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Solar Turbines Incorporated

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Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
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TABLE OF CONTENTS

1

SUMMARY

A total of twelve low NOx combustor configurations, embodying three different combustion concepts, were designed and fabricated as modular units. These configurations were evaluated experimentally for exhaust emission levels and for mechanical integrity. Many of the configurations were rejected because of mechanical integrity problems while others were modified to rectify problem areas. Emissions data were obtained in-depth on two of the twelve configurations.

Three fuels were utilized during the experimental evaluation; these being ERBS fuel (approximately Diesel No. 1), Solvent Refined Coal (SRC-II), and residual fuel oil. The SRC-II had fuel-bound nitrogen levels of the order of 1.0 percent by weight while ERBS had less than 0.02 percent and residual approximately 0.3 percent.

The three concepts evaluated were (1) a rich primary zone coupled to a lean secondary zone, (2) a modified conventional combustor and (3) a lean primary zone with a premixed fuel-air charge Of these, the first was intended to provide low NOx when burning fuels with high chemically bound nitrogen levels; while the lean premixed system was intended to provide ultra-low NOx with clean fuels. The modified conventional system was intended to provide both a basis to evaluate the performance of the other two and to investigate the effects of droplet size on emissions.

The emphasis of the program was placed mainly on obtaining low NOx with both residual and coal derived fuels (SRC-II) which have high levels of chemically bound nitrogen utilizing various versions of the rich-lean concept. NOx levels obtained with the rich-lean system although above the baseline program goals when burning SRC-II, were below the corrected EPA requirements for industrial gas turbines. The operating conditions of the combustors evaluated were typical of a 12:1 industrial gas turbine engine. EPA NOx limits for such engines were obtained by correcting the base limit of 75 ppm corrected to 15% O_2 for the nitrogen level in the fuel and for the efficiency of the overall system. Typical EPA limits for NOx for existing cogeneration/combined cycle engine systems are of he order of 300 ppm for SRC-II and 160-170 for ERBS type fuels.

2

INTRODUCTION

The specter of diminishing petroleum supplies in the United States of America, has spurred growing interest in utilizing coal-derived fuels, especially for utility and industrial gas turbines. Liquid fuels derived from coal are particularly suitable for gas turbines although they contain higher levels of fuel bound nitrogen than the equivalent petroleum fuel whose displacement is desired.

At present, on a national basis the stationary gas turbine engines used in industrial and utility applications provide a relatively small contribution to air pollution. Although those low levels represent a small contribution to the deterioration of air quality on a national scale, it can cause significant local concern, especially in the vicinity of engine installations where the background pollution level from other sources might already be objectionably high. In addition it is anticipated that considerably more gas turbines will be used in the future because of their inherent fuel flexibility. It is necessary, therefore, that means be developed for reducing the concentrations of undesirable exhaust emissions, such as NOx, from stationary gas turbines engines, particularly those burning fuels containing high levels of fuel bound nitrogen.

The Environmental Protection Agency has recently promulgated emission standards for stationary gas turbines. These standards are described in the Federal Register, Reference 1 and, at present are composed of three groups differentiated by the energy input level into the gas turbine. For engines with energy input levels below 10.7 GJ/h, no NOx emission limits have been imposed. Gas turbines with fuel energy inputs between 10.7 and 107.2 GJ/h are required to meet a limit of 150 ppm NOx converted to 15% O_2 for the next five years and then 75 ppm thereafter. Engines with higher than 107.2 GJ/h energy needs must meet the NOx level of 75 ppm corrected to 15% O_2. Emission regulations for CO and hydrocarbons are not presently required and do not appear likely in the future in light of high efficiency requirements normally imposed by the user.

Currently, the only means available for meeting these standards involves water (or steam) injection into the combustor. Unfortunately, this technique involves a capital cost of at least $10 to $15/kW and a fuel consumption increase on the order of 2 to 3 percent. In addition, increased maintenance costs are probable. For these reasons a system which can avoid the use of water injection is desirable.

The species that make up the total oxides of nitrogen (NOx) are nitric oxide (NO) and nitrogen dioxide (NO_2). Of the two, the concentration of NO is generally the higher. Other invisible gaseous species of concern consist principally of unburned polycyclic hydrocarbons and aldehydes. The variety

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of polycyclic hydrocarbons and aldehydes that have been identified in gas turbine exhaust streams is large and at present no regulations have promulgated for control Particulate pollutants are also produced in conventional gas turbine combustors especially when fuels with a high carbon to hydrogen ratio, such as coal-oils, are burned. The particulates of most concern are the soot-like matter generated in fuel-rich regions of the combustion chamber, which are discharged from the engine as a visible plume. Particulate materials in such plumes are dispersed widely. Most of the particles are essentially pure carbon, however, certain aromatic compounds are often found adsorbed or absorbed onto the particles. These latter compounds may possibly prove to be health hazards, and are presently under investigation.

Some of the earliest work on "rich-lean" combustors although not named as such was performed by White, et al., in 1973 (Ref. 2). This work defined the general shape of the NOx emissions signature as a function of equivalence ratio. Later work by Heap et al., (Ref. 3), on both liquid and solid fuels confirmed the general shape of these NOx emission curves. Work by Pierce et al (Ref. 4) has shown that low emissions can be achieved with a rich-lean type of combustor operating with fuels having a high bound-nitrogen content.

The goals of the work described herein were to obtain NOx emissions of 75 ppm corrected to 15% O_2 for fuel bound nitrogen levels up to one percent by weight and 37 ppm at 15% 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 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.

The data generated during the subject program confirmed the fact that the design procedures for low NOx combustors have been advanced. Novel approaches that provide effective rich primary zone cooling and rapid fuel vaporization have 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 low emission combustor 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 900°F passes into the combustor, vaporizing and mixing with the fuel in a short internal passage. By the time that the combustion reactions are initiated, the fuel is near fully vaporized and 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.

Table 1

Combustor Test Conditions Adopted

	Combustor Inlet Conditions	Engine Power Condition					
		Peak Load 118% Power	Baseload	70% Power	50% Power	Spinning Idle	Cold Start
kPa	Pressure, P_{in}	1240	1213	1055	910	303	103
°C	Temperature In T_{in}	376	361	334	308	143	Amb
°C	Temperature Out T_{out}	1057	982	882	810	546	649
J/kg(air)	Energy Fuel Ratio	921096	830382	732690	646628	588478	729895
%	Air Flow	103.7	100	90	81	33	6

Note: 100% air flow corresponds to 1.59 kg/s/can

A second contribution that has been provided is that of a lean secondary zone with a performance approaching that of a well-stirred reactor. Because of its unique design, the secondary zone can provide, within the torodial vortex flow reversal, a mass flow ratio in excess of one. This means that the flow reversal has associated with it a greater mass flow than the entering mass flow. A toroidal vortex reversal of this type provides particles with exceptionally long residence times which allows for the burn-out of the undesirable by-products of the rich primary zone. The torodial vortex which is like a smoke ring has a velocity profile cross-section similar to a free vortex. This type of vortex flow field tends to create a "trap" effect for particles, such as soot or smoke. Particles are retained in the vortex until the particle size has decreased to some critical value through oxidation. Paricles with dimensions below critical value can escape

The main purpose of the subject effort is to provide, at the end of the program a design of a prototype engine combustor which could handle fuels containing high levels of chemically bound nitrogen, with low NOx and smoke emissions that meet the proposed EPA standards for industrial use. This end-product was to be achieved by evaluating a variety of combustor configurations embodying several different concepts. These concepts included a premixed lean combustion system, a rich-lean concept, and a modified conventional system

Conventional in this sense refers mainly to the internal equivalence ratio values and zonal distribution within the combustor. It also refers to the injection of the fuel directly as a spray into the combustor without recourse to premixing of the air and fuel. The major problem that surfaced during the investigation and which eliminated the majority of the combustor configurations was overheating. In particular the integrity of the convectively cooled primary zone of the rich-lean system proved to be a great stumbling block in that local wall temperatures were higher than critical levels for reasonable life. Although the final design has limitations, it has been developed so that it can be operated over the maximum power range of a typical 12:1 pressure ratio industrial gas turbine engine, up to the maximum power point. The main limitation of the final developed design is that there is a limiting inlet air temperature above which operation is impossible due to incipient wall failure. Design modifications of the primary zone convective cooling system have the potential of overcoming this limitation to allow operation at any inlet air temperature. Once a design is fixed, however, operation at inlet air temperatures above the design maximum would be difficult.

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 while maintaining near constant pressure drop. Fuel staging between zones although potentially viable has proven to be difficult in implementation due to mechanical integrity problems. These latter problems consist of severe overheating of the secondary zone forward dome and of the transition piece adjacent to the dome.

Potential designs of an available geometry rich-lean combustor are provided in later sections of this report.

3
COMBUSTOR DESIGN

3.1 PROGRAM GOALS

The primary goal of this program was to design, develop and demonstrate 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 simulated those that would be produced at the combustor by a nominal 12:1 industrial gas turbine. Table 3 shows the operating conditions generated for a typical 12:1 pressure ratio engine that were adopted for test purposes.

A set of twelve different combustor configurations were to be evaluated. These are listed in Table 4. Of these twelve, the prime approach was the fuel rich primary/lean secondary zone (rich-lean) system with a near premixed fuel and air charge. Because of the potentially narrow fuel-air ratio low NOx operating band, various methods of extending the range have been included, notably fuel staging and simulated variable geometry. Other approaches such as lean primary systems were also included primarily to obtain ultra low NOx with distillate fuels and also to provide "baseline" NOx levels for the heavy fuels.

The purpose of these goals was to provide the technology and design data necessary to provide, at the end of the program, a design of a prototype engine combustor. This recommended combustor would, in essence, be the proposed configuration for Phase II of the overall program.

3.2 DESIGN PHILOSOPHY

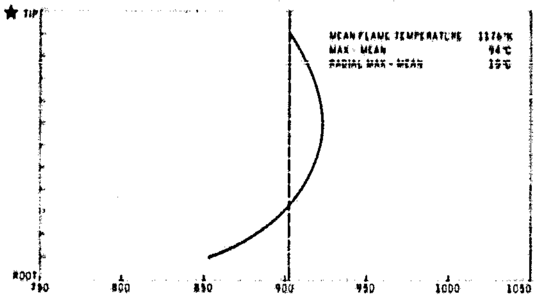
3.2.1 General Description of Configuration

The twelve configurations described above encompass only three basic concepts, these being;

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	



- Rich Primary Zone Coupled to a Lean Secondary Zone
- Lean Premixed Primary With or Without Lean Secondary Zone
- Lean Semi-Conventional Primary Zone

Of these three the rich-lean system is the main concept that represents "new technology". Thus the design philosophies of each of the three concepts discussed below concentrates mainly on this latter concept.

A total of four configurations, involving staged combustion (rich primary zone coupled to a lean secondary zone) were postulated as potential low NOx systems. These four although of fixed geometry were designed so that simulated variable geometry could be employed. It was felt that the goals of the program were to develop the technology for low emissions, not to develop means to surmount the technical problems of variable geometry linkages and alignments. Thus, no true variable geometry was designed. Of these four configurations (see Table 4) the one that was considered the most promising

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}	1310	1213	1055	910	303	103
Temperature In T_{in}	376	361	334	308	143	Amb
Temperature Out T_{out}	1057	982	882	810	546	649
Energy Fuel Ratio	220	198	175	154	140	178
Air Flow	103.7	100	90	81	33	6

Note: 100% air flow corresponds to 1.59 kg/s/can

was configuration 1-2, which had a rich primary zone with an air-assist atomizer coupled to a lean secondary zone with opposed jet mixing. The selection of the various design features for this configuration illustrate well the evolution of the various combustor designs.

Essentially the primary zone design evolved as a compromise between wall cooling requirements, and the provision of a near premixed fuel/air charge without autoignition. Having decided on augmented (trip-strip) convective cooling for the primary zone it was realized that the heated air could be used to help vaporize the heavy fuels proposed for use on the program if the cooling air were arranged in a reverse flow fashion. It was also realized that a radial inflow swirler was the best fit to direct this heated air into the primary zone as this was the only system that could be arranged to readily turn the air through 180 degrees with minimal losses. Thus swirl stabilization was chosen over heat stabilization techniques more for its fit with the cooling system than any other reason. The mixing section for the primary air and fuel was the annulus (annulii because of their small radial dimensions per unit flow area are ideal for mixing ducts) formed by the fuel injector (as a centerbody) located in the swirler exit. Because of the high air temperature, this mixing length was relatively short, although a degree of variability was built into the design by allowing the swirler exit length to be increased if necessary.

Table 4

Combustion Concepts and Configurations

FIRST CONCEPT

1. Staged Combustion (Rich Primary Zone - Lean Secondary Zone)

CONFIGURATIONS

1.1 Rich primary zone with spinning-cup and central recirculation coupled to a lean secondary zone with opposed jet mixing.

1.2 Rich primary zone with air atomizing fuel injector coupled to a lean secondary zone with opposed jet mixing.

1.3 Rich primary zone with spinning-cup and central recirculation coupled to a lean secondary zone with reduced area transition piece mixing section.

1.4 Rich primary zone with air atomizing fuel injector coupled to a lean secondary zone with reduced area transition piece mixing section.

SECOND CONCEPT

2. Staged Combustion and Staged Fuel Injection (Rich Primary Zone - Lean Secondary Zone)

CONFIGURATION

2.1 Rich primary zone with spinning cup and central recirculation coupled to a lean secondary zone with opposed fuel/air mixed jets.

THIRD CONCEPT

3. Lean Combustion with Direct Fuel Injection

CONFIGURATION

3.1 Lean primary zone with spinning cup fuel injector coupled to a lean secondary zone.

3.2 Lean primary zone with air atomizing fuel injector coupled to a lean secondary zone.

FOURTH CONCEPT

4. Lean Combustion With Fuel Staging

CONFIGURATION

4.1 Lean primary zone with spinnng cup and/or air atomizing fuel injector coupled to a lean secondary zone with port fuel injection.

FIFTH CONCEPT

5. Lean Premixed Combustor/Fuel Staging

CONFIGURATION

5.1 Lean premixed primary zone with multiple premixing ports.

5.2 Lean premixed primary zone coupled to a lean premixed secondary zone (fuel injection in both).

5.3 Lean premixed secondary zone coupled to conventional primary zone

SIXTH CONCEPT

6. Annular Staged Combustion

CONFIGURATIONS

6.1 Annular rich primary zone with stratified fuel injecton coupled to an annular lean secondary zone employing jet mixing.

6.2 Annular rich primary zone with air assist atomization coupled to an annular lean secondary zone employing jet mixing.

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The premixing of the air and fuel is necessary as the low NOx operating range is generally found to occur over a narrow range of fuel/air or equivalence ratios. Thus a system that provides a non-premixed charge would tend to produce a range of equivalence ratios around the mean value, thus raising the overall NOx level. The equivalence ratio chosen for the primary zone was 1.3, which represented a level compromise between the data shown in Figure 1 (Ref. 5) and that of Table 5. Figure 1 indicates that low fuel bound NOx conversion occurs between values of 1.25 and 1.45 for the primary zone equivalence ratio. Table 5, however, shows that smoke is likely to be produced at equivalence ratios of 1.45 and greater, thus 1.3 was selected as a reasonable compromise.

Mixing between the rich primary zone exhaust and the secondary had to be sufficiently rapid to prevent any significant residence time at stoichiometric conditions. The mixing must take place faster than the chemical reactions at those particular stoichiometric conditions if all local high temperatures, and high local NO production rates are to be avoided. To aid in this problem the intent was to try to provide carbon monoxide (CO) as the main unburned material exiting from the rich primary zone, and to minimize other species. Carbon monoxide reacts relatively slowly and by providing rapid mixing flow arrangements, it should be possible to mix sufficiently quickly to avoid the rapid reaction of CO. One of several approaches that were proposed is the opposed jet mixing system. This involves direct jet on eject impact mixing. In general, mutual impingement of two gaseous jets is unstable although mixing rates are known to be high in those instances when, through careful control of

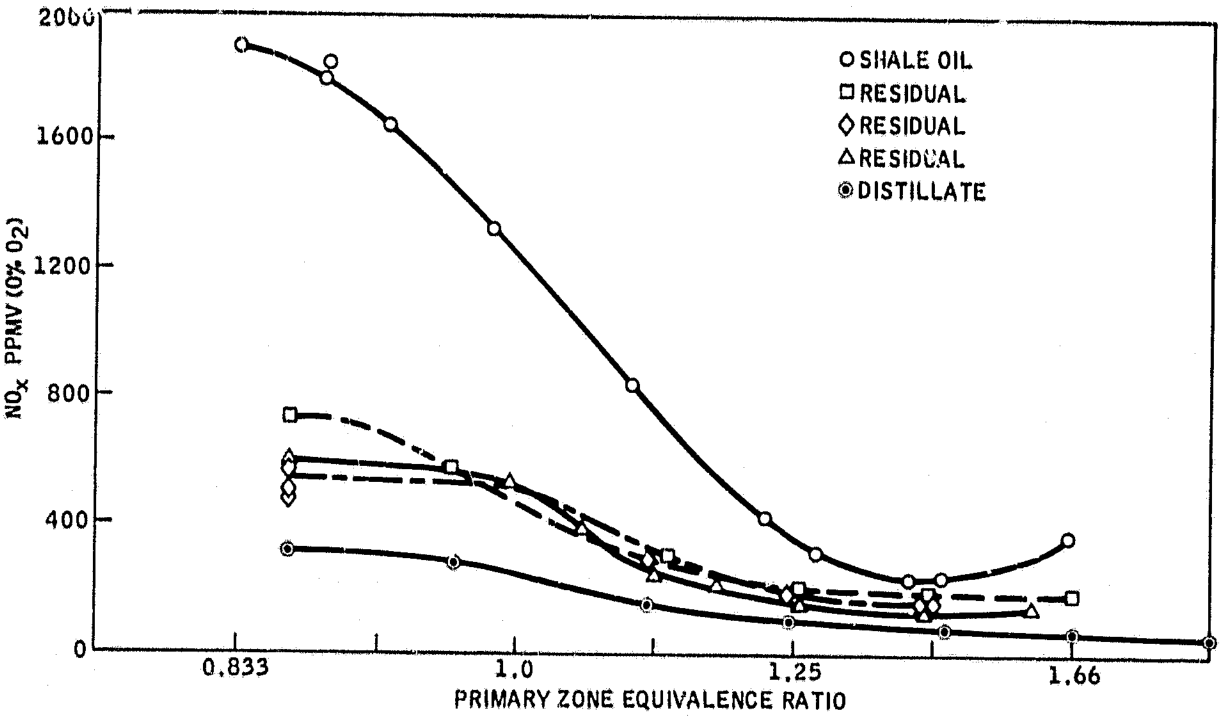

Figure 1. NOx Emissions Signature for Various Heavy Fuels

Table 5

Carbon Formation Limits for Various Fuels

Fuel	Critical O/C Ratio for Incipient Carbon Formation		Equivalence Ratio for Carbon Formation at Well-Stirred Conditions
	Pre-Mixed Bunsen Flame	Well-Stirred Condition	
Ethylene	1.67	1.43	2.11
Propylene	1.69	1.40	2.16
Butene	2,08	1.48	1.04
Benzene	1.75	1.75	1.538
Toleune	1.92	1.71	1.5
Xylene	2.08	1.80	1.46

the jet velocity profiles, stable flow have been achieved. As designed, a series of air jets at the rear of the secondary zone were arranged to mutually impinge at an angle on the centerline of the secondary zone and the derived jet resulting from this impact was constrained to impinge directly with the exhaust from the primary zone. This was based on the evidence of Figure 2 which shows resulting mixed levels for jet on jet impact, will achieve the required mixing rate.

The mixing factor quoted in the above figure is the actual level of one of the components divided by the level, if fully mixed, expresed as a percentage. The correlating parameter on the abscissa is a modified momentum ratio with ρ being the density, V the velocity and d the diameter of the jet.

In addition to providing rapid and near complete mixing, the jet arrangement in the secondary produces a strong toroidal vortex having a recirculation ratio greater than one. This means that there are packets of fluid in this vortex with ages greater than the mean residence time, this tends to allow equilibrium to be reached in a smaller volume device. This vortex also has the velocity characteristics of a free vortex, which provides the phenomenon of "trapping" particles or smoke for long periods of time. This is due to the fact that in a free vortex, the tangential velocity increases as the inverse of the radius to conserve angular momentum. A particle of a given size in such a flow field will find a circular orbit in which it stays suspended and in which it rotates steadily, the radius of the orbit being that at which

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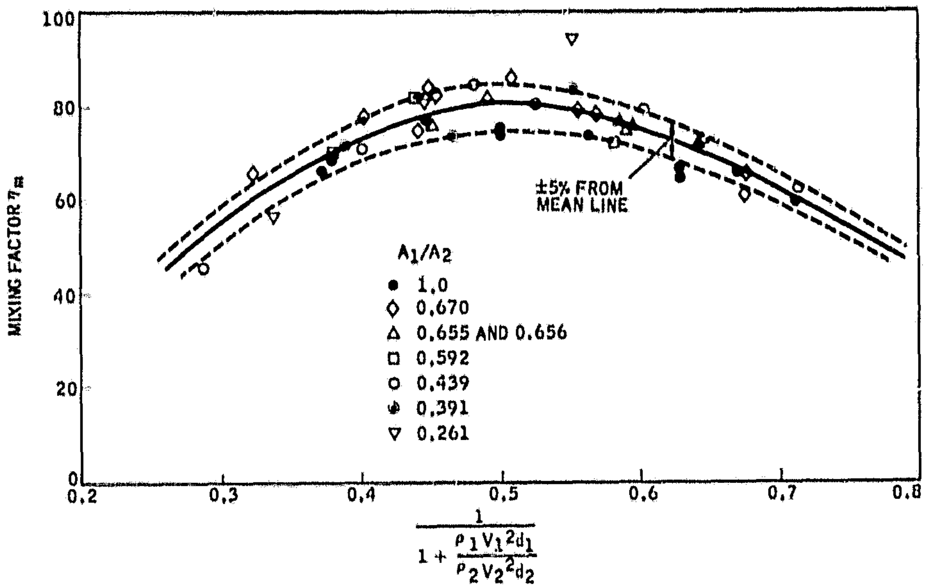

Figure 2. Mixing Efficiency as a Function of Modified Momentum Ratio

centrifugal forces of the particle spinning with the air is equal to the inward drag force on the particle caused by the relative inward radial flow of air past it (see Fig. 3).

As the particle decreases in size it will move inward and at some critically small dimension it will escape and will be swept out of the secondary zone.

