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Interim Report on Performance Tests of a Total Flow Impulse Turbine
for Geothermal Applications

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INTERIM REPORT ON PERFORMANCE TESTS
OF A TOTAL FLOW IMPULSE TURBINE FOR
GEOTHERMAL APPLICATIONS

W. J. Comfort III

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Abstract

A new two-phase expander has been successfully tested in the laboratory. The expander is a part of the energy conversion effort of Lawrence Livermore Laboratory's geothermal energy development program. This report: 1) briefly describes the machine, 2) presents the experimental results, 3) compares a numerical performance model to the results, 4) presents predictions of the same rotor with full admission, and 5) indicates how performance improvements can be made.

An engine efficiency of 23 percent for a single-nozzle test was measured. Performance predictions of a numerical model agree well with experimental results. Full admission performance, based upon the numerical model and achievable nozzle thrust coefficients indicate that full admission engine efficiency between 38 and 48 percent can be realized with present technology. Droplet size reduction and enhanced two-phase flow analysis techniques should make it possible to reach the research goal of 70 percent engine efficiency.

Introduction

A new single-stage two-phase expander has been tested in the laboratory. The expander is a part of the energy conversion effort of the Lawrence Livermore Laboratory (LLL) geothermal energy development program to develop the Total Flow concept [1], [2]. The Total Flow process consists of expanding the total wellhead brine-steam mixture directly through a mixed phase expander to sub-atmospheric condensing conditions for maximum energy extraction to drive an electric generator. If successfully developed, the Total Flow process has the potential for the highest electrical energy output per unit mass of wellhead fluid. This will result in a corresponding reduction in number of wells and, hence, initial capital costs. In addition, because of inherent design simplicity there are increased opportunities for solution of the serious chemistry problems associated with high salinity brines, and perhaps more importantly, reduction in capital costs of conversion systems.

This interim report: 1) briefly describes the machine, 2) presents the experimental results, 3) compares a numerical performance model to the results, 4) presents predictions of the same rotor with full admission, and 5) indicates how performance improvements can be made. A later report [3] describing the selection, design, fabrication, and detailed analysis of the expander is being prepared.

After consideration of many candidate expanders, an axial flow impulse turbine was chosen as the most promising expander type for the design

conditions of the Total Flow process. In order to understand its performance, a laboratory test machine was built for testing under the controlled conditions available in the LLL geothermal test facility [4], using a two-phase steam/water mixture to simulate the fluid dynamics and thermodynamics of the output of wells drilled into typical water dominated geothermal reservoirs.

The LLL geothermal program is specifically aimed at solution of the problems associated with utilization of the high temperature, high salinity brines found in the Salton Sea Geothermal Field (SSGF) near Niland, California. The engineering development of Total Flow expanders and their mechanical performance testing is purposely limited initially to the laboratory. A parallel field program is ongoing to solve problems of scale control, erosion, and corrosion of turbine components. The results of these major activities will ultimately be combined to design a brine tolerant Total Flow system for field testing by operation with the SSGF brines. At this time, the impulse turbine tested in the laboratory is solely for the purpose of determining performance characteristics, and is not intended for field testing because of material limitations. Consequently, this report will cover only this activity. A companion report on the program achievements in scale control and materials testing is in preparation.

Description of the Expander and Test Apparatus

A special rotor was built so that wide blades with sharp leading and

trailing edges could be designed in anticipation of special problems associated with two-phase fluid dynamics. Blade width was made large to increase the radius of curvature of the flow path so that droplets would have a maximum probability of passing between the blades without impacting their surfaces. Sharp edges were necessary in order to minimize phenomena associated with supersonic relative velocities within the blade passage.

In order to simplify construction, shorten development time, and to minimize costs, the rotor was designed with integral blades and was fabricated from a 7075-T6 aluminum forging. Aluminum was chosen for its high strength-to-weight ratio and machinability. Figure 1 shows the rotor being machined on LLL's numerically controlled 5-axis milling machine. The 1.27 cm. diameter cutter is shown in Figure 2 during the machining process. Figure 3 shows the rotor before an electroless nickel plating [5] was applied to the blade surfaces for water droplet erosion protection. The plated rotor is not suitable for brine exposure. It was designed to have sufficient life for the laboratory testing program. A complex plating process was developed for this specific application. An overview of the eight large tanks required to form the plating line is shown in Figure 4.

