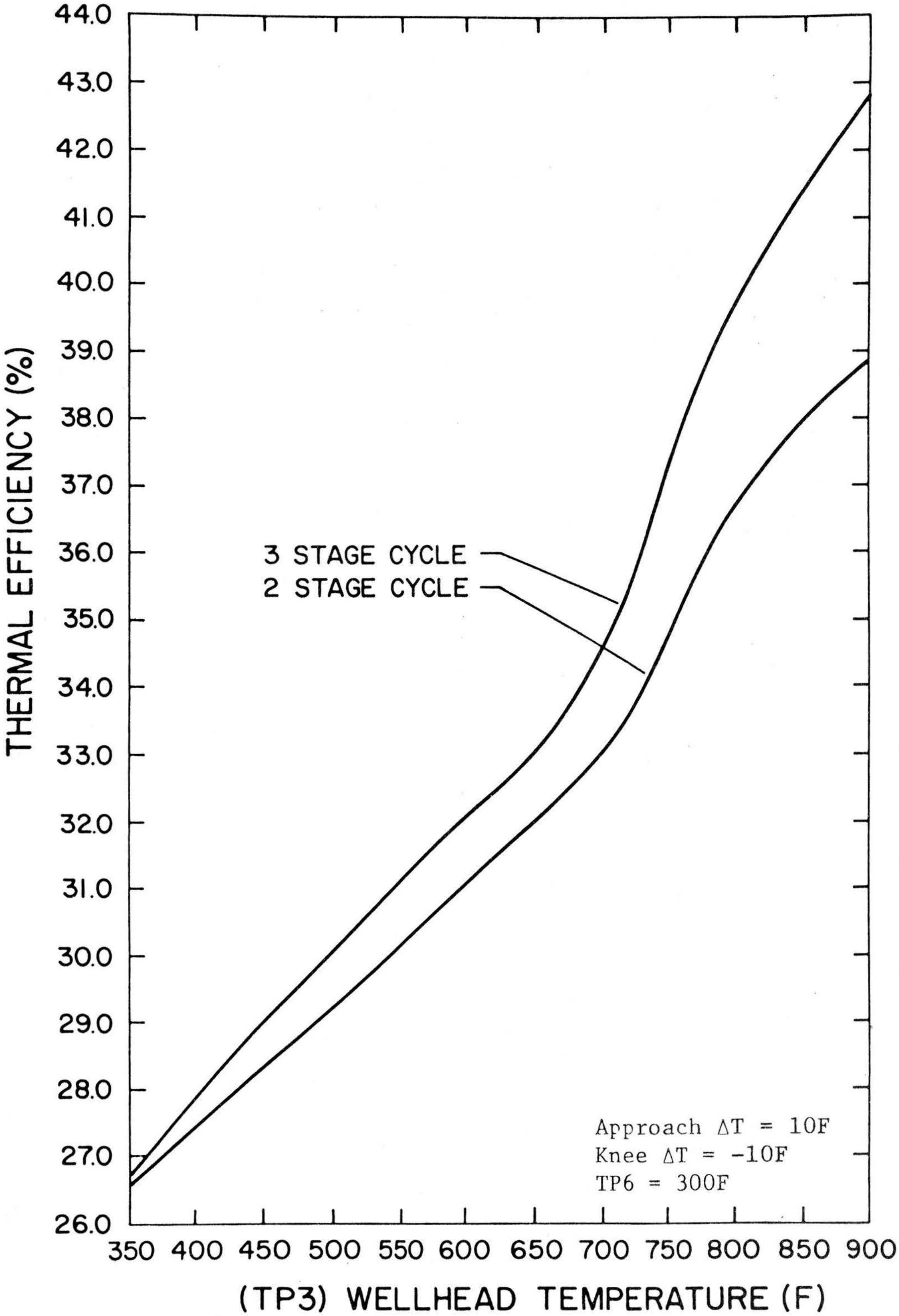


Figure 16. Thermal Efficiency Versus Wellhead Temperature for a Two and Three Stage Cycle

