PRELIMINARY ASSESSMENT OF THE VELOCITY PUMP
REACTION TURBINE AS A GEOTHERMAL TOTAL-FLOW
EXPANDER

O. J. Demuth

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O. J. Demuth

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ABSTRACT

A preliminary evaluation was made of the Velocity Pump Reaction Turbine (VPRT) as a total flow expander in a geothermal-electric conversion cycle. Values of geofluid effectiveness of VPRT systems were estimated for conditions consisting of: a 360° geothermal resource, 60°F wet-bulb ambient temperature, zero and 0.003 mass concentrations of dissolved noncondensable gas in the geofluid, 100 and 120°F condensing temperatures, and engine efficiencies ranging from 0.4 to 1.0. Achievable engine efficiencies were estimated to range from 0.47 to 0.77, with plant geofluid effectiveness values ranging as high as 9.5 Watt hr/lbm geofluid for the 360°F resource temperature. This value is competitive with magnitudes of geofluid effectiveness projected for advanced binary plants, and is on the order of 40% higher than estimates for dual-flash steam and other total flow systems reviewed. Because of its potentially high performance and relative simplicity, the VPRT system appears to warrant further investigation toward its use in a well-head geothermal plant.

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I. SUMMARY

1. Background

Total-flow expanders have been recognized as having the potential for improving available-energy efficiency of geothermal-electric energy conversion systems relative to the more widely used flash-steam systems without introducing significant additional plant expense or operational complexity. Axial-flow impulse total-flow machines have been investigated and have been found to have limitations imposed on achievable isentropic efficiency largely because of excessive droplet impingement occurring on rotor blading.

Consideration of centripetal accelerations of the liquid droplets during their passage through rotor blading of two-phase turbo-expanders, for both impulse and reaction devices, has suggested that for equal enthalpy drops per stage these centripetal accelerations, as well as accelerations in the streamline direction, would be reduced significantly through the use of reaction rather than pure impulse machines. Droplet impingement on blading might then be reduced for given droplet sizes, and engine efficiency improved. The Velocity Pump Reaction Turbine (VPRT), invented at the Lawrence Livermore Laboratory (LLL), appears to have extremely advantageous characteristics in this regard. The VPRT (Figure 1) is comprised basically of two concentric rotors for which saturated or compressed-liquid geofluid enters the center of the inner rotor. The inner rotor is driven externally (probably with power extracted from the outer rotor), in order to pump and redirect the geofluid such that it is accelerated to about twice the speed of the inner rotor in a gap between the rotors while maintaining the pressure equal to the inlet pressure. The geofluid in the gap then flows radially outward through passages in the outer rotor to nozzles near the outer periphery. The geofluid is maintained in a liquid state from the VPRT inlet to the entrance of the outer-rotor nozzle. The final two-phase expansion process is carried out in the outer-rotor nozzles, where the geofluid

is accelerated, linearly, in an axial direction relative to the nozzles. Torque and power are produced by the outer rotor due to the changes in the moment of momentum of the geofluid as it passes through the outer rotor.

2. Approach

The approach taken in the present preliminary assessment of the VPRT was first to review its performance characteristics. The relationships between engine isentropic efficiency and engine configurational and operating conditions, such as outer-rotor speed, rotor speed ratio, rotor radius ratio, and velocity coefficient for the inner and outer rotors, were explored. The review was concluded by estimating the range of expected isentropic efficiency achievable with the VPRT based on existing velocity-coefficient data for two-phase nozzles, and on performance data for a similar reaction turbines without the inner rotor. Isentropic efficiencies for the VPRT as high as 0.77 appear achievable.

Second, a VPRT energy conversion cycle (Figure 5) was envisioned which could accommodate geofluids having moderate concentrations of noncondensable gasses. The conversion system was comprised of (1) a preflash vessel to remove a major fraction of the noncondensibles from the geofluid upstream of the VPRT, (2) the VPRT, itself, (3) a shell-and-tube condenser, (4) a vapor turbine/vapor compressor unit to derive power from the preflash vapor and pump noncondensibles (and water vapor) from the condenser, (5) a wet cooling tower, and (6) liquid pumps to reinject condensate and pump cooling water as required. The system is configured to be compatible with a Stretford H_2S abatement system.

Cycle analyses were conducted for the VPRT energy-conversion system under the following conditions:

- (1) 360°F geofluid supplied at 273 psia using a downhole pump,
- (2) zero and 0.003 mass concentrations of noncondensable gases, kept in solution at the 273 psia plant inlet condition,
- (3) ambient wet bulb temperature of 60°F,
- (4) condensing temperatures of 100 and 120°F,
- (5) engine isentropic efficiencies of 0.4, 0.7, and 1.0.

For each condition the preflash temperature was found which would maximize net plant power when the vapor turbine was driven by the preflash vapor and provided the required power to operate the condenser vent compressor.

The results of the cycle analyses indicated that the variable having major influence on the net plant geofluid effectiveness is the engine isentropic efficiency; the effects of condensing temperature and noncondensable gas content were significantly smaller. At the 360°F resource temperature assumed, net plant geofluid effectiveness reached values as high as about 9.5 Watt hr/lbm geofluid for engine efficiencies in the 0.75 range.

In the third phase of the study estimates of geofluid effectiveness were assembled for competing conversion systems. The primary comparison was made with an advanced binary plant. Systems reviewed which derived power directly from expanding part or all of the geofluid were the dual-flash steam system, the helical-screw expander system, the Bi-Phase Rotary Separator Turbine system, the Jet Propulsion Laboratory (JPL) and LLL axial-flow impulse turbines, and the LLL Radial Outflow Reaction Turbine. The comparisons indicated that the higher end of the range of expected performance for the VPRT was competitive with the performance projected for the advanced binary

plant. The plant geofluid effectiveness for all of the systems expanding geofluid directly fell in the lower half of the potential performance range projected for the VPRT.

