NV

DOE/NASA/0145-2
NASA CR-167876

Low NOx Heavy Fuel Combustor Concept Program Addendum

D.J. White, R.T. LeCren and A.J. Kubasco
Solar Turbines Incorporated
Subsidiary of Caterpillar Tractor Co.
P.O. Box 80966, San Diego, CA 92138

May 1982

Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Under Contract DEN3-145

for
U.S. DEPARTMENT OF ENERGY
Fossil Energy
Office of Coal Utilization

1. Report No.
NASA CR-167876

2. Government Accession No.

3. Recipient's Catalog No.

4. Title and Subtitle
LOW NOx HEAVY FUEL COMBUSTOR CONCEPT PROGRAM

5. Report Date

6. Performing Organization Code

7. Author(s)
David J. White
Alan J. Kubasco

8. Performing Organization Report No
SR82-R-4994-08

9. Performing Organization Name and Address
Solar Turbines Incorporated
2200 Pacific Highway
San Diego, California 92138

10. Work Unit No.

11. Contract or Grant No.
DEN3-145

13. Type of Report and Period Covered
Contractor Report

12. Sponsoring Agency Name and Address
Department of Energy and
National Aeronautics & Space Administration
Washington, DC

14. Sponsoring Agency Code
DOE/NASA/0145-2

15. Supplementary Notes
Project Manager Joseph Notardonato, NASA-Lewis Research Center, Cleveland, OH 44135

16. Abstract
Three simulated coal gas fuels based on hydrogen and carbon monoxide were tested during an experimental evaluation with a rich-lean can combustor: these were a simulated Winkler gas, Lurgi gas and Blue-Water gas. All three were simulated by mixing together the necessary pure component species, to levels typical of fuel gases produced from coal. The Lurgi gas was also evaluated with ammonia addition.

The emphasis of the program was placed on burning the fuels in a rich-lean mode. Only the Blue-Water gas, however, could be operated in such a fashion. This showed that the expected NOx signature form could be obtained, although the absolute values of NOx were above the 75 ppm goals for most operating conditions.

Lean combustion produced very low NOx well below 75 ppm with the Winkler and Lurgi gases. In addition, these low levels were not significantly impacted by changes in operating conditions.

17. Key Words (Suggested by Author(s))
Low NOx
Low Energy Fuels - Simulated Coal Gas
Rich/Lean Combustor
Gas Turbines

18. Distribution Statement
Unclassified - Unlimited
Star Category No. 44
DOE Category UC-90d

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
UNCLASSIFIED	UNCLASSIFIED	45	

* For sale by the National Technical Information Service, Springfield, Virginia 22161

TABLE OF CONTENTS

ORIGINAL PAGE IS
OF POOR QUALITY

LIST OF FIGURES

LIST OF FIGURES (Continued)

LIST OF TABLES

ORIGINAL PAGE IS
OF POOR QUALITY

1

SUMMARY

Two potential low NOx combustor configurations derived from units produced on an earlier program were evaluated experimentally for exhaust emisisons and mechanical integrity problems. One of the configurations proved to be unsuitable and was not utilized. Modifications could, however, be instituted to allow the unit to perform.

Three fuels were utilized during the experimental evaluations: these being a simulated Winkler gas, Lurgi gas and Blue Water gas. All three were simulated by mixing together the necessary pure component species, to levels typical of fuel gases produced from coal. The Lurgi gas was also evaluated with ammonia addition. This latter compound is a trace material likely to be produced during coal gasification.

The emphasis of the program was placed on burning the fuels in a rich-lean mode. Only the Blue Water gas, however, could be operated in such a fashion. This showed that the expected NOx signature form could be obtained, although the absolute values of NOx were above the goals for most operating conditions. The NOx levels were also found to be sensitive to inlet pressure and temperature levels.

Lean combustion produced very low NOx with the Winkler and Lurgi gases. In addition, these low levels were not significantly impacted by changes in operating conditions. Generally the emissions goals could be met at all conditions with these two fuels.

2

INTRODUCTION

Secondary fuels derived from coal, especially low and medium energy content gases, have taken on great importance as the petroleum and natural gas supplies of the United States of America diminish. These gases are particularly suitable as fuels for gas turbines, although they often include higher levels of nitrogen containing species than conventional gases such as natural gas or propane. Because of these higher concentrations of nitrogen compounds, the oxides of nitrogen (NOx) emission levels in gas turbine exhausts is usually much higher than if natural gas were burned.

In general NOx emissions are produced by two separate pathways, the first is purely thermal and involves the fixation of atmospheric nitrogen while the second takes place via a nitrogen containing free radical mechanism, the radicals being produced from those fuel molecules containing nitrogen. This latter mechanism, depending on the concentration of nitrogen in the fuel, can dominate the NOx emission emissions. It has been found that the most successful combustion concept that minimizes NOx from both thermal and fuel-nitrogen sources is a staged combustion approach. The primary zone of this staged combustion system is designed to operate in a fuel rich mode (at an equivalence ratio greater than one), while the secondary zone is designed to be sufficiently fuel lean to maintain the reaction temperatures below 1540C (2864°F). This latter temperature is maintained to ensure that minimal thermal NOx is produced. Above this temperature substantial thermal NOx can be produced by nitrogen fixation.

The goals of the work described herein were to obtain NOx emissions of 75 ppm corrected to 15 percent O_2 for fuel bound nitrogen levels up to one percent by weight and 37 ppm at 15 percent O_2 for the fuels with essentially no fuel bound nitrogen. These emission goals were to be attained without any sacrifice in engine efficiency. Thus, limits on combustion efficiency (99.0%), pressure drop (6%), and pattern factor (0.25) were imposed at all conditions including base load power and peak power conditions. Allowances for cycle efficiency and engine size were permitted as per the Federal Register (Ref. 1).

These goals and limits were to be met while operating at a set of conditions that simulated those that would be produced at the combustor by a nominal 12:1 pressure ratio industrial gas turbine. Table 1 shows the operating conditions estimated for a typical 12:1 pressure ratio engine that has been adopted for test purposes.

In addition to the above NOx goals, a secondary goal of providing near zero smoke levels was also adopted. Smoke is a severe problem with any gas turbine that utilizes heat exchange equipment in the exhaust, in that it causes foul-

ORIGINAL PAGE IS
OF POOR QUALITY

Table 1

Adopted Combustor Test Conditions

Combustor Inlet Conditions	Peak Load 118% Power	Engine Power Condition				
		Baseload	70% Power	50% Power	Spinning Idle	Cold Start
Pressure, P_{in} kPa	1310	1213	1055	910	303	103
Temperature In T_{in} C	376	361	334	308	143	Amb
Temperature Out T_{out} C	1057	982	882	810	546	649
Energy Fuel Ratio GJ/kg_{air}	220	198	175	154	140	178
Air Flow kg/s	103.7	100	90	81	33	6
Note: 100% air flow corresponds to 1.59 kg/s/(can-combustor)						

ing of the heat exchange surfaces. To ensure low smoke emissions, the lean secondary zone was designed to have an internal flow field that had particle retention properties. Essentially this flow-field consisted of a toroidal vortex (driven by a series of jets) that resembled a "smoke-ring" in its particle retention properties.

A novel approach that provides effective rich primary zone cooling has been proven successful. Upper limits on the inlet air temperature, however, must be imposed for any given design of the cooling system. The basic new contribution to combustor wall technology is the use of primary zone regenerative cooling. All the primary combustion air in this arrangement is first used to cool the primary zone walls. After cooling the walls the preheated air at temperatures in excess of 482C (900°F) passes into the combustor, mixing with the fuel in a short internal passage. By the time that the combustion reactions are initiated, the fuel is well mixed with the air. This system avoids external premixing of air and fuel with its attendant problems of autoignition and flashback. In addition it largely eliminates the normal high levels of carbon monoxide, unburned hydrocarbons, and smoke at low power conditions. Even at ambient light-off conditions combustion efficiencies in excess of 99 percent are obtained. This latter effect is due primarily to the significant increase in the effective inlet air temperature.

To ensure that the combustor provides low NOx emissions over the entire range of engine operation, some form of variable geometry will probably have to be incorporated. Using simple valves presently in use on commercial engines, it should be possible to effect a change in the flow split between the secondary and primary zones. Although potentially viable, this has proven to be difficult to implement due to mechanical integrity problems. These problems consist of severe overheating of the secondary zone forward dome and of the transition piece adjacent to the dome.

3

COMBUSTOR DESIGN

3.1 PROGRAM GOALS

The primary goal of this program was to design, develop and demonstate a combustor concept or concepts that had exhaust emission levels below the values shown in Table 2, while burning fuels with high fuel-bound nitrogen levels. In addition, a second goal was the attainment of the performance specification shown in the lower part of Table 2, while meeting the above emission goal.

A modification of the above goals was also included. NOx emission levels one half those shown in Table 2 would have to be obtained while burning distillate fuels.

All of the above goals were to be met while operating at a set of conditions that simulate those that would be produced at the combustor by a nominal 12:1 pressure ratio industrial gas turbine. Table 3 shows the operating conditions generated for a typical industrial engine that were adopted for test purposes.

The purpose behind these goals was to produce the technology and design data necessary to provide, at the end of the program, a design of a prototype engine combustor capable of burning low and medium Btu coal gases. This recommended combustor would, in essence, be the proposed configuration for Phase II of the overall program.

3.2 DESIGN PHILOSOPHY

The combustor adopted as the prime candidate for evaluation of the low and medium energy content gases was the rich-lean combustor as developed, on the earlier program (Ref. 2), (see NASA CR-165481; DOE/NASA 0145-1, for details). The combustor as developed is shown mounted in the rig casing in Figure 1. Although this latter figure is schematic in form it does display all the salient features of the combustor. In this particular combustor all the primary air and part of the secondary air first enters into an annular cooling passage surrounding the primary zone proper. The primary air flows forward toward the dome, cooling the main part of the primary zone and then enters the primary zone via a radial inflow swirler. The secondary air that enters with the primary air, separates from the latter and flows rearward cooling the rear conical portion of the primary zone. This secondary air enters the combustor via holes in the throat of the transition piece between the primary and secondary zones. The remainder of the secondary air enters

ORIGINAL PAGE IS
OF POOR QUALITY

Table 2

Emissions and Performance Goals

	MAXIMUM LEVEL	OPERATING CONDITION
DESIGN EMISSION GOALS		
POLLUTANT		
Oxides of Nitrogen	75ppm at 15% O_2	All
Sulfur Dioxide	150 ppm at 15% O_2	All
Smoke	20 S.A.E. Number	All
DESIGN PERFORMANCE GOALS		
Combustion Efficiency	99% at all Operating Conditions	
Total Pressure Loss	6% at Base Load Power	
Outlet Temperature Pattern Factor	0.25 at Base Load and Peak Load Power	
Combustor Exit Temperature Profile*	Peak at 70% Span	

Table 3

Adopted Combustor Test Conditions

Combustor Inlet Conditions	Peak Load 118% Power	Engine Power Condition				
		Baseload	70% Power	50% Power	Spinning Idle	Cold Start
Pressure, P_{in} kPa	1310	1213	1055	910	303	103
Temperature In T_{in} C	376	361	334	308	143	Amb
Temperature Out T_{out} C	1057	982	882	810	546	649
Energy Fuel Ratio GJ/kg	220	198	175	154	140	178
Air Flow kg/s	103.7	100	90	81	33	6

Note: 100% air flow corresponds to 1.59 kg/s/(can-combustor)

