DESIGN, INSTALLATION, AND TEST OF A STEAM INJECTION SYSTEM ON AN EARLY FRAME 3 GAS TURBINE IN A COMBINED CYCLE PIPELINE COMPRESSOR STATION

by

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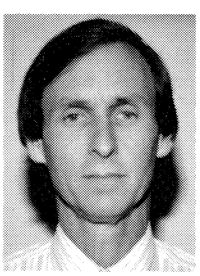

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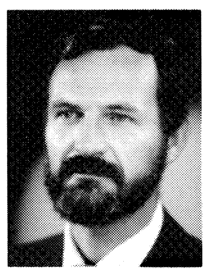

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ABSTRACT

The Frame 3 Gas Turbines have proven to be reliable drivers for pipeline compressor stations over the last 25 years. At Transco's combined cycle compressor station, a steam injection system for power augmentation was developed and found to be a cost effective alternative to the OEM's turbine upgrade offering.

The design, build, and test of this unique steam injection system required with the "straight-through combustion" configuration of the early Frame 3 Gas Turbines is described.

Due to the unavailability of basic thermodynamic design data and constraints from the OEM, three gas turbines at the station were performance tested. A performance model for the gas turbine was developed and validated. It was supplemented with steam system test data to develop the combined cycle model of the station. The model was used to assess the feasibility of, and to provide a design basis for the steam injection system. Overall combined cycle efficiency changes with steam injection system were evaluated.

A steam injection system was designed, fabricated and tested to evaluate its field performance. Field testing confirmed the expected horsepower increase, but an unexpected increase in the exhaust thermocouple spreads at higher steam injection rates was also experienced. Correlation of the combustor liner *vs* the exhaust thermocouple readings showed some interesting characteristics of these turbines. Operating guidelines for steam injection operation are presented.

INTRODUCTION

Transco has been operating its centrifugal compressor stations at two of its mainline locations since the early 1950s. These stations started out as steam plants, i.e., steam turbines were the prime movers for centrifugal boosters with gas fired boilers supplying the steam. The plant thermal efficiency was quite low. In the early 1960s, Frame 3 gas turbines were added as the throughput of the pipeline was increased. To improve the fuel efficiency, these gas turbines were equipped with supplementary fired heat recovery steam generators (HRSG) to recover heat from the exhaust gases [1]. This steam was used to drive the steam turbines and the older boilers were retired. One boiler is still being used to facilitate initial startup and to maintain control during plant upsets. The current equipment at the Billingsley, Alabama, station is described in Table 1. The equipment at the other station in Tylertown, Mississippi, is similar. These configurations are quite unique within the pipeline industry, as most pipeline stations do not have steam turbines at their sites.

Table 1. Description of Units. Transcontinental Gas Pipe Line Corporation Compressor Station 100, Billingsley, Alabama.

Unit No.	Driver/Compressor	HP Rating	Yr. Installed
1 – 3	Westinghouse/Clark Steam Turbine	5620	06/51
4	Westinghouse/Clark Steam Turbine	5600	04/57
5	G.E. MS3002B/DeLaval	6800*	11/62
6	G.E. MS3002F/DeLaval	7210	11/66
7	G.E. MS3002F/DeLaval	7210	09/68
8	G.E. MS3002F/DeLaval	8460	12/72

* Unit 5 was upgraded to 8780 HP as a result of the Steam Injection System

The four gas turbines were equipped with prespeedtronic hydraulic fuel regulator controls. Over the years, these controls were becoming difficult to maintain due to age, unavailability of quality spare parts and field support [2]. Although more modern controls were being offered, it was difficult to justify upgrades due to cost and experience with upgrades by other users. A decision to implement cost effective control upgrades on a unit by unit basis was made in 1987. One unit at each location was upgraded in 1988 with a new system with overall good results. A less expensive upgrade was also explored [3]. It was becoming evident to management that cost effective solutions require additional engineering resources and extensive user involvement. To assure long term reliability of this equipment, it was critical to develop relationships with specialized vendors.

A pipeline expansion project required an additional 2000 "peaking" horsepower at this station. The OEM recommended a horsepower upgrade option [4] that would have required the change out of most of the hot gas path components at an installed cost of about $1100/hp. All of the gas turbine parts except for the air compressor and the exhaust duct work would have been replaced. An alternate proposal was developed to increase the horsepower of the unit via installation of a steam injection system. The project scope included the following:

- Conduct a major overhaul on the unit to upgrade components to "as new" condition.
- Evaluate the plant demineralized water supply and quality and upgrade as necessary.
- Upgrade the control system for the gas turbine and integrate the steam injection controls as necessary.
- Design, install and test a steam injection system to upgrade horsepower.

The overall cost for this project was estimated to be $500/hp. Based on several considerations including cost and delivery, a decision was made to proceed with the installation of the steam injection system.

STEAM INJECTION TECHNOLOGY REVIEW

Steam injection is fast becoming a mature technology that has gained varied and wide application within the utility, cogeneration, and petrochemical industries. Most of the turbine manufacturers are utilizing this technology [5] for NOx reduction and power augmentation. However, injecting water or steam into a gas turbine is not a new idea. Water injection for short periods of thrust augmentation was at one time common in jet aircraft engines, although fans serve this purpose today [6].

Various benefits of steam injection have been known for a long time [7]. However, steam injection application got a significant boost due to mandated NOx emission requirements in California and Japan. In the early 1980s, steam injection was a technology of choice for controlling NOx, especially for combined cycle plants [8, 9, 10]. Steam injection was also being used to augment horsepower and cycle efficiencies of cogeneration applications that would typically use aircraft derivative gas turbines [11, 12].

Steam injection systems have been successfully retrofitted for NOx control [13] in 251B8 gas turbines and for NOx and power augmentation [14] in Frame 5 gas turbines. Although steam injection has been successfully utilized for many years, the OEMs have shown reluctance to promote these systems for retrofit applications.

With a steam injection system, a typical gas turbine fired on natural gas produces a NOx emission level of 45-55 ppm at base load rating [15]. Use of SCR systems (DeNOx systems) for more stringent NOx control is necessary, and a Nox range of about 15 ppm is achievable. These systems are becoming more reliable and cost efficient, due to better than expected catalyst life.

PROJECT OVERVIEW

After project approval, an overall plan of action was developed. For the older "straight-through" gas turbines, retrofit design for power augmentation required a careful review of the unit's mechanical and performance characteristics. Prior to shutdown, a performance test of the gas turbines at the station was conducted to evaluate their condition. The results of these tests are discussed later.

In addition, a plant wide study of steam purity was conducted to determine if water treatment needed to be upgraded to minimize the possibility of hot gas corrosion on gas turbine blades. The oxygen scavenger was changed from a corrogen (catalyzed sodium sulfite) to a more volatile one. This is a volatile oxygen scavenger that contributes no inorganic solids to feedwater. The capacity of the plant demineralized water system was to be increased by 40 gpm to meet the steam injection requirements. A duplicate of an existing demineralzer plant increased the rated capacity from 75 gpm to 150 gpm.

Careful inspection of gas turbine internals was conducted at site during unit overhaul. This review confirmed that the turbine was in good mechanical condition. A decision was made to apply a thermal barrier coating on the combustor liners and the transition pieces to increase service life. In addition, a special wheel inspection program was conducted to evaluate the life of existing "composite design" wheels on this unit. The wheels had no indication of any defects in the bore or under the dovetail roots. The welded joints were in excellent condition, based on the NDT and transverse hardness tests.

To minimize risks associated with retrofit designs, a decision was made to work with vendors who had either the know how or demonstrated experience with such steam injection systems.

The system feasibility study along with various design features are outlined. The steam injection system was fabricated, installed, and tested within budget and on schedule. The results of these efforts are also described.

GAS TURBINE PERFORMANCE MODEL

In March 1989, limited performance tests were conducted on Unit 5 (Model B) to establish a performance model for the gas turbine. Some data were gathered on Units 6 and 7 (Model F) to validate this model. The model is based on the schematic outlined in Figure 1.

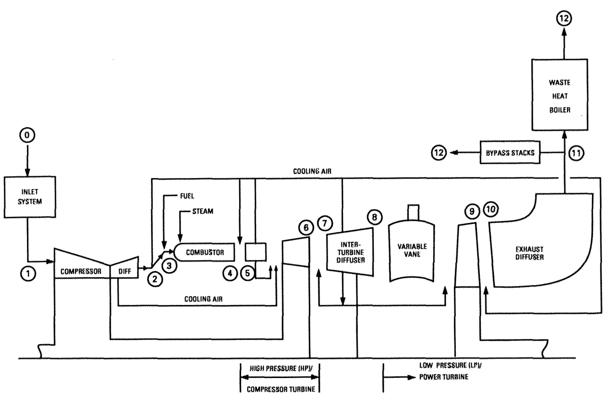

Figure 1. MS 3002 Performance Program Schematic.

Flows, temperatures, and pressures at each of locations 0 to 12 are determined from component characteristics, and fuel (and steam) input. A steady state solution is achieved when cycle parameters upstream and downstream of each component are consistent with the component's aerodynamic performance.

Some points of interest in the performance model development are:

• Compressor turbine vane and blade geometry was input to a meanline aerodynamic analysis program and stage characteristics (swallowing capacity, efficiency and exist flow angle) were calculated. The efficiency at which power is extracted from the gases is shown in Figure 2, flowing from compressor turbine inlet pressure to exit pressure as a function of stage inlet temperature and shaft speed.