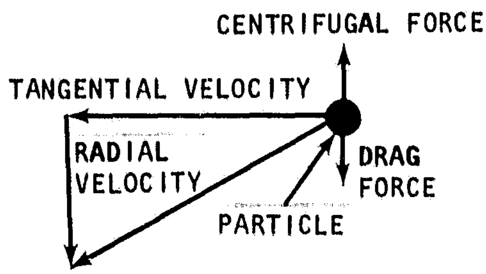

Figure 3. Forces on Particle in a Free Vortex

It should be noted that the ports producing these secondary air jets could have their diameters changed by inserting inserts or sleeves with the desired flow areas. This then allowed changes to the primary/secondary flow split.

Of the three related configurations, 1.1 simply had a spining cup atomizer instead of the air assist system. Configuration 1.3 and 1.4 involved injecting the secondary air into the throat or transition piece located between

the primary and secondary zones. This involved conventional multiple radial jet mixing rather than the opposed jet system of 1.1 and 1.2. This transition piece utilized, in essence, design technology derived from that developed for dilution zones. The main difference being that flow back toward the dome had to be avoided or at least minimized. This was accomplished by designing a slim triangular hole jet system with the apex toward the dome. Because jets produced by such a hole tend to bend downstream before they mutually impact, only a small reverse flow derived jet is produced. An additional approach was also designed and this involved injecting the air in a series of jets semi-tangentially so that no centerline interaction was involved. A swirl was thus introduced and this was oriented in opposite to that of the gases swirling out of the primary zone so as to maximize the mixing rate between the hot primary exhaust and the cold secondary air. The difference between 1.3 and 1.4 was that 1.3 had a spinning cup fuel injector and the 1.4 air-assist system.

A fifth related configuration 2.1, utilizes the configuration of 1.2 modified to allow fuel injection into the secondary zone via the ports, in a similar manner to that used with the lean premixed combustor, which is described below. This secondary zone fuel injection was intended to allow the low NOx operating range to be extended by staging fuel between the primary and secondary zones. Such staging could allow the primary zone equivalnce ratio to be maintained near constant over a wide range of overall fuel-air ratios. There would be the disadvantage however that the thermal NOx produced in the secondary zone could increase. This effect could offset any advantages offered by near constant primary zone equivalence ratio.

Configurations 3.1 and 3.2 are essentially versions of 1.1 and 1.2. The major difference between the configuration of concept 3 and concept 1 is the primary zone equivalence ratio. In configuration 3.1 and 3.2 the primary zone swirler was larger than that of 1.1 and 1.2 and the associated increased airflow was designed to move the design point equivalence ratio from 1.3 to something between 0.8 and 1.0. Fuel injection systems for these two configurations also had to be changed to match the larger high flow swirlers. Configuration 3.1 had spinning-cup injector, while 3.2 had an air-assist system, both similar to those used on configuration 1.1 and 1.2.

The fourth concept (single configuration) was essentially a fuel staged version of 3.1 or 3.2 much the same as configuration 2.1. This fuel staging was intended to extend the range (if needed) of the low NOx operating point. Fuel staging was considered to be a more practical alternative to variable geometry.

Three configurations were viewed as potential low emissions systems for the fifth concept which was a lean premixed combustion system. This general concept was intended to provide ultra-low emssions for the combustion of clean fuels (low fuel-bound nitrogen levels). Configuration 5.1 was a basic lean premixed primary zone system based on the extensive past work in this area that Solar has accummulated. In essence the basic premixed combustor used the jet induced circulation (JIC) principle, in which multiple forward angled jets converged on the centerline of the primary zone to provide a major derived jet moving toward the primary zone dome. On impinging on the

dome this jet is constrained by the dome and the walls of the primary zone to reverse its direction and flow rearward along the primary zone walls. A large part of this fluid is then entrained by the entering jets providing a strong recirculation of combustion products for exceptional stability The ports that produce the jets in this concept double as fuel-air premixing ducts. Fuel is injected at the entrance to the ports where it undergoes air-blast atomization. It then vaporizes and mixes with the air as it passes down the ports which are sized to provide near complete vaporization and mixing without autoignition. Fuel injection is accomplished via a number (four) of small tubes located in the bell-mouth inlet to the ports. Because of the rapidly accelerating flow field in the bell-mouth and because of the high air to fuel ratios involved, excellent atomization quality can be achieved (SMD approximately 75 micron) The ports or mixing ducts were designed to have a smooth bore and blends of no more than 30 degrees to prevent secondary flows. Any large drops that pair into the primary zone are constrained to remain there until they vaporize and/or burn This is due to the nature of the toroidal vorex produced by the jets. The velocity profiles within the vortex are essentially similar to those found in a free vortex, in which the velocity varies as the inverse of the vortex radius. This form of velocity profile provides, for a given size drop, a circular orbit where the centrifugal forces of the particle are balanced by the inward drag forces. Thus a drop tends to spiral inward vaporizing and burning until it reaches some critical diameter at which it can escape. Usually this phenomenon takes place rapidly as there is a strong relative air velocity with respect to the droplet which causes rapid vaporization and combustion. With reasonable premixing of the fuel and air, and maintaining the primary zone at an equivalence ratio of less than 0.55, very low NOx emissions can be obtained of the order of 20 to 30 ppm corrected to 15 percent O_2 with clean fuels.

Two other versions of the lean premixed system were also proposed. They were designed to provide an extension to the narrow range of equivalence ratios that low NOx can be obtained. Configuration 5.2 utilized fuel stabilization to accomplish the range improvement. In this system two lean premixed "primary zones" were joined serially to allow low power, low NOx operation with the true primary zone (no fuel in the secondary) and low NOx full power operation with both primary and secondary zones fueled

The last of the lean premixed configurations used a pilot zone consisting of a small conventional primary zone attached to the dome of the lean primary zone. Although the conventional primary zone could produce high NOx levels, it would only burn approximately 10 percent of the fuel and thus its total contribution to NOx would be small While this arrangement would not provide low NOx at the low turndown condition, the engine could be operated without undergoing lean extinction. The lean extinction limit bounds the low NOx primary zone equivalence ratio operating range and limits the fuel flow turndown. Most engines require a fuel-air or equivalence ratio operating range of 3 to 3.5:1, whereas the low NOx fuel-air ratio range is of the order of 2:1 for a lean premixed system.

The final concept and configuration was an annular system which can best be visualized by considering it to be configuration 1.3 with a centerbody The

main advantage of an annular geometry is that enhanced mixing is possible in the transition between the primary and the secondary zone. This would enable the secondary air to be more readily mixed with the primary exhaust.

The general design follows that of concept 1 very closely, in most ways, and for the design philosophy refer back to configuration 1.1 through 1.4. This particular combustor, although designed, was never fabricated or tested because of its high manufacturing cost. It was the only configuration that could not be assembled form the basic modules that were used for the other configurations.

To minimize cost and manufacturing delay times it was decided that a modular approach to the construction of the combustor would be adopted wherever possible. In essence this meant that each combustor concept would be assembled from a set of "standard" or common parts.

This modular approach was provided with some extra impetus when it was realized that the lean secondary zone of the rich-lean system would have to be virtually identical in size to the primary zone of the lean premixed primary type combustor.

It was also realized that the rich primary zone module could be operated in a lean mode by installing the requisite swirler and could then be operated in a semi-conventional fashion. Thus by providing one rich primary zone module, and one lean secondary/primary zone system, coupled to a common dilution zone piece, all three combustor concepts could be assembled from the same set of modules. To completely accomplish this, two different swirlers had to be provided together with transition pieces between the rich and lean zones. Unfortunately the fuel injection systems did not lend themselves to modular construction and had to be designed to fit the particular concept. The rich primary zone had, for example, two optional fuel injection schemes, a spinning cup and an air-assist atomizer. Two swirler fuel injectors were also designed for the "rich" zone when it operated lean. Fuel injection on the lean premixed approach was of the air-blast type and was designed as an integral unit with the fuel/air mixing ports or ducts.

In summary, the following modules and components were designed so as to complete the construction of the three concepts.

- Rich Primary Zone (includes cooling system)
- Rich Primary Zone Swirler
- Torch Igniter System
- Lean Primary Zone Swirler (converts rich primary zone module to a lean one)
- Lean Secondary/Primary Zone
- Transition Pieces (secondary air injection)
- Diluton Zone (variable length)

- Rich and Lean Spinning-Cup Fuel Injector

- Rich and Lean Air-Assist Atomizer Overlip (two versions)

3.2.2 Rich-Lean Combustor

This particular combustor as envisioned above would consist of the rich primary zone module, rich primary zone swirler, rich fuel injection systems, torch igniter, transition piece, lean secondary/primary zone, and dilution zone. The design philosophy for the primary zone reflected the problems imposed by the use of a rich primary zone to achieve low NOx, these being:

- wall cooling (without using air-film cooling)

- accurate maintenance of primary zone fuel/air ratio (for Low NOx)

- premixing fuel and air without autoignition

- rapid mixing of primary exhaust with the secondary air

- long residence time lean secondary zone for low temperature burnout of particulates

<u>Wall Cooling</u>

Wall cooling in a rich primary zone is a problem in that conventional air-film cooling cannot be utilized, because of the potential creation of local hot spots due to the formation of stoichiometric conditions near the wall. To satisfy the requirement to externally cool the primary zone, an augmented convective cooling approach was decided upon. This cooling scheme was based on existing in-house designs and consisted of an annular passage surrounding the primary zone with a series of "trip-strips" on both passage walls. These trip-strips (see Fig. 4) consisted of a helical wire coil brazed to each of the annular internal surfaces. In operation the wire trips the boundary layer, and thus provides a local wall velocity considerably higher than would be obtained without the strips. By having the trip-strips on both surfaces it prevents the velocity profile from becoming unduly skewed. This construction effectively produces, in the cooling annulus, a near top-hat type velocity profile, which maximizes the local convective heat transfer coefficients.

In designing the cooling system a longitudinal temperature profile for the wall was first assumed and then, with a knowledge of the internal combustion temperatures and the air inlet temperature, a thermal transfer rate was calculated. The temperature values adopted initially and the calculated heat transfer rates are shown in Figure 4. These heat transfer rates were then compared with experimentally determined levels obtained from earlier in-house rich-lean combustion systems. This latter data was unfortunately obtained with natural gas as a fuel and estimates had to be made as to the differences

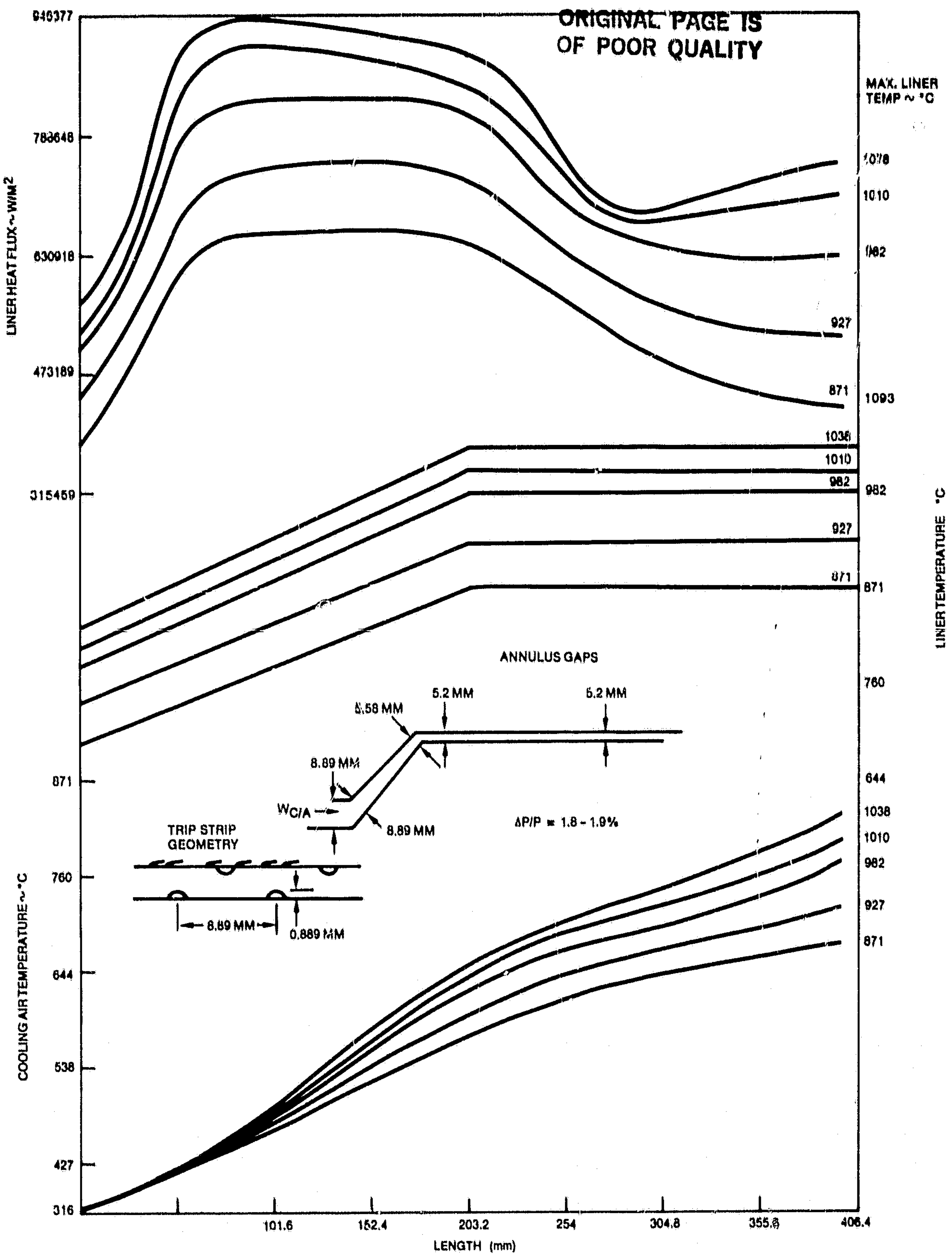

Figure 4. Combustor Cooling Annulus With Trip Strip - Preliminary

between natural gas and SRC-II. Initial estimates showed that the calculated heat transfer rates for the primary zone length adopted and the estimated rates from experimental data were approximately equal. Thus the trip strip system was adopted.

Fuel Air Premixing

In addition to cooling the combustor it was decided to utilize the preheated air for the primary combustion air. By passing the preheated primary air over the injected fuel, rapid vaporization of even the "heaviest" of fuels could be accomplished. Effectively the premixing section for the air and fuel is reduced in length considerably, the requirements being of the order of one inch or less. This of course requires the air introduction method and the fuel injection technique to be integrated to ensure that the fuel and air come into intimate contact as quickly as possible. To accomplish this a radial inflow air swirler was used to direct the preheated air from the convective cooling passage into the combustor proper. See Figure 5 for the general arrangment. The exit from the air swirler was arranged to be an annulus surrounding the fuel injector. Fuel injected by the two selected methods, air assist or spinning-cup, is sprayed radially outward (at an angle) into the swirling heated air. It vaporizes (flash vaporization) and mixes rapidly with the air before it ignites and burns, as a near premixed air-fuel charge.

As shown in Figure 5, all the primary air enters the annular cooling passage at the rear of the primary zone and flows forward, entering the combustor proper via the radial inflow swirler which turns the flow 180 degrees. This preheated air (circa 540°C [1000°F]) is brought into initimate contact with the fuel as mentioned above.

Rich Primary Zone Geometry

It was postulated, as a basis for the rich primary zone design, that the reactions were kinetically controlled. This was justified by arguing that the fuel would be vaporized rapidly by the preheated inlet air (preheated in the regenerative cooling of the walls), and the fuel air charge would be nearly completely premixed, before the reactions commenced. Thus mixing of the fuel and air as a rate controlling step would be eliminated.

If this assumption is valid and a bimolecular reaction is assumed, then a global combustion efficiency (η_c) correlation of the following form can be derived

$$\eta_c = f\left(\frac{P^2 * Vol * \exp(T/K)}{W}\right) * (f.a.r.)^c = \theta$$

where K is a constant of the order of 300
and c is a constant in the range of 0.75-1.0

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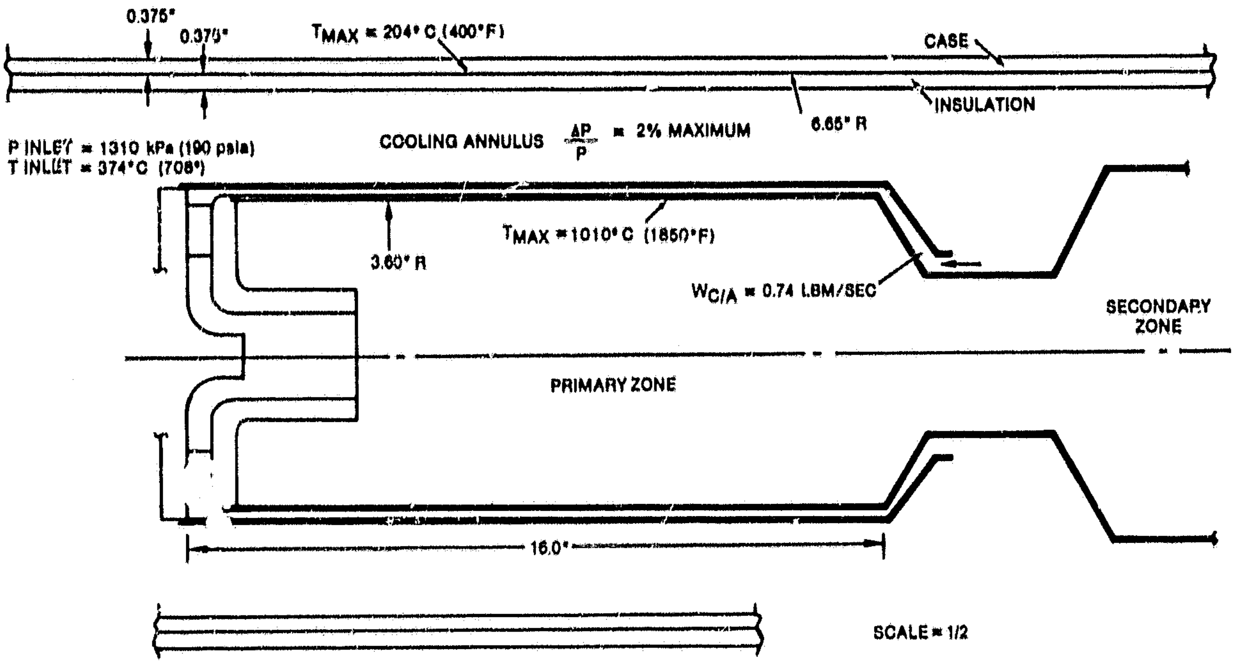


Figure 5. Combustor Primary Zone

By using values of the inlet pressure (P), temperature (T), mass flow (w), fuel/air ratio (f.a.r.), and combustor primary zone volume associated with existing or previously evaluated combustors, a correlation of θ versus combustion efficiency (η_c) can be produced. Because of the lack of data on rich primary zones it was possible only to provide an approximate value of θ for 99.9 percent efficiency. The work of S. A. Mosier et al. (Ref. 4) was utilized to estimate a value of x as defined above. The value of θ, using K = 300 and c = 0.62 derived from the above work, was approximately 101 atm^2-s-m^2/kg. For the design of the proposed rich primary zone, the desired operating conditions at maximum power were input into the θ correlation to obtain a value for the combustor volume. A value of the fuel-air ratio corresponding to an equivalence ratio of 1.2 was used based on the work of Heap et al., Reference 3.

This volume was taken to be the minimum allowable primary zone volume. A value of the length to diameter (L/D) ratio was chosen to be 1.1 initially, based on past experience. This length however, was found to be insufficient to provide a residence time of the order of that recommended in the work of Reference 4. The residence time with a length to diameter ratio of 1:1 was approximately 30 ms based on the combustor inlet conditions. Data provided in the above reference, showed that a longer residence time was required to provide the lowest possible NOx levels. To achieve this, the diameter as defined by the assumed 1:1 ratio was adopted and the length simply increased to provide an overall length to diameter ratio of 2:2. This provided a residence time of approximately 60 ms. With an inside diameter of 184 mm (7.25 in.), this produced a primary zone of 405 mm (16 in.) long.

Figure 6. Rich Primary Zone Module (Side View)

This calculated length of the primary zone was used in the wall cooling calculations discussed above, which showed that it was potentially possible to operate without catastrophic overheating of the walls.

The pressure drop over the primary zone included both the frictional pressure loss produced by the air flowing through the cooling passge and the combined frictional and momentum losses incurred by the air swirler and the primary zone itself. The pressure loss from the inlet of the swirler to the primary zone throat, was designed to be (at the maximum power point) approximately 3.2 percent of the combustor inlet pressure. The pressure drop in the cooling annulus amounted to some 2.3 percent providing a total design point pressure loss of 5.5 percent. It should be noted that the throat pressure loss was designed to be small, less than 0.5 percent. This low throat loss was utilized to prevent the pressure drop from varying significantly with combustion temperature. If the design involved a large throat pressure drop (which was based on a particular primary exhaust temperature) then, as this temperature varied, so would the pressure drop. This could change the mass flow splits of the combustor significantly. Generally, as the primary exhaust temperature increases the throat pressure drop would increase, which in turn would tend to decrease the air flow through the primary zone. If the zone is designed to be rich, this decreasing mass flow-effect would tend to decrease the temperature and thus it would be self compensating to a certain extent It would, however, tend to force the primary zone into a smoke producing region of high primary zone equivalence ratio.

If the zone were operating lean, the presure loss affect would tend to move the primary zone equivalence ratio toward stoichiometric, thus worsening the situation and increasing the potential for wall failure.

Thus a low pressure loss potentially involving only a small change in mass flow, would on a percentage basis, have less impact on the mass flow distribution than a high pressure loss.

Rich Primary Zone Swirler

The radial inflow swirler design was based on data generated in-house during the development of a small portable gas turbine emergency generating set. This latter unit (the Gemini™) utilizes a radial airflow swirler, of the type adopted, mounted on a single can-combustor. Design data for the swirler is thus considered proprietary, although it does not differ markedly from data available in the literature (see Ref. 5). It was machined from a block Inconel 600, with thick end walls (see Fig. 7). It was designed to have a swirl number of the order of 1.0 thus ensuring vorex collapse and recirculation. In addition it was designed to provide an exit swirling air cone angle of approximately 60 degrees.

Figure 7. Radial Inflow Swirler for Rich Primary Zone (Exploded View)

Torch Igniter System

The ignition system used was a slightly modified Centaur™ engine torch-igniter. This particular unit was operated on natural gas for ease of ignition and general control. In operation the "flame" existing from the torch was constrained to pass through the radial inflow swirler (made from Inconel 600), and thus come into intimate contact with the fuel. This "hot-shot" ignition system is similar in concept to that employed by aeroengine designers to ignite afterburners. The main reason for the adoption of this ignition method was to avoid the creation of an access port through the double wall of the rich combustor. Such a port would be a potential leak point where air could pass into the rich primary zone and create local stoichiometric conditions with subsequent local wall failure.

