TWO-PHASE REACTION TURBINE

Technical Progress Report
For the Period July-December 1999

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During the initial part of this period the concentrated effort was placed on getting comprehensive analysis and design of the turbine prototype. This was in order to be able to initiate its fabrication as needed for its building, assembling and timely testing.

In the second part of this period the effort was placed on design and other considerations needed to acquire the test rig within the limited budget based on subsequent grant from the DOE. This certainly is very challenging undertaking.

1) Analyze Other Mechanical Engineering Turbine Design Issues

Design of any turbine prototype is very challenging task in itself. Repeated iteration and extended reconsideration are needed.

Due to large volume of steam at the turbine exhaust a large flow cross-section is necessary. For this reason the stationary housing is designed in form of dome with its open end facing downwards. This dome, i.e. the turbine housing, is going to be placed on the top of a separator tank.

We have decided to avoid cantilever shaft-rotor design. Two stainless steel bearing are placed on each side of the impeller and at a short distance from it. In

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this way the critical speed of rotation will be much higher avoiding a possibility of the rotor vibrations.

Hot high-pressure water will be introduced in to the impeller through a hollow shaft. Two soft brass labyrinth seals inserts will be placed at two ends of the shaft. At the lower side, the differential pressure across the labyrinth seal is much higher. We were also concerned about leaking of some of the incoming hot water flow through the labyrinth seal. For this reason we have designed introduction of small amount of pressurized steam or air into the middle of the lower labyrinth seal. This should completely stop leaks of the main hot water supply.

The impeller itself is sandwich-like being split into two pieces horizontally through the middle of the curved nozzles. Curving of the reaction nozzles is calculated using a computer code written by us. A thin sheet of soft copper of thickness of 0.004 inches will be placed between two impeller halves. The two halves of the impeller halves will be held and be sandwiched together with about twenty recessed allen type screws. Perfect tight alignment of the two impeller halves will be accomplished by a push fit on the shaft plus a push fit on one or two dowel pins. These dowel pins will also go through the large 3.5-inch diameter part of the shaft. In such a way torque will be completely securely transferred from the impeller to the shaft. The maximum torque would occur when the shaft is jammed shut and the maximum flow is passing through the curved nozzles. Assuming maximum momentum no separation jets discharging the nozzles we get the maximum torque of 36.7 Nm. Further calculations show that two dowel pins of 1/8 inches in diameter placed at radii of 1.5 inches can carry six times larger torque than the maximum predicted.

Calculation of stress for a flat rotating disc of uniform thickness and diameter of 8.6 inches and with a hole at its centerline shows that for rotational speed of 20,000 RPM the maximum stress will be 330 MPa and located at the inner radius. Based on this and various other considerations we have chosen the rotor to be made out of stainless steel 17-PH4 which has the tensile strength of 1,310 MPa.

In addition to the above stress calculation, our design has further substantially decreased the maximum stress by more than a factor of two. This is achieved by two design modifications. First, the thickness of the rotor at the inner radius, where it attaches to the shaft, is about two times larger than at the outer radius. This considerable decreased the maximum stress.

Second, when there is no central hole in a rotating disc of uniform thickness then the maximum stress is only half as high compared to the case with a hole at the disc center? In our case for no-hole design the maximum stress would be only 165 MPa. We have taken advantage of this feature by observing that the upper part of the shaft has no hole in the center. For this reason we will machine the upper shaft from 3.5 inch diameter stock steel with an internal recess which will

help "hold" the impeller (which has much larger outer diameter) as shown on the enclosed Figures. This will reduce the maximum stress in the impeller. We expect the actual maximum stress for the rotational speed of 20,000 RPM to be at about 160 MPa, i.e. eight times lower than the maximum strength of the chosen stainless steel.

Earlier we intended to make rotor 16 inches in diameter so that it would have the same diameter as the rotor of the LLL two-phase turbine. This turned out to be too costly and too bulky to manufacture. Accordingly we have decreased the rotor to eight inches. The housing has also decreased in diameter by factor of approximately two. The weight and the manufacturing costs are basically proportional to the square of the diameter. This means that we have reduced the actual manufacturing costs by factor of close to four.

One negative consequence of the two times smaller radius is two times higher rotational speed. The biggest inconvenience that this has caused was the difficulty of finding commercially available dynamometer. However even for 10,000 RPM and 30kW power the dynamometer would cost about \$25,000. We have resolved this issue by using high-speed electrical generator with resistive load banks as a dynamometer. In order to increase the accuracy of measuring the power output, the stator of the electrical generator will be held by a flange type torque cell. In such a way exiting torque of the turbine shaft will be measured very accurately. The RPM-s will also be measured very accurately by using optical tachometer.