• Power turbine vane and blade geometry was determined from the available site manuals, drawings, and throat gauging information. Because the power turbine vane can vary from minus five degrees (fully closed) to plus 22 degrees (fully open) during turbine operation, and stage characteristics do not vary linearly

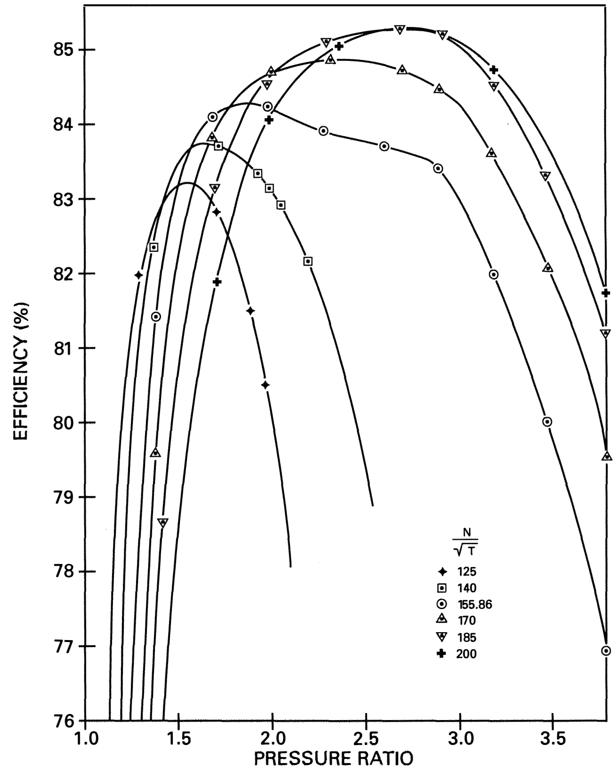

Figure 2. MS 3002 Compressor Turbine Efficiency.

with vane position, it was necessary to produce aerodynamic characteristics at minus five degrees, minus one degree, plus three degrees, plus seven degrees, plus 12 degrees, and plus 22 degrees. The vane midsection suction side and trailing edge is shown in Figure 3 at these positions. A brief illustration of the variation in properties with vane angle is shown in Figure 4 for peak efficiency at each vane position.

All detailed characteristics were incorporated into the performance model and compressor design point values adjusted until a

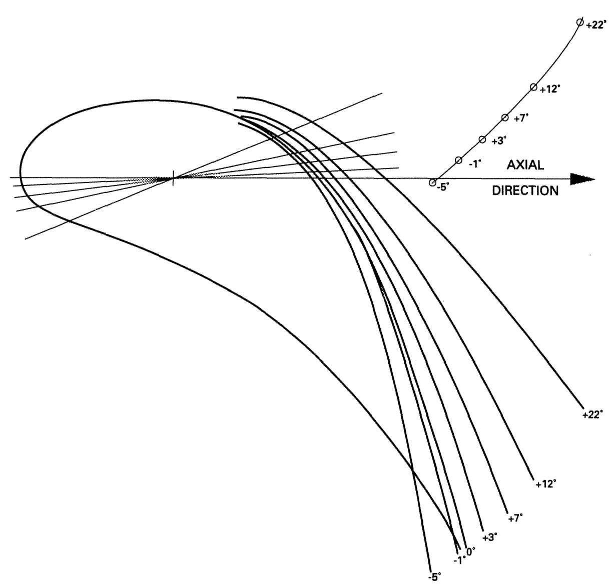

Figure 3. Power Turbine Vane at Various Positions.

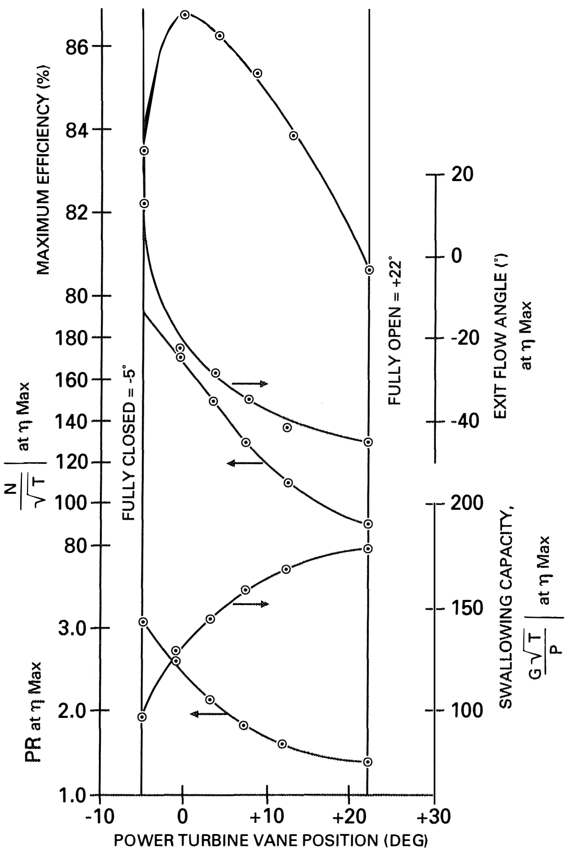

Figure 4. Power Turbine Characteristics vs *Vane Position.*

reasonable match of Unit 5's March test data was achieved, as shown in Table 2.

The model results were then compared to the available quoted data for Unit 5. At quoted exhaust temperatures, calculated horsepower was within 300 hp of quoted, similar to the match against measured data.

Table 2. Unit Number 5. Comparison of Calculated and Measured Performance Data.

DATA		Run 3 Mar. 8/89	Run 4 Mar. 8/89	Run 5 Mar. 8/89
Compressor Delivery Press (PSIG)	Calculated	66.4	69.0	65.2
	Measured	66.3	68.5	65.0
Fuel Flow (MSCFH)	Calculated	108.1	110.3	98.7
	Measured	106.2	109.2	97.1
Exhaust Temp (°F)	Calculated	885	875	820
	Measured	883	877	822
Horsepower	Calculated	7505	7690	6470
	Measured	7764	7853	6770
Power Turbine Vane Angle(°)	Calculated	+1.5	+2.3	+3.0
	Measured	+2.0	+3.0	+4.0

It was concluded that the gas turbine model was valid for steam injection studies and that Unit 5 was performing close to original quotations.

COMBINED CYCLE MODEL AND STEAM INJECTION POTENTIAL

For the purpose of studying the effect of steam injection on combined cycle efficiency, the combined cycle was defined around an individual turbine, boiler, and steam turbine, as shown in Figure 5.

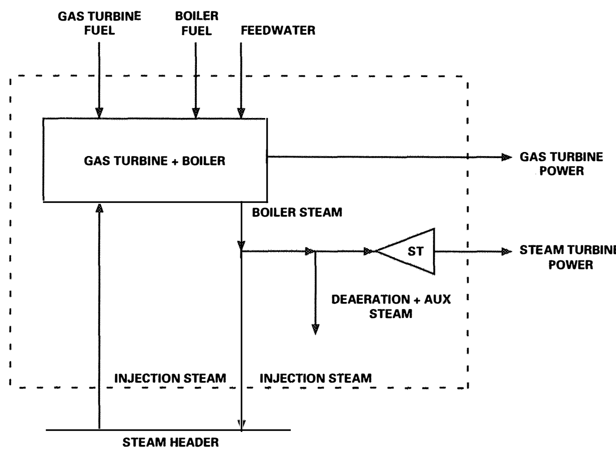

Figure 5. Combined Cycle Schematic Representation.

Boiler evaporator, superheater, and economizer surface areas given by Vogt, were combined with representative heat transfer coefficients for each surface to predict steam output. Model *vs* quoted boiler performance with the boiler fired is shown in Table 3. Gas turbine exhaust was from Unit 5 at site elevation and 59°F ambient temperature, and the boiler was fired to 1140°F (243°F fired temperature rise) by burning 28.9 MSCFH of fuel. It can be seen that at 60,000 pph steam flow, calculated stack temperature and water temperature leaving the economizer were similar to quoted values, while the predicted superheater exit temperature was 23°F below quoted. This 23°F difference was considered acceptable for the steam injection study.

Table 3. Comparison of Calculated and Quoted Vogt Boiler Performance Data.

DATA	CALCULATED	QUOTED
Steam Flow (PPH)	60,000	60,000
Superheater Exit Pressure (PSIG)	625	625
Feedwater Temperature (°F)	225	225
Superheater Exit Temperature (°F)	727	750
Water Temperature Leaving Economizer (°F)	435	450
Stack Temperature (°F)	413	409

Steam turbine power was calculated assuming:

- Steam available for expansion through the steam turbine equalled 87 percent of (boiler steam production - injection steam extraction).
- Steam lost 10°F and 10 psi between superheater exit and steam turbine inlet.
- Steam turbine exit pressure was constant at 2.5 in HGA, and
- Steam turbine efficiency was 68.9 percent.

As discussed earlier, steam injection was an economical means of uprating compressor station throughput, without changing turbine hardware, by utilizing the supplementary fired capacity of the waste heat boiler. The amount of steam generated from the turbine exhaust is not sufficient to operate the steam turbines at full rated power. It is necessary to supplementary fire the waste heat boiler; thus, any extra steam which can be raised by firing can be injected into the gas turbine to boost gas turbine (and thus, combined cycle) power. Combined cycle power is presented in Figure 6 as a function of steam injection and ambient temperature.

