

General Disclaimer

One or more of the Following Statements may affect this Document

- This document has been reproduced from the best copy furnished by the organizational source. It is being released in the interest of making available as much information as possible.
- This document may contain data, which exceeds the sheet parameters. It was furnished in this condition by the organizational source and is the best copy available.
- This document may contain tone-on-tone or color graphs, charts and/or pictures, which have been reproduced in black and white.
- This document is paginated as submitted by the original source.
- Portions of this document are not fully legible due to the historical nature of some of the material. However, it is the best reproduction available from the original submission.

✓ DOE/NASA/13111-13
NASA TM-82998

Combustion Characteristics of Hydrogen—Carbon Monoxide Based Gaseous Fuels

(NASA-TM-82998) COMBUSTION CHARACTERISTICS
OF HYDROGEN. CARBON MONOXIDE BASED GASEOUS
FUELS (NASA) 16 p HC A02/MF A01 CSCL 21D

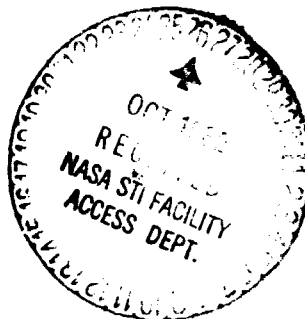
N83-10560

Unclas
G3/44 35580

David J. White, Alan J. Kubasco, and Richard T. LeCren
Solar Turbines Incorporated

and

Joseph J. Notardonato
National Aeronautics and Space Administration
Lewis Research Center



Work performed for
U.S. DEPARTMENT OF ENERGY
Fossil Energy
Office of Coal Utilization and Extraction

Prepared for
Joint Power Conference
Denver, Colorado, October 17-21, 1982

Combustion Characteristics of Hydrogen—Carbon Monoxide Based Gaseous Fuels

David J. White, Alan J. Kubasco, and Richard T. LeCren
Solar Turbines Incorporated
San Diego, California

and

Joseph J. Notardonato
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135

Work performed for
U.S. DEPARTMENT OF ENERGY
Fossil Energy
Office of Coal Utilization and Extraction
Washington, D.C. 20545
Under Interagency Agreement DE-AC01-77ET13111

Prepared for:
Joint Power Conference
Denver, Colorado, October 17-21, 1982

COMBUSTION CHARACTERISTICS OF HYDROGEN - CARBON MONOXIDE BASED GASEOUS FUELS

David J. White, Alan J. Kubasco, and Richard T. LeCren
Solar Turbines Incorporated
San Diego, California
and

Joseph J. Notardonato
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio

ABSTRACT

An experimental rig program has been conducted with the objective of evaluating the combustion performance of a family of fuel gases based on a mixture of hydrogen and carbon monoxide. These gases, in addition to being members of a family, were also representative of those secondary fuels that could be produced from coal by various gasification schemes. In particular, simulated Winkler, Lurgi, and Blue-water low and medium energy content gases were used as fuels in the experimental combustor rig. The combustor used was originally designed as a low NOx rich-lean system for burning liquid fuels with high bound nitrogen levels. When used with the above gaseous fuels this combustor was operated in a lean-lean mode with ultra long residence times. The Blue-water gas was also operated in a rich-lean mode. The results of these tests indicate the possibility of the existence of an "optimum" gas turbine hydrogen - carbon monoxide based secondary fuel. Such a fuel would exhibit low NOx and high efficiency over the entire engine operating range. It would also have sufficient stability range to allow normal light-off and engine acceleration. Solar Turbines Incorporated would like to emphasize that the results presented here have been obtained with experimental rig combustors. The technologies generated could, however, be utilized in future commercial gas turbines.

INTRODUCTION

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

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 that contain nitrogen. This latter mechanism, depending on the concentration of nitrogen in the fuel, can dominate the NOx 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 1540 C (2804°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. For clean fuel gases (non-nitrogen containing) an effective approach is to utilize a lean primary zone with a premixed air-fuel charge. Low NOx can be obtained with such a combustion system.

The goals of the work described herein were to obtain NOx emissions of 75 ppm corrected to 15 percent O₂ for fuel bound nitrogen levels up to one percent by weight and 37 ppm at 15 percent O₂ 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.

A novel approach that provides effective rich primary zone cooling had been developed for the combustor utilized. The basic new contribution to combustor wall cooling technology was the use of primary zone regenerative cooling. All the primary combustion air in this arrangement was first used to cool the primary zone walls. After cooling the walls the preheated air at temperatures in excess of 482 C (900°F) passed into the combustor, mixing with the fuel in a short internal passage. By the time that the combustion reactions were initiated, the fuel was well mixed with the air. This system avoided external premixing of air and fuel with its attendant problems of autoignition and flashback. In addition it largely eliminated the normal high

E-1434

Table 1. Adopted Combustor Test Conditions

Combustor Inlet Conditions	Peak Load 110% Power	Engine Power Condition				
		Base Load	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

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 were obtained. This latter effect was due primarily to the significant increase in the effective inlet air temperature.

