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**DEVELOPMENT OF HIGH-TEMPERATURE TURBINE SUBSYSTEM
TECHNOLOGY TO A "TECHNOLOGY READINESS STATUS"**

Phase II

Quarterly Report for the Period January—March 1982

April 1982

Work Performed Under Contract No. AC01-76ET10340

**General Electric Company
Schenectady, New York**

U. S. DEPARTMENT OF ENERGY

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DEVELOPMENT OF HIGH-TEMPERATURE TURBINE SUBSYSTEM TECHNOLOGY TO A "TECHNOLOGY READINESS STATUS" PHASE II

**Quarterly Report for the Period
January — March 1982**

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Prepared for

**UNITED STATES DEPARTMENT OF ENERGY
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Under Prime Contract No. DE-AC01-76ET 10340
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OBJECTIVE AND SCOPE OF WORK

The objective of the DOE-HTTT (High-Temperature Turbine Technology) Program is to bring to "Technology Readiness,"* over a six- to ten-year duration, a high-temperature gas turbine for use in a combined-cycle power plant, with coal-derived fuel at a firing temperature† of 2600 °F and with growth capability to 3000 °F. This program has been divided into three phases:

- Phase I — Program and System Definition
- Phase II — Technology Testing and Test Support Studies
- Phase III — Technology Readiness Verification Test Program

The three phases of the program encompass the design, development, manufacture, and test of a GE-TRV gas turbine system** at a DOE facility.

Phase I was completed, and Phase II commenced on August 1, 1977.

The objectives of Phase II are to: (1) perform component design and technology testing in critical areas; (2) perform system design and trade-off analyses in sufficient depth to support the component design and test tasks; and (3) update the Phase I combined-cycle plant studies to evaluate the commercial validity of a GE-TRV gas turbine system.

* "Technology readiness" is defined as "that stage of system, subsystem, or component development where all major problems associated with the performance specification goals have been solved, and where the solutions to these problems have been demonstrated successfully. At that stage, no major risks remain for an agency or contractor in scaling up the technology (if full-scale demonstration has not been performed) and in proceeding with mission (or commercial) development of the system, subsystem, or component."

† Firing temperature is defined by the General Electric Gas Turbine Division as the inlet temperature to the first-stage buckets after all of the first-stage nozzle cooling flows and seal leakages have mixed with the combustor discharge flows.

** GE-TRV (General Electric Technology Readiness Vehicle) approximates the GE MS-6000 gas turbine in size.

SUMMARY OF PROGRESS AND CONCLUSIONS

The work in Phase II of the HTTT Program began in August of 1977. The total work-scope was defined in eleven reporting categories. Program management, evaluation, and documentation were included in a separate reporting category. Table S-1 contains a capsule review of the program status at the present time. To date, eight reporting categories are complete, and work is continuing in the other three categories.

Reporting Category I – HOT GAS PATH DEVELOPMENT TESTS

All testing activities commensurate with the requirements and objectives of this reporting category have been completed. All testing in the Hot Gas Path Development Test Stand has been successfully concluded. All testing in the Rotor Water Delivery and Distribution Test Rig likewise has been successfully concluded, and a topical report summarizing the activity has been released. All testing of both the low-Btu coal gas sectoral combustor and the alternate fuel combustor has been completed. All nondestructive and destructive evaluations of tested components have been completed. Evaluation of the combustion test results is continuing.

The principal activity in this reporting category during the reporting period has been the preparation of results for inclusion into the outstanding topical report "Hot Gas Path Development Test, Part II" and into the Final Technical Report. The issuance of these reports will conclude the total efforts required in this reporting category.

Reporting Category II – TURBINE SIMULATOR TESTS

This task was completed with the publication of Topical Report FE-1806-92 in November 1980.

Reporting Category III – SHOCK TUNNEL TESTS

This task was completed in April 1981.

Reporting Category IV – LOW-TEMPERATURE LOW-Btu GAS CLEANUP SYSTEMS TESTS

This task was completed with the publication of Topical Report FE-1806-78 in February 1980.

Reporting Category V – AERODYNAMIC TESTS

This task was completed during the last quarter of 1979.

Reporting Category VI – MOTORIZED RIG TESTS

A single specimen was tested extensively during the reporting period. The tested specimen, with a 0.083-inch-diameter passage STEM-drilled in an IN718 tube, is prototypical of eight cooling passages in the Stage 1 bucket and all cooling passages in Stages 2 and 3. The performance of this specimen was better than the performances of all smooth specimens tested previously, both in terms of power at boiling transition and in terms of wetted area. These favorable results have been used to review the cooling flows and the need for turbulence promoters in TRV buckets.

Table S-1
PROGRAM STATUS

A. Work is completed in the following reporting categories:

Reporting Category	Purpose	Status
II	Test water-cooled composite nozzle under a close simulation of service conditions of an IGCC plant	Completed: topical report issued (FE-1806-92)
III	Develop turbine gas-side heat transfer relationships	Shock tunnel tests and analysis complete
	Improve analytical techniques with computer program for boundary layer analysis	Analysis with computer program complete
IV	Investigate a coal-derived low-Btu gas cleanup capability for use in integrated gasification combined-cycle simulation testing	Completed: topical report issued (FE-1806-78)
V	Obtain data to verify turbine nozzle aerodynamic design assumptions	Completed
VII	Determine the effect of bucket passing frequency on heat transfer to downstream nozzle	Completed
VIII	Improve the estimate of advanced combined-cycle power plant performance with coal-derived fuels by updating the Phase I plant designs	Completed: two topical reports issued (FE-1806-83 & 84)
IX	Reference Turbine Subsystem Design	Completed: topical report issued (FE-1806-95)
X	TRV Controls System Study	Completed

B. Work is continuing in the following reporting categories:

Reporting Category	Purpose	Status
I	Prove technology readiness by test of TRV turbine nozzles	Complete: HGPD Facility Description (Part I), topical report (DOE/ET/10340-122), has been released: a draft HGPD Tests (Part II) topical report is being written
	Prove design reliability by cyclic testing of 1st and 2nd stage nozzles in HGPDTs	Completed
	Engineering evaluation testing; fault mode tests	Completed
	Prove technology readiness by test of a sectoral	Completed
	Development of an Alternate Fuels Combustor	Completed
	Test of Alternate Fuels Combustor	Test program complete. Data evaluation in progress
	Construction and evaluation testing of Rotor Water Delivery and Distribution System	Completed; Topical report has been released (DOE/ET/10340-120)
VI	Develop turbine bucket water cooling technology by model tests	Concept demonstration complete
	Determine sensitivity of bucket water cooling to pressure, tilt and length	Testing complete: analysis complete
	Demonstrate crimp manufacture and study erosion, corrosion, and deposition	Crimp manufacture demonstrated. Erosion, corrosion, and deposition testing curtailed by facility problems
	Optimize crimp location	Crimp location and spacing selected
XI	Develop TRV components for the Category I tests; develop key materials and processes for other components; advance the TRV design to support component development	Continuing

Reporting Category VII – AIR TURBINE TESTS

This task was completed during the last quarter of 1979.