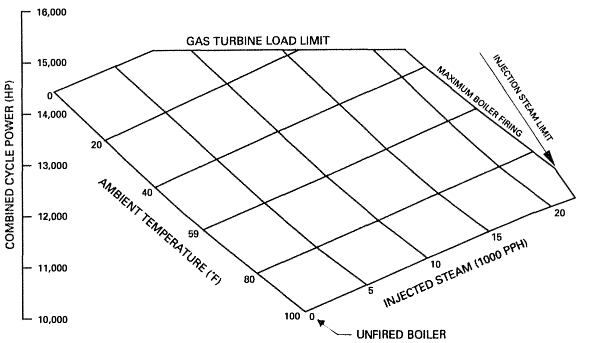

Figure 6. Combined Cycle Power with Steam Injection. With PAMB = 14.48 psia, boiler firing; NC/T = 6950 rpm; 1450°F T.I.T.

Within the limits of gas turbine load and maximum boiler firing, it can be seen that maximum steam injection will occur at an ambient temperature of about 59°F, where combined cycle power will rise from 11,700 hp to 14,100 hp, a 21 percent increase.

The effect of boiler firing and steam injection on combined cycle efficiency can be calculated as follows:

$$\text{Efficiency} = \frac{(\text{HP GT} + \text{HP ST}) \times 550./778.16}{(\text{LHV} \times (\text{GF GT} + \text{GF B}) + \text{G STINJ} \times [\text{h ST} - \text{h SHE}])} \times 100.$$

where:

HP GT	=	gas turbine horsepower
HP ST	=	steam turbine horsepower
LHV	=	fuel lower heating value, BTU/LB
GF GT	=	gas turbine fuel consumption, LB/SEC
GF B	=	boiler fuel consumption, LB/SEC
G STINJ	=	injection steam flow, LB/SEC
h ST	=	enthalpy of injection steam supplied from the steam header, BTU/LB
h SHE	=	enthalpy of steam leaving the superheater, BTU/LB

The injection steam term in the above equation is included to maintain consistency with the combined cycle control volume, because in the actual plant, steam can be supplied by four different waste heat boilers and one stand-alone boiler to the header from which injection steam is extracted.

The variation of combined cycle efficiency with injection steam flow and ambient temperature is presented in Figure 7. It can be seen that boiler firing results in a decreased combined cycle efficiency; at 59°F ambient temperature, combined cycle efficiency drops from 29.3 percent to 27.2 percent, a 7.2 percent decrease.

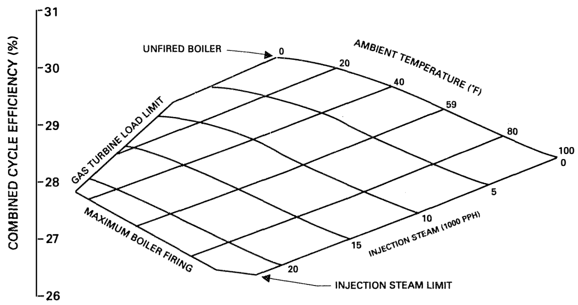

Figure 7. Combined Cycle Efficiency with Steam Injection. With: PAMB = 14.48 psia, boiler firing; NC/T = 6950 rpm; 1450°F T.I.T.

Combined cycle efficiency decreases as the boiler is fired because fuel energy required to raise, and heat, injection steam is converted to horsepower at an incremental efficiency of only 18 percent. When the extra power produced by injection steam at 18 percent efficiency is added to the combined cycle power produced at 29.3 percent before boiler firing, the resulting total power is generated at only 27.2 percent.

The reason for the low steam side efficiency (18 percent) becomes apparent when enthalpy changes experienced by the steam are positioned on a h-s diagram for water vapor. Injection steam enthalpy and entropy is traced in Figure 8, from entry into the cycle as boiler feed water (Point 1), to exit from the cycle as water vapor in the stack (Point 9). A simple division of enthalpy produced as injection steam expands through the compressor turbine and power turbine, by enthalpy added through combustion

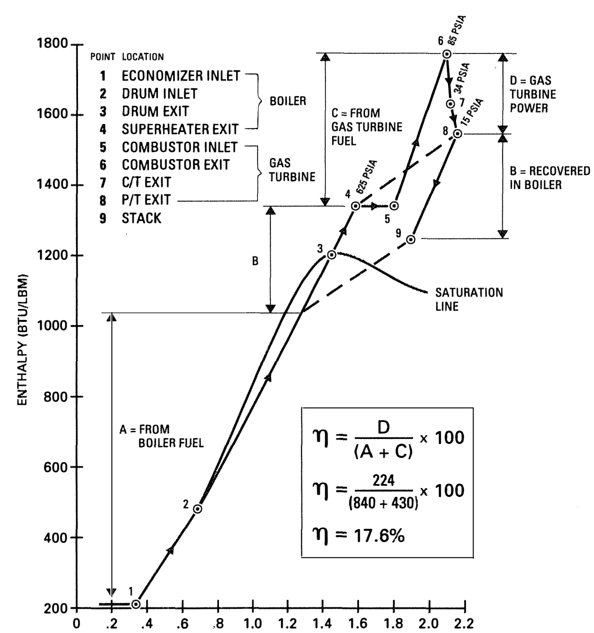

Figure 8. Incremental Efficiency of Injection Steam at 59°F Ambient.

of fuel in the boiler and gas turbine, gives 18 percent injection steam efficiency.

The seven percent reduction in efficiency means that increased station throughput resulting from the 21 percent increase in power will be slightly reduced, due to the extra fuel consumption, but the overall economic evaluation remains positive.

STEAM INJECTION SYSTEM DESIGN

When steam injection is retrofit to an industrial gas turbine for power augmentation, the steam is usually injected into the combustor shell [14] where it mixes with compressor delivery air, before entering the combustors. The early MS 3002 turbines do not have a combustor shell; instead, each of the six liners is enclosed in separate casing, as shown in Figure 9. Compressor delivery air discharges directly into the liner dome region, as shown in Figure 10.

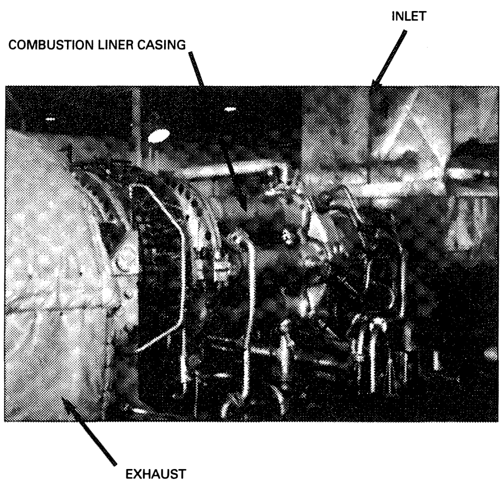

Figure 9. MS 3002 B Unit Number 5 at Billingsley, Alabama.

Since steam was to be injected for power augmentation, not NOx control, the objective was to achieve combustion stability at the highest possible steam/fuel (mass) ratio, while maintaining low liner wall temperatures. Ecob [16] measured a steam-to-fuel mass ratio (SFR) of about 2.2:1 for natural gas fuel, with steam injected in the primary zone before blow out. Experience with compressor delivery air/steam mixing in the combustor shell [14] with some steam reaching the primary zone, showed SFRs of 4.6:1 could be achieved before combustion efficiency started to decrease. Therefore, it was felt that steam introduction downstream of the primary zone would be desirable.

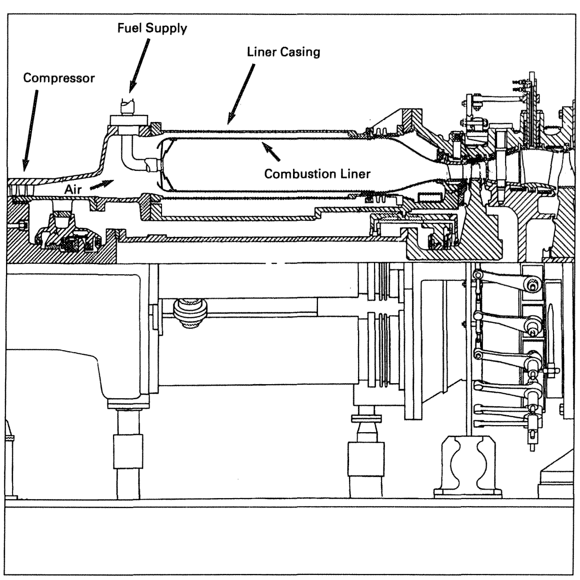

Figure 10. MS 3002 Gas Turbine.

The "straight-through combustor" configuration allowed this objective to be met by the addition of a steam supply manifold around the upstream end of each liner casing, as shown in Figure 11. To allow access to two liner casing flange bolts, the manifold was split into a 46 degree segment and a 283 degree segment. Steam was injected through the liner casing into the annulus around the liner in two rows of 13/64 in diameter holes, equispaced at 20 degree increments and separated axially by 0.8 in. Manifold supply steam was introduced into a nozzle block on the 283 degree manifold from which point it supplied 30 liner casing holes, along with a crossover pipe that supplied the remaining six holes.