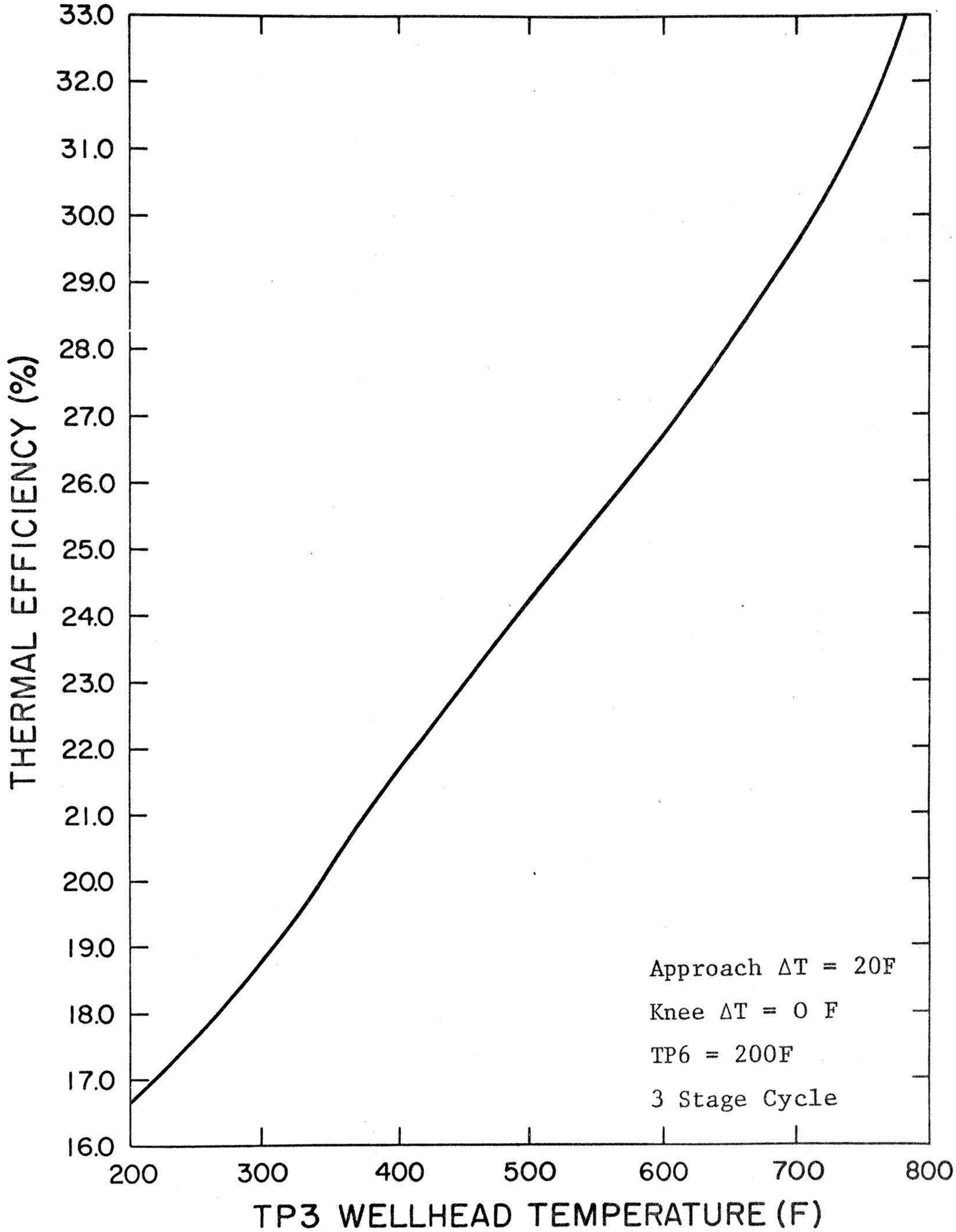


Figure 17. Proposed Thermal Efficiency Versus Wellhead Temperature for Optimization of the DGPG Operating Cycle

III. OPTIMUM WELL OPERATING CHARACTERISTICS FOR MAXIMUM ENERGY PRODUCTION

The third and final goal of the research effort was to determine the well operating conditions that would produce the maximum electrical energy, $E = \int P dt$ over a given period of time and thereby minimize its cost. As determined by Fink [3] and illustrated in Fig. 18, the well bottom temperature, TP2, diminishes with time as heat is removed from the earth. Over the same time period, the well exit temperature TP3 initially rises to a peak and then falls. The initial rise is due to the time required to remove the 10-mile column of cold water in the inner pipe. If allowed to run continuously the well exit temperature will fall exponentially to a temperature equal to TP6, the steam generator exit temperature at which time no power will be produced. For these reasons it was concluded that cyclic operation in which the mass flow was stopped and the surrounding earth temperature regenerated, was necessary to produce the optimum electrical energy.

The variables to be optimized in this intermittent well operation were

(a) the temperature at which mass flow should begin (cut-on temperature),

(b) the temperature at which it should end (cut-off temperature), and

(c) the mass flow rate.

In order to simplify the computer programming, the cut-on and cut-off temperatures were based on the well bottom temperature, TP2. Also, to

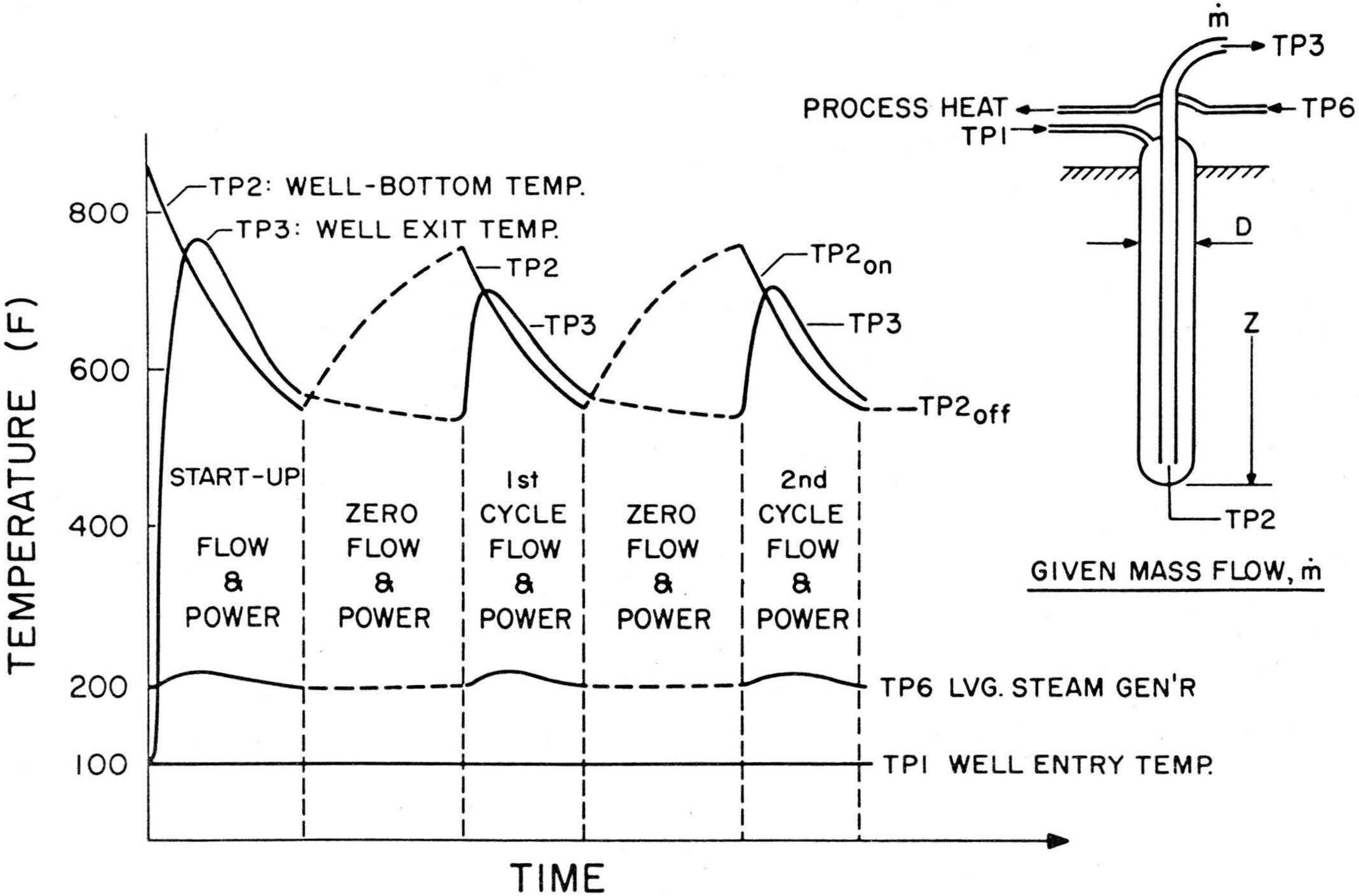


Figure 18. Optimization of the DGPG Operating Cycle

take advantage of natural circulation of the water in the well, a maximum inlet temperature (TP1) of 100F was set, and a mimimum heat exchanger exit temperature (TP6) of 200F. The thermal energy between TP6 and TP1 was considered as "process heat" for sale, as shown in the economic analysis of reference [10]. Uses of such process heat are shown in Table V.