Manufacturing issues for the turbine and the whole test rig are continuously being reexamined and solutions improved? It is very hard and overly time consuming and ineffective to in written form explain many many issues and aspects of the design considerations. Just for an example we were given bids for appropriate dynamometers by two well established companies in Southern California, of about \$80,000 each. This was of course financially completely not possible within the project's budget. We have done very intensive and wide consideration and design of alternative dynamometer. We are satisfied that we have come up with much more economical and effective device to measure the power output of the turbine.

We also had to choose thermodynamic and flow parameters at and for which to design the first turbine prototype. Many important factors had to be weighted in choosing the key parameters. One of the key ones being available funds for building and testing the turbine prototype. Some of these considerations are actually covered by our next project, which are the building and the testing of the turbine prototype.

2) Develop the Engineering Drawings of the Turbine

The initial most innovative part of the design of the prototype turbine is design of its curved 100% reaction nozzles. Based on the governing equations we have developed a computer code, which calculates the curvature of the nozzles in approximately 1500 incremental finite difference steps along the nozzles. Cross-sectional flow areas of the nozzles were calculated using the governing equations as well as thermodynamic properties of state.

Two nozzles, rather than four, were used. In this way the cross-sectional areas are twice as large than in the LLL nozzles. We believe that this will be beneficial since the boundary layers would occupy smaller fraction of the total cross-sectional areas.

The assembly drawing explains the most about the design of this turbine. It would be way too extensive to start to elaborate on details of design of the turbine. Drawings of various parts are enclosed.

3) Design Test Facility

We have undertaken, within a modest budget, to design and (subsequently) build a test rig needed to verify the performance from our turbine. Instead of a continuously running facility we have chosen a blow down test rig, which is shown on an enclosed figure. This facility will have 200 GL high temperature (400F) and pressure (250PSIA) hot water tank connected to appropriated gas burning heating coils (tubes) of heating capacity of about 300,000 BTU/hr. The tank and the heating coils replace usual much more expensive high temperate and pressure boiler. Using this system we will first have to heat the water for about 5 hours to 400F, then we should be able to achieve high flow steady state operation of our turbine for at least 10 minutes. During the steady state operation time we will take all needed data to calculate the performance of the turbine.

The other large saving in the cost of the facility we are achieving by having an atmospheric separation tank downstream of the two-phase turbine instead of low-pressure condenser. About 85%, by mass, of the turbine discharge flow, is water. It will be contained into 300 GL separator tank. The steam is 15% of the discharge flow. The steam cannot be saved but it will be continuously discharged for the separator tank via one-foot diameter duct, which will exit through the roof of the building.

Drawings of the high temperature, and of the separator, tanks are enclosed.

Third very large savings will be accomplished in terms of equipment to measure the power output of the turbine. We have received two bids from established Southern California companies to such supply such equipment. Each of these two bids is for about \$80,000 just to measure the power output of the turbine. Subsequently we have made our own design of the assembly of equipment

needed to measure the power output of the turbine.

The main part of the dynamometer will be electrical generator. We have already purchased an electric generator capable of delivering 30kW or more. It is a surplus generator of an aircraft bomber. It is designed to operate at 6,000 RPM but also it can run at 9,000 RPM. We expect that the turbine impeller will rotate at up to 20,000RPM. Accordingly it is necessary to have a transmission RPM reduction system from the turbine shaft to the generator shaft. We have done extensive considerations of transmission systems and have selected sprocket and chain system as the most appropriate. We have already purchased a chain and two sprockets. Drawing showing the turbine assembly, the electrical generator and the transmission system. Please note that the turbine and generator shafts and the transmission are completely enclosed by thick steel shrouds. This is done in order to achieve high degree of safety.

The output of the electric generator will be taken to and electrical load bank. The load bank has various resistors so the loading in the generator, i.e. simultaneously on the turbine, can be changed at any particular constant RPM.

4) Analyze Potential Applications of the FAS Two-Phase Reaction Turbine

We were asked to compare performance of geothermal power plants that do not or do employ our turbine in various thermal cycles possible. There are many many considerations that need to be taken in account. The main considerations and comparisons of improvements in performance have been given in our earlier proposals and answers to questions submitted to the DOE.

In order to be able to more directly compare the thermodynamic and power output performance we have assumed the same state parameters of the brine entering the assumed plant and the same condensation temperature of the working fluid. If geothermal fluid now entering the plant is 90% in liquid and 10% in steam state, this that there is flashing of the liquid only brine within last top parts of the upcoming well in the ground. This can be rectified and appreciable benefit gained by using larger diameter hole for the upcoming flow, or by using a down-hole pump, which, in this case, would require a relatively small power input.

