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TURBINE TEST REPORT

Addendum to Final Report

By
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W. D. Batton

July 1984

Work Performed Under Contract No. AC03-81CS30576

Barber-Nichols Engineering Co.
Arvada, Colorado

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Addendum to Final Report
July, 1984

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1.0 ABSTRACT

The radial inflow turbine developed for the NASA 404 program 25-ton solar air conditioner (RCWS-2-2753-G0) was tested for performance as Task 6 of contract DE-AC03-81CS30576. Using the converging-only nozzles designed for this system, a peak efficiency of 86% was measured at a pressure ratio of 2.7 and a velocity ratio of 0.66. Near the design pressure ratio of 3.5 and velocity ratio of 0.645, the measured pressure ratio of 3.5 and velocity 0.645, the measured efficiency was 84% instead of the predicted 82%. Data is presented for pressure ratios of 2.7, 3.6, and velocity ratio ranges of 0.20 to 0.85. This covers the normal operating range of interest in this machine.

The performance is better than predicted. This indicates that some of the loss coefficient values assumed during the original analysis were conservative.

2.0 INTRODUCTION

Organic Rankine Cycle Engines (ORCE) are used to convert solar energy to shaft power which can be used to drive electrical generators and air conditioning compressors. The prime power-producing element in the ORCE is the turbine. The engine power output is directly proportional to the efficiency of this turbine. Therefore, the efficiency of turbines operating on organic fluids is a very important performance parameter. The turbine type of interest here is the radial inflow (RIF) turbine.

Because of the characteristics of a typical solar power system, the engine must operate over a wide range of conditions. A typical operating day would have the machine turning on early in the morning at low solar insolation levels. This condition provides low energy levels to the turbine. As the day progresses and the sun rise to its zenith, the energy levels increase to a maximum. The turbine, at this maximum, produces high power levels from the high heads available. As the sun then heads for the horizon the energy

decreases to a minimum and the turbine shuts off.

Due to this daily operating cycle it is desirable to know the turbine performance over a wide range of conditions, not just the design point. Unfortunately, there is not a lot of information available on off-design performance of organic radial inflow turbines. This program was intended to fill in some of these gaps in knowledge.

The objective of this program was to test the radial inflow turbine from the low temperature (200°F), 25-ton solar power air conditioner originally developed for the NASA 404 program. These tests were over a wide range of operating conditions. This report describes the tests conducted and the results of those tests.

3.0 TURBINE DESCRIPTION

The turbine tested in this program is a 356 aluminum radial inflow turbine wheel. A cross section of the wheel is shown in Figure 3.1. The wheel was cast by the lost wax process from a wax produced from an RTV Silicon rubber mold. The rubber mold was cast over a master wheel. The wheel was heat treated to a T6 condition. The wheel rotates clockwise looking at the exducer end.

The converging-only nozzles were machined from 304 stainless steel on a numerically controlled mill. The nozzle flow passages are rectangular in cross section. The nozzle passage configuration is shown in Figure 3.2. The nozzle throats (15 total) are 0.269 inches wide by 0.330 inches high for a total nominal area of 1.33 square inches. The turbine inlet blade height of 0.510 and the nozzle height of 0.330 give a lap ratio of 1.55. This rather large value is due to a last minute programmatic change that reduces design power levels from 25 hp to 20 hp. There was only time to reduce nozzle height to reduce power rather than to redesign the entire turbine assembly.

-3-
FIGURE 5.1
TURBINE WHEEL
FULL SCALE

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DIMENSIONS IN INCHES

These nozzles are choked at nearly all operating points and a free expansion occurs after the nozzle exit. Barber-Nichols considered designing converging-diverging instead of converging-only nozzles for this turbine, thereby eliminating the free expansion. The area ratio of the divergent section (exit area to throat area) that such nozzles would have is very nearly unity. Therefore, the performance benefit from the converging-diverging over converging-only would have been small and the potential for poorer off-design performance existed with the converging-diverging nozzles when the nozzles operated unchoked. The converging-only nozzles were chosen for the turbine design for the above reasons and also for their ease of manufacture.

The turbine wheel rotates at 21,140 rpm and a velocity ratio of 0.645. The specific speed of 84.8 is favorable for high efficiency. The R-113 enters as nearly a saturated vapor (slightly superheated) at a pressure of 38.5 psia. The design exhaust pressure is 11 psia with a total adiabatic head of 7.65 Btu/lb. The predicted efficiency was 82%.