3. Conclusions

The overall conclusion reached is that the VPRT appears to be a good candidate for a well-head total-flow geothermal-electric plant for moderate resource temperatures in the 360°F range. The maximum expected geofluid effectiveness is competitive with that expected for an advanced binary plant, and the system should be relatively inexpensive to construct and operate relative to the binary plant which uses expensive or flammable working fluids. The VPRT configuration attacks the major problem encountered with axial-flow impulse total-flow turbines, that of droplet impingement, by maintaining the geofluid in a single phase liquid state during all flow direction changes within the VPRT.

The present assessment indicates that the VPRT concept is worthy of additional investigation. The next logical step would be the experimental determination of geofluid effectiveness using subscale hardware with which different outer-rotor nozzle configurations can be tested and evaluated.

II. INTRODUCTION

1. Energy Conversion Concepts

The basic energy-conversion concepts existing today for generation of electrical energy from liquid-dominated hydrothermal resources, consist of flashed steam, binary and total-flow systems. At the present time only the flashed steam system is in significant commercial use. Because of the predominance of the lower temperature geothermal resources (below, say 400°F) with correspondingly low magnitudes of available energy per unit mass of geofluid, geothermal-electric energy is expensive to produce, and motivation exists to improve the available-energy efficiency of the conversion process. Toward this objective binary plants are attracting current attention; the 50 MW Heber binary plant, for example, is being constructed in California's Imperial Valley and the 10 MW Magmamax binary plant has been producing power for several months in that same area. An alternate, and probably a simpler approach, potentially leading to improvements in available-energy efficiency and therefore net plant geofluid effectiveness (power output per unit mass flow of geofluid), is the "total-flow" concept.

2. Total Flow Systems

The total-flow concept consists basically of expanding the entire geofluid flow from the well through an energy conversion device to produce mechanical work, and in turn, electrical energy using an electric generator. In actual practice with geofluids containing noncondensable gasses for which expansion to subatmospheric pressure is needed to achieve adequate geofluid effectiveness, a mild preflash may be beneficial. This preflash will eliminate most of the noncondensable gas from the geofluid upstream of the total-flow expander in order reduce the amount of noncondensibles which must be vented and pumped from the condenser. Under these conditions a small amount of steam with the noncondensable gases would bypass the total

flow expander, and the geofluid entering the expander would be close to the saturated liquid state.

The total flow system has a potential thermodynamic advantage, relative to the flashed-steam system, in that thermodynamic irreversibilities occurring in the isenthalpic flash processes are not present, and no geofluid (containing available energy) must be discarded as is done following the last flash of a flashed steam system. Relative to the binary system, thermodynamic irreversibilities, generated because of the differences in temperature between the geofluid and the binary working fluid during heat addition, are not present. Although a potential performance advantage exists for total-flow systems, relative to binary and flashed-steam systems, the overall merit of the total-flow system for a given application will be strongly dependent on the performance parameter engine isentropic efficiency.

3. Performance Limitations of Axial Flow Impulse Turbines

Axial-flow impulse turbines have been selected for study by the Jet Propulsion Laboratory (JPL) (Reference 1) and the Lawrence Livermore Laboratory (LLL) (Reference 2) as total-flow expanders which hold promise for producing good isentropic efficiencies, and which can carry out geofluid expansions through the very large specific volume ratios demanded of total-flow systems. During their experiments performance limitations (in terms of engine isentropic efficiency) encountered by both JPL and LLL were judged to result largely from the liquid phase, in the form of droplets, impinging on the blading. This droplet impingement formed a liquid film on the blading and created major departures of actual fluid velocities from the design fluid velocities; conversion of fluid kinetic energy to useful mechanical work was impacted significantly.

4. Reaction Turbomachines for Total Flow Expanders

A semiquantitative comparison of nozzle, blade, and fluid velocities between an impulse turbine and a 50% reaction turbine, maintaining equal enthalpy drops per stage, indicated that absolute fluid accelerations in the blading (both parallel and perpendicular to the flow direction) should be on the order of half as large for the reaction turbine as for the impulse turbine. In addition, the nozzle exit velocity for the reaction turbine would be about 1/2 times that for the impulse turbine. These results suggested that for equal droplet sizes a reaction turbine might suffer smaller droplet velocity lags in the nozzles and substantially smaller droplet impingement effects in the blading. During our review of various reaction-turbine types, a conceptual design and analysis study by LLL on the "Velocity Pump Reaction Turbine" (VPRT) was encountered (Reference 3). This configuration appears to circumvent a large portion of the droplet-impingement problems found during experiments conducted with the axial-flow impulse turbines in the total-flow applications.

5. Report Content

This report summarizes a review made of the VPRT performance characteristics, presents results of cycle calculations made for systems incorporating the VPRT, and compares the resulting geofluid effectiveness magnitudes with those of several competing total-flow systems. The work was supported by the U.S. Department of Energy, Deputy Assistant Secretary for Renewable Energy, Geothermal and Hydropower Division under contract No. DE-AC07-76ID01570.

III. VPRT PERFORMANCE CHARACTERISTICS

1. Description of VPRT

The Velocity Pump Reaction Turbine (VPRT) is shown schematically in Figure 1; its description and a derivation of the relationships defining its performance are presented in Reference 3. Conceptually, the pump consists of two concentric rotors with the geofluid entering the device at the center of the inner rotor. The fluid is pumped to a higher stagnation pressure as it passes radially outward through the inner rotor which is driven by an external power source (or by a drive system receiving power from the outer rotor). The fluid then is accelerated tangentially, relative to the inner rotor, as it passes through the inner-rotor nozzle at radius, r , and is discharged into the circumferential gap between rotors. The pressure in the gap depends on the peripheral speed of the inner rotor, the VPRT inlet pressure, and the fluid velocity at the nozzle exit of the inner rotor (relative to the rotor). For the present application the fluid in the gap (inner-rotor nozzle exit) is assumed to be kept in the saturated liquid (or slightly subcooled) state to maintain a single phase fluid. To minimize fluid friction effects in the gap, the outer rotor speed at its inner surface should be close to the absolute velocity of the fluid in the gap. If saturated liquid enters the VPRT, if frictional effects are neglected, and if the gap between rotors is small, those conditions are approximately satisfied when the inner-rotor-nozzle fluid velocity (relative to the inner rotor) equals the peripheral speed of the nozzle itself, and the angular velocity of the outer rotor is twice that of the inner rotor.