The final machining on the rotor hub was completed after plating and the finished part was then assembled with the steel shafting. The rotor assembly was balanced, Figure 5, and assembled into the rotor housing, Figure 6.

Figure 1
Rotor being machined on LLL's 5-axis milling machine.

Figure 2

Detail showing 1.27 cm diameter cutter removing material from the blade passages. Blade height was 12.2 cm.

Figure 3

Rotor after machining of blades was completed.

Figure 4

Overview of electroless nickel plating line showing the eight large tanks used for the process. The rotor, mounted in a special fixture to maintain rotation during plating is shown about to enter the first cleaning solution.

Figure 5
Balancing being done at the Westinghouse Machine Maintenance
Plant in Emeryville.

Figure 6

Rotor assembled into rotor housing. The front half of the housing has not been installed in this photograph.

Figure 7
Nozzle mounted on four bar linkage for thrust measurement.

Figure 8

Dynamometer assembly to which the expander was later attached for performance testing.

A single-nozzle was used for the machine because of limited flow-rate capabilities of the test facility. The nozzle [6] was suspended with a four-bar linkage so that axial thrust could be measured independently during performance testing of the expander, Figure 7.

A dynamometer assembly, Figure 8, consisting of a hydraulic pump/motor was used both as a brake and as a motive source. Performance of the expander was measured by using the brake mode, while disk friction and windage losses were measured by using the hydraulic motor to drive the rotor.

Single-Nozzle Expander Experimental Results

The operating conditions used for the performance testing are summarized in Table 1. -These operating conditions are selected to be representative of the thermodynamic conditions of well outputs from the reservoirs of the SSGF. Figure 9 shows the measured engine efficiency versus blade speed ratio for the machine. The maximum engine efficiency for the single-nozzle test was observed to be 23 percent at a blade speed ratio (the ratio of the mean blade speed to the homogeneous isentropic mixture velocity of the fluid exiting the nozzle) of .24. Testing was done on three separate occasions and showed good repeatability. Operation was smooth and quiet.

The nozzle thrust coefficient was .83 during testing with the rotor. This was lower than the .94 thrust coefficient measured during previous

Table 1 Test conditions

Fluid state at nozzle entrance

$$P_o = 2.53 \text{ MPa (367 psia)}$$

$$X_o = \text{quality} = 14\%$$

$$T_o = 224^{\circ}\text{C (434}^{\circ}\text{ F)}$$

$$h_o = 1.2 \text{ MJ/kg (526 BTU/lbm)}$$

$$\dot{m} = 0.59 \text{ kg/s (1.31 lbm/s)}$$

Exhaust pressure and temperature

$$P_e = 0.0137 \text{ MPa (2 psia)}$$

$$T_e = 126^{\circ}\text{F}$$

Fig. 9. Measured engine efficiency versus blade speed ratio.

tests of the nozzle alone. The nozzle thrust coefficient, C_T , is defined as:

$$C_T = \frac{T}{\dot{m}V_{is}} \quad (1)$$

where T is the measured thrust, $\dot{m}$ is the measured mass flow rate, and V_{is} is the homogeneous isentropic mixture velocity.

Although radial pressure gradients associated with blade rotation are not as severe in two-phase conditions as would be expected in standard vapor turbines, they were probably large enough to explain the lower performance. Larger separation between the nozzle and rotor as well as improved nozzle design should eliminate this problem.

Performance Model

A numerical model was developed and was used to predict expander performance. The model included calculation of droplet trajectories within the blade passage to determine what happened to the liquid. In addition to standard vapor turbine losses, special losses associated with droplet impact and removal from the blades were included in the model. Static cascade measurements in two-phase flow were used to determine the performance of blades identical to those on the rotor. This information was used to normalize the vapor flow portion of the model. Measured values of combined disk friction and windage were used to normalize the disk friction and windage model in the predictive technique.

The model was designed to allow extrapolations of performance both for full admission and for changes in droplet diameter. Single-nozzle performance predictions were very successful, as will be shown, and provide strong credibility to the extrapolations.