Reporting Category VIII – UPDATE OF GAS-FUELED AND LIQUID-FUELED COMBINED-CYCLE POWER PLANT DESIGNS

The work required by this task was completed with the publication of Topical Reports FE-1806-83 and FE-1806-84 during the third quarter of 1980.

Reporting Category IX – UPDATE OF REFERENCE TURBINE SUBSYSTEM DESIGN

This task was completed with the publication of Topical Report FE-1806-95 during the second quarter of 1981.

Reporting Category X – TRV CONTROLS SYSTEM STUDY

This task was completed in the second quarter of 1980.

Reporting Category XI – TRV DESIGN AND COMPONENT DEVELOPMENT

The configuration status of the TRV, the flange-to-flange water-cooled gas turbine (Figure S-1) remains unchanged from that which was reported in the last quarterly report.

The blading status (Table S-2) remains unchanged from that of the previous report. Figures S-2 through S-4 provide details of the nozzle and bucket designs. The low-Btu gas combustion system is described on Table S-3. The alternate fuel combustion system description is provided on Table S-4.

Table S-2
TRV BLADING STATUS

	First-Stage Nozzle (Figure S-2)	First-Stage Bucket (Figure S-2)	Second-Stage Nozzle (Figure S-3)	Second-Stage Bucket (Figure S-3)	Third-Stage Nozzle (Figure S-4)	Third-Stage Bucket (Figure S-4)
Design Concept	Composite	Monolithic	Monolithic	Monolithic	Monolithic	Monolithic
Number per Machine	48	80	40	84	40	84
Airfoil Height, in.	3.8	4.1	7.2	7.48	11.87	12.3
Number Cooling Passages	28	28	47	20	16	9
Water Flow, pps	25.0	5.75	25.0	5.83	50.0	3.15
Water Inlet Temp., °F	225	100	225	100	400	100
Water Outlet Temp., °F	459	320	397	272	470	228
Materials						
Airfoil Spar	Nitronic-50	IN718	IN718	IN718	IN718	IN718
Airfoil Copper	Forged MZC	Forged	Cast	Forged	Cast	Forged
Airfoil Skin	IN617					
Airfoil Tubes	Nitronic-50

DECEMBER 1981

**Figure S-1. Technology Readiness Vehicle (TRV) configuration status
(with alternate Fuel Combustor)**

ISO Day Operating Conditions

$T_{\text{gas max}} = 3010^{\circ}\text{F}$
 $M_o = 0.155$
 $P_{T_{\text{gas}}} = 166.2 \text{ psia}$
 $W_{\text{throat}} = 305.9 \text{ pps}$

Dimensions

Airfoil height = 3.8 in
 Axial chord = 3.47 in
 Pitch diam = 65 in
 TE thickness = 0.15 in

COOLING TUBES

	FEED	RETURN
NUMBER	12	16
OD - in	0.094	0.125
ID - in	0.080	0.095
$T_{\text{skin max}}^{\circ}\text{F}$	944	840
$T_{\text{cu max}}^{\circ}\text{F}$	705	691
$T_{\text{tube OD}}^{\circ}\text{F}$	639	657
$T_{\text{tube ID}}^{\circ}\text{F}$	534	574

Per Nozzle Coolant Flows @ Full Load

a) Nozzle

ISO Day Operating Conditions

$T_{\text{gas max}} = 2530^{\circ}\text{F}$ (Relative)
 $M_{R_1} = 0.414$
 $P_{T_1} = 105.6 \text{ psia}$

Dimensions

Airfoil height = 4.1 in
 Axial chord = 3.5 in
 Pitch diam = 65.1 in
 TE thickness = 0.15 in

3 leading edge passages (0.062-in. dia)
 3 trailing edge passages (0.062-in. dia)
 2 passages of 0.141-in dia in trailing edge
 20 passages of 0.094-in dia elsewhere
 28 passages total
 0.05-in (passage edge-to-surface)

b) Bucket

Figure S-2. First-stage blading design summary

ISO Day Operating Conditions

$T_{\text{gas max}} = 2460^{\circ}\text{F}$
 $M_o = 0.25$
 $P_{T\text{gas}} = 79 \text{ psia}$
 $W_{\text{throat}} = 323.9 \text{ pps}$

Dimensions

Airfoil height = 7.2 in
 Axial chord = 8.58 in
 Pitch diam = 66.3 in
 TE thickness = 0.15 in

COOLING PASSAGES

	FEED	RETURN
NUMBER	25	22
Dia (in)	0.07 0.06 (TE)	0.07
$T_{\text{hole surface}}^{\circ}\text{F}$	340	450

Per Nozzle Coolant Flow @ Full Load

a) Nozzle

ISO Day Operating Conditions

$T_{\text{gas max}} = 2100^{\circ}\text{F (Relative)}$
 $M_{R1} = 0.326$
 $P_{T1} = 50.9 \text{ psia}$

Dimensions

Airfoil height = 7.5 in
 Axial chord = 3.305 in
 Pitch diam = 66.4
 TE thickness = 0.15 in

Passages 1-8 and 16-18 are 0.062 in. dia
 Passages 9-15 and 19-20 are .093 in. dia
 20 - STEM-drilled passages total

b) Bucket

Figure S-3. Second-stage blading design summary

ISO Day Operating Conditions

$T_{\text{gas max}} = 1880^{\circ}\text{F}$
 $M_o = 0.271$
 $P_{T_{\text{gas}}} = 35.2 \text{ psia}$
 $W_{\text{throat}} = 330.9 \text{ pps}$

Dimensions

Airfoil height = 11.87 in
 Axial chord = 9.167 in
 Pitch diam = 67.78
 TE thickness = 0.15 in

a) Nozzle

COOLING PASSAGES

NUMBER	FEED	RETURN
OD (in)	0.3125 (4 places)	0.3125
	0.20 (1 place)	
	0.09 (2 places)	
	0.07 (1 place)	
	0.05 (1 place)	
$T_{\text{Hole surface } ^{\circ}\text{F}}$	480	525

Per Nozzle Water Coolant Flow @ Full Load

100 Day Operating Conditions

$T_{\text{gas max}} = 1615^{\circ}\text{F}$ (Relative)
 $M_{R1} = 0.309$
 $P_{T1} = 23.4 \text{ psia}$

Dimensions

Airfoil height = 12.15 in
 Axial chord = 3.5 in
 Pitch diam = 67.8 in
 TE thickness = 0.15 in

b) Bucket

PASSAGES DIAMETER (in.)