Due to the heat losses from riser to downer, it was found that in order to keep the well exit temperature, TP3 above 200F, the cut-off temperature, TP2 must not be less than 300F. Therefore, using the results produced by Fink [3] and the graph of thermal efficiency versus wellhead temperature (TP3) shown in Fig. 17, the first results of the optimization were produced. As seen in Fig. 19, for a 30 in. well, and a 300F cut-off, at the end of 60 days the maximum power was produced with a mass flow rate of 2×10^6 lbm/hr and a cut-on temperature of 550F, at these particular conditions of 300F off, 550F on, there was a temperature difference, $\Delta T = 250F$. The analysis was continued by increasing the cut-off temperature and determining the power produced at the end of 60 days, for the same $\Delta T = 250F$ and 2×10^6 lbm/hr mass flow rate. From Fig. 20, it is seen that increasing the cut-off temperature does, in fact, increase the power produced during a 60-day period. The same procedure was continued for $\Delta T = 100F$ and $\Delta T = 50F$, each time the power continuing to rise. However, this part of the optimization was stopped at this point due to the limit of the finite difference technique used by Fink [3]. To be sure that stopping here was correct, continuous runs were made for 1×10^6 and 2×10^6 lbm/hr. In 2×10^6

TABLE V. THE REQUIRED TEMPERATURE FOR PROCESS HEAT APPLICATION [11]

^{o}F	^{o}C		
392	200		
374	190		
356	180	Evaporation of highly concentrated solutions Refrigeration by ammonia absorption Digestion in paper pulp, kraft	
338	170	Heavy water via hydrogen sulphide process and drying of diatomaceous earth	
320	160	Drying of fish meal or timber	Conventional power production
302	150	Alumina via Bayer's process	
284	140	Drying farm products at high rates and canning of food	
266	130	Evaporation in sugar refining, and extraction of salts by evaporation and crystalization	
248	120	Fresh water by distillation, most multiple effect evaporation concentration of saline solution	
230	110	Drying and curing of light aggregate cement slabs	
212	100	Drying of organic materials seaweeds, grass, vegetables, and washing and drying of wool	
194	90	Drying of stock fish and intense de-icing operations	
176	80	Space heating; greenhouses heated by space heaters	
158	70	Refrigeration (lower temperature limit)	
140	60	Animal husbandry, greenhouses by combined space and hotbed heating	
122	50	Mushroom growing and balneological baths	
104	40	Soil warming	
86	30	Swimming pools, biodegradation, fermentations, warm water for year-around mining in cold climates, and de-icing	
68	20	Hatching of fish, fish farming	