The present numerical model is only adequate for impulse designs and cannot be used to select detailed blade contours because the vapor and liquids are not fully coupled, i.e. droplets are affected by the vapor, but the vapor is assumed to be unaffected by the droplets. In order to adequately design more advanced blade and nozzle contours, a completely coupled model for two-dimensions, based upon an existing one-dimensional two-phase flow calculation technique [7] is necessary and is currently being developed.

The details of the numerical model are beyond the scope of this presentation. To provide the reader with some insight, however, into the extrapolation between the single-nozzle and full admission results, a simplified model will be presented here to demonstrate some of the characteristics to be expected.

Assuming for this simplified model that no slip exists between the vapor and liquid entering the turbine blades, one may write the expression for blade efficiency e_b . For an impulse turbine with the ratio of mass flow rate of vapor to mixture equal to R , relative velocity coefficients for the vapor and the liquid equal to λ_v and λ_L , blade speed ratio (blade speed/nozzle velocity) β , and nozzle angle θ , one may write:

$$e_b = 2\beta[R(1 + \lambda_v) + (1 - R)(1 + \lambda_L)](\cos\theta - \beta) \quad (2)$$

For a given blade speed ratio and nozzle angle, no higher conversion efficiency can be achieved than that indicated by (2) with $\lambda_V = \lambda_L = 1$. If all of the droplets impact the blade, then $\lambda_L = 0$, which reduces the efficiency of the liquid kinetic energy conversion by 50 percent.

If some of the liquid escapes impact, the blade efficiency must be expanded to include those effects. To maintain a simplified approach, the analysis herein will assume that all of the liquid impacts the blade, i.e. $\lambda_L = 0$. Actual numerical results to be presented in the next section do not require this simplification.

Defining the optimum speed ratio, (2) may be differentiated with respect to the speed ratio and set equal to zero as shown:

$$\frac{\partial e_b}{\partial \beta} = 0 = 2[R(1 + \lambda_V) + (1 - R)(1 + \lambda_L)](\cos\theta - 2\beta) \quad (3)$$

with the result that:

$$\beta|_{\text{optimum}} = \frac{1}{2}\cos\theta$$

It should be observed that the optimum blade speed ratio is independent of the velocity coefficients and is only weakly dependent on the nozzle angle for angles less than 20 degrees.

To convert the blade efficiency to an engine efficiency, e_e , the losses due to disk friction and to radial liquid removal, e_d and e_c , must be

subtracted from the blade efficiency modified by the square of the nozzle thrust coefficient:

$$e_e = C_{Tb}^2 - e_c - e_d \quad (4)$$

where

$$e_c \propto \alpha f(\beta^2)$$

$$e_d \propto g(\rho, \beta^{2.85})$$

The quantity α , in the expression for the liquid removal represents the fraction of the liquid mass flow rate which must be removed radially, while ρ , in the disk friction and windage loss expression refers to the effective density of the fluid surrounding the rotor.

The previously derived optimum speed ratio is valid only when the disk friction and windage, and pumping losses are small relative to the first term in (3). For a single-nozzle test, the disk friction and windage and liquid removal losses can severely affect the blade speed ratio associated with the peak efficiency, shifting it toward a lower value. This, in turn, decreases the blade efficiency because the blade speed ratio is reduced. For full admission, the disk friction and windage loss, for a given blade speed ratio, is inversely proportional to the mass flow rate, while the liquid removal loss is proportional to the mass flow rate if all droplets impact the blade and are radially removed.

Thus, the liquid removal loss is not diminished relative to power output for increased admission, while the disk friction and windage loss fraction is reduced. As admission is increased, the optimum blade speed ratio increases, but since the liquid removal loss is proportional to the blade speed ratio squared, the loss fraction increases.

With this brief introduction to some of the phenomena involved, the actual numerical model results will next be compared to the experimental results and then the numerical model will be used to extrapolate to full admission and to an advanced design.

Single Nozzle Comparisons and Extrapolations to Full Admission

Figure 10 also shows verification that the numerical model compares very well with the experimental results. As noted, the upper calculated band assumes that disk friction and windage losses are associated with the vapor density only, while the lower band assumes that they are a function of the two-phase mixture density. The upper and lower limits of each of the bands are determined by nozzle modeling uncertainty and will be fully described in [3].