The turbine speed is reduced to electric motor speed through a double reduction gearbox. The intermediate shaft (4,108 rpm) carries the system feed pump sealed from the gearbox with a face seal. The output shaft (1,200 rpm) is sealed with a face seal and an outboard lip seal with an intervening oil buffer. This gearbox is lubricated with a novel system using a mixture of oil and R-113 that is recycled through the Rankine engine. This lubrication system greatly reduces gearbox losses by eliminating a seal on the high speed shaft. The gearbox then becomes part of the hermetic system of the Rankine engine.

4.0 TEST SYSTEM

The test system was assembled from a converted 25-ton, low temperature solar-driven air conditioner originally assembled for the NASA 404 program (model RCWS-2-2753-Go, S/N 6). The original engine was used as a source of hot, pressurized R-113 for turbine drive, as a source of gearbox lubrication, and for condensing the

turbine exhaust. The turbo-gearbox originally designed and built for this system was the test item. The system was modified for variable speed operation and was outfitted with proper instrumentation. The test assembly is shown in schematic form in Figure 4.1.

Referring to Figure 4.1, the test turbine receives slightly superheated R-113 at state 0. This fluid is accelerated through the converging-only nozzles to state 1 where the pressure is measured. After flowing through the turbine, the hot vapor exits at state 2 where it passes through the regenerator and to the condenser at state 3. After being condensed in the water-cooled condenser, the fluid is pumped up to high pressure with the feed pump. For the present test the feed pump is electrically driven rather than being gearbox-driven. The flow is then metered through an ASME sharp-edged orifice. The fluid then flows through the regenerator where it picks up heat from the turbine exhaust vapor before entering the preheater. In the preheater/boiler the fluid is heated with hot, pressurized water. This water loop is heated by steam injection. After exiting the boiler, state 8, the entrained liquid is separated and the vapor passes to the turbine. The liquid separated out is used as gearbox lubricant (it is 50% oil and 50% R-113). The drain of the gearbox is connected to the condenser where this liquid is recycled.

The gearbox is loaded with a hydraulic system that is coupled to the gearbox through a torquemeter. This hydraulic system consists of a fixed displacement pump (sometimes acting as a motor) that drives a variable displacement motor (sometimes a pump). The variable displacement motor drives an induction motor being used as a generator. Through the displacement control for the motor, the relative speed of the hydraulic motor-pump system can be adjusted. This hydraulic system can be used as the driver to drive the gearbox for gearbox loss measurements as well as being driven as a load for the turbine.

Primary test measurements included pressure and temperature at turbine inlet, pressure at nozzle exit, pressure at turbine exit, speed, torque, and R-113 flowrate. In addition, several

Other pressures, temperatures, flow rates, current flows, etc., were measured for control purposes and to insure steady operation. A listing of the test instrumentation used appears in Appendix A. All instrumentation was calibrated prior to testing.

4.1 TEST PROCEDURE

The first portion of testing involved measuring gearbox losses. This required operating the entire system as the gearbox lubrication flow was furnished by the engine. The turbine wheel was removed for this test to eliminate that windage loss. The gearbox-mounted feed pump was removed for the entire test sequence in order to eliminate the necessity for correcting the turbine power developed for pump power consumed. The pump seal was also removed to eliminate those losses. The gearbox was driven in the direction of normal rotation in order to eliminate the additional losses associated with introducing the lubricant spray on the entering side of the gear mesh. The gearbox losses were low enough not to cause any significant problems with thrust reversal due to changing the roles of driver and driven.

The turbine testing was carried out over several test days. The turbine inlet condition and exhaust pressure were established and steadied out. Then a series of test points was obtained at various turbine speeds. This resulted in data series at essentially constant pressure ratio across the turbine with variable velocity ratio (U/C_o). Each data point was allowed to stabilize but, since the thermal conditions were constant, the stabilizing period was rather short.

The range of data was limited by equipment; namely, the hydraulic system capacity and the pressure relief settings. The hydraulic system capacity was limited by the displacement of the hydraulic pump and motor and by the power absorption capacity of the induction generator. These limitations in the hydraulic system limited the power output levels of the turbine and the operating speeds. The pressure relief settings on the engine limited the upper temperature limits.

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SUBJECT
TEST SCHEMATIC
Figure 4.1

The raw data was corrected for pressure gage line heights, gage calibrations, barometric readings, etc. The corrected data was then used to determine power output, fluid thermodynamic state properties, speed, torque, torque coefficient, reaction, etc. All computations were conducted on a HP 9825. The test results are reported in the next section.

