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IDAHO NATIONAL ENGINEERING LABORATORY

DESIGN CONCEPTS FOR FLASH STEAM SYSTEMS FOR USE WITH MEDIUM TEMPERATURE GEOTHERMAL WATER

J.F.Whitbeck

ENERGY RESEARCH & DEVELOPMENT ADMINISTRATION
USA

PREPARED FOR THE
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DESIGN CONCEPTS FOR FLASH STEAM SYSTEMS FOR USE WITH MEDIUM TEMPERATURE GEOTHERMAL WATER

Prepared by Idaho Geothermal Project

J. F. Whitbeck

SUBMITTED TO THE IDAHO OPERATIONS OFFICE AND THE DIVISION OF GEOTHERMAL RESEARCH

U. S. ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION

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ABSTRACT

The abundance and potential of geothermal resources for providing part of our future energy needs has been established. The temperature of the resources vary, with the abundance increasing rapidly as the needed temperature range is lowered. Presently high temperature (>200°C, 392°F) resources appear to be able to furnish electrical power at competitive costs to more conventional resources. In the range of 150°C (302°F), the economic competitiveness is not clear, though the abundance of the fluids is much greater at readily retrievable depths. The Raft River Area of Southern Idaho is believed to be typical of such moderate temperature reservoirs throughout the United States. The referenced documentation discusses the availability of such geothermal waters, while this document discusses one method of utilizing these waters to produce electricity.

Medium temperature water can be utilized for production of electrical energy when it is available in massive quantities. The design concepts herein are to provide a base for feasibility studies and evaluate processes with consideration of the economics of developing this electrical energy on a commercial scale.

Two methods of producing electrical energy with geothermal water are being considered. The methods discussed in this document are by the flashing process of producing steam for driving turbine-generators. The second method is exchanging heat to a secondary working fluid that would drive the turbine. It is generally recognized that the second method can potentially provide greater power per unit of geothermal water than a steam system.

The magnitude of the general technology requirements of the heat exchange method is greater than the steam method. The known technical problems to be resolved in the heat exchanger hardware are extensive. Development of the flash steam method offers the possibility of getting a plant and related facilities in operation in less time.

Flash steam systems were evaluated for use with 300°F water. Single and multiflash systems were evaluated and component size sensitivity to operating pressures were studied. It was determined a double flash system is the most practical system. Net power production of approximately 2.4 megawatts/million pounds per hour of brine is estimated for the double flash system which operates at an initial flash pressure of 30 psia and a second stage pressure of 13 psia. Flash pressures below atmospheric are not recommended due to oxygen leakage into the system. Sensitivity analysis has indicated that the power output is not highly sensitive to the first stage flash pressure. A significant loss in power output occurs if the second stage pressure is increased significantly.

The quantity and substance of non-condensables in the geothermal water has yet to be determined. The removal of non-condensables from the condenser could consume up to 25% of the power generated if the steam contains 10% non-condensables. In the event such large quantities of non-condensables are encountered, trade-off studies which consider a preliminary flash for venting a majority of the non-condensables will be required. Geothermal water analysis from deep wells in the Raft River Valley indicate that steam will contain less than 0.1% non-condensables.

Acknowledgment

The author wishes to acknowledge the help of Mr. S. C. Caywood who aided in organizing the material, preparing the final figures, editing and taking care of the many details required in the publication process.

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FIGURES

TABLES

NOMENCLATURE

A	Area
b	Brine
C	Cooling water
c	Specific heat of cooling water, Btu/lb-^{o}F
D	Diameter, ft
d	Diameter, inch
e	Turbine exit or condenser inlet point
F	Flashed
f	Fluid
H	Enthalpy
H_b	Enthalpy, brine, Btu/lb
H_{e_a}	Enthalpy, turbine exit, actual, Btu/lb
$H_{e'}$	Turbine exit, isentropic
H_f	Enthalpy, fluid, Btu/lb
H_{fg}	Enthalpy, fluid latent, Btu/lb
H_{f_n}	Enthalpy, fluid, efficiency Btu/lb
H_g	Enthalpy, steam, Btu/lb
p	Pressure, psia
p_s	Steam pressure, psia
p_{cond}	Condenser pressure
Q	Work
q	Heat
Q_{turb_i}	Work, turbine, ideal Btu/hr
$Q_{turb-gen_a}$	Work, turbine-generator, actual, Btu/hr
Q_{q_C}	Work, heat released to Cooling water
S	Entropy
s	Steam

NOMENCLATURE

T	Temperature, thermodynamic
T_{Ci}	Temperature, ^{o}F, cooling water inlet
T_{Co}	Temperature, ^{o}F, cooling water outlet
T_{cond}	Temperature, sink, condenser
T_{i}	Temperature, flashed, turbine inlet
T_{s}	Temperature steam
V	Velocity, ft/sec
V_{i}	Velocity, ft/sec, inlet
w	Flow rate
w_{b}	Flow rate of brine
w_{s}	Flow rate of steam
w_{C}	Flow rate, cooling water
w_{m}	Flow rate, mixture, lb
w_{ncd}	Flow rate, non-condensables
n	Efficiency
n_{gen}	Efficiency, generator
$n_{turb-gen}$	Efficiency, turbine generator
n_{turb}	Efficiency, turbine
$n_{turb-stg}$	Efficiency, turbine stage
ν	Specific volume
ρ	Density
X	Quality, mass fraction of vapor in a two phase mixture

<u>Subscript Numbers</u>

1	Brine
2	Condenser conditions
3	Actual exhaust end point
3'	Isentropic turbine exhaust end point
4	Condensate

1. INTRODUCTION

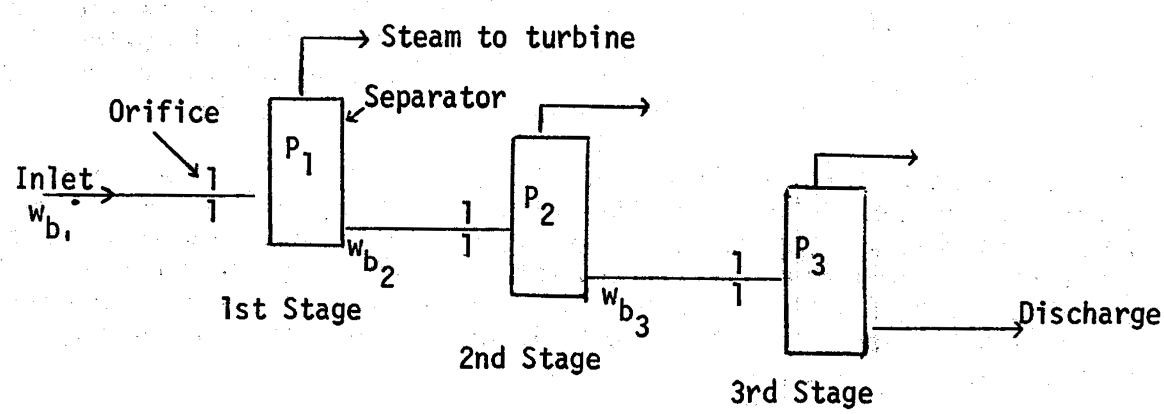
The production of electrical power in the United States from a geothermal energy is restricted solely to the geyser area in Northern California. The Northern California steam wells provide steam for direct use in steam turbines. Such steam sources are unique, the geyser area being one of two known fields in the world. A much larger geothermal resource is thought to exist in hot water fields. One method of utilizing hot water is to permit this water to flash to steam. The steam is then used to drive a turbine-generator. This principle has been applied in Wairakei, New Zealand, and in Cerro Prieto, Mexico, where high temperature water is allowed to flash as the fluid rises to the surface of the earth. The steam is separated from the water at the well site and then piped to the power plant. Because of the low density over a substantial depth of the well, the wells flow by themselves.

Studies have been completed over the past year to evaluate the use of medium temperature ($\sim 300^{o}F$) geothermal water for the production of power. The purpose of this work is to report on the studies concerned with flash steam systems. The initial phase of this work was primarily concerned with obtaining estimates of equipment size and sensitivity to operating conditions for a single flash system of approximately 10 MW(e). This was followed by sizing of a double flash system.

The results of these studies are reported herein. They provide information as to the specific usable energy that can be derived from a flash steam system operating with $300^{o}F$ geothermal water and provide some insight into the equipment sizes required.

1.1 Flash Steam Thermodynamics

Steam generation is provided by causing the water pressure to drop to a pressure lower than the saturation pressure corresponding to its temperature. Typically, the pressurized brine is throttled by a valve or orifice and the resulting steam separated from the two-phase mixture in a separation tank. Several stages can be put in series resulting in a multi-flash system producing steam at each stage.



Since the throttling process is assumed to the isenthalpic, the fraction of steam generated in each stage is computed by:

$$x_n = \frac{H_{f_{n+1}} - H_{f_n}}{H_{fg_n}}$$

$$w_{s_n} = x_n \, w_{b_n}$$

or in terms of the inlet brine flow, the steam flow from each stage can be expressed as:

$$w_{s_n} = x_n \left[(1-x_1)(1-x_2) - - - -(1-x_{n-1})\right] w_{b_1}$$

The work that can be done by the steam generated at each flash stage is:

$$Q_n = x_{s_n} (H_g - H_e) \, n_{turb\text{-}gen}$$

$$Q_n = n_{turb\text{-}gen} \, x_n \left[(1-x_1)(1-x_2) - - - -(1-x_{n-1}) \, w_b\right] (h_g - h_e)_n$$

The specific power/stage, Btu/lb of brine supplied is:

$$\frac{Q_n}{w_b} = n_{turb\text{-}gen} \; x_n \left[\left(1-x_1\right)\left(1-x_2\right) - - - -\left(1-x_{n-1}\right)\right] \; (h_g-h_e)_n$$

and the total power is the sum of the stages:

$$Q_T = \sum_{1}^{n} Q_n$$

Considering, only one stage, one can see that as the stage pressure is decreased, the amount of steam generated is increased, however, as the pressure is decreased the enthalpy drop across the turbine also decreases. Thus, as the power output is a product of steam flow and enthalpy drop, a maximum will exist. The maximum output will exist when the flash points are located at a temperature giving an equal interval between the brine temperature and the sink temperature. This can be shown as follows. For the simple system of a single flash, assuming an idealized cycle:

$$Q = w_s \, (T_F - T_{cond})(S_1 - S_{cond})$$

and

$$\frac{w_s}{w_b} = x = \frac{H_b - H_f}{H_{fg}} \approx \frac{(T_b - T_f)}{\text{Constant}}$$

$$Q \approx \frac{(T_b - T_f)(T_f - T_{cond})}{\text{Constant}}$$

The maximum is found by differentiating with respect to T_f after expanding:

$$Q = \frac{T_f T_b - T_f^2 - T_{cond} T_b + T_f T_{cond}}{\text{Constant}}$$

$$\frac{dQ}{dT_f} = (T_b - 2T_f + T_{cond})(\text{Constant}) = 0$$

$$T_f \; \frac{T_b + T_{cond}}{2}$$

A similar result can be obtained for a double flash by expanding the double flash expression and taking the partial derivatives of the power with respect to the two flash temperatures and setting the result to zero and solving for the flash temperatures for the case of the double flash,

$$T_{f_1} = \frac{2T_b + T_{cond}}{3}$$

$$T_{f_2} = \frac{T_b + 2T_{cond}}{3}$$

which can be rearranged so that,

$$T_b - T_{f_1} = \frac{1}{3}\left(T_b - T_{cond}\right)$$

$$T_b - T_{f_2} = \frac{2}{3}\left(T_b - T_{cond}\right)$$

The theoretical optimum occurs when the difference between the source temperature and the condenser temperature is evenly divided. This result is based on an ideal cycle and simplifying assumptions. A calculation of maximum output that can be expected from a flash steam system, can be computed by using this result. The results are shown in Figure (1) for up to four flashes. Figure (2) shows the gross power output of a turbogenerator set at a flowrate of 10 million lb/hr. A specific power output of about 4 watt-hrs/lb of brine is possible with a four flash system, however, a double flash is probably the practical limit, at least for low temperature systems at about 3.5 watt-hrs/lb of brine. This value is the ideal condition. It does not include system losses or the effect of selecting an operating point other than the thermodynamic maximum. The sensitivity of the power output of a single flash system to a variation in the temperature of the geothermal fluid is shown in Figure 3.

For systems with more than one flash, one or more of the flash pressures will be subatmospheric if the guide lines for the thermo dynamic optimum are observed. Subatmosphere operation is generally undesirable because the large increases in specific volume of the steam requires large piping and components, plant start up is more complicated and most important, leakage of oxygen into the system may cause excessive corrosion. For these reasons, the pressure of the lowest flash stage will be maintained above atmospheric pressure.

1.2 A Comparison of Performance With Flash Pressure

A comparison of flash systems with 300^{o}F geothermal supply was made to determine the thermodynamic value of the following:

a. The sensitivity of power output to increasing the pressure of the initial flash. This information will be useful in future tradeoff studies when the use of steam from the initial flash will be considered for air-ejector drive and also in tradeoffs with the turbine design.

b. The advantage gained by reducing condenser pressure to 1.5 in. Hg abs as may be possible if a cold water aquifer can be used for condenser cooling.

c. The advantage of subatmospheric flashes. Subatmospheric flashes have not been seriously considered to date because of the corrosion problem that could result due to oxygen leakage into the system.