The fluid in the gap is then picked up by the outer rotor, flows radially outward (relative to the outer rotor), and becomes pumped to a higher stagnation pressure at the outer rotor nozzle at radius, R . The fluid then turns through 90° (relative to the rotor) in a

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Figure 1. Velocity Pump Reaction Turbine (VPRT) concept.

subcooled liquid state, and is expanded through the exhaust nozzle in the outer rotor to a two-phase state. A reaction torque is imparted to the outer rotor, and power is generated.

Two concepts of significance should be introduced at this time. First, the incorporation of the inner rotor results in higher engine isentropic efficiencies than could be obtained with a single-rotor reaction device (such as Hero's turbine). Generally these peak efficiencies can be obtained at lower maximum rotor speeds and lower exhaust nozzle expansion ratios for the VPRT than for the single-rotor radial outflow reaction turbine.

Second, the VPRT inner rotor serves the function of the nozzle assembly in a more conventional turbine; that is, the inner rotor accelerates the fluid before it enters the rotating component that generates the shaft power output. The VPRT rotor, however, provides an additional degree of control; it permits accelerating the fluid prior to its entry into the outer rotor, but at the same time creating a pressure which will maintain the accelerated fluid in the single-phase liquid state while the various required changes in flow direction occur upstream of the exhaust nozzle. The final two-phase expansion in the exhaust nozzle, then, can be made without major centripetal accelerations of the droplets, and droplet-impingement effects on performance should be substantially reduced relative to conventional two-phase turbomachines.

2. Equations Describing Engine Efficiency

In order to amplify some of the assumptions and viewpoints presented in Reference 3 during the derivation of the performance relationships, a development of the equations, with simplifications made as required to maintain tractability, is included in this section. Some discussion of the simplifications is included along with examples of errors introduced.

First, the assumption is made that the inlet geofluid is saturated or subcooled liquid having a density, ρ lbm/ft³, and an inlet enthalpy h_{in} ft lbf/lbm. Further, it is assumed that the geofluid is kept in the liquid state until it reaches the inlet to the outer-rotor nozzle, and accordingly, the density is assumed constant to that point.

Inner Rotor. Referring to Figure 1, the geofluid enters the inner rotor at its center, and its pressure is increased as it is pumped because of the inner rotor turning at a speed ω rad/sec. The increase in stagnation pressure from the VPRT inlet to the inner-rotor nozzle inlet at radius r , ft, is:

$$\Delta p = \int_0^r \frac{\rho \omega^2 r dr}{g} = \frac{\rho \omega^2 r^2}{2g} \quad \frac{\text{lbf}}{\text{ft}^2} \quad (1)$$

The ideal work done on the fluid during this pumping is:

$$W = \int_0^{\Delta p} \frac{dp}{\rho} = \frac{\omega^2 r^2}{2g} \quad \frac{\text{ft lbf}}{\text{lbm}} \quad (2)$$

The fluid then expands through the inner-rotor nozzle into the gap to a pressure equal to the VPRT inlet pressure and with an ideal velocity, v_{1i} , relative to the inner rotor.

$$v_{1i} = \sqrt{\frac{2g\Delta p}{\rho}} = \omega r \quad \text{ft/sec} \quad (3)$$

If a velocity coefficient C_{v1} is assigned to represent the deviation of the actual relative velocity in the gap from the ideal relative velocity, the actual absolute gap velocity at the same pressure can be written:

$$V_1 = \omega r (1 + C_{v1}) \quad (4)$$

From the moment of momentum relationship the torque, T , supplied externally to the inner rotor must be:

$$T_1 = \frac{\dot{m}}{g} \omega r^2 (1 + C_{v1}) \quad \text{ft lbf} , \quad (5)$$

and the power, P_1 , required to drive it is:

$$P_1 = \frac{\dot{m}}{g} \omega^2 r^2 (1 + C_{v1}) \quad \frac{\text{ft lbf}}{\text{sec}} \quad (6)$$

where $\dot{m}$ is the mass/flow rate, lbm/sec.

Gap Pressure Required to Maintain Liquid State. At this point a little digression may be in order. A value of $C_{v1} < 1.0$ introduces two related effects. First, frictional effects throughout the inner rotor, nozzle, and gap region are implied which generate irreversibilities in the machine (to be discussed later). Second, the irreversibilities correspond to an increased geofluid temperature at the gap pressure (assumed to equal the VPRT inlet pressure). Therefore, if the inlet state is saturated liquid, some vapor would form in the gap, which is a condition we may want to prevent by increasing the inner-rotor speed, ω . To estimate the magnitude of such a required increase in ω , we can first approximate the temperature increase, which is related to the difference between the kinetic energy in the gap, KE_1 , (inertial reference) and the power per unit mass flow supplied to the rotor.

$$P_1 - KE_1 = \frac{\dot{m}}{g} \omega^2 r^2 (1 + C_{v1}) - \frac{\dot{m}}{2g} \omega^2 r^2 (1 + C_{v1})^2 ,$$

or

$$P_1 - KE_1 = \frac{\omega^2 r^2}{2g} (1 - C_{v1}^2) \quad \text{ft lbf/lbm} .$$

Using the Clapyron Equation and the specific heat of liquid water, C , the required pressure increment to maintain saturated liquid at the increased temperature can be approximated as:

$$\Delta p' = \left(\frac{\partial p}{\partial t} \right)_{\text{sat}} \Delta T = \frac{h_{fg}}{v_{fg} T} \frac{\omega^2 r^2}{2gC} (1 - C_{v1}^2) \quad \frac{\text{lbf}}{\text{ft}^2}$$

where h_{fg} and v_{fg} are the latent enthalpy and specific volume of vaporization, and T and p are the absolute saturation temperature and pressure of the geofluid in the rotor.

$$\frac{\Delta p'}{\Delta p} = \frac{h_{fg} (1 - C_{v1}^2)}{T v_{fg} C \rho} .$$

For saturated liquid at 360°F, for example, and for $C_{v1} = 0.95$,

$$\frac{\Delta p'}{\Delta p} = 0.00063 ,$$

which could be produced by a fractional change in rotor speed of 0.0003. This effect, therefore, appears negligible and will not be treated further, analytically; its negligibility would be confirmed in any experiments conducted.