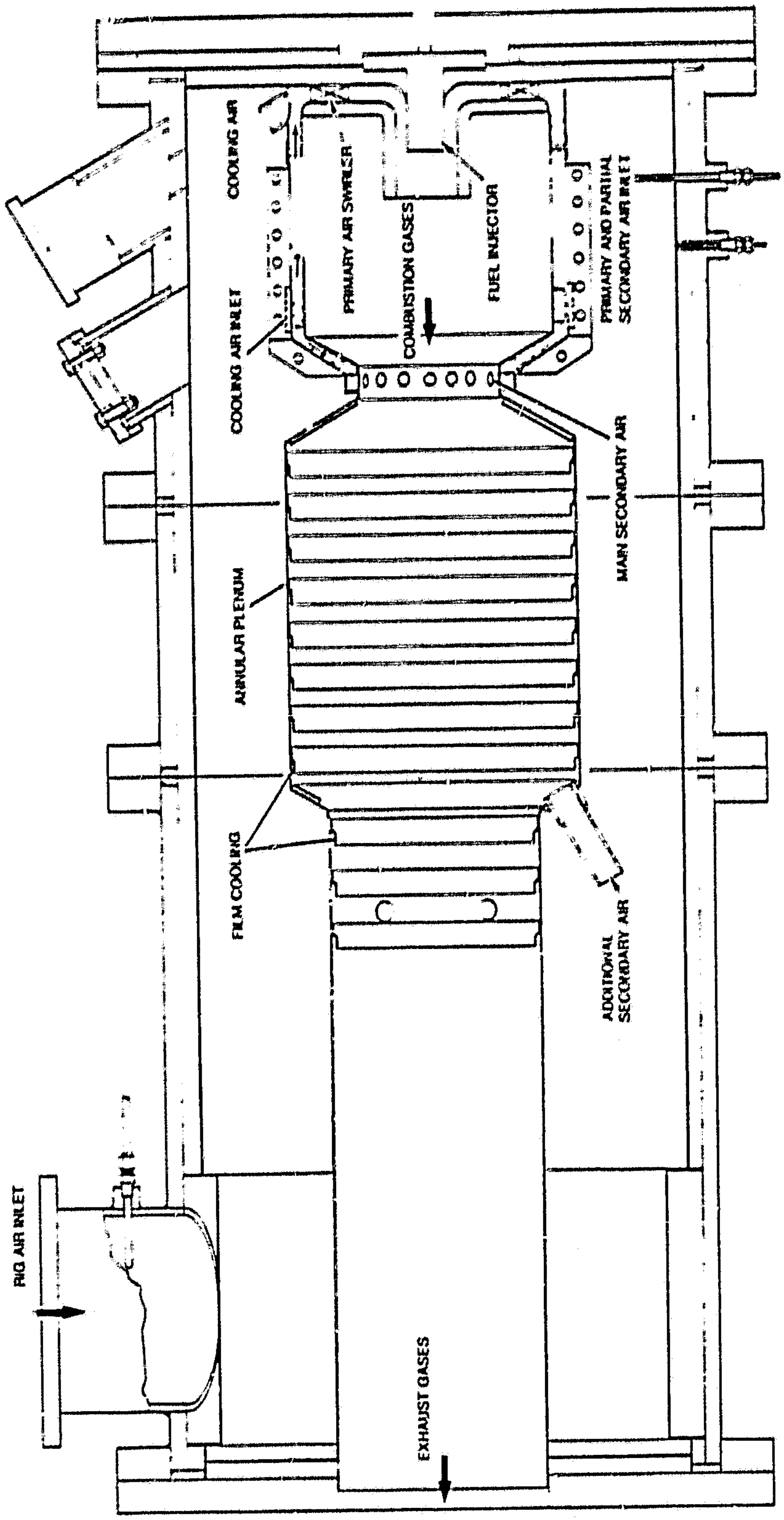

Figure 1. Rich-Lean Combustor

through ports at the rear of the secondary zone. These ports are angled forward and the jets of air produced inside the secondary zone merge at the centerline. Two resultant jets are produced from the interaction of these secondary air jets, one flowing on the axis toward the transition piece and one flowing toward the exit of the secondary zone. The jet flowing toward the transition exit contains the major portion of the secondary air mass flow. In addition, the momentum of this major derived jet (which decreases rapidly to approximately 60 percent of the initial jet momentum) is arranged to balance the momentum of the gases exiting from the transition piece within the secondary zone proper. This later effect is utilized to ensure that the transition through stoichiometric takes place within the secondary zone. Opposed jet on jet mixing is also one of the more effective methods of mixing two fluids rapidly. Rapid mixing is required to ensure that the time period spent by the reacting gases at stoichiometric is minimzed. This minimization of the residence time at stoichiometric in turn minimzes the thermal NOx production.

The rich-lean combustor stoichiometry or air flow splits were designed for typical petroleum and coal-based liquid fuels having stoichiometric fuel-air ratios in the range of 0.067 to 0.068. Because of the cost of either introducing variable geometry or of modifying the combustor to emit the stoichiometry of the low and medium energy content coal gases it was decided to leave the geometry as developed for liquid fuels. This latter geometry allowed only the medium energy content fuel to be burned in the primary zone in a rich mode. The two low energy content gases effectively were burned in a lean-lean mode of operation, even though the combustor would commonly be referred to as a rich-lean system. This can be appreciated better by recognizing that the simulated Winkler gas 4,097,811 J/m^3 (110 Btu/scf) and the simulated Lurgi gas 6,183,968 J/m^3 (166 Btu/scf) have stoichiometric fuel-air ratios by weight of 0.55 and 1.04 respectively. When these ratios are compared to the design primary zone equivalence ratio design point range of 1.2 to 1.4 (based on an average stoichiometric fuel-air ratio of 0.0675) it is obvious that even allowing for the reductions in heating value compared with conventional fuels, that neither could be operated in a rich mode. The "Blue-Water" gas on the other hand, could and was operated in a rich-lean fashion due to the relatively low value of its stoichiometric ratio (0.248) and its high lower heating value 10,207,272 J/m^3 (274 Btu/scf).

It was originally thought that rich primary zone operation could be achieved by adding air in the dilution section when operating with the two low energy content gases. When this was attempted it was found that lean combustion occurred in the dilution section, causing excessive local wall temperatures. The reduction in mechanical integrity that occurred proved to be unacceptable, thus the combustor was operated in the as-developed configuration.

3.2.1 Fuel Injection (For the Rich-Lean Combustor)

The original air-assist liquid fuel injector was capable of being modified to handle the low and medium energy content gases at the low flow (low power) conditions. This injector was modified so that the fuel gas was injected into the combustor via the air-assist passages, while the liquid-fuel injection ports were blocked. Because the air-assist passagss were not sufficiently large to allow the maximum gas fuel flow, a new fuel-injector was designed and built. To ensure the maximum of fuel flexibility the injector was designed from the onset to handle low, medium and high energy content gases together with most liquid fuels. This injector is shown in Figure 2. In the low energy content gas combustion mode, gas would be injected via both the central passage (air assist in liqid fuel operation) and the annular passage immediately surrounding it. When operating with medium and high energy content gases either the central passage or the surrounding annular passage could be used depending on whether rich or lean combustion is desired. For lean combustion the annular passage would be utilized for gas injection while air would be introduced through the central passage which has a swirler to ensure rapid mixing of gas and air. For rich combustion with high energy content gases, the central passage could be used. Liquid fuels would be injected through the outermost series of passages and holes, while air would be introduced through the center passage with a swirling motion. This latter swirling air is required during liquid fuel injection to ensure stable film formation and subsequent fine atomization from the sharp edge of the injector.

For dual-fuel (liquid/gas) operation, liquid fuel would be injected as described above while the medium or high energy content gas could be injected concurrently through the annular passage surrounding the central air-assist supply tube. In the case of dual fuel (liquid/gas) combustion where the gas is a low energy content fuel a more complex mode of operation would be required. As outlined above, the low energy content gas would be injected through both the central air-assist tube and the surrounding annular passage. During the change to liquid fuel operation the gas flow in the central tube would be turned down as the liquid fuel was introduced and air introduced as a replacement fluid. Only after air had been substituted for all the low energy gas in the central tube, would the gas flow in the surrounding annular passage be reduced. This staged action would ensure that the liquid fuel film would remain in a stable condition.

A detailed description of the design of each of the component parts of the rich-lean combustor is given in NASA CR-165481 (Ref. 2) and this report should be used if further information is needed.

3.3 LEAN PREMIXED COMBUSTOR

The second approach proposed for investigation was a lean premixed combustor, utilizing external mixing ports or ducts. In general, the configuration of the adopted lean combustor is referred to as the "JIC" or jet induced circulation combustor. Flame stabilization in this combustor occurs in the toroidal vortex driven by the forward flowing angled jets. This vortex has properties resembling those of a "smoke-ring", and this ensures that any liquid fuel droplets not vaporized externally in the mixing ducts are retained in the primary zone until they are consumed. In the case of gas combustion this stabilization method is not absolutely necesary. It does, however,

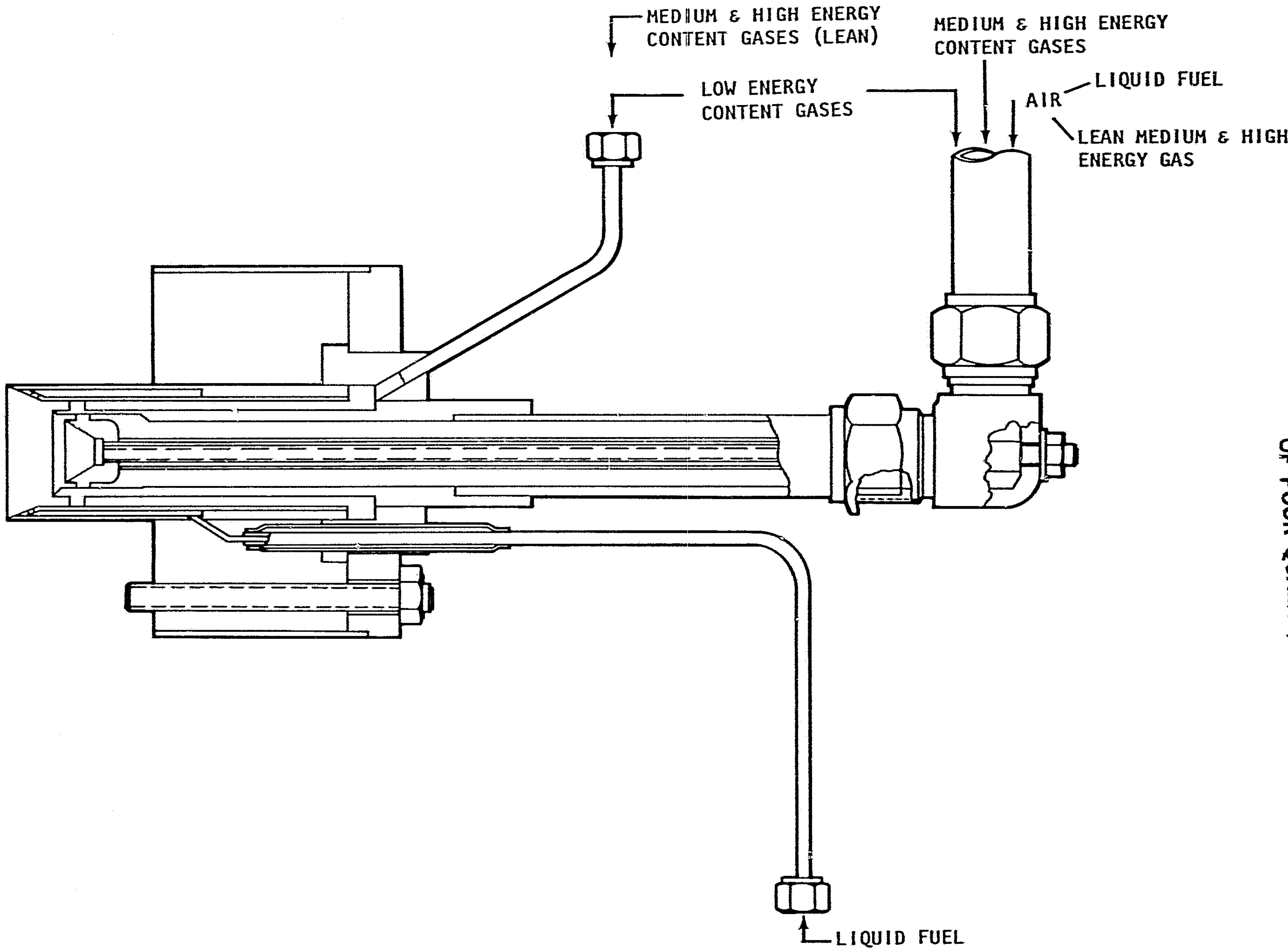

Figure 2. Advanced Gas-Liquid Fuel Injector

provide a means of ensuring that any carbon particulates produced are retained and consumed before the associated combustion gases can exit.

This particular combustor had been utilized extensively in past work to provide low oxides of nitrogen and low smoke, while burning a wide variety of fuels including high and reduced energy content gases.

A detailed description of both the lean premixed combustor system and the rich-lean system is provided in Reference 2.

4

FUELS AND FUEL SYSTEM

4.1 TEST FUELS

Three different simulated fuel gases that can be obtained from coal were chosen for investigation. These covered the range of energy contents in terms of lower heating value that lay between 110 and 273 Btu/ft^3. Table 4 shows a wide range of gaseous fuel compositions that could be produced from coal through reaction with air or oxygen and steam in various combinations. The three that were chosen for testing were the Winkler (air blown), Lurgi (air blown) and Blue Water gases. Each of these fuels were simulated by mixing the appropriate pure compounds together in a five component on-line mixer. Typically these included hydrogen (H_2), carbon monoxide (CO), methane (CH_4), carbon dioxide (CO_2), and nitrogen (N_2). The nominal compositions adopted for each of the gases is provided in Tables 5, 6 and 7.

In most cases the experimentally reported compositions of these gases did not include either steam or hydrogen sulfide. The Lurgi gas did, however, have as part of its composition, a 0.25 percent level of steam and 0.6 percent of hydrogen sulfide and these were ignored in simulating the gas.

4.2 FUEL SYSTEM

All of the component gases needed with the exception of the carbon monoxide were obtained from existing facility sources. The carbon monoxide was purchased and delivered in tube-trailer lots. In operation the tube-trailer gas manifold was connected to a mating tube fitting mounted on a stanchion that also anchored the trailer. A schematic of the carbon monoxide fuel system as developed on the program is shown in Figure 3. Soft copper tubing and silver soldered brass-fittings were used throughout to ensure minimal leakage. Multiple remotely controlled regulator valves were installed to provide fine control of the flow rates. Fast acting shutoff valves were also installed as a safety measure. Hydrogen was also delivered by tube-trailer, but in this case the gas was transferred to an existing multiple tube fixed facility when delivered. As in the case of the carbon monoxide, undergound heavy wall soft copper pipe was used to convey the hydrogen to the test cell. All fittings were of brass or copper and were either brazed or silver-soldered together. Multiple pressure regulator valves were also used to ensure precise flow control. See Figure 4 for a schematic of the line.