COMBUSTOR DESIGN

The combustor used for evaluation of the low and medium energy content hydrogen/carbon monoxide gases was the rich-lean combustor as described in Reference 2. The combustor as developed during this referenced previous work 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 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 momenta) is arranged to balance the momentum of the gases exiting from the transition piece within the secondary zone proper. This latter effect is utilized to ensure that the transition through stoichiometric takes place within the secondary zone for rich-lean operation. 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 (during rich-lean combustion) is minimized. This minimization of the residence time at stoichiometric in turn minimizes the thermal NO_x production.

The rich-lean combustor used for the tests had stoichiometry or air flow splits that were designed for typical petroleum and coal-based liquid fuels having stoichiometric fuel-air ratios in the range of 0.067 to 0.068. 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 $4,097,811 \text{ J/m}^3$ (110 Btu/scf) and the simulated Lurgi gas $6,183,968 \text{ J/m}^3$ (166 Btu/scf) have stoichiometric fuel-air ratios by weight of 0.55 and 1.04 respectively. When compared to the

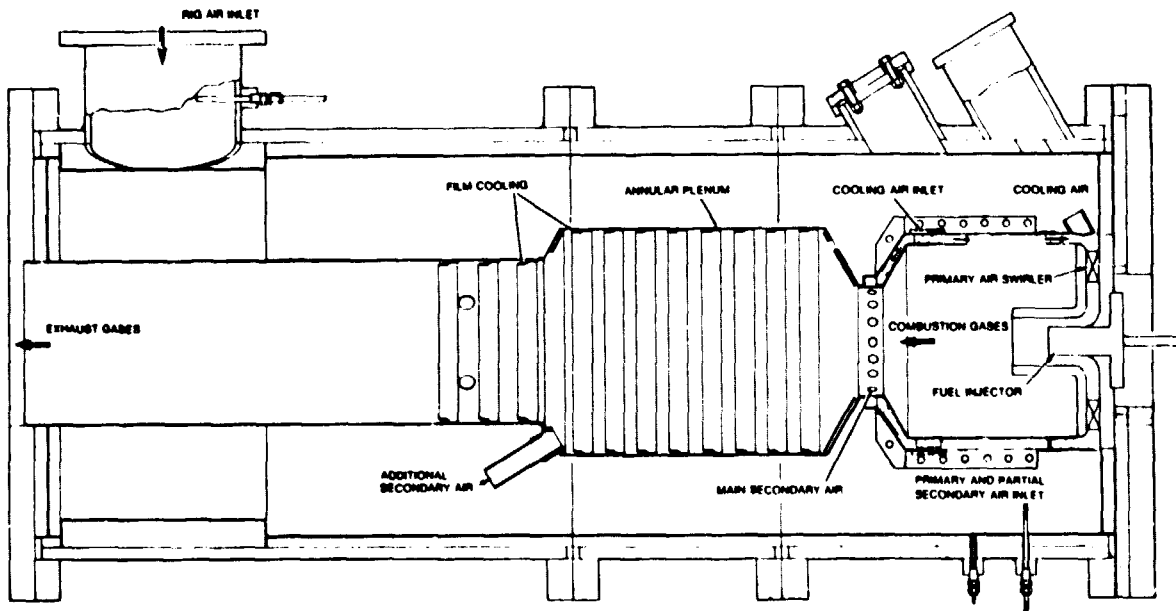


Fig. 1 Rich-Lean Combustor

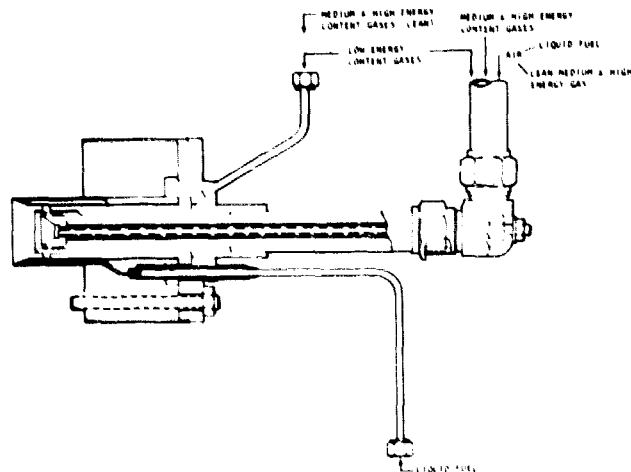


Fig. 2 Advanced Gas-Liquid Fuel Injector

combustor 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 $10,207,272 \text{ J/m}^3$ (274 Btu/scf).

Fuel Injection

To ensure the maximum of fuel flexibility an injector was designed from the outset 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 liquid 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 the central passage could be used. Liquid fuel 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 intro-

duced 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.

FUELS AND FUEL SYSTEM

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 $4,097,811$ and $10,207,274 \text{ J/m}^3$ (110 and 273 Btu/ft³) and formed a family of fuels consisting mainly of CO/H₂ mixture. Table 2 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 gases included hydrogen (H₂), carbon monoxide (CO), methane (CH₄), carbon dioxide (CO₂), and nitrogen (N₂). The nominal compositions adopted for each of the gases is provided in Tables 3, 4 and 5.

