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TWO-PHASE REACTION TURBINE

Technical Progress Report
For the Period January-May 1999

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1) Investigation of Relevant Literature on Two-Phase Flow Turbomachinery and Flashing

We have updated our familiarity with relevant literature of direct and indirect importance for success of this project. These included not only fluid dynamics of two-phase flashing flows but also of number of other practical hardware oriented subjects.

We have assembled a collection of over 130 technical articles, reports, books, dealing with technical issues of importance for this project. We have further, in critical and practice-oriented manner, studied and analyzed this literature. Many detailed pieces of valuable information and knowledge have been gained and further clarified.

In addition we have made copies and analyzed about 40 USA and foreign patents dealing with two-phase flow turbomachines. Reading these patents gave us wider insights into thinking of the inventors about various aspects of energy conversion involving two-phase flow. Search of these patents, however, showed that our invention is unique in drastically reducing the two main efficiency loss mechanisms in "prior art" two-phase reaction turbines.

Summary of the finding from the literature can be the best explained via enclosed Figures 1 through 34 and their titles.

2) Computation of Rotating Two-Phase Flashing Flows

We have discussed this issue with Professor Clayton Crowe who is the USA leading expert in computational fluid dynamics of two-phase flows. He is the principal author of a textbook on that subject. In addition he had spent extended time at the Lawrence Livermore Laboratory during their R&D work on analysis and testing of the LLL two-phase reaction turbine. He was the key in-house expert and consultant on computational fluid dynamics for that project.

We have also attended one-day workshop on computational fluid dynamics for turbomachines. The workshop has been organized by AEA Technology, Engineering Software Inc., which is a worldwide leading company with very sophisticated software for computation of three-dimensional fluid flow in high-speed turbomachinery. Their codes are used extensively by companies such as Solar Turbines International, General Electric, ABB, Siemens, and many others.

Subsequently we had discussions with key scientists and software engineers at the AEA. Their codes are indeed very sophisticated but for single phase fluid dynamics. In two-phase flow fluid dynamics, especially involving flashing, very substantial advancements are still needed. AEA is at this time developing computer code that would be capable of handling fluid dynamics of non-equilibrium condensation of steam flowing through high-speed cascade of turbine blades.

Another project that AEA is just now undertaking is development of a sophisticated code for predicting cavitation flow through turbomachinery. These are development in (for our project) right direction but are still not good enough for modeling of flashing two-phase flow through. AEA has told us that development of a sophisticated flashing two-phase fluid dynamic code for turbomachinery would cost about \$100K and would take a year to develop. We believe that this would definitely worthwhile undertaking. It, however, is beyond funding and schedule capability of the present project. Development of such code would be more appropriate when larger two-phase flow turbines would have to be designed and when a budget and schedule for such very substantial computational task would be appropriate.

Fortunately, our extensive analysis of the physics and practical parameter of two-phase flow in rotating turbine indicate that there is substantial redundancy in design of curved rotating two-phase nozzles which drastically reduce separation forces and abruptness in flashing. This means that a less sophisticated computer code can be reasonably used to design the curved rotating nozzles.

We have acquired such less sophisticated but reasonable predictive two-phase flow code from Professor Clayton Crowe. We are in process of making modifications in that code in order to predict needed curvatures of the rotating

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nozzles. These nozzles are then designed in such a way that it can be realistically expected that lateral separation forces are reduced by two orders of magnitude (about 100 fold) compared to the LLL turbine. In the same time the rate of depressurization is also reduced by two orders of magnitude (about 100 fold) as compared to the LLL turbine. In this way dramatic improvements in the nozzle and turbine efficiencies are expected.

3) Design of a Prototype of Two-Phase Flow Turbine

In order to be able to conclusively evaluate the improvement in performance due to new turbine design it should be attempted to keep all other facts unchanged. This means that the testing parameters such as the inlet and the outlet pressures, inlet temperature, fluid flow rate, should be kept the same as in the case of tests of the LLL turbine. Diameter of the turbine rotor will also be the same as in the LLL turbine; i.e. it will be 16 inches. Number of nozzles will also be unchanged, i.e. four nozzles.