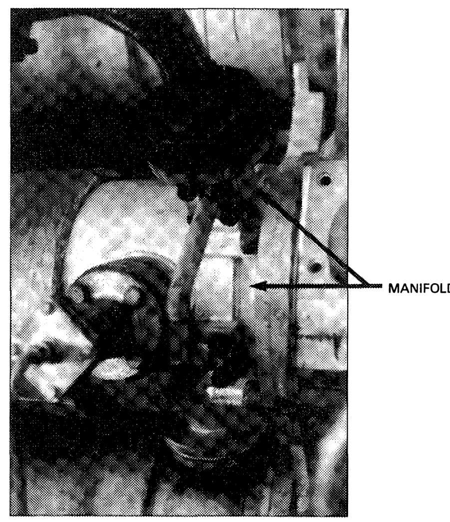

Figure 11. Steam Supply Manifold.

The six individual steam manifolds were supplied from a 280 degree pipe through lined flexibles, as shown for three combustors in Figure 12. To achieve a circumferentially uniform distribution

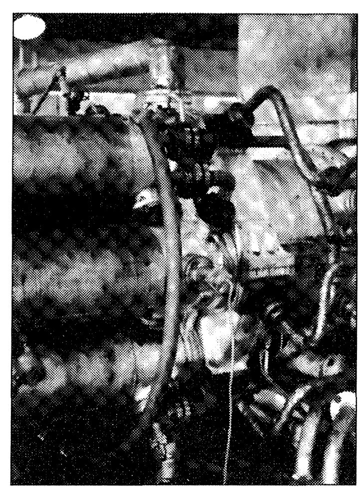

Figure 12. Steam Supply Piping.

of steam into the annulus around the liner, to ensure that the 36 steam jets did not impinge on the liner wall, and to ensure that maximum design steam rate of 30,000 pph would not be exceeded, a flow analysis through the liner casing steam manifold system (Figure 13) was conducted.

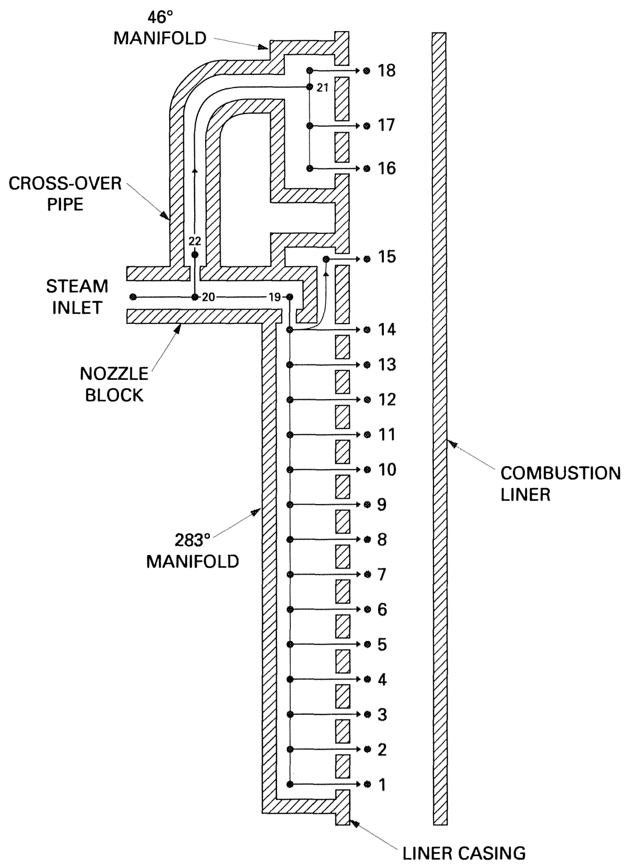

Figure 13. Steam Supply Model.

The effect of approach velocity and wall thickness on discharge coefficients was included using the method suggested by McGreeham [17]. Maximum steam jet penetration into compressor delivery air flowing through the annulus between the liner casing and the liner was calculated using the experimental modification found by Abuaf, et al., to Lefebvre's proposed expression [18]. This maximum penetration was at a distance of three diameters downstream of the hole. With an injection hole diameter of 13/64 in, maximum penetration was held to 0.96 in at maximum stream flow of 30,000 pph. The variation in mass flow through the 18 holes equispaced around the circumference of the liner casing was limited to ± 6.0 percent, as shown in Figure 14.

To minimize steam velocity through the lined flexible hoses, the orifices supplying the six and 30 hole manifolds were sized to pass a total of 30,000 pph steam with the steam throttle valve 100 percent open. This results in a pressure in the flexes of about 425 psia, and a pressure drop across the supply orifices of 300 psi. The large restriction in each liner manifold forces equal flow to each liner.Therefore, steam supply system design ensured that equal steam would flow to each liner, and would be uniformly distributed around each liner.

Injection steam was taken from a nearby three inch starting turbine supply line and routed to the turbine, as shown in Figure 15. In addition to the requirement for steam flow measurement and

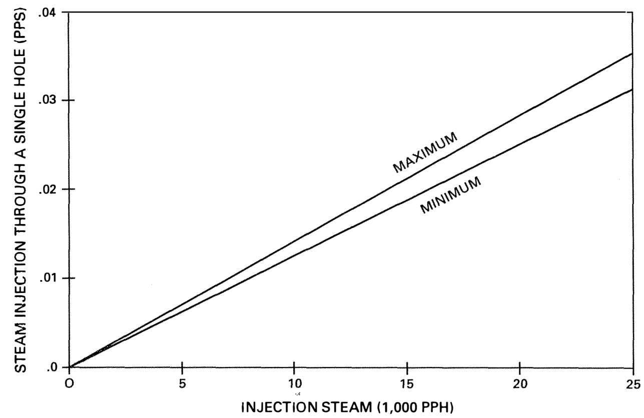

Figure 14. Mass Flow Variation Through Steam Injection Holes.

control, protection against water injection was provided by including a "knockout" drum and steam trap. The "knockout" drum, designed as a cyclone separator, removes water droplets and particulates from the vapor, and provides an accumulator volume for upset conditions.

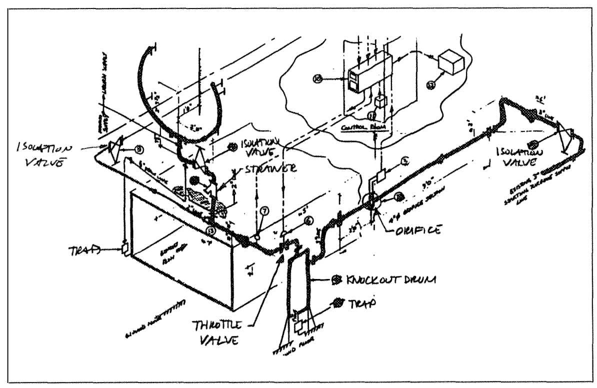

Figure 15. Injection Steam Supply System.

Average combustor exit temperature for the MS 3002 turbine was 1450°F; therefore, it was decided to add an adjustable monitoring thermocouple in the downstream end of each liner, as shown in Figure 16, to see if changes in combustion characteristics could be detected as increasing quantities of steam were injected.

TEST RESULTS

Steam was injected into Unit 5 on November 27, 1990 in gradually increasing quantities (up to 13,000 pph), and all available engine data were recorded. Turbine output increased approximately as expected as the quantity of injection steam was increased, while exhaust temperature spread jumped about 50°F, once more than 10,000 pph of steam was injected (Figure 17). Exhaust spread without steam injection was 120°F which is somewhat higher than normal. Therefore, increase in the spread to 170°F was worrisome.

To gain additional information, the new combustor thermocouples were wired in, adjusted to their maximum penetration of 3.5 in inside the liner (Figure 16a), and at base load without steam injection, a surprisingly high 900°F temperature spread was measured, as shown in Figure 18.

A photograph of a MS 3002 liner (Figure 19a), shows the placement of primary, secondary, and the four, 2⅛ in diam. dilution air holes. The combustor thermocouple was located 13 in

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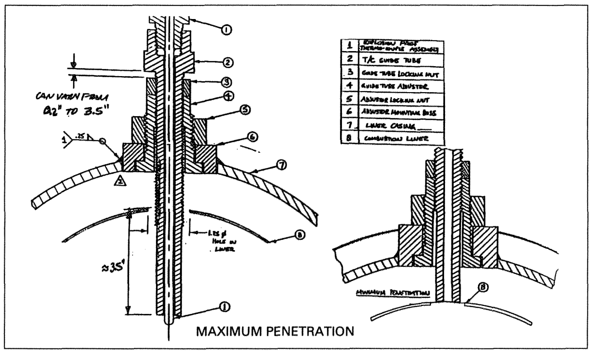

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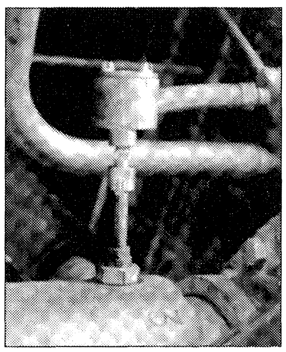

Figure 16. Adjustable Combustor Thermocouple.