All of the component gases needed with the exception of the carbon monoxide were obtained from existing facility sources. The carbon monoxide was 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, underground 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 (CH₄), carbon dioxide, and nitrogen were utilized. Each of these facility sources were "piped" to the test cell using 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 was reduced or increased together as necessary for the range near the maximum flow requirements. Combustion low flow conditions were obtained by bleeding a portion of the flow to a flare which burned the gas in ambient air. In typical operation

Table 2. Gaseous Fuels From Coal Analyses and Properties

Class	Low Btu Gas							Medium Btu Gas				
Process	Producer Gas From Coal	blast Furnace Gas	Lurgi	Lurgi	Fluidised Bed Coal Gas	Bureau of Mines BOM 7644	Winkler #1	Blue Water Gas	Coal Gas (Vertical Retort w/Steaming)	Winkler #2	Lurgi	Koppers Totzek
Air												
Oxygen												
Steam												
Analysis (% of Vol)												
O ₂	--	--	--	--	--	--	--	--	0.4	--	--	--
H ₂	52.4	60	40.26	30.2	50.4	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	20	--
C ₂ H ₆	--	--	--	--	--	--	--	--	2	--	--	--
C ₃ H ₈	--	--	--	--	--	--	--	--	--	--	--	--
C ₄ H ₁₀	--	--	--	--	--	--	--	--	--	--	--	--
C ₅ H ₁₂	--	--	--	--	--	--	--	--	--	--	--	--
H ₂ O	--	--	0.257	27.8	0.5	0	--	--	--	--	0.26	0.37
H ₂ S	--	--	0.63	0.5	0.7	0	--	--	--	--	1.0	0
Net 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
(BTU) stoich (C)	2871	2205	1986	2312	3007	2638	2438	3769	3601	3501	2057	2500
(P/A) stoich												
(by wt)	0.711	1.462	0.5519	0.712	0.674	0.770	1.040	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	0.44
Fuel Flow Mass Ratio	9.56	18.0	8.13	10.5	8.85	10.6	13.7	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
Btu/lb _{air} stoich	1600	1746	1460	1453	1637	1556	1641	1613	1395	1597	1415	1637
Limits of Flammability												
Lower L ₂₅	17.38	37.59	12.6	18.0	15.05	16.6	20.5	6.38	5.6	7.48	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 3. 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%
Lower heating value 116 Btu/ft ³	

Table 5. 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	

Table 4. 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	

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 increments 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 was diverted to the flare. By adjusting the flow to the flare the desired range of combustion fuel/air ratios could be obtained.

To ensure the complete combustion of a wide range of low and medium energy content gases, each

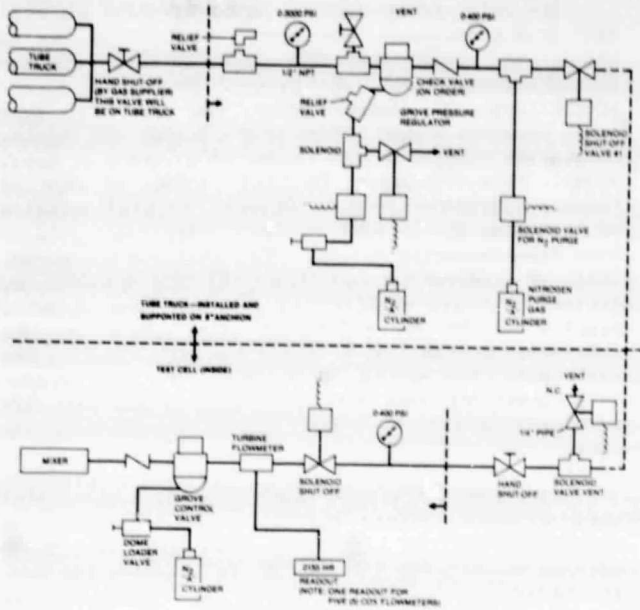


Fig. 3 Schematic of the Carbon Monoxide Supply System

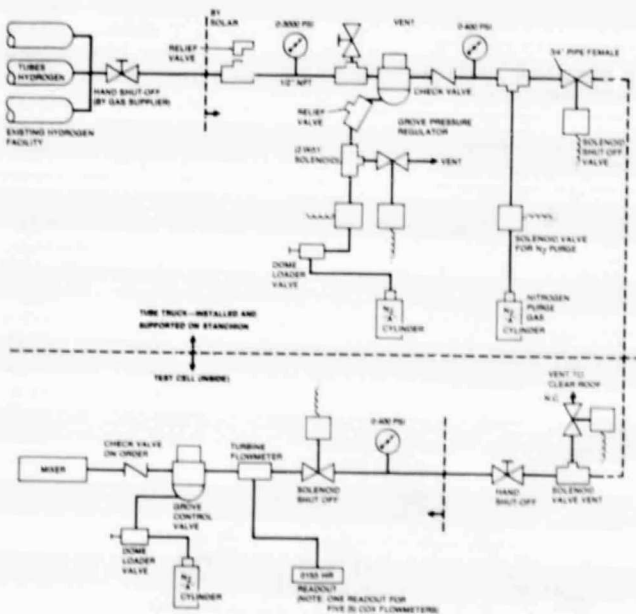


Fig. 4 Schematic of the Hydrogen Supply System

having widely ranging flow rates, the flare was equipped with a permanent natural gas fueled pilot. This latter pilot flame was arranged to entrain the gases to be burned before entraining the necessary air for combustion.