FIGURE 1

MAXIMUM OUTPUT EXPECTED FROM A SYSTEM OF UP TO 4 FLASHES

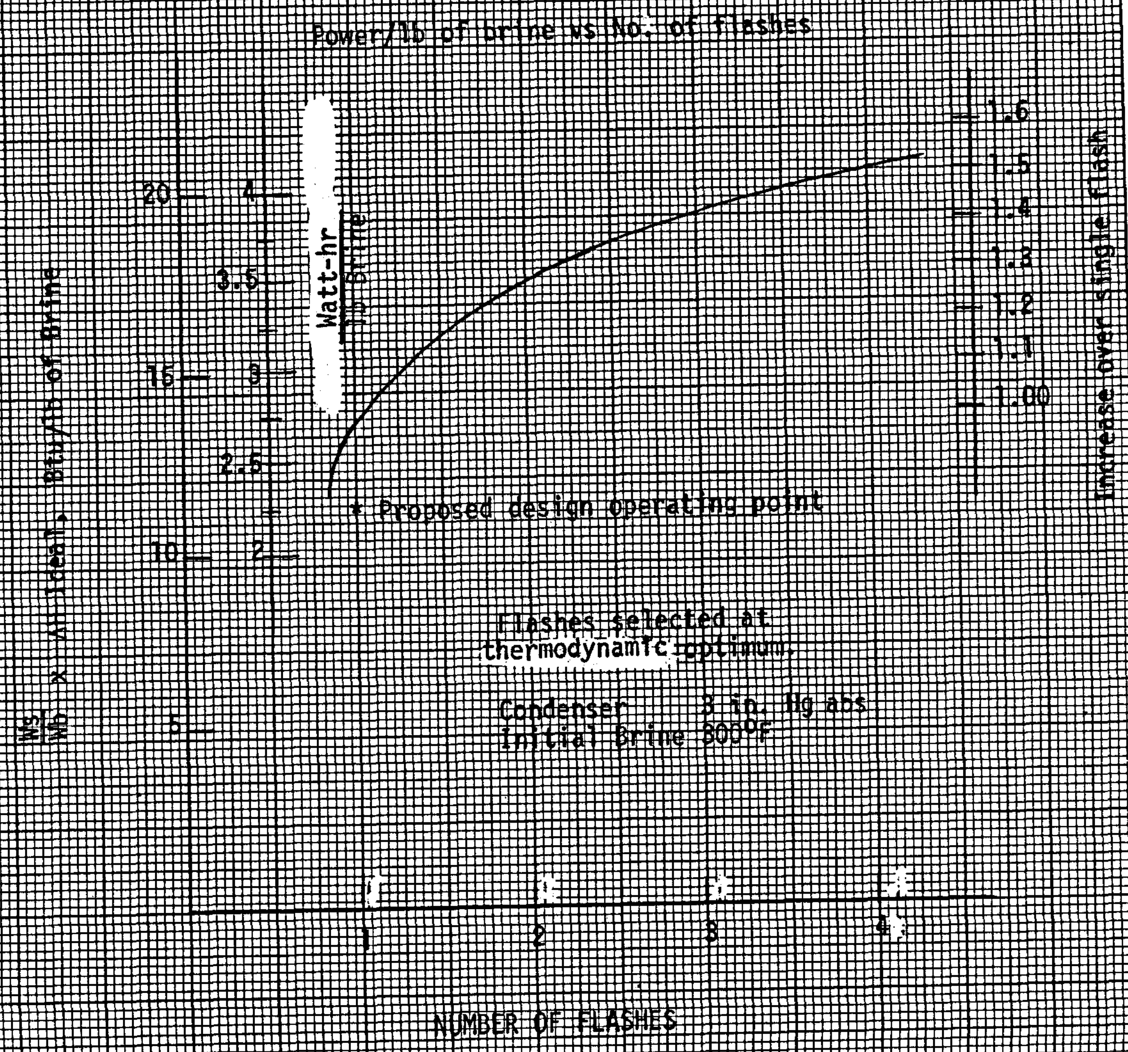


FIGURE 2

GROSS POWER OUTPUT OF TURBO GENERATOR AT
10 MILLION POUNDS/HOUR BRINE FLOW RATE

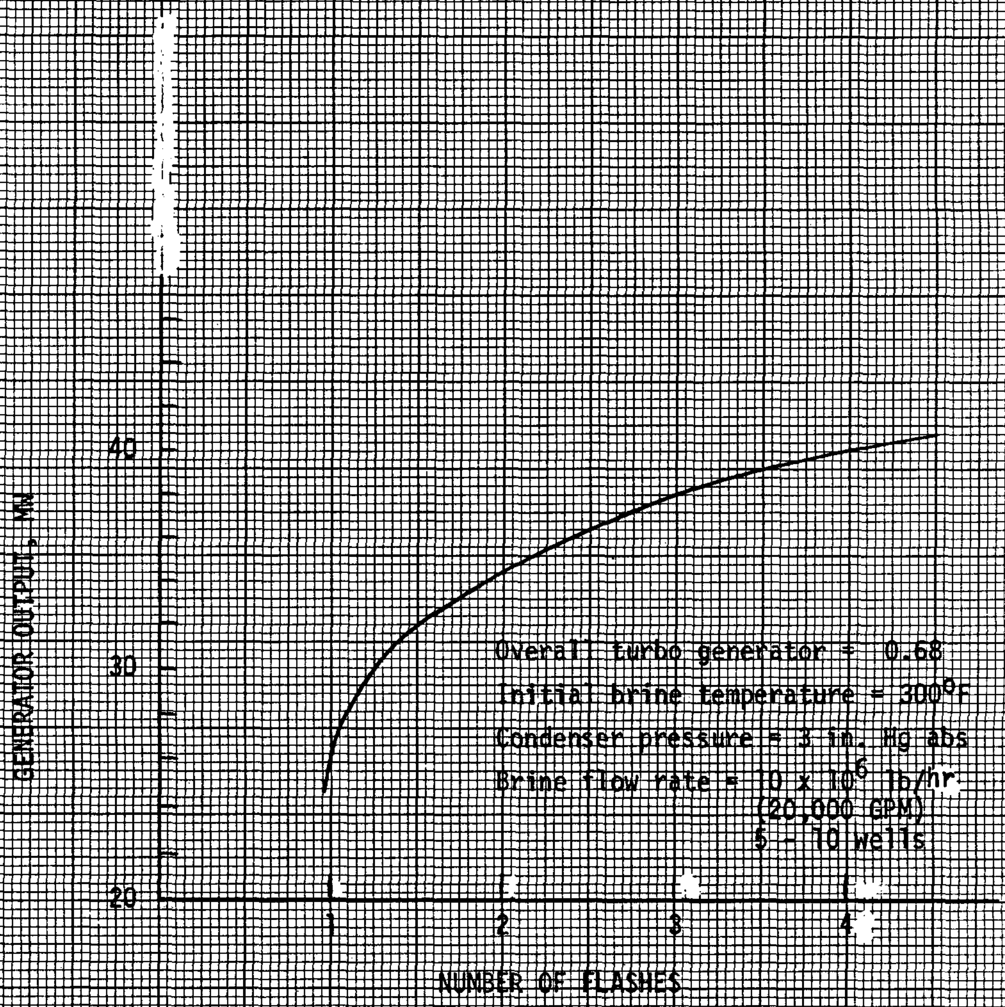
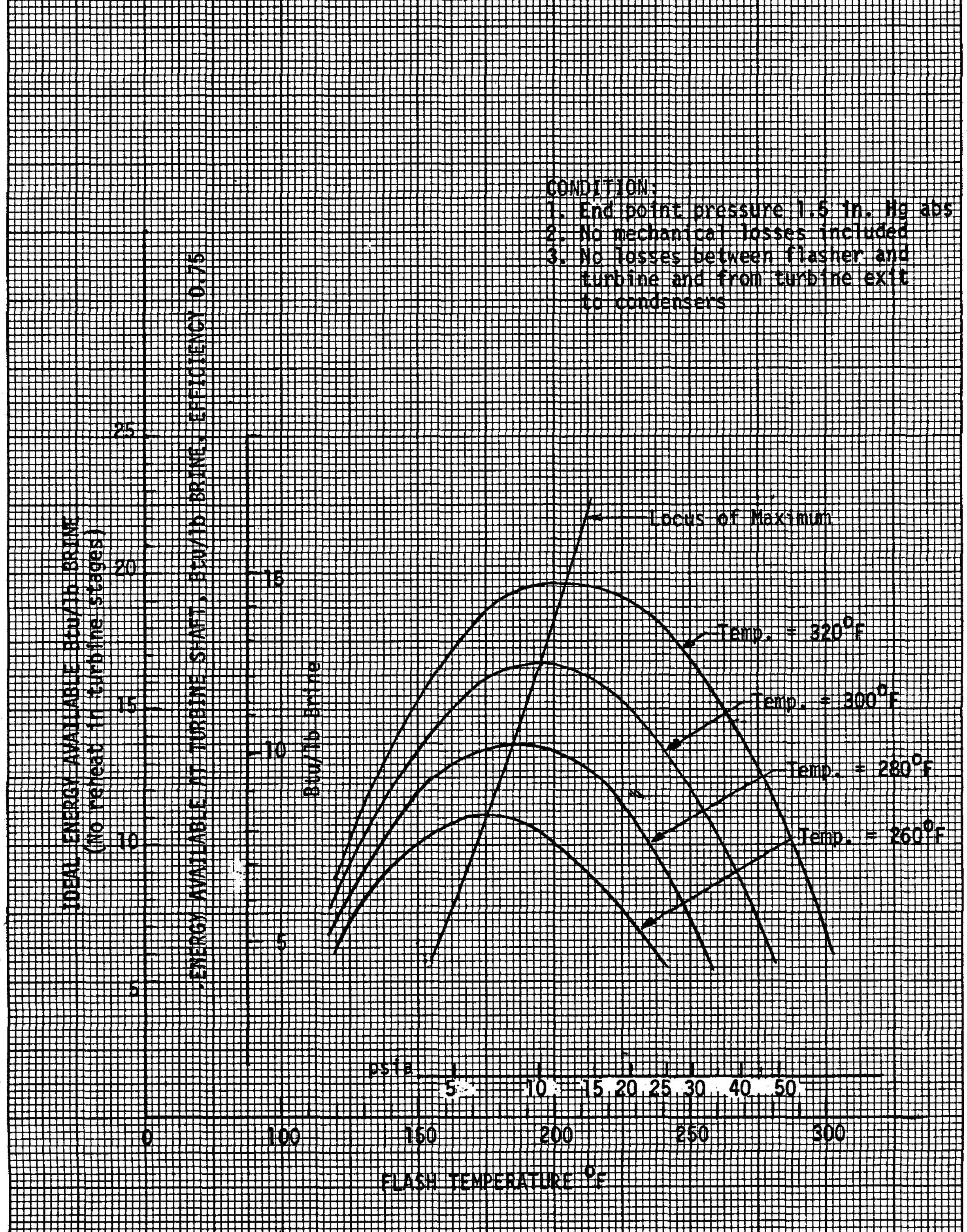


FIGURE 3

ENERGY AVAILABLE FOR PRODUCTION OF POWER AS FUNCTION OF INITIAL BRINE TEMPERATURE AND SINGLE FLASH TEMPERATURE



The results of this work is summarized in Table I. The cycle power is given in terms of watt-hrs/lb of brine (equivalent to MW/million lb/hr of brine). This power is based on a combined turbine and generator efficiency of 68%. It does not include system losses.

Cases 1 through 5 indicate the sensitivity to the pressure/temperature selected for the first flash of a dual flash system when the pressure of the second flash is held essentially constant. Case 4 is the reference case. The power output is relatively insensitive to the pressure of the first flash over the range considered. Case 6 shows the advantage (21% increase) that could be obtained by reducing the condenser pressure to 1.5 in. Hg abs as may be possible if the cold water aquifer can be used for cooling.

The advantage of a subatmospheric second flash is shown in case 7 (compare with case 5) where an 11% improvement occurred. The reduction in output (approximately 12%) caused by increasing the second flash pressure is shown by comparing cases 8 and 9 with cases 1 and 2, respectively. Some multiflash (4) results are shown in cases 10 through 13. Comparing case 10 with 4 shows only a 6% improvement is obtained by the addition of two flash stages above atmospheric pressure. Subatmospheric flashes combined with low condenser pressure yield 4.93 $MW/10^6$ lb/hr of brine, a 50% improvement over the referenced case. Cases 11 and 12 utilize two condensers in series to reduce cooling water requirements.

1.3 System and Equipment Analysis Base

The following sections present some of the major basis on which performance studies were made. In particular, the single and double flash systems discussed later are based on the information and assumptions in this section.

1.3.1 Basic Assumptions and Equations

1.3.1.1 Flash Process

The flash process was assumed to be an isenthalpic process where flashing is assumed to occur across a throttling valve. The geothermal brine was considered to be pure water and therefore, any effects of dissolved gases and minerals were ignored.

For simplicity in all calculations it was assumed that 100% quality steam is produced and that flasher/separator pressure and turbine inlet pressure are the same.

1.3.1.2 Turbine Process

For all calculations a stage efficiency of $\eta_S = 0.8$ was used. The assumptions are; turbine steam inlet pressure is 100%, flash pressure and turbine inlet pressure are equal.

TABLE I

Comparison of Various Flash Cases

Case No.	No. Flashes	Flash Press/Temp psia/^{o}F	No. Cond.	Cond. Pressure in. Hg abs	Watt-hrs lb Brine
1	2	40/267 14/209	1	3	3.19
2	2	35/259 14/209	1	3	3.21
3	2	30/250 14/209	1	3	3.22
4	2	30/250 13/206	1	3	3.28 Ref. Case 7% less than Case 7
5	2	25/240 14/209	1	3	3.18
6	2	30/250 13/206	1	1.5	4.0 21% better than case 4
7	2	25/240 7/177	1	3	3.52 best thermo. at 3 in. Hg
8	2	40/267 20/227	1	3	2.85
9	2	35/259 20/257	1	3	2.85
10	4	40/267, 31/252 22/233, 13/206	1	3	3.49 All flashes above atmospheric pressure
11	4	46/276, 31/252 20/228, 12.5/204	2	4 1.5	3.97
12	4	40/267, 23/235 12/202, 6/170	2	4 1.5	4.29
13	4	40/267, 23/235 12/202, 6/170	1	1.5	4.93

Mechanical efficiency, η_m, was assumed in the order of 94% and the generator was assumed to have an overall efficiency, η_g, of 90%. The overall efficiency factor to be applied to the isentropic work is:

$$\eta_o = \eta_s \times \eta_m \times \eta_g = 0.8 \times 0.94 \times 0.9$$
$$= 0.68$$

1.3.1.3 Separator Performance and Size Requirements

Reference (1) was used as the basis for separator performance evaluation and separator sizing. This reference provided test results on a cyclone separator with a bottom steam outlet used in the geothermal plant in Wairakei, New Zealand. The report indicates that with steam velocities of up to 180 ft/sec at the separator inlet that moisture can be maintained less than 0.1%. Separator size was given in the reference in terms of the inlet diameter. Figure 4 reproduces a sketch of the separator dimension in terms of inlet pipe diameter. No detailed calculations of performance were made for this study. The separator was sized solely on the basis of the referenced report.