5.0 RESULTS

The results of these tests are presented graphically in Figures 5.1 through 5.7. They consist of the gearbox losses, turbine efficiency, turbine torque coefficient, and turbine reaction. These parameters have been correlated in two ways. What follows is a brief description of the graphical results. These results will be discussed in more detail in Section 6.0.

5.1 GEARBOX LOSS RESULTS

The first figure given in this section, Figure 5.1, presents the results of the gearbox loss test. The gearbox was driven in the same direction of rotation as it was later powered by the turbine and the input power to the gearbox was measured. The gearbox power loss was then plotted as a function of output shaft speed. These tests were performed without the turbine wheel being present on the high speed shaft of the gearbox. Later, when the turbine performance was being calculated from data, these gearbox losses were added to the measured gearbox output power to give the turbine output power.

5.2 TURBINE PERFORMANCE RESULTS

The turbine performance parameters are given in the form of efficiency, η , and torque coefficient, C_t , as a function of isentropic velocity ratio, U/C_o . The torque coefficient is given by the following equation:

$$\begin{aligned} C_t &= \tau K / r_m D \sqrt{\Delta H_s} \\ t &= \text{Torque (lb}_f\text{-in)} \end{aligned}$$

FIGURE 5.1
B-N 25 Ton (Low Temp) Gearbox Loss Vs RPM
Normal Direction of Rotation, CW looking At Output Shaft
40% oil, 60% Freon 113 Mixture

FIGURE 5.2
TORQUE COEFFICIENT AND
TOTAL-TO-STATIC EFFICIENCY VS.
ISENTROPIC VELOCITY RATIO
TURBINE PRESSURE RATIO = 2.7

TURBINE TYPE: RADIAL INFLOW
TURBINE OUTSIDE DIA.: 4.33 in.
TURBINE EYE DIA. = 3.66 in.
DESIGN SHAFT SPEED = 21,140 RPM
DESIGN PRESSURE RATIO = 3.5
WORKING FLUID: R113

U = WHEEL TIP SPEED
 C_0 = ISENTROPIC EXPANSION VELOCITY
 C_t = TORQUE COEFFICIENT (NON-DIMENSIONALIZED TORQUE)
 η_{ts} = TOTAL-TO-STATIC EFFICIENCY

FIGURE 5.3
TORQUE COEFFICIENT AND
TOTAL-TO-STATIC EFFICIENCY VS.
ISENTROPIC VELOCITY RATIO

TURBINE PRESSURE RATIO = 3.6

TURBINE TYPE: RADIAL INFLOW
TURBINE OUTSIDE DIA. = 4.33 in.
TURBINE EYE DIA. = 3.66 in.
DESIGN SHAFT SPEED = 21,140 RPM
DESIGN PRESSURE RATIO = 3.5
WORKING FLUID: R113

U = WHEEL TIP SPEED
 C_0 = ISENTROPIC EXPANSION VELOCITY
 C_T = TORQUE COEFFICIENT (NON-DIMENSIONALIZED TORQUE)
 η_{TS} = TOTAL-TO-STATIC EFFICIENCY

FIGURE 5.4
TORQUE COEFFICIENT AND
TOTAL-TO-STATIC EFFICIENCY VS
ISENTROPIC VELOCITY RATIO

TURBINE PRESSURE RATIO = 4.1

TURBINE TYPE: RADIAL INFLOW
TURBINE OUTSIDE DIA. = 4.33 in.
TURBINE EYE DIA. = 3.66 in.
DESIGN SHAFT SPEED = 21,140 RPM
DESIGN PRESSURE RATIO = 3.5
WORKING FLUID: R113

U = WHEEL TIP SPEED
 C_o = ISENTROPIC EXPANSION VELOCITY
 C_T = TORQUE COEFFICIENT (NON-DIMENSIONALIZED TORQUE)
 η_{TS} = TOTAL-TO-STATIC EFFICIENCY

FIGURE 5.5

DESIGN INLET PRESS. = $P_0 = 38.5$ psia

DESIGN INLET TEMP = $T_0 = 17.6^\circ\text{F}$

WORKING FLUID : R113

η_{TS} = TOTAL -TO-STATIC EFFICIENCY

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93	94	95	96	97	98	99	100
---	---	---	---	---	---	---	---	---	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	-----

$$\frac{HP}{P_0 \sqrt{V_0}} \times 10$$
$$\frac{HP}{P_0 \sqrt{T_0}} \times 10$$

TUE

FIGURE 5.6
NOZZLE PRESSURE RATIO VS.
CORRECTED TURBINE SHAFT SPEED WITH
LINES OF CONSTANT TURBINE PRESSURE RATIO

TURBINE TYPE : RADIAL INFLOW

TURB. OD. = 4.33 in.