EXPERIMENTAL APPARATUS

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 as shown in a casing in a reverse flow configuration. The forward end of the combustor is rigidly attached

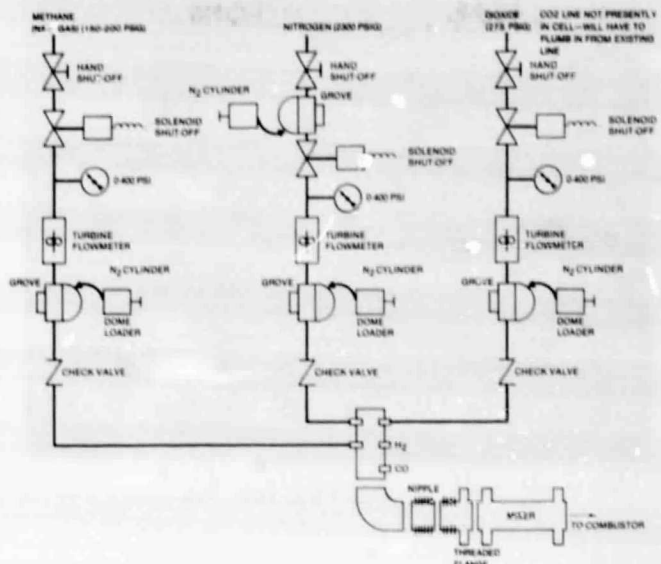


Fig. 5 Generalized Piping Diagram



Fig. 6 Stainless Steel Gas Mixer

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.

The 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.

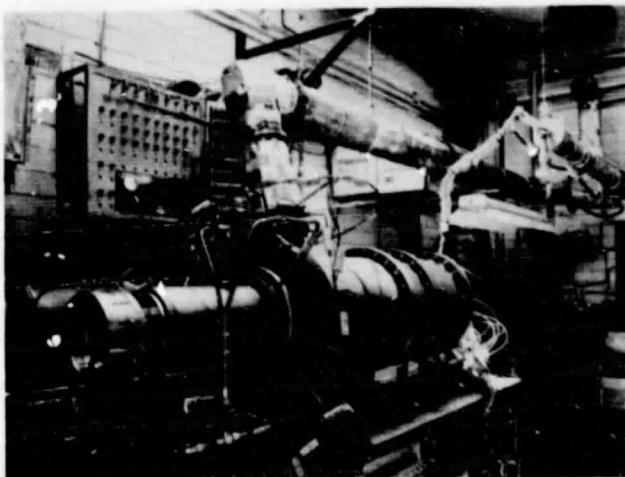


Fig. 7 Combustor Test Rig

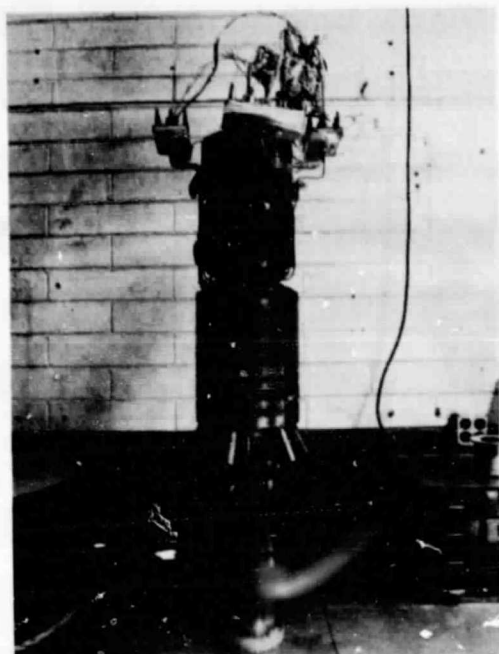


Fig. 8 Rich-Lean Combustor

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, and six Kiel type total pressure probes. Each of the probes and thermocouple was 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 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 facility 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 361°C (682°F) for this particular air flow.

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 Type K thermocouples located at the centers of equal areas in the exhaust flow stream. These are not shown in Figure 9.