1.3.1.4 Effect of Flash Pressure on Flasher/Separator Size

The size of the flasher is assumed to be only a function of the inlet pipe diameter (Ref. 1). The inlet pipe diameter, assuming a given velocity is defined by the specific volume of the mixtures. Assuming a homogeneous mixture, the variation in specific volume as a function of flash pressure is shown in Figure 5. The diameter of the separator inlet is defined by the equation:

$$D_i = \sqrt{\frac{4}{\pi} \frac{w_b \nu_i}{V_i}}$$

Since w_b and V_i are functions of flash pressure the flasher inlet pipe diameter can be determined with respect to pressure for one flasher handling full flow. Combining this with the flasher criteria of $D_{fl} = 3\ D_i$ and $h_{fl} = 11.5\ D_i$ results in flasher volume $V_{fl} = 81.28\ D_i^3$. Figure 6 showed the volume and surface area of a single flasher as a function of pressure. The sizes are generally extreme for a single unit.

1.3.1.5 Condenser

All sizing studies in this report assumed the use of a barometric condenser. References (2) and (3) were used to establish the size of the barometric condenser. The heat released to the condensing water is given by:

$$Q_C = w_s \left(H_e - H_4\right)$$

The load due to cooling of non-condensables was neglected as well as the load taken by the air ejectors.

The gas content of the steam will affect the cooling requirements due to its affect on the total pressure within the condenser. Gas content in geothermal steam varies between 1/3% to 10% or more (Ref. 4) and is mainly CO_2, H_2S and NH_3, which have molecular weights of about 44, 34, and 17, respectively. Air has a molecular weight of 28.9. Since the type and amount of the dissolved gases is not known, air will be used as the basis and assumed to be in amount of 10% of steam by weight. Molecular weight of steam is 18:

$$\frac{P_v}{P_e} = \frac{\text{partial pressure of vapor}}{\text{total pressure of mixture}} = \frac{\text{moles of condensables}}{\text{moles of mixture}}$$

For: $$w_m = w_s + w_{ncd} = w_s + 0.1\ w_s$$

$$\frac{P_v}{P_e} = \frac{w_S/18}{\frac{w_S}{18} + 1.0\,\frac{w_S}{28.97}} = \frac{1}{1 + \frac{0.1 \times 18}{28.97}} = 0.941$$

$$P_v = 0.941\ P_e$$

where: P_e is the suction pressure to the condenser (corresponds essentially to turbine end point).

The value of T_v corresponding to P_v is the temperature to which the cooling water can be heated theoretically. Assuming a significant amount of non-condensables and a well designed condenser the minimum temperature difference between the vapor temperature and the cooling water is 5°F. Thus, the cooling water outlet temperature is:

$$T_C = T_{s_e} - 5$$

$$Q_C = w_C\ C_{p_c}\ (T_{Co} - T_{Ci})$$

Equating for $C_{p_c} \approx 1$:

$$w_c / w_s = \frac{H_e - H_4}{T_{co} - T_{ci}}$$

For all calculations a cooling water inlet temperature of 75°F was assumed.

The maximum inlet velocities of saturated steam to the condenser is given by Reference (2) as a function of suction pressure, according to the equation:

$$V^2 \rho = 400$$

since:

$$V = 4 \frac{w_s \nu_{e_T}}{\pi D^2}$$

$$\frac{16 w_s^2 \nu_e^2}{\pi^2 D_i^4} \times \frac{1}{\nu_e} = 400$$

$$D_i \approx \frac{1}{4} w_s^{1/2} \nu_e^{1/4}$$

where:

w_s = lb/sec

D_i = ft.

The diameter of the main body of the condenser should be about 1.6 times as large as the vapor inlet diameter and the length to diameter ratio, L/D = 2 (Reference 3).

The minimum effective tail pipe height based on 5000 ft elevation:

$$L = \frac{P_s \times 144}{\rho} \approx 28.2 \text{ ft, (29 ft)}$$

Additional considerations are:

(1) The tail pipe must extend at least one foot below the hot well level and not less than D/2 above the bottom of the hot well.

(2) The hot well must contain sufficient excess fluid to fill the tail pipe during startup.

(3) The hot well size was based upon a hot well area to tail pipe area ratio of 40.

(4) Cooling water inlet pipe area based on a velocity of 6 ft/sec.

FIGURE 4

RECOMMENDED PROPORTION OF BOTTOM OUTLET CYCLONE SEPARATOR WITH SPIRAL INLET

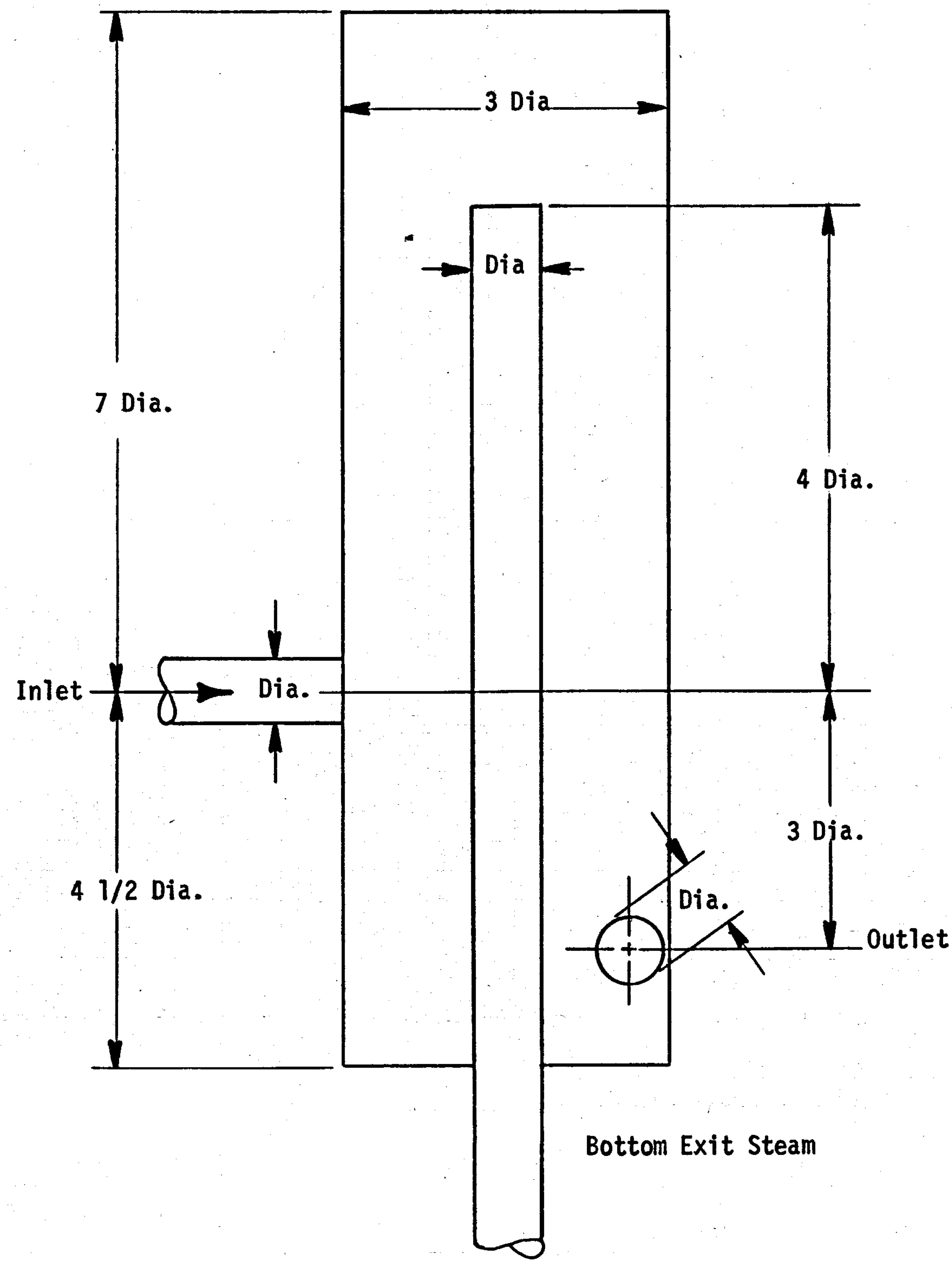


FIGURE 5

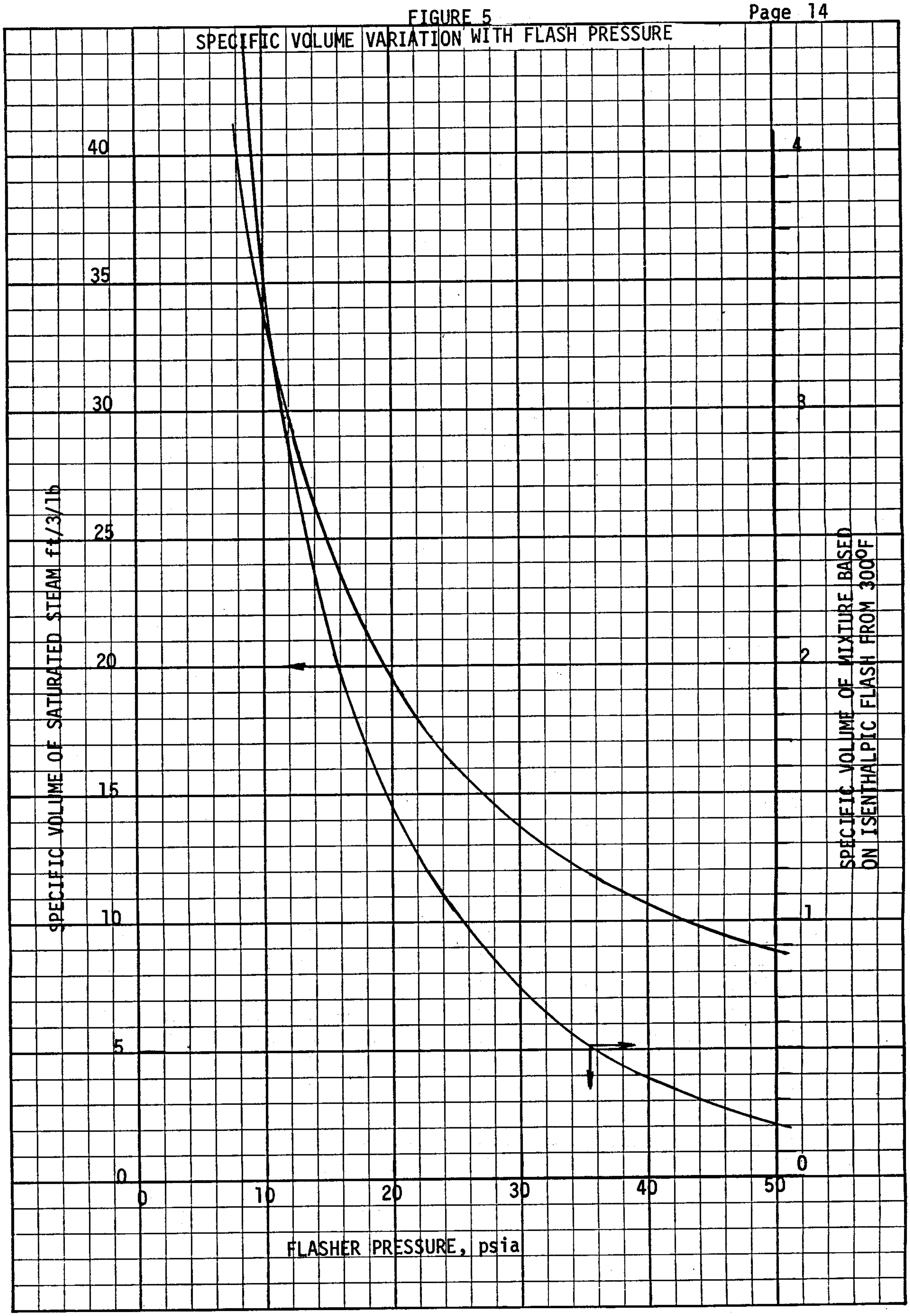
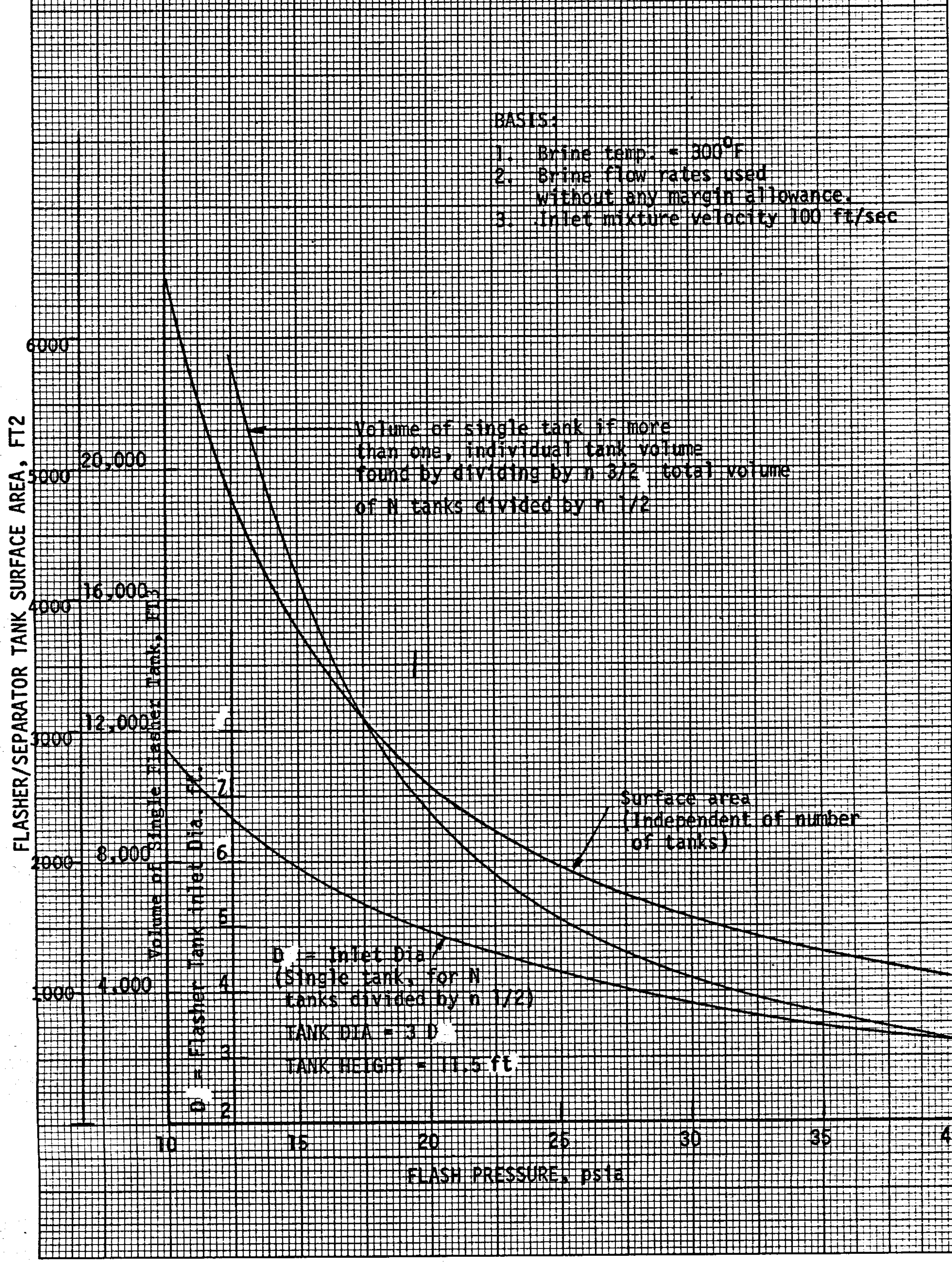