TURB. EYE DIA. = 3.66 in.

DESIGN SHAFT SPEED = 21,140 RPM

DESIGN PRESSURE RATIO = 3.5

DESIGN INLET TEMP. = $T_0 = 176^\circ\text{F}$

DESIGN INLET PRESS. = $P_0 = 38.5 \text{ psia}$

WORKING FLUID : R113

η_{TS} = TOTAL-TO-STATIC EFFICIENCY

NOZZLE
PRESSURE
RATIO 2.2

(Inlet Press.)
(Exit Press.)

3.0

2.8

2.6

2.4

2.2

2.0

1.8

1.6

1.4

1.2

1.0

0

.1

.2

.3

.4

.5

.6

.7

.8

.9

1.0

1.1

1.2

1.3

1.4

1.5

1.6

$\frac{N/\sqrt{T_0}}{(N/\sqrt{T_0})_{DES}}$

TURB P/R = 4.1

TURB P/R = 3.6

TURB P/R = 3.2

TURB P/R = 2.7

η_{TS}

FIGURE 5.7
TURBINE REACTION VS.
CORRECTED TURBINE SHAFT SPEED WITH
LINES OF CONSTANT TURBINE PRESSURE RATIO

TURBINE TYPE: RADIAL INFLOW
TURB. O.D. = 4.33 in.
TURB. EYE DIA. = 3.66 in.
DESIGN SHAFT SPEED = 21,140 RPM
DESIGN PRESSURE RATIO = 3.5
DESIGN INLET TEMP = $T_0 = 176^\circ\text{F}$
WORKING FLUID: RI13

REACTION = $\Delta H_s - \text{ROTOR} / \Delta H_s - \text{TURBINE OVERALL}$
 ΔH_s = ISENTROPIC ENTHALPY DIFFERENCE

$\dot{m}$	=	Mass Flowrate (lbm/sec)
D	=	Turbine Diameter (in.)
ΔH_s	=	Isentropic Enthalpy Drop (Btu/lbm)
K	=	Constant for Nondimensionality:

The isentropic velocity ratio is given by the following equation:

$$\eta_{is} = U/C_o = (\pi DN/720)/\sqrt{2gJ\Delta H_s}$$

$$N = \text{Shaft Speed (rpm)}$$

$$J = 778.17 \text{ (ft-lb}_f\text{/Btu)}$$

$$g = 32.17 \text{ (lbm-ft/lb}_f\text{-sec}^2\text{)}$$

The turbine efficiency given in this report is the total-to-static efficiency. This is defined as the actual enthalpy drop divided by the isentropic enthalpy drop across the turbine, based on inlet total and exhaust static conditions.

Three different pressure ratios (2.7, 3.6, and 4.1) were maintained across the turbine while the shaft speed was varied, producing three different efficiency versus velocity ratio curves. These are given in Figures 5.2, 5.3, and 5.4. Also shown on the same plots are torque coefficients.

Another way to present the data is to present turbine efficiency (total-to-static) and corrected output power versus corrected shaft speed. The corrected output power and corrected shaft speed are given by the following equations:

$$HP/P_o\sqrt{T_o} = \text{Corrected Output Power}$$

$$N/\sqrt{T_o} = \text{Corrected Shaft Speed}$$

$$P_o = \text{Turbine Inlet Total Pressure}$$

$$T_o = \text{Turbine Inlet Total Temperature}$$

Another useful parameter related to the above is the corrected mass flow:

$$\dot{m}\sqrt{T_o}/P_o = \text{Corrected Mass Flow}$$

In these tests the corrected maximum flow was essentially constant, since the nozzles were choked for nearly all data points. The variation of the corrected mass flow amounted to $\pm 1.2\%$ for nearly all data points, which indicates the error based on measuring the mass flowrate. The nozzle flow coefficient (the ratio of actual to ideal flow through the nozzles) also reflects the accuracy of the mass flow measurement. For this test, the flow coefficient was between .98 and 1.02 for almost all of the data presented. The value should be slightly below 1.00, but it is not unusual to see values of flow coefficient at or slightly above 1.0 due to small inaccuracies in the measurement and also due to leakage of the flow around the nozzles.