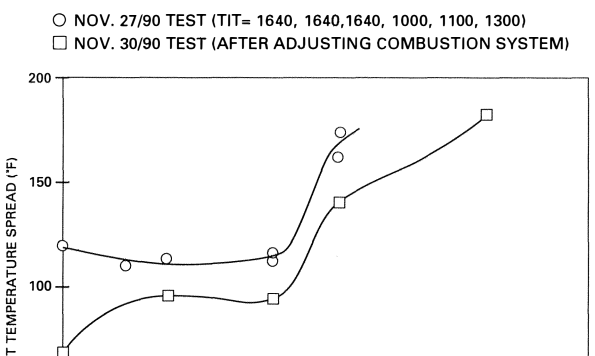

Figure 17. Exhaust Temperature Spread vs *Steam Injection Flow.*

downstream of the center of the dilution holes (6 hole diameters), and circumferentially between them (Figure 19b), a position far enough downstream that considerable mixing of hot gas with the cold dilution jets should have occurred, as shown in Figure 19c (Cameron, et al. [19], measured mixing patterns downstream of wall injection jets in a can combustor which support this hypothesis).

For the November 28, 1990, run, compressor delivery (dilution jet) temperature was about 450°F while upstream of the dilution holes, the mass averaged temperature inside the liner was about 2100°F. At the plane of the thermocouples, mass averaged temperature was about 1470°F, so the average measured combustor temperature of 1523°F was reasonable for a probe located in the mixing region.

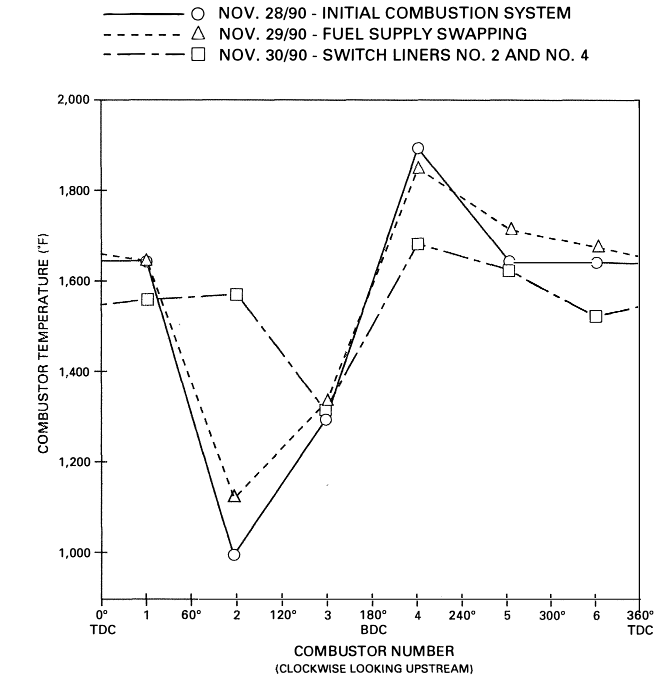

Figure 18. Combustor Temperature Profiles. With: no steam injection; thermocouples at maximum penetration.

Therefore, it was concluded that combustion system component tolerances were causing the cold core of the four merged dilution jets to shift towards, or away from, the combustor thermocouple tip, especially in liners No. 2, 3, and 4.

To see if the liner spread could be decreased, the following steps were taken:

• New fuel nozzles were substituted at the hottest and coldest positions. The unit was retested and no change in liner temperatures was found.

• All 90° fuel supply elbows (Figure 10) were removed, cleaned internally, and all fuel tip holes checked for correct diameter and (lack of) internal burrs. Liners were checked for fit-up of crossfire tubes and all were found to be visually correct. Fuel elbows were reinstalled in different locations and the engine retested; the result being only a slight change, as shown in Figure 18 (spread reduced to 735°F).

• Liners in positions 2 (cold) and 4 (hot) were interchanged and the turbine retested. This action caused a significant reduction in spread, to 368°F as shown in Figure 18.

In addition to reducing combustor thermocouple spread to 368°F, the changes to combustion hardware also significantly reduced exhaust temperature spread from 140°F to 75°F, as shown in Figure 20.

From this exercise, a number of tentative conclusions were drawn:

• Variation in combustion system temperature profile is accompanied by variation in exhaust temperature profile, and

• These turbines have been operating for many years with relatively high exhaust spreads (up to 150°F) without unusual distress to hot end components. Thus, low exhaust spread is desirable, but not necessary for gas turbine performance and life for the MS 3002.

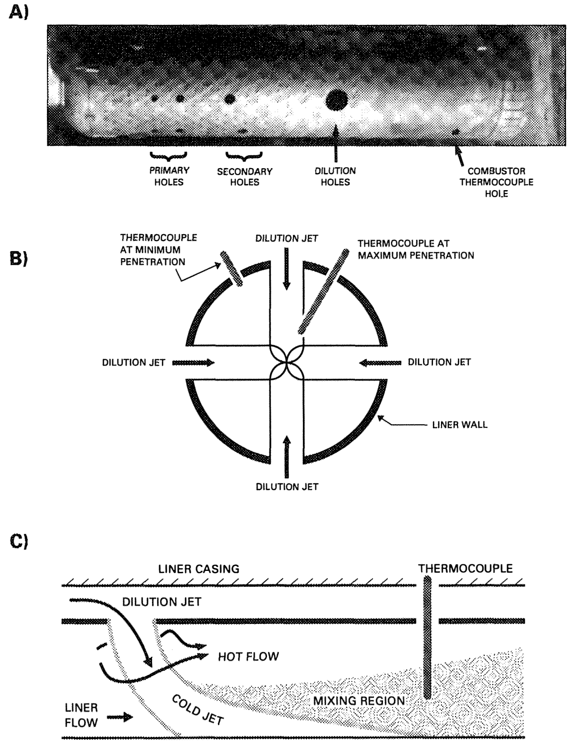

Figure 19. MS 3002 B Combustion Liner.

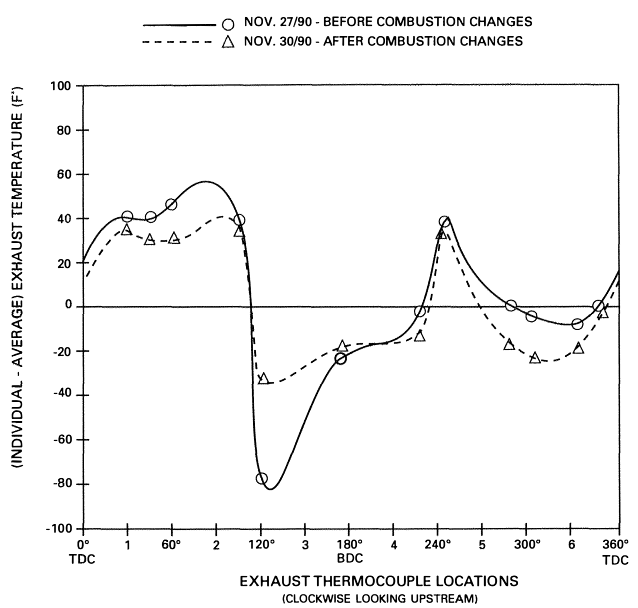

Figure 20. Exhaust Temperature Profiles. With: no steam injection.

With the improved combustion uniformity, it was decided to retest steam injection. This time, up to 20,000 pph of steam was injected and again, increases in gas turbine power with steam injection were approximately as expected. However, past 10,000 pph, exhaust spread increased as before, as shown in Figure 17.

This time, combustor outlet temperatures were available, as plotted in Figure 21. From 0 to 10,000 pph, temperatures changed somewhat; between 10,000 and 15,000 pph, temperature change reversed direction, and increased dramatically; and between 15,000 and 20,000 pph, temperatures continued to shift, but not as rapidly. It was obvious from these data that increasing quantities of steam were causing significant changes in combustor mixing patterns, the results of which were being detected by increased exhaust temperature spread. Exactly why these combustion changes occurred was not clear, thus, the long term implication for engine operation was unknown. The overall engine performance was considered acceptable.

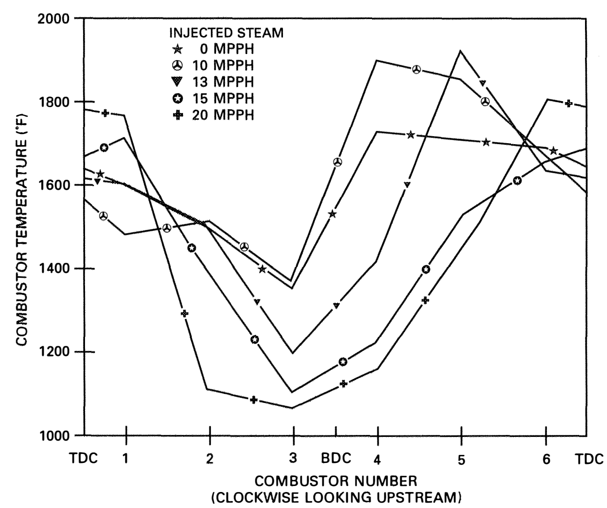

Figure 21. Combustor Temperature Profiles with Steam Injection. With: thermocouples at maximum penetration constant mass averaged turbine inlet temperature.

In March 1991, Unit 5 was retested in an attempt to understand the relationship between steam injection and exhaust temperature spread. During the intervening four months, the following changes had occurred:

- Axial compressor blading was changed.
- All liners were completely dimensionally checked (including louvre gauging).
- A new flow matched gas tip set was tested at the OEM's overhaul facility and installed, and
- The combustor thermocouples were retracted to their minimum penetration position of just piercing the liner wall (Figures 16a and 18b).

With zero steam injection, exhaust spread was found to be greater now than during the November 27, 1990, tests, as shown in Figure 22, despite the fact that a matched gas tip set had been installed, and all liners and transitions had been carefully inspected for fitup both before, and during installation.