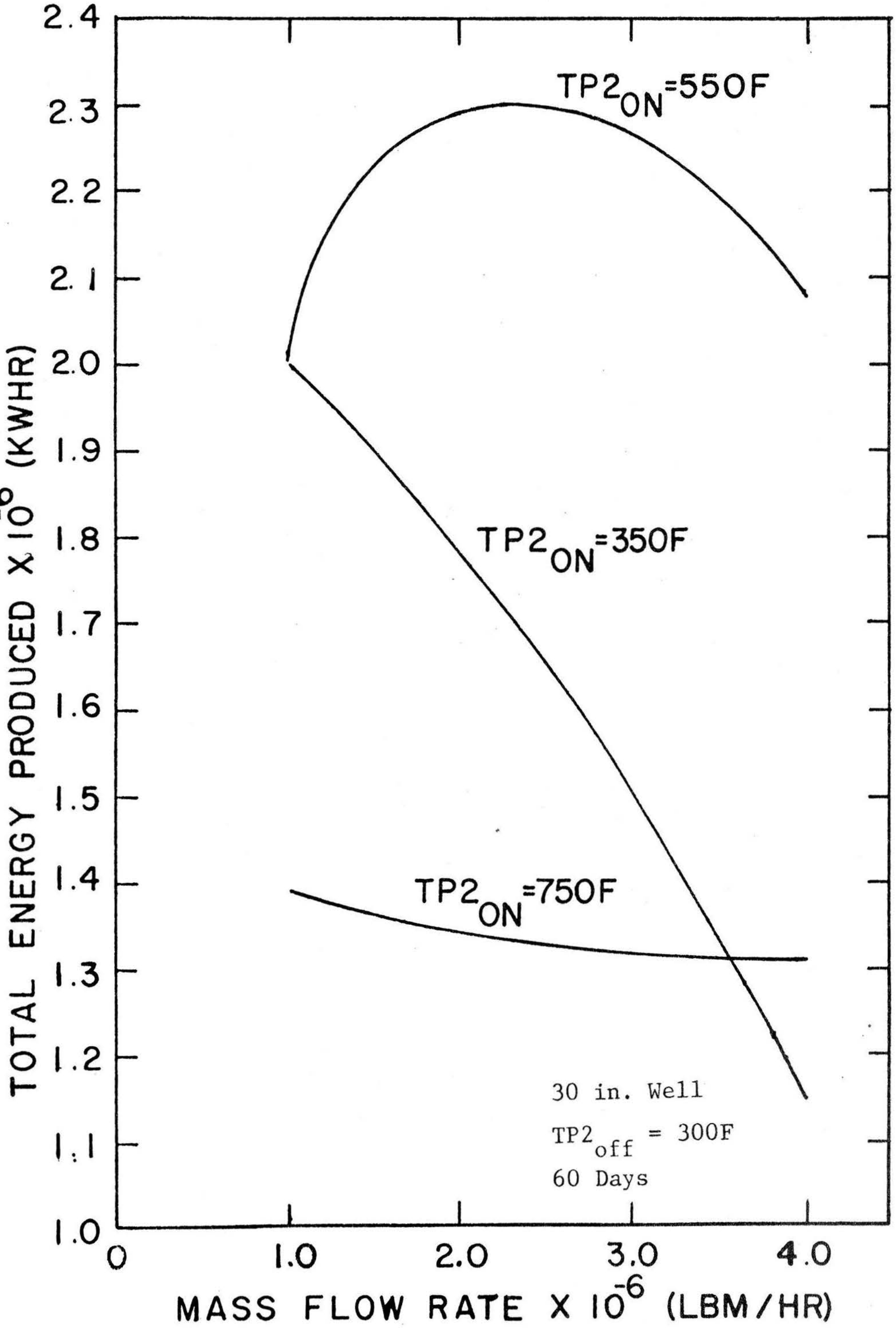


Figure 19. Total Energy Produced Versus Mass Flow Rate

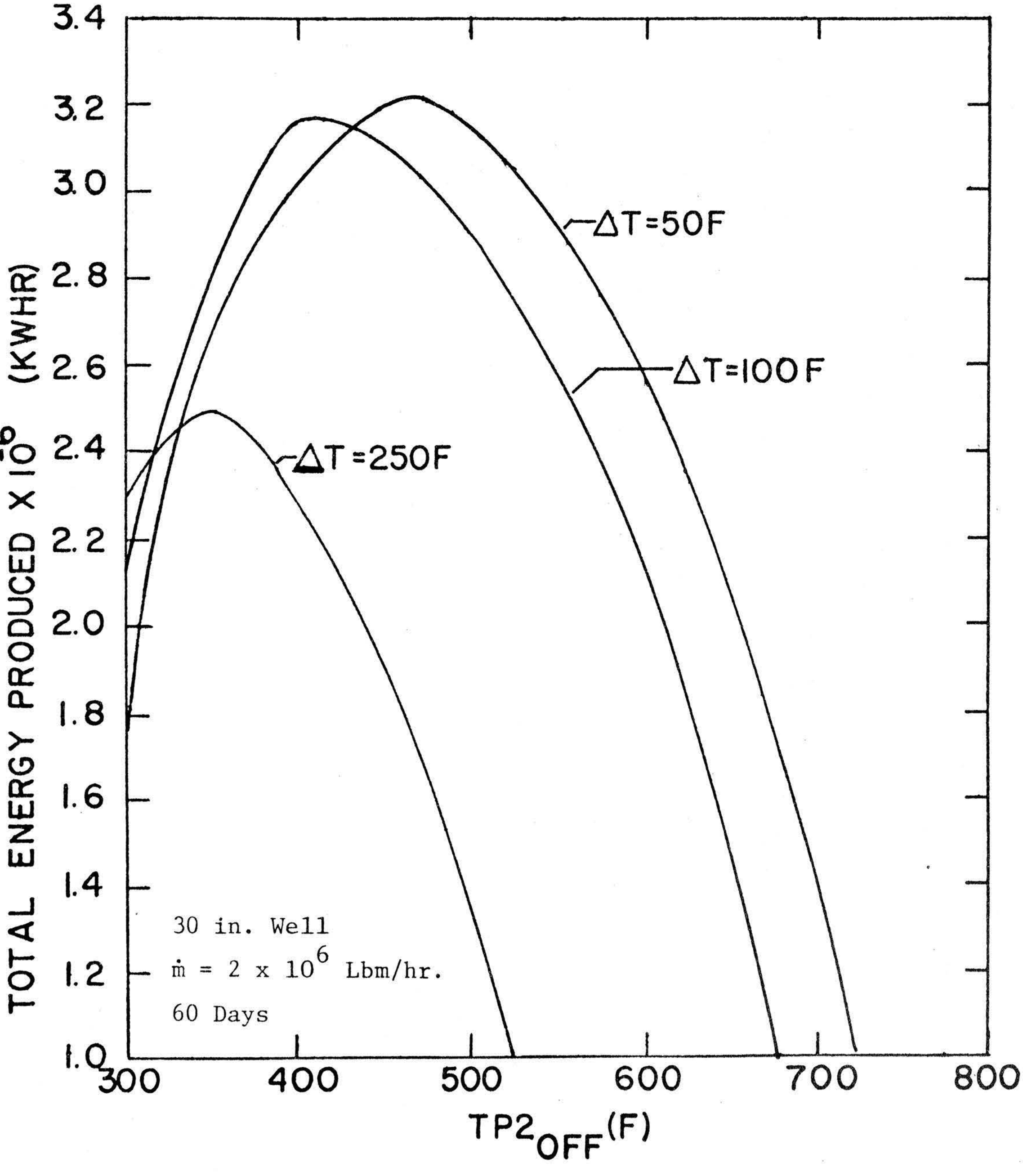


Figure 20. Total Energy Produced Versus $TP2_{off}$

lbm/hr case, TP3 fell below 200F within 24 hrs. and in the 1 x 10^6 lbm/hr case the same occured within 10 days. In both cases the power produced was far below that produced by the intermittent runs.

In order to estimate the cost of the geothermal power, or the justifiable well cost for such an operation, the optimum case of 2 x 10^6 lbm/hr, $TP2_{on}$ = 450F and ΔT = 50F was run for 120 days and is shown in Fig. 21. A straight line extrapolation of the total power produced versus time was made and it was found that 14,280,000 kwhr of electricity would be produced along with 169,000 million Btu(mkB) of process heat at the end of the first year. If the price of electricity at the busbar is $.02/kwhr and if process heat is $3.00/mkB the net income at the end of the first year, allowing for a $50,000 operating cost is $740,000.00. By amortizing this income over a 50 year period at the rate of 10% per year, the justifiable plant would be $7,400,000, of which the surface equipment would be approximately $400,000.

It is worth noting that for a 10 ft diameter well, rather than the 30 in. diameter well as calculated above, at the end of 60 days approximately three times the energy will be produced. It is therefore, perhaps more realistic to think of drilling a 10 ft diameter well for only a realitively small jump in cost compared to the power output received.