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

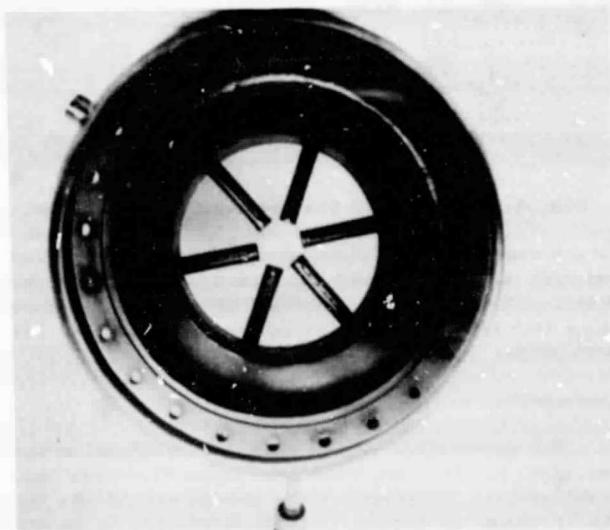


Fig. 9 Instrumentation Rig

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 temperature). 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 through a 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 from 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.

Test Procedures

The following procedure was used to conduct the

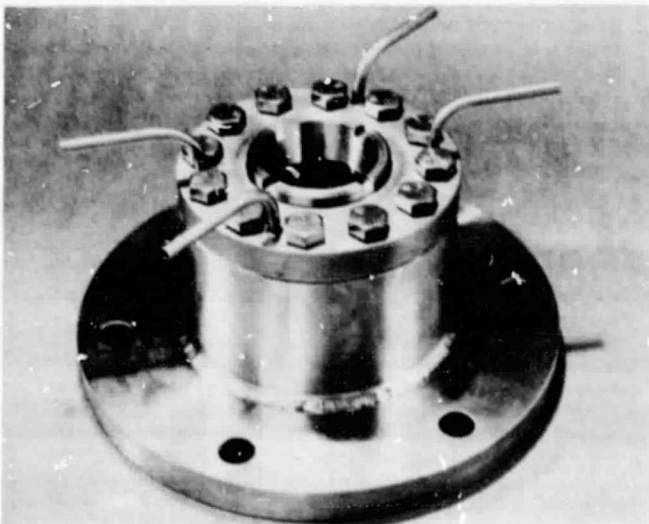


Fig. 10 Quartz Window



Fig. 11 Control Panel

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 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 then the inlet air temperature was varied by changing the preheater setting and the 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, and cooling the rig by continuing to flow air through it.

TEST RESULTS AND DISCUSSION

Emission signatures for the developed rich-lean combustor were determined over the full range of simulated engine conditions (see Table 1). Specifically combustor exhaust emission levels of NO_x, CO, UHC, smoke and CO₂ were determined for each of the engine test points, with each of the three fuels. These 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 NO_x 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 120°F (the maximum safe operating temperature of the cylinders) was of the order of 1930 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. A need to redesign the ammonia system was thus apparent. However, because of limitations imposed by the schedule, redesign was not accomplished, and operation at lower pressures was all that could be achieved.

Fuels

Three fuels all based on CO/H₂ mixtures, were chosen for experimental purposes and these are described in detail in the Fuels and Fuel System section above. 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/m³ (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/m³ (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 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 between 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, when 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; and 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 baseline 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 in which to obtain low NOx emissions.

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 6 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. Only at conditions above 690 kPa (100 psia) (and its associated compression temperature) were the reaction rates sufficient 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,

Table 6. 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 (100 Btu Btu)	1213	161	1.500	982	Baseline	1000
	1055	114	.41	882	70% power	1100
	910	108	1.28	810	50% power	1200
					Spinning Idle	1300
	303	143	0.52	546		
Winkler Gas (110 Btu Btu)	1213	161	1.500	982	Baseline	1400
	1055	114	1.43	882	70% power	1500
	910	108	1.28	810	50% power	1600
					Spinning Idle	1700
	303	143	0.52	546		
Blue Water Gas (166 Btu Btu)	1213	161	1.500	982	Baseline	2000
	1055	114	1.41	882	70% power	2100
	910	108	1.28	810	50% power	2200
					Spinning Idle	2300
	303	143	0.52	546		

Sixteen test conditions each involving at least five test points to provide an emission signature.

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 to around the 50 percent power point. When this latter point was reached the Winkler gas was 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 shown in Figure 12, and shows a characteristic typical 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 approx-

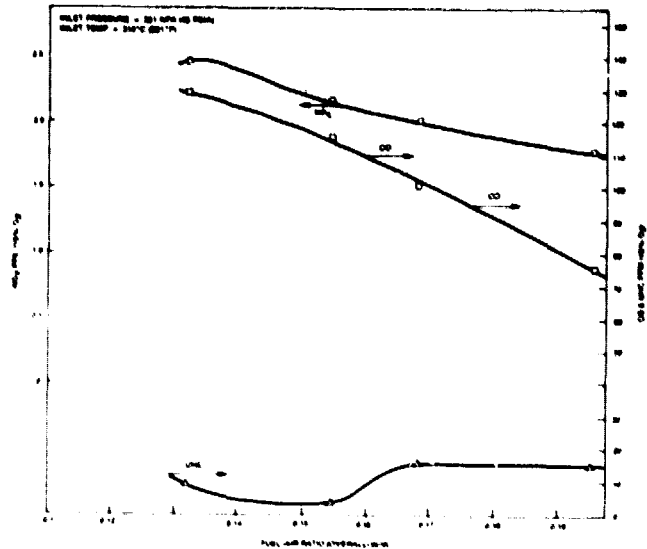


Fig. 13 Winkler Gas Emissions Signature
(Hot Idle Point)

imately 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.