SPECIFIC VOLUME VARIATION WITH FLASH PRESSURE
SPECIFIC VOLUME OF SATURATED STEAM ft^3/lb
40
35
30
25
20
15
10
5
0
0
10
20
30
40
50
FLASHER PRESSURE, psia
SPECIFIC VOLUME OF MIXTURE BASED ON ISENTHALPIC FLASH FROM 300°F
4
3
2
1
0

FIGURE 6

SEPARATOR TANK VOLUME AND SURFACE AREA
AS A FUNCTION OF FLASH PRESSURE



1.3.1.6 Geothermal Wells and Brine Pumping Power

At the present time the location and depth of the geothermal supply and reinjection wells is not known. Layouts have been made which indicate that an average supply well will be located 0.4 miles from the power house and the reinjection wells located a maximum of 9 miles from the power house. The wells are assumed to be at least 4000 ft deep.

Because of the temperature difference between the supply and return wells a driving force potentially exists, $(\rho_H - \rho_C)h$, to force the hot water to the surface. In all calculations it has been assumed that no pumping power is required to either lift the supply water from the well or reinject it. The only pumping power requirement for the brine are those associated with friction and elevation between the wells and the power house.

In some calculations of pumping power the following assumptions were made:

(1) Supply and return lines were lumped together in one run of 49,620 ft. A +60 ft change in elevation was assumed. Equivalent length of fittings and valves was assumed 10% of the total pipe run. A friction factor of f = 0.013 was assumed.

(2) A pump efficiency of 85% was assumed and a motor efficiency of 93% (79% overall).

(3) Economic pipe diameter was based upon the equation:

$$d = 0.611\left(\frac{w_b}{1000}\right)^{.45}$$

where: d = in.

w_b = lb/hr

This equation was obtained from Reference (5) and may not reflect current economic trends.

1.3.1.7 Cooling Water Wells and Cooling Water Pumping Power Assumptions

Wells into the aquifer can provide water at about 50^{o}F. This low temperature is desirable to increase the efficiency of the cycle. Calculations involving the cooling water pumping power requirement when cooling water is obtained from wells are based on the following:

(1) Supply well lift = 400 ft

(2) Excess pressure at reinjection well head = 200 head ft

(3) Friction factor, f = 0.013, distance 10,300 ft, 10% allowance for fittings and valves.

(4) Pump and motor efficiencies of 85 and 93%, respectively. Pumping powers in the order of 2800 kW result.

2.0 Description of a Single Flash Steam Plant

The major equipment of a single flash steam plant consists of wells, pumps, piping, valves, flasher/separators, steam turbine, generator, condenser cooling tower and auxiliary equipment. The primary state point variables are flash pressure and condenser pressure. This section is intended to present a view of the sensitivity of system performance to a variation in the main state points and a feel for the sensitivity of equipment size to the primary variable, flash pressure, associated with a single flash electrical power plant with a gross output of 11 MW derived from 300^{o}F geothermal water.

Although it is unlikely that a single flash system would be considered solely electrical power production (in view of higher performance of multi-flash system) it may interface well with non-electric processes.

2.1 System Performance Sensitivity

As the flash pressure of the system is lowered from the saturation pressure corresponding to the geothermal brine temperature an increasing fraction of the geothermal fluid is flashed into steam. However, as the pressure decreases, the available enthalpy drop across the turbine also decreases. These relationships are shown in Figure (7). Since the total output power is a product of the steam flow rate and enthalpy drop a maximum exists as shown on Figure (3) as a function of temperature and Figure (8) as a function of pressure. That the potential plant output is quite sensitive to the condenser is also shown on Figure (8) where the gross output varies between 2 and 3.5 MW/million/pounds/hr of brine flow. The magnitude of the brine and steam flow rate as a function of flash and condenser pressure is given in Figure (9) and (10) for a 11 MW (gross) plant. The brine flow requirements increase with an increasing rate as flash pressure is raised.

A significant consideration in selecting an operating point is the cooling water requirements or its trade off with condenser pressure. It is evident from the preceding figures, that the back pressure significantly affects the power output, however, excessive cooling requirements must be avoided. Figure (11)indicates the cooling water flow(in terms of its ratio to steam flow) as a function of condenser pressure for a 11 MW plant. The information is shown for two temperatures; 75^{o}F, representative of that obtainable from a wet cooling tower and 50^{o}F which is representative of the best that could be achieved with once through cooling supplied by a cold water aquifer. The higher temperature implies a condenser pressure in the range of 3 in. to 4 in. Hg abs to avoid excessive cooling water flow. The cooling water is not sensitive to flash pressure as shown in Figure (12).

2.2 System Description and Component Sensitivity

Figure 13 shows a composite schematic representation of the system. To limit the size of the equipment, three flasher/separator and hot well units were selected to handle full load requirements. An additional assembly including flasher, separator, hot well, isolation valves, and brine return pump are provided as a spare. The spare unit is considered desirable at this time in anticipation of potential maintenance problems due to mineral accumulation. The design conditions for the system are as shown in Table II.

A 30 inch header collects the brine from the various wells and supplies the flasher separator units. Each unit can be valved off from the supply header. The distribution lines to the flasher valve are 15 inch and include an isolation valve. The lines between the flasher valve and the separator tank are 30 inch. This line enters the tank at a tangent so that a cyclone action is produced to separate the steam from the water. The steam rises to the upper portion of the separator where it turns and enters a 30 inch steam pipe which enters the bottom of the tank and raises to 7 ft of the top of the tank. The steam leaves the flasher through the 30 inch line that contains an isolation valve, and enters a 48 inch steam header.

The steam is then piped to the turbine. The main steam line contains a fast-closing isolation valve and a throttle valve (butterfly type) to protect the turbine from over-speed. Steam velocity is 150 ft/sec in the 48 inch line. The steam exhaust enters the barometric condenser with approximately 10% moisture at a velocity of 300 ft/sec. Cooling water from a cooling tower enters spray jets in the condenser and condenses the exhaust steam. The condensate and cooling water leave the condenser through the 48 inch condenser tail pipe and enter the condenser hot well from which it is recirculated to the cooling tower.

Within the separator the waste water drops to the bottom of the tank and flows by gravity to an associated hot well tank. Pumps which take suction from the hot well tank feed the 30 inch brine return header which is connected to the 30 inch line to the reinjection wells.

A flasher/separator operating pressure of 25 psia was selected based on an attempt to minimize brine flow requirements while limiting equipment size. The thermodynamic optimum of 10.6 psia requires very large equipment. An operating pressure in the range of 20 to 35 psia appears reasonable. The value of 25 psia is at this time, an arbitrary choice.

At 25 psia a turbine inlet pressure of 300,000 lb/hr of steam is required to produce 11,000 kW with a condenser pressure of 3 in. Hg abs (see Figure 10). Allowing a 10% margin for auxiliary use, overload, etc., raised the total product at the flashers to 330,000 lb/hr. The selection of condenser pressure will be discussed later. The brine flow required is 5.1×10^6 lb/hr (Figure 8). Referring to Figure 8, note that the selection of 25 psia operating pressure is to the right of the maximum and, therefore, if not limited by equipment, power output can be increased by lowering flasher pressure.

The ratio of condenser cooling water flow to steam flow varies with condenser pressure and cooling temperature. The variation with turbine inlet pressure is small (see Figures 11 and 12).

Cooling water for the condenser will be provided by a cooling tower. Twenty one thousand gpm at 75°F will be required. Use of a cooling tower rather than the 50°F aquifer is based on pumping costs. The cycle advantage of aquifer cooling is apparent in Figure 11. Cooling water wells would be at least 400 ft deep. For reinjection an added pressure of 200 ft head is required at the well. Thus, a 600 ft head requirement exists without considering frictional losses. In Figure 11, the cooling water to steam flow ratio is 15 to 20 ($4.95 - 6.6 \times 10^6$ lb/hr) resulting in pumping power requirements of 2000 kW. Based on suppliers information approximately 750 kW will be consumed by a cooling tower, excluding power to circulate water (~100 kW).

FIGURE 7

FRACTION OF GEOTHERMAL BRINE FLASHED TO STEAM AS A FUNCTION OF FLASH PRESSURE

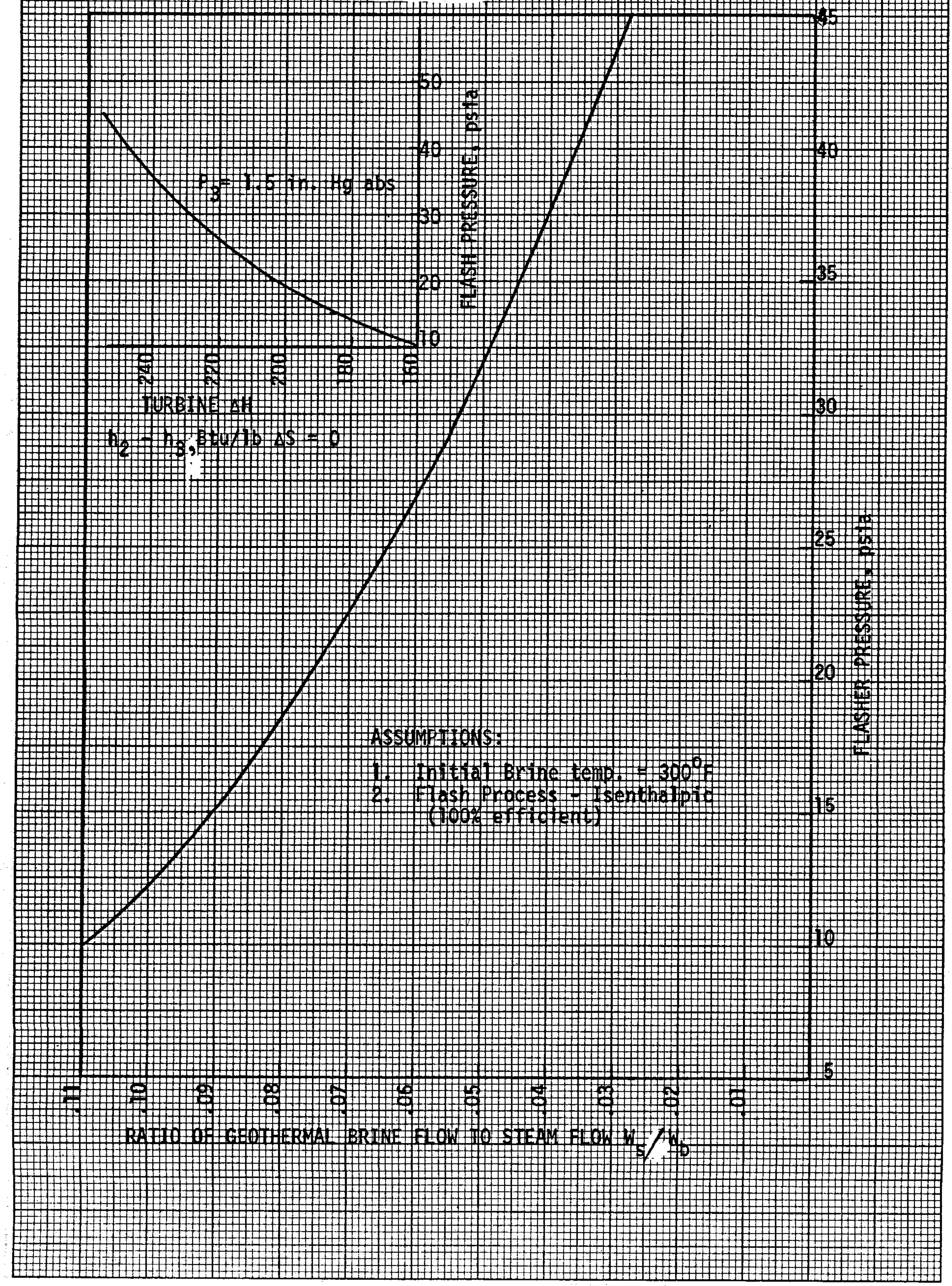
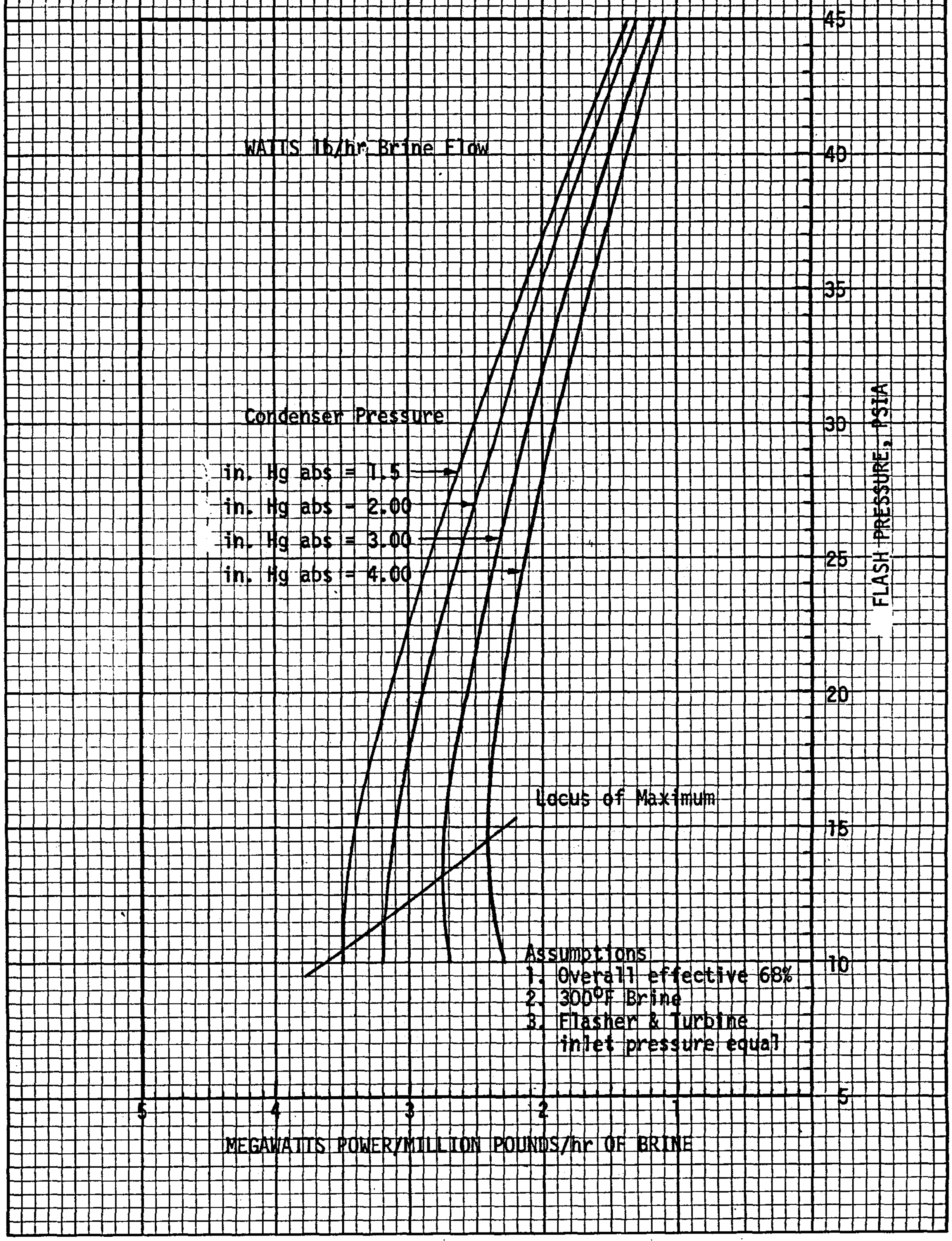