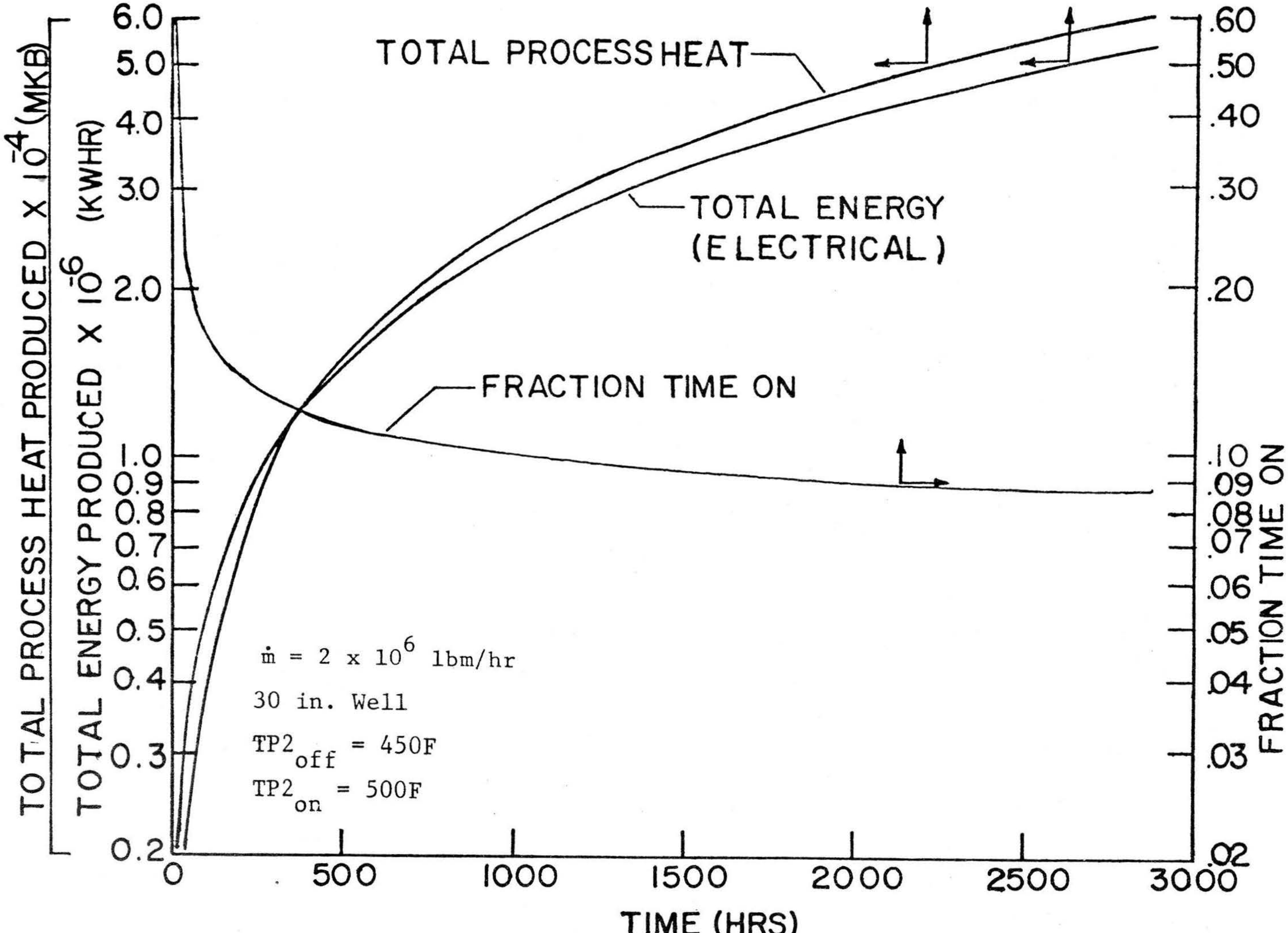


Figure 21. Total Energy, Total Process Heat, and Fraction Time On for Optimum Well Design

CONCLUSIONS AND RECOMMENDATIONS

The design and installation of an insulated coaxial pipe in a 50,000 ft drilled geothermal well has been shown to be feasible. Design details, however, must be worked out in future engineering studies before actual construction can be made.

Thermal efficiencies in the range of 17.0 - 33.0% for wellhead temperatures of 200F to 800F, respectively, can be attained from the surface equipment by using a three-stage power conversion cycle. Due to the changing wellhead temperature, the efficiencies and power outputs of the system will continuously vary throughout the entire well operation.

Intermittent operation of the well, in order to allow the surrounding temperature of the earth to be regenerated, appears to be best for maximum energy production. For this type of operation, justifiable well costs appear to be approximately $7,000,000 which is within future technological possibility. A series of wells using the same surface equipment will provide continuous power. In the author's judgement geothermal power will be more practical for meeting the daily peaking load, for which power companies are willing to pay premium prices in order to avoid increasing base load plants; however, further investigation into the optimum well operating characteristics should be made.

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APPENDIX A: COMPUTER OUTPUTS FOR THE OPTIMIZATION OF THE ABOVEGROUND POWER GENERATION SYSTEM

2 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP5 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.2030	16808.71	200.0	350.0	-10.0	10.0
0.2028	16826.09	200.0	350.0	-10.0	20.0
0.2027	16840.38	200.0	350.0	-10.0	30.0
0.1937	17618.41	200.0	350.0	0.0	10.0
0.1935	17641.14	200.0	350.0	0.0	20.0
0.1933	17660.38	200.0	350.0	0.0	30.0
0.1841	18542.73	200.0	350.0	10.0	10.0
0.1838	18571.81	200.0	350.0	10.0	20.0
0.1835	18597.35	200.0	350.0	10.0	30.0
0.2660	12829.43	300.0	350.0	-10.0	10.0
0.2660	12831.21	300.0	350.0	-10.0	20.0
0.2660	12831.32	300.0	350.0	-10.0	30.0
0.2588	13188.66	300.0	350.0	0.0	10.0
0.2587	13192.26	300.0	350.0	0.0	20.0
0.2587	13194.09	300.0	350.0	0.0	30.0
0.2513	13580.44	300.0	350.0	10.0	10.0
0.2512	13586.01	300.0	350.0	10.0	20.0
0.2511	13589.53	300.0	350.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)
** APPROACH=(TP3-T4A)=(TP4-T4B)

2 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP5 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.2140	15949.45	200.0	400.0	-10.0	10.0
0.2137	15971.63	200.0	400.0	-10.0	20.0
0.2134	15991.45	200.0	400.0	-10.0	30.0
0.2051	16641.21	200.0	400.0	0.0	10.0
0.2048	16668.17	200.0	400.0	0.0	20.0
0.2045	16692.67	200.0	400.0	0.0	30.0
0.1959	17422.55	200.0	400.0	10.0	10.0
0.1955	17455.32	200.0	400.0	10.0	20.0
0.1952	17485.50	200.0	400.0	10.0	30.0
0.2749	12415.95	300.0	400.0	-10.0	10.0
0.2747	12422.68	300.0	400.0	-10.0	20.0
0.2746	12427.76	300.0	400.0	-10.0	30.0
0.2680	12736.34	300.0	400.0	0.0	10.0
0.2678	12744.52	300.0	400.0	0.0	20.0
0.2677	12751.13	300.0	400.0	0.0	30.0
0.2609	13083.72	300.0	400.0	10.0	10.0
0.2607	13093.65	300.0	400.0	10.0	20.0
0.2605	13101.98	300.0	400.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)
** APPROACH=(TP3-T4A)=(TP4-T4B)

2 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP5 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.2246	15195.93	200.0	450.0	-10.0	10.0
0.2242	15221.14	200.0	450.0	-10.0	20.0
0.2239	15244.51	200.0	450.0	-10.0	30.0
0.2161	15794.18	200.0	450.0	0.0	10.0
0.2157	15823.77	200.0	450.0	0.0	20.0
0.2153	15851.41	200.0	450.0	0.0	30.0
0.2073	16463.94	200.0	450.0	10.0	10.0
0.2069	16498.79	200.0	450.0	10.0	20.0
0.2065	16531.48	200.0	450.0	10.0	30.0
0.2836	12034.88	300.0	450.0	-10.0	10.0
0.2834	12044.80	300.0	450.0	-10.0	20.0
0.2832	12053.63	300.0	450.0	-10.0	30.0
0.2770	12322.98	300.0	450.0	0.0	10.0
0.2767	12334.35	300.0	450.0	0.0	20.0
0.2765	12344.55	300.0	450.0	0.0	30.0
0.2701	12633.88	300.0	450.0	10.0	10.0
0.2699	12646.91	300.0	450.0	10.0	20.0
0.2696	12658.72	300.0	450.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)
** APPROACH=(TP3-T4A)=(TP4-T4B)