3.3 LEAN PREMIXED COMBUSTION

In the design of a lean premixed combustor, as in the design of any combustion system, two sets of parameters can be considered as crucial in dictating the ultimate geometric configuration. These two sets are the required operating conditions and the desired performance characteristics, respectively.

By utilizing experimentally obtained correlations of all of the various performance parameters with "correlation functions" composed of combinations of operating conditions and geometric functions, a preliminary combustor design can be obtained. In general, correlation functions that describe performance parameters are thus the prime method by which the geometry of the lean premixed combustor system is obtained. The geometric results obtained from such correlations however, must, in general, be checked to ensure that they do not fall outside of certain limits which are set to prevent extrapolation of the experimental data. These latter limits have usually been empirically determined. Other secondary limits, based on known phenomena such as autoignition and flashback, that could affect performance or mechanicl integrity must also be included.

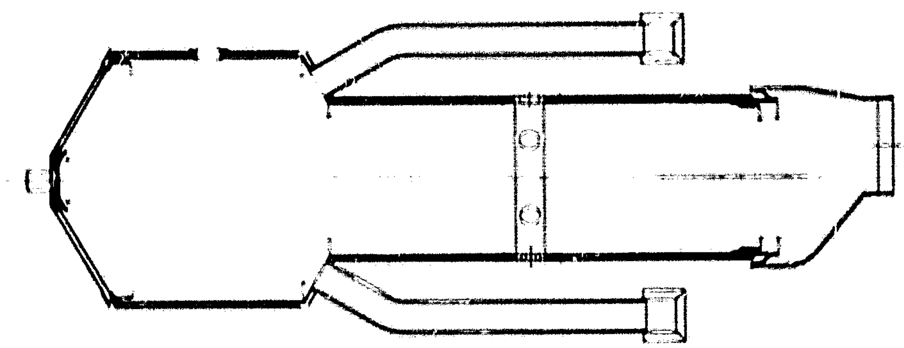
The general configuration of a JIC combustor is shown in Figure 8. For such a combustion system, it has been convenient to divide the mechanical design into three basic areas, whih can be described as:

(a) bulk geometry determination

(b) open hole area and hole distribution estimation

(c) fuel/air mixing section dimensions determination

The definition of the geometry associated with each of these areas is a complex procedure which involves interactions between all three of the above areas and both sets of the input parameters, described earlier.



JIC

Figure 8. Geometric Size of a JIC Low Emission Combustor

3.3.1 General Design Method

The "logic flow" involved in the JIC combustor design procedure is shown in simplified form in Figure 9. The three major columns shown cover the three basic geometry defining areas described above (bulk geometry, hole area and fuel/air mixing duct). Each utilizes the combustor operating conditions shown at the head of the columns and the performance requirements shown (as entering from the right) to determine, at the first level of the calculation procedure (top row), the various key geometry correlation parameters. At subsequent calculation levels (lower rows) more detailed definition of the geometry is obtained.

Calculation of Combustor Geometry

The first column of Figure 9 shows the logic used in determining the volume of both the primary and secondary zones, and subsequently the lengths and diameters of each of these combustor sections (Fig. 8).

To size the primary zone of a JIC combustor, it must be determined, from past NOx emissions results with the fuel in question, what value of the primary zone equivalence ratios (ϕ_{pz}) is required. A plot of a typical NOx emissions curve used for the purpose of designing the combustor is shown in Figure 10. To maintain reasonable NOx emissions levels (below 75 ppm corrected to 15% O_2), primary zone equivalence ratios at the desired low emission point have to be within the range of 0.35 to 0.55. This latter equivalence ratio value proves some operating range in terms of turndown before blow-out occurs. It should be noted that with lean premixed primary zone combustion systems low levels of NOx can only be obtained at conditions near to the lean extinction limit (levels of CO however can be arranged to be low).

Using this value of the equivalence ratio and the inlet conditions that provide the minimum value of θ', where $\theta' = (p^{1.75} * \exp T/540)/w$, a volume is

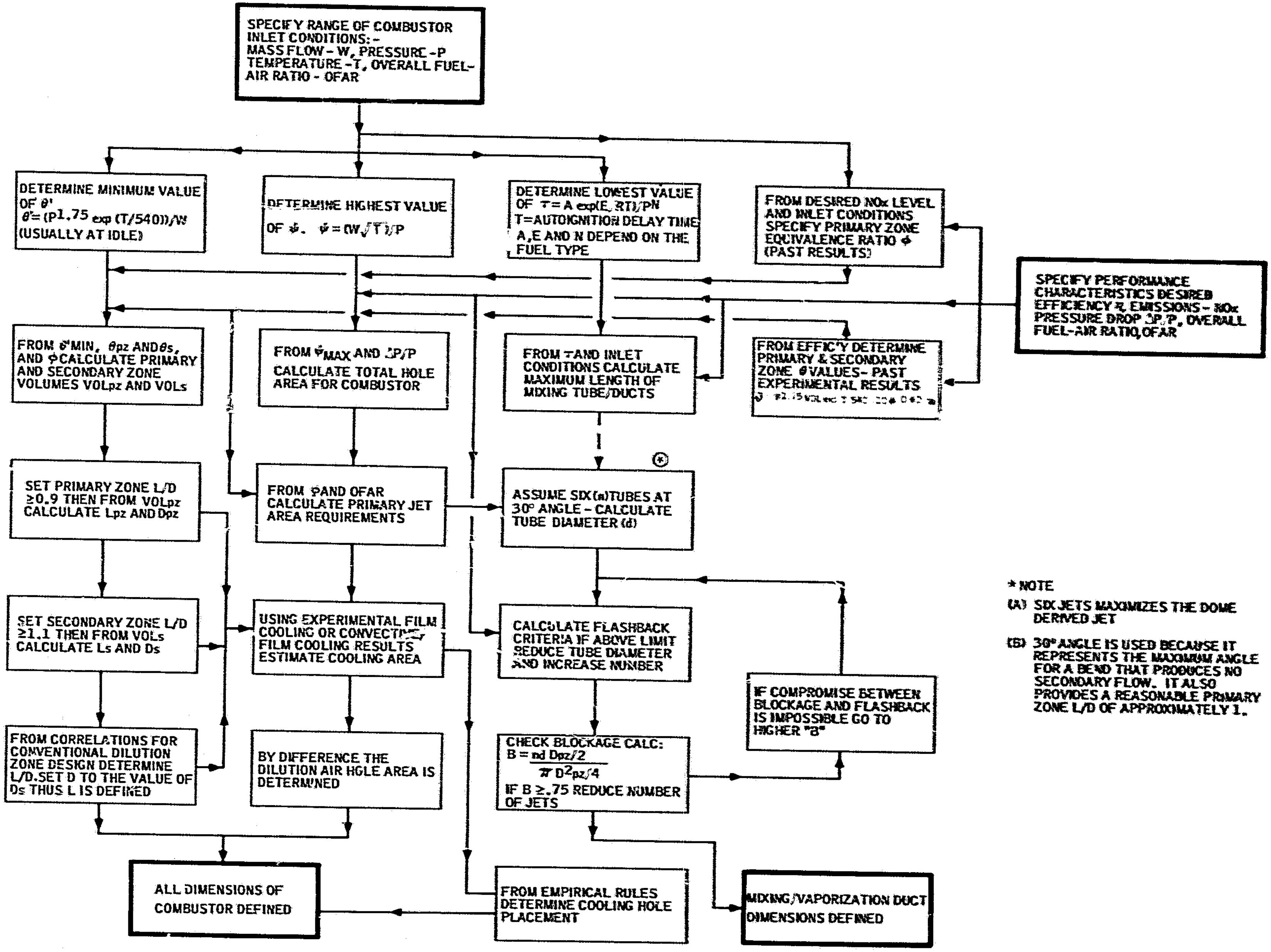


Figure 9. Design Logic for a Lean JIC Type Combusor

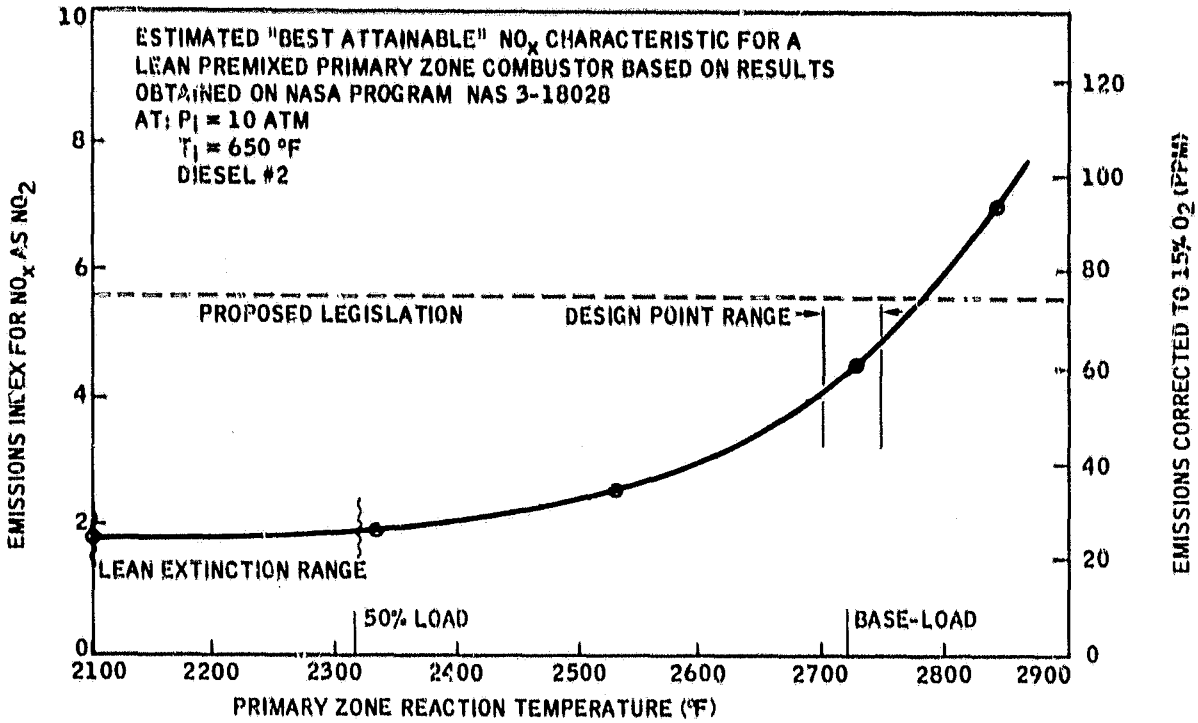


Figure 10. Typical Lean Combustor NOx Characteristics

calculated from an experimentally determined value

$$\theta = \frac{P^{1.75} * vol * \exp(T/540)}{W} * (10 * \phi)^{0.62}$$

If the operating conditions for minimum θ' differ from those at the desired low emission design point then ϕ_{pz} must be adjusted accordingly to the value associated with that point. (Assume a constant combustor air flow split, initially derived from the low emissions point, desired ϕ_{pz}, and the overall fuel/air ratio at the same point.) Using a semi-empirical ratio for the length to diameter of 1.5 for the primary zone (it must be sufficiently long to allow the 30 degree angle jets to mutually impinge before the resulting derived jet impinges on the dome), the necessary diameter can be obtained from the volume term.

A similar design procedure is used for the secondary (or carbon monoxide burn-out zone) which precedes the dilution zone. A separate and distinct value is used, together with an empirical length/diameter ratio of 1:1 to solve for the dimensions. The two values used for the design of the present combustor are for the primary and secondary zones respectively, $5.8 * 10^6$ and $4.0 * 10^6$. The units utilized for calculating these two values being psia, in^3, °F, lb/sec, each corresponding to the pressure, volume, temperature and mass flow terms in the correlation.

The determination of the dimensions of the dilution zone follows conventional practice, in that the length is chosen to provide a particular desired pattern factor. A length to diameter ratio correlation with pattern factor for various pressure drop levels (pressure drop over the combustor divided by the dynamic head of the jets entering the combustor; $\Delta P/q$) is shown in Figure 11. From this correlation, and for a particular desired pattern factor and pressure drop level, a length to diameter ratio can readily be estimated. The diameter of the dilution section is taken as that of the secondary zone and thus the length is then defined. This later length is taken as the distance between the dilution hole plane and the exit plane of the combustor.

With the above general dimensions defined, the next phase of the design procedure involves determining the total open-hole area requirements of the system.

Open-Hole Area Determination

In general terms the total hole area available can be determined from pressure drop considerations at a given set of inlet conditions using either incompressible or compressible flow equations.

The total flow, and thus the open area for cooling purposes, is defined next by using for each of the sections (primary zone, dilution and secondary zone) an empirical correlation of cooling mass flow per unit of surface area cooled versus a wall temperature function. This latter function is defined as the

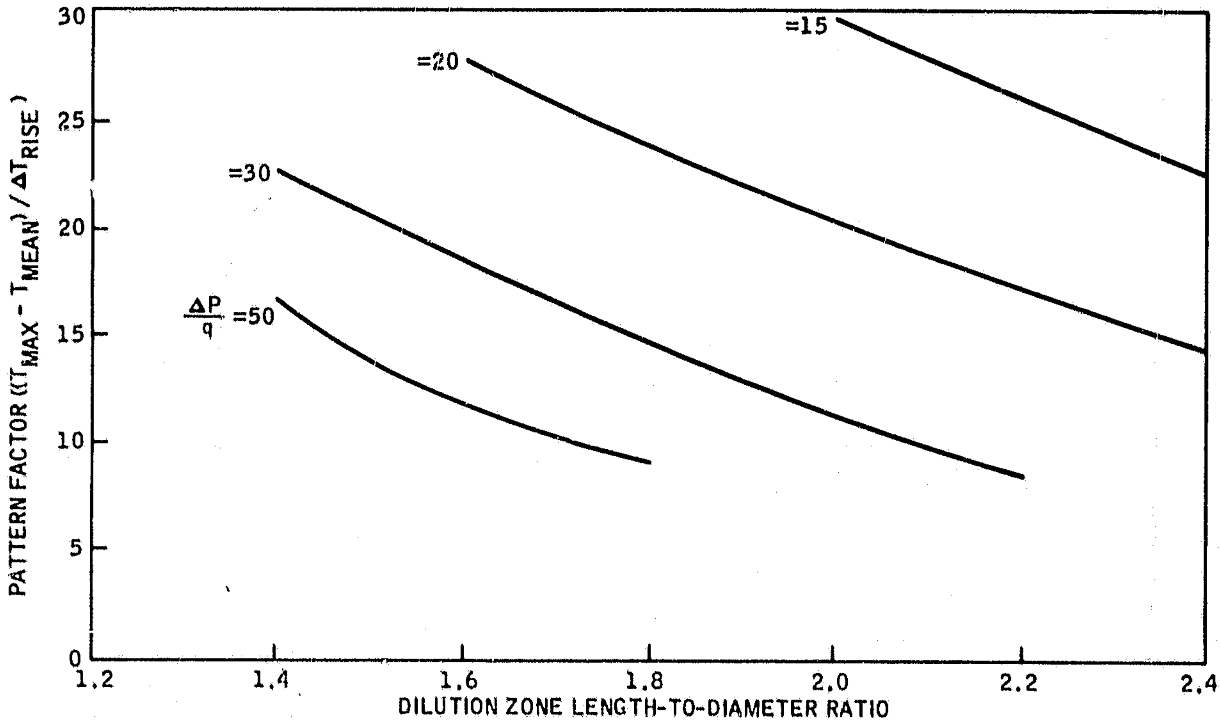


Figure 11. Dilution Zone Sizing Correlation

desired temperature of the wall minus the inlet temperature divided by the temperature rise over the combustor. Typical values of this function are 0.01 $lb/sec/in^2$ for secondary and dilution zones and 0.003 $lb/sec/in^2$ for primary zones using combined convective/film cooling. The data available to date is such that a general correlation between the above two groups is not possible, however, discrete values such as those quoted can be utilized.

After the total mass flow required for cooling is obtained from the above correlations, the open-hole function associated with it is then estimated by multiplying the total hole area by the cooling mass flow fraction. In addition the primary zone air mass flow function and thus the associated open area function is estimated from the inverse ratio of the primary zone to overall equivalence ratios. The two area functions (cooling and primary zone) are summed and subtracted from one to provide the dilution zone area fraction. If the sum of the two area fractions exceeds one then dilution may be dispensed with and the combustor redesigned to minimize cooling air requirements (less surface area).

From the dilution area fraction, and assuming six holes with a discharge coefficient of 0.6, an estimate of the size of the holes can be obtained. A similar calculation is also made for the primary ports. The detail design of the primary ports is covered however, by the next section that describes the design requirements of the fuel/air preparation section.

Fuel/Air Preparation Section

The fuel/air preparation and premixing section of the JIC combustor consists of a series of tubes or ducts (usually more than three) of circular cross-section arranged to introduce the air and fuel mixture as jets into the primary zone. These jets are dispersed so as to mutually impinge on the primary zone centerline to form a main derived jet that moves toward and impinges on the primary zone dome wall. After impingement on the dome, the resulting "spent" jet flow is constrained by the walls of the primary zone to flow rearward and between the incoming jets.

The incoming "fresh" jets entrain a proportion of the "spent" jet fluid to create a torodial vortex flow pattern, the mass flow of which is usually greater than the inlet flow.

The geometry of the tubes or mixing ducts directly affects the vaporization and mixing levels. Tube length or residence time is a particularly critical parameter that affects autoiginition while the diameter of the tube has a definite effect on mixing rate. Generally the tubes should be as long as possible, and there should be as large a number as possible to promote mixing and vaporization. In addition, they should be as small a diameter as possible, again to ensure rapid mixing and complete vaporization. There are limitations however, to both the length and the number of ports that can be employed. Autoignition of the air and fuel limits the length of the mixing duct according to the highest value of θ produced by the general operating conditions.

Jet blockage/stability considerations limit the number and diameter of the jets depending on the dimensions of the combustor proper. Generally, if a blockage factor (B), defined as

$$B = (N * d * D_p^2/2)/(\pi D_p^2 2/4)$$

where

N is the number of jets or tubes
d is the tube diameter
D_p^2 is the primary zone diameter

exceeds a value of 0.75 to 0.8 then the number of jets will have to be reduced from the initial starting value of 6. This latter value, it should be noted, is near the optimum number for maximizing the upstream recirculating flow (see Ref. 1).

In addition, the tube diameter has to be checked against a flashback correlation to ensure the ultimate mechanical integrity of the ports. The particular correlation used for this check (see Fig. 11) is based on very few data points which thus produces a large uncertainty in the predictions. If the velocity gradient (G), defined as

$$G = f\ U\ Re/(2d)$$

where

f is the Fanning friction factor
U is the mean air velocity
Re is the Reynolds number
d is the tube diameter

exceeds a certain critical value then flashback can occur. If the critical value is exceeded, then the number of tubes has to be increased with a concomittant decrease in the diameter.

Fuel injector performance and type do have a significant impact on the fuel vaporization and mixing through its control of the droplet dimensions and distribution of drop sizes and also the initial fuel dispersion.

Although the highest initial fuel dispersion possible is the goal for any fuel injection system there are other limits that must be considered. In particular the fuel injection device should not place fuel on the wall of the premixing duct or tube. In additon, the injection mechanical blockage should be a minimum to avoid creating local recirculating zones that enhance autoignition and which could also act as flameholders. Generally these restrictions impose the requirements that the fuel droplet motion be dominated by the airflow and that the airflow is non-swirling. Wakes or recirculating zones have to be elminated through the design and placement of the injector outer shell. Usually the device has to be streamlined or placed upstream of an accelerating flow field which tends to reduce recirculating flow levels.

To date the only successful fuel injection arrangement discovered (fuel pressure atomizers, air blast and air-assist atomizer have been considered) is the air-blast system. The main reason for this is that the other systems tend to deposit fuel on the duct walls that in turn tends to cause flashback.

3.3.2 Lean Semi-Conventional Primary Zone

This particular combustor concept was basically a modification of the rich primary zone module. In essence, a larger flow radial inflow swirler was substituted for that used during rich mode operation, so that the primary zone equivalence ratio at the design point could be adjusted to lie in the range of 0.8 to 1.0. Different sizes of fuel injectors capable of integration with this swirler were also provided. These fuel injectors, one of which was an air-assist system and the other a spinning-cup, were designed specifically to fit the geometry of the lean swirler.

Transition Pieces

Three different transition pieces that fitted between the primary and secondary zones were designed (see Fig. 12). Two of these were intended to provide secondary air injection into the small diameter transition piece. The injection geometry was designed to provide rapid mixing of the secondary air with the hot exhaust gases exiting from the rich primary zone. In operation these transition pieces would essentially replace the air injection system usually provided in the lean secondary zone by the reverse flow ports.

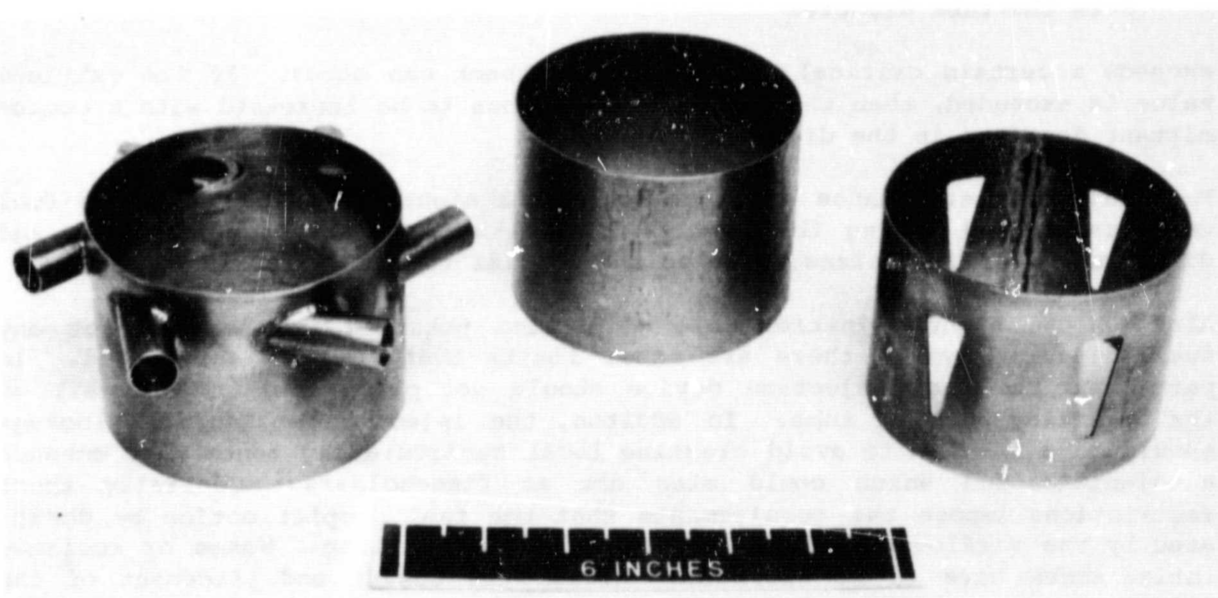


Figure 12. Rich-Lean Transition Pieces

One of these transition pieces utilized triangular holes which provided jets that "bend over" early although part of the jet penetrated to the centerline. This phenomenon allows mass addition along the radius that is the axis of the jet. The overall effect is to provide more rapid mixing. A second approach was also designed in which the secondary air would enter through a series of tubular ports to produce a swirling flow, swirling in the opposite direction to the main flow. This also had the potential, based on past experience of providing rapid mixing.

