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**NEW CONCEPTS FOR CONVERTING THE
ENERGY IN LOW- TO MEDIUM-TEMPERATURE
LIQUIDS, WITH EMPHASIS ON
GEOTHERMAL APPLICATIONS**

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PREFACE

The concepts described in this report were developed solely by P. A. House as part of his work in the Lawrence Livermore Laboratory Geothermal Program. This report is written to encourage further development by those interested in and dependent on advances and innovation in energy conversion technologies.

A. L. Austin

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NEW CONCEPTS FOR CONVERTING THE ENERGY IN LOW- TO MEDIUM-TEMPERATURE LIQUIDS, WITH EMPHASIS ON GEOTHERMAL APPLICATIONS

ABSTRACT

The Geothermal Development Program at Lawrence Livermore Laboratory has produced several novel expanders for liquids of low to medium temperatures ($\sim 180^{\circ}\text{C}$). A unique radial outflow reaction turbine (RORT) has been developed and laboratory-tested; results indicate that 50% engine efficiency is achievable. This work has led to a new concept called the velocity pump reaction turbine (VPRT), which could significantly increase the gross engine efficiency of the RORT. VPRT and its modifications are a unique family of turbines created specifically for expanding liquids to produce shaft work at potential engine efficiencies of up to 70%. Such devices, if used between the two separation stages of a double-flash system, could increase the overall power output by 15-20%, reducing power costs by at least 10% for about a 3% increase in capital costs. We discuss geothermal applications with emphasis on geopressured resources. Also, these machines are suitable for utilizing solar heated fluids and waste heat sources from industrial processes.

INTRODUCTION

Until recently, turbine prime movers using water as the working fluid were developed as either vapor turbines or hydraulic turbines. Because fossil fuels were abundantly available to produce the high-temperature heat of vaporization needed for economical binary cycles or Rankine cycles, little attention was paid to processes and turbomachinery for the direct expansion of hot water. Development of turbines based on the Rankine cycle and operating at high temperatures dominated the direction and growth of energy conversion technologies.

Now, however, increasing energy consumption and dependence on oil imports have created a need to generate power from alternative, nonfossil energy sources characterized by lower temperatures. Two examples are geothermal water deposits and waste heat. For the rare case of vapor-dominated geothermal resources, conventional steam turbines suffice; for the more abundant hydrothermal reservoirs, in which the wellhead products range from totally liquid (hot water) to a two-phase mixture that is mainly liquid (80% or more of the mass flow rate), there is need for a turbine that can expand hot water directly. Also, this type of turbine is becoming important for power production from industrial waste heat, which exists mainly in the form of hot liquids or gases too low in temperature to drive a binary or Rankine cycle economically.

The Lawrence Livermore Laboratory Geothermal Program, recently terminated by the Department of Energy's Division of Geothermal Energy, included development of expanders for two-phase mixtures, with emphasis on application of the total flow concept. The total flow concept postulates direct single-stage expansion of two-phase wellhead products. Theoretically, the concept represents the thermodynamic upper bound on cycle efficiency; potentially, it will produce the maximum work output per pound of inlet fluid. Practical applications depend on the development of efficient two-phase expanders.

For high-temperature ($\sim 300^{\circ}\text{C}$) resources yielding two-phase wellhead products, the single-stage impulse turbine is the most suitable for direct expansion to subatmospheric conditions for maximum power

output. For low- to medium-temperature ($\sim 180^\circ\text{C}$) resources with two-phase products, there are several candidate expanders—impulse, impulse/reaction, and pure reaction turbines and, to a limited extent, positive displacement machines. Among these, pure reaction turbines appear most promising for expanding saturated and compressed liquids.

Our studies have produced several novel expanders for the low- to medium-temperature range. A unique radial outflow reaction turbine (RORT) has been designed and laboratory-tested. The results are very encouraging and suggest that a 50% engine efficiency is achievable. This work has led to development of a new concept called the velocity pump reaction turbine (VPRT). Analyses show that the VPRT concept may increase the RORT's gross engine efficiency significantly. Also considered are variations specifically suited to utilizing geopressured geothermal resources.