The Table III lists parameters and performance data for testing of the LLL two-phase reaction turbine. The inlet hot water temperatures were from 345 to 371 F. The inlet hot water pressures were from 188 to 200 PSIA. The exhaust backpressures were from 2.54 to 3.67 PSIA. The water flow rates were from 1.37 to 2.3 lb./s. It is our intention to match these parameters as close as possible.

Using the modified one-dimensional two-phase flow nozzle code we are exploring possible curvatures of the nozzles for some of the flow parameters used at the LLL. The optimal change in the shape of the nozzles does not change much with changing parameters. This is encouraging, meaning that the curvature of the nozzles is not much sensitive to the actual flow parameters. On the other hand in the LLL tests only one shape of the nozzles was used for all the test runs as well. The shape of the LLL nozzles was drastically different from an optimal shape. In that way the LLL nozzles exposed the two-phase flow to lateral separation forces of the order of 6000 g-s (gravitational accelerations), and to drastic abruptness in flashing. This was the main cause of low nozzle and turbine efficiencies of the LLL turbines.

It is known that, for stationary two-phase flow nozzles, the optimal shape of the centerline is a straight line. The reason for it is that the longitudinal (stream-wise along the nozzles) acceleration is on the order of 10,000 g-s (g is gravitational acceleration). In the same time nominal lateral acceleration is only one g.

In rotational straight nozzles used at the LLL the lateral acceleration were up to 6000 g-s. It is therefore completely reasonable to expect that appropriately curved rotational two-phase flow nozzles would perform considerable better than the straight nozzle. The improved performance of the nozzles means higher discharge bulk (mainly of the liquid) fluid velocity at the discharge of the nozzle.

4) Computation of Optimal Curvature of the Two-Phase Rotating Nozzles

We have developed method of calculation of the optimal curvature of the rotating two-phase flow flashing nozzles. This method calculates the shape of the nozzles in such way that the lateral separation forces are reduced theoretically to zero but in practice probably reduced by factor of order of 100 compared to those forces in the LLL nozzles. The methods also reduce the abruptness in flashing by factor of about 100 as compared to the LLL rotating nozzles.

At the present time we are in process of improving the computer code for calculation of curvature of the rotating nozzles.

The above "curvature" computation is coupled to an algorithm for computation of nozzle performance. The computation of the nozzle performance permits for variation of most of the governing parameters such as flow rate, inlet and exit pressure, inlet temperature, cross-sectional shape of the nozzles, etc. The rotational speed parameter can be varied in the curvature computational part of the algorithm.

5) Analysis of Other Two-Phase Turbine Design Issues

Extensive design analysis has been carried out. There are many factors that mutually interact and influence the actual design. Some of these factors are: sealing, bearing, materials, critical speed, manufacturability, connection of turbine shaft to dynamometer or electricity generator, introduction of the inlet fluid into the rotating hollow shaft, ease of voluminous steam discharge and its effective condensation, prevention of air leaks into the condenser, continuous evacuation of air from the condenser, possible recovery of some of the exiting kinetic energy of the liquid, separation of two discharged phases, etc.

It is next to impossible to, in written form, describe various considerations and mutual interaction of the above parameters. Design process is in many ways continuous iterative improvement process. Its outcome are engineering drawings and subsequently manufactured prototype.

6) Engineering Design Drawings of Two-Phase Turbine

Based on all numerous above mentioned considerations engineering design of a 50 kW prototype of the novel two-phase turbine is now underway. Extensive discussions of various designs have been done.

Due to the fact that the working fluid will be hot water and back pressures will be

as low as 2.5 PSIA the volume of exhausting two-phase flow, at quality of 0.2, will be considerable. This imposed necessity of having as short distance as possible between the turbine (actually) rotor discharge and a condenser. This was the main reason that we chose the turbine housing to be a half-dome shape bolted directly on the condenser.