2 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP5 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.2351	14517.37	200.0	500.0	-10.0	10.0
0.2347	14544.38	200.0	500.0	-10.0	20.0
0.2342	14569.99	200.0	500.0	-10.0	30.0
0.2269	15039.52	200.0	500.0	0.0	10.0
0.2265	15070.42	200.0	500.0	0.0	20.0
0.2260	15099.91	200.0	500.0	0.0	30.0
0.2185	15619.37	200.0	500.0	10.0	10.0
0.2180	15654.98	200.0	500.0	10.0	20.0
0.2175	15689.02	200.0	500.0	10.0	30.0
0.2924	11673.79	300.0	500.0	-10.0	10.0
0.2921	11686.18	300.0	500.0	-10.0	20.0
0.2918	11697.60	300.0	500.0	-10.0	30.0
0.2860	11934.61	300.0	500.0	0.0	10.0
0.2856	11948.28	300.0	500.0	0.0	20.0
0.2853	11961.01	300.0	500.0	0.0	30.0
0.2794	12214.71	300.0	500.0	10.0	10.0
0.2791	12229.87	300.0	500.0	10.0	20.0
0.2787	12244.09	300.0	500.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)
** APPROACH=(TP3-T4A)=(TP4-T4B)

2 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP5 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.2456	13894.06	200.0	550.0	-10.0	10.0
0.2452	13922.02	200.0	550.0	-10.0	20.0
0.2447	13948.88	200.0	550.0	-10.0	30.0
0.2378	14352.92	200.0	550.0	0.0	10.0
0.2373	14384.36	200.0	550.0	0.0	20.0
0.2368	14414.66	200.0	550.0	0.0	30.0
0.2297	14858.82	200.0	550.0	10.0	10.0
0.2291	14894.40	200.0	550.0	10.0	20.0
0.2286	14928.81	200.0	550.0	10.0	30.0
0.3013	11326.64	300.0	550.0	-10.0	10.0
0.3010	11340.73	300.0	550.0	-10.0	20.0
0.3006	11354.07	300.0	550.0	-10.0	30.0
0.2951	11564.09	300.0	550.0	0.0	10.0
0.2947	11579.34	300.0	550.0	0.0	20.0
0.2944	11593.89	300.0	550.0	0.0	30.0
0.2888	11818.01	300.0	550.0	10.0	10.0
0.2884	11834.60	300.0	550.0	10.0	20.0
0.2880	11850.48	300.0	550.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)
** APPROACH=(TP3-T4A)=(TP4-T4B)

2 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP5 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.2567	13295.68	200.0	600.0	-10.0	10.0
0.2562	13323.79	200.0	600.0	-10.0	20.0
0.2556	13351.09	200.0	600.0	-10.0	30.0
0.2491	13699.55	200.0	600.0	0.0	10.0
0.2486	13730.76	200.0	600.0	0.0	20.0
0.2480	13761.17	200.0	600.0	0.0	30.0
0.2413	14141.88	200.0	600.0	10.0	10.0
0.2407	14176.71	200.0	600.0	10.0	20.0
0.2402	14210.69	200.0	600.0	10.0	30.0
0.3108	10981.82	300.0	600.0	-10.0	10.0
0.3104	10997.07	300.0	600.0	-10.0	20.0
0.3099	11011.77	300.0	600.0	-10.0	30.0
0.3048	11198.73	300.0	600.0	0.0	10.0
0.3043	11215.07	300.0	600.0	0.0	20.0
0.3039	11230.85	300.0	600.0	0.0	30.0
0.2986	11429.62	300.0	600.0	10.0	10.0
0.2982	11447.20	300.0	600.0	10.0	20.0
0.2977	11464.19	300.0	600.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)
** APPROACH=(TP3-T4A)=(TP4-T4B)

2 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP5 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.2788	12242.60	200.0	700.0	-10.0	10.0
0.2782	12270.03	200.0	700.0	-10.0	20.0
0.2775	12296.95	200.0	700.0	-10.0	30.0
0.2716	12564.12	200.0	700.0	0.0	10.0
0.2710	12593.93	200.0	700.0	0.0	20.0
0.2704	12623.29	200.0	700.0	0.0	30.0
0.2643	12912.00	200.0	700.0	10.0	10.0
0.2637	12944.59	200.0	700.0	10.0	20.0
0.2630	12976.70	200.0	700.0	10.0	30.0
0.3302	10335.73	300.0	700.0	-10.0	10.0
0.3297	10352.26	300.0	700.0	-10.0	20.0
0.3292	10368.46	300.0	700.0	-10.0	30.0
0.3244	10521.54	300.0	700.0	0.0	10.0
0.3238	10538.92	300.0	700.0	0.0	20.0
0.3233	10556.00	300.0	700.0	0.0	30.0
0.3185	10717.41	300.0	700.0	10.0	10.0
0.3179	10735.80	300.0	700.0	10.0	20.0
0.3174	10753.89	300.0	700.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)
** APPROACH=(TP3-T4A)=(TP4-T4B)

2 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP5 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.3231	10562.82	200.0	800.0	-10.0	10.0
0.3225	10584.57	200.0	800.0	-10.0	20.0
0.3218	10606.13	200.0	800.0	-10.0	30.0
0.3167	10776.22	200.0	800.0	0.0	10.0
0.3160	10799.31	200.0	800.0	0.0	20.0
0.3154	10822.20	200.0	800.0	0.0	30.0
0.3102	11002.12	200.0	800.0	10.0	10.0
0.3095	11026.71	200.0	800.0	10.0	20.0
0.3086	11051.12	200.0	800.0	10.0	30.0
0.3684	9263.42	300.0	800.0	-10.0	10.0
0.3678	9278.70	300.0	800.0	-10.0	20.0
0.3672	9293.89	300.0	800.0	-10.0	30.0
0.3628	9407.51	300.0	800.0	0.0	10.0
0.3622	9423.30	300.0	800.0	0.0	20.0
0.3616	9438.99	300.0	800.0	0.0	30.0
0.3571	9556.74	300.0	800.0	10.0	10.0
0.3565	9573.15	300.0	800.0	10.0	20.0
0.3559	9589.41	300.0	800.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)
** APPROACH=(TP3-T4A)=(TP4-T4B)

2 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP5 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.3420	9980.91	200.0	900.0	-10.0	10.0
0.3412	10001.97	200.0	900.0	-10.0	20.0
0.3405	10022.95	200.0	900.0	-10.0	30.0
0.3354	10176.50	200.0	900.0	0.0	10.0
0.3346	10198.75	200.0	900.0	0.0	20.0
0.3339	10220.89	200.0	900.0	0.0	30.0
0.3287	10381.87	200.0	900.0	10.0	10.0
0.3280	10405.41	200.0	900.0	10.0	20.0
0.3273	10428.85	200.0	900.0	10.0	30.0
0.3899	8754.07	300.0	900.0	-10.0	10.0
0.3892	8769.14	300.0	900.0	-10.0	20.0
0.3885	8784.18	300.0	900.0	-10.0	30.0
0.3838	8892.41	300.0	900.0	0.0	10.0
0.3831	8907.95	300.0	900.0	0.0	20.0
0.3825	8923.45	300.0	900.0	0.0	30.0
0.3778	9032.90	300.0	900.0	10.0	10.0
0.3772	9048.95	300.0	900.0	10.0	20.0
0.3765	9064.96	300.0	900.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)
** APPROACH=(TP3-T4A)=(TP4-T4B)