A plot showing turbine efficiency and corrected output power versus corrected shaft speed and total-to-static pressure ratio is given in Figure 5.5.

The final data presented in this report concerns the turbine reaction and the nozzle pressure ratio. The turbine reaction is defined as that fraction of the isentropic enthalpy drop across the nozzle and rotor that occurs across the rotor. Since this is intimately related to the rotor inlet pressure (identical to nozzle exit pressure), the nozzle pressure ratio (inlet pressure divided by nozzle exit pressure) conveys basically the same information. Figure 5.6 shows nozzle pressure ratio versus corrected turbine speed with lines of constant turbine pressure ratio. Figure 5.7 gives turbine reaction versus corrected speed. In addition to the previously described pressure ratios, these two plots show data for a turbine pressure ratio of 3.2 which yielded good pressure information but could not be used elsewhere in this report due to a lack of data during those tests.

6.0 DISCUSSION

The following sections present discussions of the results of the turbine testing, including the gearbox loss test and the turbine performance tests.

6.1 DISCUSSION OF GEARBOX LOSS TEST

Figure 5.1 shows the data from these tests along with the least squares curve fit of the data. The value of $R1$ (the confidence factor of the curve fit) is .9913. A value of $R2$ equal to 1.0 would mean a perfect fit. It is interesting to note that this curve fit has an exponent equal to 1.79. This is the type of exponent one would expect for bearing loss as a function of speed for ball bearings, which are used in the Barber-Nichols gearbox. The value of the gearbox loss at the gearbox out shaft design speed of 1200 rpm is .42 hp, which agrees well with Barber-Nichols' predictions. This is an extremely low value for a loss from a gearbox of this size (100 hp potential) but this low loss is due to the lack of a high speed seal as described in Section 3.0.

The accuracy of the gearbox loss test is important for the turbine performance test, since the turbine output power is not directly measured. The gearbox output power is measured and the turbine output power must be calculated as the difference between the measured value and the gearbox loss.

6.2 RESULTS OF TURBINE PERFORMANCE TESTS

Figures 5.2 through 5.4 present the traditional method of presenting turbine efficiency, i.e., efficiency versus isentropic velocity ratio. Isentropic velocity defined in Section 4 is known as a velocity diagram parameter. The velocity diagrams (triangles) or the way the gas velocity vectors align with the rotating blade section, determine the turbine efficiency. Therefore, the velocity ratio, which is a measure of the velocity diagram, is a perfect method of correlating the turbine efficiency. At one certain velocity diagram, the turbine efficiency will be at its maximum. This will occur at one certain value of the velocity ratio. As the conditions in the turbine change, either the shaft speed or the pressure ratio, the velocity diagrams will change. This will have an impact on efficiency. The figures mentioned above are typical in that the shape of the efficiency versus velocity ratio curve is basically a parabola, with the peak value being

dependent upon turbine geometry and the thermodynamic conditions across the machine.

The data presented in Figures 5.2 through 5.5 show slight scattering, primarily for the lowest pressure ratio tested. This data scatter is isolated to only a few points. The quality of the torque coefficient data indicates the efficiency measurement are good. Our confidence in this data is quite high, especially since the gearbox loss data was of such high quality. Data concerning the turbine reaction shows more data scatter than the efficiency data, but it still is quite acceptable. The turbine reaction data will be discussed later on in this section.

Figures 5.2 through 5.4 show that the peak value of efficiency decreases with increasing turbine pressure ratio. The lowest pressure ratio, 2.7, gives a peak efficiency of .86 while the highest pressure ratio, 4.1, yields .81 for a value of peak efficiency. The value of the velocity ratio at which the peak efficiency occurs rises as the pressure ratio rises, from $U/Co = .66$ at a pressure ratio of 2.7 to $U/Co = .69$ at a pressure ratio of 4.1.

These effects are understandable in that the peak efficiency should occur at lower turbine pressure ratios because the turbine utilizes converging-only nozzles. Higher pressure ratios would cause free expansions after the nozzles which may lead to shock waves and, hence, poorer rotor inlet conditions. The fact that peak efficiencies occurred at higher values of velocity ratio as the pressure ratio increased is also understandable. As the pressure ratio increases, the available energy to the turbine increases. The absolute value of energy available to the nozzle also increases (the reaction decreases as pressure ratio is increased and speed remains constant). This extra energy converted into velocity by the nozzles forces the peak efficiency point to occur at a higher value of U/Co because the velocity diagram has also changed.