Furthermore we consider it worth while to explore if some kinetic energy of discharging liquid, which will have 80% mass fraction of total flow, could be recovered. In the chosen design the liquid impinges tangentially on the housing and keeps spinning maintaining major part of its kinetic energy. This liquid flow is then "scooped up" with pick-ups, which contain short diffusers. In this way an appreciable portion of the exiting kinetic energy will be converted in increased pressure of flowing liquid. We should say that this recovery of exiting kinetic energy is of secondary importance and of some "bonus" value. By far the greatest increase in the turbine efficiency will achieve by drastically reducing the two main loss mechanisms in the rotor as discussed earlier.

The turbine rotor has short 1.5 inch diameter shaft, which is supporter by ball bearing on both sides. This arrangement makes the rotor very stable, sturdy and pushes the critical rotating speed very high, i.e. much higher than the operational speeds.

The hot water is introduced through a hollow shaft from bellow. Only lower part of the shaft is hollow. The rotor

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FIGURES 1-34

SE : separator expander
ST : steam turbine

Figure 1: Schematic and T-s Diagram for a Single Flash Direct Brine Power Plant. (SE represents some type of two-phase flow expander delivering useful mechanical power.)

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TWO-PHASE ENERGY CONVERSION FOR SPACE

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- AXIAL FLOW IMPULSE TURBINE
- ROTARY SEPARATOR TURBINE (RST) (RADIAL IMPULSE TURBINE)
- LIQUID REACTION ROTOR TURBINE
- TWO-PHASE REACTION ROTOR TURBINE (HERO)
- TWO FLUID COMPONENTS TURBINES

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Figure 16: Air Velocity for Onset of Entrainment in Horizontal Parallel Flow

Figure 17: The Three Periods of Bubble Growth.

Figure 18: Temperature Distributions at a Bubble Interface.

FLASHING OF LIQUID JETS

Fig. 6. Flashing jet 10X. Type B, $D = 0.020$ in., $P = 120$ lb./sq. in. $T = 284^{\circ}\text{F}$.

Fig. 5. Flashing jet 10X. Type B, $D = 0.031$ in., $P = 120$ lb./sq. in. $T = 295^{\circ}\text{F}$.

Fig. 4. Flashing jet 10X. Type A, $D = 0.030$ in., $P = 131$ lb./sq. in. $T = 287^{\circ}\text{F}$. One inch from orifice.

Fig. 3. Flashing jet 10X. Type A, $D = 0.040$ in., $P = 120$ lb./sq. in. $T = 286^{\circ}\text{F}$.

Figure 19: Flashing of Liquid Jets.

Figure 20: Flashing of Liquid through a Short Nozzle.

CHOKED FLOW FLASHING

Mass flux variation with L/D for D = 3.2 mm.

Figure 21: Effect of Sub-Cooling Liquid (Excess above Saturation Pressure) on Critical Mass Flow through Discharge Tubes.

Figure 22: Effect of Initial Bubble Density on Critical Mass Flux (Sozzi and Sutherland)

Figure 23: Flashing of Highly Over-Pressurized Diesel Fuel in IC Engine.

Figure 24: Flashing of Highly Over-Pressurized Diesel Fuel in DeLaval Nozzles.

NOZZLE PRESSURE DISTRIBUTIONS

LLL ROTATING NOZZLES

DIMENSIONS IN MILLIMETERS

"STRAIGHTENED-OUT" FAS ROTATING NOZZLES

Figure 25: Nozzle Pressure Distributions and Nature of Flashing in the LLL and FAS Nozzles.

RORT engine efficiencies as a function of angular velocity and the liquid capture factor, α . This RORT had a 16-in.-diam rotor; a 350-psia saturated liquid inlet; a 20-psia exhaust (20% quality); a 60% slip; 95% liquid and 75% two-phase steam nozzle efficiencies; and 8° steam and 4° liquid discharge angles.