The unique contribution is a family of expanders especially designed for expanding liquids to produce shaft work at potential engine efficiencies in the 50-70% range. Such devices, if used as liquid expanders between the two separation stages of a double-flash system, could increase the overall power output by about 15-20%, reducing power costs by at least 10% for an increase in capital costs of only about 3%.

RADIAL OUTFLOW REACTION TURBINE (RORT)

The RORT is specifically designed¹ to produce power from the expansion of saturated or compressed liquid at temperatures below 180°C . This machine employs a two-stage nozzle configuration for efficient expansion of the compressed liquid. Results of the tests are very encouraging and are reported under separate cover.²

In these tests, the highest efficiency measured was 32.7%, which was below the peak efficiency for this small prototype. Extrapolations to larger machines indicate that an efficiency near 50% is credible. Thus, the RORT could perform attractively either in conjunction with conventional flashed steam systems or alone, as a total flow expander for liquid. A rough estimate indicates that a 7.5-MW RORT would cost somewhere between \$0.5 and \$1.5 million and could add about 15% to the power output of a conventional 50-MW plant. This means a 10% reduction in busbar energy cost for a very modest capital cost.

The important result of this work is that we have established the feasibility of an efficient low-cost liquid expander. Further, the basic concept broadens the opportunities for economical power production from lower temperature geothermal sources and, possibly, for utilization of waste heat from industrial processes.

We also present new design concepts that should give even greater performance than the RORT. The first is the velocity pump reaction turbine (VPRT) and the others are variations specifically modified for expansion of geopressured fluids that contain both hydraulic and thermal energy components. Theory and applications of each are discussed in the following sections.

VELOCITY PUMP REACTION TURBINE (VPRT)

With reference to Fig. 1, the inner rotor of radius, r , rotates with angular velocity, ω , and the outer rotor of radius, R , rotates with angular velocity, Ω . Liquid of enthalpy, h , enters the center rotor and is pumped outward. The pump work is

$$\int \frac{dp}{\rho} = \int_0^r r\omega^2 dr = \frac{r^2\omega^2}{2} .$$

Hence, the ideal relative velocity leaving the center rotor is

$$v_{1i} = \sqrt{2 \left(\frac{r^2\omega^2}{2} \right)} = r\omega . \quad (1)$$

Fig. 1. Velocity pump reaction turbine (VPRT) concept.

The absolute velocity is

$$V_{1i} = v_1 + u_1 = r\omega + r\omega = 2r\omega \quad (2)$$

and the power supplied to the velocity pump is

$$P_{1i} = \dot{m}(r_1\omega_1) (2r\omega) = 2\dot{m}r^2\omega^2, \quad (3)$$

where $\dot{m}$ is the mass flow rate of fluid. Let the nozzle velocity coefficient be η_1 . Then,

$$v_1 = \eta_1 r\omega,$$

$$V_1 = (1 + \eta_1)r\omega \quad , \quad (4)$$

$$P_1 = \dot{m}(r\omega) (1 + \eta_1)r\omega = \dot{m}(1 + \eta_1)r^2\omega^2 \quad (5)^*$$

For the outer rotor, the ideal moment of momentum, T_i , is

$$T_i = \dot{m}[2r^2\omega + R(v_{2i} - R\Omega)] \quad . \quad (6)$$

$$v_{2i} = \sqrt{V_{is}^2 + \Omega^2(R^2 - r^2)}$$

where V_{is} is the ideal velocity resulting from the isentropic expansion of the fluid through the outer nozzle due to the enthalpy drop, Δh_{is} ; hence,

$$V_{is} = \sqrt{2\Delta h_{is}}$$

as if the nozzle were fixed in space. The term $\Omega^2(R^2 - r^2)$ is the centrifugal head. Let η_2 be the nozzle velocity coefficient of the outer nozzle, so that the actual relative velocity is

$$v_2 = \eta_2 \sqrt{V_{is}^2 + \Omega^2(R^2 - r^2)} \quad . \quad (7)$$

Using Eqs. (4) and (7), the actual moment of momentum is

$$T = \dot{m}[(1 + \eta_1)r^2\omega + R\eta_2\sqrt{V_{is}^2 + \Omega^2(R^2 - r^2)} - R^2\Omega] \quad .$$