Existing facility sources of methane (CH_4), carbon dioxide, and nitrogen were utilized. Each of these facility sources were "piped" to the test cell using

Table 4

Gaseous Fuels Derived From Coal

Class	Low Btu Gas							Medium Btu Gas				
Process	Producer Gas From Coal	Blast Furnace Gas	Lurgi	Lurgi	Fluidized Bed Coal Gas	Bureau of Mines BOM 7644	Winkler #1	Blue Water Gas	Coal Gas (Vertical Retor w/Steaming)	Winkler #2	Lurgi	Koppers Totzek
Air												
Oxygen												
Steam												
Analysis (% of Vol)												
O_2	—	—	—	—	—	—	—	—	0.4	—	—	—
N_2	52.4	60	40.26	30.2	59.4	54.5	55.3	4.5	6.2	1	0.32	1.13
CO_2	4	11	14.18	10.7	0.5	7.2	10	4.7	4	19	28.87	6.21
CO	29	27	16.71	10.7	31.8	2.0	22	41	18	38	19.52	55.11
H_2	12	2	22.98	15.7	15.6	15.5	12	49	49.4	40	38.80	37.18
CH_4	2.6	—	4.97	4.4	0.5	2.8	0.7	0.8	20	2	11.20	—
C_2H_6	—	—	—	—	—	—	—	—	2	—	—	—
C_3H_8	—	—	—	—	—	—	—	—	—	—	—	—
C_4H_{10}	—	—	—	—	—	—	—	—	—	—	—	—
C_5H_{12}	—	—	—	—	—	—	—	—	—	—	—	—
H_2O	—	—	0.257	27.8	0.5	0	—	—	—	—	0.26	0.37
H_2S	—	—	0.63	0.5	0.7	0	—	—	—	—	1.8	0
Mol Wt (kg/k mol)	25.2	29.25	29.08	22.4	23.97	24.8	26.4	15.9	13.7	20.4	27.94	30.11
LHV												
(Btu/scf)	150	92.3	166	121	154	132	110	274	422	250	277	279
(Btu/lb_m)	2250	1194	2646	2041	2429	2021	1578	6504	11,626	4641	4058	5476
(ΔT) stoich (°C)	2871	2205	1986	2312	3007	2638	2438	3769	3601	3501	2057	2588
(F/A) stoich												
(by wt)	0.711	1.462	0.5519	0.712	0.674	0.770	1.049	0.248	0.120	0.344	0.3486	0.299
(by vol)	0.817	1.448	0.6684	0.923	0.815	0.899	1.141	0.450	0.252	0.488	0.3871	0.445
Fuel Flow Mass Ratio	9.56	18.0	8.13	10.5	8.85	10.6	13.6	3.31	1.85	4.63	5.30	3.93
Wobbe Index	161	91.8	166	138	169	143	115	370	614	298	282	274
Fuel Flow Volume Ratio	5.88	10.3	5.70	6.86	5.60	6.62	8.23	2.50	1.54	3.17	3.35	3.45
But/lb_{air} stoich	1600	1746	1460	1453	1637	1556	1641	1613	1395	1597	1415	1637
Limits of Flammability												
Lower I_{25}	17.38	37.59	12.6	18.0	15.05	16.6	20.5	6.38	5.6	7.49	7.50	7.56
Higher U_{25}	66	73.06	51.8	60.57	69.80	64.95	71.17	69.97	38.8	66.7	45.89	72.6
Ratio	3.80	1.94	4.11	3.36	4.64	3.91	3.47	10.96	6.91	8.92	6.12	9.60
H_2/CO (by vol)	0.414	0.074	1.375	1.467	7.75	7.75	0.545	1.195	2.744	1.053	1.988	0.675

Table 5

Air-Blown Lurgi Gas

Component	Composition (%) Volume
Nitrogen	N_2 - 40.5%
Carbon Dioxide	CO_2 - 14.5%
Carbon Monoxide	CO - 17.0%
Hydrogen	H_2 - 23.0%
Methane	CH_4 - 5.0%
Note: Water and hydrogen sulfide have been ignored	
Lower heating value 166 Btu/ft^3	

Table 6

Air-Blown Winkler

Component	Composition (%) Volume
Nitrogen	N_2 - 55.5%
Carbon Dioxide	CO_2 - 10.0%
Carbon Monoxide	CO - 22.0%
Hydrogen	H_2 - 12.0%
Methane	CH_4 - 0.5%
Lower heating value 110 Btu/ft^3	

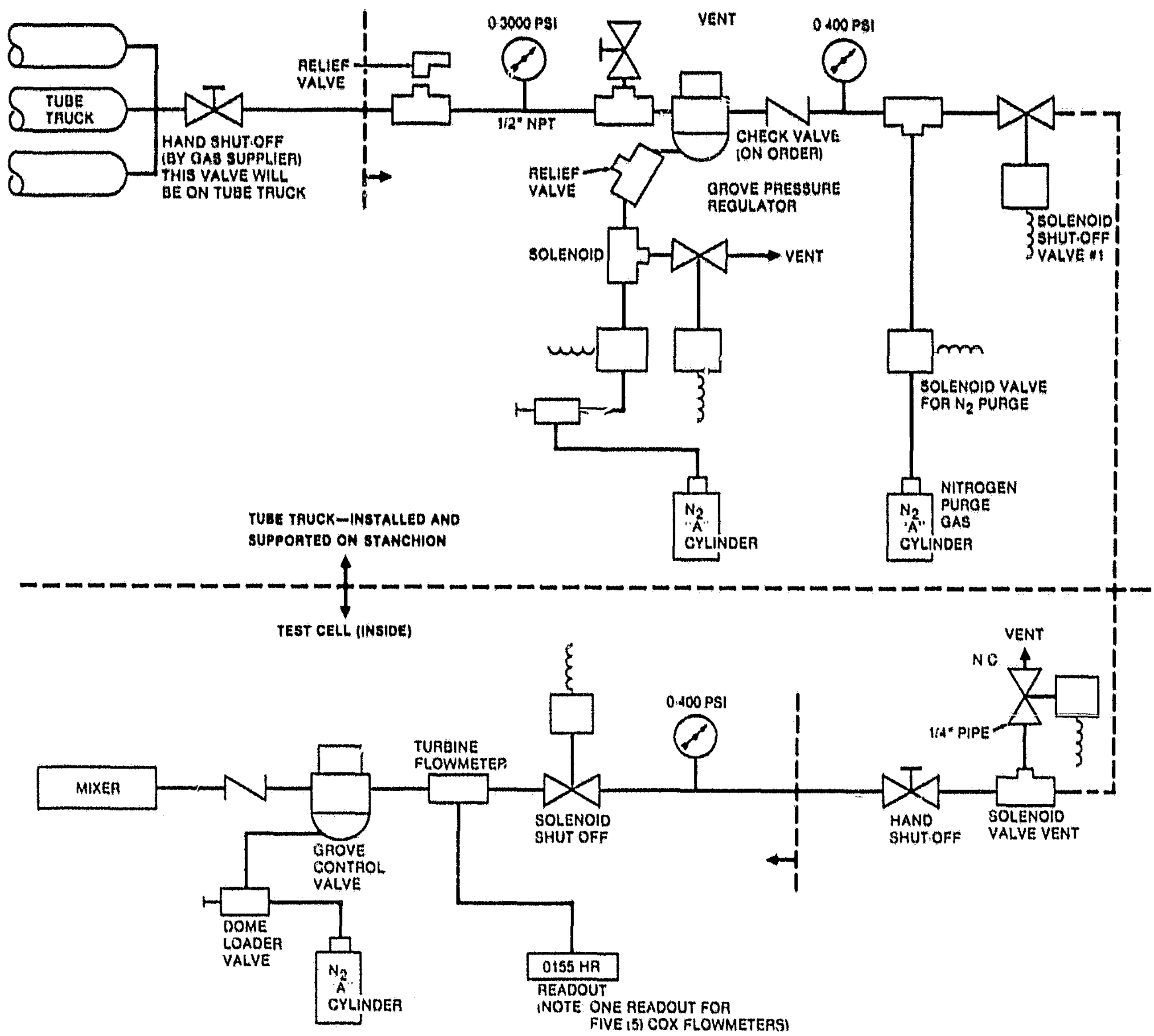

Figure 3. Schematic of the Carbon Monoxide Supply System

copper tubing as in the case of hydrogen and carbon monoxide. Fuel line schematics for these three gases are shown in Figure 5, together with the mixer. Entry points for hydrogen and carbon monoxide are shown.

Each of the gases was fed into the on-line mixer or blender (see Fig. 6). This mixed the gases to a level of plus or minus one percent of the minor component. To control the total mixed fuel gas mass flow entering the combustor, each of the individual gas flows were reduced or increased together as necesary, for the range near the maximum flow requirements. Combustion at low flow conditions was obtained by bleeding a portion of the fuel gas flow to a flare which burned the gas in ambient air. In typical operation

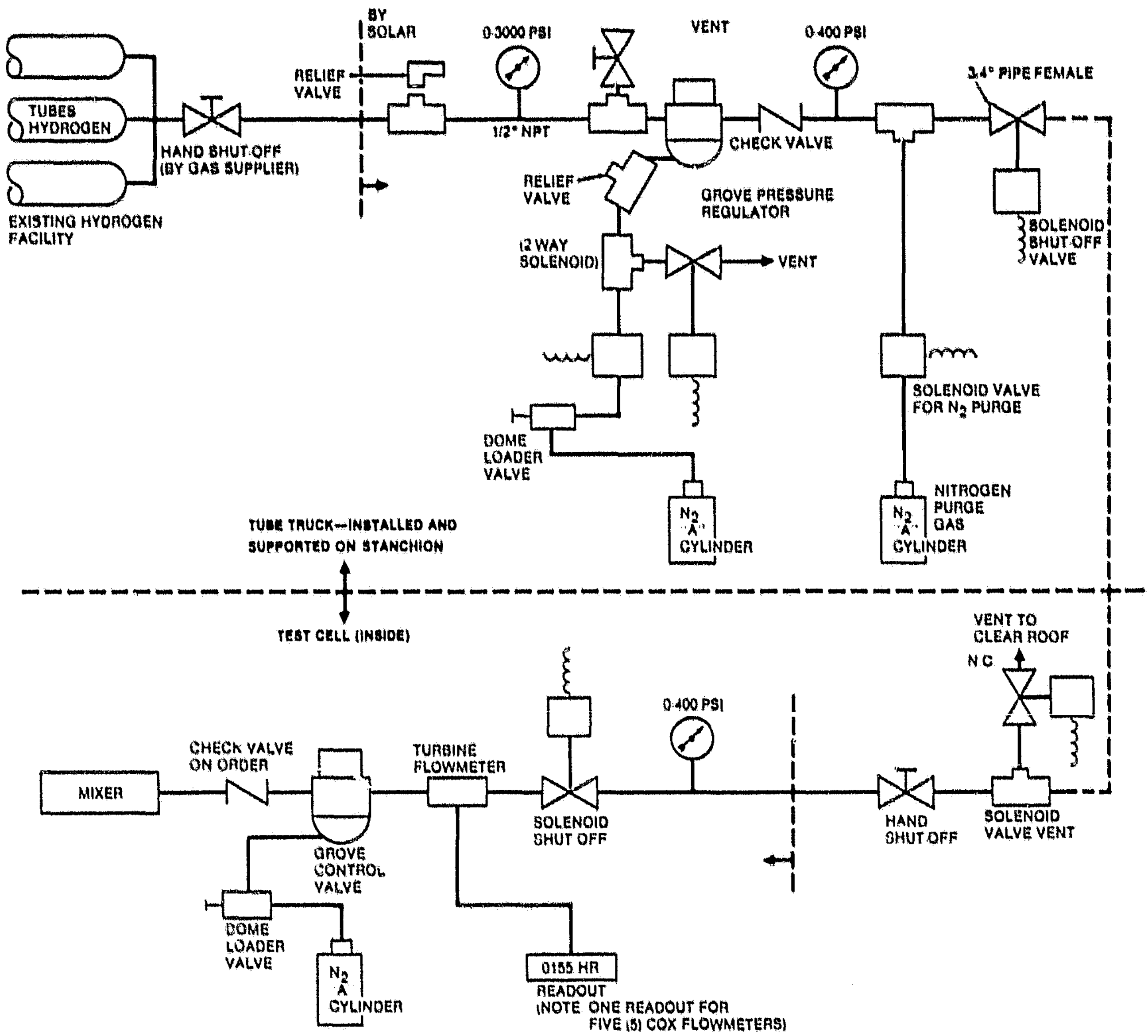

Figure 4. Schematic of the Hydrogen Supply System

all five gases would be set to the flow conditions corresponding to the maximum fuel-air ratio of the combustor at the simulated power-point.

The gas flows would then be reduced together in a series of decrements to lower the fuel/air ratio and thus obtain emission data as a function of fuel/air ratio. At some point the specie with the lowest concentration reaches a flow detection or measurement limit. At this point, part of the flow leaving the mixer is diverted to the flare. By adjusting the flow to the flare the desired range of combustion fuel/air ratios could be obtained.

ORIGINAL PAGE IS
OF POOR QUALITY

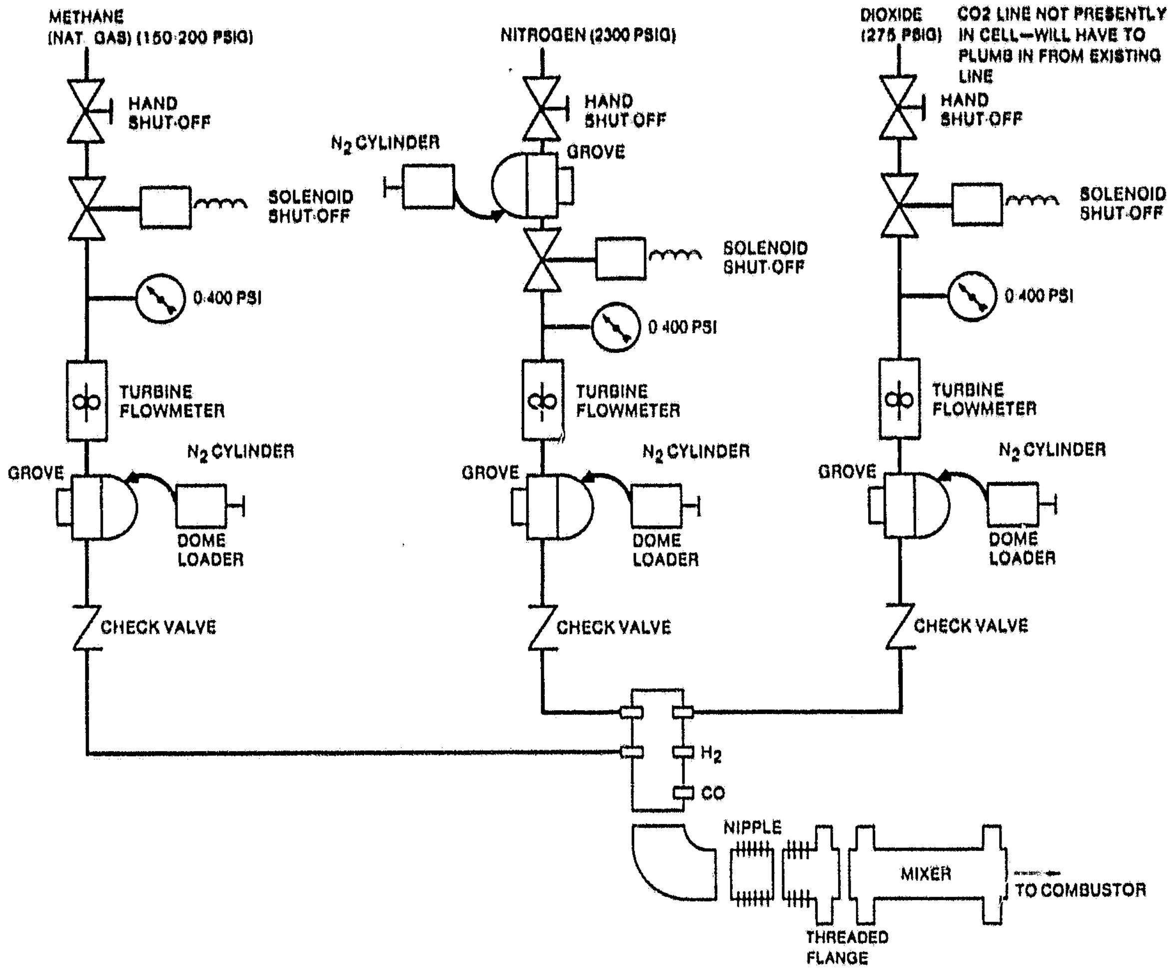

Figure 5. Generalized Piping Diagram

To ensure the complete combustion of a wide range of low and medium energy content gases, each having widely ranging flow rates, the flare was equipped with a permanent natural gas fueled pilot. This latter pilot flame was arranged so as to entrain the gases to be burned, before entraining the necessary air, for combustion.