Figure 26: Lawrence Livermore Laboratory Predicted Efficiencies of their 25kW Two-Phase Turbine.

Figure 27: Component Accelerations in Curved Rotating Nozzles.

REACTION ROTOR

$$0 < w < u$$

Accelerations:

- 1) $\frac{u^2}{R}$ centripetal
- 2) $2\frac{u}{R}w$ Coriolis
- 3) $\frac{w^2}{r}$ centripetal in relative motion
- 4) $\frac{dw}{dt}$ along streamlines (not shown)

Figure 28: Example of Two-Phase Rotating Nozzles which Avoid Separation and Abrupt Flashing.

Figure 29: Centerline Curvature of Two-Phase Rotating Nozzles as Function of the Velocity Slip (Very Small Influence).

1. Base Case
2. +10% Slip
3. +20% Slip

CURVING OF RELATIVE MOTION

Figure 30: Curvature of Relative Motion as Function of Ratio of Relative Fluid to Local Rotor Velocity Showing Effective Entrainment of Liquid Boundary Layers.

Figure 31: Radius of Relative Motion for No Acceleration Transverse to Streamlines

Legend

State

Fluid	P(MPa)	h(kJ/kg)
T(C)	Flow(kg/s)	x(kg vapor/kg total)

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CONVENTIONAL SINGLE FLASH GEOTHERMAL POWER PLANT

SINGLE FLASH GEOTHERMAL POWER PLANT RETROFITTED WITH TWO-PHASE TURBINE

NOTE: NET INCREASED POWER OUTPUT IS 35%

Figure 32: Example of Improvement Achieved by Replacing Single Flash Throttling Valve by an Efficient Two-Phase Turbine.

GEO THERMAL BINARY TRILATERAL CYCLE POWER PLANT

6/29/93

CONDENSATION
TEMPERATURE
IS 32.22 C

LEGEND.

State

Fluid	P(Bars)	H(kJ/kg)
T(C)	Flow(kg/s)	X(kg vapor/kg total)

Brine

R-11

Air-

Figure 33: Geothermal Trilateral R-11 Cycle Power Plant Employing Efficient Two-Phase Turbine (Condensation Temperature = 32.22 C)

GEO THERMAL BINARY TRILATERAL CYCLE POWER PLANT

6/29/98

CONDENSATION
TEMPERATURE
IS 26.67 C

LEGEND

State

Fluid	P(Bars)	H(kJ/kg)
T(C)	Flow(kg/s)	X(kg vapor/kg total)

Brine
R-11
Air

Brine	6.178	675.54
60	114.53	0

Brine	5.498	132.74
31.67	114.53	0

R-11	23.90	85.29
26.67	513.013	0

R-11	1.1193	83.103
24.36	513.013	0

R-11	22.55	206.42
153.89	513.013	0

ELECTRICAL OUTPUT		
8588 kWe		
-390 kWe		
-1056 kWe		
7142 kWe NET		
BRINE UTILIZATION		
17.32 W-hr/kg		

R-11	1.1193	189.68
26.67	513.013	0.583

R-11	1.1193	264.33
26.67	299.087	1

Air	1.01325	454.37
21.11	5154.0	1.0

Air	?	444.26
11.11	5154.0	1.0

Figure 34: Geothermal Trilateral R-11 Cycle Power Plant Employing Efficient Two-Phase Turbine (Condensation Temperature = 26.67 C)