3 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP6 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.2115	16138.25	200.0	350.0	-10.0	10.0
0.2114	16147.56	200.0	350.0	-10.0	20.0
0.2113	16153.85	200.0	350.0	-10.0	30.0
0.2026	16845.59	200.0	350.0	0.0	10.0
0.2024	16859.00	200.0	350.0	0.0	20.0
0.2023	16869.10	200.0	350.0	0.0	30.0
0.1934	17645.53	200.0	350.0	10.0	10.0
0.1932	17663.70	200.0	350.0	10.0	20.0
0.1931	17678.31	200.0	350.0	10.0	30.0
0.2677	12749.20	300.0	350.0	-10.0	10.0
0.2677	12750.12	300.0	350.0	-10.0	20.0
0.2677	12748.48	300.0	350.0	-10.0	30.0
0.2606	13098.12	300.0	350.0	0.0	10.0
0.2605	13100.55	300.0	350.0	0.0	20.0
0.2605	13100.94	300.0	350.0	0.0	30.0
0.2532	13477.97	300.0	350.0	10.0	10.0
0.2532	13481.89	300.0	350.0	10.0	20.0
0.2531	13484.41	300.0	350.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)=(TP6-T3C)
** APPROACH=(TP3-T4A)=(TP4-T4B)=(TP5-T4C)

3 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP6 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.2250	15169.26	200.0	400.0	-10.0	10.0
0.2248	15182.29	200.0	400.0	-10.0	20.0
0.2246	15192.89	200.0	400.0	-10.0	30.0
0.2166	15756.07	200.0	400.0	0.0	10.0
0.2164	15772.54	200.0	400.0	0.0	20.0
0.2162	15786.42	200.0	400.0	0.0	30.0
0.2080	16412.02	200.0	400.0	10.0	10.0
0.2077	16432.64	200.0	400.0	10.0	20.0
0.2075	16450.57	200.0	400.0	10.0	30.0
0.2791	12230.23	300.0	400.0	-10.0	10.0
0.2790	12234.73	300.0	400.0	-10.0	20.0
0.2789	12237.52	300.0	400.0	-10.0	30.0
0.2723	12532.88	300.0	400.0	0.0	10.0
0.2722	12538.72	300.0	400.0	0.0	20.0
0.2721	12542.99	300.0	400.0	0.0	30.0
0.2654	12860.09	300.0	400.0	10.0	10.0
0.2652	12867.52	300.0	400.0	10.0	20.0
0.2651	12873.34	300.0	400.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)=(TP6-T3C)
** APPROACH=(TP3-T4A)=(TP4-T4B)=(TP5-T4C)

3 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP6 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.2379	14348.83	200.0	450.0	-10.0	10.0
0.2376	14364.38	200.0	450.0	-10.0	20.0
0.2374	14378.08	200.0	450.0	-10.0	30.0
0.2299	14844.67	200.0	450.0	0.0	10.0
0.2296	14863.18	200.0	450.0	0.0	20.0
0.2294	14879.77	200.0	450.0	0.0	30.0
0.2217	15393.64	200.0	450.0	10.0	10.0
0.2214	15415.67	200.0	450.0	10.0	20.0
0.2211	15435.61	200.0	450.0	10.0	30.0
0.2900	11768.80	300.0	450.0	-10.0	10.0
0.2898	11776.09	300.0	450.0	-10.0	20.0
0.2897	11781.97	300.0	450.0	-10.0	30.0
0.2836	12034.51	300.0	450.0	0.0	10.0
0.2834	12042.94	300.0	450.0	0.0	20.0
0.2832	12050.18	300.0	450.0	0.0	30.0
0.2770	12320.40	300.0	450.0	10.0	10.0
0.2768	12330.25	300.0	450.0	10.0	20.0
0.2766	12338.79	300.0	450.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)=(TP6-T3C)
** APPROACH=(TP3-T4A)=(TP4-T4B)=(TP5-T4C)

3 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP6 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.2504	13628.90	200.0	500.0	-10.0	10.0
0.2501	13646.14	200.0	500.0	-10.0	20.0
0.2498	13661.91	200.0	500.0	-10.0	30.0
0.2429	14053.37	200.0	500.0	0.0	10.0
0.2425	14073.18	200.0	500.0	0.0	20.0
0.2422	14091.43	200.0	500.0	0.0	30.0
0.2351	14519.64	200.0	500.0	10.0	10.0
0.2347	14542.43	200.0	500.0	10.0	20.0
0.2344	14563.61	200.0	500.0	10.0	30.0
0.3008	11346.63	300.0	500.0	-10.0	10.0
0.3005	11356.07	300.0	500.0	-10.0	20.0
0.3003	11364.48	300.0	500.0	-10.0	30.0
0.2947	11582.57	300.0	500.0	0.0	10.0
0.2944	11592.98	300.0	500.0	0.0	20.0
0.2942	11602.49	300.0	500.0	0.0	30.0
0.2884	11835.08	300.0	500.0	10.0	10.0
0.2881	11846.68	300.0	500.0	10.0	20.0
0.2878	11857.21	300.0	500.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)=(TP6-T3C)

** APPROACH=(TP3-T4A)=(TP4-T4B)=(TP5-T4C)

3 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP6 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.2630	12979.38	200.0	550.0	-10.0	10.0
0.2626	12997.68	200.0	550.0	-10.0	20.0
0.2622	13014.94	200.0	550.0	-10.0	30.0
0.2557	13346.73	200.0	550.0	0.0	10.0
0.2553	13367.22	200.0	550.0	0.0	20.0
0.2550	13386.61	200.0	550.0	0.0	30.0
0.2483	13747.31	200.0	550.0	10.0	10.0
0.2479	13770.40	200.0	550.0	10.0	20.0
0.2475	13792.20	200.0	550.0	10.0	30.0
0.3117	10950.69	300.0	550.0	-10.0	10.0
0.3114	10961.83	300.0	550.0	-10.0	20.0
0.3111	10972.18	300.0	550.0	-10.0	30.0
0.3058	11162.20	300.0	550.0	0.0	10.0
0.3054	11174.12	300.0	550.0	0.0	20.0
0.3051	11185.31	300.0	550.0	0.0	30.0
0.2997	11387.43	300.0	550.0	10.0	10.0
0.2994	11400.31	300.0	550.0	10.0	20.0
0.2991	11412.47	300.0	550.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)=(TP6-T3C)
** APPROACH=(TP3-T4A)=(TP4-T4B)=(TP5-T4C)