The net power output of the system is

$$P = T\Omega - P_1 \quad ,$$

which is

$$P = \dot{m} [(1 + \eta_1)r^2\Omega\omega + R\Omega\eta_2\sqrt{V_{is}^2 + \Omega^2(R^2 - r^2)} - R^2\Omega^2 - (1 + \eta_1)r^2\omega^2] \quad .$$

This reduces to

$$P = \dot{m}[(1 + \eta_1)r^2(\Omega\omega - \omega^2) + R\Omega\eta_2\sqrt{V_{is}^2 + \Omega^2(R^2 - r^2)} - R^2\Omega^2] \quad . \quad (8)$$

To find the optimum conditions, we can set

$$\frac{\partial P}{\partial \omega} = (1 + \eta_1)r^2(\Omega - 2\omega) = 0.$$

*Note: We should also include a pump efficiency, but, since the efficiency is unknown, we will not charge it to the VPRT in this analysis.

Hence,

$$\omega = \frac{\Omega}{2}. \quad (9)$$

This condition is also obvious from the continuity requirement that the liquid annulus must have the same tangential velocity as the inner surface of the outer rotor, i.e.,

$$v_1 + u = r\omega + r\omega = r\Omega, \therefore \omega = \frac{\Omega}{2}.$$

Substitute Eq. (9) into Eq. (8) and define the velocity ratio as $V = R\Omega$. Thus,

$$\beta = \frac{R\Omega}{V_{is}} = \frac{U}{V_{is}}. \quad (10)$$

Then,

$$P = \frac{\dot{m}V_{is}^2}{2} \left[\left(\frac{1+\eta_1}{2} \right) \frac{r^2}{R^2} \beta^2 + 2\beta\eta_2 \sqrt{1 + \left(1 - \frac{r^2}{R^2} \right) \beta^2} - 2\beta^2 \right], \quad (11)$$

and the gross engine efficiency of the VPRT is

$$e_{VPRT} = \frac{P}{\frac{\dot{m}V_{is}^2}{2}} = \left(\frac{1+\eta_1}{2} \right) \frac{r^2}{R^2} \beta^2 + 2\beta\eta_2 \sqrt{1 + \left(1 - \frac{r^2}{R^2} \right) \beta^2} - 2\beta^2. \quad (12)$$

It is interesting to note then where $r = 0$, the configuration in Fig. 1 reverts geometrically to the RORT and Eqs. (11) and (12) reduce to the performance characteristics of the RORT, as they should.

Because the fluid remains in the liquid state through the expander, the inner nozzle is very efficient. Hence, without much loss in accuracy and generality, we can set $\eta_1 \cong 1.0$ so that Eq. (11) reduces to

$$P \cong \frac{\dot{m}V_{is}^2}{2} \left[\left(\frac{r^2}{R^2} + 2 \right) \beta^2 + 2\beta\eta_2 \sqrt{1 + \left(1 - \frac{r^2}{R^2} \right) \beta^2} \right] \quad (13)$$

and the engine efficiency is

$$e_{VPRT} = \frac{P}{\frac{\dot{m}V_{is}^2}{2}} = \left(\frac{r^2}{R^2} + 2 \right) \beta^2 + 2\beta\eta_2 \sqrt{1 + \left(1 - \frac{r^2}{R^2} \right) \beta^2}. \quad (14)$$

The distinguishing feature of the VPRT is that adding a velocity pump allows maximum performance at finite velocity ratios. Figure 2, a plot of Eq. (14) with $\eta_2 = 1.0$, shows this.