1	0.063
2	0.070
3	0.105
4	0.127
5	0.127
6	0.127
7	0.127
8	0.127
9	0.063

Figure S-4. Third-stage blading design summary

Table S-3
SECTORAL COMBUSTOR DESIGN DESCRIPTION SUMMARY

• Basic Configuration	Low-Btu gas, two-stage, cooled shell and panel, 16 combustors per machine.	
• Fuel LHV	Design will accommodate LHV range of 127.5 to 150.0 Btu/scf without changes.	
• Fuel Flow	94.59 pps Low-Btu (127.5 Btu/scf) gas at ISO day FSFL	
• Fuel Inlet Temp.	420 °F (Low-Btu Gas)	
• Liner Airflow Splits	Cooling Passages	48.2%
	Second Stage	24.2%
	Pilot Zone	21.0%
	Secondary Dilution	4.6%
	Crossfire	2.0%
		<u>100.0%</u>
• Operating Characteristics	Start on No. 2 petroleum distillate. Transfer to low-Btu gas in pilot zone initiated at 8% and completed by 18% load. Stage in the secondary nozzles at 44% load.	
• Corrosion/Erosion	Designed for clean low-Btu fuel — no special provision needed for hot corrosion, particulate erosion, or plugging.	
• Emissions (Based on Illinois No. 6)	CO < 200 ppm	
	SO _x within EPA standards	
	NO _x within EPA standards.	
	(except during start to 20% load on No. 2 petroleum distillate)	

Table S-4

ALTERNATE FUEL COMBUSTOR DESIGN DESCRIPTION SUMMARY

• Basic Configuration	Single-stage, single nozzle, shell and panel downstream end, slot-cooled reaction zone, 16 combustors per machine	
• Fuel	Light distillate; FBN < 0.05%; 9.8 pps at ISO day FSFL	
• Fuel Nozzle	Air atomized; standard	
• Emissions	NO _x control by water injection	
• Liner Airflow Splits	Cooling air	25.0%
	Dilution zone air	15.0
	Reaction zone air	60.0
		<hr/> 100.0%

Review of First-Stage Bucket Design. The first-stage bucket design for the TRV has been reviewed during this reporting period in view of the recent test results on the heat transfer performance of a prototypical bucket cooling passage in the Pressurized Test Facility. As a direct consequence of these test data, the first-stage bucket cooling design has been simplified to delete all requirements for "reamed" cooling passages and for "turbulence promoters" in any of the cooling passages. To complete the changes, and to reduce to a minimum the cooling water requirement for this stage, it has been decided that the resistance flow dividers for this stage, when designed for the TRV application, will utilize three flow levels to service the 28 cooling passages of the bucket.

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Reporting Category I

HOT GAS PATH DEVELOPMENT TESTS

The objectives of the Hot Gas Path Development Tests (HGPDT) were to evaluate the performance of the Technology Readiness Vehicle (TRV) hot gas path components at simulated service conditions and to demonstrate key aspects of providing water cooling to the TRV rotating components.

All required activities commensurate with these objectives have been completed. All testing in the Hot Gas Path Development Test Stand has been successfully concluded. All testing in the Rotor Water Delivery and Distribution Test Rig likewise has been successfully concluded, and a topical report summarizing the activity has been released. All testing of both the low-Btu coal gas sectoral combustor and the alternate fuel combustor has been completed. All nondestructive and destructive evaluations of tested components have been completed. Evaluation of the combustion test results is continuing.

The principal activity in this reporting category during the reporting period has been the preparation of results for inclusion into the outstanding Hot Gas Path Topical Report (Part II) and into the Final Technical Report. The issuance of these reports will conclude the total efforts required in this reporting category.

Reporting Category II

TURBINE SIMULATOR TESTS

The objective of the Turbine Simulator Tests was to evaluate the erosion, ash deposition, heat transfer, and corrosion characteristics of gas turbine first-stage water-cooled nozzles. These test characteristics were evaluated in an environment representative of a high-temperature gas turbine that is burning coal-derived, low-Btu gas fuel. Combustion characteristics were also evaluated. This task was completed with the publication of Topical Report FE-1806-92 in November 1980.

Reporting Category III

SHOCK TUNNEL TESTS

The objective of the shock tube and shock tunnel tests was to obtain hot gas-side local heat transfer coefficients for the TRV turbine airfoils. The shock tube tests, which used a flat plate as the test model, were designed to resolve the question of the appropriate temperature for the selection of fluid properties. The shock tunnel tests used both a flat plate and TRV half-scale first-stage nozzles to experimentally determine local heat transfer coefficients.

The results of the shock tube tests have been reported in Quarterly Report FE-1806-100 for the October-December 1980 period. The results of the shock tunnel tests and boundary layer analysis have been reported in Quarterly Report FE-1806-107 for the January-March 1981 period. The Phase II Final Report will present the overall results and summary.

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Reporting Category IV

LOW-TEMPERATURE, LOW-Btu GAS CLEANUP SYSTEM TESTS

The objective of this task was to demonstrate the low-temperature, low-Btu gas cleanup capability assumed for the combined cycle plant design in Phase I. This task was completed with the publication of Topical Report FE-1806-78 in January 1980. The task was completed in the last quarter of 1979, and the results were presented in Quarterly Report FE-1806-76.

Reporting Category V

AERODYNAMIC TESTS

The objective of this task was to study the airflow along the contoured endwalls (step-ups) of the TRV turbine nozzles. Cold air cascade tests were conducted in a transonic wind tunnel with several nozzle endwall configurations. Calculations were performed to determine the effect of the endwall geometry on cascade aerodynamic losses.

The task was completed in the last quarter of 1979, and the results were presented in Quarterly Report FE-1806-76. The Phase II Final Report will present the overall results and summary.

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Reporting Category VI

MOTORIZED RIG TESTS

The objective of this task is to acquire heat transfer data needed to design the cooling passages within the rotating components of the TRV gas turbine. This task includes the theoretical analyses needed to prove the applicability of results.

A single specimen was tested extensively during the reporting period. The tested specimen, with a 0.083-inch-diameter passage STEM-drilled in an IN718 tube, is prototypical of eight cooling passages in the Stage 1 bucket and all cooling passages in Stages 2 and 3. The performance of this specimen was better than the performances of all smooth specimens tested previously, both in terms of power at boiling transition and in terms of wetted area. These favorable results have been used to review the cooling flows and the need for turbulence promoters in TRV buckets.

6.1 SPECIMEN DESCRIPTION

A 0.083-inch-diameter passage STEM-drilled in an IN718 tube is prototypical of eight cooling passages in the Stage 1 bucket and all cooling passages in Stages 2 and 3. Specimen 19 with this STEM-drilled passage was modified to incorporate a platform cooling passage upstream of the radial airfoil cooling passage. A stainless steel extension which incorporated the platform cooling passage was brazed to the inlet of the specimen. The final length of the radial cooling passage was 3.5 in. This modified specimen, which has been designated Specimen Number 819, is shown on Figure 6-1a.