It is known that the throttling is constant enthalpy process in which the available mechanical energy is wasted. Based on the parameters given to us (335F, 10% steam quality) we have calculated the more appropriate input (**the same enthalpy**, and 0% quality) to be saturated brine at temperature of 214C. The condensation temperature we have taken to be the same in all cases, at 25C. Based on these basic assumptions, and a brine flow rate of 100 kg/s, we have calculated the following results:

a) Single Flash Conventional Plant

Assuming the efficiency of the steam turbine is 0.85 the plant power output is 7,733kW.

b) Single Flash Plant Retrofitted with Two Two-Phase Flow Turbines.

Assuming the efficiency of the two-phase flow turbines is 0.75 the plant power output is 12,727kW.

c) Dual Flash Conventional Plant

Assuming the efficiency of steam turbine to be 0.85 the plant power output is 9,775kW.

d) Dual Flash Plant Retrofitted with Three Two-Phase Flow Turbines.

Assuming the efficiency of the two-phase flow turbines is 0.75 the plant power output is 13,009kW.

e) Open Brine Trilateral Cycle

Assuming the condensation temperature at 40C and the efficiency of the two-phase flow turbine of 0.75, the plant power output is 11,782kW.

f) Open Brine Trilateral Cycle

Assuming the condensation temperature at 26.67C and the efficiency of the two-phase flow turbine of 0.75, the plant power output is 13,770kW.

g) Trilateral Cycle with Binary Fluid

Assuming the temperature difference in the receiving heat exchanger of 5C, the condensation temperature of 40C, efficiency of the two-phase flow turbine of 0.75, the plant power output is 12,053kW.

TEST RIG SET UP

TWO-PHASE TURBINE TEST RIG

TURBINE-GENERATOR SCHEMATIC DIAGRAM

PART NAMES

PART NAMES

PART NAMES

DETAIL DRAWINGS

P2

Horizontal
Dimension

Cross-Section B-B

Thread size = 1.125 in.
12 threads per inch
Minor Diameter = 1.0258 in.
Pitch Diameter = 1.0709 in.

P2

Vertical
Dimension

Key Dimension

P2

Tightening Bolt for the Rotor

Thread size = 1.125 in.
12 threads per inch
Minor Diameter = 1.0348 in.
Pitch Diameter = 1.0709 in.

P3

Vertical Dimension

Horizontal Dimension

Rotor Top View

Bolt Position Table

Position	Angle(degree)	Radius(inch)
1	157	0.8
2	57	0.8
3	85	1.35
4	120	1.83
5	152	2.25
6	182	2.74
7	200	3.06
8	220	3.35
9	9	1.35
10	52	1.83
11	101	2.40
12	138	2.98
13	165	3.42
14	182	3.77
15	198	4.04

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Dimensions in Inches

Angle	Radius	Alpha	A	B	X	Y	A/Cos(Alpha)
12.032	0.670	60.846	0.394	0.550	0.659	0.140	0.809
22.918	0.866	46.696	0.394	0.435	0.798	0.337	0.574
34.378	1.035	37.815	0.315	0.347	0.855	0.585	0.399
45.837	1.189	32.315	0.275	0.275	0.828	0.853	0.325
48.702	1.228	31.284	0.275	0.275	0.811	0.923	0.322
51.566	1.264	30.424	0.275	0.275	0.786	0.990	0.319
57.296	1.342	28.992	0.275	0.298	0.725	1.130	0.314
68.755	1.492	27.216	0.275	0.340	0.541	1.391	0.310
80.214	1.645	25.669	0.275	0.382	0.280	1.622	0.305
91.674	1.803	24.294	0.283	0.413	-0.053	1.802	0.310
114.592	2.146	21.887	0.295	0.491	-0.893	1.951	0.318
137.510	2.500	19.996	0.315	0.559	-1.844	1.689	0.335
160.429	2.870	18.449	0.335	0.627	-2.704	0.961	0.353
183.347	3.260	17.132	0.354	0.689	-3.254	-0.190	0.370
206.266	3.673	15.986	0.394	0.840	-3.294	-1.626	0.410
226.319	4.000	15.126	0.433	0.879	-2.763	-2.892	0.448
229.319	4.057	15.126	0.439	0.885	-2.644	-3.077	0.455
232.319	4.114	15.126	0.445	0.891	-2.515	-3.256	0.461
235.319	4.172	15.126	0.451	0.897	-2.374	-3.431	0.467
238.319	4.231	15.126	0.457	0.903	-2.222	-3.600	0.473
241.319	4.291	15.126	0.463	0.909	-2.059	-3.764	0.480
244.319	4.352	15.126	0.469	0.915	-1.886	-3.922	0.486
247.319	4.413	15.126	0.475	0.921	-1.702	-4.072	0.492
250.319	4.475	15.126	0.481	0.927	-1.507	-4.214	0.500

Flow Discharge

Cross-Sectional View E-E

P4

Horizontal Dimension

O-ring Spec.