Table 7

Blue-Water Gas

Component	Composition (%) Volume
Nitrogen	N_2 - 4.5%
Carbon Dioxide	CO_2 - 4.5%
Carbon Monoxide	CO - 41.0%
Hydrogen	H_2 - 49.0%
Methane	CH_4 - 1.5%
Lower heating value 274 Btu/ft^3	

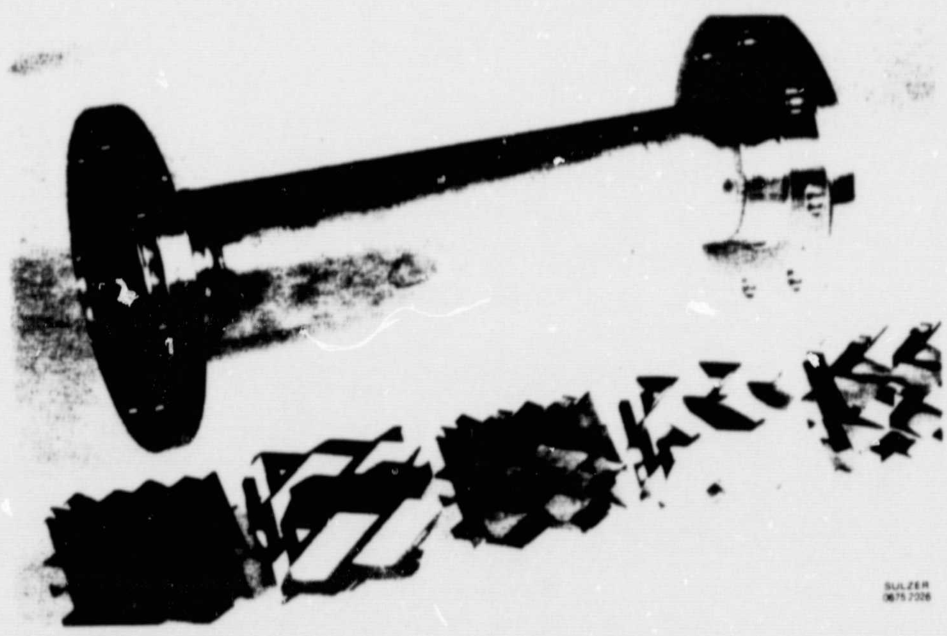

Figure 6. Stainless Steel Gas Mixer

5
EXPERIMENTAL APPARATUS

5.1 EXPERIMENTAL COMBUSTOR HARDWARE

The arrangement of the rich-lean combustion system used in the experimental evaluation has been shown earlier in Figure 1. A photograph of the test rig is shown in Figure 7. The combustor is mounted in a casing in a reverse flow configuration. The forward end of the combustor is rigidly attached to a mounting plate which bolts to one end of the rig casing. The rear end of the combustor is supported by a slip joint which accommodates axial movement induced by thermal expansion of the combustor. All combustor instrumentation is routed through the combustor mounting plate via removable instrumentation ports. This allows the combustor to be removed from the mounting plate without removing the instrumentation. Figure 8 shows the rich-lean combustor with instrumentation attached. The removable instrumentation ports can be seen at the top of the combustor.

Each combustor was instrumented with chromel/alumel (Type K) thermocouples and static pressure taps to measure: (1) liner skin temperatures; (2) air temperature, pressure and pressure drop across the primary air swirler; and (3) combustor pressure loss from the combustor inlet to the combustor throat. The skin thermocouples were tack welded to the skin in an open junction fashion and then covered with Inconel foil which was also tack welded to the combustor.

The 1/16 inch diameter thermocouple leads were strapped to the combustor with Inconel foil and loops were provided for thermal expansion. Static pressure lines from the combustor throat were similarly routed.

During operation all skin temperatures were continuously monitored to avoid damage to the combustor liner. Combustor pressure drops were used to indicate any mechanical failures. The primary swirler air temperature, pressure and pressure drop were also used to calculate primary air flow.

The combustor rig case was a high pressure pipe section made of mild steel. This was insulated on the inside with ceramic fiber, held in place by a thin sheet of stainless steel (314).

The inlet to the casing was a six-inch diameter stainless steel pipe positioned at right angles to the casing. Located in this inlet section were six static pressure taps, six exposed junction chromel/alumel (Type K) thermocouples (1/8 in. diameter), and six Kiel type total pressure probes. Each of the probes and thermocouples were located at the center of a series of equal areas. This allowed a weighted average of each of these measurements to be obtained. Located upstream of the inlet was an ASME standard sharp-edged

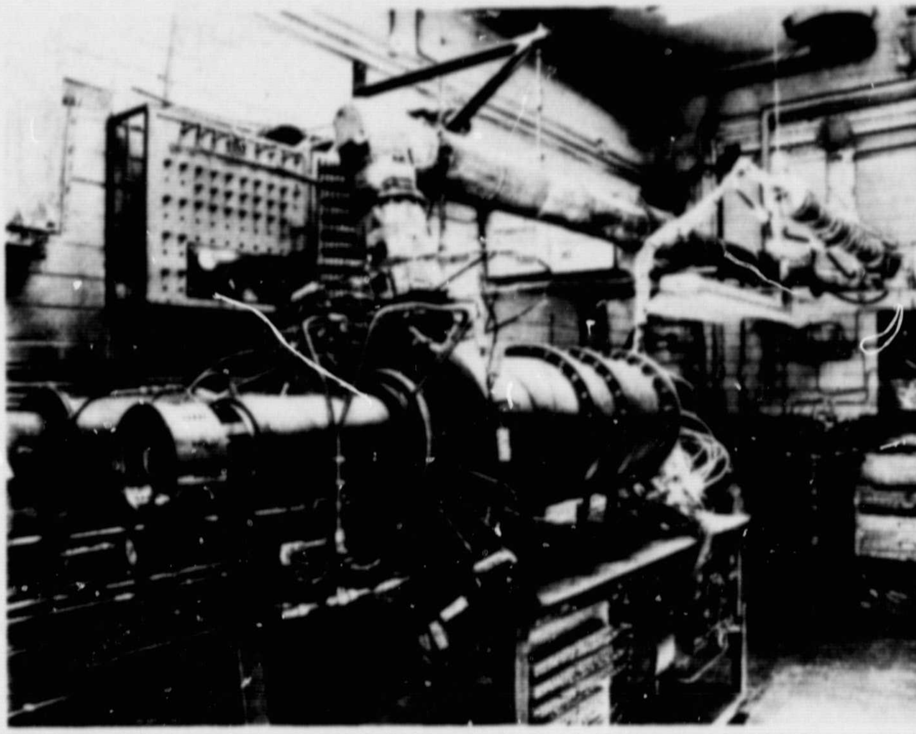

Figure 7. Combustor Test Rig

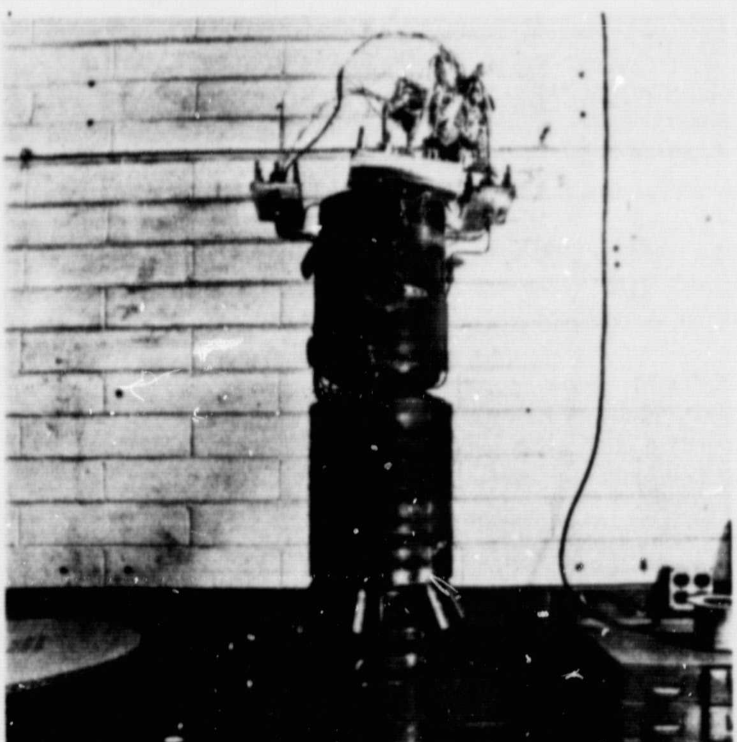

Figure 8.
Rich-Lean Combustor

ORIGINAL PAGE
BLACK AND WHITE PHOTOGRAPH

orifice mass flow measuring device. This utilized the normal upstream diameter tap and downstream half diameter tap system. Air was supplied to this six-inch diameter orifice run by an eight-inch pipe which brought indirectly heated high pressure air into the test cell. The maximum flow, pressure, and temperature conditions were 1.59 kg/s (3.5 lb/s), 1213 kPa (176 psia) and 454C (850°F) for this particular air flow.

A second air supply system was piped into the cell for the air assist fuel injector. This system provided 2750 kPa (400 psia) air at room temperature and low flow rates of the order of 0.04 kg/s (0.2 lb/s). Control of this flow and pressure was obtained through the use of multiple regulators.

Ignition of the main combustor was accomplished using a spark ignited natural gas torch which was mounted on the rig casing. A flame from this torch entered the primary air swirler and ignited the fuel in the primary combustion zone. The torch natural gas flow rate was measured using a turbine-type flow meter and torch ignition was verified by observing the temperature at the torch exit, using a Type K thermocouple.

At the exit of the combustor the exhaust gases passed through a water-cooled instrumentation ring. This ring contained emissions sampling probes which allowed exhaust gases to be drawn from many points in the gas flow area to obtain an average sample. Figure 9 shows these probes in the instrumentation ring. These gases then flowed through a heated line to an emissions analyzer. Also in the instrumentation ring were 12 exposed junction 1/8 inch Type K thermocouples located at the centers of equal areas in the exhaust flow stream. These are not shown in Figure 9.

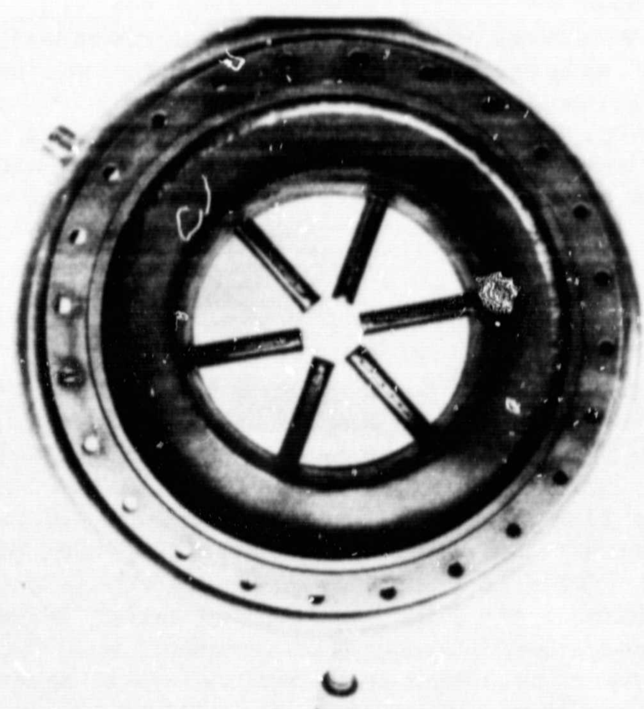

Figure 9. Instrumentation Rig

The cooling water from the instrumentation ring was dumped into the exhaust gas flow downstream of the emissions probes. This water served to cool the exhaust gases before they reached the butterfly valve used to control back pressure and air flow through the rig. After passing the back pressure valve, the exhaust gases flowed through a silencer and then rose through an exhaust stack and exited to the atmosphere.