FIGURE 8

POWER GENERATION AS A FUNCTION OF CONDENSER AND FLASH PRESSURE AND BRINE FLOW



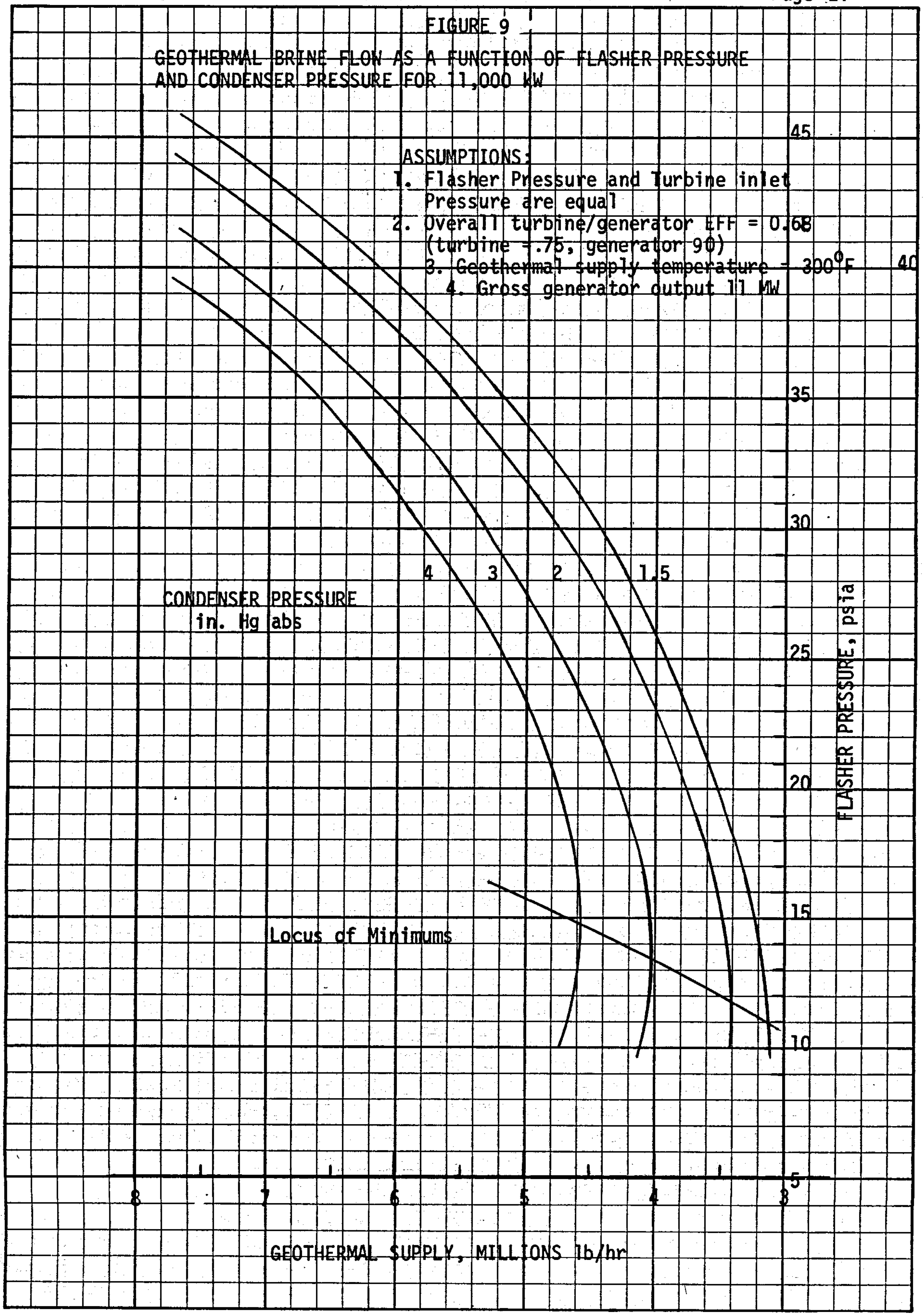


FIGURE 9
GEOTHERMAL BRINE FLOW AS A FUNCTION OF FLASHER PRESSURE AND CONDENSER PRESSURE FOR 11,000 kW
ASSUMPTIONS:
1. Flasher Pressure and Turbine inlet Pressure are equal
2. Overall turbine/generator EFF = 0.68 (turbine = .75, generator 90)
3. Geothermal supply temperature = 300°F
4. Gross generator output 11 MW
CONDENSER PRESSURE in. Hg abs
4
3
2
1.5
Locus of Minimums
45
40
35
30
25
20
15
10
5
FLASHER PRESSURE, psia
8
7
6
5
4
3
GEOTHERMAL SUPPLY, MILLIONS lb/hr