3 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP6 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.2760	12367.25	200.0	600.0	-10.0	10.0
0.2755	12386.23	200.0	600.0	-10.0	20.0
0.2751	12404.41	200.0	600.0	-10.0	30.0
0.2690	12687.12	200.0	600.0	0.0	10.0
0.2686	12707.92	200.0	600.0	0.0	20.0
0.2682	12727.86	200.0	600.0	0.0	30.0
0.2619	13033.48	200.0	600.0	10.0	10.0
0.2614	13056.46	200.0	600.0	10.0	20.0
0.2610	13078.45	200.0	600.0	10.0	30.0
0.3231	10562.20	300.0	600.0	-10.0	10.0
0.3228	10574.68	300.0	600.0	-10.0	20.0
0.3224	10586.61	300.0	600.0	-10.0	30.0
0.3174	10753.15	300.0	600.0	0.0	10.0
0.3170	10766.27	300.0	600.0	0.0	20.0
0.3166	10778.77	300.0	600.0	0.0	30.0
0.3115	10955.40	300.0	600.0	10.0	10.0
0.3111	10969.32	300.0	600.0	10.0	20.0
0.3108	10982.59	300.0	600.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)=(TP6-T3C)
** APPROACH=(TP3-T4A)=(TP4-T4B)=(TP5-T4C)

3 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP6 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.3021	11298.90	200.0	700.0	-10.0	10.0
0.3015	11318.42	200.0	700.0	-10.0	20.0
0.3010	11337.50	200.0	700.0	-10.0	30.0
0.2955	11551.69	200.0	700.0	0.0	10.0
0.2949	11572.48	200.0	700.0	0.0	20.0
0.2944	11592.79	200.0	700.0	0.0	30.0
0.2887	11821.93	200.0	700.0	10.0	10.0
0.2882	11844.23	200.0	700.0	10.0	20.0
0.2876	11866.00	200.0	700.0	10.0	30.0
0.3468	9841.77	300.0	700.0	-10.0	10.0
0.3463	9856.06	300.0	700.0	-10.0	20.0
0.3458	9870.19	300.0	700.0	-10.0	30.0
0.3412	10003.97	300.0	700.0	0.0	10.0
0.3407	10018.63	300.0	700.0	0.0	20.0
0.3402	10033.05	300.0	700.0	0.0	30.0
0.3355	10174.05	300.0	700.0	10.0	10.0
0.3350	10189.19	300.0	700.0	10.0	20.0
0.3345	10204.04	300.0	700.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)=(TP6-T3C)

** APPROACH=(TP3-T4A)=(TP4-T4B)=(TP5-T4C)

3 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP6 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.3581	9529.72	200.0	800.0	-10.0	10.0
0.3575	9546.77	200.0	800.0	-10.0	20.0
0.3569	9563.79	200.0	800.0	-10.0	30.0
0.3518	9701.51	200.0	800.0	0.0	10.0
0.3512	9719.04	200.0	800.0	0.0	20.0
0.3505	9736.50	200.0	800.0	0.0	30.0
0.3455	9879.20	200.0	800.0	10.0	10.0
0.3448	9897.34	200.0	800.0	10.0	20.0
0.3442	9915.35	200.0	800.0	10.0	30.0
0.3984	8567.14	300.0	800.0	-10.0	10.0
0.3977	8582.03	300.0	800.0	-10.0	20.0
0.3970	8597.10	300.0	800.0	-10.0	30.0
0.3907	8734.82	300.0	800.0	0.0	10.0
0.3901	8749.42	300.0	800.0	0.0	20.0
0.3894	8764.10	300.0	800.0	0.0	30.0
0.3856	8851.03	300.0	800.0	10.0	10.0
0.3850	8865.72	300.0	800.0	10.0	20.0
0.3843	8880.46	300.0	800.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)=(TP6-T3C)
** APPROACH=(TP3-T4A)=(TP4-T4B)=(TP5-T4C)

3 STAGE CYCLE

THERMAL EFFICIENCY	HEAT RATE (BTU/KWHR)	TP6 (DEG.F)	TP3 (DEG.F)	KNEE* (DEG.F)	APPROACH** (DEG.F)
0.3858	8845.81	200.0	900.0	-10.0	10.0
0.3851	8862.82	200.0	900.0	-10.0	20.0
0.3843	8879.93	200.0	900.0	-10.0	30.0
0.3786	9014.18	200.0	900.0	0.0	10.0
0.3779	9031.53	200.0	900.0	0.0	20.0
0.3772	9048.94	200.0	900.0	0.0	30.0
0.3715	9186.79	200.0	900.0	10.0	10.0
0.3708	9204.61	200.0	900.0	10.0	20.0
0.3701	9222.44	200.0	900.0	10.0	30.0
0.4289	7956.86	300.0	900.0	-10.0	10.0
0.4281	7972.21	300.0	900.0	-10.0	20.0
0.4273	7987.88	300.0	900.0	-10.0	30.0
0.4208	8110.78	300.0	900.0	0.0	10.0
0.4200	8125.82	300.0	900.0	0.0	20.0
0.4192	8141.11	300.0	900.0	0.0	30.0
0.4137	8249.50	300.0	900.0	10.0	10.0
0.4130	8264.47	300.0	900.0	10.0	20.0
0.4122	8279.63	300.0	900.0	10.0	30.0

* KNEE=(TP4-T3A)=(TP5-T3B)=(TP6-T3C)
** APPROACH=(TP3-T4A)=(TP4-T4B)=(TP5-T4C)

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DESIGN ANALYSIS

OF A

DRILLED GEOTHERMAL WELL

BY

JEFFERY H. WARREN

(ABSTRACT)

An insulated coaxial pipe for a 50,000 ft drilled geothermal well was designed, in order to remove heat from the earth's crust by the circulation of water through a single well. The design took into consideration means of insulating and supporting the inner pipe, thermal expansion, the large hydrostatic pressures involved, and a feasible means of installing the ten mile pipe.

A thermodynamic analysis of the aboveground power generation system, including a once-through steam generator, condensor, circulating pumps, and a feed-water heater system, was made in order to obtain the maximum thermal efficiency. The drilled geothermal power generation equipment is different from that of conventional fossil or nuclear-fired equipment in that the water temperature arriving at the wellhead is continuously varying due to the removal of heat from the earth's crust.

Due to the fact that the earth is a poor conductor of heat, continuous operation of the well results in the water at the wellhead falling below an acceptable power generation temperature after a relatively

short time period. Therefore, intermittent operation of several wells in staggered fashion using the same aboveground power generation equipment is required. A determination of the well operating characteristics including optimum mass flow and well cut-on and cut-off temperatures was made. From this a justifiable well cost was determined in order to provide geothermal electrical power and process heat at a resonable cost.