The emissions analyzer contained equipment for measuring unburned hydrocarbons, carbon monoxide, carbon dioxide, nitrous oxides, oxygen and smoke. Unburned hydrocarbons were measured continuously using a Beckman Model 402 Flame Ionization Detector (high temperauture). Carbon monoxide and carbon dioxide were measured by the nondispersive infrared method using a Beckman Model 315 Dual Stacked Cell Infrared Analyzer. Oxides of nitrogen were determined by the chemiluminescence method using a Thermo Electron Corporation Chemiluminescent Analyzer Model 10A. Oxygen was determined by measuring an electrical current developed by an amperometric sensor in contact with the sample. This sensor was electrically connected by a multi-conductor shielded cable to a Beckman Model 742 Oxygen Analyzer. Smoke was measured on a continuous basis using the Von Brand method. When smoke was detected the standard ASME (ROSECO) smoke analysis system was brought on-line to provide a detailed definition of the levels.

Testing was conducted from a control room separated from the actual rig. A window allowed visual inspection of the rig during testing. A view of the flame was provided by using a mirror inside the cell to look into a quartz window located in the back end of the test rig. This window consisted of two 2-1/2 inch diameter quartz glass lenses. The cavity between these was pressurized with nitrogen to prevent leakage from the rig. Figure 10 shows this quartz window. Pressures were observed on a combination of mechanical gauges and both water and mercury manometers. Temperatures were monitored on both analog and digital meters and on a CRT output provided by a data acquisition system. Air, natural gas and liquid fuel flows were observed on digital panel meters and were controlled entirely from within the control room. Figure 11 shows the control room.

5.2 TEST PROCEDURES

The following procedure was used to conduct the testing: First the rig was heated by flowing air through with no combustion occurring in the test combustor. The air was heated to the desired inlet test temperature by an indirect-fired heat exchanger. Once the desired inlet test temperature was reached, the test combustor was lighted by the means of a torch ignitor at low airflow and near ambient pressure. The airflow rate and inlet test pressure were controlled by a system of valves in the inlet plumbing and, a butterfly type back pressure valve downstream of the rig. For any one series of tests the airflow, inlet pressure, and temperature were held constant and the fuel flow rate varied. Data points were selected in order to allow determination of the emission signature of the test combustor at the particular operating condition and on the particular test fuel. The data consisted of basically three groups. First combustor skin, fuel and air temperatures were continually monitored and then recorded on printed paper tape. Second, the rig operating

ORIGINAL PAGE IS
OF POOR QUALITY

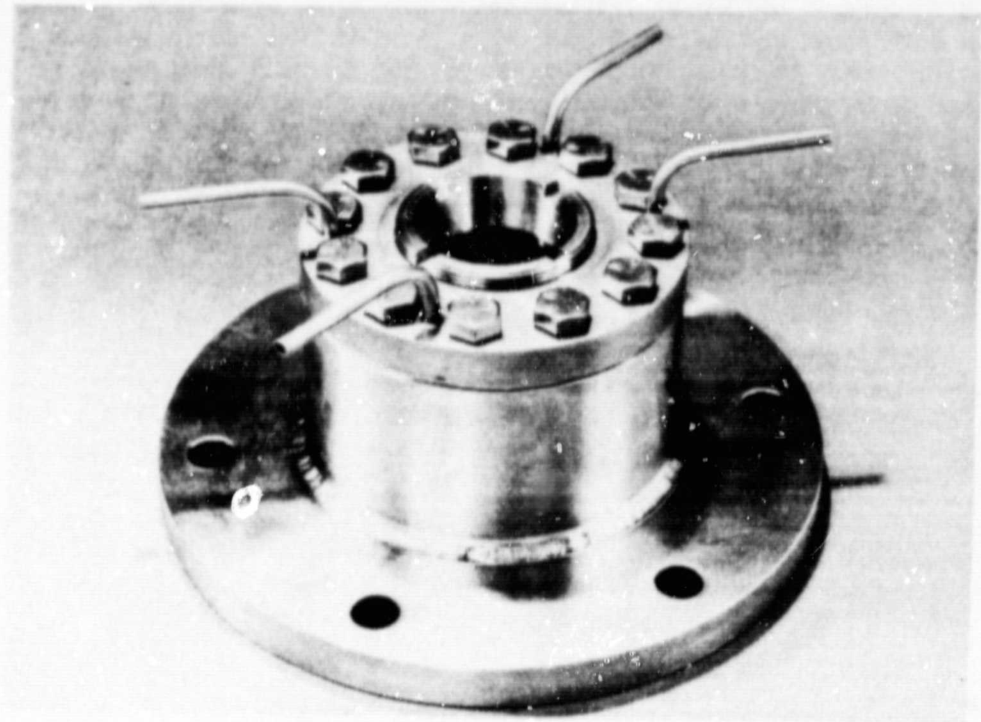

Figure 10. Quartz Window

Figure 11. Control Panel

data were recorded by hand. These data included pressures, pressure drops, flowrates and some additional temperatures. Third, the emissions data were monitored on both strip chart recorders and digital meters and recorded by hand. When one series of tests was completed and another begun, the inlet air temperature was first varied by changing the preheater setting and then the new airflow pressure condition achieved by manipulating the inlet control and backpressure valves. Shutdown consisted of extinguishing the flame in the test combustor, turning off the preheater, cooling the rig by continuing to flow air through it.

6

TEST RESULTS AND DISCUSSION

6.1 GENERAL

The combustor utilized for the majority of the tests was the developed rich-lean combustor as used in Reference 2. Emission signatures for this particular combustor have been determined over the full range of simulated engine conditions (see Table 1). Specifically combustor exhaust emission levels of NOx, CO, UHC, smoke and CO_2 were determined for each of the engine test points, with each of the required fuels. The three fuels were treated approximately equally in terms of the test hours associated with each of them. Ammonia was added to one of the gases and certain test points were retested, to provide a comparative NOx emission level. It was found that the generated ammonia pressure could not be increased sufficiently to allow operation at the maximum power condition. To obtain high ammonia pressures the cylinders containing liquid ammonia were immersed in an electrically heated water bath. The pressure obtained at 50C (120°F) (the maximum safe operating temperature of the cylinders) was of the order of 1430 kPa (280 psia). Although this pressure should have been sufficient to inject and mix the ammonia into the fuel gas, an unforeseen phenomenon occurred that reduced the pressure substantially. It was found that at high pressures ammonia reacted with the seal materials used in the flow measurement system, even though these materials were recommended for ammonia operation. At normal low pressure conditions the reaction rates between ammonia and the polymeric seal materials apparently are very low, however, at high pressures these reaction rates increase dramatically. The products of the reaction quickly blocked the lines and prevented ingress of the ammonia into the gas mixer. Redesign of the ammonia system was thus required, however, because of limitations imposed by the schedule, this could not be accomplished, and operation at lower presures was all that could be achieved.

6.2 FUELS

Three fuels all based on CO/H_2 mixtures, were chosen for experimental purposes and these are described in detail in Section 4. Two of the fuels were low energy content gases and a third was of medium energy content. The lowest energy content gas chosen was that produced by the combined air and steam blown Winkler gasifier. This was a gas that had a lower heating value of 4,097,811 J/m^2 (110 Btu/scf) making it difficult to burn although it typified gases produced by existing technology. Integrated gas turbine and gasification units could readily use the Winkler (simple fluidized bed) technology which was one of the reasons for choosing it. The integration may require, however, a pressurized fluidized bed rather than the true Winkler system which is atmospheric. It is believed however, that the gas compositions produced would not vary significantly with pressure. The second gas was that typically provided by an air blown (fixed bed) Lurgi system, and had a lower heating value of 6,183,468 J/m^2 (166 Btu/scf). This system also has the potential of being integrated with a gas turbine.

Essentially the low energy content gases derived from coal would have to be utilized at the source, because the costs of transportation quickly exceed the energy value delivered. Generally a supply radius of the order of 50 miles is considered to be the limit. Thus it is anticipated that low energy content gas producers will have total integration with an on-site gas turbine. This would allow the two units to share "waste-heat" so as to improve the overall cycle. Typically this would involve using the gas turbine exhaust waste heat (and possibly the exhaust gases directly) to preheat the reactants entering the gasifier. Additionally heat exchange beween the hot fuel gases and the compressor discharge air prior to gas cleaning could also aid in improving the combined unit efficiency.

Medium energy content gases could be considered transportable, and if produced are likely to be used at some distance from the generating source. Many of these gases are close relatives of the low energy content gases. The difference is that oxygen is substituted for the air used in the gasifier, where a medium energy content gas is desired. The majority of medium energy content gases rely on the reaction between coal and steam to produce a mixture of hydrogen and carbon monoxide. This reaction is endothermic and various means of supplying the necessary energy are utilized. Generally oxygen (air for low energy gases) is introduced in parallel with the steam. The exothermic combustion reactions that take place offset the endothermic reactions.

Blue Water gas is a medium energy content gas that is produced by the action of steam on coal with the energy for the reaction supplied externally. Thus this gas can be considered as a baseline medium energy gas in that it is not contaminated with combustion products.

Because of this characteristic it was chosen as a representative fuel. In addition because of its lack of "inerts" and high flame temperature it was felt that this would be one of the more difficult fuels with which to obtain low NOx emissions.

6.3 COMBUSTION SYSTEMS

Two distinctly different combustion systems were considered for low emissions evaluation. The primary combustor utilized in the evaluation was the rich-lean combustor as developed for liquid fuel operation, and described in Section 3 and Reference 2.

The second combustor was a premixed lean primary zone system with a conventional dilution zone.

6.3.1 Rich-Lean Combustor Results

Sixteen test conditions each involving a minimum of five test points (differing fuel-air ratios) were defined to provide a series of detailed emission signatures of the rich-lean combustor, when operated on each of the three test fuels. These test conditions are provided in Table 8 with each set of conditions referenced to an engine power point. This latter engine being a hypothetical 12:1 compression ratio simple cycle industrial gas turbine, based on the Solar Centaur.

The simulated Winkler gas was chosen as a typical, difficult-to-burn low energy content, coal gas produced by combined air/steam blowing in a fluidized bed. It was found that this gas was difficult to burn in a controlled manner in the rich-lean combustor below the inlet pressures and temperatures associated with 6:1 compression. Apparently the concentration of carbon dioxide and nitrogen was sufficiently high that it significantly reduced the flame speed or reaction kinetics. At conditions above 690 kPa (100 psia) (and its associated compression temperature) the reaction rates were apparently sufficiently high to provide normal stable combustion. These reduced reaction rates created conditions such that the performance (that is, stability and efficiency, not NOx emissions) was sensitive to changes in the fuel gas composition, especially to the hydrogen concentration. Increases in hydrogen concentration caused increases in the overall reaction rate, and this, in turn, altered the efficiency. Small changes in inlet temperature also significantly changed stability and the combustion efficiency. This latter effect presumably was produced by changes in reaction rate created by deviations in the inlet temperature.

Very low NOx emissions could be obtained with the Winkler gas although, in practice, there would probably be limited operational range due to excessive carbon monoxide emissions. In operation the gas turbine would probably have to utilize an auxiliary fuel (such as propane or natural gas) for ignition and engine acceleration at least to the 50 percent power point. When this latter point was reached the Winkler gas could be gradually introduced to replace the initial fuel. Because of the above limitations, especially the sensitivity to fuel composition, it is felt that the Winkler type gas is not a prime fuel for gas turbine use.

In addition to the above thermochemical problems, it was found during testing that the high nitrogen flow requirements of the Winkler gas quickly exceeded the capability of the facility to supply vaporized nitrogen. This phenomenon lead in turn to failure of the facility nitrogen pumps through liquid nitrogen flashing in the pump housings. Because of the numerous nitrogen pump failures during Winkler gas testing, only limited emission data could be obtained. Generally when the nitrogen pumps failed, the lower heating value of the fuel increased rapidly causing overheating of the combustor walls and consequent shut-down of the experiment.

In light of the above, two operating points were concentrated on, the idle point and the 70 percent power point. The emissions signature at the idle power point is provided in Figure 12, and it shows a characteristic typical

Table 8

Rich-Lean Combustor Test Matrix

	Inlet Pressure (kPa)	Inlet Temperature (C)	Air Flow (kg/s)	Outlet Temperature (C)	Operating Conditions	Test Point Reference
Lurgi Gas (166 Btu/SCF)	1213	361	1.588	982	Baseload	1000
	1055	334	1.43	882	70% power	1100
	910	308	1.28	810	50% power	1200
	303	143	0.52	546	Spinning Idle	1300
Lurgi Gas (166 Btu/SCF) plus % NH_3	1213	361	1.588	982	Baseload	1400
	1055	334	1.43	882	70% power	1500
	910	308	1.28	810	50% power	1600
	303	143	0.52	546	Spinning Idle	1700
Winkler No. 1 (110 Btu/SCF)	1213	361	1.588	982	Baseload	3000
	1055	334	1.43	882	70% power	3100
	910	308	1.28	810	50% power	3200
	303	143	0.52	546	Spinning Idle	3300
Blue Water Gas (274 Btu/SCF)	1213	361	1.588	982	Baseload	2000
	1055	334	1.43	882	70% power	2100
	910	308	1.28	810	50% power	2200
	303	143	0.52	546	Spinning Idle	2300

(Sixteen test conditions each involving at least five test points to provide an emissions signature.)