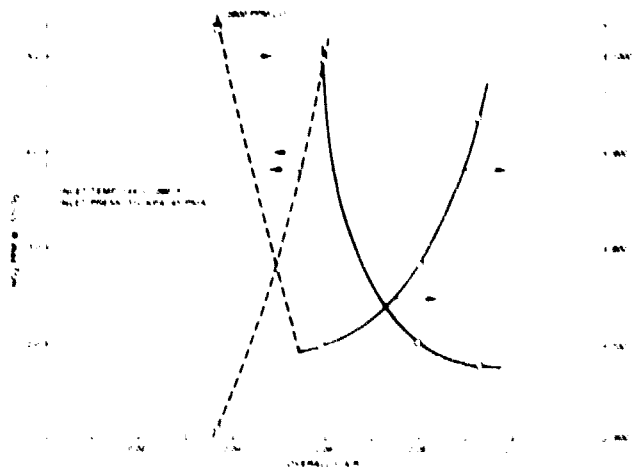


Fig. 12 Winkler Gas Emissions Signature
(Idle Point)

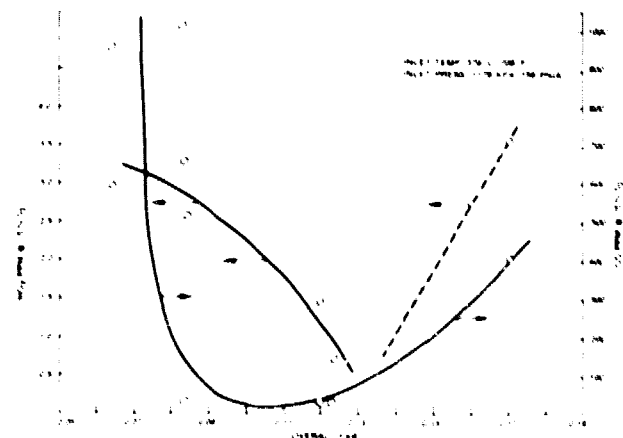


Fig. 14 Winkler Gas Emissions Signature
(70% Power Point)

A Lurgi gas having a lower heating value of 61,839,583 J/m³ (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 of the Lurgi gas at the idle conditions is shown in Figure 15. As can be seen, exceptionally low NO_x 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 NO_x emissions, it can be considered to be an attractive gas turbine fuel.

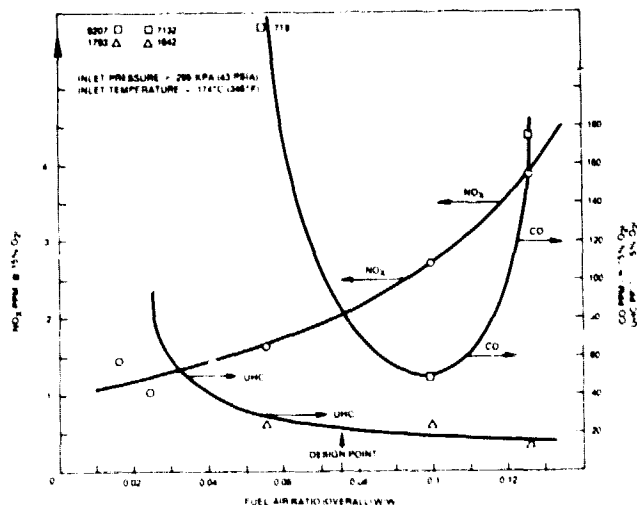


Fig. 15 Lurgi Gas Emissions Signature
(Idle Power Point)

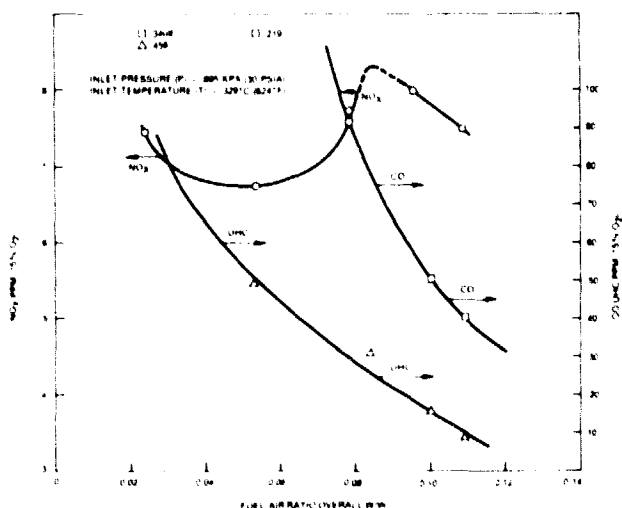


Fig. 16 Lurgi Gas Emissions Signature
(50% Power Point)

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 (NH₃) 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 NO_x levels produced are high in both cases. They