FIGURE 10

STEAM FLOW REQUIRED TO PRODUCE 11,000 kW GROSS

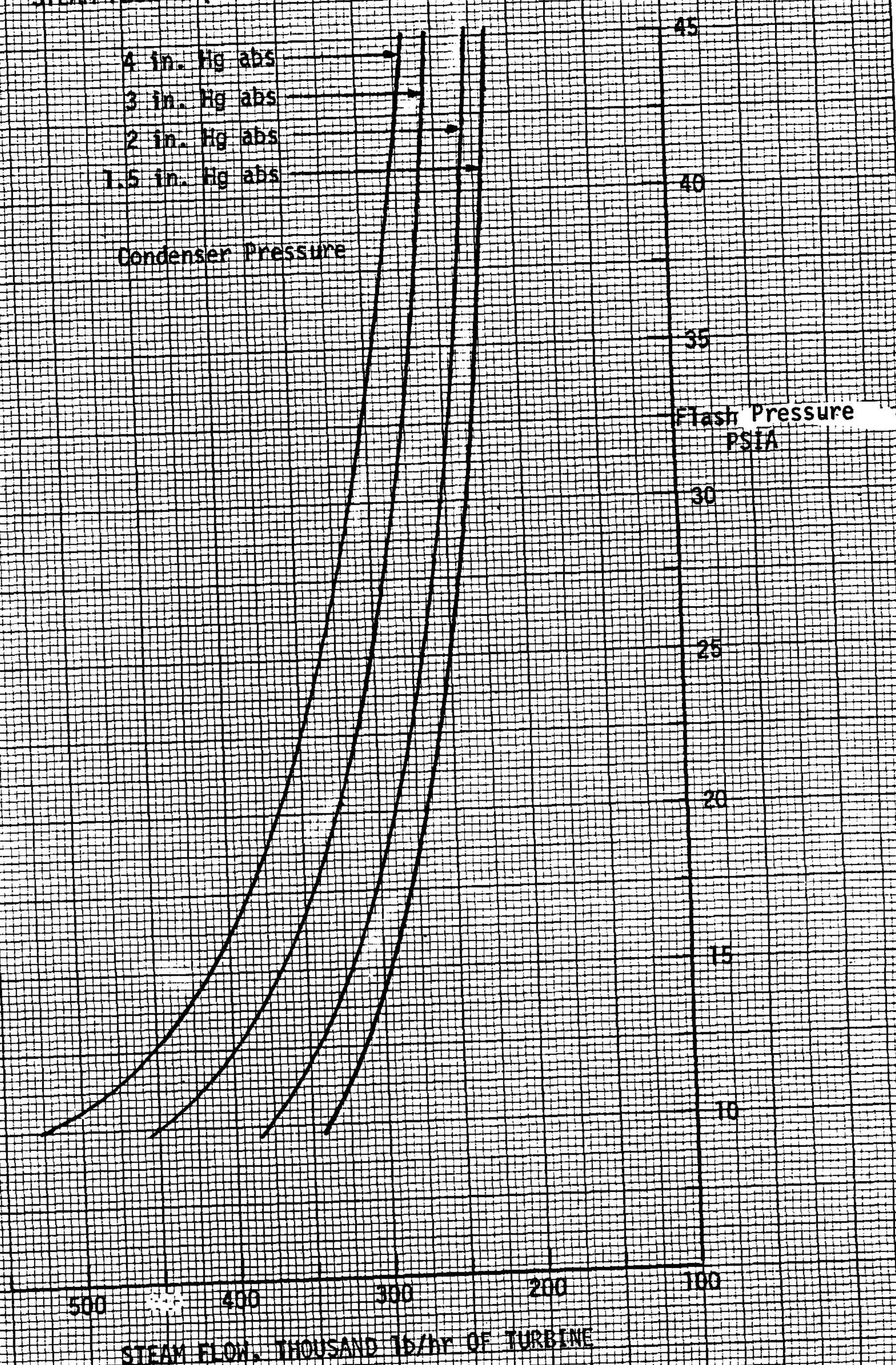


FIGURE 11

RATIO OF COOLING WATER FLOW TO STEAM FLOW
AS A FUNCTION OF CONDENSER PRESSURE

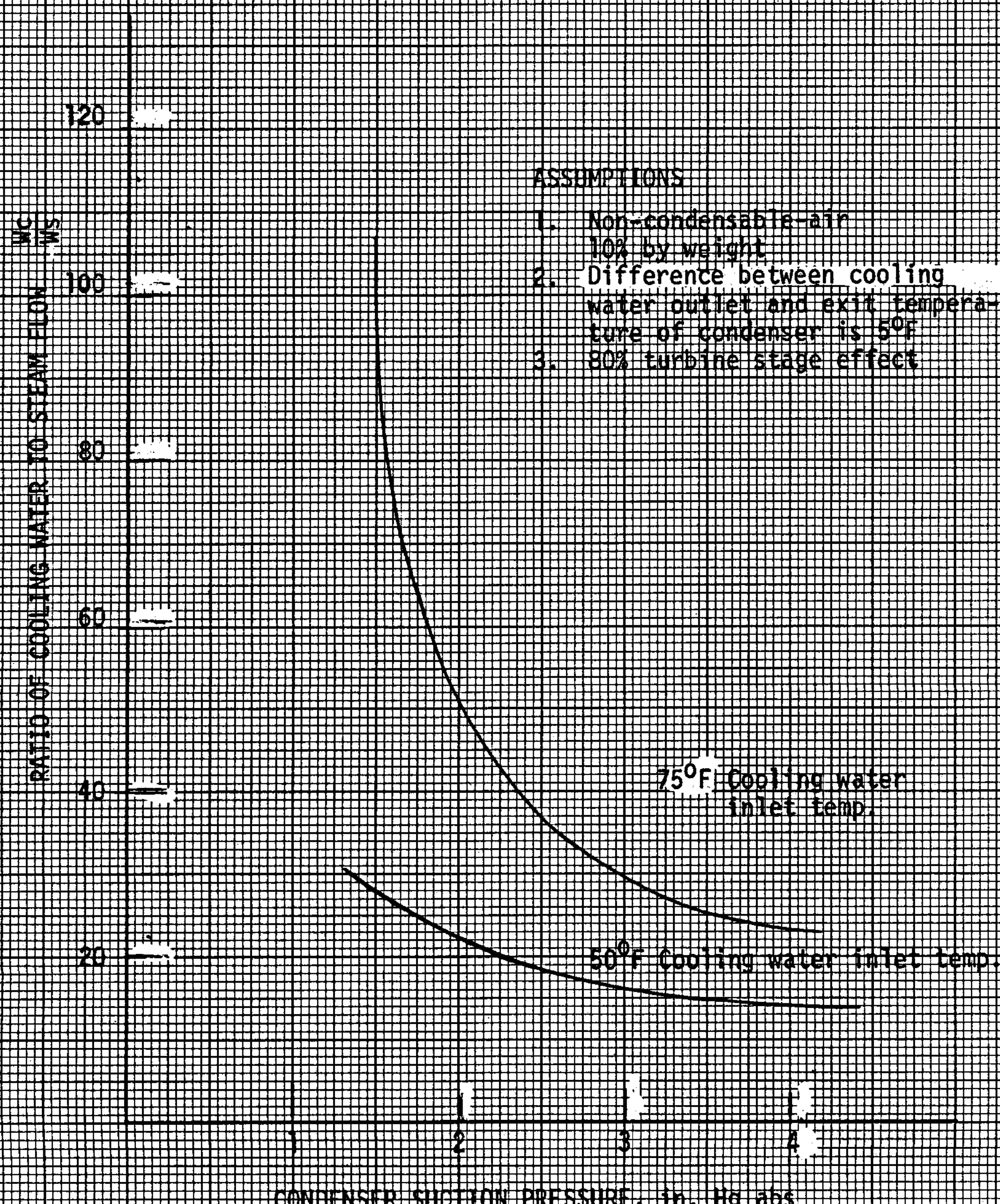


FIGURE 12

SENSITIVITY OF COOLING WATER REQUIREMENTS TO FLASHER AND CONDENSER PRESSURE

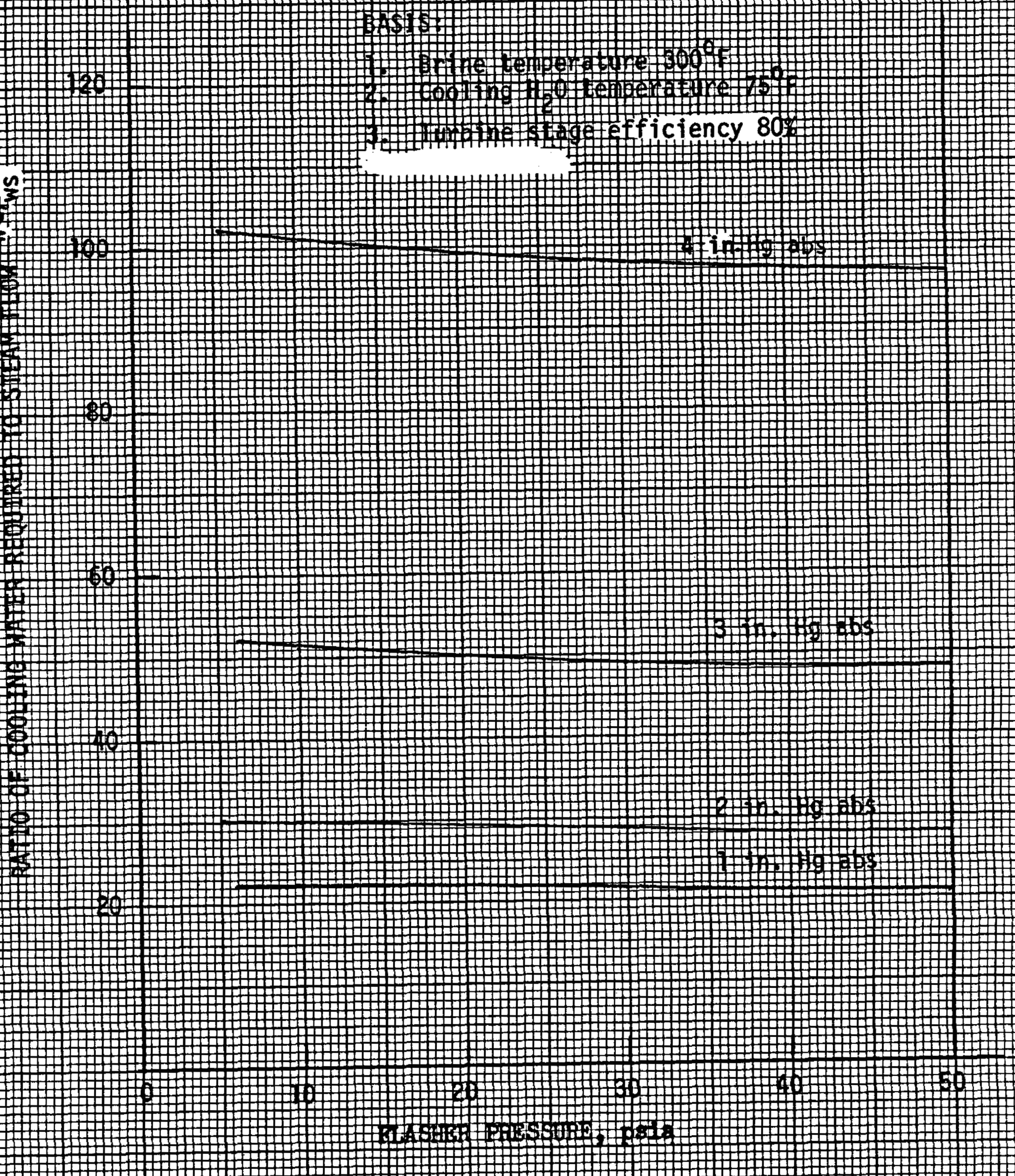


TABLE II

Single Flash Steam Parameter List

Gross electrical power, kW	11,000
Btu/hr	37,500,000
Geothermal brine, supply	
Number of supply wells required at 10^6 lb/hr/well	~5
Average distance from well to power house, ft	2100
Elevation difference between well and power house	0
Pump requirements (each) head, ft	155
flow, gpm	2000
Temperature, ^oF	300
Flow rate, lb/hr (total design)	5.1×10^6
Normal operating, lb/hr (100% power)	4.65×10^6
Geothermal brine, return	
Flow total, lb/hr design	4.77×10^6
Separator tank outlet, lb/hr	1.59×10^6
Temperature	140^oF
Distance between power house and reinjection wells, ft	47,520
Elevation difference between well and power house, ft	0
Pump requirements (each) head, ft	180
gpm	3400
Flasher/separator	
Number required for full power	3
Pressure of flasher valve inlet, min. psi	58
Temperature, flasher valve inlet, ^oF	300
Flow rate, lb/hr, each design	1.7×10^6
Separator pressure, 100% power, ^oF	240

TABLE II (cont'd)

SINGLE FLASH STEAM PARAMETER LIST

Steam flow conditions	
Turbine flow, 100% power, lb/hr	300,000
Design (max. output 3 flashers), lb/hr	330,000
Pressure, psia	25
Steam Quality, %	99.5
Condenser conditions	
Condenser suction power, 11 in. Hg abs	3
Inlet quality, %, max.	12
Non-condensables (air), % by weight	10
Partial pressure of vapor, in. Hg abs	2.82
Temperature, cooling water out, ^{o}F	108
Temperature, cooling water inlet, ^{o}F	75
Cooling water flow rate, gpm	21,000
Effective height of tail pipe, ft	29

FIGURE 13
SINGLE FLASH SYSTEM SCHEMATIC

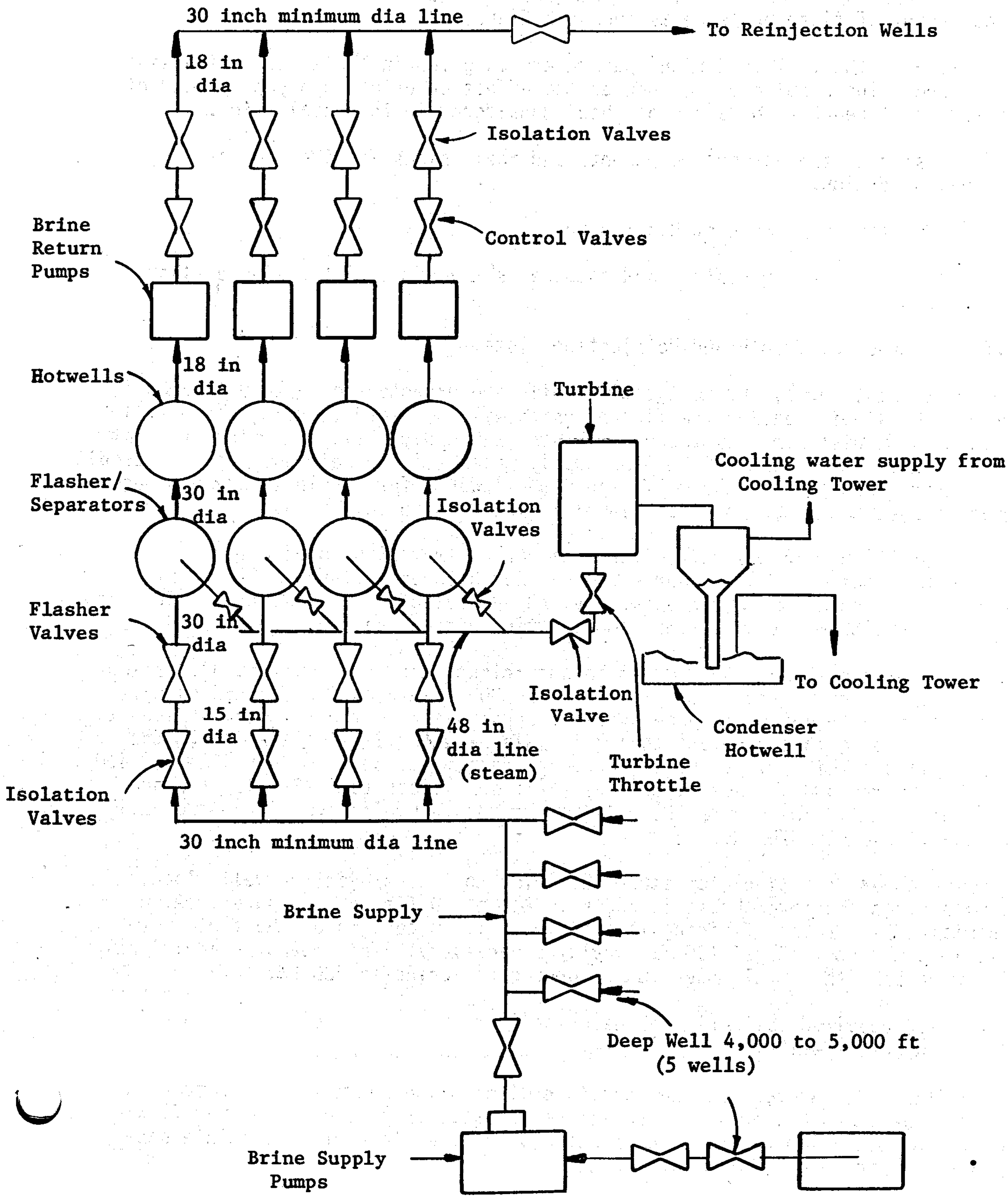


Figure 11 shows the cooling water requirements as a function of condenser pressure. From this figure it is evident that for 75°F water a condenser pressure of 3 to 4 in. Hg abs is reasonable. The cooling water requirements are essentially independent of flash pressure as shown in Figure 12.

A summary of the full load plant parameters is given in Table II. Off design point operating requirements, such as the effect on brine supply pump head of operating at reduced loads has not been considered in this evaluation.

A discussion of the various components and their sizes is given in the following section.

2.3 Subsystem and Component Description

The following major subsystems and components are parts of the single flash steam plant.

2.3.1 Geothermal Supply and Reinjection System

The geothermal supply and reinjection wells are expected to be 4000 to 6000 ft deep. The liquid leaving the flasher units will be piped to reinjection wells located some distance from the power house and supply wells and reinjected into the ground. It is assumed that the supply wells will be self-pumping. No credit for natural pressure at the well head nor allowance for supplementary pumping power to get the water to ground level is included in this study.

The land altitude in the Raft River area varies from 5000 to 5100 feet. The location of the wells and power house has not been established. Since this study is for comparative purposes only and any selection of elevation differences would be arbitrary at this time, no elevation differences were considered.

Pressure at the flasher inlet must be sufficient to prevent premature flashing in the pipe line. Saturation pressure for 300°F fluid is 67 psia (55 psig). The considered pumping distance from the supply well to the flasher is 2100 ft. Fifty five psig is equivalent to 127 ft, allowing for 20 ft elevation in the power plant. This brings the total, (excluding friction), to approximately 150 ft which requires 360 kW (see Figure 14). Friction losses for 1/2 mile will add an additional 12 kW (Figure 15) raising the total pumping power from the well to the flasher to 375 kW.

Pumping power (friction) to return the brine to the reinjection wells located 9 miles from the power house is 250 kW. Again, sufficient pressure must be maintained to prevent flashing in the line. If no cooling of the fluid was assumed, at least 13 psi (30 ft) would be necessary. This means an additional 30 kW raising the total power for returning the brine to 280 kW.

The power required to transport the brine from the supply well and return it to the reinjection wells will be 655 kW.

Each flasher/separator unit was provided with an independent brine return pump feeding a common header. This selection is based on providing maximum flexibility of operation of the power plant and consideration of pump maintenance problems which at this time are unknown.

number of supply wells that will be required is unknown at this time; therefore, will be assumed that a minimum of 200 gpm will be provided by each well which implies 5 wells and thus 5 supply pumps each rated at 2000 gpm and 155 ft of head. The return pumps each handle 1/3 of the brine or 3350 gpm each. Their operating head is approximately 185 ft.

2.3.2 Flasher/Separator System

The flasher/separator system consists of a flash control valve and a tank in which steam water separation takes place; the cylindrical tank is 8 ft in diameter and 30 ft high.

The flasher design is based solely on Reference (1). Thus, the diameter and height are a direct function of the inlet pipe size which in turn is strongly affected by the specific volume of the flashed mixture (which is a function of flash pressure) and the allowable inlet velocity. An inlet velocity of 100 ft/sec was selected for the design point. This velocity provides a sufficient margin for operation at pressures less than design pressure without decreasing the steam quality. Figure 6 which shows the flasher volume and surface area variation with flash pressure was considered in the selection of the 25 psia operating point. The surface area of the system required, increases sharply at pressures below 20 psia. It was thought that cost would be minimized if the flasher/separator unit was of a size that would not require on-site fabrication. When the fraction of cost represented by the flasher/separator is better known the operating point could be reevaluated. The decision to provide a spare unit, and keep the units of a size that could be easily shipped overland, resulted in the selection of three units, plus one spare. Each unit is designed to handle 1.73×10^6 lb/hr of brine. The variation of flasher/separator size with brine flow and operating pressure is shown in Figure 16. The nominal size of the flasher/separator has an 8 ft diameter by 30 ft height.