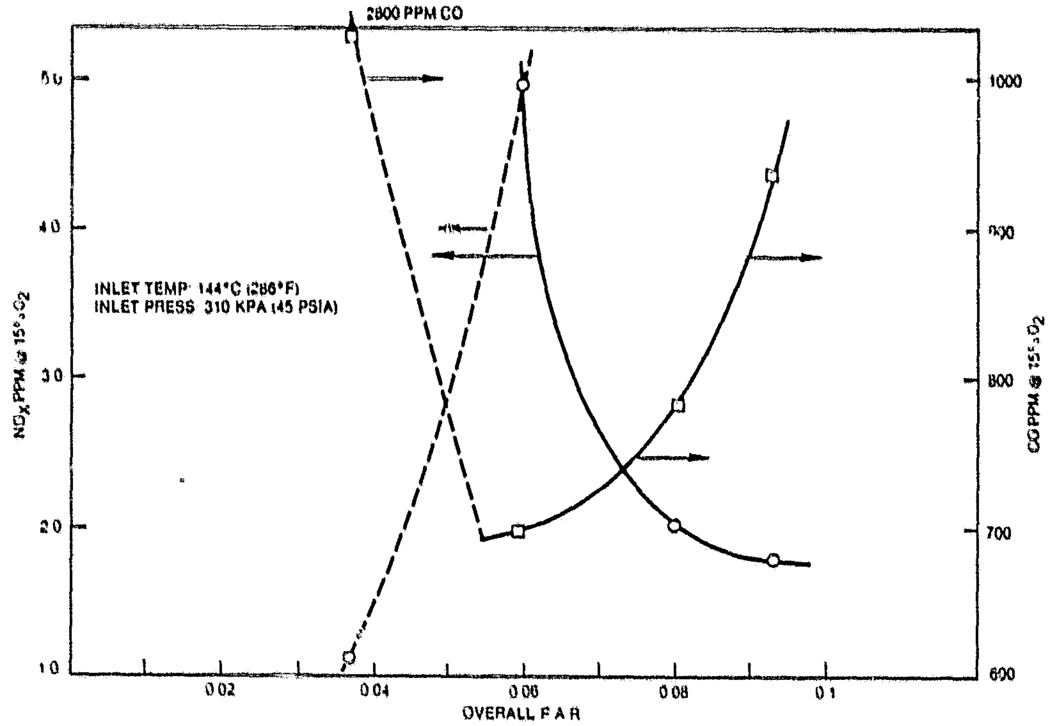

Figure 12. Winkler Gas Emissions Signature (Idle Point)

of a lean premixed combustion system with exceptionally low NOx levels. An attempt was made to extend this signature to much higher fuel-air ratios to ascertain the NOx emission trends. The results of this latter experiment are shown in Figure 13. NOx emissions although higher than before were still very low and appeared to be approaching an asymptote. The limiting fuel-air ratio was approximately 0.2 (primary zone equivalence ratio 0.875) and at this point a combination of various mechanical integrity problems forced the experiment to be terminated.

Data at the baseload power condition could not be obtained as the nitrogen flow requirements exceeded the available supply. As a consequence operation at the 70 percent power point was pursued and the results are shown in Figure 14. As previously experienced, the NOx emissions were very low although at these higher pressure conditions, the shape of the curves has changed. A NOx minimum is now apparent although it is not well defined.

In all cases the carbon monoxide (CO) emissions were high at conditions close to the lean extinction limit and reasonable at fairly high primary zone equivalence ratios. These high CO emissions would limit operation of the combustor primary zone to stoichiometric or greater, so as to provide reasonable combustion efficiencies over the turn-down range.

A Lurgi gas having a lower heating value of 61,839,583 J/m^3 (166 Btu/scf) was also evaluated, and it was found that the higher hydrogen concentration and higher heating value allowed much better control. The emissions signature

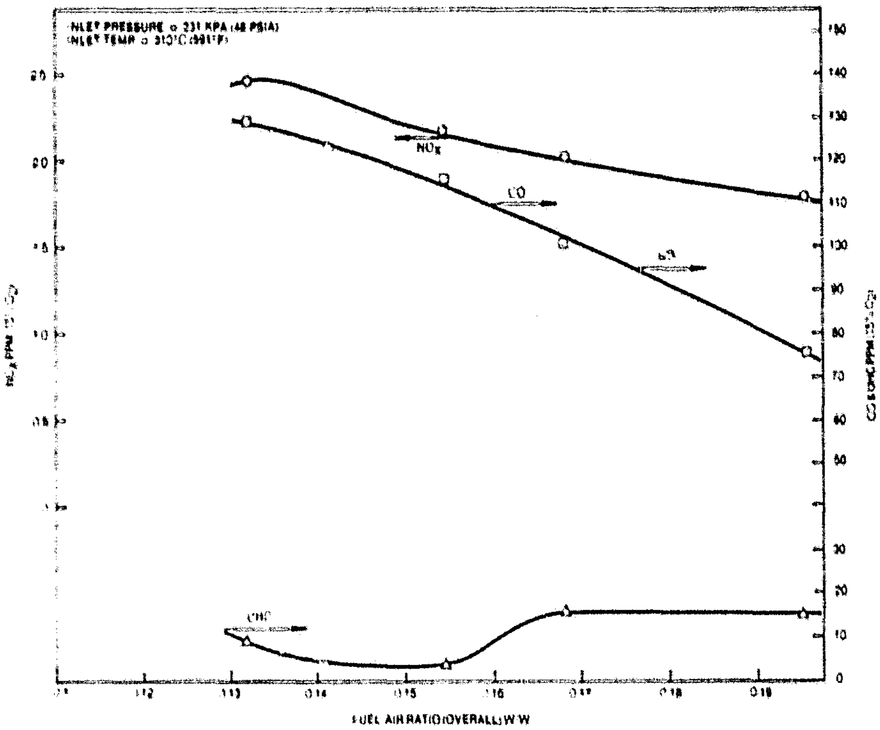

Figure 13. Winkler Gas Emissions Signature (Hot Idle Point)

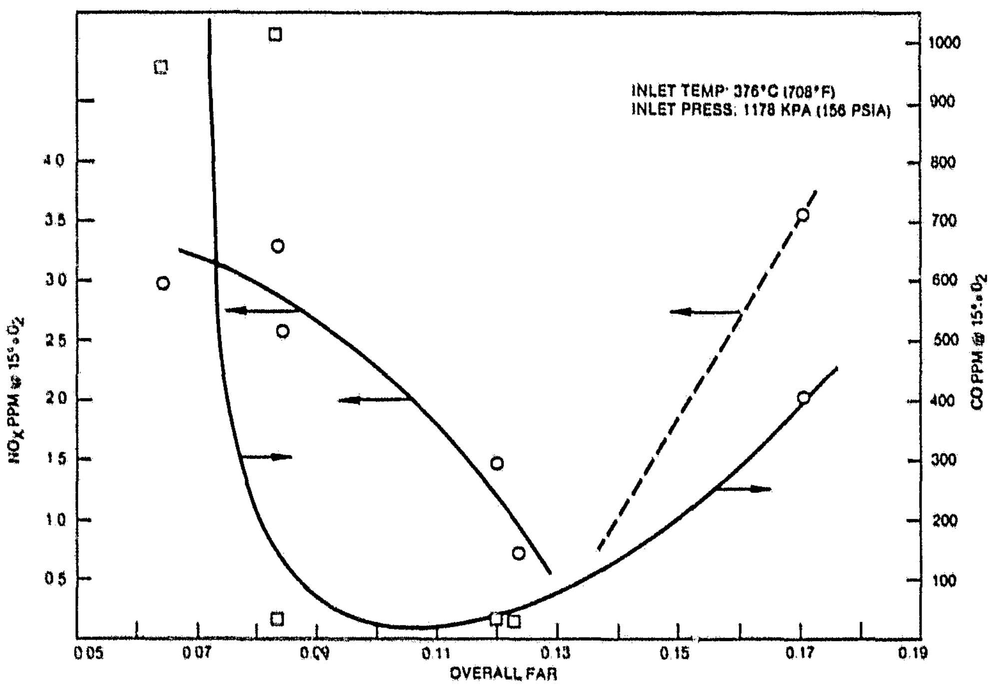

Figure 14. Winkler Gas Emissions Signature (70% Power Point)

of the Lurgi gas at the idle conditions is shown in Figure 15. As can be seen, exceptionally low NOx emissions and reasonable CO emissions can be obtained. Similar results were obtained at the 50 percent power point, 70 percent power point, and 100 percent power or baseload condition (see Figs. 16 and 17). Since this gas provided adequate operating range with reasonable efficiencies and low NOx emissions, it can be considered to be an attractive gas turbine fuel.

Ammonia was injected into the Lurgi gas to produce a concentration of the order of 1% v/v. Two sets of data were obtained with ammonia (NH_3) addition, one at the idle operating conditions and the other at the 50 percent power point. These data are shown in Figures 18 and 19, respectively. The maximum NOx levels produced are high in both cases. They are of the order of 50 percent of the level that would have been produced if all the ammonia had been converted. Thus, even though these reactions took place in a lean mode rather than a rich-lean staged combustion system, only a part of the ammonia has been converted to NOx.

The last gas to be evaluated on the rich-lean combustor was the Blue Water gas, chosen because of its potentially high NOx emission characteristic. This particular gas because of its high hydrogen content provides very high combustion temperatures, and as a consequence it was expected to produce high thermal NOx levels.

Because of the stoichiometric requirements of the Blue Water gas, it could be operated in a staged combustion rich-lean mode. The emissions signature at the idle power point (Fig. 20) shows a NOx minimum typical of rich-lean

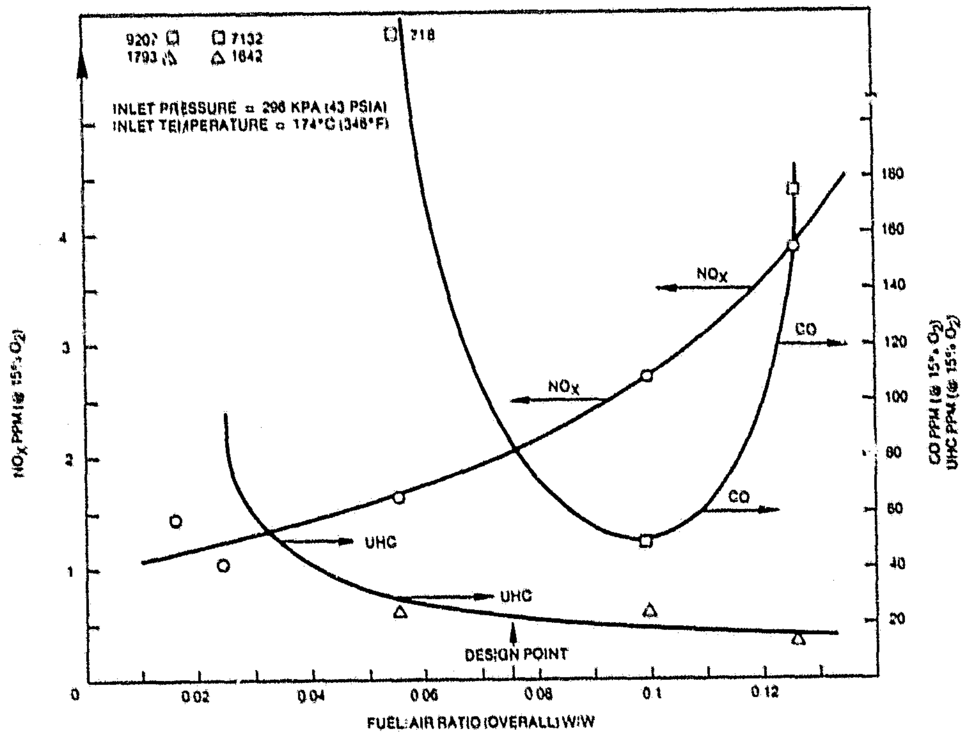

Figure 15. Lurgi Gas Emissions Signature (Idle Power Point)

ORIGINAL PAGE IS
OF POOR QUALITY

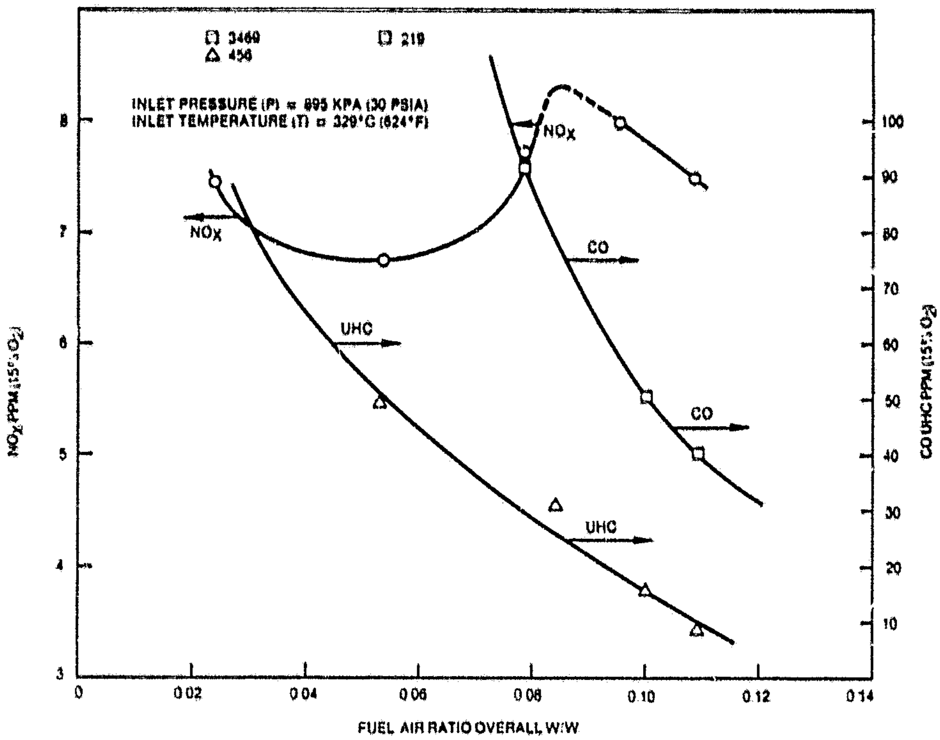

Figure 16. Lurgi Gas Emissions Signature (50% Power Point)