Outer Rotor. The geofluid leaving the gap, and passing radially through the outer rotor, has its pressure increased because of the pumping effect of the outer rotor speed, ω . It is assumed here, for purposes of simplification and to be mildly conservative, that none of the velocity head of the gap fluid is recovered as it enters the outer rotor. That is, the inlet stagnation pressure in the outer rotor (relative to the rotor) is assumed to equal the gap static pressure.

Neglecting frictional effects in the outer rotor upstream of the nozzle, the change in stagnation pressure between the outer rotor inlet and its nozzle inlet is:

$$\Delta p_2 = \rho \frac{\omega^2 (R^2 - r^2)}{2g} \frac{\text{lb f}}{\text{ft}^2} \quad (7)$$

The work done on the fluid to produce this effect is:

$$\frac{\Delta p_2}{\rho} = \frac{\omega^2 (R^2 - r^2)}{2g} \frac{\text{ft lb f}}{\text{lb m}} \quad (8)$$

The next step in the development of VPRT performance predictions is to determine the approximate optimum rotor speed ratio and the corresponding isentropic efficiency of the VPRT. First, the assumption is made that the entire thermodynamic process experienced by the geofluid in moving from the VPRT inlet to the inlet of the outer-rotor nozzle is reversible. Under that condition the isentropic discharge enthalpy for the machine, $h_{\text{ex is}}$, (at the discharge pressure) which is used to define the "spouting velocity," $C_o = \sqrt{2g(h_{\text{in}} - h_{\text{ex is}})}$, equals the isentropic discharge enthalpy for the outer-rotor nozzle. For a reversible process through the inner rotor and gap C_{v1} must equal unity; the power (per lbm/sec) supplied to the inner rotor (Equation 6) just equals the absolute kinetic energy of the gap fluid; and the static enthalpy in the gap equals the VPRT inlet enthalpy, h_{in} . The square of the isentropic discharge velocity from the outer nozzle (relative to the outer nozzle) can be written:

$$v_{2i}^2 = 2g \left[h_{\text{in}} + \frac{\omega^2 (R^2 - r^2)}{2g} - h_{\text{ex is}} \right] \quad (9)$$

Incorporating the definition of the "spouting velocity," (Equation 9) can be rewritten as:

$$v_{2i}^2 = C_o^2 + \Omega^2(R^2 - r^2) \quad (10)$$

The absolute velocity of the exhaust geofluid is:

$$V_2 = C_{v2} \sqrt{C_o^2 + \Omega^2(R^2 - r^2)} - \Omega R \quad (11)$$

where C_{v2} is the velocity coefficient of the outer nozzle. From the moment-of-momentum relationship the torque produced by the outer rotor is:

$$T_2 = \frac{\dot{m}}{g} \left[R C_{v2} \sqrt{C_o^2 + \Omega^2(R^2 - r^2)} - \Omega R^2 + 2r^2 \omega \right] \quad (12)$$

The net VPRT power is:

$$P = \frac{\dot{m}}{g} \left[R \Omega C_{v2} \sqrt{C_o^2 + \Omega^2(R^2 - r^2)} - \Omega R^2 + 2r^2 \omega - 2\omega^2 r^2 \right] \frac{\text{ft lbf}}{\text{sec}} \quad (13)$$

To find the value of $\frac{\omega}{\Omega}$ at maximum net power for the assumption of reversible internal fluid thermodynamics (the flow through the outer nozzle is treated as irreversible since $C_{v2} < 1.0$), the relationship $\frac{\partial P}{\partial \omega} = 0$ can be introduced.

$$\text{Then } \frac{2\dot{m}}{g} (\Omega - 2\omega) = 0 ,$$

or

$$\Omega = 2\omega \quad (14)$$

The expression for net power with $\omega = \frac{1}{2} \Omega$ is obtained from (Equation 13), and becomes:

$$P = \frac{\dot{m}}{g} \left[R \Omega C_{v2} \sqrt{C_o^2 + \Omega^2 (R^2 - r^2)} - \Omega^2 R^2 + \frac{1}{2} \Omega^2 r^2 \right]. \quad (15)$$

Writing $\beta = \frac{\Omega R}{C_o}$, as is done in Reference 3, (Equation 15) can be rewritten:

$$P = \frac{\dot{m} C_o^2}{2g} \left[2 \beta C_{v2} \sqrt{1 + \beta^2 \left(1 - \frac{r^2}{R^2}\right)} - \beta^2 \left(2 - \frac{r^2}{R^2}\right) \right]. \quad (16)$$

Using the standard definition of engine isentropic efficiency,

$$e = \frac{P}{\dot{m} (h_{is} - h_{ex is})} = \frac{P}{\frac{\dot{m} C_o^2}{2g}}, \text{ and}$$

$$e_{VPRT} = 2 \beta C_{v2} \sqrt{1 + \beta^2 \left(1 - \frac{r^2}{R^2}\right)} - \beta^2 \left(2 - \frac{r^2}{R^2}\right). \quad (17)$$

Effects of Entropy Production Within VPRT. As indicated, (Equations 14) and (17) were derived assuming reversible, adiabatic (isentropic) conditions within the VPRT from the input to the entrance of the outer nozzle. The question to be answered next is, "What are the consequences of the presence of irreversibilities in the rotor if the rotor speed ratio, ω/Ω , is held at 1/2 as shown in (Equation 14)?" For $C_{v1} < 1.0$ (Equation 6) describes the power to drive the inner rotor. If $\omega = \frac{1}{2} \Omega$ (Equation 6) becomes:

$$P_1 = \frac{\dot{m}}{4g} \Omega^2 r^2 (1 + C_{v1}). \quad (18)$$

The stagnation pressure at the entrance to the outer nozzle is not influenced by the magnitude of C_{v1} assuming zero recovery of the relative-velocity in the gap, but the stagnation enthalpy at that location must be increased because of two effects. First, the difference between the power required to drive the inner rotor and the absolute kinetic energy in the gap must be added.