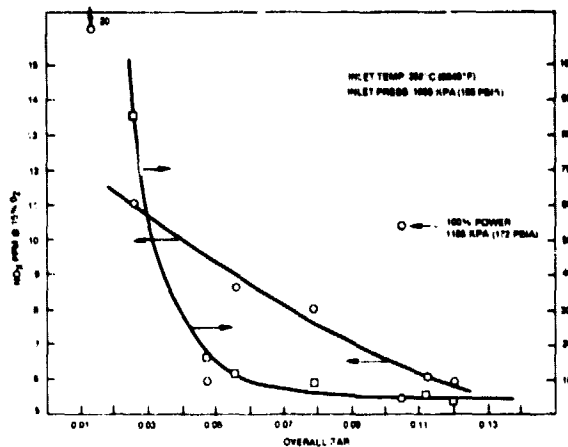


Fig. 17 Lurgi Gas Emissions Signature
(70% Power Point)

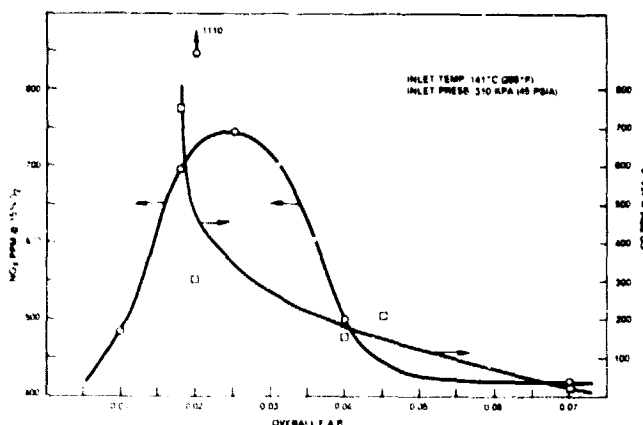


Fig. 18 Lurgi Gas Emissions Signature
(1% v/v Ammonia Added)

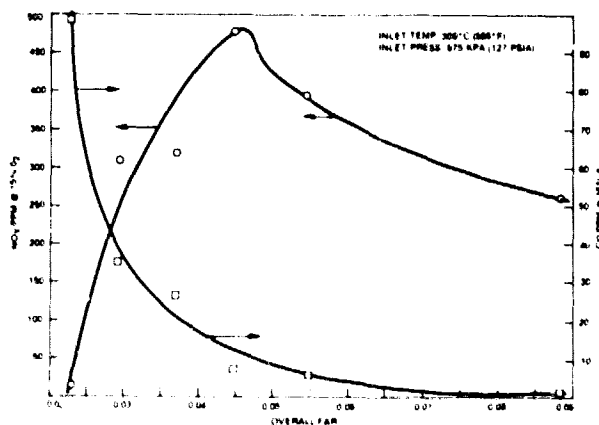


Fig. 19 Lurgi Gas Emissions Signature
(50% Power Point) 1% NH₃

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 NO_x.

ORIGINAL PAGE IS OF POOR QUALITY

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 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 here 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% O₂ 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 sensi-