This rapid mixing of the secondary air with the fuel rich primary zone exhaust is essential if low NOx emissions are to be obtained. The general intent is to mix the secondary air and the fuel-rich primary flow together in a manner that would provide minimal residence time at stoichiometric or near stoichiometric mixture strengths. When the requisite amount of secondary air is added, the resultant equivalence ratio would be of the order of 0.55. This value being determined from previous work with lean reaction zone systems as the maximum equivalence ratio, for the initial mixed temperature, that would provide low thermal NOx.

Dilution Zone

A conventional dilution zone was provided with six circular dilution ports.

Conventional film cooling was also used in this zone. The length and diameter were estimated in the manner described above under the heading of Lean Premixed Primary/Secondary Zone (see also the work in Ref. 7), while the port or dilution hole sizes were calculated form the overall combustor stoichiometry.

Rich and Lean Spinning Cup Fuel Injector

A spinning cup aerosol generator can produce in excess of 10^7 near monodisperse particles per second with sizes adjustable over a range from 20 to 60 microns when operating in the drop formation mode (see Figs. 13 and 14). To produce the aerosol, the particle formation material (fuel) is fed onto the center of a stainless steel cup which rotates at from 10,000 to 100,000 rpm. The liquid is atomized into two discrete droplet sizes. Primary droplets, varying in diameter from 20 to 150 microns in diameter, are formed during liquid break-up, and form the bulk of the near homogeneous test aerosol. Smaller satellite droplets are also formed during liquid break-up and rapidly vaporize. These smaller droplets constitute by mass a relatively small proportion of the total spray, usually of the order of eight percent.

The size of the primary droplet d_p is related to the angular cup speed , cup diameter D, fluid surface tension σ, and fluid density ρ, by the approximate expression:

$$d_p = K \sigma/\rho\omega^2 D^{1/2} \qquad (1)$$

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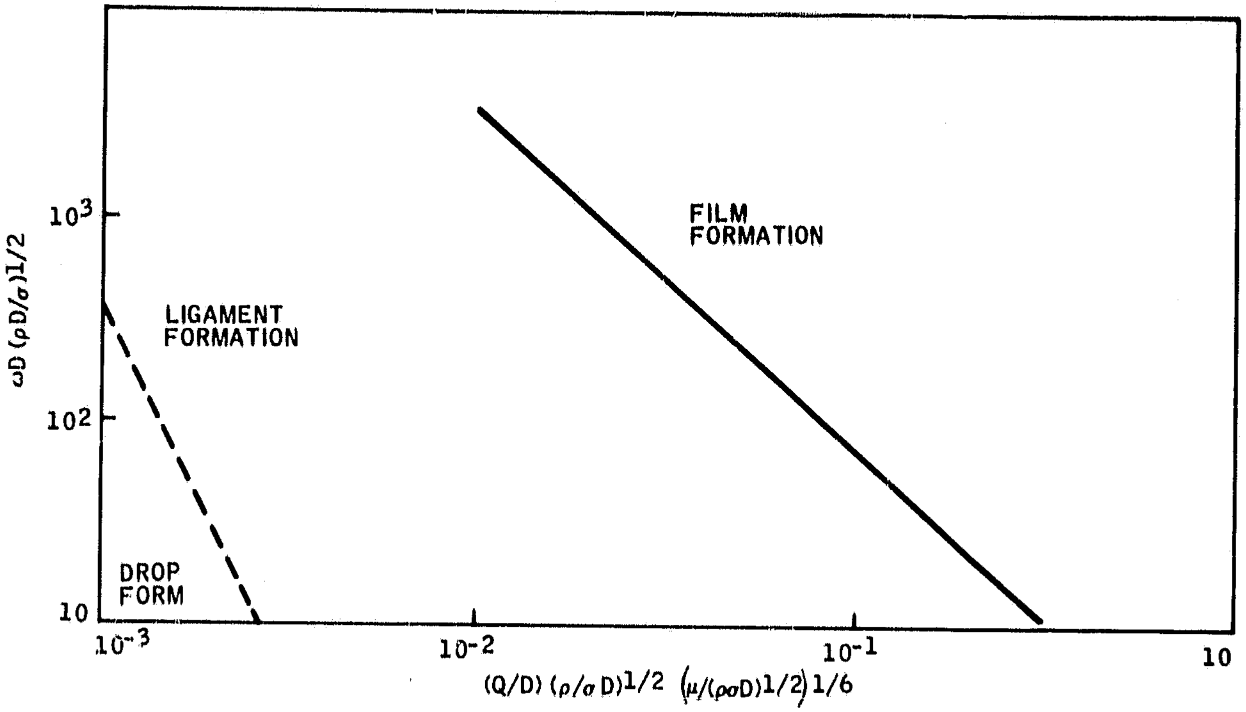

D - cup diameter
Q - volume flow rate
ρ - fuel density

σ - fuel surface tension
μ - fuel absolute viscosity
η - fuel kinematic viscosity

Figure 13. Operational Modes of a Spinning Cup Atomizer

The constant K is theoretically equal to $(12)^{1/2}$. Experiments show that K can vary from 2 to 7 depending on the disk speed and on the liquid used.

With respect to particle monodispersity, the primary aerosol size distribution can be approximated by a log-normal relationship with a geometric standard deviation of 1.05 to 1.15. The total horsepower used by a spinning cup can be estimated from the curve of Figure 15. This particular curve was obtained with water as the working fluid, however, it does provide a close approximation for paraffinic fuels.

A re-examination of the data of Walton and Prewett at Battelle Memorial Institute (A. A. Putnam & C. C. Miesse), employing dimensional analysis, has led to the following equation that takes into consideration the effects of the liquid viscosity (η).

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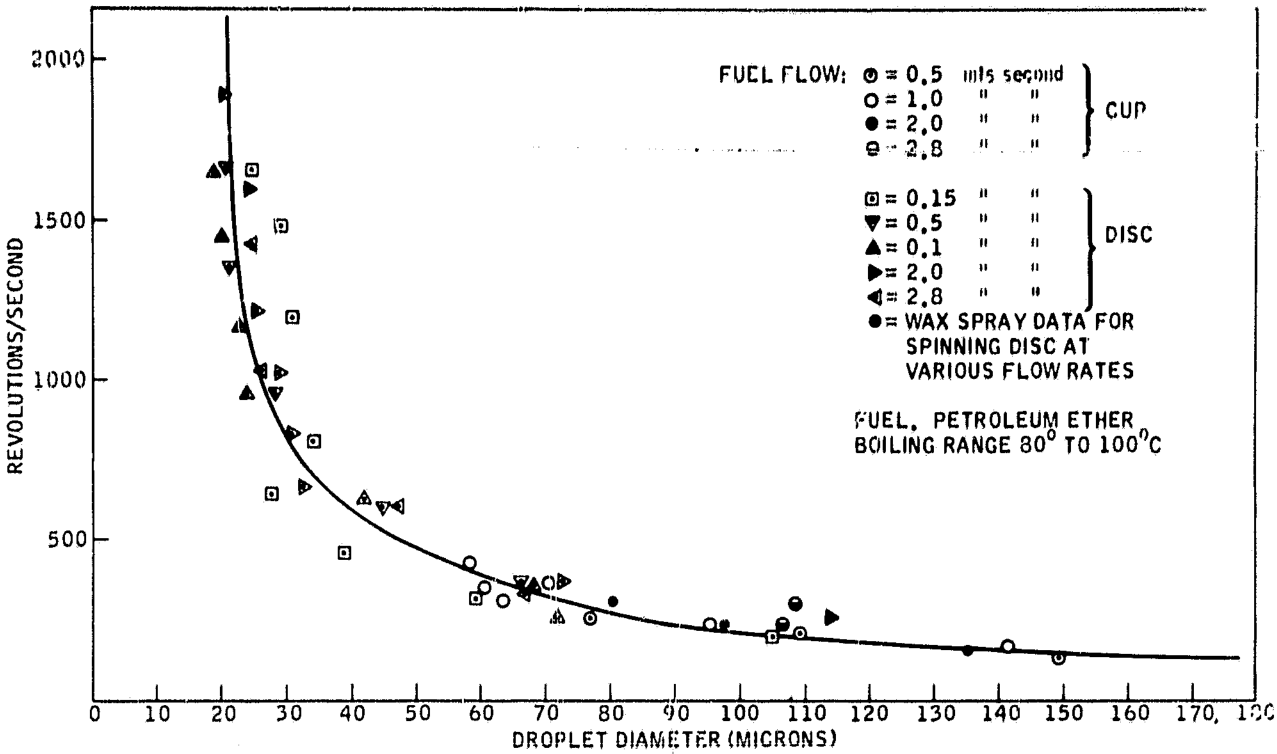

Figure 14. Mean Droplet Diameter of a Rotating Cup Fuel Injector at Various Cup Speeds

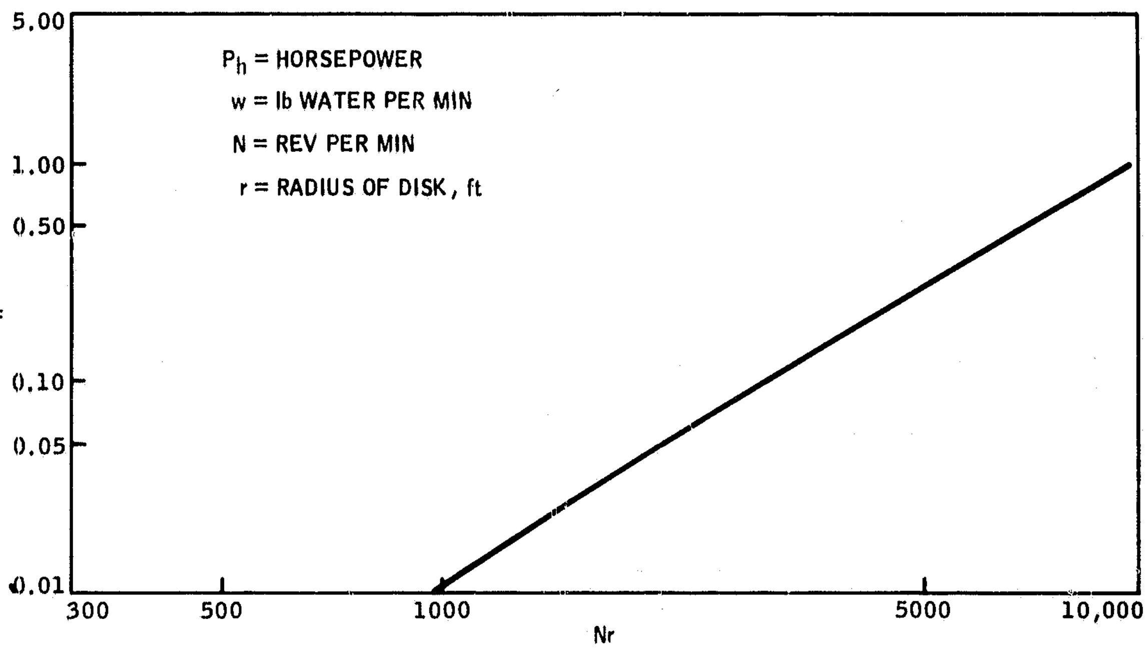

Figure 15. Power Requirements of Spinning Cup

$$d_p D = 6.855 \, (\eta^2/\rho\sigma D)^{0.041} \, (\omega^2 \, d_p^3/ \,)^{0.522} \quad (2)$$

D = cup diameter
Q = volume flow rate
ρ = fuel density
σ = fuel surface tension
μ = fuel absolute viscosity
η = fuel kinematic viscosity

This latter equation provides a reasonably accurate correlation of the monodisperse drop size as proven by a limited series of actual measurements.

From the foregoing and from the fixing of the cup geometry (1.987 inches diameter) by the combustor and swirler sizes, it is estimated that a cup speed of the order of 40,000 rpm with a horsepower requirement of 1.25 HP could be needed.

Rich and Lean Air-Assist Atomizers

Both rich and lean air assist atomizers utilized the same design techniques. Additionally they were both designed in modular fashion so that different geometries, that changed internal flow splits, could readily be incorporated. Two basic designs were considered for each of the rich and lean atomizers, One involved "overlip" fuel filming and the other "underlip" fuel filming. In the "overlip" case the fuel was injected as a film onto the inner surface of an annular passage containing a swirling air flow. This inner wall terminated in a sharp edge off of which the fuel was atomized. In addition the cylindrical passage formed by the annulus inner wall had an extra swirling air flow which, with air in the annular passage, caused a shearing action (see Figs. 16 and 17). Various swirler inserts could be mounted in each of the two fuel injectors (rich and lean) produced. This changing of swirlers varied the air flow split above and below the fuel film.

Initially only one underlip configuration was produced, this being for the lean primary combustor, and this is shown in Figure 18. The fuel in this instance was injected as a film onto the outer surface of the center cylindrical duct. An annular concentric duct was also provided that had a swirling air flow constraint to pass along it. Swirling air also passed along the center cylindrical duct and this helped stabilize the film on the wall and allowed wide turndown without film disruption. Later in the program one of the overlip atomizers for rich operation was modified to an underlip configuration.

3.4 FABRICATION

All the fabrication of the moduluar unit rigs and component parts was subcontracted to small specialist metal-working shops. Raw materials for all of the combustors and rigs were supplied by Solar Turbines International, who maintained material control records for this material.

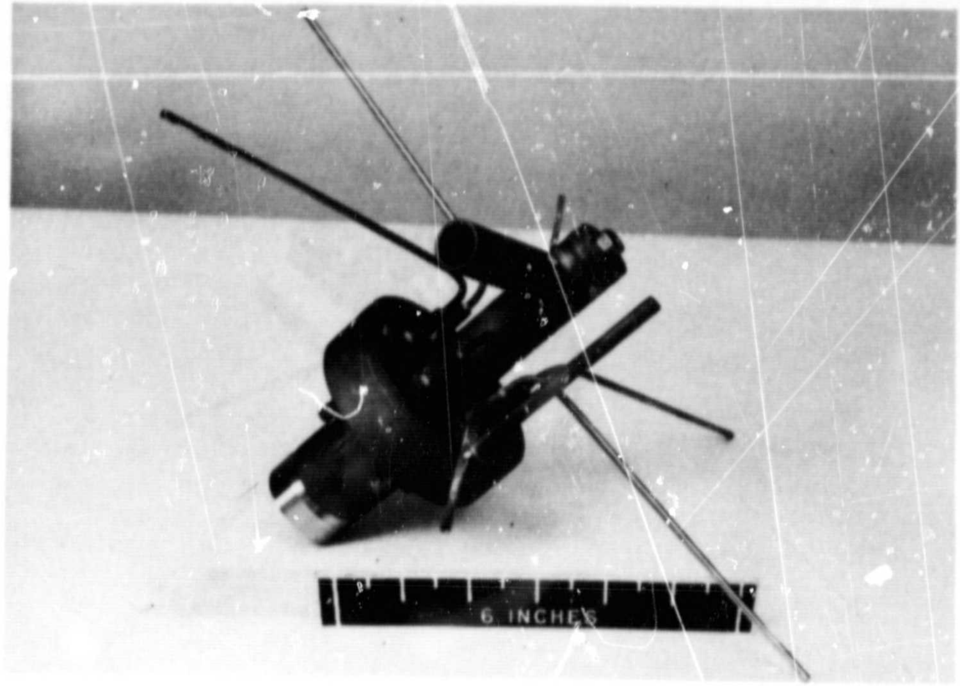

Figure 16. Overlip Fuel-Filmed Vortex Injector (Side View)

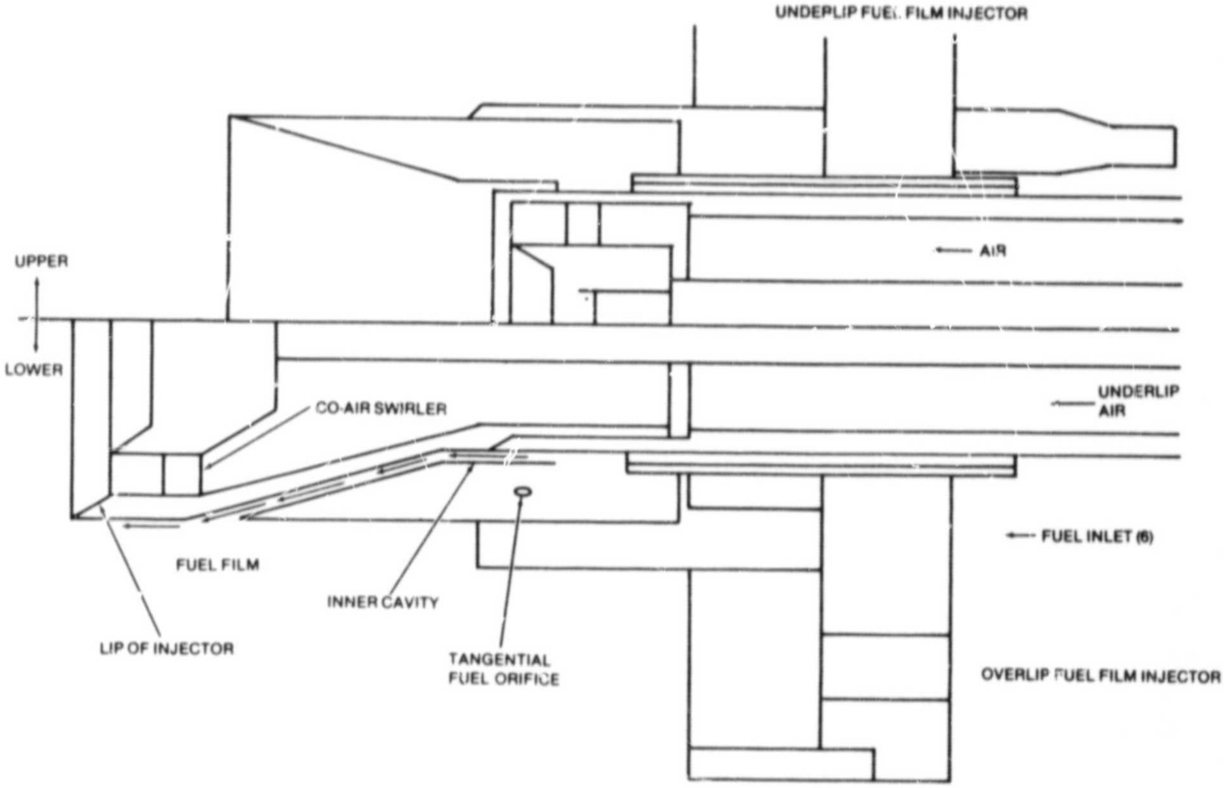

Figure 17. Upper Half Depicts Underlip Fuel Film Injector; Lower Half Depicts Overlip Fuel Film Injector

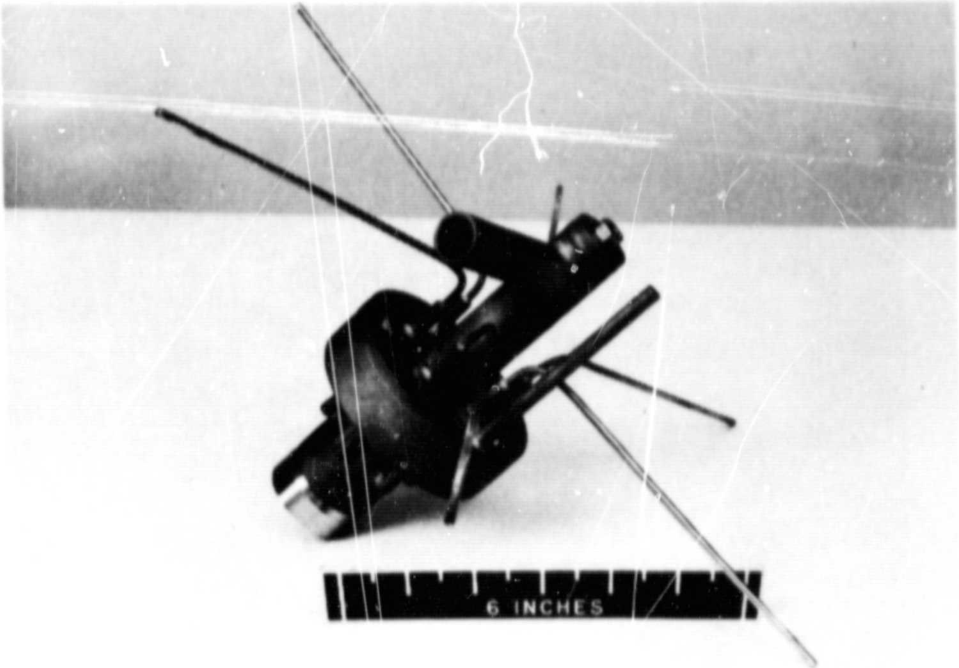

Figure 18. Underlip Fuel Filmed Vortex Injector (Side View)

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4

FUELS AND FUEL SYSTEM

4.1 TEST FUELS

Three different fuels were supplied by NASA-LeRC, as government furnished equipment (GFE), to Solar Turbines Inc. for evaluation on the program described herein. These three fuels consisted of a pretroleum based middle distillate fuel developed by NASA as an Experimental Research Broadened Specification Fuel (ERBS), a middle distillate fuel derived from coal, produced by Gulf Oil Co. and called Solvent Refined Coal II (SRC-II), and a petroleum based residual fuel oil.

All three fuels were analyzed in detail with particular emphasis being placed on the SRC-II. Such emphasis reflected the general lack of familiarity with such a fuel. The results of these analyses, which are presented in terms of the various key ASTM tests, are shown in Tables 6, 7 and 8. For reference purposes each table also contains the limits normally imposed by Solar Turbines Incorporated (STI) on fuels to be used in their engines.