$$\text{For } \Omega = 2\omega, P_1 - KE_1 = \frac{\Omega^2 r^2}{8g} (1 - C_{v1}^2) \quad \text{ft lbf/lbm} .$$

Second, the kinetic energy in the gap, KE_2 , (relative to the outer nozzle) must be added:

$$KE_2 = \frac{\left[\Omega r - (1 + C_{v1}) \frac{\Omega}{2} r \right]^2}{2g} \quad \frac{\text{ft lbf}}{\text{lbm}} ,$$

or

$$KE_2 = \frac{\Omega^2 r^2}{8g} (1 - C_{v1})^2 .$$

The total increase in enthalpy from the VPRT inlet to the outer-nozzle inlet, beyond that shown in (Equation 8) is:

$$(P_1 - KE_1) + KE_2 = \frac{\Omega^2 r^2}{4g} (1 - C_{v1}) \quad \text{ft lbf/lbm} .$$

The isentropic enthalpy change through the outer nozzle will be affected only slightly by this increase in stagnation enthalpy above that of the reversible internal flow case because in both cases (before and after accounting for the increase in outer-nozzle stagnation enthalpy), the expansions occur between the same pressures. Since

$$\left(\frac{\partial H}{\partial S} \right)_p = T ,$$

the difference between the values of the enthalpy changes for the two outer nozzle expansions is approximately $(T_1 - T_2) \Delta S$, where T_1 and T_2 are the inlet and exhaust temperatures of the outer-nozzle flow for the case having reversible VPRT flow, and ΔS is the difference in entropy between the two isentropic expansion processes.

$$\Delta S \approx \frac{1}{T_1} \frac{\alpha^2 r^2}{4g} (1 - C_{v1}) ,$$

and the difference in isentropic enthalpy change between the expansion processes for the irreversible VPRT flow case and the reversible-flow case ($C_v = 1.0$) is:

$$\left(1 - \frac{T_2}{T_1}\right) \frac{\alpha^2 r^2}{4g} (1 - C_{v1}) .$$

The isentropic enthalpy change for the VPRT for the imposed inlet and discharge pressures is:

$$h_{in} - h_{ex is} = \frac{C_0^2}{2g} .$$

The impact on engine efficiency associated with the effect of increasing the inlet enthalpy of the outer nozzle is then:

$$\Delta e = \frac{\alpha R^2}{2C_0^2} \frac{r^2}{R^2} \left(1 - \frac{T_2}{T_1}\right) (1 - C_{v1}) . \quad (19)$$

As an example, for $\beta = \frac{\alpha R}{C_0} = 1.0$, $C_{v1} = 0.95$, $T = 360^\circ\text{F}$,

$T_2 = 100^\circ\text{F}$, and $\frac{r}{R} = 0.85$, $\Delta e = 0.5 \left(1 - \frac{560}{820}\right) 0.05 \times 0.85^2 = 0.0057$;

this effect on engine efficiency is completely negligible. (Note that in this treatment wall friction within the outer rotor up to the nozzle entrance has been neglected.)

VPRT Performance Equations when $\Omega = 2\omega$. The torque produced by the outer rotor for $\Omega = 2\omega$ can be written, modifying (Equation 12) to account for $C_{v1} < 1.0$:

$$T_2 = \frac{\dot{m}}{g} \left[RC_{v2} \sqrt{C_0^2 + \Omega^2 (R^2 - r^2)} - \Omega R^2 + \left(\frac{1 + C_{v1}}{2} \right) r^2 \Omega \right] . \quad (20)$$

The net power becomes:

$$P = \frac{\dot{m}}{g} \left[\Omega RC_{v2} \sqrt{C_0^2 + \Omega^2 (R^2 - r^2)} - \Omega^2 R^2 + \frac{\Omega^2 R^2}{2} \left(\frac{1 + C_{v1}}{2} \right) \frac{r^2}{R^2} \right] .$$

Introducing $\beta = \frac{\Omega R}{C_0}$

$$P = \frac{\dot{m} C_0^2}{2g} \left[2\beta C_{v2} \sqrt{1 + \beta^2 \left(1 - \frac{r^2}{R^2} \right)} - \beta^2 \left(2 - \frac{1 + C_{v1}}{2} \frac{r^2}{R^2} \right) \right] , \quad (21)$$

and the engine isentropic efficiency is:

$$e_{VPRT} = 2\beta C_{v2} \sqrt{1 + \beta^2 \left(1 - \frac{r^2}{R^2} \right)} - \beta^2 \left[2 - \left(\frac{1 + C_{v1}}{2} \right) \frac{r^2}{R^2} \right] . \quad (22)$$

Strictly speaking, the optimum speed ratio should have been found by differentiating a complete expression which includes the effects of internal VPRT irreversibility on the ideal exit velocity of the outer

nozzle; however, from the numerical evaluation of (Equation 19) the final result presented in (Equation 22) is adequate for performance predictions of the VPRT.

VPRT Performance Results. To find the optimum β (β_{opt}), e_{VPRT} in (Equation 22) can be differentiated with respect to β , and the derivative set equal to zero. Performing these operations results in:

$$C_{v2} = \frac{\beta_{opt} \left[2 - \left(\frac{1 + C_{v1}}{2} \right) \frac{r^2}{R^2} \right]}{\beta_{opt}^2 \left[1 + \beta_{opt}^2 \left(1 - \frac{r^2}{R^2} \right) \right]^{-1/2} \left(1 - \frac{r^2}{R^2} \right) + \left[1 + \beta_{opt}^2 \left(1 - \frac{r^2}{R^2} \right) \right]^{1/2}} \cdot \quad (23)$$

From (Equation 23), β_{opt} can be found for given values of C_{v1} , C_{v2} , and $\frac{r}{R}$.