Specimen 819 was installed in the pressurized test facility (PTF) such that the platform cooling passage was in the plane of rotation. With "forward" rotation of the PTF disc, the specimen simulated a suction-side cooling passage. With "reverse" rotation of the PTF disc, the specimen simulated a pressure-side passage. Additional tests were performed with the specimen rotated 90°, such that the platform cooling passage was parallel with the axis of rotation.

To further determine the effects of the platform cooling passage on performance, Specimen 819 was later modified by drilling through the stainless steel extension to bypass the portions of the passage that simulated a platform cooling passage. The modified specimen, designated Number 829, is shown on Figure 6-1b. To prevent any flow from detouring through the platform cooling passage, a new nipple was constructed and a 0.060-inch-diameter tube was brazed into the nipple. This tube extended all the way through the stainless steel extension on the specimen, colinear with the cooling passage. Cooling water was discharged from this tube at the entrance to the 0.083-inch-diameter STEM-drilled IN718 cooling passage. The actual cooling passage was only 3.5 in. long, the same as that in Specimen 819.

Figure 6-1(a). Specimen 819 inlet arrangement

Figure 6-1(b). Specimen 829 inlet arrangement

6.2 REPEATABILITY DEMONSTRATED

Tests of Specimen 819 were conducted for 13 days. At the beginning of each day, two reference points were repeated to establish confidence in the measurements. Fifty-five hours of testing time was accumulated and 105 sets of boiling transition data were measured.

Table 6-1 shows statistics for the reference points. For the four tests at atmospheric pressure conditions, the standard deviation of the boiling transition power levels is essentially equal to the minimum control increment in heater power. At 95 psia the standard deviation is somewhat higher, partly because of the wide range of inlet water temperatures and the effect that inlet water temperature has on the power at boiling transition.

Table 6-1
STATISTICS FOR REPETITIONS OF SPECIMEN 819 PERFORMANCE POINTS

Flow: 0.02 gpm Speed: 3500 rpm Radial Length of Cooling Passage: 3.5 in. Rig: Pressurized Test Facility									
Pressure, psia	15	15	15	15	95	95	95	95	
Inlet Vent	Closed	Closed	Open	Open	Closed	Closed	Open	Open	
Rotation	F	R	F	R	F	R	F	R	
No. of Repetitions	10	4	3	1	18	5	4	2	
Mean Power at B.T., watts	1902	1854	1890	1868	1673	1623	1957	2005	
Std. Deviation, watts	45	15	27	—	96	225	73	96	

Note: Experimental precision (minimum control increment in heater power) is 30 W.

When Specimen 19 was tested in the PTF in June 1981, the power at boiling transition was only 1400 W at 15 psia and 0.02 gpm, and 900 W at 95 psia and 0.02 gpm. The recent improvement in performance was achieved by the elimination of silicone grease contamination in the PTF and by thorough cleaning of Specimen 819 prior to testing. On Figure 6-2 there is an indication of slight degradation in wetted area during the period of testing. Specimen temperatures were higher late in the testing for the same heat input. This degradation could have been avoided by restoration of the effectiveness of the water conditioning system. However, the deposit patterns on the cooling passage were considered valuable for understanding flow patterns and verifying wetted area, so testing was intentionally performed with substandard water conditions. Below boiling transition power levels there was no indication of degradation in performance from deposits or contamination.

6.3 BOILING TRANSITION POWER

6.3.1 High Power at Boiling Transition

Figure 6-3 shows the power at boiling transition for Specimen 819 at 95 psia. At low flow rates the rotational direction has no significant influence on performance. The additional 0.6 in. in the TRV bucket radial passages is predicted to increase the power at boiling transition by 17% relative to the measured performance of Specimen 819.

Figure 6-2. Performance data for Specimen 819

Figure 6-3. Power at boiling transition as a function of water flow rate

6.3.2 Effect of Excess Flow

Although the performance of Specimen 819 is very repeatable at flows close to the TRV design flow, the results from tests with very high flows show more scatter. Shown on Figures 6-4 and 6-5 are data indicating an actual decrease in the power at which boiling transition occurs at very high water flow rates. The lines connecting data points indicate the sequence of tests and imply nearly constant inlet water temperature. The data show that opening the inlet vent to permit reverse flow of steam increases the performance at low flow rates but has little influence at high flow rates. This supports the hypothesis that steam flow is necessary for effective wetting of the cooling passage surface. At the high water flow rates the steam flow is quite low because most of the heat is absorbed in raising the water to the saturation temperature. Without the steam flow to spread the water and keep the passage fully wet, the wetted area can drop rapidly at the high heat flux conditions. Thus, it is postulated that the scatter in the data for high flow rate tests indicates the dependence of wetted area on steam flow.

6.3.3 Effect of Reverse Steam Flow

A vent port at the inlet to the specimen permitted a reverse flow of steam when the vent was open. This reverse flow did not influence performance at atmospheric pressure. However, at 95 psia it did result in an increase in power at boiling transition shown on Figure 6-6. Utilization of reverse steam flow in the design provides a means for further improvements in bucket cooling performance.

6.3.4 Effect of Subcooling

Data plotted on Figure 6-6 also show a slight reduction in power at boiling transition as inlet water temperature is raised corresponding to the reduced sensible heat.

6.3.5 Effect of Rotational Speed

Power at boiling transition is strongly influenced by rotational speed between 0 and 2000 rpm. (See Figure 6-7.) Above 2000 rpm, the data show a gradual increase in performance with increasing speed. Since the tests were performed at a smaller radius and smaller rotational speed than TRV conditions, this increase in performance with increasing centrifugal acceleration gives confidence in the application of these test data to the TRV.

The behavior of the specimen in the vicinity of 2000 rpm was particularly interesting. There was essentially no change in specimen temperature for a major (50%) increase in power from the 2500 rpm boiling transition condition. Thus, the added steam velocity must have caused an increase in wetted area proportional to the increase in power. It is postulated that the steam forces controlling the spreading of the water film are balanced by Coriolis forces tending to narrow the wetted area at speeds between 1500 and 2000 rpm in the PTF, which has an 11 in. specimen tip radius.

A similar behavior was observed in prior testing with the crimped specimen tilted at 10° in the Motorized Test Rig.