The effect of an imperfect nozzle ($\eta_2 \neq 1.0$) is shown in Fig. 3 for the specific cases of $r/R = 0.7$ and $r/R = 0.8$. These curves also show the reduction in the optimum velocity ratio to values much lower than 1.0. The advantages are obvious: lower tip speed, U , results in lower wheel stresses, which in turn reduce design problems.

It should be noted that these performance curves do not include the effect of the velocity pump's efficiency. Since pump efficiency is unknown, we choose here to plot only the gross engine efficiency, as noted in Figs. 2 through 4.

Figure 4, a parametric plot of Eq. (14), is included to illustrate an advantage of the VPRT. The dashed line represents the upper bound on performance. Of course, $r/R = 1.0$ is not practicable, nor are nozzle coefficients greater than 0.9 likely in the rotating nozzle.

Fig. 2. Ideal performance characteristics of the VPRT, with $\eta_2 = 1.0$.

Fig. 3. Effect of nozzle performance on VPRT output: (a) $r/R = 0.7$.

Fig. 3. Effect of nozzle performance on VPRT output: (b) $r/R = 0.8$.

Fig. 4. VPRT performance.

Fig. 5. Optimum velocity ratios to produce maximum power output.

For any given nozzle coefficient and any desired r/R , Fig. 5 gives the optimum velocity ratio, β_{OPT} , for maximum engine efficiency. For example: With a nozzle coefficient of 0.8 and an $r/R \cong 0.9$, Fig. 4 shows that the peak engine efficiency is about 60%—a 50% increase over the output of the RORT ($r/R = 0$). Figure 5 shows that the mechanical design must accommodate the increase in speed ratio from $\beta = 0.58$ to $\beta = 0.78$. This may be restrictive for higher enthalpy drops. However, these results show that the VPRT has the potential to be a significant improvement in the technology of liquid expanders, particularly for the lower temperature energy sources.

APPLICATIONS TO GEOPRESSURED RESOURCES

Geopressured geothermal resources are unusual and potentially large energy sources. The hot water deposits are as deep as 15,000 ft and are subjected to overburden pressures as high as 10,000 psia downhole. A unique feature is that these deposits may contain large quantities of dissolved methane. Because of the low ($\sim 150^\circ\text{C}$) temperatures and high well costs, it is unlikely that central station power production from the thermal and hydraulic energy components will be economically competitive. The economics change, however, if significant quantities of methane can be produced. Hence, geopressured resources should be thought of as fossil energy—not geothermal—resources. If the resource is developed for natural gas, geothermal energy becomes a by-product. Consequently, it will be important to develop a low-cost efficient energy conversion system specifically suited for utilizing low-temperature geothermal energy components. Applicability of the VPRT to this situation is discussed in the following section.

Geopressure Velocity Pump Reaction Turbine (GVPRT)

For a given, Δp_G , with overpressure due to the geopressure, the head is

$$H = \frac{\Delta p_G}{\gamma} , \quad (15)$$

where γ is the specific weight of the fluid. With reference to Fig. 1, the ideal relative velocity leaving the center rotor is

$$v_{1i} = \sqrt{2g \left(H + \frac{r^2 \omega^2}{2g} \right)} \quad (16)$$

and the absolute velocity at r is

$$V_{1i} = v_1 + u_1 = \sqrt{2gH + r^2 \omega^2} + r\omega \quad (17)$$

Thus, the power supplied to the velocity pump is

$$P_{1i} = \dot{w} \left(\frac{V_1^2}{2g} - H \right) = \dot{m} \left(r^2 \omega^2 + r\omega \sqrt{2gH + r^2 \omega^2} \right) \quad (18)$$