2.3.3 Brine Hot Well Tank

Each separator is provided with a separate tank to provide additional water storage in the system. The tanks are 8 ft in diameter by 18 ft in height. The water level in the tanks is monitored and provides input to the brine return control valves. There is sufficient water stored in the tanks to provide 15 seconds to shut the system down in the event of a brine supply failure. In an actual system it is likely that a large single tank would be provided rather than the single units indicated.

2.3.4 Turbine-Generator

The turbine-generator unit sizes have been discussed with a commercial supplier to determine the size of a turbine-generator set over the range of operating pressures from 10 psia to 30 psia. All proposed units were double flow with sizes listed below as a function of inlet pressure.

psia	Steam rate, lb/hr	TYPE	Approximate length of last bucket, inches
10	445,000	2 stages, Double flow (4 rows)	14
20	304,000	3 stages, Double flow	11
30	257,000	4 stages, Double flow (8 rows)	10

noted the turbines are all double flow. A single flow unit would be quite large. At 10 psia the inlet pipe size becomes the limitation since the largest size that can be accommodated is two 36 inch diameter lines. A significantly more economic unit would result with the 20 to 30 psia pressure range. This provided an additional basis for selecting the operating pressure of 25 psia.

FIGURE 14

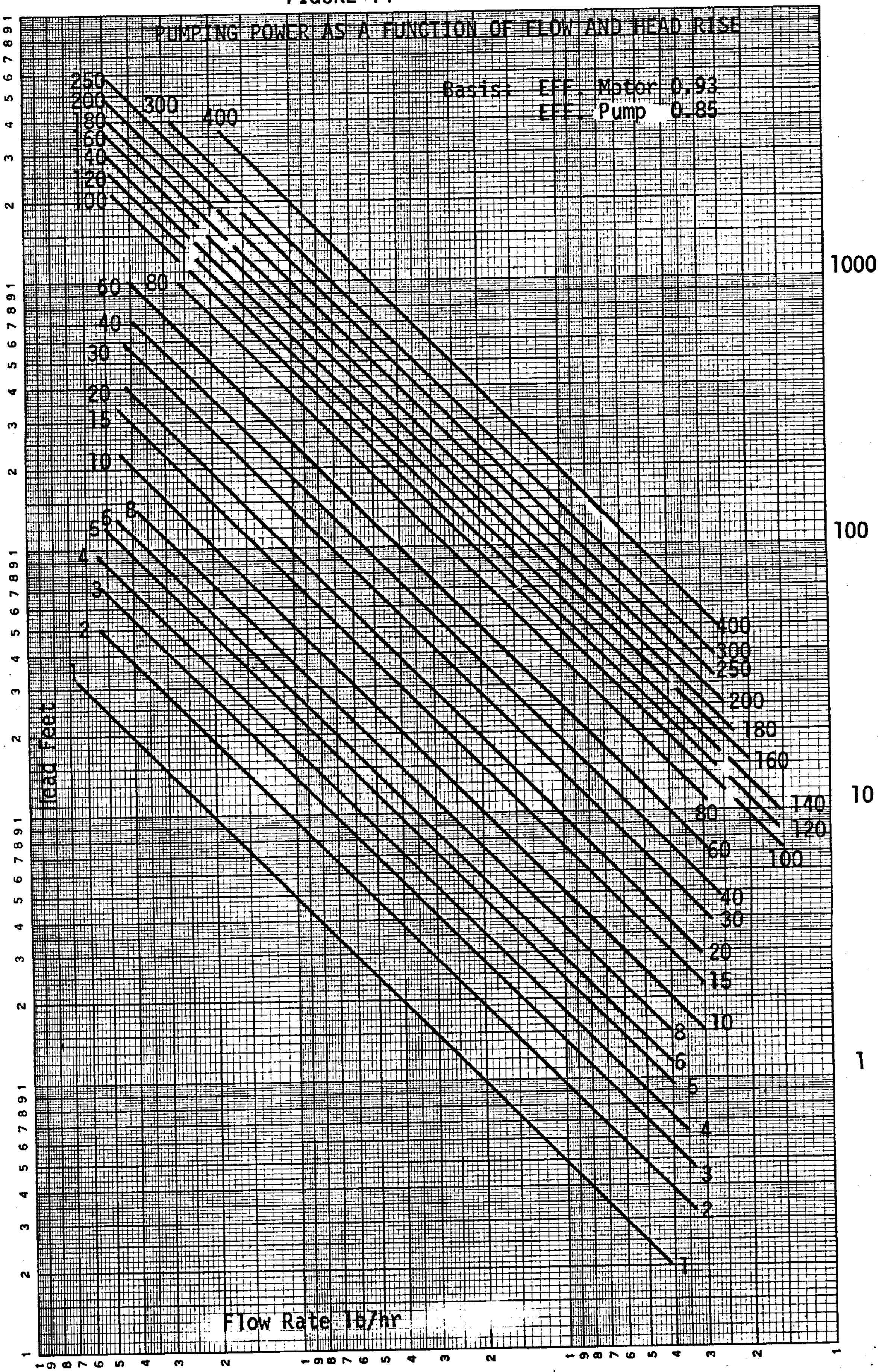
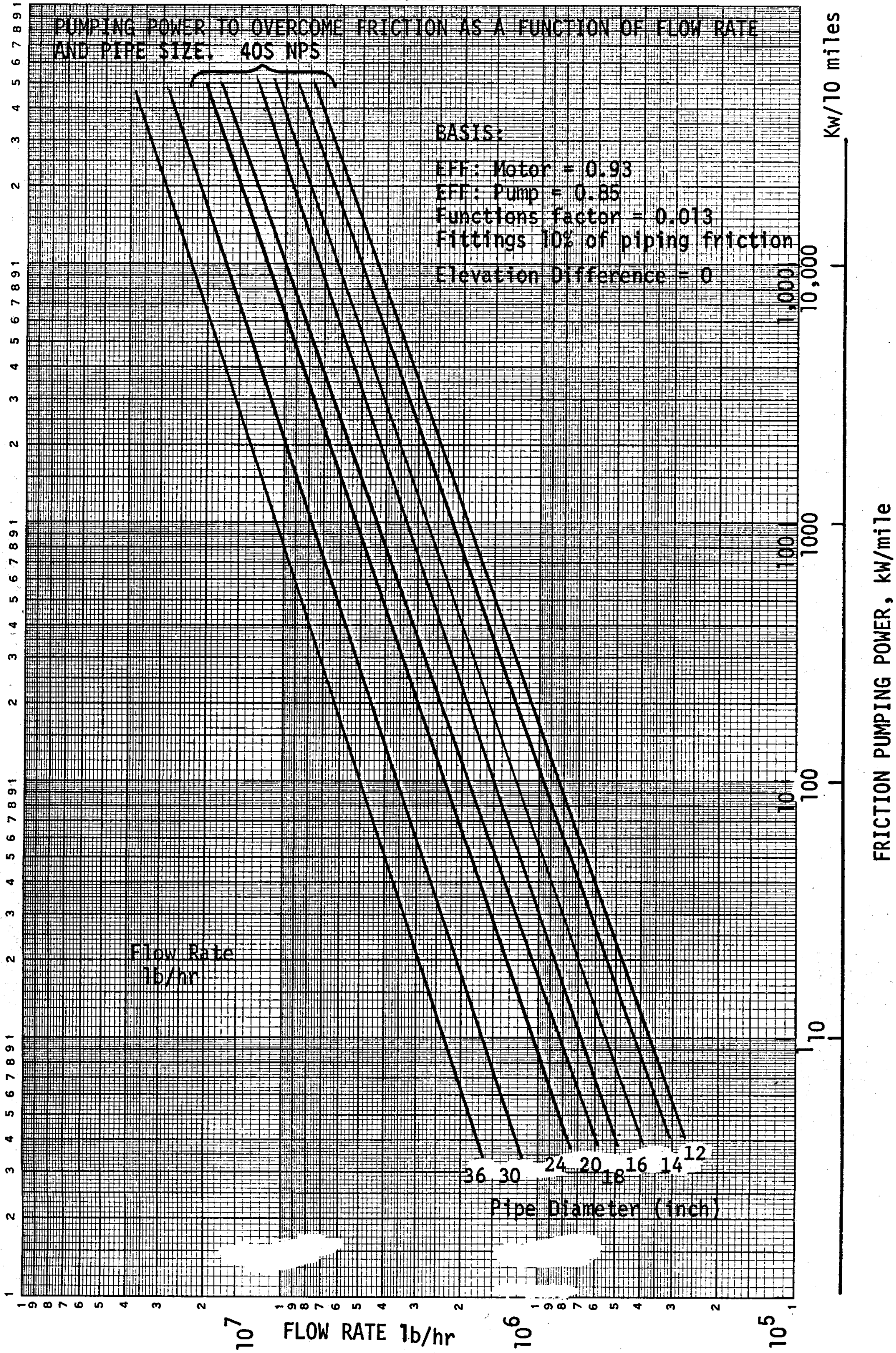

PUMPING POWER AS A FUNCTION OF FLOW AND HEAD RISE
Basis: EFF. Motor 0.93
EFF. Pump 0.85
Head Feet
Flow Rate lb/hr
Pumping Power, kW
1000
100
10
1
250
200
180
160
140
120
100
300
400
60
80
40
30
20
15
10
8
6
5
4
3
2
1

FIGURE 15


PUMPING POWER TO OVERCOME FRICTION AS A FUNCTION OF FLOW RATE AND PIPE SIZE. 40S NPS
BASIS:
EFF: Motor = 0.93
EFF: Pump = 0.85
Functions factor = 0.013
Fittings 10% of piping friction
Elevation Difference = 0
Flow Rate lb/hr
36
30
24
20
18
16
14
12
Pipe Diameter (inch)
10,000
1000
100
10
1,000
100
10
Kw/10 miles
FRICTION PUMPING POWER, kW/mile
10^7
FLOW RATE lb/hr
10^6
10^5

The turbine-generator set will be provided by the manufacturer with all auxiliaries and controls necessary for its operation. Calculations were made to determine if last stage moisture was not excessive. The results show that over our normal operating range the moisture will not exceed 12% when operating with a condenser pressure of 3 in. Hg abs.

2.3.5 Condenser

A barometric condenser has been tentatively selected. This type condenser minimizes its height. It is assumed that the cooling water can be supplied by a vacuum drag from the cooling tower. The diameter of the condenser tank is 14-1/2 ft and its length is 29 ft. The tail pipe was sized to provide 4 ft/sec maximum velocity of the cooling water and condensate. The wall thickness of the condenser body is 1 inch including corrosion allowance if the shell is not stiffened. The condenser tail pipe extends about 2 ft below the surface of the hot well allowing ample variation and still preserving the 1 ft minimum required between the end of the tail pipe and surface of the hot well water. The minimum allowable distance from the end of the tail pipe to the bottom of the hot well is 2 ft. Three feet is provided. Air ejectors or air pumps will be required in conjunction with the condenser to remove non-condensables. At this time no sizing calculations have been made for the air ejector.

The condenser size is determined primarily by the steam flow and condenser pressure. The variation in condenser size is given in Figure 17.

2.3.6 Cooling Tower System

The cooling water for the condenser will be provided by a wet cooling tower. One shallow well will provide makeup water to the cooling tower. Normally, the addition of the condensate to the cooling water (about 5% of cooling water flow) is sufficient to makeup for evaporation losses. However, addition of water is required to prevent mineral concentration.

Cooling tower suppliers have indicated that the power requirements of the cooling tower are about 850 kW for fan and water circulation.