Plots of (Equation 22) have been shown, in several formats, for $C_{v1} = 1.0$ in Reference 3. To illustrate the VPRT performance behavior, two of these figures have been included as Figures 2 and 3 in this report. Figure 2 illustrates the variation of VPRT isentropic efficiency with rotational speed at several values of the outer-nozzle velocity coefficient for the ratio of rotor radii of 0.8 and for the inner-rotor velocity coefficient, C_{v1} , of unity. Figure 3 shows the results of combining (Equations 22) and (23) for $C_{v1} = 1.0$.

3. Estimated Bounds on VPRT Efficiency

Figure 4 was prepared to illustrate the predicted VPRT efficiency as a function of outer-nozzle speed (ωR) for a rotor radius ratio, $\frac{r}{R}$, of 0.85, and for different values of nozzle velocity coefficients. Also shown for comparison is a similar curve for a device without the inner rotor $\frac{r}{R} = 0$. The curves were plotted for a 360°F resource with a condensing temperature of 100°F. Two points are noteworthy here. First, the presence of the pump rotor (inner rotor) increases the VPRT efficiency by about 30% at a nozzle speed of 1200 ft/sec.

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Figure 2. Effect of nozzle performance on VPRT output.

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Figure 3. VPRT efficiency.

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Figure 4. Effects of velocity coefficients on VPRT efficiency, resource temperature = 360°F.

Second, the pump velocity coefficient, C_{v1} , of 0.95 results in only a small (3%) penalty in efficiency relative to $C_{v1} = 1.0$, whereas engine efficiency is seen to be quite sensitive to the outer-nozzle velocity coefficient, C_{v2} .

Based on the preceding calculations and on measurements of velocity coefficients in two-phase-expansion nozzles by LLL (Reference 2) as high as 0.94 (with some inlet quality), an engine isentropic efficiency of about 0.77 appears to represent a reasonable upper limit for use in VPRT cycle calculations and net-plant-geofluid-effectiveness predictions.

To estimate a lower bound on engine isentropic efficiency for the VPRT, use was made of actual test data obtained with a radial outflow reaction turbine by LLL as presented in Reference 4. The configuration tested corresponded to a VPRT without the inner rotor ($\frac{r}{R} = 0$). In these tests brake power, flow, inlet temperature and pressure, discharge pressure, and rotor speed were measured. From the data and calculated brake engine efficiency presented in Reference 4, values of C_o , β , and nozzle velocity coefficient were determined (using Equation 22) for several test points. Values of velocity coefficient calculated at the highest values of β tested were 0.76, 0.75, 0.74, 0.75, and 0.75 for β 's ranging from 0.20 to 0.38. Since the brake engine efficiency presented included mechanical as well as all fluid friction effects, a value of C_{v2} of 0.75 is believed to represent a conservative lower bound on the outer-nozzle velocity coefficient for a VPRT. A corresponding lower limit for the expected engine efficiency was calculated to be about 0.47 using (Equation 22) assuming $C_{v1} = 1.0$, $C_{v2} = 0.75$, $r/R = 0.8$, and $\beta = \beta_{opt}$.

IV. POWER PLANT CYCLE EVALUATION

1. Purpose and Scope

The overall purpose of the performance analyses conducted for the various VPRT cycles considered here was to compare the resulting values of net plant geofluid effectiveness (net plant power output per unit mass flow of geofluid), with those of other competing systems, in particular advanced binary plants. Included were effects of some of the non-ideal conditions existing for geothermal plants such as the presence of dissolved gas in the geofluid. In addition, approximate values of geofluid effectiveness projected for other competing systems as found in several references are included for comparison.

A typical moderate-temperature resource (360°F) was assumed, having pumped wells which provide well-head pressures sufficiently high to keep dissolved gasses in solution and to supply saturated or slightly subcooled liquid geofluid to the plant. A wet-bulb temperature of 60°F was assumed for all the cycles studied; however, because of some uncertainty in the ability to effectively package large expansion ratio nozzles in the outer rotor of the VPRT, condensing temperatures of both 100 and 120°F were analyzed. VPRT isentropic efficiencies assumed ranged from 40 to 100%.

Total dissolved gas mass concentrations of zero and 0.003 were assumed for the geofluid, with average properties of carbon dioxide (CO_2).

It was assumed that hydrogen sulfide (H_2S) would be present, however, so the cycles were configured to discharge the H_2S (mixed with steam) to a Stretford process at one atmosphere pressure, and a shell-and-tube condenser was incorporated to prevent H_2S dissolved in the condensate from being discharged to the atmosphere in the wet cooling tower. Reference 5, published in 1980, indicates that the Stretford process, under those conditions, adds between 1 and

2 mills/KWh to the cost of power for plants in the 10 to 50 megawatt power range, a cost increment well within the uncertainty of the present cycle performance estimates.

2. Description of Cycles.

The VPRT is the primary power producer for the cycles analyzed here, and to achieve good cycle performance the turbine discharge pressure was assumed to be subatmospheric, corresponding to the 100 and 120°F condensing temperatures mentioned previously. Figure 5 is a schematic diagram of the cycle. Because of the dissolved noncondensable gas concentrations assumed for the geofluid, the condenser must be vented and pumped. To minimize the performance penalty associated with the dissolved gas a mild preflash was assumed ($\Delta T \sim 4^\circ\text{F}$) which would eliminate most of the noncondensibles passing through the VPRT. The steam and noncondensibles generated from the preflash were assumed to drive a conventional vapor turbine, whose discharge is at one atmosphere and is directed to the Stretford H_2S abatement system. The power from the turbine drives the condenser-vent compressor, whose discharge is also directed to the Stretford system. As mentioned, the condenser was assumed to be the shell-and-tube type, and is supplied with cooling water from a wet cooling tower. Makeup water for the cooling tower was considered to be supplied by the condensate assuming the dissolved H_2S level permitted this arrangement. The remainder of the condensed geofluid is reinjected.