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Figure 6-4. Effect of excess coolant flow with inlet vent sealed

Figure 6-5. Effect of excess coolant flow with inlet vent opened

Figure 6-6. Effect of subcooling and reverse steam flow on power at boiling transition

Figure 6-7. Effect of rotational speed on power at boiling transition

6.4 WETTED AREA

The model used to analyze wetted area consists of a convection coefficient of 30,000 Btu/hr-ft²-F for a turbulent "river" of coolant water on the Coriolis side (10% of circumference), 12,000 Btu/hr-ft²-F where the wall is wetted by a thin film, and 200 Btu/hr-ft²-F where flowing steam is the only coolant in contact with the passage wall. Using this model, the average value for wetted area was 56% of the circumference for the 18 data points at 95 psia, 3500 rpm forward rotation with the inlet vent plugged to prevent reverse steam flow. The standard deviation was only 9%. At higher flow rates the scatter was somewhat greater, but the wetted area was also greater.

6.4.1 Effect of Flow Rate

There is a trend of increasing passage wetted area with increasing coolant flow rate. This is indicated by the data on Figure 6-8a and b which show cooler passage wall temperatures at higher flow rates. The 0.03 and 0.05 gpm test data on Figure 6-8b show abrupt drops in temperature for slight increases in power. It is postulated that increased steam velocity caused improved wetting at this point as the force balance changed.

6.4.2 Effect of Reverse Steam Flow

The data shown on Figure 6-9 indicate a benefit in both the power at boiling transition and the wetted area when the vent at the inlet to the specimen is opened to permit reverse flow of steam. At cool inlet temperatures there is a 10% increase in wetted area with the inlet vent open.

6.4.3 Effect of Pressure

To display the influence of pressure on the wetted area, data from Specimen 819 have been plotted against the difference between measured specimen temperatures and saturation temperature at vessel pressure on Figure 6-10. The temperature of the flow exiting from the specimen is depressed below saturation temperature as much as 30 °F at 3500 rpm. Comparisons using the depressed temperature as the sink temperature might be more appropriate but would not change the relationships shown on Figure 6-10.

Because the saturation temperature is 112 °F higher for 95 psia than for 15 psia, the water remains below saturation temperature for a greater fraction of the passage length. Thus it is not surprising to observe that at low power the pressurized test appears cooler. In fact, the sink temperature is below the saturation temperature, so the comparison is inappropriate. At high power the temperature differences appear to converge implying that the wetted area fractions are about the same at high power levels regardless of the pressure.

6.4.4 Effect of Inlet Temperature

Cooler inlet temperatures result in cooler specimen temperatures over the full range of powers. Figures 6-11 and 6-12 show temperature data at the pitch and tip sections of Specimen 819. In these tests, only the supply temperature of the cooling water was varied. Even though the heat removed in the first half of the cooling passage is sufficient to warm all the water to saturation temperature and evaporate a small fraction, the influence of inlet water temperature persists all the way to the tip of the specimen. If the water were all heated to saturation temperature before any evaporation occurred, it would be expected that the tip temperature would show very little influence of inlet water temperature.

(a) Reverse rotational speed

(b) Forward rotational speed

Figure 6-8. Effect of coolant flow rate on cooling passage wall temperature

Figure 6-9. Effect of inlet vent on power-temperature data

Figure 6-10. Effect of pressure on power-temperature data

Figure 6-11. Effect of inlet temperature on pitch temperature

Figure 6-12. Effect of inlet temperature on tip temperature

Previous paragraphs, which have indicated the importance of steam velocity on wetted area, support a thesis that the thermal conductance to the water at the tip should be poor for the lower steam velocity condition resulting from cool inlet temperature. A higher specimen temperature would thus be expected at the tip. Since the opposite is observed (Figure 6-12), it is concluded that subcooled liquid remains all the way to the tip of the passage. Evaporation must be taking place at the edges of the liquid film, and rivulets of subcooled boiling must be occurring along the full length of the passage.

6.4.5 Variation Along Passage Length

The difference in the shape of the pitch and tip temperature curves shown on Figures 6-11 and 6-12 show that boiling transition initiates at the tip and progresses toward the root as power is increased. Figure 6-13 compares root, pitch, and tip temperatures for one cold and one hot inlet temperature test series. In both cases the root temperature is much cooler than the pitch and tip because all the water is below saturation temperature. Steam velocity is low at the root, but borescopic inspection of deposits (resulting from the use of ordinary tap water in the experiment) indicates that flow patterns are fully developed within the first quarter inch of the passage and do not change along the specimen length despite the increasing flow rate of steam. From Figure 6-13 it is seen that the tip goes from cooler-than-pitch to warmer-than-pitch as boiling transition initiates at the tip.

6.4.6 Effect of Rotational Speed

The influence of rotational speed on the power at boiling transition was shown on Figure 6-7. Data from the same series of tests are plotted on Figure 6-14 to show the influence of speed on wetted area. The difference between 1500-2000 rpm data and other data is readily apparent. It is also apparent that the specimen is cooler at 3500 rpm and at 3000 rpm than at 2500 rpm. This is the case for both directions of rotation. In fact, the direction of rotation has essentially no influence on the specimen temperature at 3500 rpm.

Since the specimen temperature is hotter for a given power setting when operating at 3500 rpm than at 1500-2000 rpm, it can be concluded that wetted area fraction is lower at the high speed. Assuming that wetted area is 100% at 2000 rpm, it is deduced that the wetted area at 3500 rpm is not more than 83%. Data from the highest power at which stable data were obtained at 3500 rpm indicate 67% of the wetted area of the 2000 rpm data. This is an indication of how far the boiling transition zone has migrated upstream from the tip.

6.5 PLATFORM PASSAGE ORIENTATION

Figure 6-14 does not indicate any difference in wetted area between forward and reverse rotation. At the conclusion of tests of Specimen 819, the specimen was rotated 90°, such that the platform cooling passage was parallel with the axis of rotation instead of being in the plane of rotation. Test results in this arrangement do not show any difference from in-plane results. Figure 6-15 shows temperature data in the out-of-plane arrangement to be at the center of the cluster from in-plane tests. Boiling transition power levels duplicated the in-plane results. It is concluded that the orientation of the platform passage has no influence on performance of the radial cooling passage.

Figure 6-13. Variation in temperature along passage length

Figure 6-14. Effect of rotational speed and direction

Figure 6-15. Influence of platform passage orientation

6.6 STRAIGHT-THROUGH PERFORMANCE

The prototype STEM-drilled specimen, Number 819, was modified such that flow bypasses the portions of the passage that simulated a platform cooling passage. The performance in the straight-through configuration, designated Specimen Number 829, was essentially unchanged. It is concluded that the very good performance reported above is possible from all clean STEM-drilled holes and is not dependent on abrupt bends in the entrance to the passage.

6.6.1 Boiling Transition Power

The reference points used to demonstrate repeatability with Specimen 819 were duplicated with modified Specimen 829. Boiling transition power results are compared in Table 6-2. At atmospheric pressure the power at boiling transition was 13% higher than the average of the 10 points tested previously. At 95 psia the performance was essentially the same. From these data it is concluded that the good performance is not dependent on flow patterns set up by abrupt bends in the passage entrance. This is consistent with borescope inspections of Specimen 819, which indicated that flow patterns stabilize very quickly and remain unchanged after the first 1/4 in.