Seal Science, Inc. (714) 957-1007

ID=1.114 $\pm$ 0.010

OD(Mean)=1.254

Width=0.070 $\pm$ 0.003

P4

Vertical Dimension

Insert Piece # 1

Insert Piece # 2

Labyrinth Seal Dimension

P5

Vertical Dimension

Horizontal Dimension

P6

Vertical Dimension

Horizontal Dimension

P7

T-Beam Support

Cross Section A-A

Support Pieces

Support Pieces

P8

Vertical Dimension

Horizontal Dimension

P9

Vertical Dimension

Horizontal Dimension

90 Degree Elbow

P10

Shield Box Top View

Shield Box Side View

P11

Generator Support Top View

Generator Support Side View

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NAVAIR 03-5AS-5

B25

Technical Manual Overhaul Instructions

AC GENERATOR

TYPE NO.

28B54-1-A

28B54-9-A

28B54-1-B

28B54-9-B

(Bendix)

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1 September 1967

SECTION I

INTRODUCTION

1-1. GENERAL.

1-2. This manual contains overhaul instructions and test procedure for type 28B54-1-A, Alternating Current Generator, manufactured by The Bendix Corporation, Electric Power Division, Eatontown, New Jersey, and is prepared under contract N00383-67-C-2571.

1-3. PURPOSE OF EQUIPMENT. The ac generator (figure 1-1) is used to provide an output voltage for the alternating current, aircraft electric system.

1-4. LEADING PARTICULARS. The leading particulars of the alternating current generator are listed in table I.

1-5. DIFFERENCE DATA SHEETS.

1-6. Sections II and III of this handbook contain overhaul instructions and test procedure for type 28B54-

1-A AC Generator. Overhaul instructions and test procedure for additional types are provided in Section IV by the use of Difference Data Sheets. The additional types covered in Section IV are listed in Section IV.

1-7. Overhaul instructions and test procedure for the types covered in Section IV are the same as those given in Sections II and III, except for the specific differences noted by the applicable Difference Data Sheet.

Note

"Many parts for equipment covered in this publication are provided in the form of kits. (See applicable Illustrated Parts Breakdown for details.) However, cleaning, inspection, testing and repair information is included for all parts which can be repaired to cover any emergencies caused by shortages in supply."

TABLE I. LEADING PARTICULARS

Rated Voltage (Line to neutral)	120 v
Rated Voltage (line to line)	208 v
Rated Load	30 kva
Power Factor	0.75
Frequency	400 cps
Phase	3
Rated Speed	5700 rpm
Maximum Speed for Regulation	7000 rpm
Rotation (Viewing drive end)	Counterclockwise
Weight (Max)	70 lb
Bolt Circle Diameter	10.000 in.
Drive Spline	16-tooth

NAVAIR 03-5AS-6

Technical Manual
Illustrated Parts Breakdown

AC GENERATOR

TYPE NO.

28B54-1-A

28B54-9-A

28B54-1-B

28B54-9-B

(Bendix)

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Figure 4-4. AC Generator Type 28B54-9-B

1. Generator to air duct adapter	42. Dirt slinger
2. Caplug	43. Ball bearing
3. Screw	44. Self-locking nut
4. Lockwasher	45. Lockwasher
5. Washer	46. Washer
6. Terminal block cover	47. Screw
7. Screw	48. Electrical lead
8. Washer	49. Elastic stop nut
9. Self-locking nut	50. Rectifier
10. Washer	51. Shim
11. Round nut	52. Washer
12. Washer	53. Nut
13. Beveled retaining ring	54. Insulating washer
14. Permanent magnet rotor core	55. Resistor
15. Rotor shaft	56. Rotor assembly
16. Woodruff key	57. Inner bearing retainer
17. Drive shaft assembly	58. Dust cap
18. Self-locking nut	59. Electrical receptacle connector assembly
19. Washer	60. Screw
20. Spring	61. Lockwasher
21. Front plate	62. Washer
22. Lock ring	63. Connector adapter
23. Fan assembly	64. Motor generator exciter stator
24. Woodruff key	65. Screw
25. Stator and panel assembly	66. Square washer
26. Screw	67. Terminal block
27. Lockwasher	68. Screw
28. Washer	69. Lockwasher
29. Screw	70. Washer
30. Lockwasher	71. Jumper strap
31. Washer	72. Jumper strap
32. Service liner	73. AC stator assembly
33. Screw	74. Screw
34. End bell	75. Identification plate
35. Dirt slinger	76. Screw
36. Ball bearing	77. Service liner
37. Bearing retainer	78. Screw
38. Outer bearing retainer	79. Screen
39. Screw	80. Screw
40. Bushing	81. Spacer
41. Bearing spacer	82. Housing

Figure 2-1. Type 28B54-1-A, AC Generator, Main Assembly