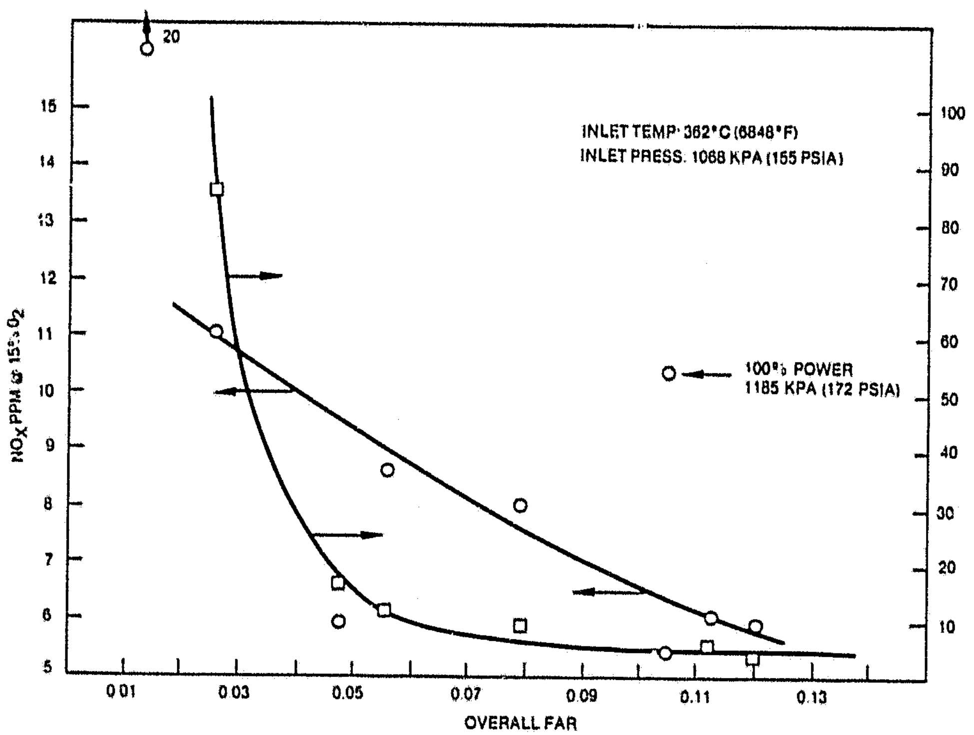

Figure 17. Lurgi Gas Emissions Signature (70% Power Point)

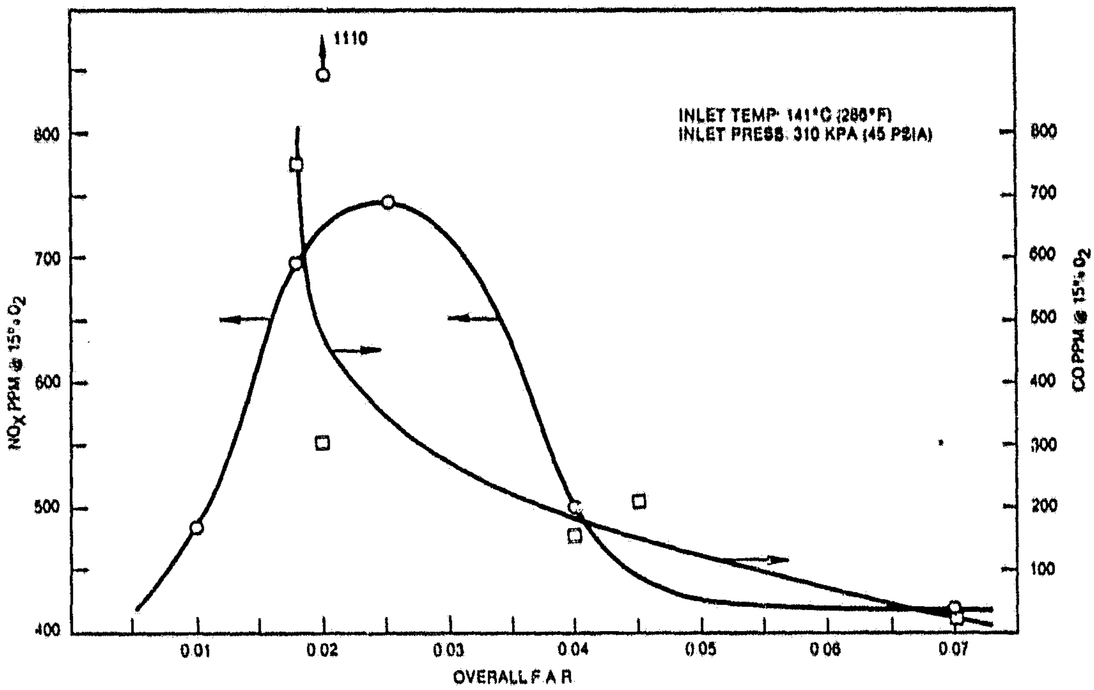

Figure 18. Lurgi Gas Emissions Signature (1% v/v Ammonia Added)

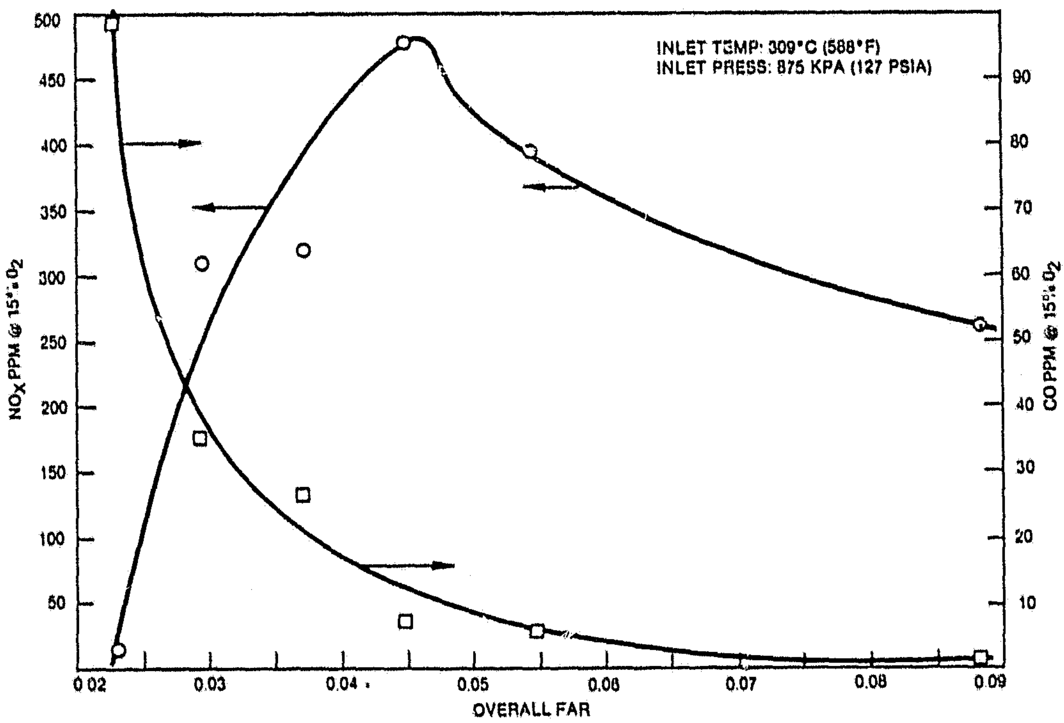

Figure 19. Lurgi Gas Emissions Signature (50% Power Point) 1% NH_3

ORIGINAL PAGE IS
OF POOR QUALITY

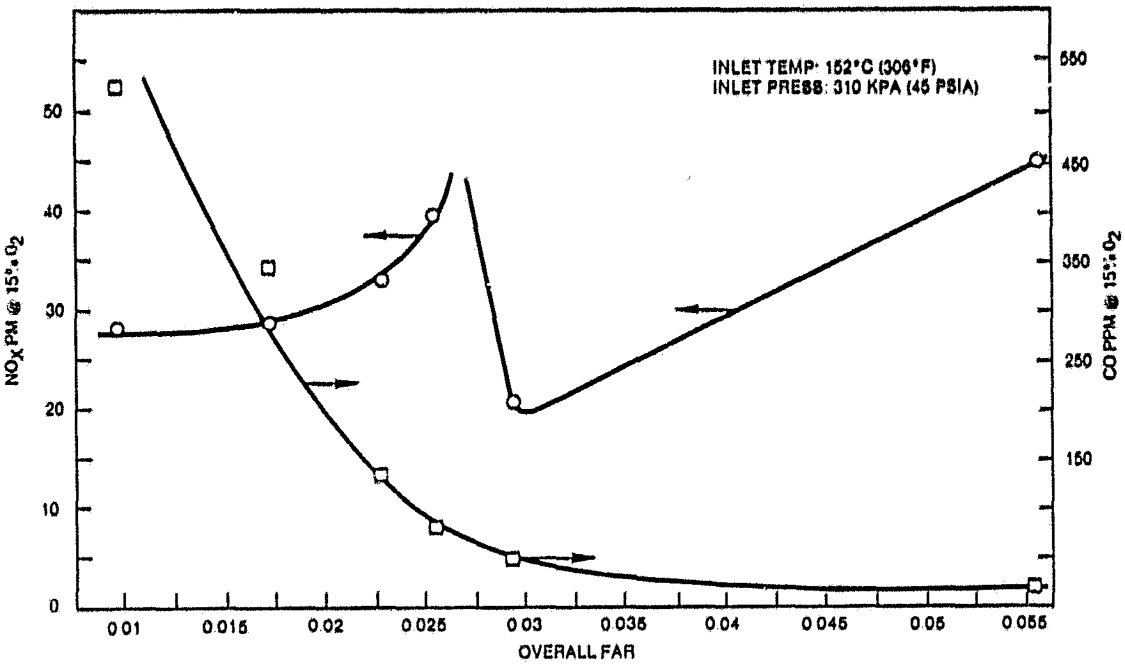

Figure 20. Blue Water Gas Emissions Signature

operation, however, the range at the minimum point appears to be very limited. A steep gradient is also present in the NOx curve as the primary zone fuel-air ratio is moved toward the stoichiometric point. At the 50 percent power point there is a significant shift upward in the general NOx levels (See Fig. 21) although the minimum point is still evident. Under these conditions it is apparent that the NOx emission goal of 75 ppm @ 15% O_2 could not be obtained. Similar results were obtained at the higher pressure conditions (see Figs. 22 and 23). As the inlet pressure and temperature increase it can be seen that the general NOx level increases. Of the three gases tested this is the only one that showed significant changes in emission signature with inlet temperature and pressure. This lack of sensitivity to inlet or operating point conditions of the NOx emissions for low energy content gases may be a factor in selecting the gas or family of gases which is to be produced from coal.

It is expected that if the composition of the medium energy fuel gas were changed drastically through methanation, then significant changes in NOx level would occur. Decreasing carbon monoxide and hydrogen levels, and increasing methane levels should result in reduced NOx levels. To evaluate this possibility a comparison between the Blue Water gas emissions and those of a methane-nitrogen mix of the same lower heating value was made. Figure 24 shows the emissions produced by a methane-nitrogen mix having the same lower heating value as the Blue Water gas. As can be seen, it is a totally different characteristic from that provided in Figure 20, with generally lower NOx levels. The fact that methanation of medium energy content gases