TABLE I-III

TABLE 1. ANALYSIS OF LATERAL FORCES IN ROTATING TWO-PHASE NOZZLES

FACTS	LLL TURBINE	TURBINE INVENTED BY DR. FABRIS
1) AXIS OF ROTATING NOZZLES	TWO STRAIGHT PIECES	APPROPRIATELY CURVED TO CANCEL OUT LATERAL SEPARATION FORCES
2) CROSS-SECTIONAL SHAPE OF NOZZLES	THREE FOLD ABRUPT ENLARGEMENT AT THE TROAT	NORMAL DELAVAL CROSS- SECTIONAL AREAS
3) MAGNITUDE OF LATERAL FORCES(CENTRIFUGAL AND CORIOLIS) SEPARATING TWO PHASES IN THE HIGHLY ACCELERATING NOZZLE FLOW	ON AVERAGE 6000 TIMES LARGER THAN GRAVITATIONAL WEIGHT OF LIQUID	ON AVERAGE LESS THAN 50 TIMES WEIGHT OF LIQUID
4) LATERAL MEAN LOCATION OF TWO PHASES	ON AVERAGE LARGELY SEPARATED	ON AVERAGE THE SAME (HOMOGENEOUS)
5) EFFECTS ON STREAMWISE ACCELERATION OF FLUID	LARGE STREAMWISE VELOCITY SLIP BETWEEN TWO PHASES	MUCH LOWER VELOCITY SLIP BETWEEN TWO PHASES
6) SLIP LOSS IN NOZZLE EFFICIENCY	30%	4%

**TABLE II. ANALYSIS OF FLASHING CHARACTERISTICS IN ROTATING TWO-PHASE
NOZZLE**

FACTS	LLL TURBINE	TURBINE INVENTED BY DR. FABRIS
1) (P-P _{sat}) AT THE NOZZLE ENTRANCE DUE TO CENTRIFUGAL FORCES IN ROTATING UPSTREAM PASSAGES	10 * P _{sat}	~ 0.1 * P _{sat}
2) LENGTH OF THE CONVERGING PART OF THE NOZZLE	0.05 R	0.3 R
3) NUCLEATION SITES(TINY BUBBLES) IN LIQUID NEEDED TO INITIATE FLASHING	ABSENT(DISSOLVED)	PRESENT
4) LIQUID DEPRESSURIZATION RATE	100000 [P _{sat} /s]	100 [P _{sat} /s]
5) INITIAL LOCATION OF FLASHING	COUPLE OF INCHES DOWNSTREAM OF THE TROAT	UPSTREAM OF THE TROAT
6) FLASHING DYNAMICS	ABRUPT, EXPLOSIVE, TOO FAR DOWNSTREAM OF EQUILIBRIUM LOCATIONS	GRADUAL, CLOSE TO PROPER EQUILIBRIUM LOCATIONS
7) EFFECT ON NOZZLE EFFICIENCY	INEFFICIENT REACTIVE PROPULSION	EFFICIENT REACTIVE PROPULSION
8) QUANTITATIVE LOSS IN NOZZLE EFFICIENCY	30%	3%

TABLE III: Performance Data From Test Runs on 18 July 1978

Point	Inlet Conditions			Case	Turbine	Brake	Brake Eng.
	Temp. (°F)	Press. (PSIA)	Flowrate (LB/Sec)	Press. (PSIA)	Speed (RPM)	Power (H.P.)	Efficiency %
1	345.6	191.5	1.37	2.57	4161	13.36	22.11
2	361.6	200.1	1.33	2.54	4062	14.10	20.75
3	366.3	195.5	1.42	2.64	4489	16.65	22.46
4	367.3	188.2	1.44	2.64	4602	17.07	22.44
5	369.0	196.2	1.53	2.68	4927	19.59	24.02
6	369.7	191.1	1.54	2.77	5046	19.98	24.55
7	369.7	191.7	1.70	2.90	5587	23.74	26.74
8	369.7	195.5	1.71	2.90	5524	23.44	26.36
9	370.4	195.1	1.83	2.99	6022	26.42	27.82
10	370.8	195.8	1.85	3.00	6022	26.35	27.54
11	369.9	193.0	1.98	3.13	6552	29.18	28.99
12	370.0	193.2	1.99	3.16	6586	29.44	29.41
13	370.6	193.2	2.13	3.30	7085	32.34	30.42
14	370.2	194.3	2.14	3.31	7080	32.28	30.19
15	369.3	194.7	2.29	3.44	7549	34.25	30.76
16	369.3	193.2	2.29	3.47	7545	34.15	30.67
17	369.9	192.9	2.30	3.67	8043	36.05	32.59
18	370.6	193.6	2.30	3.67	8066	36.24	32.71