In comparing the results obtained with SRC-II with those recommended by STI, it can be seen there are significant differences between the two. The SRC-II fuel has a lower heating value than that normally required for conventional fuels. Its viscosity is higher and so is its nitrogen content. Corrosive elements such a sodium, potassium, sulfur and vanadium are, however, much lower.

In general these tests reveal that with the exception of the nitrogen content the SRC-II fuels would make a reasonable gas turbine liquid fuel. The nitrogen content is sufficiently high, however, that even with 40 percent conversion (assuming an average 1.2 percent nitrogen content) the approximate contribution would be 65 ppm of NOx at 15 percent O_2, leaving a margin of 10 ppm for any thermal NOx contribution. SRC-II would probably provide a less corrosive exhaust stream than most petroleum fuels having a similar physical specification, because of the absence or minimized levels of vanadium, sulfur and alkali metals such as sodium.

A qualitative analysis of the constituent parts of the SRC-II fuel has also been made and the results are summarized in Table 9. The general conclusion of this analysis is that the material consists mostly of olefinic and aromatic compounds. The nitrogen in the fuel apparently is present mostly in the "heavier" aromatic compounds, which are reasonably strong bases. This indicates that the nitrogen may be present in compounds such as aniline and its homologues and possibly also secondary amines.

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Table 6

ERBS Fuel Properties

Fuel Property	Solar Limits	Test Method	Data
Kinematic Viscosity	Maximum: 12 Centistokes Minimum: 1 Centistoke @ 100°F + (210°F)	ASTM D 445	1.68 Centistokes @ 100°F 0.83 Centistokes @ 210°F
Gravity	30 to 51 Degrees API	ASTM D 287	38.78 Degrees API (60°F)
Cloud and Pour Point	Cloud = 10°F Below Amb. Max. Pour = 10°F Below Cloud Max.	ASTM D 2500 ASTM D 97	Cloud = -0.4°F Pour = -54°F
Flash Point	100% Minimum or Legal Limit	ASTM D 93	180°F Flash & Fire
Distillation	90% Evaporated 625°F Maximum End Point 675°F Maximum.	ASTM D 86	90% Evaporated 575°F End Point 645°F (98% Recovery)
Sulfur	1% by Weight Maximum	ASTM D 129	0.054% By Weight
Lower Heating Value	18,000 Btu/lb Minimum	ASTM D 240	18251 Btu/Lb
Carbon Residue on 10% Residum	0.35% Maximum	ASTM D 524	0.52%
Ash	0.005% by Weight Maximum	ASTM D 482	0.002% By Weight
Vanadium	0.5 Parts Per Million by Weight Maximum	ASTM D 2787	0 Parts Per Million
Sodium Plus Potassium	1 Part Per Million by Weight Maximum	ASTM D 2788	0 Parts Per Million
Calcium	1 Part Per Million by Weight Maximum	ASTM D 2788	0 Parts Per Million
Lead	2 Parts Per Million	ASTM D 2787	0 Parts Per Million

A mass spectrometer analysis of the SRC-II fuel was also made. The results confirm, to a certain extent, the initial qualitative analyses (Table 10). This analysis provided clear evidence of the high phenolic levels and the high naphthalene concentration, which are known to cause problems due to solvency with many of the polymeric sealants used in fuel systems.

The high levels of polynuclear hydrocarbons and their derivatives indicates that the chances of smoke and soot production are high in comparison say with the ERBS fuels.

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Table 7

Solvent Refined Coal (SRC-II) Physical Properties

Fuel Property	Solar Limits	Test Method	Data
Kinematic Viscosity	Maximum: 12 Centistokes Minimum: 1 Centistoke @ 100°F	ASTM D 445	4.76 Centistokes @ 100°F 1.35 Centistokes @ 210°F
Gravity	30 to 51 Degrees API	ASTM D 287	9.85 Degrees API (60°F)
Pour Point	Pour = 10°F Below Cloud Max.	ASTM D 97	Pour = -40°F
Flash Point	100% Minimum or Legal Limit	ASTM D 93	200°F
Distillation	90% Evaporated 625°F Maximum End Point 675°F Maximum.	ASTM D 86	90% Evaporated 615°F End Point >750°F (95% recovery)
Sulfur	1% by Weight Maximum	ASTM D 129	0.097% by weight
Lower Heating Value	18,000 Btu/lb Minimum	ASTM D 240	16590 Btu/lb
Carbon Residue on 10% Residum	0.35% Maximum	ASTM D 524	1.10%
Ash	0.005% by Weight Maximum	ASTM D 482	0.0004% by weight
Vanadium	0.5 Parts Per Million by Weight Maximum	ASTM D 2787	0 Parts Per Million
Sodium Plus Potassium	1 Part Per Million by Weight Maximum	ASTM D 2788	0.40 Parts Per Milion
Calcium	1 Part Per Million by Weight Maximum	ASTM D 2788	0 Parts Per Million
Lead	2 Parts Per Million	ASTM D 2787	0 Parts Per Million
Engineering Comments:	Carbon 84.55/84.35% Hydrogen 8.73/8.87%		

Viscosity relationships for each of the fuels (as a function of temperature) were determined and they are shown in Figures 19 and 20 for ERBS and Residual Fuel. In addition the viscosity of SRC-II as a function of temperature is provided in Figure 21. The visocisty of the residual fuel was so high that it could not be measured directly at 37.8°C (100°F) which is the standard ASTM requirement. The visocisity was determined instead by extrapolating the

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Table 8

Residual Fuel Properties

Fuel Property	Solar Limits	Test Method	DATA
Kinematic Viscosity	Maximum: 12 Centistokes Minimum: 1 Centistoke @ 100°F + (210°)	ASTM D 445	4100 Centistokes @ 100°F 10.64 Centistokes @ 210°F
Gravity	30 to 51 Degrees API	ASTM D 287	17.4 Degrees API (60°F)
Flash Point	100% Minimum or Legal Limit	ASTM D 93	300°F 362°F - Fire
Distillation	90% Evaporated 625°F Maximum End Point 675°F Maximum	ASTM D 86	70% Evaporated 730°F End Point 730°F (79% recovery)
Sulfur	1% by Weight Maximum	ASTM D 129	0.056% By Weight
Lower Heating Value	18,000 Btu/lb Minimum	ASTM D 240	17867 Btu/Lb
Carbon Residue	0.35% Maximum	ASTM D 524	5.37%
Ash	0.005% by Weight Maximum	ASTM D 482	0.5% by Weight
Vanadium	0.5 Parts Per Million by Weight Maximum	ASTM D 2787	10.5 Parts Per Million
Sodium Plus Potassium	1 Part Per Million by Weight Maximum	ASTM D 2788	Na = 22.3 k = 25.0
Calcium	1 Part Per Million by Weight Maximum	ASTM D 2788	82.0 Parts Per Million
Lead	2 Parts Per Million	ASTM D 2787	0 Parts Per Million

results of viscosity mesurements made with the residual fuel diluted with diesel fuel of a known viscosity. By taking measurements at several different dilutions, a graph can be drawn showing viscosity at a particular temperature as a function of composition (see Fig. 22). By extrapolting these later relationships to a compositon consisting of pure residual fuel a viscosity can be determined that is representative of the undiluted residual fuel.

4.1.1 Fuel Systems

The fuel system design that was adopted after long consideration of all of the options is shown in Figure 23. This system, although complex, allowed the accurate metering of a wide range of fuels differing in viscosity and density.

Table 9

Qualitative Tests for Bases in SRC-II Mid-Distillate Fuel

Test Method

One-hundred (100) milliliters of this fuel was extracted with 10 mls of 0.1 N HCl in water and this 10 ml portion was subsequently made alkaline by the addition of 20 mls of 0.1 N NaOH. After extracting the 30 mls of alkaline water containing the basic constituents with chloroform, the chloroform was dried with sodium sulfate and evaporated to concentrate the basic constituents.

Infrared and ultra violet spectrophotometry were used to investigate this residue and a series of thin-layer chromotograms were also performed.

The infrared spectrum of the "as-evaporated residue" indicated the presence of both aromatic components and primary or secondary amines as well as aliphatic components. The thin-layer chromatograms indicated that the residue consists of three components, which are described below.

Results

- The most mobile is not aromatic but a conjugated olefinic material which does not contain nitrogen.
- The other less mobile species may be somewhat aromatic in character and contain nitrogen, but it is too weak a base to react with bromecresol green indication.
- The non-mobile component also has aromatic character, contains nitrogen and is a strong enough base to react with bromecresol green indication.
- Ultra violet and infrared analyses of the individual components were not able to identify them.

It was designed to operate from fuel drums which held the desired fuel or fuel mixture. Fuel mixtures were blended in separate batch mixing tanks on a weight basis. The resultant mixtures were then placed in specially lined fuel drums that replaced the standard fuel drums containing the as-received fuels. The fuel system was able to maintain accurate fuel temperatures through the use of a water bath and insulated lines. Electrical trace heating was also used when necessary to achieve a final tailoring of the fuel temperature. With an accurate fuel viscosity-temperature correlation the viscosity of the fuel is thus known and any necessary corrections for viscosity applied to the turbine meter reading, thus providing accurate mass flow readings.

Fuel in this system was taken directly from the drums that it was delivered in through a boost pump into a main high pressure pump. After leaving the high pressure pump the fuel can be passed through a water-bath to heat it to temperatures up to approximately 88°C (190°F). This heated fuel could, if desired, be recirculated back to the drum several times to ensure that the

Table 10

Mass Spectrometer Analysis of SRC-II Fuel

	Volume, %
Paraffins	2.43
Naphthenes	4.75
Alkylbenzenes	12.16
Indanes/Tetralins	13.31
Idenes	5.45
Napthalenes	16.55
Acenaphthenes	4.76
Acenaphythalenes	1.51
Tricyclicaromatics	1.01
Phenols	24.44
Unidentified components	13.62

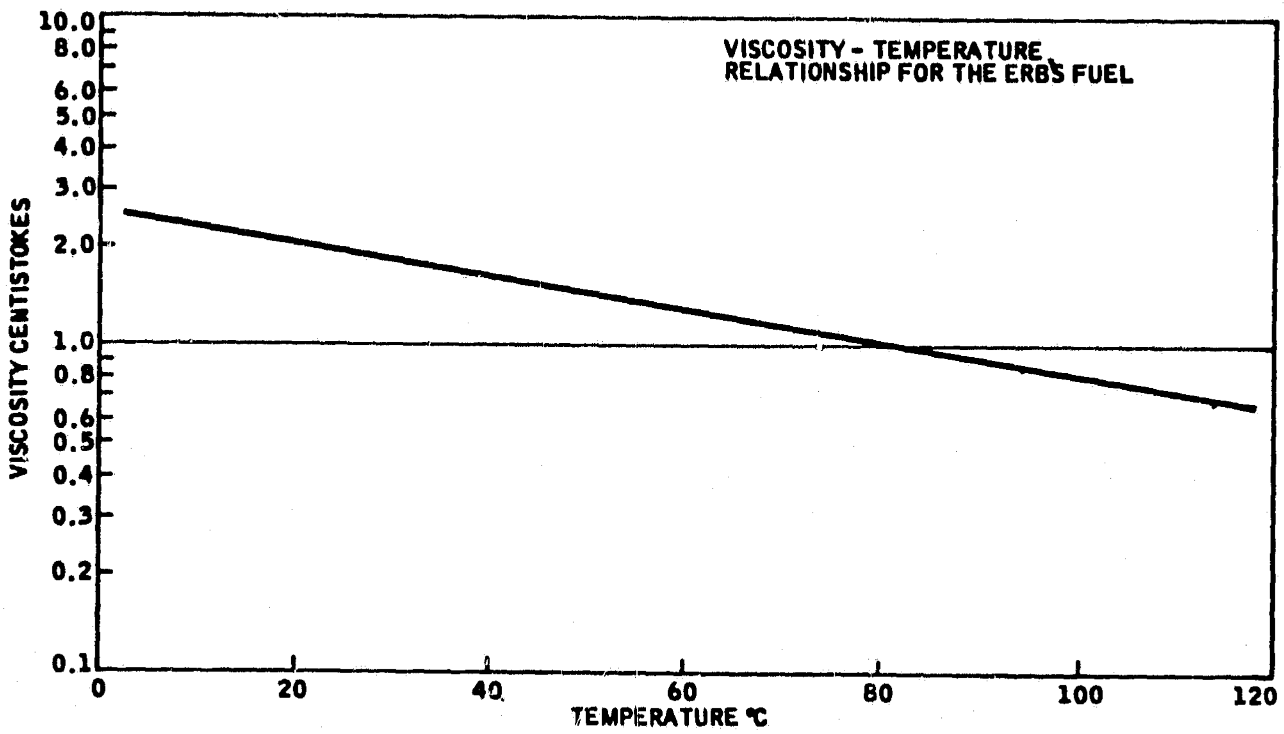

Figure 19. Fuel Viscosity-Temperature Relationships

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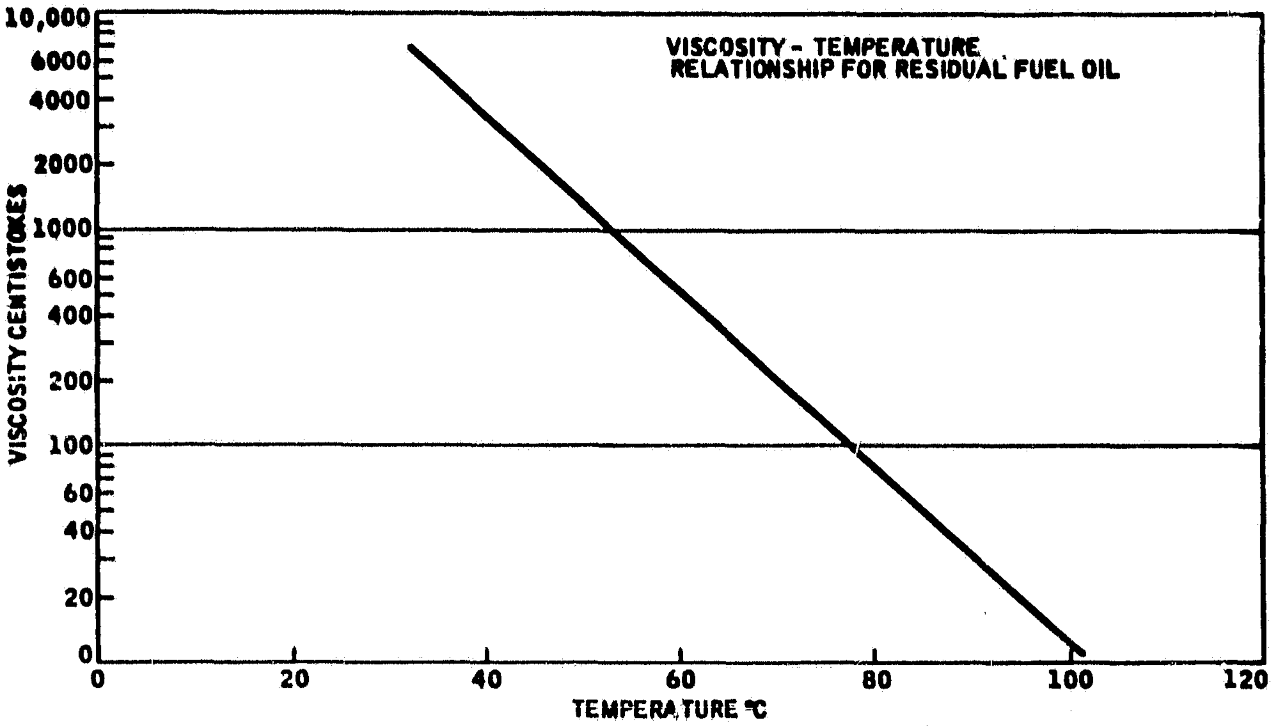

Figure 20. Residual Fuel Viscosity-Temperature Relationship

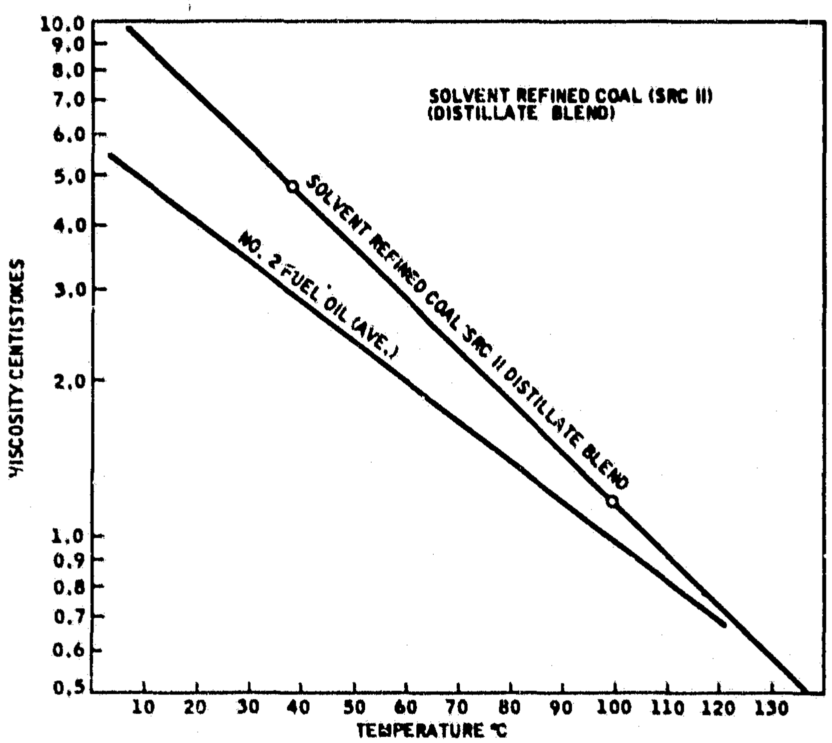

Figure 21. Comparative SRC-II Viscosity-Temperature Relationship

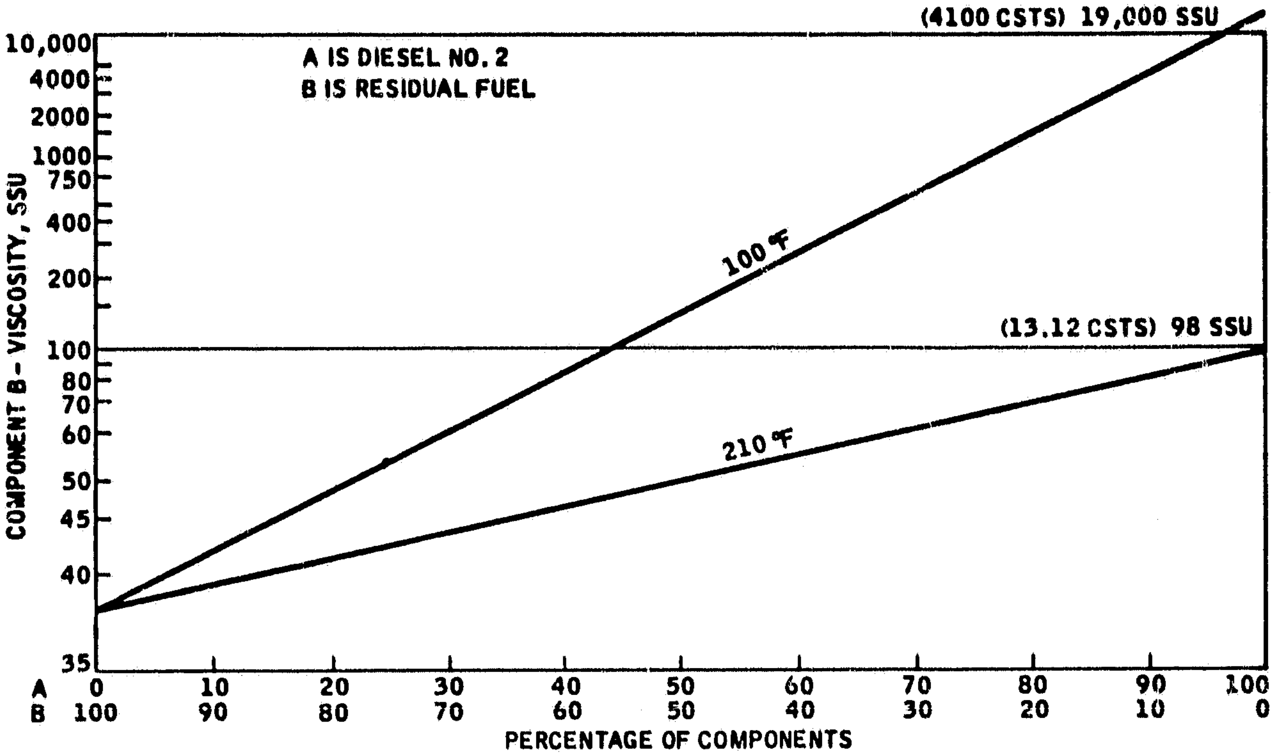

Figure 22. Blended Diesel and Residual Fuel Viscosities

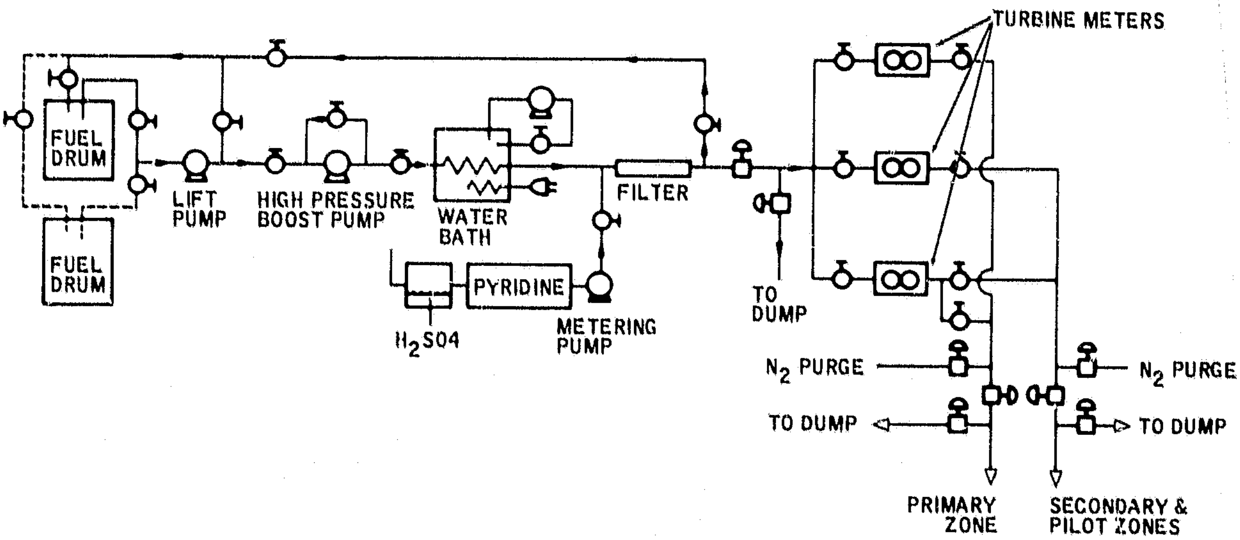

Figure 23. Fuel System Schematic

total fuel batch temperature is near constant. Monitoring of the fuel flow was accomplished using multi-viscosity turbine meters in three flow sizes. These latter three turbine-meters were selected to cover the flow range to be covered in the various tests. From the turbine meters the fuel was directed to the test rig and to the particular desired fuel injection point on the combustor. Flow control was provided by utilizing a coarse needle-valve and a fine micrometer-valve in series for each of the individual fuel lines. By measuring the fuel temperature just ahead of the flow meter and using a correlation of viscosity versus temperature, an accurate viscosity value could be obtained. This was then input into a correlation of a flow correction factor versus viscosity, to obtain the necessary corrections to the usual flow readings.