FIGURE 16

FLASHER/SEPARATOR SIZE AS A FUNCTION OF FLASH PRESSURE AND BRINE FLOW

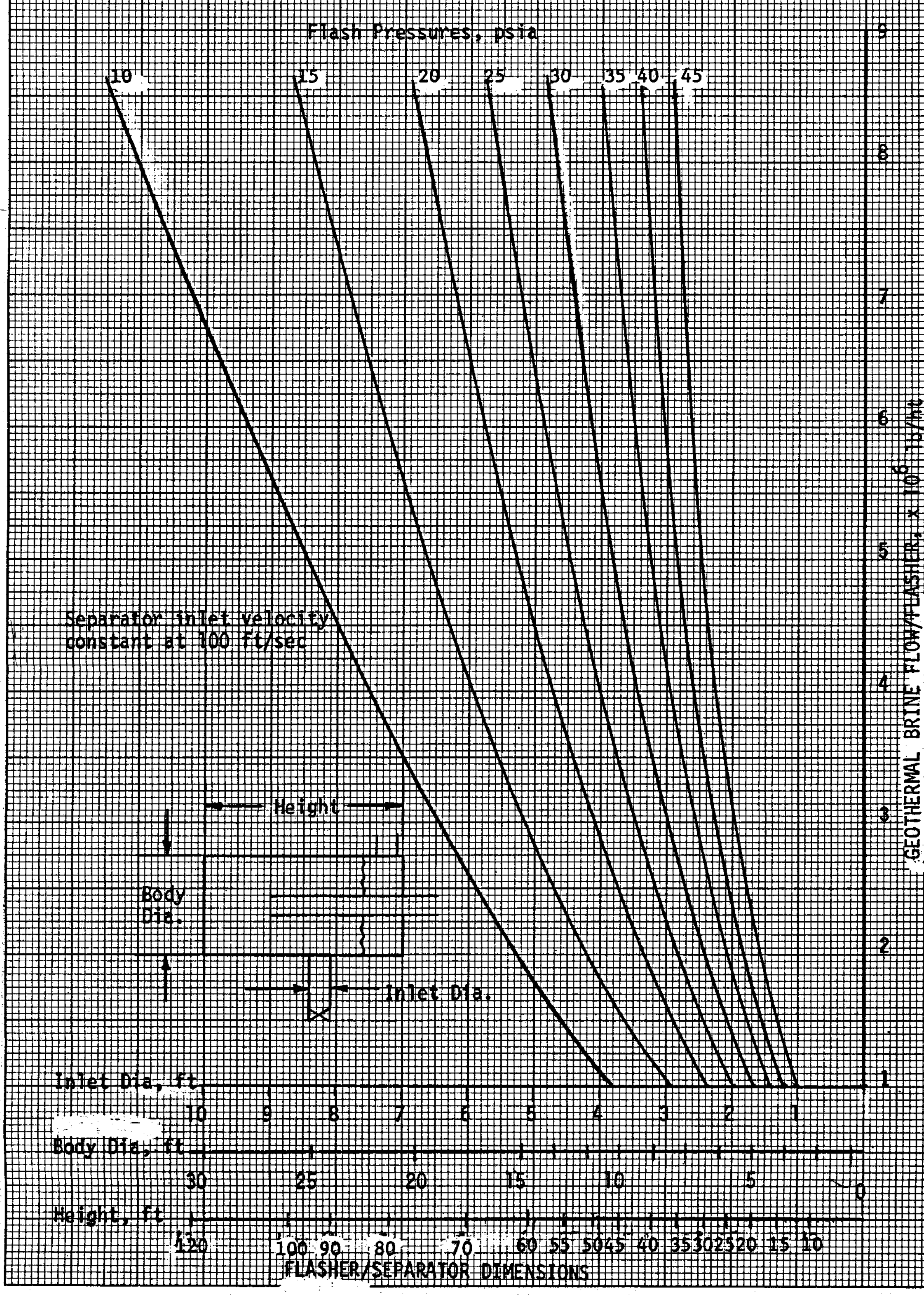
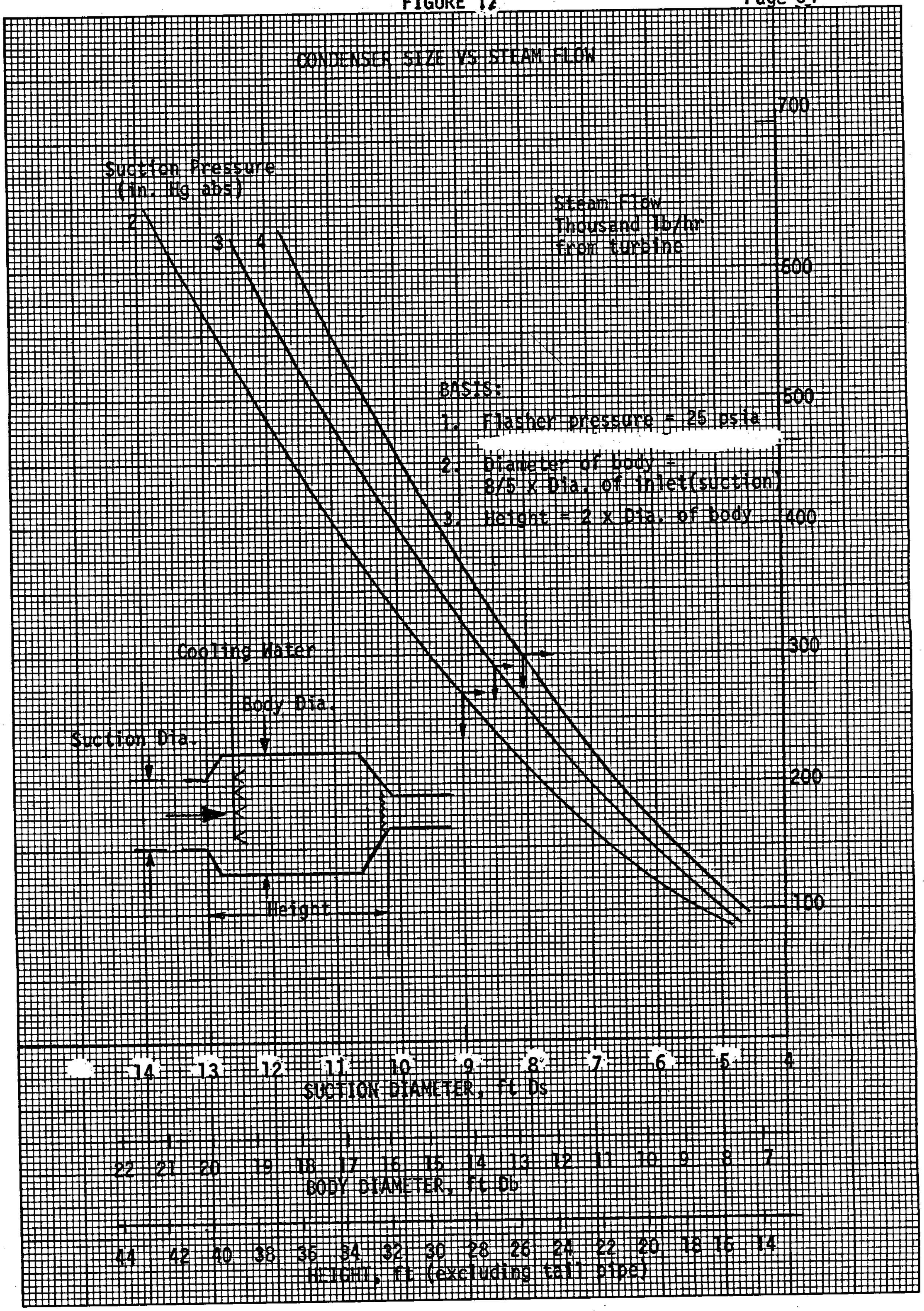


FIGURE 17


CONDENSER SIZE VS STEAM FLOW
Suction Pressure
(in. Hg abs)
2
3
4
Steam Flow
Thousand lb/hr
from turbine
700
600
500
400
300
200
100
BASIS:
1. Flasher pressure = 25 psia
2. Diameter of body =
8/5 x Dia. of inlet(suction)
3. Height = 2 x Dia. of body
Cooling Water
Body Dia.
Suction Dia.
Height
14 13 12 11 10 9 8 7 6 5 4
SUCTION DIAMETER, ft Ds
22 21 20 19 18 17 16 15 14 13 12 11 10 9 8 7
BODY DIAMETER, ft Db
44 42 40 38 36 34 32 30 28 26 24 22 20 18 16 14
HEIGHT, ft (excluding tail pipe)

Double Flash Steam System and Comparison with Single Flash

A double flash system was sized based on criteria and assumptions of the studies discussed in 1.3. The studies indicate the gross output of a plant could be improved 34% by the addition of a second flash system. A third flash system would increase the output by 10%.

A double flash system has been defined in which the first flash occurs at 30 psia and the second flash occurs at 13 psia. The second flash pressure was selected to be just above atmospheric pressure to prevent air leakage at flanges, valves, etc. These pressure conditions are not at the thermodynamic optimum conditions for a 300^{o}F inlet temperature with 3 in. Hg abs condenser pressure. However, the gross power produced will be approximately 93% of the ideal condition.

For convenience of calculation the steam expansion of both flashes were assumed to be independent for each turbine. Condenser sizes were estimated for both a single and a combined turbine discharge as well as separate condensers. The probable turbine configuration will be double flow unit with a later stage admission.

The double flash system assumes a brine flow rate of (5.1×10^6 lb/hr) that provides a 10% margin. A summary tabulation of operating conditions and equipment size is given in Table III and a listing of system losses and their basis is given in Table IV. A summary of the double flash results follows:

Nominal gross output, kW		15,100
Losses, kW		
Geothermal brine supply	405	
Geothermal brine return	649	
Cooling tower	1127	
Air ejection	(see discussion)	
Total losses (less air ejection)		2,181
Net power, kW		12,919

The wells should be located in relation to the generating station to minimize elevation loads. Table IV discusses losses associated with terrain differences. Supply wells located at different elevations relative to each other and the power house may present a major problem of matching outputs. A large common collection tank may be required from which the plant will be serviced. Tables III and IV indicate that 840 kW (5.5% of gross output) is required for possible elevation differences.

A significant unknown at this time is the work required for removal of non-condensables from the condensers. The problem depends on the amount of air in the brine. If non-condensables are as low as at Wairakei (0.3% by weight in steam) little concern exists. However, if it approaches 10% by weight of steam as in some plants (Ref 4) the load is huge. Ideal isentropic work to remove the air is the order of 1642 kW.* Steam air ejectors operating at the first flash pres-e appear impractical due to the low pressure.

* Water analysis of the deep geothermal wells drilled on the Raft River Valley the Spring of 1975 show a gas content that would result in only .05% in the steam.

A driving steam rate at 30 psia in the order of 2.5×10^6 lb/hr is required to remove 10% air (without considering the vapor removed with the air). For 100 psi driving steam it is estimated about 210,000 lb/hr of driving steam required (again without considering the vapor removed with the air). Some sources indicate that the ratio of vapor to dry air is in the order of 2.5. The above calculation aasumed the non-condensable to be air which has a lower molecular weight than most non-condensables found in geothermal suppliers.

It is unfair at this point to penalize the plant for what may be far too conservative estimates of non-condensables. The magnitude of the potential problem has been estimated, it was not included in the net output calculation.

For the single flash system some of the loads would be reduced. The problems on the brine supply and return are the same. The cooling tower load is reduced to 186 kW for pumping and 526 for fans (total, 712 kW) using the same assumptions as was used in the double flash calculation. The single flash summary becomes (for 5.1×10^6 lb/hr brine):

Nominal gross output, kW		11,000
Losses, kW		
Geothermal brine supply	405	
Geothermal brine return	649	
Cooling tower	712	
Total losses (less air ejection)		1,766
Net plant output, kW		9,234

The double flash system then provides an increase of almost 40% over the single flash (the ideal increase was in the order of 34%). The further increase is due to the brine pumping loads being independent of number of flashes.

TABLE III

Summary of Double Flash System

Initial brine temperature = 300°F Flow = 5.1×10^6 lb/hr (10,200 gpm
(10% added for MW gross)

	Flash #1	Flash #2
Flasher/Separator/Turbine Conditions		
Flasher operating pressure, psia	30	13
Flasher operating temperature, °F	250.3	205.9
Flasher inlet enthalpy (liquid, Btu/lb)	269.59	218.82
Ratio of steam to initial brine flow	0.0537	0.04362
Steam produced (100% quality), lb/hr	273,870	222,460
Nominal turbine steam flow, lb/hr	246,483	200,215
Nominal power at overall efficiency = 68%, MW (15.1)	9.52	5.56
Maximum power (using full steam production) MW (16.7)	10.58	6.18
Turbine inlet enthalpy, Btu/lb	1164.1	1148.1
Turbine Δh isentropic, Btu/lb	194	139.45
Turbine exit enthalpy at stage efficiency = 80%, Btu/lb	1009	1036
Turbine exit moisture, %	10.1	7.3
Condenser Conditions		
Condenser pressure, in. Hg Abs	3 in.	3 in.
Ratio of cooling water flow to steam flow	28.41	30.57
Condenser cooling water flow, gpm	15,563	13,603
Temperature of cooling water inlet, °F	75	75
Temperature of cooling water outlet, °F	108	108
Individual Separator Conditions		
Inlet brine flow, gpm	3,400	3,217
Exit brine flow, gpm	3,217	3,069
Flasher exit area, ft^2	3.56	6.26
Flasher exit inside diameter, in.	25.5	33.8
Separator, steam line exit diameter, in.	25.5	33.8
Brine discharge exit diameter, in.	25.5	33.8
Separator diameter, ft	6.38	8.47
Separator height, ft	24.5	32.47

TABLE III (cont'd)

	Flash #1	Flash #2
Condenser Size		
Inlet diameter, ft	8.3	7.5
Body diameter, ft	13.3	12
Height (excluding tail pipe), ft	26.6	24
Single condenser sized for discharge of both turbines or a single turbine with second stage admission	- - - - - - - -	- - - - - - - -
Inlet diameter (equivalent), ft	11.2	
Body diameter, ft	17.9	
Height, ft	36	

TABLE IV

Losses Associated with A Double Flash Plant

<u>Supply System</u>	
Friction losses [1]	
Average distance from well	2,100 ft
Pipe size	9.625 OD x 8.75 ID
Roughness (equivalent to asphalted cast iron)	0.0004 ft
Relative roughness	$\varepsilon/D = 0.00055$
Reynolds No.	$\sim 5 \times 10^6$
Friction factor	f = 0.017
Fitting allowance	10%
f L/D	48
Friction loss at 1×10^6 lb/hr	100 ft
Power, friction only	50 kW/well
Total friction power, 4 remote wells	200 kW[2]
Elevation head losses	
Elevation from header to flasher #1, assumed 22 ft	57 kW
Elevation from 4 wells to power house or common tank, 60 ft	123 kW
Allowance for positive pressure at flasher inlet (5 psi), 10 ft	25 kW
Total elevation losses	205 kW
Total brine supply losses	405 kW

Remarks: Elevations are at this time only estimates; however, since they represent ~50% of total requirement the importance of well and generating site location is indicated. Friction loss could be reduced by larger pipe, since flow is constant, friction power will vary directly with head, which varies inversely with d^5. (A 50% increase in pipe ID will reduce friction power to about 1/8 of value shown).

TABLE IV (cont'd)

Brine Return Lines	
Friction Losses	
Pipe size	30 in. nominal, Sch. 10, ID $\approx$ 29 in.
Roughness	ε = .0005 ft
Relative roughness	ε/D = 0.000165
Reynolds No.	$\approx 2 \times 10^6$
Friction factor	f $\approx$ 0.013
Fitting allowance	10%
f L/D + K	$\approx$ 32.7
Flow rate	5.1×10^6 ;b/hr (3)
Friction loss at 5.1×10^6 lb/hr	18.27 ft loss/mile
	7.75 psi/mile
Power loss, friction/mile	47 kW
For: 9 miles total friction	423 kW
6 miles total friction	282 kW
Elevation losses	
Potential elevation difference between generating station and reinjection wells (5100 - 4960)	140 ft(4)
Power to overcome elevation of 140 ft	367 kW
Total brine return power (6 miles)	649 kW
Cooling Tower Loads	
Cooling water flow, total	
Fan power (estimated at 285 kW/10,000 gpm)	833 kW
Total lift from condenser hot well, 40 ft max.(5) power to lift 29,166 gpm, 40 ft	294 kW
Total cooling tower power	1127
Air ejection loads	
Total lb/hr air (assuming 10%)	49,633 lb/hr
Ideal isentropic work to pump to atmospheric pressure	1642 kW
Actual work, assuming efficiency = 60%	2738 kW

TABLE IV (cont'd)

(1) Pressure drop for headers and branches within power station are not included.

(2) Pressure drop calculations are frequently considered accurate to ±20%. Thus, an additional 40 kW could be considered.

(3) Assumes flexibility to return all brine to reinjection.

(4) If power plant is located near the "narrows" area, elevation is about 4960 ft. The reinjection location is at 5100 ft.

(5) A minimum of 20-50 ft is required within cooling tower, an additional 10 ft was allowed for relative placement of tower and hot well.

4. REFERENCES

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