FIGURES E1-E9

ENGINEERING ASSEMBLY DRAWING OF NOVEL TWO-PHASE REACTION TURBINE

Figure E1

FAS ENGINEERING, INC.

2 PHASE TURBINE
FRONTAL CROSS-SECTION

DATE: MAY 3, 1999

DIMENSION: INCHES

SCALE: 1mm=0.14in

DRAWING NUMBER 1 OF 2

Indication of Positions of
Cross-Sectional Cuts at
B-B & C-C

Figure E2

FAS ENGINEERING, INC.	2 PHASE TURBINE Cross-Sectional Cuts Positions	DATE: May 3, 1999
DIMENSION: INCHES	SCALE: 1mm=0.14in	DRAWING NUMBER 2 OF 2

Positions of O-rings and Their Inside Diameters

Figure E3

FAS ENGINEERING, INC.	2 PHASE TURBINE Inside Diameters of O-rings	DATE: May 3, 1999
DIMENSION: INCHES	SCALE: 1mm=0.14in	DRAWING NUMBER 1 OF 1

CROSS-SECTION B-B SHOWING
THE TOP VIEW OF ROTOR

Figure E4

Curved Delaval Nozzles

FAS ENGINEERING, INC.

2 PHASE TURBINE
Cross Section B-B

DATE: May 3, 1999

DIMENSION: INCHES

SCALE: 1mm=0.14in

DRAWING NUMBER 1 OF 1

CROSS-SECTION C-C

Figure E5

FAS ENGINEERING, INC.	2 PHASE TURBINE CROSS-SECTION C-C	DATE: MAY 3, 1999
DIMENSION: INCHES	SCALE: 1mm=0.167in	DRAWING NUMBER 1 OF 1

Rotor & Shaft

Figure E6

FAS ENGINEERING, INC.

2 PHASE TURBINE
Rotor & Shaft

DATE: MAY 3, 1999

DIMENSION: INCHES

SCALE: 1mm=0.098in

DRAWING NUMBER 1 OF 1

UPPER BEARING (partial dimensions)

Figure E7

FAS ENGINEERING, INC.	2 PHASE TURBINE UPPER BEARING	DATE: MAY 3, 1999
DIMENSION: INCHES	SCALE: 1mm=0.085in	DRAWING NUMBER 1 OF 2

UPPER BEARING (additional dimensions)

Figure E7a

FAS ENGINEERING, INC.	2 PHASE TURBINE UPPER BEARING	DATE: MAY 3, 1999
DIMENSION: INCHES	SCALE: 1mm=0.08in	DRAWING NUMBER 2 OF 2

HOT WATER INLET AREA (partial dimensions)

Figure E8

FAS ENGINEERING, INC.	2 PHASE TURBINE HOT WATER INLET	DATE: MAY 3, 1999
DIMENSION: INCHES	SCALE: 1mm=0.08in	DRAWING NUMBER 1 OF 2

HOT WATER INLET AREA (additional dimensions)

Figure E8a

FAS ENGINEERING, INC.	2 PHASE TURBINE HOT WATER INLET	DATE: MAY 3, 1999
DIMENSION: INCHES	SCALE: 1mm=0.078in	DRAWING NUMBER 2 OF 2

Figure E9

FAS ENGINEERING, INC.

2 PHASE TURBINE
Side Wall

DATE: MAY 3, 1999

DIMENSION: INCHES

SCALE: 1:0.071

DRAWING NUMBER 1 OF 1