A series of drains and flush lines were provided to avoid contamination between the various fuels to be used, and to allow the removal of debris and varnish deposits in the turbine meters and other sensitive equipment pieces. The flushing fluid used was either an ASTM 1-A type of fuel or Diesel No. 2. Arrangements were also made in the fuel system design to allow flushing either the entire system or just the turbine meter section.

Nitrogen could also be used to purge and dry fuel lines. During nitrogen purging, care was taken to avoid passing the gas through the turbine meters (these would overspeed with nitrogen) and the purging system was arranged so that this could easily be achieved.

A high pressure metering pump that could directly inject pyridine into the high pressure fuel line was included as part of the fuel system. Pyridine, for safety purposes, was contained in glass carbon cushioned within an outer stainless jacket. The vent or air inlet line contained a sulfuric acid bubbler that both dried the air entering and prevented pyridine vapor from being vented to the atmosphere. It was necesary to design the pyridine injection system such that the pyridine was directly injected into the high pressure fuel line. If it were premixed into the fuel, its concentration would tend to diminish with time since the fuel was stored in vented drums. Thus the actual concentration would at any point in time be unknown with this latter technique.

The pyridine was to be used to tailor the nitrogen content of the particular fuel being used. When injected into the fuel, samples of the mixed fuel were to be analyzed using wet chemistry methods. Samples would have been taken before and after the test.

5

EXPERIMENTAL APPARATUS

5.1 EXPERIMENTAL COMBUSTOR HARDWARE

After approval of the preliminary designs was granted by NASA project management, final designs for each configuration and their component parts were produced. As described in Section 3, a modular approach to the production of each configuration was adopted as a cost saving expedient. These final designs, when approved, were then fabricated by outside vendors and assembled by Solar. Specialized items and procedures, such as the application of thermal barrier coatings, were also provided by Solar.

The modular approach in the design of the combustor hardware involved the standardization of the main primary zone, as the rich design, and main secondary zone, which could be converted to a primary zone and was based on the lean premixed primary zone design. These two units could be assembled using various interconnecting transition pieces to produce a wide variety of rich-lean combustor configuration.

The secondary zone also could be reversed to allow the secondary jets to penetrate either backward or forward. In addition, this lean secondary zone could be used by the addition of a dome plate and by extending the ports to allow premixing of air and fuel, to take the place of a premixed lean primary zone. By changing the radial inflow swirler assembly the "rich primary zone module" could also be arranged to run in a lean mode as described earlier.

The arrangement of the combustors during their experimental evaluation is illustrated by the rich-lean combustion system shown in Figure 24. A photograph of this test rig is shown in Figure 25. 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 26 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.

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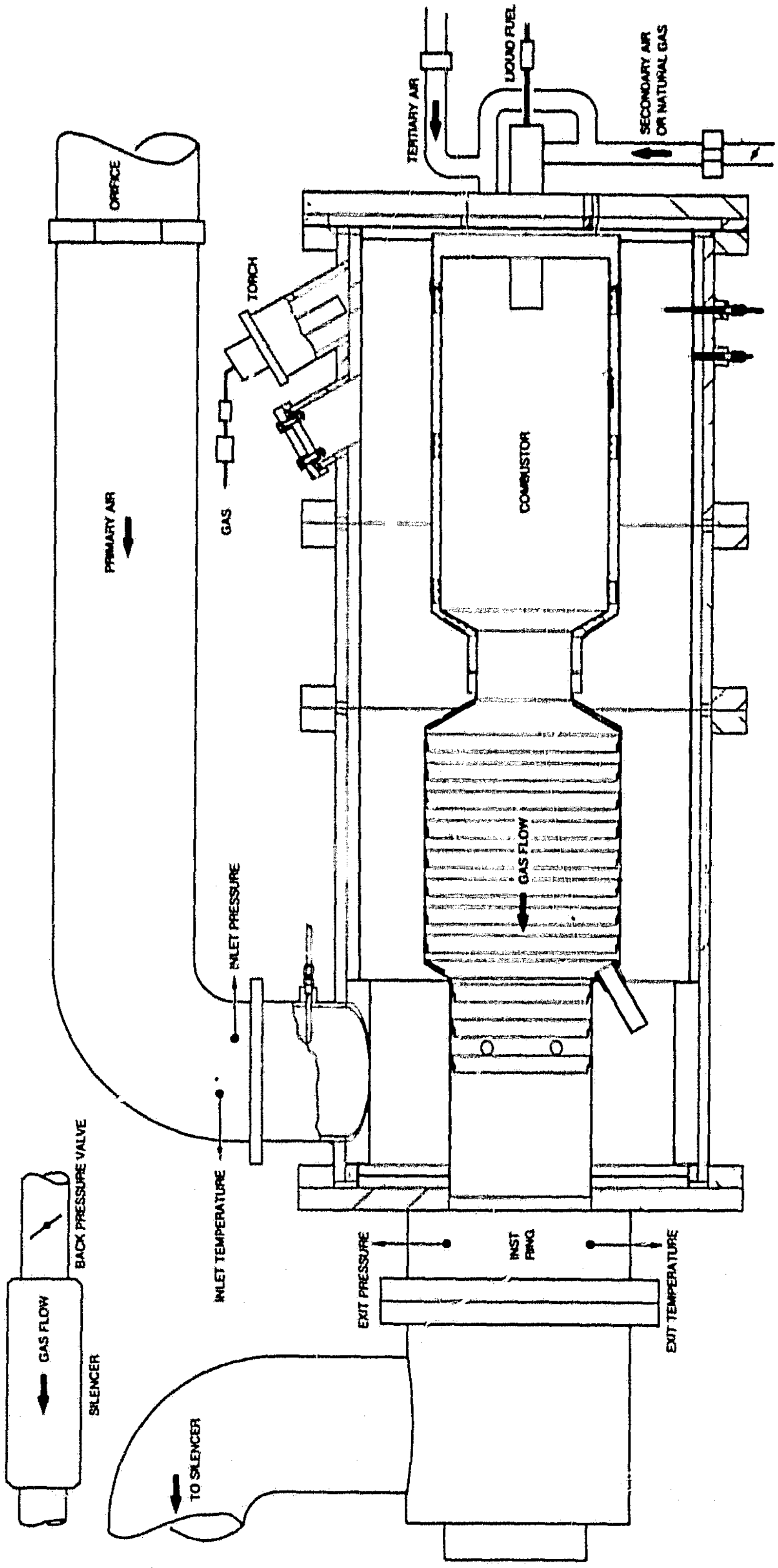

Figure 24. Adopted Test Rig Configuration

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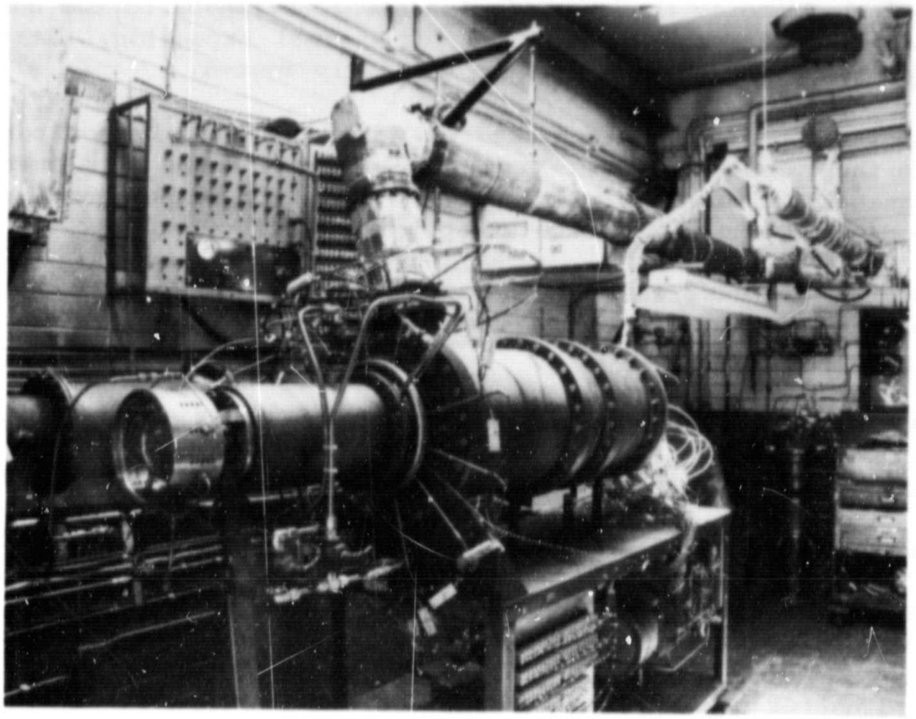

Figure 25. Combustor Test Rig

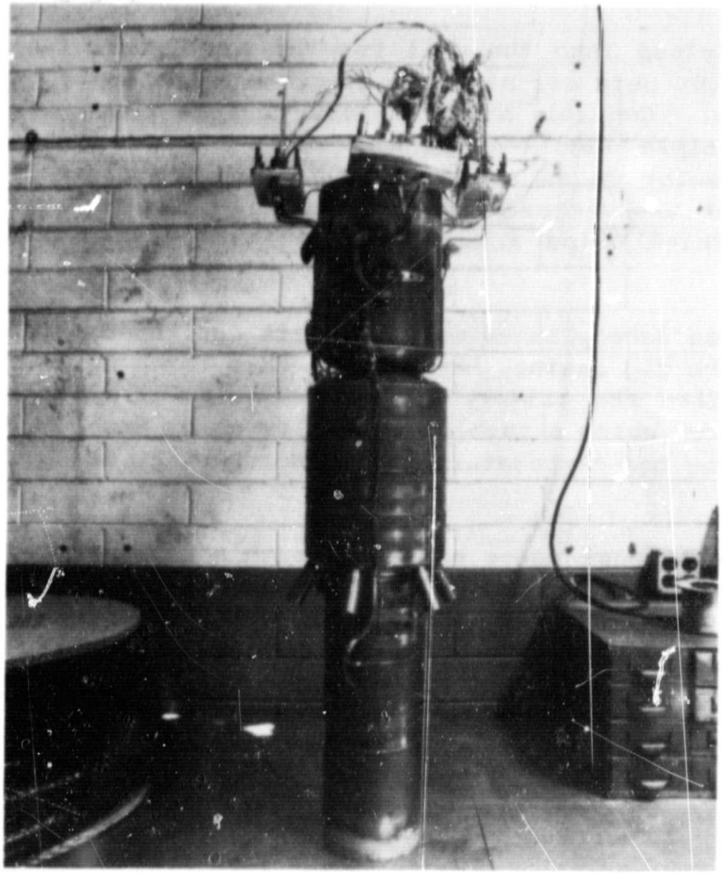

Figure 26.

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 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 open tip 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 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 indirectly brought heated high pressure facility air into the test cell. The maximum flow, pressure, and temperature conditions were 3.5 lb/s, 176 psia and 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 400 psia air at room temperature and low flow rate of the order of 0.2 lb/s. Control of this flow and pressure was obtained through the use of multiple regulators. This air system was also used to drive the spinning cup motor on an as-required basis. The pressure and temperature of the air assist was measured before entering the combustor and air assist airflow was measured using a turbine type flowmeter with a digital readout.

Ignition of the main combustor was accomplished using a spark ignited natural gas torch which was mounted on the rig casing. Flame from this torch entered the primary air swirler and ignited 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 27 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 open-face 1/8 inch Type K thermocouples located at the centers of equal areas in the exhaust flow stream. These are not shown in Figure 27.

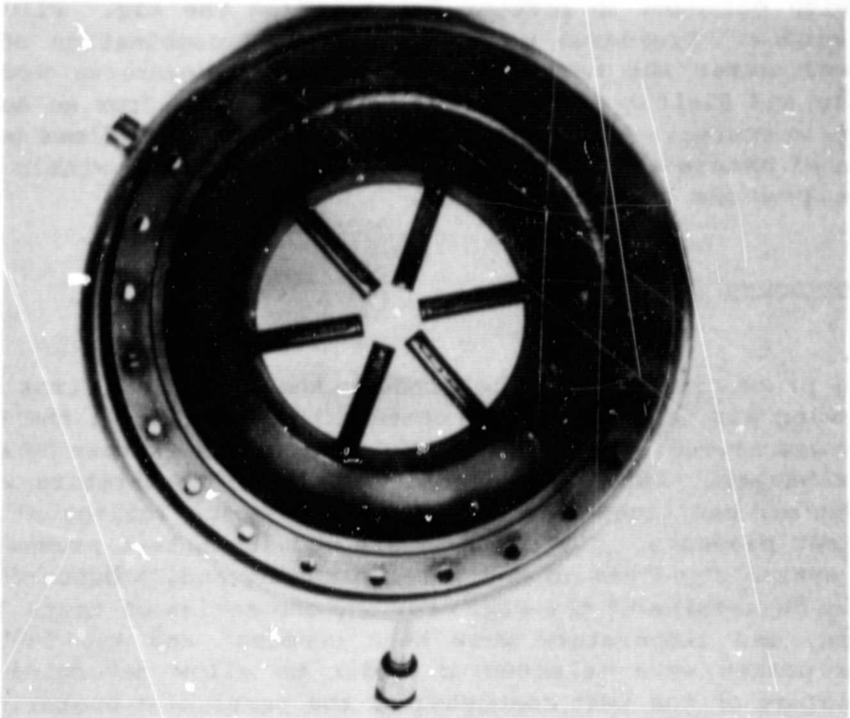

Figure 27. Instrumentation Rig

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 using the Von Brand method.

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 28 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 an Autodata Nine 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 29 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 both combustor skin temperatures and fuel and air temperatures were continually monitored and then recorded on printed paper tape. Second,

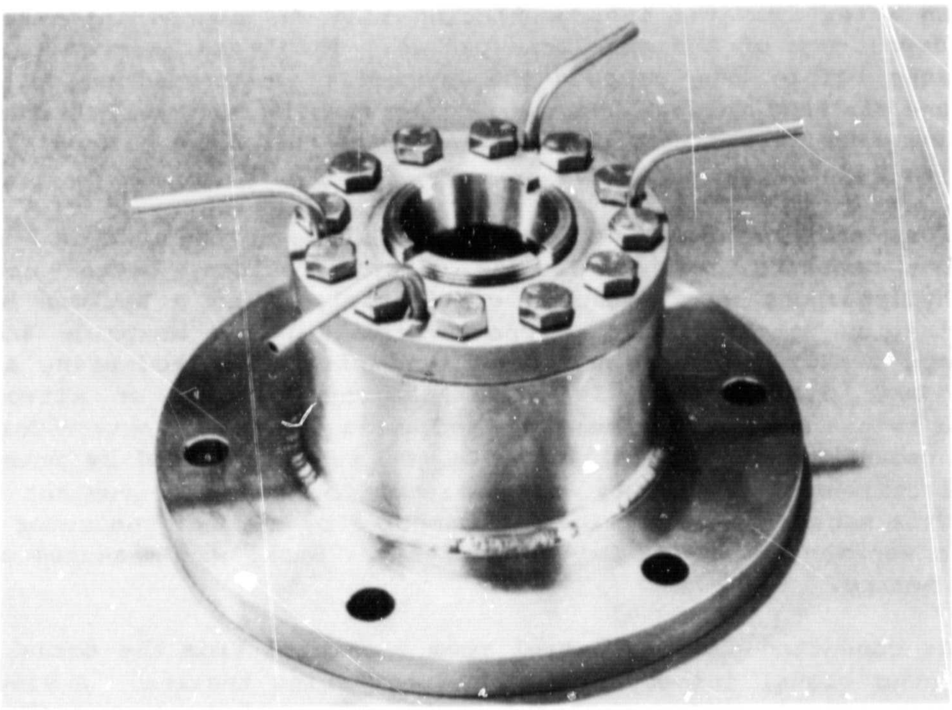

Figure 28. Quartz Window

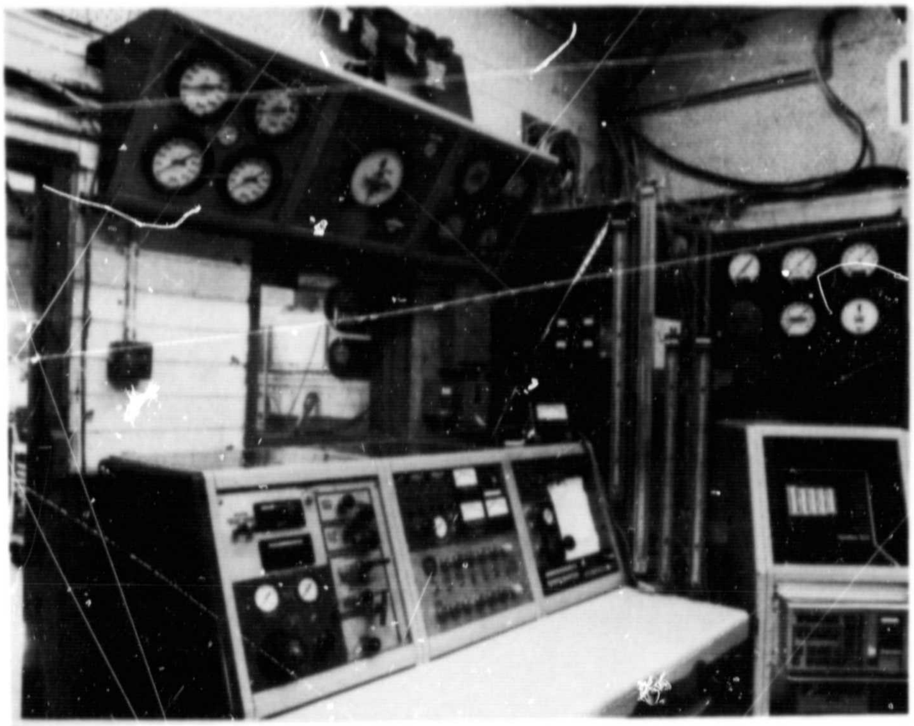

Figure 29. Control Panel

the rig operating data were recorded by hand and 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, cooling the rig by continuing to flow air through and back-purging the fuel system to a drum.

6

TEST RESULTS AND DISCUSSION

6.1 TEST RESULTS

The prime combustor approach adopted to provide low Nox emissions was the rich primary-lean secondary zone concept which consisted of configurations 1.1, 1.2, 1.3 and 1.4. Initially the evaluations concentrated on configuration 1.2 which utilized an air-assist fuel injector. Mechanical integrity problems involving primary zone liner temperatures above 1050°C (1922°F) limited initial operation to low air inlet temperatures and pressures. To remedy these problems a developmental program was undertaken that resulted in a shorter primary zone. This shortening of the primary zone reduced the surface area that required cooling. The total heat flux originally estimated for the rich primary zone was much lower than that found in practice. A peak level of 1,900,000 W/m^2 (600,000 Btu/hr ft^2) was measured in operation, whereas during design a peak level of only 950,000 W/m^2 (300,000 Btu/hr ft^2) was assumed. After shortening it was found that the main hot-spots in the primary zone were located in the transition piece and the rear primary zone conical contracting piece. By revising the primary air inlet, and adopting a combination of features of 1.4 and 1.2, a satisfactory solution to the mechanical integrity problem was provided. In essence the primary air inlet was moved from the rear of the zone to a point approximately 2/3 of the zone length from the inlet swirler. Secondary air holes were added to the transition piece, to allow partial entry of secondary air at that point as in configuration 1.4. The remainder of the secondary air was injected at the rear of the secondary zone through forward directed entry ports. These jets mutually impinged on the centerline. This latter form of air injection provides the secondary zone with a stable torodial vortex with a high level of recirculation. A vortex arrangement with recirculation mass ratios greater than one, when used as a combustion device, approaches the behavior of a well-stirred reactor. This behavior has been found in practice. A schematic of the final combustor configuration mounted in the test rig is provided in Figure 30. A skin temperature profile for the final combustor development when operating at maximum power is shown in Figure 31. The peak temperatures shown are at the throat (transition) and the rear of the primary zone. These peak temperatures are limiting values for mechanical integrity, and higher air inlet temperatures could not be tolerated.