3. Approach and Assumptions.

The general approach taken for the evaluation of geofluid effectiveness for the VPRT system shown in Figure 5 was to conduct cycle analyses at each of a number of conditions for which the ranges covered were discussed in Section IV.1. These conditions were: inlet geofluid temperature and pressure = 360°F and 273 psia; reinjection pressure = 273 psia; wet bulb temperature = 60°F; condensing

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Figure 5. VPRT cycle analyzed.

temperatures = 100 and 120°F; dissolved noncondensable gas mass fractions (CO_2 properties) = zero and 0.003; engine isentropic efficiencies = 0.4, 0.7, 1.0.

For each set of cycle conditions selected, calculations were made for several preflash pressures, and the condensing pressure was found for each, at which the vapor turbine power output just matched the vapor compressor input power required. VPRT output power, total power required to pump condensate, and cooling tower parasitic power requirements were estimated for each flash temperature (or flash pressure), and the flash temperature resulting in maximum net plant power was determined. Figure 6 illustrates a typical optimization of flash temperature for 3000 ppm dissolved noncondensibles (0.003 mass fraction), an engine isentropic efficiency of 0.7, and a condensing temperature of 100°F. The figure shows gross power, the several parasitic losses, and net plant output, all expressed in terms of Watt hr/lbm geofluid, plotted as functions of the flash temperature. Also shown is the condenser pressure. The optimum flash temperature for these conditions was found to be about 355°F. Similar flash temperature optima were found for the other conditions.

Preflash Estimates. The amount of dissolved gas coming out of solution during the preflash was estimated using mass and energy balances for the noncondensable gas and geofluid, assuming zero heat of solution for the dissolved gas, and a modified Henry Constant of

$$H' = 25.6 \times 10^{-6} \frac{\text{mass fraction in solution}}{\text{psia partial pressure}}$$

for the noncondensable gas dissolved in geofluid. This constant was estimated from Reference 6 as appropriate for CO_2 .

Vapor Turbine Approximations. For simplicity in estimating power produced by the vapor turbine, two turbines were visualized, one expanding the steam from the flash, the other expanding the

Figure 6. Effect of preflash temperature on VPRT cycle performance.

noncondensable gas (CO_2 properties). Each turbine was assumed to expand its own fraction of the total vapor flow from the preflash with an inlet temperature equal to the flash temperature and an inlet pressure equal to the partial pressure of the corresponding vapor from the flash. The exit conditions were found for which the turbines expanded the vapors such that their exit temperatures were equal and the sum of their exit pressures was equal to one atmosphere. This approach was selected rather than treating the mixture as a perfect gas, because it was recognized that the water vapor would undergo some condensation during the expansion.

Vapor Compressor Approximation. For the condenser-vent vapor compressor the mixture of water vapor and noncondensable gas was treated as a perfect gas mixture. The values of specific heat and gas constant were estimated based on the mass fractions of the constituents at each condensing temperature and partial pressure of the noncondensable gas present in the condenser.

Condensate Pumping Assumptions. Condensate pumping requirements were estimated by assuming that the condensate used for makeup water was sufficient to replace 1.33 times the water evaporated (4 cycles of concentration), and was pumped from the condenser pressure to the cooling tower. The remainder of the condensate was assumed to be pumped to the geofluid inlet pressure.

General Assumptions. Several additional assumptions, generally applied throughout the calculations, are listed below:

1. Downhole geofluid pumping requirements (Watt hr/lbm geofluid) were assumed to be the same for all cases (including the requirements of the advanced binary cycle which was compared with the VPRT cycle), and were not included.
2. Piping and condenser frictional pressure drops were neglected.

3. Vapor turbine efficiencies were assumed to be 80% with a 1% degradation for each "average" percent of moisture formed in the turbine.
4. Vapor-compressor and liquid-pump efficiencies were assumed to be 0.75 and 0.80.
5. The wet cooling towers were assumed to provide cooling water at 70° to the condensers. Cooling water flow was selected to maintain minimum approach temperature differences (pinch points) in the condensers of 10°F.
6. As discussed in Reference 7, cooling-tower parasitic losses in Watts were estimated from earlier work as 0.077 times the cooling water flow in lbm/hr for a cooling-water temperature rise (ΔT_{CW}) of 20°F. For $\Delta T_{CW} = 20^\circ\text{F}$, adjustments to this factor were made to account for changes in pumping power required for the modified cooling water flows.
7. Water properties were taken from the 1967 ASME Steam Tables. The specific heat at constant pressure and gas constant used for CO_2 were 0.1989 and 0.0451 Btu/lbm°R.

4. Results of Power Cycle Evaluation

Values of net plant geofluid effectiveness for the optimum flash temperature at each of the VPRT cycle conditions chosen were plotted as functions of engine isentropic efficiency in Figure 7.

Effectiveness values are shown for zero and 0.003 mass concentrations of dissolved noncondensable gas, and for 100 and 120°F condensing temperatures. The curves show that the variable having major influence on geofluid effectiveness is engine efficiency. Condensing temperature and noncondensable gas concentration, over the ranges considered, show significantly smaller effects.

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Figure 7. Plant performance for Velocity Pump Reaction Turbines.

The crosshatched region shows the range of expected conditions and the resulting range of net plant geofluid effectiveness expected for the VPRT systems. The minimum engine efficiency, e_{VPRT} , for the crosshatched region is shown as 0.47, a value which was determined for the lower bound of outer-rotor nozzle efficiency (C_{v2}) of 0.75 and a rotor radius ratio of 0.8. The maximum engine efficiency for the crosshatched region is shown as $e_{VPRT} = 0.77$ which corresponds to a rotor radius ratio of 0.85, perhaps a little more difficult to design, and a C_{v2} of 0.90, slightly less than the velocity coefficient measured by LLL (Reference 2) for a nozzle having a small inlet quality. Values of system geofluid effectiveness are predicted to reach values as high as perhaps 9 to 9.5 Watt hr/lbm geofluid if the outer-nozzle velocity coefficient of 0.9 can be obtained when supplied with zero-inlet-quality geofluid.