The results of static cascade testing indicated that all droplets impacted the blade passages and exited at negligible velocity. Droplet size estimates indicate that the droplets are probably between 2.7 and 7 microns in diameter. Results of the performance predictions indicate that they are greater than 2.2 microns because this is the largest size

Fig. 10. Comparison of experimental data with turbine performance model predictions

of droplet that can pass through the blade passage without impact. Hence, if droplets are greater than 2.2 microns in diameter, all droplets impact the blade passage--as is indicated by the results.

The blade speed ratio indicated for the peak power point of the experimental data lies between the two calculated bands. This is significant because, as was shown, the optimum blade speed ratio for an impulse turbine is primarily dependent upon the disk friction and windage and losses associated with radial removal of liquid droplets which have impacted the blades. The fact that the optimum blade speed ratio for the bounding calculations spans the measured optimum speed ratio indicates that the mechanisms of disk friction, windage, and liquid removal are well modeled. Assuming all droplets impact the blades and are removed radially, one would expect that the vapor density would best approximate the fluid surrounding the majority of the rotor. The results indicate that this is not the case. Liquid droplets may be splashing off of the rotor housing walls and remixing with the vapor, increasing the effective density of the fluid mixture around the rotor and consequently increasing the friction forces as well as the power necessary to accelerate relatively low velocity water particles.

Figure 12 shows extrapolations to a full admission performance for the existing rotor design and to an advanced design. It should be noted that the full admission results reflect the nozzle thrust coefficients previously measured for the nozzle in the laboratory. Otherwise, the calculation contains the same parameters for the full admission calculation as for the partial admission calculation, except that the

Fig. 11. Extrapolations using the numerical performance model

admission was increased. The increase in performance is due primarily to the diminishment of the effects of disk friction and windage since this loss becomes very small in comparison to full admission power input, while it is rather large compared to that of partial admission. This diminishment allows the rotor to attain a higher speed ratio, enhancing the engine efficiency

Two bands are used to represent the extrapolated performance of a full admission design for droplet diameters greater than 2.2 microns, and with an assumed nozzle thrust coefficient of .94. The difference between the upper and lower band (dashed and solid lines, respectively) is the result of radial liquid removal losses. The loss associated with the radial removal of liquid from the blades is included in the lower band (solid lines). Though this loss is certain in the single-nozzle design, it is not clear that it will exist in the full admission machine because the nearly continuous flow may force the liquid through the blades without significant radius change. The upper band (dashed lines) indicates the displacement of the lower band (solid lines) due to removal of the losses associated with radial pumping of the liquid from the blades. The upper bound of both bands is calculated assuming disk friction and windage loss is controlled by the vapor density and that the slip ratio (the ratio of average liquid velocity to average vapor velocity at the exit of the nozzle) is approximately .9. The lower bound of both bands is calculated assuming that the disk friction and windage loss is controlled by the mixture density and that the slip ratio is approximately .7. For the full admission performance, the slip ratio, rather than the fluid density

assumed for the disk friction and windage, is the major cause of the difference between the upper and lower bounds of each band.

The advanced design assumes that the droplet size has been reduced to 1 micron, and that the thrust coefficient remains .94. Under these circumstances it would be expected that the liquid and vapor exit the nozzle with very nearly the same velocity. The upper bound of the lower band (solid lines) is based upon vapor turbine correlations of vapor friction losses and the assumption of vapor-controlled disk friction and windage. The lower bound of the lower band is based upon blade vapor performance consistent with results from static cascade tests and assumes that the mixture density controls disk friction and windage. The upper band for this case (dashed lines) indicates the performance if liquid radial removal losses are omitted. The change resulting from the omission is not as great as in the previous case (full admission with droplet diameters greater than 2.2 microns) because only a small fraction of the liquid impacts the blades when the droplets are 1 micron in diameter.