The moment of momentum on the outer rotor is

$$T_i = \dot{m} [rV_1 + R(v_{2i} - R\Omega)] \quad (19)$$

where

$$v_{2i} = \eta_2 \sqrt{V_{is}^2 + \Omega^2 (R^2 - r^2)} ,$$

because $\Delta p_G = p_0 - p_1$ where p_1 is the saturation pressure of the liquid in the annulus at r . Further, consistent with this definition, we set, as before,

$$V_{is}^2 = \sqrt{2gJ\Delta h_{is}} ,$$

where Δh_{is} is the isentropic enthalpy drop occurring through the outer nozzle only and J is the mechanical equivalent of heat. As before, we assume $\eta_1 \cong 1.0$, so that the power output is

$$P = T\Omega - P_1 .$$

Consequently

$$P = \dot{m} \left[r^2 (\Omega\omega - \omega^2) + r(\Omega - \omega) \sqrt{2gH + r^2 \omega^2} + R\Omega\eta_2 \sqrt{V_{is}^2 + \Omega^2 (R^2 - r^2)} - R^2 \Omega^2 \right] . \quad (20)$$

From continuity in the annulus between rotors,

$$r\omega + \sqrt{2gH + r^2 \omega^2} = R\Omega ,$$

and

$$\frac{\omega}{\Omega} = \frac{1}{2} \left(1 - \frac{2gH}{r^2 \Omega^2} \right) \quad (21)$$

Recall that we defined the velocity ratio as

$$\beta = \frac{R\Omega}{V_{is}} = \frac{U}{V_{is}}.$$

If we substitute into Eq. (2) and reduce, the power output becomes

$$P = \frac{\dot{m}V_{is}^2}{2} \left[\frac{2gH}{V_{is}^2} - \left(\frac{r^2}{R^2} - 2 \right) \beta^2 + 2\beta\eta_2 \sqrt{1 + \left(1 - \frac{r^2}{R^2} \right) \beta^2} \right]. \quad (22)$$

For simplicity, we will define Z as the ratio of available thermal energy to available hydraulic energy:

$$Z = \frac{J\Delta h_{is}}{H}. \quad (23)$$

But,

$$V_{is}^2 = 2gJ\Delta h_i$$

and

$$Z = \frac{V_{is}^2}{2gH} \quad (24)$$

where $J = 778 \text{ ft-lb/Btu}$. Also, note that from Eqs. (24) and (21) that

$$\frac{\omega}{\Omega} = \frac{1}{2} \left(1 - \frac{1}{\frac{r^2}{R^2} \beta^2 Z} \right). \quad (25)$$

Since we defined the enthalpy drop, Δh_i , to be only across the outer nozzle, the engine efficiency is

$$e_{\text{GVPR}} = \frac{P}{\frac{\dot{m}V_{is}^2}{2} + mgH} = \frac{P}{\frac{\dot{m}V_{is}^2}{2} \left(1 + \frac{1}{Z} \right)}.$$

Hence, from Eq. (22),

$$e_{\text{GVPR}} = \frac{\frac{1}{Z} + \left(\frac{r^2}{R^2} - 2 \right) \beta^2 + 2\beta\eta_2 \sqrt{1 + \left(1 - \frac{r^2}{R^2} \right) \beta^2}}{1 + \frac{1}{Z}}. \quad (26)$$

This equation is consistent with Eq. (14). If no geopressure exists, then $H = 0$, $Z = \infty$, and Eq. (26) reduces to Eq. (14), as it should. So,

$$e_{\text{GVPR}} = \frac{1}{1 + Z} + \left(\frac{Z}{1 + Z} \right) e_{\text{VPR}} \quad (27)$$

Fig. 6. Performance of geopressure velocity pump reaction turbine (GVPRT). $Z = \text{thermal energy/hydraulic energy} = J\Delta h_i/H = V_{is}^2/2gH$.

where e_{VPRT} is given by Eq. (14). This result is plotted in Fig. 6. Given the e_{VPRT} from Eqs. (3) and (4), the performance for the geopressed case can be found directly once the amount of overpressure head, H , is specified. Also, the speed ratio can be found from Eq. (25) and Fig. 5. It is of interest to mention a special case which we will call the geopressure combination turbine (GCT).