Emission signatures for this particular finalized combustor have been determined over the entire engine operating range. Specifically combustor exhaust emissions levels of NOx, CO, UHC, smoke and CO_2 were determined for each of the engine test points quoted in Table 1, with each of the required test fuels, and associated blends. The fuel that most emphasis was placed on for these tests was the solvent refined coal SRC-II. Emission levels of NOx and CO when burning ERBS fuel at low pressures are shown as a function of approximate

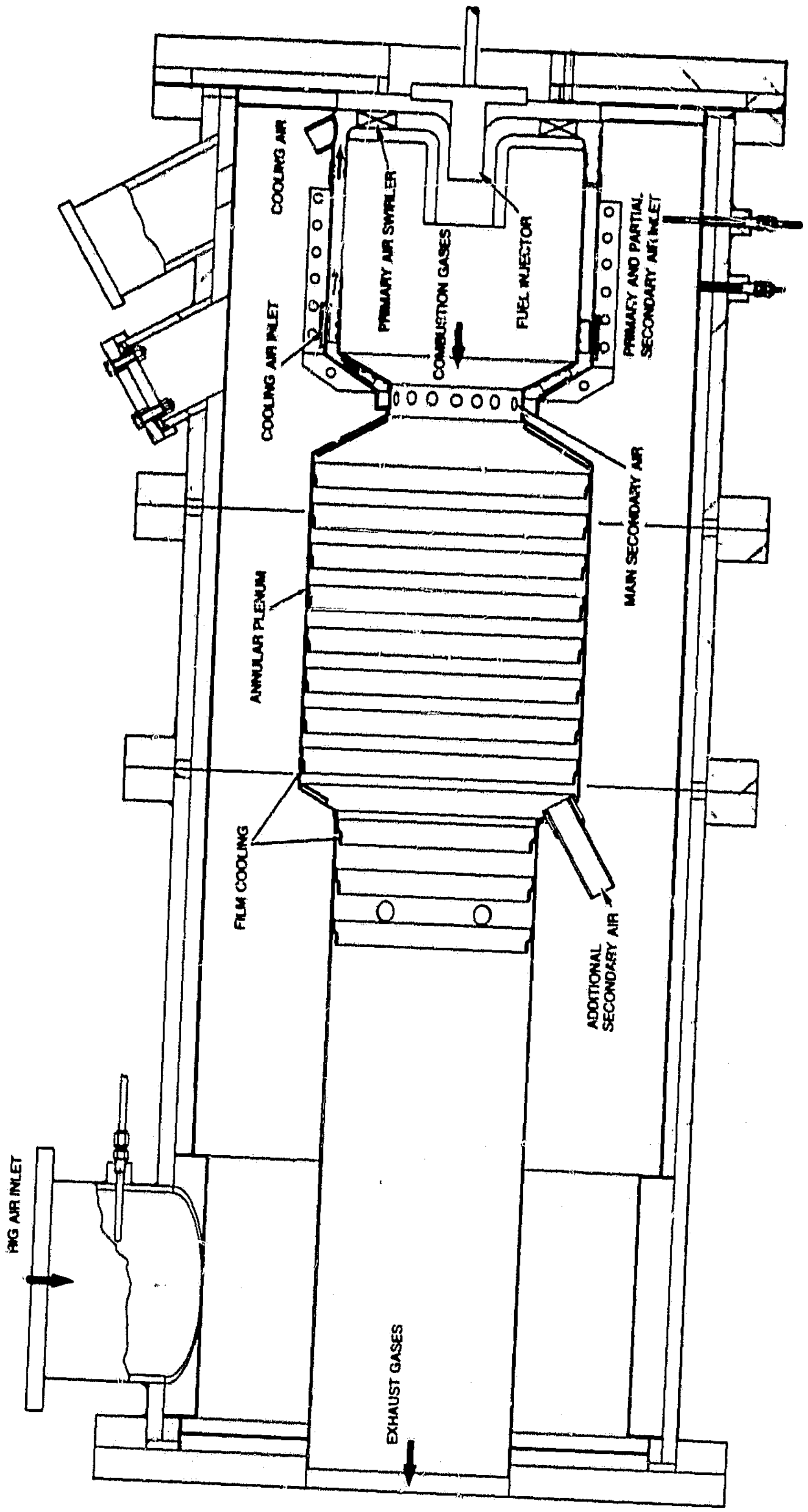

Figure 30. Final Rich-Lean Combustor Configuration

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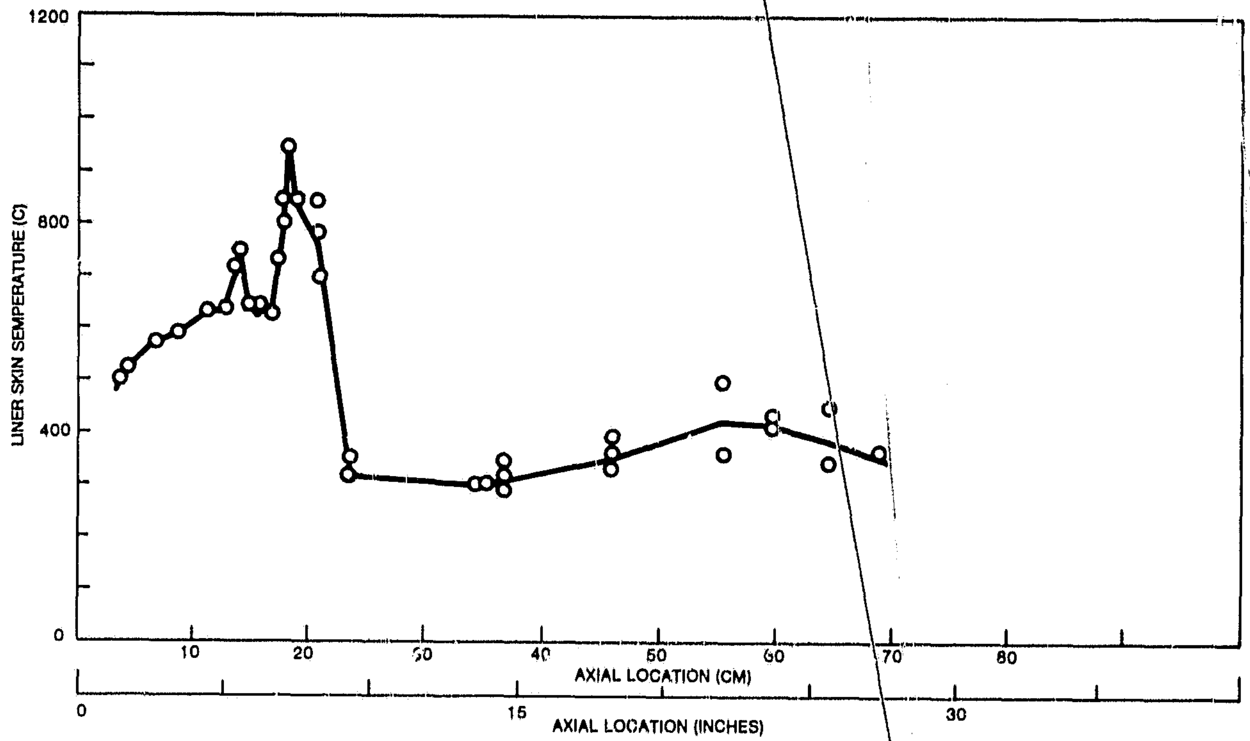

Figure 31. Skin Temperature Profile for Final Combustor Design

primary zone equivalence ratio in Figures 32 and 33. A set of data for SRC-II is provided in Figures 34, 35, 36 and 37. Figures 38 and 39 represent complete operating maps at close to the maximum power point of a typical 12:1 ratio gas turbine engine for SRC-II and ERBS respectively. The lowest recorded values of the equivalence ratio on these graphs being the last completely stable operating point before lean extinction. These latter curves which show NOx emission levels at both rich and lean primary zone conditions make clear the superiority of operating with a rich primary zone for high fuel-bound-nitrogen fuels. The emissions obtained burning residual fuel at the highest practical temperature and pressures allowed by mechanical integrity are shown in Figure 40. Essentially the SCR-II fuel contained 0.9 percent by weight of fuel bound nitrogen. The ERBS fuel contained roughly 0.01 percent by weight, and thus a blend of 50 percent ERBS and 50 percent SRC-II by weight provided a fuel bound nitrogen content of 0.455 percent. A blend of 75 percent SRC-II and 25 percent ERBS provided 0.6775 percent fuel bound nitrogen and a blend of 25 percent SRC-II and 75 percent ERBS produced 0.2325 percent. Emissions signatures of each of these blended fuels and the pure base fuels are comparatively displayed in Figures 41 and 42. This shows that the minimum NOx value produced under rich conditions (at a primary zone equivalence ratio of approximately 1.6), is relatively insensitive to the level of chemically bound nitrogen in the fuel. This phenomenon is important in that a properly designed combustor could be capable of handling a wide variety of fuels.

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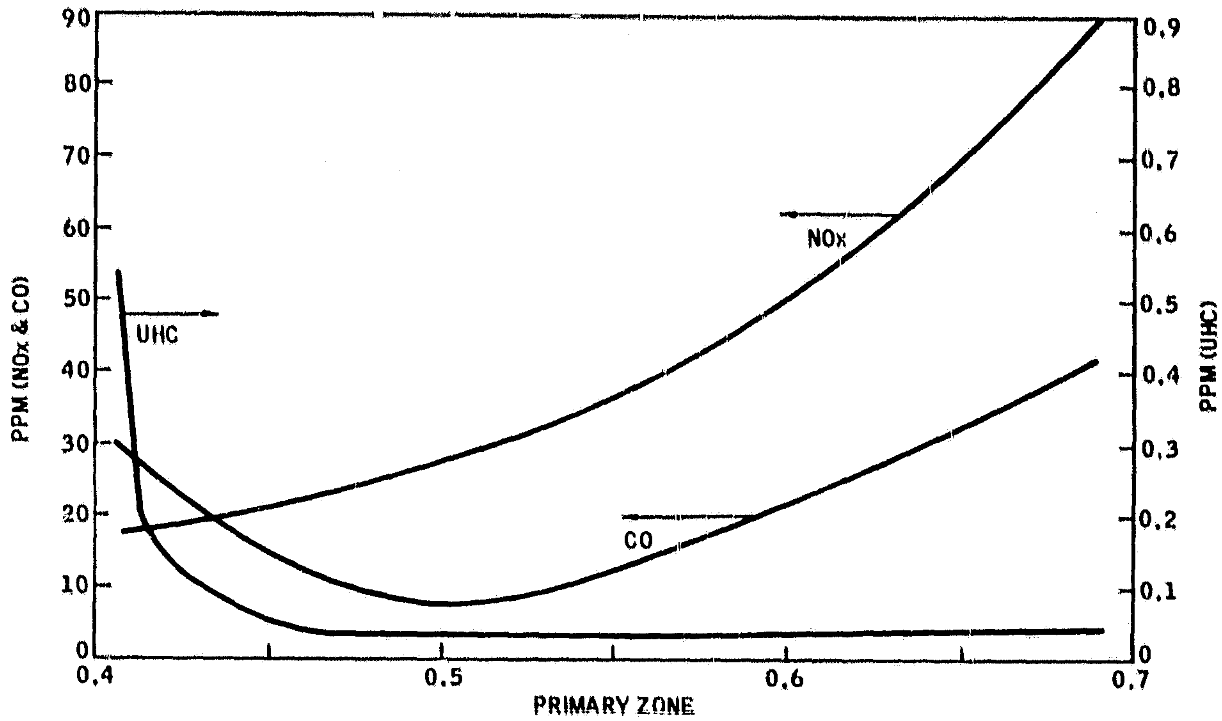

Figure 32. Atmospheric Emissions Signature

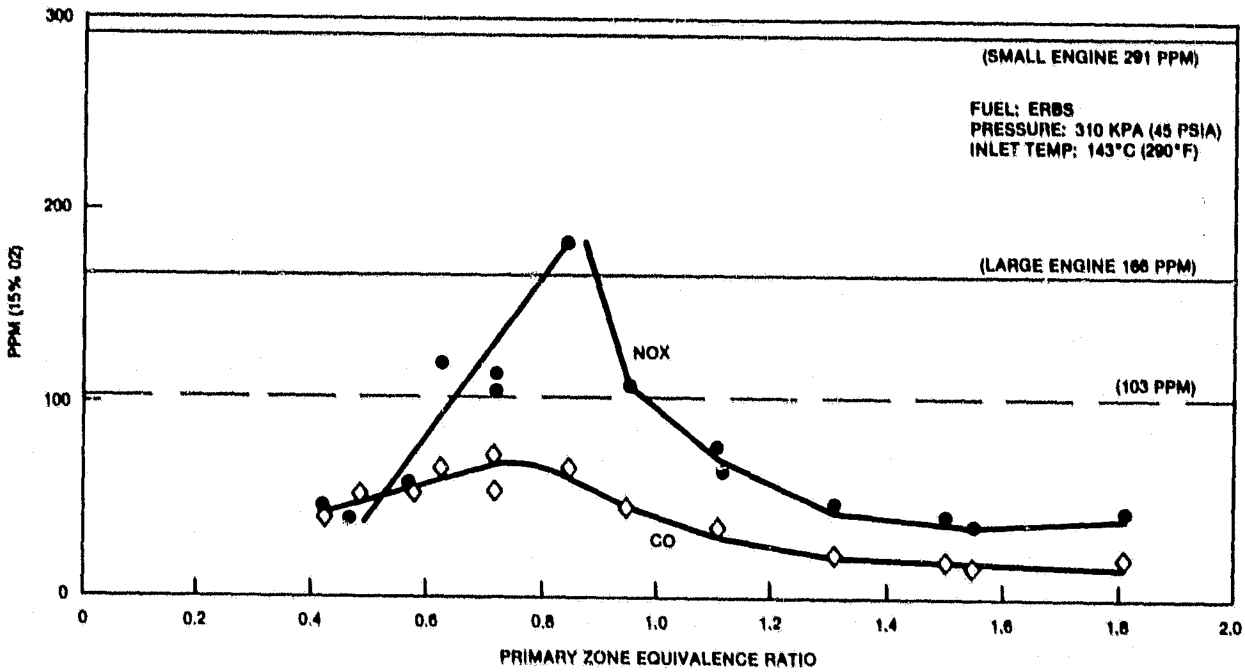

Figure 33. Low Pressure Emissions Signature With ERBS

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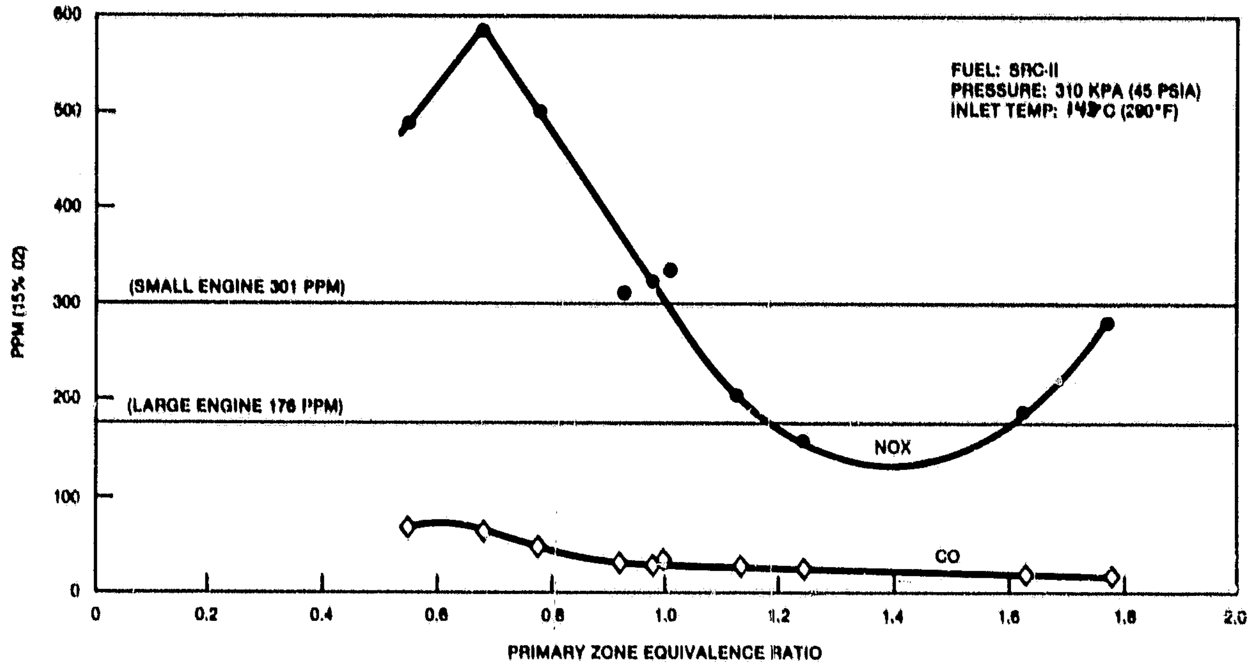