Before proceeding with steam injection, exhaust spread was investigated further. Exhaust temperatures from Units 6 and 7 were recorded and compared to those from Unit 5. All three units had similar spreads (115°F to 135°F), all three units had peaks located about 70 degrees clockwise from top dead center (TDC), while Unit 5 had a second distinct peak at 240 degrees clockwise from top dead center, as shown in Figure 23.

All three turbines have identical exhaust systems (leading to waste heat boilers), but their inlet systems are different. Units 6 and 7 have straight down inlets, while Unit 5 has a side/down inlet (Figure 9). It may be possible that inlet/exhaust configuration causes circumferentially nonuniform flow through the air compressor (into the liners), so that even with identical combustion

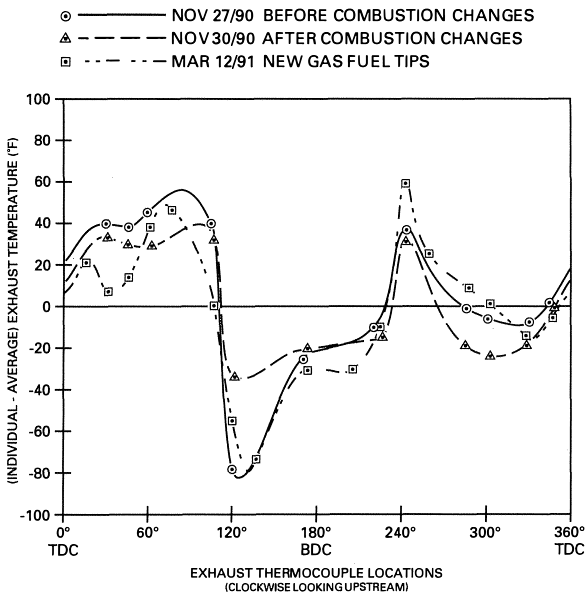

Figure 22. Exhaust Temperature Profile. With: no steam injection.

hardware, different air/fuel ratios would exist in adjacent combustors, and low exhaust spread would never be achieved.

A touch pad temperature probe was available, so to investigate the previously mentioned possibility, the exposed liner casing surface temperature was measured one inch upstream of the insulation sheath and circumferentially centered between the igniter mounting flange and the cross flame tube pipe (Figure 11) for all three units and plotted in Figure 24. Because of natural convection of the room air around the engine, the top two (1 and 6) liner casings would be expected to run a little hotter than would the bottom two (3 and 4) liner casings. Units 6 and 7 tend to follow this trend while casing 6 on Unit 5 is about 20°F lower than would be expected, possibly confirming the idea that compressor exit flow is not uniformly distributed into Unit 5's liner casings.

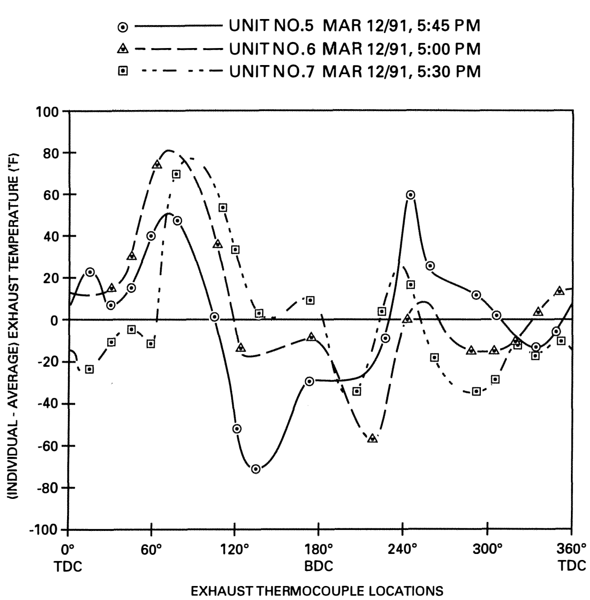

Figure 23. Exhaust Temperature Profiles. With: base firing, no steam injection.

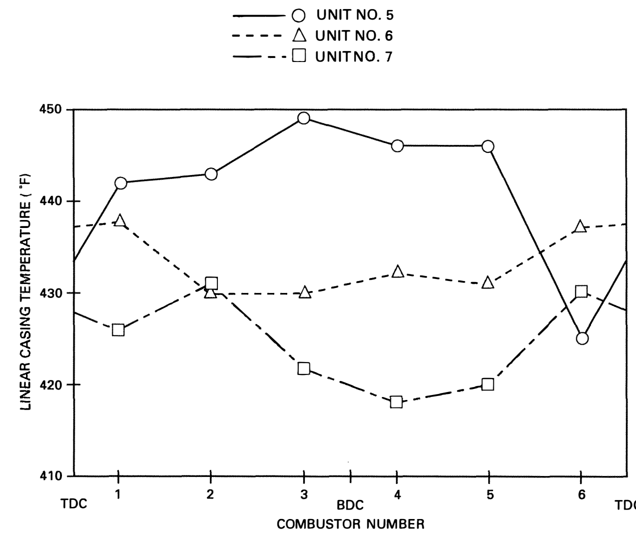

Figure 24. Linear Casing Upstream Temperature Differences.

Midway between the downstream edge of the liner casing insulation sheath and the bellows clamp (Figure 16b), the exposed liner casing surface temperature was measured over the outboard 180 degrees (from TDC to BDC). If combustion uniformity exists, temperatures at corresponding positions on liner pairs 1 and 6, 2 and 5, and 3 and 4, should be similar. The maximum differences between pairs at similar surface locations are shown in Figure 25.

Unit 7 has consistently similar liner casing temperatures, while Unit 5 has variations of up to 95°F, implying significant differences in combustion patterns between liners. The conclusion reached at this stage was that variation in liner-to-liner combustion patterns

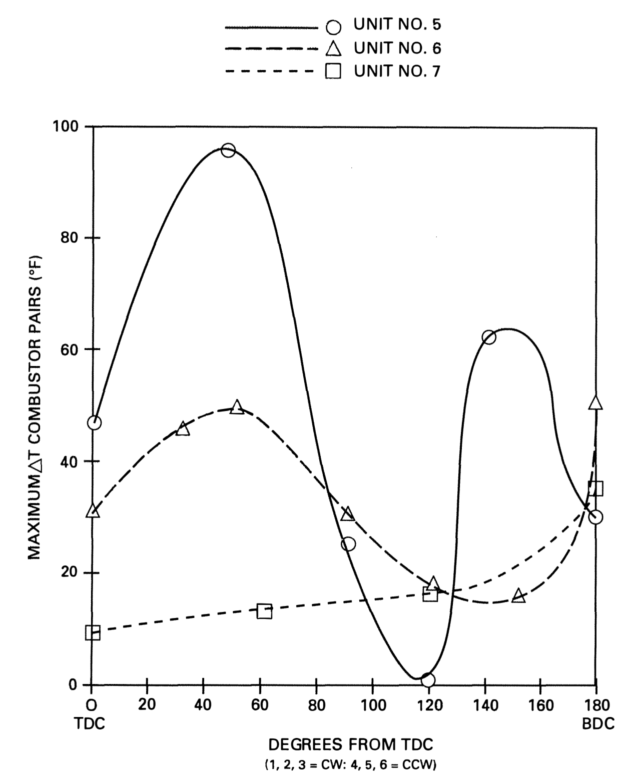

Figure 25. Linear Casing Downstream Temperature Differences.

which evidence themselves as large circumferential exhaust temperature variations, were strongly influenced both by inlet/exhaust configuration and by small (acceptable) combustion component tolerances.

Injection steam was then admitted in quantities of up to 25,000 pph, and full performance data recorded. Because of previously encountered rapid changes in both combustor and exhaust temperatures, steam was injected in 1000 pph increments between 10,000 and 15,000 pph.

Exhaust spread initially decreased, jumped rapidly between 10,000 and 15,000 pph, and then held constant as greater than 15,000 pph steam was injected, as shown in Figure 26.

When exhaust temperature profiles were plotted for 0, 11,000, 15,000, and 25,000 pph of injected steam, as shown in Figure 27, it became evident that exhaust temperature spread on its own is deceiving, because between 0 and 11,000 pph exhaust profile actually changed shape significantly, even though the spread stayed approximately constant. From 11,000 to 15,000 pph, exhaust profile continued to move, while between 15,000 and 25,000 pph there was much less change in the profile.

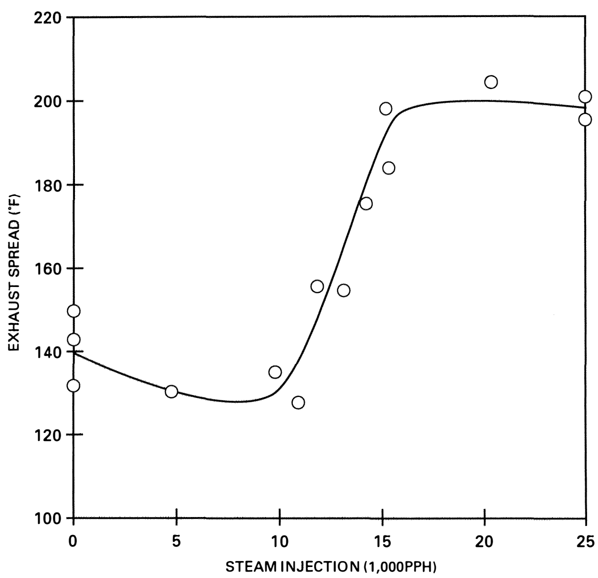

Figure 26. Unit Number 5 Exhaust Spread vs Steam Injection Flow (March 13, 1991 Data).