The single-nozzle, full admission extrapolation, and advanced design results may be compared in another way. Table 2 represents a summary of losses associated with the peak efficiency points corresponding to the data and extrapolations shown in Figure 11. It should be remembered that this table does not indicate causality, but rather a loss summary at a particular operating condition. For example, from the single nozzle results, one might conclude that the disk friction is not a deterministic loss mechanism. However, disk friction is not an independent function,

Table 2. Loss distribution at peak efficiencies for single nozzle and for extrapolations to full admission and advanced designs

	Single Nozzle D > 2.2 μm Percent	Full Admission Bounds Existing Design D > 2.2 μm		Full Admission Bounds Advanced Design D $\leq$ 1 μm	
		Percent	Percent	Percent	Percent
<u>Loss Distribution</u>					
● Slip, Liquid/Blade, Vapor Friction	17	33	34	14	10
● Liquid Pumping	2	6	0	2	0
● Disk Friction and Windage	4	<1	<1	<1	<1
● Geometry & Non-Optimum Blade Speed Ratio	33	13	7	6	6
● Nozzle Inefficiency	21	10	11	11	11
<u>Total Losses</u>	77	62	52	33	27
<u>Engine Efficiency</u>	23	38	48	67	73

just as the other losses listed are not independent functions. In the case of disk friction, its nearly cubic relationship with blade speed ratio and its large relative magnitude compared to the power output capability of a single nozzle test forces the blade speed ratio to be considerably off optimum. This, in turn, results in the significant loss ("geometry and non-optimum blade speed ratio") which results directly from blade efficiency relationships as previously shown.

In summary, considerable improvement can be expected in turbine performance as a result of droplet size reduction. The performance indicated for droplets smaller than 1 micron also requires that sufficient two-dimensional analytical methods are developed to improve two-phase blade and nozzle design. Indeed, it appears possible to achieve the research goal of 70 percent engine efficiency for the machine. Using the test conditions (Table 1) as representative, a 70 percent engine efficiency converts to a resource utilization rate (water rate) for pure water of 47 lb/kw-hr, gross, i.e. exclusive of the parasitic loads which would occur in a typical power plant application. This represents a significant improvement over conventional conversion systems operating with the same conditions. The important implication is that the number of wells required for a given power output is correspondingly reduced, which in turn, reduces the initial capital investment since the cost of wells is dominant.

Conclusion

A two-phase axial flow impulse expander has been successfully tested in the laboratory. An engine efficiency of 23 percent for a single-nozzle test was measured. Performance predictions of a numerical model agree well with experimental results. Full admission performance extrapolations using the numerical performance model and based upon achievable nozzle thrust coefficients indicate that full admission engine efficiency between 38 to 48 percent can be realized with present technology. Droplet size reduction to 1 micron and enhanced two-phase flow analysis techniques for design of blades and nozzles should make it possible to achieve the research goal of 70 percent engine efficiency.

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References

- 1 Austin, A. L., Higgins, G. H., Howard, J. H., "The Total Flow Concept For Recovery Of Energy From Geothermal Hot Brine Deposits", UCRL-51366, April, 1973, Lawrence Livermore Laboratory, Livermore, California.
- 2 Austin, A. L., "Prospects for Advances in Energy Conversion Technology for Geothermal Energy Production", Proceedings of the Second United Nations Symposium on the Development and Use of Geothermal Resources, San Francisco, U. S. A., May 28-29, 1975, Vol. 3, pp. 1925-1937.
- 3 Comfort, W. J. III, "The Design and Evaluation of A Two-Phase Turbine for Low Quality Steam-Water Mixtures", to be published.
- 4 Weiss, H., Shaw, G., "Geothermal Two-Phase Flow Test Facility", UCRL-76409 April 1975, Lawrence Livermore Laboratory, Livermore, California.
- 5 Wiesner, H. J., Comfort, W. J. III, Frey, W. P., Gardiner, C. C. Jr., Kelley, W. K., "Electroless-Nickel Plating of a Large Turbine Rotor for the Geothermal Program", UCID-17253, September 1976, Lawrence Livermore Laboratory, Livermore, California.
- 6 Alger, T. W., Designed and tested the expander nozzle, Private Communication.
- 7 Comfort, W. J., Alger, T. W., Giedt, W. H., Crowe, C. T., "Calculation of Two-Phase Dispersed Droplet-In-Vapor Flows Including Normal Shock Waves", UCRL-78426 (Preprint), July 1976, Lawrence Livermore Laboratory, Livermore, California.

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