Figure 5.5 shows the same information as Figures 5.2 through 5.4, but is correlated with the corrected speed parameter. Also

presented in this figure is the corrected turbine output power. This figure clearly shows the effect of pressure ratio. The higher the turbine pressure ratio, the lower the peak value of efficiency and this peak value occurs at higher corrected speeds. Output power increases with pressure ratio and its peak value occurs at higher values of corrected speed as the pressure ratio increases.

The turbine reaction data presented in Figures 5.6 and 5.7 shows the trend of a radial inflow turbine. Since the enthalpy drop through the rotor of a radial inflow turbine depends upon pressure ratio and the decreasing wheel speed as the flow passes through it, a strong effect of both wheel speed and pressure ratio is seen. As the speed is increased for any given turbine pressure ratio, the nozzle pressure ratio decreases and the reaction increases. These two phenomenon mean the same thing, i.e., a higher percentage of the energy drop is occurring through the rotor. As the turbine pressure ratio is increased for any given value of corrected speed, the nozzle pressure ratio increases and the reaction drops. As the turbine speed increases, the percentage of energy available to the rotor (or to energy drop in the rotor) must increase because of the strong influence of wheel speed for radial inflow turbines. If the shaft speed is held constant and the turbine pressure ratio increases, that part of the energy drop in the rotor due to wheel speed remains constant. This results in the increase in nozzle pressure ratio and the resulting decrease in turbine reaction.

The predicted efficiency of this turbine was .82 at a velocity ratio of .645 and a turbine pressure ratio of 3.5. A comparison with the data of Figure 5.3 shows that actual efficiency for those conditions is slightly higher than .84. The peak efficiency at a pressure ratio of 2.7 is .86, which is well above what Barber-Nichols predicted. The trend towards increasing peak efficiency for decreasing turbine pressure ratio shows the effect of lower nozzle pressure ratios. Since the nozzles were converging-only, and these nozzles were operating at super critical ratios (pressure ratios higher than the sonic throat condition) over most of the

test range, their performance will decrease as the pressure ratio rises past critical. The lower nozzle pressure ratios present for the turbine pressure ratio of 2.7 indicate the nozzles were operating closer to the critical pressure ratio and subsequently had a lower performance penalty. The turbine efficiency as a whole increased due to the better operation of the nozzles.

7.0 CONCLUSIONS

The following is a list of conclusions that can be drawn from the data.

1. The quality of the data is generally quite good and shows little data scatter.
2. The original performance estimation of this turbine was conservative as the actual turbine efficiency was more than two percentage points higher than the predicted design point.
3. The data shows a strong effect of pressure ratio on turbine efficiency, which is due to the nozzles operating well past the critical pressure ratio at higher turbine pressure ratios.
4. The peak efficiency of 86% occurs at the lowest pressure ratio tested (2.7).

APPENDIX A

TEST INSTRUMENTATION

<u>Function</u>	<u>Device</u>	<u>Information</u>
R-113 Flow Rate Measurement	ASME Orifice	d=.657", D- $\frac{1}{2}$ D
Hot Water Flow Rate Measurement	ASME Orifice	d=1.3476", D- $\frac{1}{2}$ D
Torque Measurement	Torque Meter	Lebow Model 1404-2000 in-lb
Pressures:		
Turbine Inlet Pressure	0-100 psi Gauge	Helicoil
Turbine Exit Pressure	30-0-30 psi Gauge	Ashcroft
Feed Pump Exit Pressure	0-200 psi Gauge	Weksler
Nozzle Exit Pressure	30-0-60 psi Gauge	Weksler
Diffuser Pressure	0-30 in/Hg	Weksler
Hot Water Flow Diff.	0-200 in/H ₂ O	ITT Barton
R-113 Flow Differential Pressure Across Orifice	0-200 in/H ₂ O	ITT Barton
Temperatures:		
Turbine Inlet Temperature	Thermocouple	Type J
Turbine Exit Temperature	Thermocouple	Type J
Gearbox Lube Outlet Temp.	Thermocouple	Type J
Feed Pump Temperature	Thermocouple	Type J
Hot Water Inlet Temperatures	Thermocouple	Type J
Regenerator Outlet Temperature	Thermocouple	Type J