Table 6-2
BOILING TRANSITION POWER

Pressure	Specimen 819	Specimen 829
15 psia	1902	2151
95 psia	1673	1615

Conditions: 0.02 gpm, 3500 rpm (Forward), Vent closed
Experimental Precision: minimum increment in power is 30 W

The power at boiling transition for Specimen 829 is shown on Figure 6-16 as a function of cooling water flow. Whereas, for a crimped specimen, Number 811, and for Specimen 819 the power at boiling transition actually decreased at very high flow rates, there is a monotonic increase in cooling capacity with increasing water flow for Specimen 829. There is very little influence of inlet temperature on performance.

Rotational speed does influence the power at boiling transition in that it passes through a maximum at 2000 rpm. Above 2500 rpm there is very little change (see Figure 6-17).

6.6.2 Wetted Area

Analyses were based on the model which assigns a convection coefficient of 30,000 Btu/hr-ft²-F to a 10% region (turbulent river) at the Coriolis side of the passage, 12,000 Btu/hr-ft²-F to the remaining wetted region, and 200 Btu/hr-ft²-F to the remaining surface opposite the Coriolis side. The sink temperature was assumed to be 22 °F below saturation temperature in accord with water exit temperature measurements. Using the mid-span (pitch) thermocouple and the thermocouple 3/4 in. from the tip, the wetted fraction at boiling transition was found to decrease with increasing rotational speed (see Figure 6-18). This finding is consistent with the fact that Coriolis forces are increasing, drawing liquid away from the leading surface of the passage more rapidly. Furthermore, it is consistent with results for other smooth specimens published previously when plotted using a dimensionless rotation number parameter, (see Figure 6-19). Despite this indication that wetted area decreases with speed, Figure 6-17 shows slight increases in the power at boiling transition with speed. It is postulated that the Coriolis forces hold the liquid to the wall more effectively at high speed, thus suppressing critical heat flux. Even though the wetted surface is smaller at high speeds, the heat transfer remains stable at higher heat flux, permitting an increase in total power at higher speed.

A typical series of specimen temperature data is plotted on Figure 6-20. The test was conducted by raising heater power in increments and waiting (typically 6 min.) for stable temperatures to be achieved before recording data. Because the tip became cooler than the pitch section at high power, it is apparent that that tip was cooled more effectively as power was raised and steam velocity increased.

It is also apparent that boiling transition occurred first at the tip because the tip went from cooler-than-pitch to hotter-than-pitch at boiling transition. Although the highest power point was stable for several minutes, the tip was not being cooled as effectively as at the preceding power setting, indicating that boiling transition had occurred at the tip. This finding is consistent with observations from tests of Specimen 819.

Figure 6-16. Power at boiling transition for Specimen 829 at 95 psi

Figure 6-17. Effect of speed on boiling transition for Specimen 829

Figure 6-18. Wetted fraction variation with speed

Figure 6-21 shows how the wetted area changes with flow rate. No correction has been made for the changes in sink temperatures resulting from changes in flow. At the high flow rate the wetted area may be overestimated to the extent that subcooled liquid remains a significant fraction of the flow. It is presumed that 0.01 gpm flow results in steam velocity high enough to balance Coriolis forces, much as was observed when speed was varied.

Another indication of the effect of flow rate on wetted area is presented on Figure 6-22, which shows temperature data for tests at several flow rates. For a particular specimen temperature, the power is higher at each higher flow rate.

Figure 6-19. Wetted area as function of rotation number

Figure 6-20. Power-temperature data for Specimen 829

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Figure 6-21. Wetted area fraction for Specimen 829

Figure 6-22. Pitch temperature data for Specimen 829

6.7 POST BOILING TRANSITION PERFORMANCE

In order to study what happens after boiling transition occurs in a cooling passage, tests were conducted in which Specimen 829 temperatures were held constant at values significantly above the onset of boiling transition at the tip. Some of these data are shown on Figure 6-22. At 0.01 gpm the data indicate multiple stable temperatures for a given power setting near boiling transition. Specimen temperature was permitted to increase to a desired level; then the power was reduced gradually until the specimen temperature remained constant. It is postulated that the boiling transition zone expanded back from the tip by discrete amounts for each new set of test conditions. When power was reduced to stabilize the process, the water was not able to rewet the wall where boiling transition had occurred. This caused several stable specimen temperatures to exist for power settings at boiling transition.

The multistable behavior was observed at atmospheric pressure and design coolant flow rate, 0.02 gpm (see Figure 6-23). At 95 psia (Figure 6-24), the stable conditions were not multivalued. Each increase in temperature required a corresponding increase in power.

Because of the high-conductivity copper body of the tested specimen, this series of tests is not exactly representative of the process expected in a low-conductivity gas turbine bucket. With the low-conductivity bucket material there will be very little redistribution of heat along a cooling passage in the event that boiling transition occurs. This makes it less likely that boiling transition will propagate upstream in a bucket than in the specimen where the copper quickly transfers the heat upstream. In the test specimen, the heat load per unit length is necessarily constant along the cooling passage even after boiling transition commences at the tip. If boiling transition should occur at the tip of a bucket, the local region of the bucket would warm up, reducing the heat flux from the hot gas. This condition would arrest the boiling transition zone growth.

It is still desirable that sustained operation with any bucket cooling passage in boiling transition be avoided to limit potential corrosion and to minimize local stresses. Nevertheless, a momentary fluctuation in water flow or gas temperature should not damage the bucket, even if boiling transition were to be initiated at the tip.

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Figure 6-23. Power-temperature data for Specimen 829 at 0.02 gpm and 15 psi

Figure 6-24. Power-temperature data for Specimen 829 at 0.02 gpm and 95 psi

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Reporting Category VII

AIR TURBINE TESTS

The objective of the air turbine tests was to furnish the data necessary to understand the effects that bucket passing have on the heat transfer to the downstream nozzle.

The work was successfully completed in the last quarter of 1979 and reported in (Quarterly Report FE-1806-76). This work will be summarized in the Phase II Final Report.

Reporting Category VIII

UPDATE OF GAS FUELED AND LIQUID FUELED COMBINED-CYCLE POWER PLANT DESIGNS

The gas-fueled and liquid-fueled power plant designs established in Phase I were updated to:

- Confirm their capability in meeting the established performance and operational requirements
- Identify new approaches or modifications that will lower output energy cost by increasing system efficiency or lower first costs
- Identify new approaches or modifications that will enhance plant operating flexibility or reliability

This task was completed with the publication of Topical Reports FE-1806-83 and FE-1806-84 in July 1980.

An additional task, a Failure Modes and Effects Analysis for an Integrated Gasification Combined-Cycle power plant, was added to the workscope during 1980. The objective of this task was to assess qualitatively the reliability of an oxygen-blown entrained bed (Texaco process) coal gasification integrated combined-cycle (IGCC) power plant. Identification of critical components provided the basis for design modification to assure reliability.