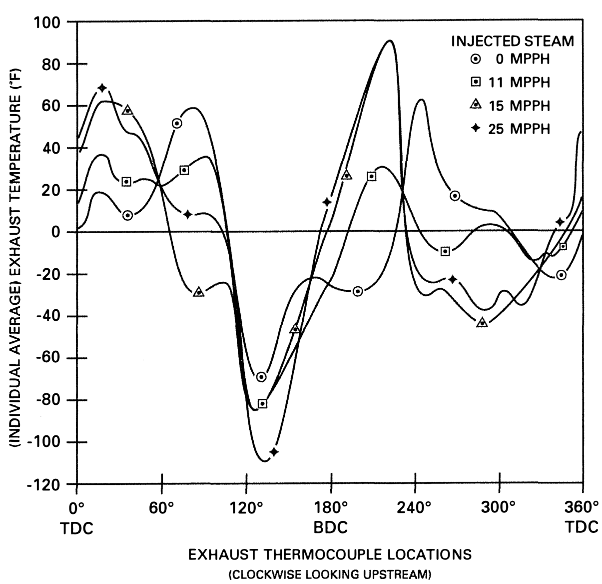

Figure 27. Exhaust Temperature Profiles with Steam Injection.

The temperatures recorded during these tests by the retracted combustor thermocouples are shown in Figure 28. Here, temperatures increased considerably between 0 and 10,000 pph, decreased rapidly between 10,000 and 15,000 pph, and then stayed about the same. Temperature measured with the probes retracted is the air temperature outside of the lining because air will flow through the liner thermocouple access hole and wash the probe tip. Air temperature outside of the lining should reflect compressor delivery temperature, injected steam quantity and temperature, and heat pickup from the liner wall. Therefore, calculated compressor delivery temperature for each run was subtracted from the average combustor temperature, and the results plotted in Figure 29. There was adiscontinuity between 11,000 and 12,000 pph which could only be related to steam injection; therefore, steam supply into the turbine was reviewed in detail.

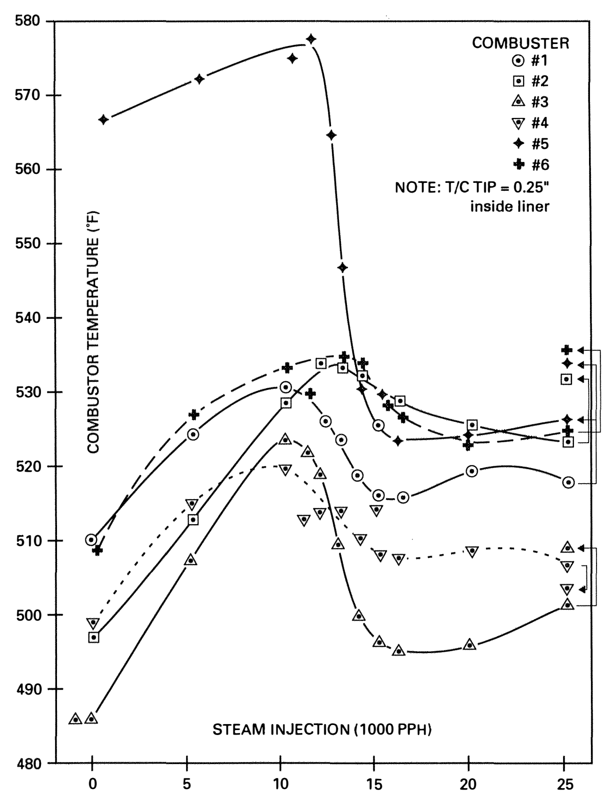

Figure 28. March 13, 1991 Steam Injection Tests.

Combustion liner (Figure 18a) air flows were calculated from measured hole and louvre dimensions using appropriate discharge coefficients, and approximate boundaries of primary-secondary, and secondary-dilution air were determined within the annulus formed by the liner casing and the liner (Figure 10).

Steam injection through the 36, 0.2 in diameter jets in the liner casing can be considered analogous to “film cooling” of the liner casing when jet velocity is low (very small steam flow), but once blowing rate, M ((jet density x jet velocity)/(main flow density x main flow velocity)) exceeds 0.5, the jet penetrates into the main flow [20], as illustrated in Figure 29 [21]. Approximate steam jet

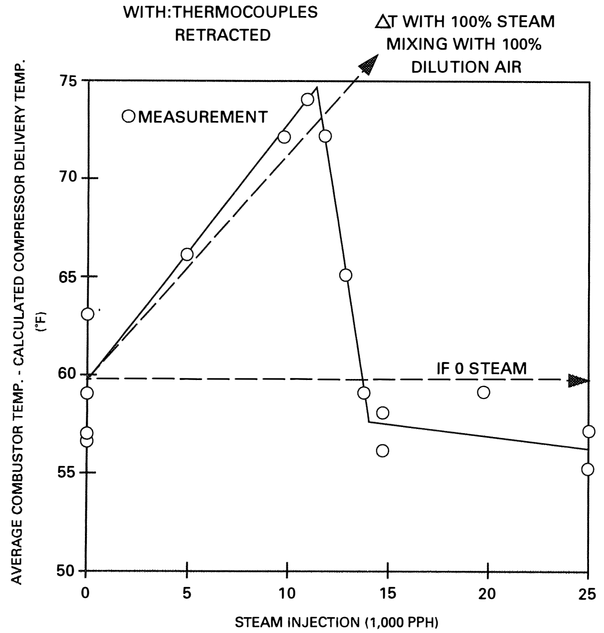

Figure 29. Jet Penetrating into a Crossflow.

centerline trajectories were calculated using Equation 7 from Abuaf, et al. [18], for increasing quantities of steam injection, and the results plotted along with combustion liner air flow, in Figure 30.

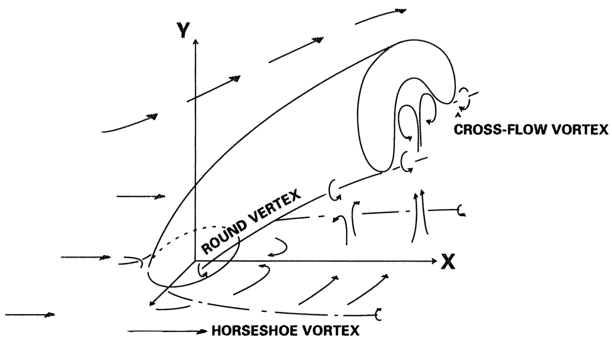

Figure 30. Steam Injection Jets Penetrating Liner Supply Air.

At 5000 pph steam, the calculated centerline trajectory falls entirely within dilution air. However, at 10,000 pph steam, the jet centerline intersects the boundary between secondary and dilution air about four inches downstream of the injection holes in the liner casing. At 15,000 pph, injection steam jets penetrate entirely through dilution air into secondary air.

To test this sequence, the temperature rise due to mixing 100 percent injection steam with 100 percent dilution air was calculated and plotted in Figure 31. It was immediately obvious that the sequence of events is true, namely:

- from 0 to 11,600 pph, all the steam mixes with dilution air,
- from 11,600 pph to 14,000 pph, progressively more of the injected steam penetrates into secondary air, leaving less steam to mix with dilution air, and
- beyond 14,000 pph steam, all of the steam has penetrated through dilution air into secondary and primary air.

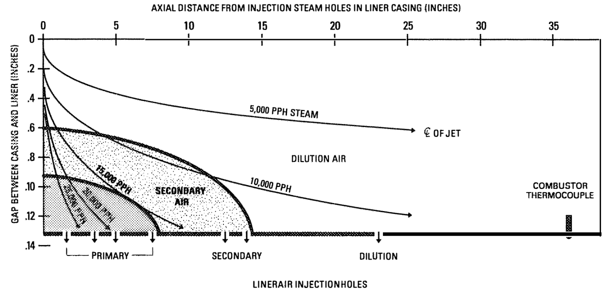

Figure 31. Relationship between Combustor Temperature and Compressor Delivery Temperature with Steam Injection.

Thus, it may be concluded that; changes in exhaust and combustor temperature profiles measured as steam was injected, were caused by steam mixing within different combustion zones, and the resulting changes in temperature profiles within the liners must have been large to cause the detected temperature changes.

To see if the large changes in combustion patterns affected engine performance, the performance program described earlier in the paper was utilized. Three March 13, 1991, runs without steam injection were matched, then using that model, performance without, and with steam injection was predicted by duplicating measured:

- ambient temperature
- barometric pressure
- compressor turbine speed, and
- exhaust temperature,

with power turbine speed matched to the March 13th power turbine load *vs* speed curve. Inherent in the prediction method was the assumption that altered (by steam) combustion patterns did not affect:

- combustion efficiency (held at 100 percent),
- compressor turbine swallowing capacity or efficiency,
- power turbine swallowing capacity or efficiency,
- inter-turbine or exhaust diffuser recoveries.

Measured to predicted changes are compared in Figures 32 and 33 in the primary indicators of turbine performance, as steam was injected. The increase in power was greater than predicted for all quantities of injected steam, fuel consumption generally increased less than expected, compressor delivery pressure was close to expected, and the power turbine vane closed a little more than predicted.