5. Performance Estimates for Alternate Systems

To show how the performance of a VPRT system might compare with competing systems, preliminary estimates of geofluid effectiveness of several alternate cycles for producing geothermal electric power were made for a resource temperature of about 360°F. The estimates have been plotted on Figure 7 as bars to indicate geofluid effectiveness without regard for isentropic efficiency.

Advanced Binary Plant. The first alternate shown consists of an advanced binary plant utilizing a mixed hydrocarbon working fluid (0.96 isobutane/0.04 heptane) vaporized at a supercritical pressure and condensed in a countercurrent flow condenser. Performance of this plant was estimated as discussed in Reference 8 for the same resource and ambient wet-bulb temperatures (360 and 60°F) as for the VPRT systems. The geofluid effectiveness of about 9.3 Watt hr/lbm for this binary plant falls near the maximum end of the expected performance range for the VPRT systems.

Dual-Flash-Steam Plant. Estimates of geofluid effectiveness made by T. W. Lawford at EG&G for dual-flash steam plants utilizing 360°F geofluid (saturated liquid) indicated magnitudes between about 4.2 and 5.5 Watt hr/lbm depending on the amounts of total dissolved solids and dissolved noncondensable gases (References 9 and 10). For the present comparisons a value at the top of this range, 5.5 watt hr/lbm, was selected, and is shown in Figure 7; it falls near the low end of the expected-performance range for the VPRT systems.

Bi Phase RST System. System studies made by Cerini and Hays (Reference 11) concluded that use of the Bi Phase Rotary Separator Turbine (RST) in combination with a steam turbine results in a "Bi Phase RST System" which provides a 10 to 20% improvement in geofluid effectiveness over that of a dual-flash steam system. Geofluid effectiveness for the Bi Phase RST System, therefore, was estimated at about 6.3 Watt hr/lbm geofluid. This value falls in the lower half of the expected VPRT performance range.

Helical Screw Expander. Elliot (Reference 12) has indicated that the peak efficiency obtained with a helical screw expander with inlet steam qualities ranging from 0 to 0.6 was 0.48 for an atmospheric exhaust. Because of the limited volume ratio of the expander (~ 17) a reduced exhaust pressure would tend to lower the expander efficiency. The engine efficiency of 0.48 corresponds to a system performance equivalent to the most pessimistic value predicted for the VPRT. In Reference 13 Elliot indicated that a plant using a 60% efficient helical screw expander in combination with a steam turbine would result in a system having a 17% higher geofluid effectiveness than for a dual-flash steam system. For a 50% efficient expander (Reference 12), the helical screw expander system would have a performance comparable to that of the Bi Phase RST System. Accordingly, the Helical Screw and Bi Phase RST System performance values are depicted by a single bar in Figure 7.

LLL Axial Flow Impulse Turbine. Because of droplet impingement effects and limited values of nozzle efficiency, the Lawrence Livermore Laboratory (LLL) axial flow impulse turbine has not achieved performance levels as high as predicted for improved designs. Austin and Lundberg in Reference 2 indicated that engine efficiencies up to a maximum of 0.5 should be achievable today (assuming a nozzle velocity coefficient of 0.94 and a full admission configuration). This engine efficiency would place the predicted geofluid-effectiveness range between about 4.9 and 6.0 Watt hr/lbm for the 360°F resource condition as shown in Figure 7.

LLL RORT System. The LLL Radial Outflow Reaction Turbine (RORT) is the VPRT without the inner rotor ($\frac{r}{R} = 0$). With a nozzle velocity coefficient of 0.9 (assumed to be a reasonable upper limit for the VPRT) the engine efficiency reaches a maximum of 0.56. The corresponding range of geofluid effectiveness values lies between 5.5 and 6.8 watt hr/lbm geofluid.

JPL Axial Flow Impulse Turbine. Elliot, in Reference 1, indicated that the efficiencies of optimized two-phase axial-flow impulse steam-water turbines of the type being investigated by the Jet Propulsion Laboratory (JPL) would be estimated to be in the mid 50% range. On this basis the geofluid effectiveness of the JPL axial flow impulse turbine system would range between 5.4 and 6.7 watt hr/lbm geofluid for the conditions assumed in the present study, and would be essentially the same as for the LLL RORT system. One bar is shown in Figure 7 to represent both systems.

V. CONCLUSIONS

The overall conclusion reached is that the Velocity Pump Reaction Turbine (VPRT) appears to be a good candidate as the primary power producer in a well-head total-flow geothermal-electric conversion system for moderate temperature resources in the 360°F range.

Specific conclusions were drawn as follows:

1. The VPRT configuration can maintain the geofluid in a pure liquid state while major changes in flow direction occur through the machine. Two-phase expansion takes place only in the outer-rotor nozzle where droplet accelerations normal to the flow direction are minimized. This characteristic should reduce or eliminate the major problem of droplet impingement in rotor blading, which has impeded the achievement of high efficiencies for the axial-flow impulse total-flow devices previously investigated.
2. Engine isentropic efficiencies as high as 0.77 can be projected for the VPRT without relying on the formation smaller droplets than those experienced in previous work.
3. Preliminary performance comparisons of various geofluid-expansion cycles indicated that all other total-flow and dual-flash systems reviewed performed in the lower half of the range of potential geofluid effectiveness values predicted for the VPRT system.
4. The upper end of the geofluid effectiveness range expected for the VPRT total-flow system is competitive with performance projections for advanced binary cycle conversion systems. Because of the relative simplicity of the VPRT

system, which avoids use of expensive or flammable working fluids utilized in binary systems, the VPRT system may prove to be attractive for a well-head plant.

5. It is believed that additional investigation of the VPRT is indicated by the present preliminary evaluation. This work should consist of the experimental determination of geofluid effectiveness for the VPRT in subscale hardware which can accommodate evaluation of a number of outer-rotor nozzle configurations.

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