This task was completed in the first quarter of 1981. The results and conclusions will be presented in the Final Report, now in preparation.

Reporting Category IX

UPDATE OF REFERENCE TURBINE SUBSYSTEM DESIGN

This task was completed with the release of Topical Report FE-1806-95.

Reporting Category X

TRV CONTROLS SYSTEM STUDY

The projected controls system has been studied in parallel with design developments of the TRV in order to identify and solve systems problems as they arose. All systems analysis work planned for Phase II is complete. The results of the study will be included in the Final Report.

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Reporting Category XI

TRV DESIGN AND COMPONENT DEVELOPMENT

The combustor and two stages of the Technology Readiness Vehicle (TRV) water-cooled nozzles have been developed for technology testing and evaluation in Phase II of the HTTT Program. Materials were evaluated and processes were developed to manufacture water-cooled turbine buckets. Bucket design was performed in order to support this process development effort. Water delivery to the turbine and water distribution to each bucket throughout the turbine was developed in this reporting category. The overall turbine system and other components of the gas turbine were designed as required in order to support the primary tasks.

11.1 REVIEW OF FIRST-STAGE BUCKET DESIGN

The first Stage bucket thermal analysis, which was completed one year ago, was reviewed again during this reporting period to assess the impact of intervening test results with both STEM and CRIMP passages. Major findings affecting bucket design are summarized below.

- Elevated ambient pressure reduces the power at boiling transition in both CRIMP and STEM passages. Stage 1 bucket cooling passages with thermal loads in excess of 2000 W will require more water flow to avoid boiling transition. Figure 11-1 shows the early design curve and recent power at boiling transition versus flow results at 95 psia.
- STEM-drilled passage performance at pressure is very nearly the same as CRIMP'ed passages (see Figure 11-1). It is therefore no longer necessary to use CRIMP'ed turbulence promoters in passages with high thermal loads. Wetted area is calculated to be greater than 50% in the STEM-drilled passage. Deposits (resulting from the intentional use of tap water in the experiment) on the anti-Coriolis side of the passage provide experimental evidence that droplet evaporation and substantial heat transfer is taking place on the dry side.
- The STEM-drilled surface condition is superior to smooth, reamed holes. Figure 11-1 shows a 25% improvement in power at boiling transition. STEM-wetted area, noted above, exceeds previous smooth passage calculations by more than 35%.
- CRIMP passage performance is sensitive to inlet subcooling and other features of CRIMP geometry, i.e., height, L/D, or section profile.
- Reverse steam flow, accomplished by a vent at the cooling passage entrance, has been shown to be an effective mechanism to achieve high power at boiling transition in both STEM and CRIMP passages. This principle, if applied to the TRV First-Stage bucket, would result in higher exit steam quality. The required stage water flow would be less than 5 lb/s, including a 20% margin for maldistribution of water.
- Tests were conducted in which the tip of the specimen was sustained in a stable post-boiling transition condition. Previously it had been postulated that the onset of boiling transition would propagate and envelop the entire passage instead of stabilizing with a short portion of the passage tip uncooled. The test results mean that the consequence of the low water flow is less threatening and can probably be detected in service by observing high temperature or distress in the bucket tip region.

- Cooling passage performance has been shown to be very repeatable. Sufficient data point repetition was performed to establish statistical confidence in the results within a few percentage points. This achievement was made by eliminating sources of surface contamination and confirming acceptable levels of cleanliness by observing the wettability of passage surfaces. Startup procedures for water-cooled turbine operation will require similar precautions. Silicone grease contamination required heating in air to 1000 °F, followed by a flush with a water-detergent solution. Trichloroethylene was effective in removing nonsilicone films.
- A platform cooling passage, in series with the airfoil cooling channel, was tested. Thermal performance is independent of platform passage orientation, i.e., leading, following, or perpendicular to the passage plane of rotation. Comparable results were also achieved with the platform passage bypassed. Therefore, it is concluded that entrance conditions are not a factor in passage performance.

Figure 11-1. Power at boiling transition for the TRV first-stage bucket (4.1 in. long, 95 psia)

TRV first-stage bucket cooling passages and their current design status flow requirements are summarized in Table 11-1 as they were specified early in 1981. CRIMP'ed turbulence promoters were specified in passages with high thermal loads and where high wetted area was required to achieve LCF life goals on the suction side of the bucket (i.e., where the liquid coolant film is remote from the surface). Reaming was specified in holes where strain levels could not accommodate a 4X reduction in LCF life, i.e., where predicted life was less than four times the design life of 3350 cycles. This factor was applied pending crack initiation evaluation of STEM-drilled IN718 materials in an aqueous environment.

The thermal load summary of platform cooling included only the vaporization thermal load, neglecting heating to saturation, since power at boiling transition was taken to be independent of subcooling. With the development of design curves, Figure 11-1, based on a specimen with a prototypical platform passage in series with the cooling passage, it is now more consistent to include the entire platform cooling load.

Table 11-2 is a revised summary of cooling requirements incorporating the revised platform thermal loads. CRIMPs are no longer specified in any passages because of the STEM passage power at boiling transition capability with some increase in flow. STEM-drilled passage wetted area and observed dry-side heat transfer is judged to be adequate to avoid degradation in predicted LCF life on the suction side of the bucket.

Reaming of cooling passages is also no longer specified. Smooth surface power at boiling transition is inferior to the STEM conditions per Figure 11-1, and tests on cast IN718 STEM material in the aqueous environment have not demonstrated a significant reduction in LCF life.

Table 11-2 tabulates the flow required to avoid boiling transition for the total thermal load for each passage. The next column includes a 20% margin for maldistribution of flow and uncertainty in the flow-versus-boiling transition design curve data supporting Figure 11-1. In previous probabilistic analyses, a 20% margin in flow was sufficient to calculate a 99% chance of avoiding boiling transition in any cooling passage. It is possible to design the Resistance Flow Divider (RFD) with variations in individual passage flow by varying the tortuous path length. Previously, as in Table 11-1, two flows were specified. With the range of flow requirements in Table 11-2, three flows are specified to minimize excess water consumption while not unduly complicating the RFD design.