ORIGINAL PAGE IS
OF POOR QUALITY

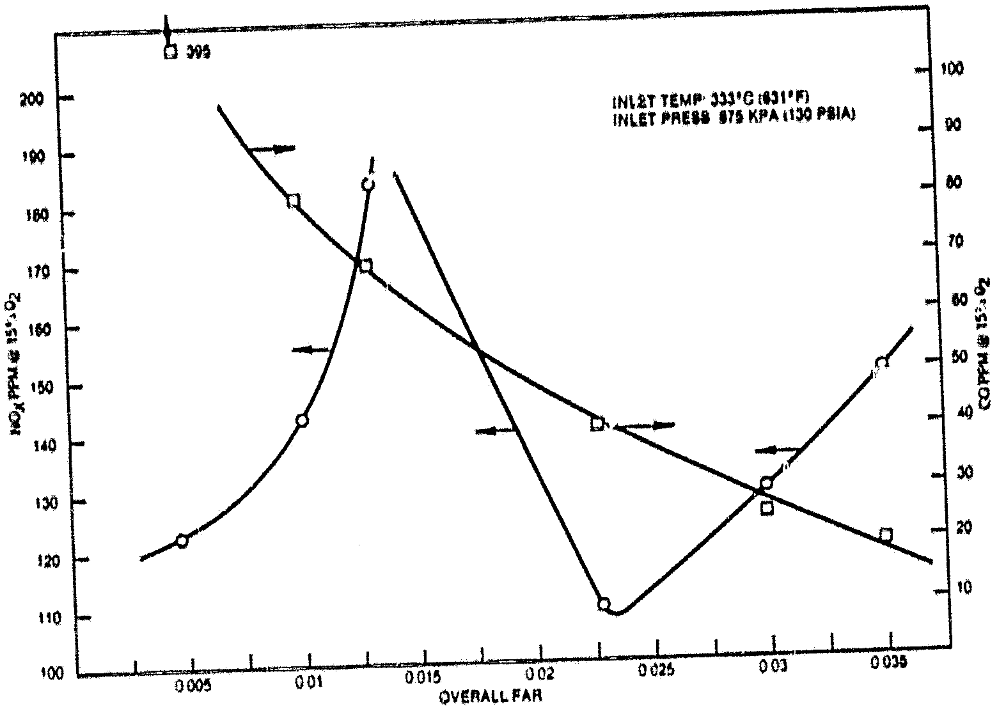

Figure 21. Blue Water Gas Emissions Signature

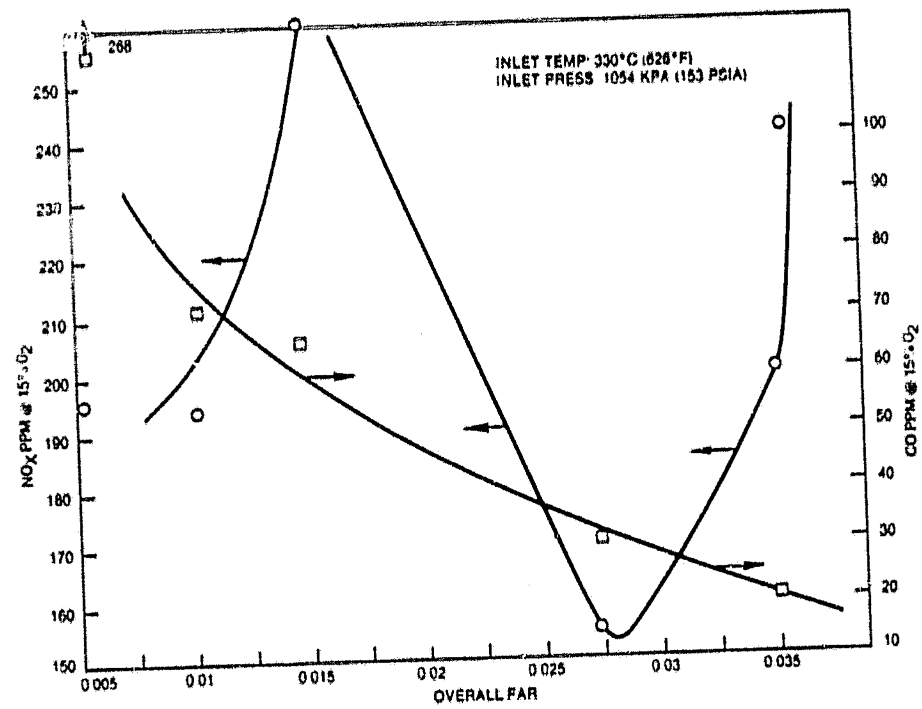

Figure 22. Blue Water Gas Emissions Signature

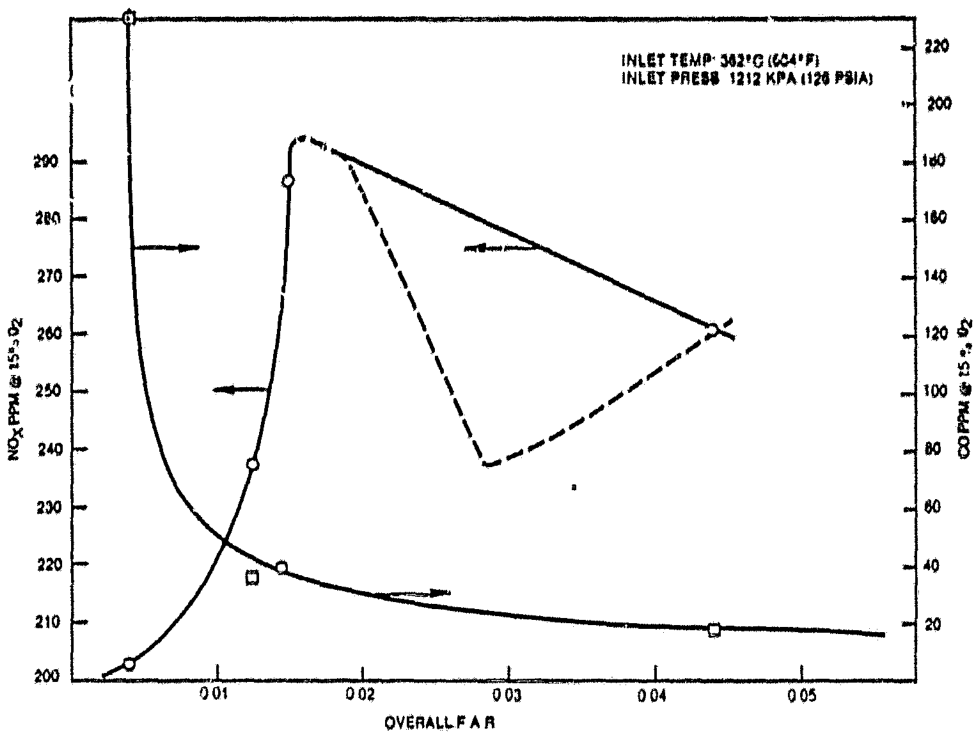

Figure 23. Blue Water Gas Emissions Signature

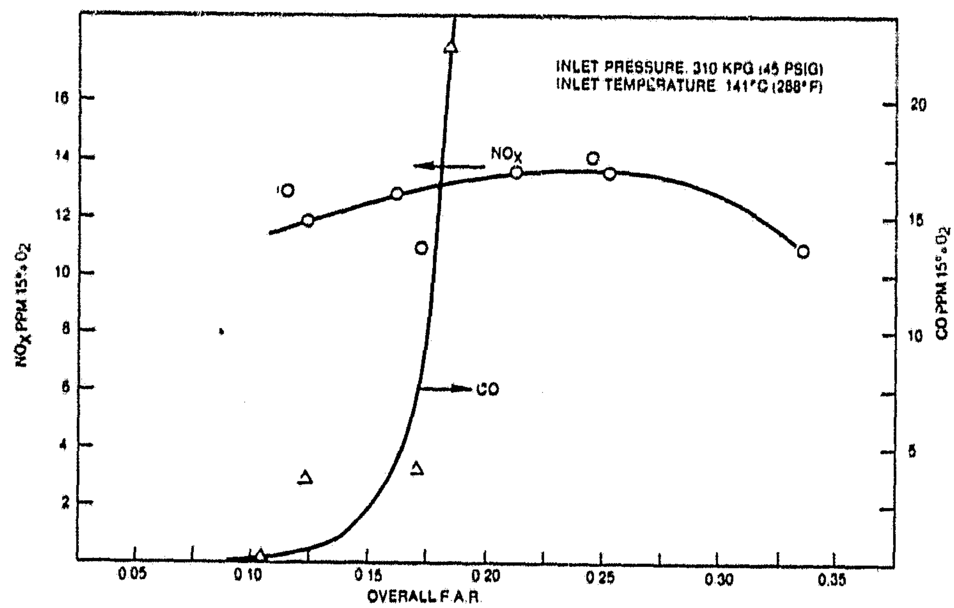

Figure 24. Methane (CH_4) and Nitrogen (N_2) Mixture Simulating Blue Water Gas Lower Heating Value

ORIGINAL PAGE IS
OF POOR QUALITY

could provide lower NOx levels during their subsequent combustion may also be a factor in both coal-gas type selection, and coal gas processing.

It should be noted that over the range of conditions tested none of the fuels produced wall overheating problems that were severe enough to preclude operation. Generally the maximum wall temperatures were found in the throat; however, they never exceeded 900°C (1650°F).

6.3.2 Lean Premixed Combustor

The combustor shown in Figure 25 was mounted in the test rig described previously, and it was planned to operate this combustor with each of the three gases and at each of the conditions listed in Table 9. This combustor had been designed originally to burn a partially vaporized kerosene type fuel in a lean primary zone. The high volume flows of the low and medium energy content gases made it difficult to inject them into each of the mixing ports (through plain end tubes). Because of the high fuel injection pressures and high flows required, some spilling of the gases around the ports occurred with subsequent ingestion into the film cooling holes. This created severe mechanical integrity problems and the approach was abandoned. Injection of the low energy content gas alone through two of the primary ports, with the remaining four being used for air, was also investigated. This approach also proved to be unsatisfactory as it caused the primary recirculation zone to become unsable. Violent oscillations were created causing the air flow control system to fail.

Larger diameter air ports were needed to allow satisfactory and safe injection of the gases into the combustor. Neither schedule nor funds permitted the redesign and rebuilding of the combustor and thus no reasonable emissions data could be obtained with this lean primary zone system.

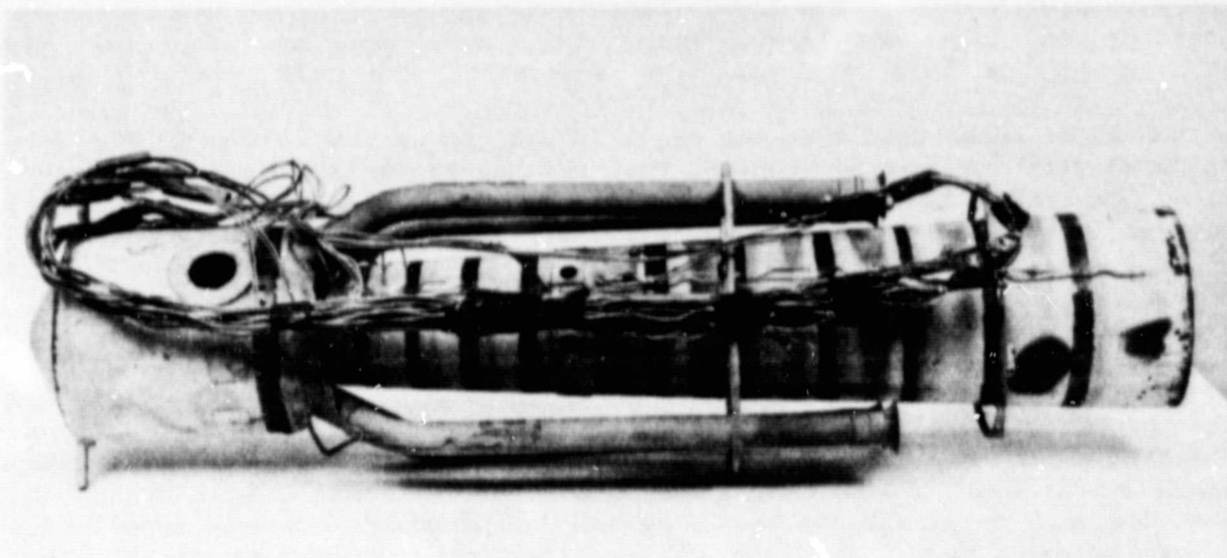

Figure 25. Lean Primary Zone Combustor

Table 9

Lean Premixed Combustor Test Matrix

Fuel Type	Inlet Pressure (kPa)	Inlet Temperature (C)	Air Flow kg/s	Outlet Temperature (C)	Operating Condition	Test Point
Lurgi (166 Btu/SCF)	1055	334	1.43	882	70% power	4000
Blue Water Gas (274 Btu/SCF)	1055	334	1.43	882	70% power	5000
Winkler No. 1 (110 Btu/SCF)	1213	361	1.43	982	Baseload	6000
Winkler No. 1 (110 Btu/SCF)	1055	334	1.43	882	70% power	6100
Four test conditions each involving at lest five test points (varying fuel-air ratio) to provide an emissions signature.						

7

CONCLUSIONS AND RECOMMENDATIONS

The rich-lean combustor system developed by Solar Turbines Incorporated during the course of an earlier program described in Reference 2, has shown itself capable of burning a wide range of low and medium energy content fuels, based on carbon monoxide (CO) and hydrogen (H_2). Operation of this combustion system involved lean primary zone operation for the low energy content fuels and a rich-lean mode for the medium energy content fuel. The lean combustion of the low energy content fuels which were a simulated air-blown Winkler gas and an air-blown Lurgi gas respectively, demonstrated that the combustor used could produce low NOx levels. These emissions easily met the goals of providing NOx levels below 75 ppm @ 15% O_2, and were generally insensitive to inlet pressure and temperature conditions.

Ammonia addition to the Lurgi gas, to provide a one percent by volume concentration, increased the NOx emissions significantly to levels well above the goals. The peak levels produced, however, were approximately one half of those that would have been obtained had all the ammonia converted into NOx. Thus a significant reduction can be claimed.

Emissions of NOx during the combustion of the medium energy content fuel (a simulated Blue Water gas) were above the goal except at low pressure conditions. The emisisons signature showed the expected "rich-lean minimum" at rich primary zone conditions. The slope of the curve, however, between stoichiometric and the NOx minimum fuel-air ratio was much steeper than that encountered with conventional petroleum fuels.

NOx emissions were expected to be high with the Blue Water gas fuel, because of its high flame temperature characteristic. Thermal NOx produced in the secondary zone proabably dominated the entire emission signature. With increasing pressure and temperature, the shape of the NOx emission curves remained essentially constant, however, their general level increased substantially. Insufficient data is available at present to extract the true dependence of the NOx levels on inlet pressure and temperature.

No severe mechanical integrity problems were encountered with the three fuels at any of the operating conditions. Generally maximum wall temperatures were below 900°C (1650°F), which is adequate for the cooling system employed.

It was found that the very low energy content gas (Winkler) had a limited controllable range in that its stability and efficiency (not NOx) were very sensitive to small changes in the hydrogen concentraiton, and inlet temperature. The Lurgi gas which had a higher hydrogen concentration did not suffer from this problem. It can be postulated then on the limited evidence available that there is probably an optimum fuel in the family of CO/H_2 gas mixtures that lies somewhere between the Winkler and the Blue Water gas. This fuel would have good stability and efficiency characteristics, but yet would have a NOx characteristic insensitive to operating conditions (unlike the Blue Water gas).

Because of the high flame temperatures of the high hydrogen content fuels such as the Blue Water gas, it may be advantageous to process the fuel further before combustion. Typically methanation, could increase the lower heating value and lower the stoichiometric flame temperatures, allowing lower NOx to be produced.

It is recommended that an investigation be made to determine the optimum low energy content coal-gas fuel for gas turbine use. The results of similar investigations should provide a guide to the developers of gasification systems. The optimum fuel (consisting of CO, H_2, CO_2 and N_2) would have good stability and efficiency with a NOx level below the goals and would be insensitive to operating conditions. This could be achieved by investigating fuels with differing CO/H_2 ratios and different levels of inerts (varying lower heating values).

Methanation of the medium energy content gases should also be evaluated as a potential aid in providing a fuel with low NOx characteristics. An investigation into the effects of methane content on NOx production could readily be implemented, and could, as above, provide a guide to coal-gas producers as to which gases to produce.

The above is a synopsis of the conclusions and recommendations of the program as performed to date. More detailed conclusions and recommendations may be produced when the work of the other contractors is analyzed.

ORIGINAL PAGE IS
OF POOR QUALITY

8

REFERENCES

1. NSPS Standards, Federal Register, October 3, 1977, Revised Sept. 10, 1979.

2. White, D. J., LeCren, R. T. and Batakis, A. P., "Low NOx Heavy Fuel Combustor Concept Program", Final Report on Contract NASA DEN3-145, NASA CR-165481, Solar Turbine Incorporated, 1982.