Geopressure Combination Turbine (GCT)

Figure 7 shows a special case of the GVPRT with fixed inner rotor and gas vent ports. This simplified version may facilitate gas removal. With these changes, Eq. (25) becomes

$$\omega = 0 = \frac{1}{2} \left(1 - \frac{1}{\frac{r^2}{R^2} \beta^2 Z} \right).$$

$$\therefore \beta^2 = \frac{1}{\frac{r^2}{R^2} Z}. \quad (28)$$

If we substitute in Eq. (26), the engine efficiency e_{GCT} is now

$$e_{GTC} = \frac{2}{1+Z} \left[1 - Z\beta^2 + \beta\eta_2 \sqrt{Z^2(1+\beta^2) - Z} \right]. \quad (29)$$

The optimum speed ratio, β_{OPT} , can be found by setting

$$\frac{\partial e_{GCT}}{\partial \beta_{OPT}} = 0$$

Fig. 7. Geopressure combination turbine (GCT). This special case of the GVPRT has a fixed inner rotor and gas vent ports.

and solving for β_{OPT} . This works out to be

$$\beta_{OPT} = \left(\frac{U}{V_{is}} \right)_{OPT} = \sqrt{\left(\frac{Z-1}{2Z} \right) \left(\frac{1}{\sqrt{1-\eta_2^2}} - 1 \right)} \quad (30)$$

as the optimum speed ratio for maximum efficiency. From Eq. (28), the corresponding value of r/R can be found as

$$\left(\frac{r}{R} \right)_{OPT} = \frac{1}{\beta_{OPT} \sqrt{Z}}, \quad (31)$$

which gives the only value of r/R for the maximum efficiency. The value Z is specified by the known thermal and hydraulic components of the geopressed wellhead product, and η_2 is given. Therefore, we can solve Eq. (30) for the optimum speed ratio. By substituting into Eq. (29), we can solve Eq. (31) for maximum efficiency as well as for r/R .

Fig. 8. Performance of geopressure combination turbine (GCT) plotted for three outer nozzle velocity coefficients.

Fig. 9. Optimum velocity ratios for geopressure combination turbine (GCT).

Figure 8 shows efficiency over a range of Z for three cases— $\eta_2 = 0.9, 0.8$, and 0.7 . Figure 9 shows the optimum speed ratio for the same values of η_2 . Evidently, there is no gain in efficiency over the GVPRT, although the machine is simpler with its fixed inner rotor. However, the operation range is limited since r/R must be less than unity.

CONCLUSIONS

A unique radial outflow reaction turbine—RORT—has been designed and laboratory-tested. This machine is especially suited to expanding low- to medium-temperature saturated liquids; test results indicate that a 50% engine efficiency can be achieved. As a result of this work a new concept, the velocity pump reaction turbine—VPRT—has been developed. Calculations show that up to a 50% increase in gross engine efficiency over the RORT can be achieved with the VPRT. It should be noted, however, that we do not consider internal losses (e.g. fluid friction and gearing performance), so the gain on net engine efficiency is not known. Nevertheless, the concept of the VPRT is a significant contribution to expander technology, and should be developed further by design and performance-testing.

The novel contribution is that a family of expanders has been created solely for expansion of liquids to produce shaft work at potential engine efficiencies of 50-70%. This means that if the VPRT is used as a liquid expander between the two separation stages in a double-flash system the overall power output can be increased by about 15-20%. Thus, power costs would be reduced by at least 10% for about a 3% increase in capital costs. Also, these machines are especially important for utilizing low temperature energy sources such as those encountered in geothermal deposits, solar heated fluids, and waste heat sources from industrial processes.

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