Table 11-1
TRV FIRST-STAGE BUCKET COOLING SUMMARY (1981)

Cooling Passage No.	Diameter, in.	Surface Condition*	Platform Passage Length, in.	Thermal Load, watts			Design Flow gpm
				Platform	Air Foil**	Total	
1	0.062	C,R	1.4		1900	1900	0.023
2	0.062	C,R	2.9	330	1610	1840	0.023
3	0.062	R	—		1620	1620	0.016
4	0.094	C,R	—		1750	1750	0.023
5	0.094	R	1.3		1540	1540	0.016
6	0.141	C,R	—		2140	2140	0.023
7	0.141	C,R	2.2	150	1650	1800	0.023
8	0.094	C,R	—		1570	1570	0.016
9	0.094	C,R	1.1		1780	1780	0.023
10	0.094	C,R	—		1610	1610	0.016
11	0.094	C,R	—		1580	1580	0.016
12	0.094	C,R	—		1580	1580	0.016
13	0.094	C,R	—		1550	1550	0.016
14	0.094	C,R	—		1510	1510	0.016
15	0.094	C,R	0.2		1490	1490	0.016
16	0.094	C,R	0.4		1500	1500	0.016
17	0.094	C,R	1.1		1500	1500	0.016
18	0.094	C,R	—		1370	1370	0.016
19	0.094	C,R	1.6	80	1250	1330	0.016
20	0.062	R	—		1060	1060	0.016
21	0.062	R	1.1		1410	1410	0.016
22	0.062	R	—		1310	1310	0.016
23	0.094	C,R	1.1		1670	1670	0.023
24	0.094	S	1.8		1600	1600	0.016
25	0.094	S	1.5		1410	1410	0.016
26	0.094	C,R	2.3	170	1530	1700	0.023
27	0.094	C,R	2.2	150	1660	1810	0.023
28	0.094	S	1.8	80	1530	1610	0.016

Bucket total flow = 0.517 gpm = 0.071 lb/s Stage flow = 5.7 lb/s

*C = CRIMP'ed turbulence promotor

R = Reamed hole

S = STEM-drilled holes

**100% coverage assumed for holes 1-11, where tilt is > 5°; all others are 75%.

Table 11-2
TRV FIRST-STAGE BUCKET COOLING SUMMARY (Revised 1982)

Cooling Passage No.	Diameter, in.	Surface Condition*	Platform Passage Length, in.	Thermal Load, watts			Flow to Avoid PBT, gpm	With Safety Margin (1.2 × Flow)	Design Flow gpm
				Platform	Air Foil**	Total			
1	0.062	S	1.4	290	1900	2190	.0316	.0379	.0438
2	0.062	S	2.9	640	1610	2250	.0365	.0438	.0438
3	0.062	S	—		1620	1620	.0130	.0156	.0168
4	0.094	S	—		1750	1750	.0152	.0182	.0276
5	0.094	S	1.3	240	1540	1780	.0186	.0223	.0276
6	0.141	S	—		2140	2140	.0286	.0343	.0438
7	0.141	S	2.2	460	1650	2110	.0275	.0333	.0438
8	0.094	S	—		1570	1570	.0124	.0149	.0168
9	0.094	S	1.1	180	1780	1960	.0214	.0257	.0276
10	0.094	S	—		1610	1610	.0130	.0156	.0168
11	0.094	S	—		1580	1580	.0125	.0150	.0168
12	0.094	S	—		1580	1580	.0125	.0150	.0168
13	0.094	S	—		1550	1550	.0122	.0146	.0168
14	0.094	S	—		1510	1510	.0116	.0139	.0168
15	0.094	S	0.2	60	1490	1550	.0122	.0146	.0168
16	0.094	S	0.4	120	1500	1620	.0130	.0156	.0168
17	0.094	S	1.1	180	1500	1180	.0140	.0168	.0168
18	0.094	S	—		1370	1370	.0100	.0120	.0168
19	0.094	S	1.6	390	1250	1640	.0136	.0163	.0168
20	0.062	S	—		1060	1060	.0068	.0082	.0168
21	0.062	S	1.1	250	1410	1660	.0138	.0166	.0168
22	0.062	S	—		1310	1310	.0093	.0113	.0168
23	0.094	S	1.1	140	1670	1810	.0165	.0198	.0276
24	0.094	S	1.8	210	1600	1810	.0165	.0198	.0276
25	0.094	S	1.5	230	1410	1640	.0136	.0163	.0168
26	0.094	S	2.3	480	1530	2010	.0230	.0276	.0276
27	0.094	S	2.2	460	1660	2120	.0282	.0338	.0438
28	0.094	S	1.8	390	1530	1920	.0196	.0235	.0276

Bucket total flow = 0.681 gpm = .095 lb/s Stage flow = 7.6 lb/s

* S = STEM-drilled holes

** 100% coverage assumed for holes 1-11, where tilt is > 5°; all others are 75%.

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Reporting Category XII

PROGRAM MANAGEMENT, REVIEWS, DOCUMENTATION, AND REPORTS

12.1 REVIEWS

No customer reviews were conducted during this reporting period.

12.2 DOCUMENTATION

12.2.1 Patent Clearance

Requests for patent clearance and technical concurrence to publish the following papers were made to DOE.

- Proposed article for the General Electric Company publication *Electric Forum* entitled "High-Temperature Water-Cooled Gas Turbine for Increased Energy Efficiency."
- A technical paper prepared for presentation at the ASME International Gas Turbine Conference in London, 1982, entitled "Development, Fabrication, and Testing of a Prototype Water-Cooled Gas Turbine Nozzle."
- A technical abstract planned for presentation at the ASME/IEEE-sponsored Joint Power Generation Conference in Denver, 1982, entitled "Water Cooled Gas Turbine Nozzle Technology Demonstration at Ultra High Firing Temperature."

12.3 NOTICE OF INVENTION DOCKETING

There were no notifications of invention docketing submitted to DOE this reporting period.

12.4 REPORTS

12.4.1 Financial Reports

The contract cost reports for December 1981, and January and February 1982 were submitted.

12.4.2 Technical Reports

The technical progress reports for the months of December 1981 (DOE/ET/10340-130), January 1982 (DOE/ET/10340-131), and February 1982 (DOE/ET/10340-133) were published.

The quarterly technical progress report for the period October – December 1981 (DOE/ET/10340-129) was published in fulfillment of contract requirements.

12.4.3 Topical Reports

The final release of the topical report *Rotor Water Delivery and Distribution Tests* (DOE/ET/10340-120) was accomplished in fulfillment of contract requirements.

The final release of topical report *Hot Gas Path Development Test, Part I* (Facility Description) (DOE/ET/10340-122) was also accomplished in fulfillment of contract requirements.

One more topical report is required by the contract. Efforts to prepare "Hot Gas Path Development Test, Part II" have been initiated and are continuing. A draft of the subject topical report is being prepared for release in the upcoming reporting period.

12.4.4 Final Technical Report

Efforts to prepare a draft of the Final Technical Report were continued this quarter.

12.4.5 Energy RD&D Project Report

The *Energy RD&D Work in Progress* report required annually by the HTTT contract was prepared and released in fulfillment of contract requirements.

12.4.6 Abstracted Technical Annual Report

The *Abstracted Technical Annual Report* for calendar year 1981 was prepared and released in fulfillment of contract requirements.