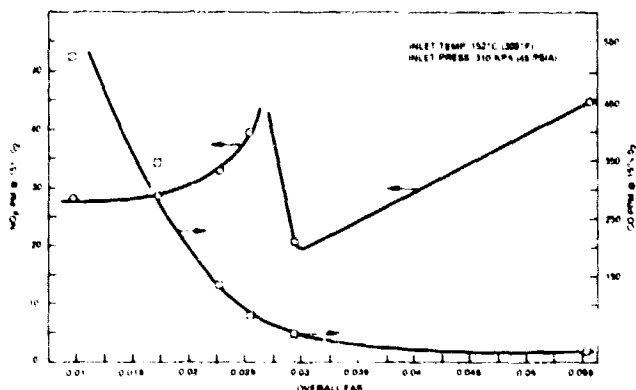


Fig. 20 Blue Water Gas Emissions Signature

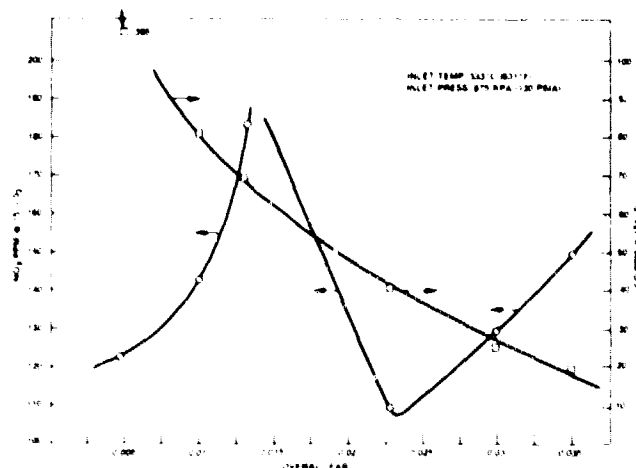


Fig. 21 Blue Water Gas Emissions Signature

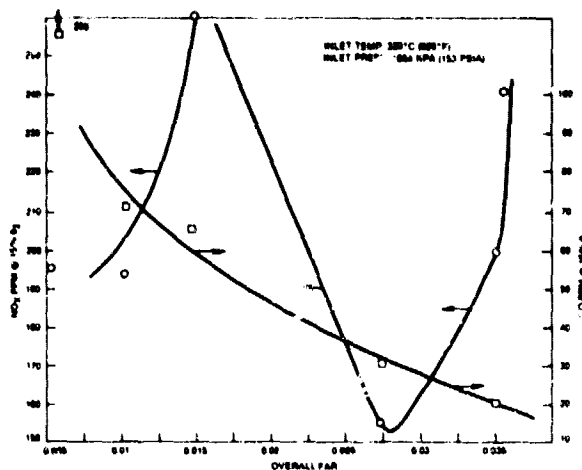


Fig. 22 Blue Water Gas Emissions Signature

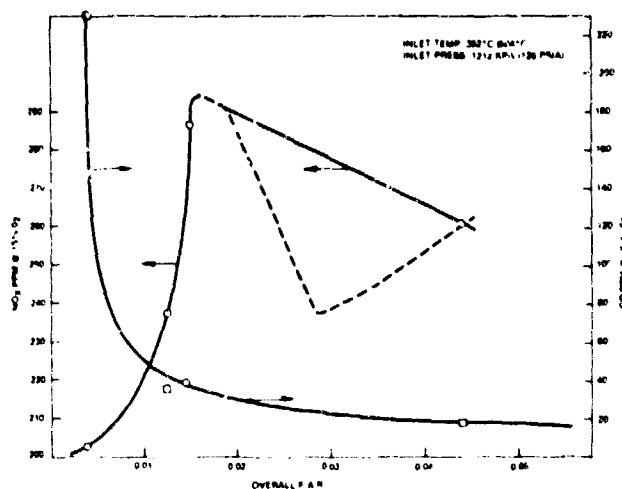


Fig. 23 Blue Water Gas Emissions Signature

tivity 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 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 over-

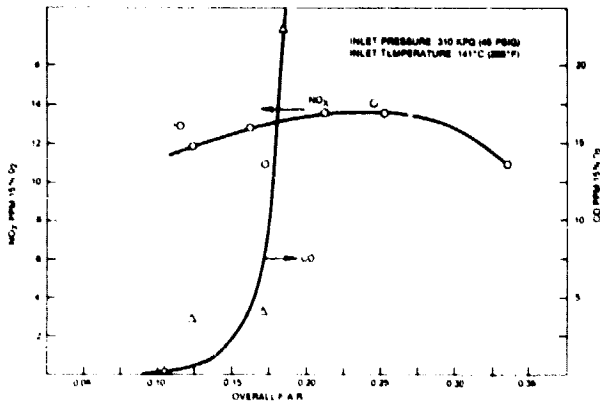


Fig. 24 Methane (CH₄) and Nitrogen (N₂) Mixture
Simulating Blue Water Gas Lower Heating
Value

heating 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).

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 (H₂). Operation of this combustion system involved lean primary zone operation for the low energy content fuel 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 NO_x levels. These emissions easily met the goals of providing NO_x levels below 75 ppm @ 15% O₂, 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 NO_x 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 NO_x. Thus a significant reduction can be claimed.

Emissions of NO_x during the combustion of the medium energy content fuel (a simulated Blue-Water gas) were above the goal except at low pressure conditions. The emissions signature showed the expected "rich-lean minimum" at rich primary zone conditions. The slope of the curve, however, between stoichiometric and the NO_x minimum fuel-air ratio was much steeper than that encountered with conventional petroleum fuels. NO_x emissions were expected to be high with the Blue-Water gas fuel, because of its high flame temperature characteristic. Thermal NO_x produced in the secondary zone probably dominated the entire emission signature. With increasing pressure and temperature, the shape of the NO_x emission curves remained essentially constant, however, their general level increased substantially. Insufficient data is available at present to extract the true

dependence of the NO_x 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 NO_x) were very sensitive to small changes in the hydrogen concentration, 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/H₂ 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 NO_x 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 NO_x 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, H₂, CO₂ and N₂) would have good stability and efficiency with a NO_x level below the goals and would be insensitive to operating conditions. This could be achieved by investigating fuels with differing CO/H₂ 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 NO_x characteristics. An investigation into the effects of methane content on NO_x production could readily be implemented, and could, as above, provide a guide to coal-gas producers as to which gases to produce.

ACKNOWLEDGEMENTS

This paper is based on work performed under contract DEN3-145. Funds for this contract were provided by the Department of Energy (DOE). Project Management was provided by the National Aeronautics and Space Administration (NASA).

REFERENCES

- 1 NSPS Standards, Federal Register, October 3, 1977, Revised September 10, 1979.
- 2 White, D. J., LeCren, R. T., and Batakis, A. P., "Low NO_x Heavy Fuel Combustor Concept Program", prepared under NASA contract DEN3-145, 1981.