In general, the MS 3002 B gas turbine responded to steam injection somewhat better than predicted. The precise cause of this improved performance could not be deduced from the available data. It is postulated that the improvement was either due to the axial compressor's efficiency being greater than assumed at the higher pressure ratios accompanying steam injection, or due to the compressor turbine's efficiency increasing with the different combustor outlet patterns accompanying steam injection.

Finally, increased exhaust temperature spread caused by injection of steam into the secondary and primary combustion regions (at overall SFRs up to 5.3:1) did not degrade the combustor or hot end performance.

OPERATING GUIDELINES

The steam injection system is available for operation on an "as needed basis" for peaking service. The pipeline operating conditions are evaluated by the gas control dispatcher to determine if

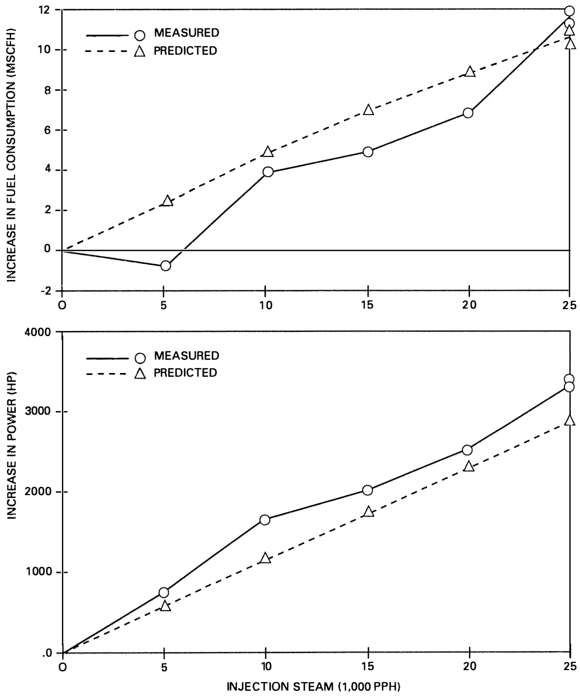

Figure 32. Increase in Power and Fuel Consumption with Steam Injection.

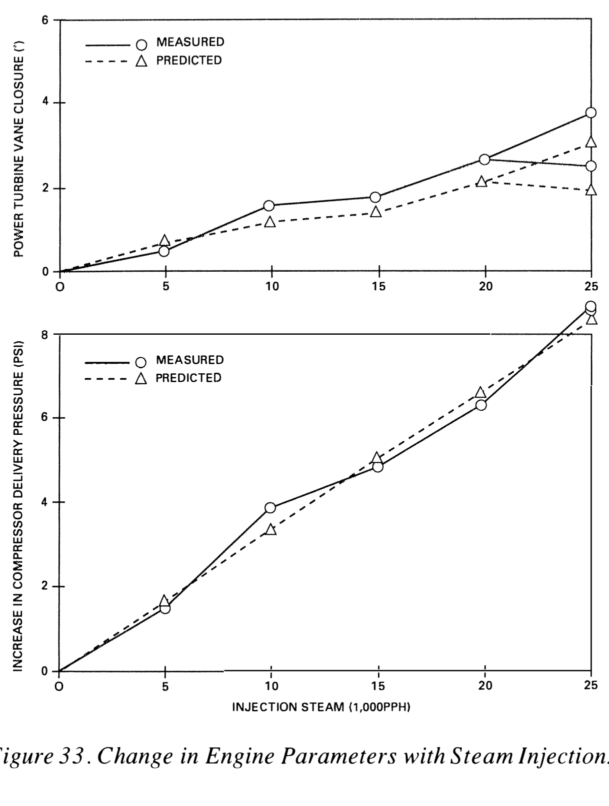

Figure 33. Change in Engine Parameters with Steam Injection.

peaking horsepower is needed. The plant operator is requested to put the steam injection system on.

A startup procedure has been developed that allows the steam piping up to the steam injection header to be warmed up to 530°F prior to opening of the steam injection control valve. The steam is injected at an incremental rate of up to 14,000 lb/hr to yield an additional 2000 hp.

CONCLUSIONS

Steam injection technology is fast becoming a mature technology with varied and wide application within the utility, cogeneration, and petrochemical industries. This technology has been successfully used for both power augmentation and NOx reduction. The steam injection systems have been either part of the original turbine design by the OEMs or have been retrofitted to the existing turbine installations.

This technology has not been utilized widely within the gas transmission industry as the majority of gas turbine horsepower involves simple cycle or regenerative cycle installations. Transco's combined cycle plants are good candidates for use of this technology due to the availability of relatively inexpensive, but good quality steam for injection.

Power augmentation of an existing gas turbine was cost effectively implemented using a steam injection system. These older "straight-through combustor" units required a special modification to the combustors. This system was designed, built and tested for adequacy. The normal "around-the-corner combustor" units have been successfully steam injected in the diffuser section of the air compressor.

The operating envelop for this unit was developed based on performance tests. It was concluded that the overall combined cycle efficiency is reduced when the unit is injected with steam. This is a surprising conclusion as steam injection is usually thought of as increasing plant efficiency, which it does if steam is raised from "waste" heat. When fuel is burned in the HRSG to produce steam for injection, plant efficiency drops because of the steam's low incremental efficiency. However, for Transco's plant, it is cost effective to generate peaking horsepower using steam injection.

At higher steam injection rates, there was an increase in exhaust temperature spreads caused by introduction of steam into different combustion zones. However, there was no degradation of combustor or hot end performance. Based on empirical correlation of combustor temperatures with the exhaust temperatures, it is postulated that the impact on mechanical deterioration of the hot end components will be minimal. The annual hot end inspections were normal. A major overhaul is planned for the summer of 1993 and further inspection and verificationof component life will be made at that time.

REFERENCES

1. Boyce, M. P., *Gas Turbine Engineering Handbook,* Houston, Texas: Gulf Publishing, (January 1992).
2. EPRI Report, "Workshop on Retrofit Controls for Combustion Turbines," EPRI Report No. AP-5098- SR Proceedings (March 1987).
3. Adams, R. L., "Low Cost Exhaust Temperature Control Enhances Gas Turbines," Pipe Line Industry, pp. 41-45 (July 1991).
4. Freeman, A., "MS 3002 Gas Turbine Modernization with Advanced Technology Parts," G.E. Gas Turbine Reference Library GER-3527, 6 pp. (April 1987).
5. Editors, "Trends in Low- Nox Combustion Design and Operating Experience," Gas Turbine World, March - April 1990, pp. 12-15.
6. Hill, P. G. and Peterson, C. R., *Mechanics and Thermodynamics of Propulsion,* New York: Addison-Wesley (1965).

7. Heard, T. C., "Reduction of Gas Turbine Fuel Consumption on Gas Pipelines," ASME Paper No. 76-GT-180 (1976).

8. Touchon, G. L., "An Experimentally Verified NOx Prediction Algorithm Incorporating the Effects of Steam Injection," Journal for Engineering for Gas Turbines and Power, 106, pp. 833-840 (October 1984).

9. Touchon, G. L., "Influence of Gas Turbine Combustor Design and Operating Parameters on Effectiveness of NOx Suppression by Injected Steam or Water," Journal of Engineering for Gas Turbines and Power, 107, pp. 706-713 (July 1985).

10. Hilt, M. B. and Waslo, J., "Evolution of NOx Abatement Techniques through Combustor Design for Heavy Duty Gas Turbines," Journal of Engineering for Gas Turbines and Power, 106, pp. 825-832 (October 1984).

11. Masserlie, R. L. and Tischler, A. O, "Test Results of a Steam Injected Gas Turbine to Increase Power and Thermal Efficiency," 18th Intersociety Energy Conversion Engg. Conference (1983).

12. Larson, E. D. and Williams, R. H., "Steam Injected Gas Turbines," Journal of Engineering for Gas Turbines and Power, 109, pp. 55-62 (January 1987).

13. Kaempffer, L. and Williams K., "Steam Injecting a Mature Gas Turbine in a Modern Cogeneration Plant," ASME Paper No. 89-GT-167 (1989).

14. Little, D. A. and Rives, J.P., "Steam Injection of Frame 5 Gas Turbines for Power Augmentation in Cogeneration Service," ASME Paper 88-GT-51 (1988).

15. Irwin Stambler, "SCR Experience in U.S. Showing Better Than Expected Performance," Gas Turbine World, pp. 16-21 (March-April 1990).

16. Ecob, D. J. and Marriott, D. G., "Development of Steam Injection for Ruston TB 5000 and Tornado Engines," ASME Paper 89-GT-253 (1989).

17. McGreeham, W. F., Schtsch, M. J., "Flow Characteristics of Long Orifices with Rotation and Corner Radiusing," ASME Paper 87-GT-162 (1987).

18. Abuaf, N., Rasmussen, N. S., and Sazema, L. C., "Flow in Liner Holes for Countercurrent Combustion Systems," ASME Paper 88-GT-158 (1988).

19. Cameron, C. D., et al., "A Detailed Characterization of the Velocity and Thermal Fields in a Model Can Combustor with Wall Jet Injection," ASME Paper 88-GT-26 (1988).

20. Ericksen, V. L., Goldstein, R. J., "Heat Transfer and Film Cooling Following Normal Injection Through a Round Hole," ASME Paper 74- GT-6 (1974).

21. Carrote, J. F. and Stevens, S. J., "The Influence of Dilution Hole Geometry on Jet Mixing," ASME Paper 89-GT-292 (1989).

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