Figure 34. Low Pressure Emissions Signature for SRC-II

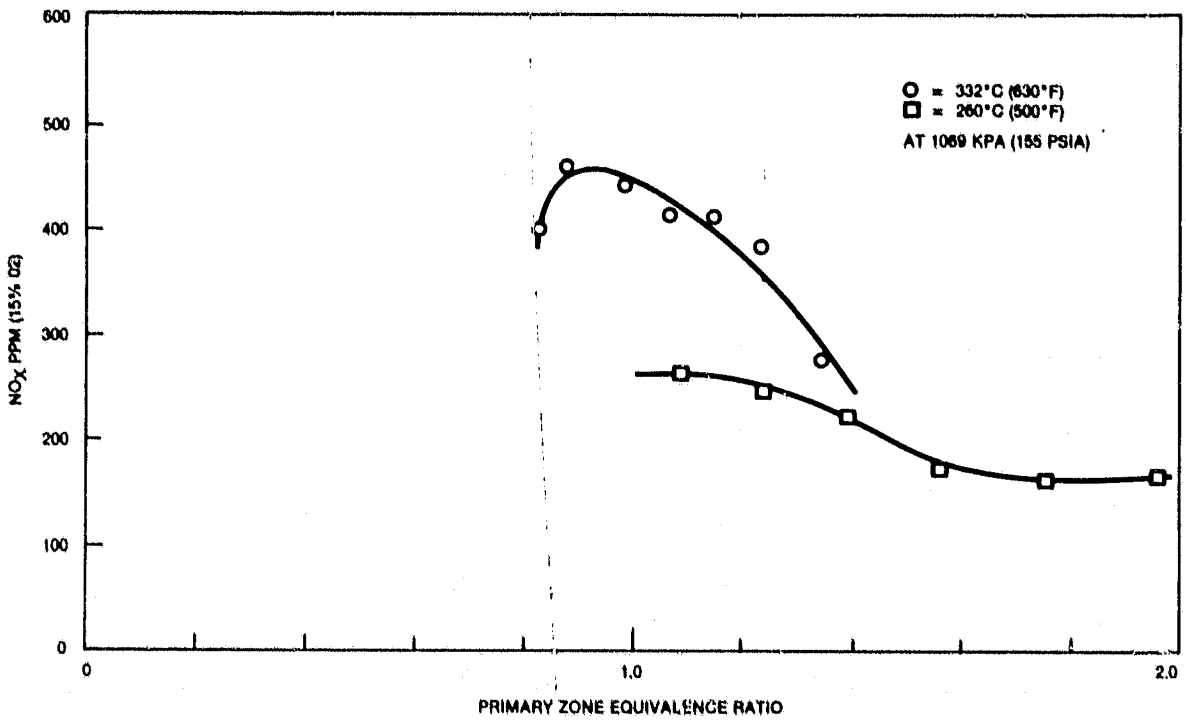

Figure 35. NOx Emissions Signature for SRC-II

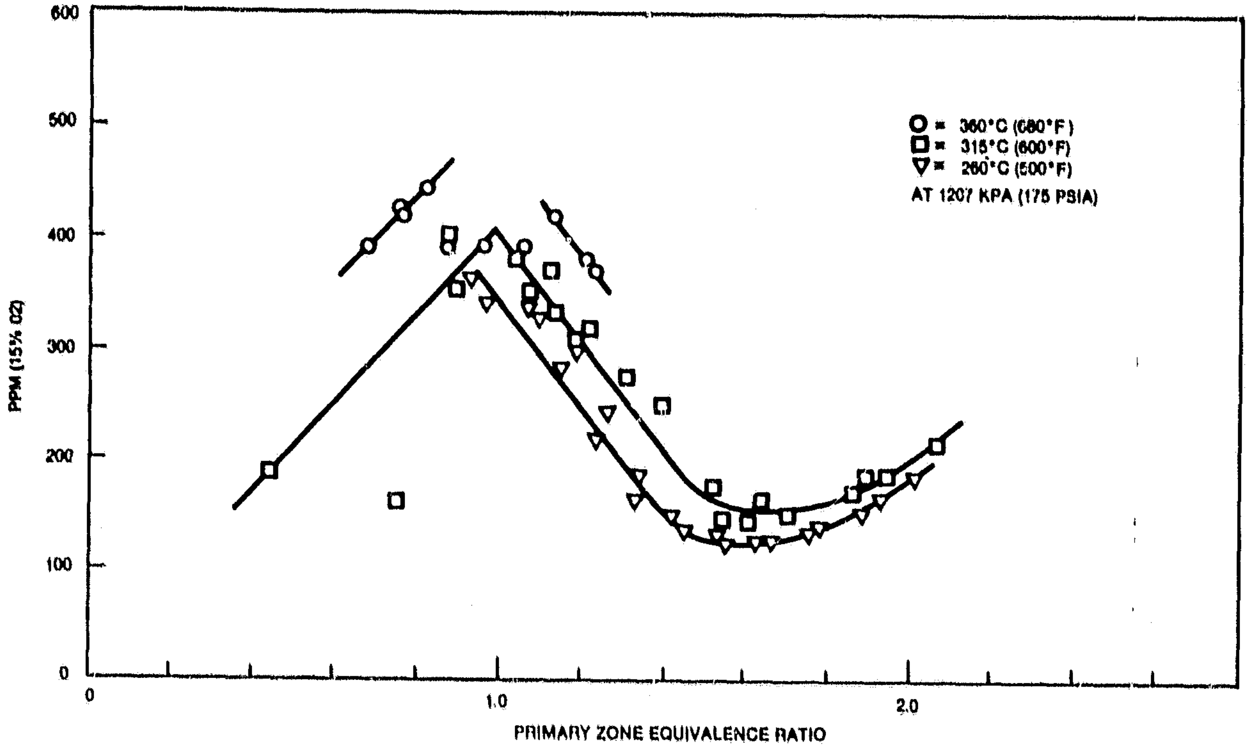

Figure 36. NOx Emissions Signature for SRC-II at Various Temperatures

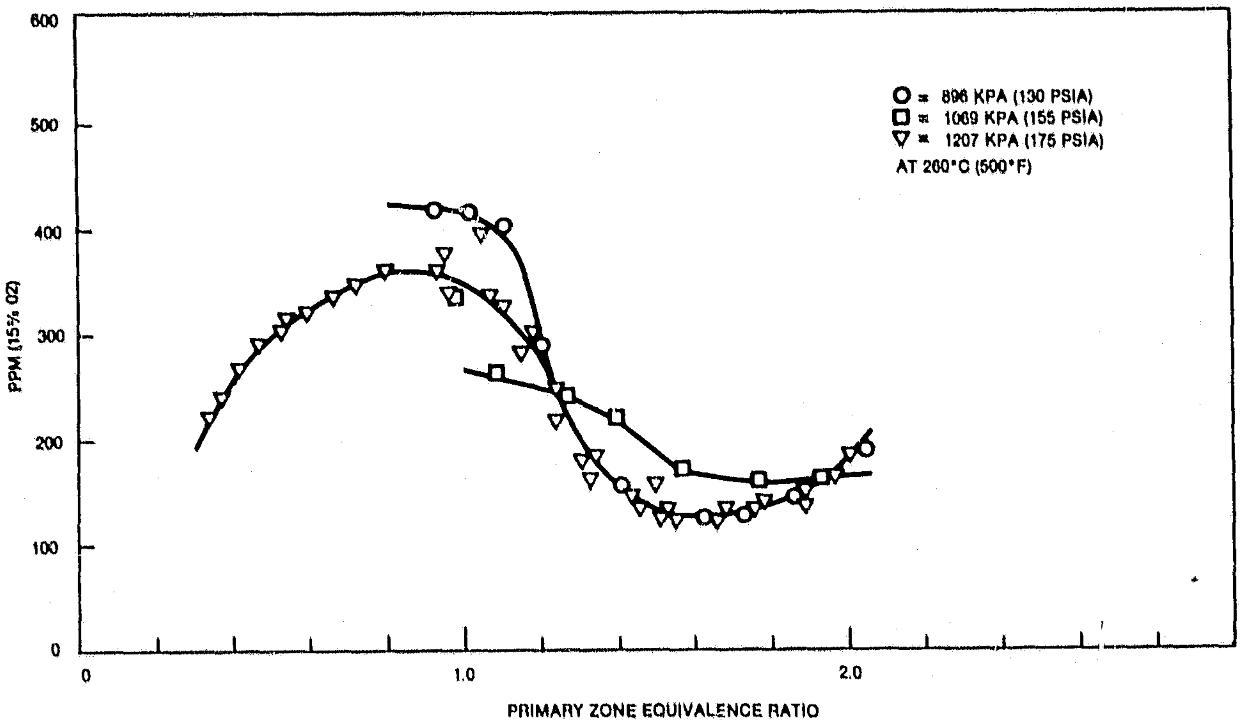

Figure 37. NOx Emissions Signature for SRC-II at Various Pressures

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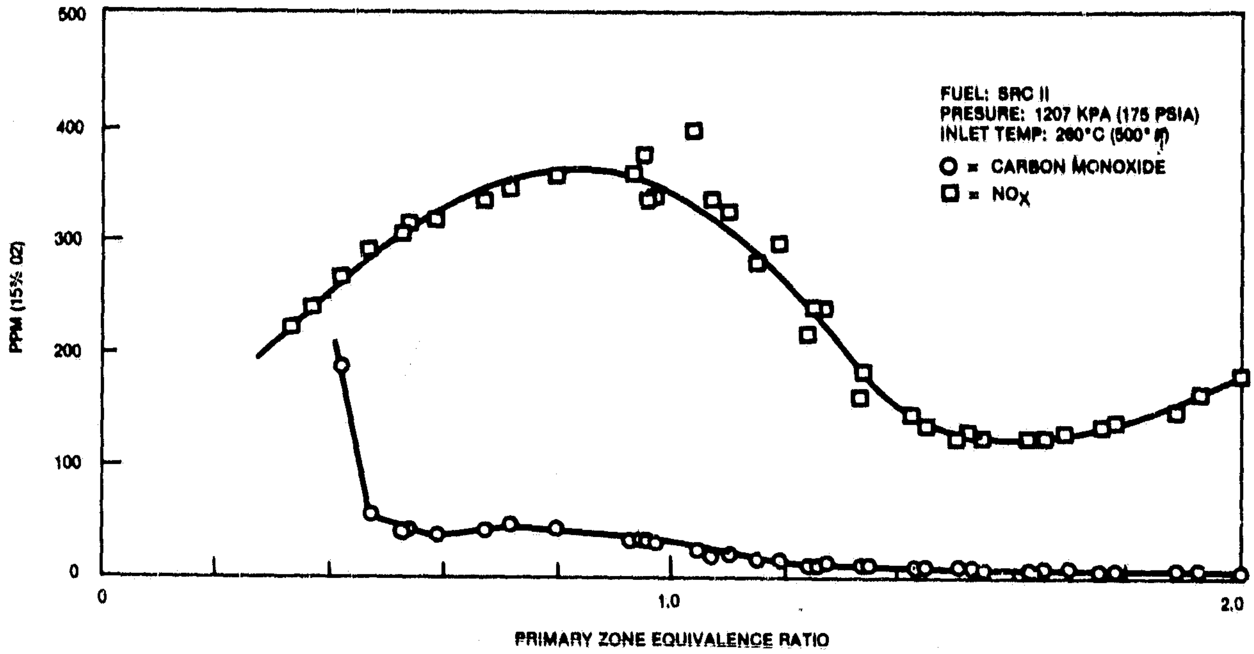

Figure 38. Complete Emissions Operating Map for SRC-II

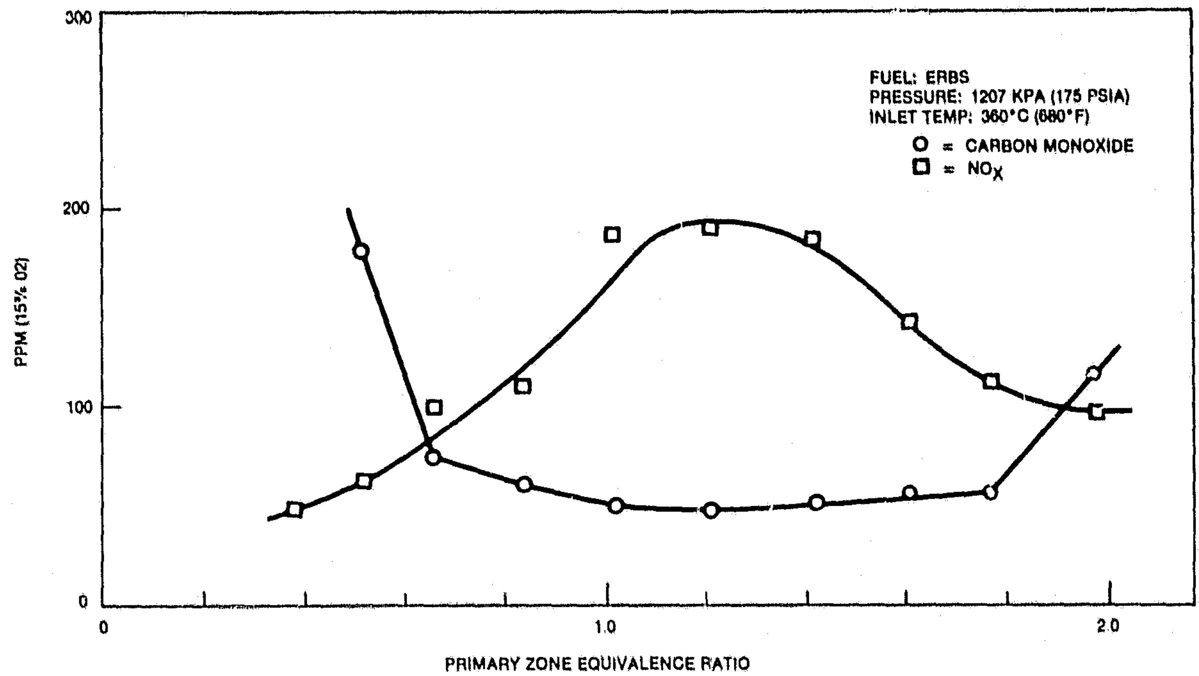

Figure 39. Complete Emissions Operating Map for ERBS Fuel

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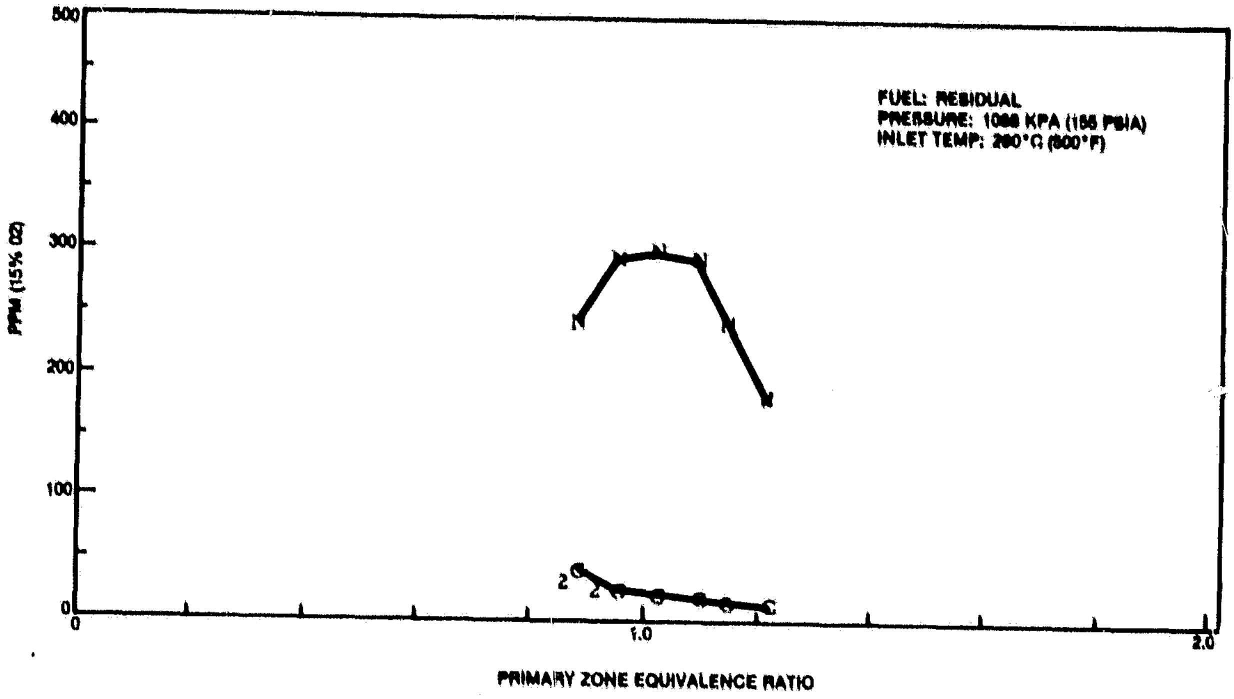

Figure 40. Emissions Signature of Residual Fuel

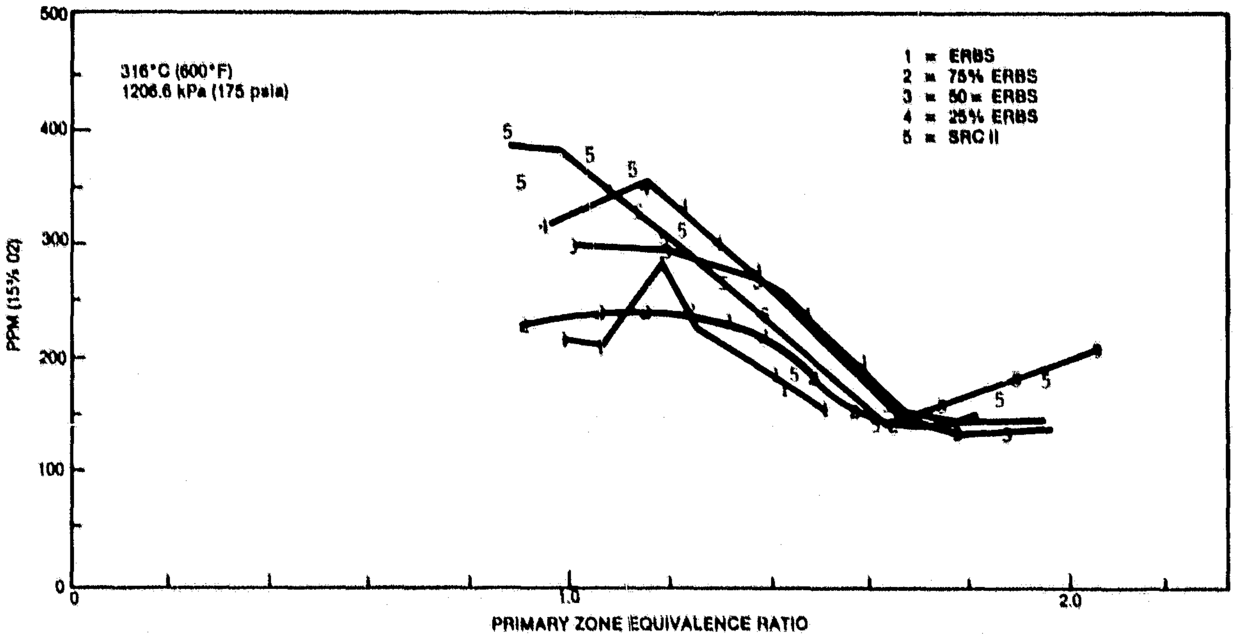

Figure 41. Comparative Emissions Signatures for SRC-II/ERBS Blends

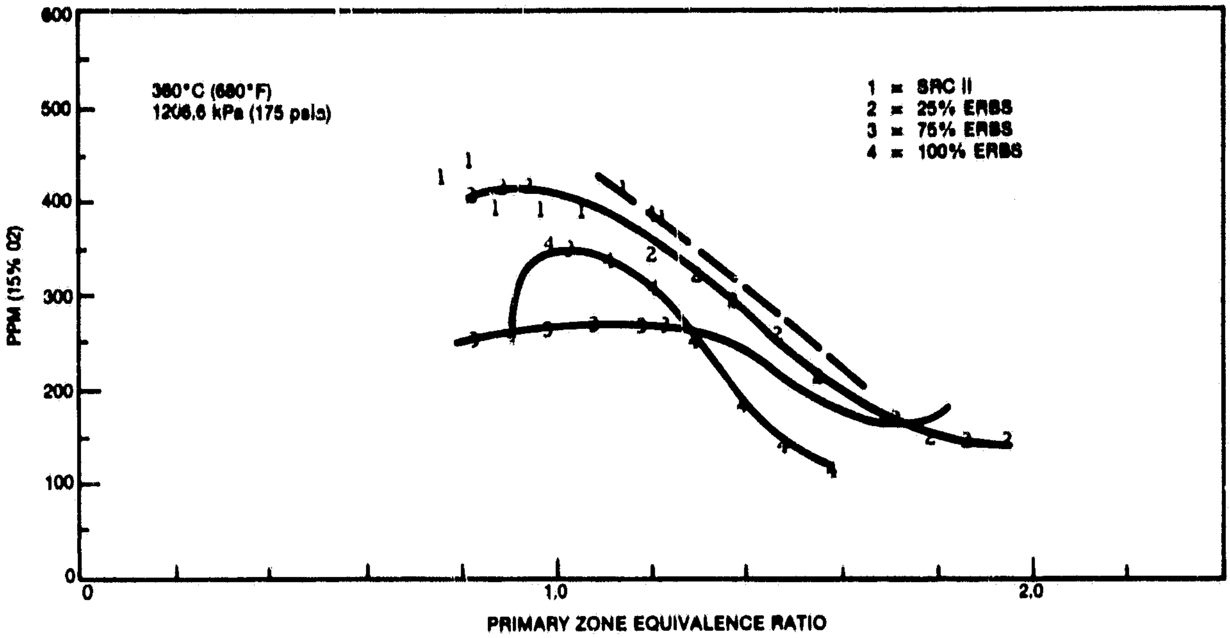

Figure 42. Comparative Emissions Signatures for SRC-II/ERBS Blends

Tabulated emissions data for the above results are provided in Appendix A.

Figure 43 shows the variation in NOx versus engine load for both SRC-II and ERBS. It can be observed that there is a significant difference in NOx levels between the fuels which would seem to contradict statements made above. The reason for the difference is that the simulated engine operating points do not occur at an equivalence ratio where the NOx is minimized. For instance (refer to Figure 42) the 100 percent load point corresponds to an equivalence ratio of 1.4. By matching the combustor at a richer primary zone equivalence ratio the NOx level on SRC-II could be reduced.

Some attempts were made to evaluate the effects of residence time by varying airflow at the same inlet pressure and temperature with the same combustor hardware. The data were not complete enough to make a definite conclusion. The data that were obtained are presented in Figure 44.

Configuration 1.3 and 1.4 were found to be impossible to operate due to severe overheating of the transition pieces. In addition, because of mechanical problems involving face-seals in the spinning-cup, no reliable data could be obtained with configuration 1.1 and 1.3.

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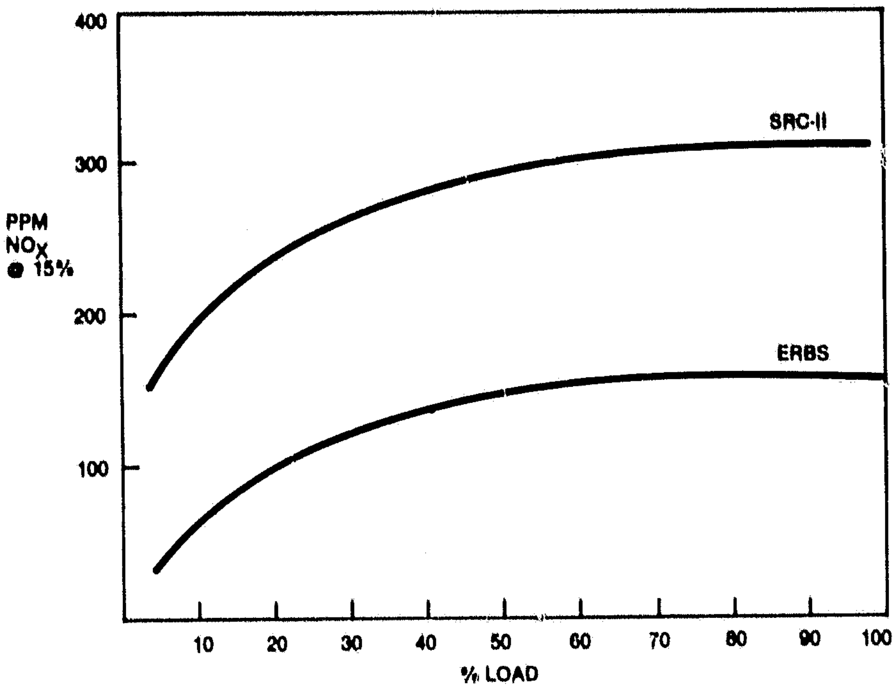

Figure 43. Variation in NOx With Engine Load for SRC-II and ERBS Fuels

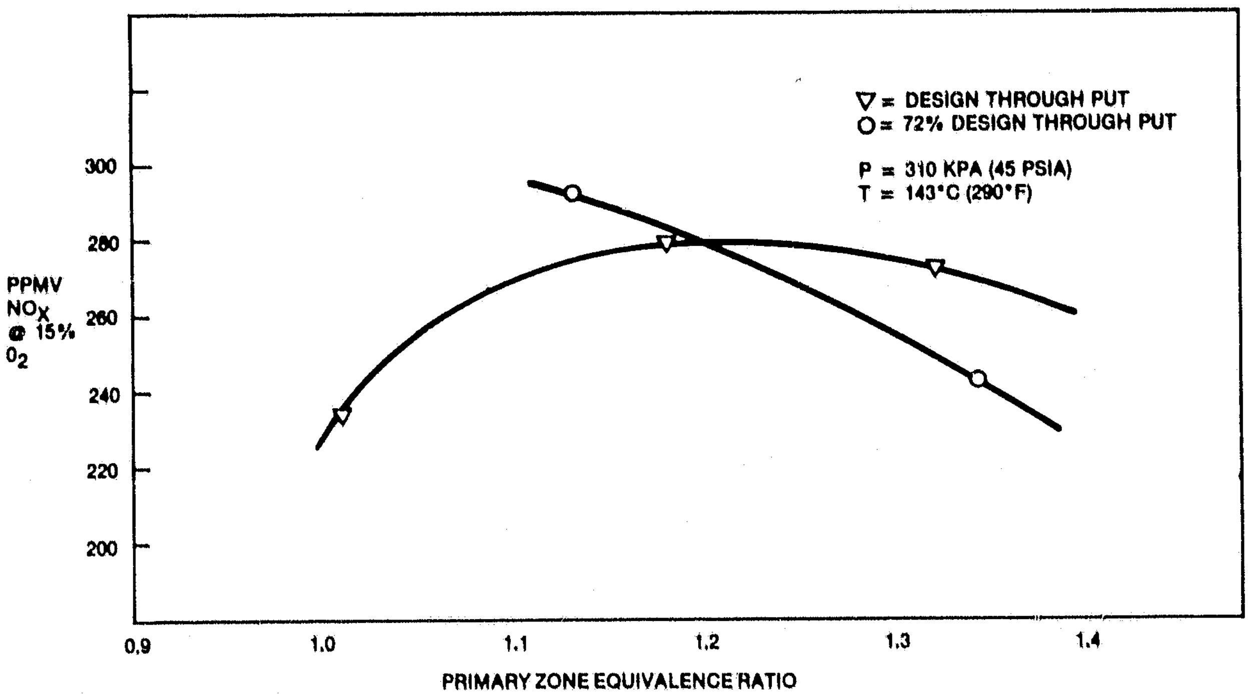

Figure 44. Effects of Throughput on NOx Production

Concept 3 (lean combustion with direct fuel injection) had two configuration versions, one employed a spinning cup for the fuel injection (3.1) and the other an air atomizing fuel injector (3.2). This particular concept utilized the rich primary zone body on first design with a larger swirler. The radial airflow swirler used was basically the same design as that used during rich operation but allowed a larger air flow into the primary zone.

No data could be obtained with configuration 3.1 which utilized the spinning cup because of problems with the face seals, which acted as a "brake". During operation the speed of the spinning cup could not be maintained at a constant value. Further development would be needed to ensure that the device would work well enough to be used experimentally as a fuel injector.

Configuration 3.2 was successfully tested and the results of this configuration were used mainly as a baseline for other configurations. If the results of the other configurations were not better than those of 3.2 then they were rejected. Essentially configuration 3.2 simulated a "leaned-out" conventional combustor. It had a variable primary zone equivalence ratio ranging from 0.6 to 0.9. These various values were obtained by changing the effective secondary port diameters through the use of inserts.

Both ERBS and SRC-II fuels were evaluated. Because of wall overheating in the primary zone at conditions approaching those of the maximum power point, a version was developed that utilized combined convective/film cooling. This consisted of adding a conventional splash-ring at the mid-point of the primary zone parallel section, and one at the rear contracting section. Air was bled from the annular convectively cooled air-space at these two locations, and then directed as a film along the inside of the primary zone wall.

The emissions results of this final developed version at various conditions are shown in Figures 45, 46, and 47 for both ERBS and SRC-II fuels. These emissions correlations show clearly the effects of operating pressure and temperature on the levels of NOx produced. At pressures in excess of 793 kPa (115 psia) the effect on NOx is significant, however, below this value pressure has little effect on the NOx produced. Inlet air temperature has a large effect on NOx levels. With increasing temperature levels the NOx increases. The difference in NOx emissions due to the use of SRC-II rather than ERBS fuel in the lean-lean combustor is shown in Figure 47. This clearly shows the strong influence of fuel-bound nitrogen on the total NOx emission level. In general these emission characteristics are similar to those obtained with a conventional combustor, and are unlike those of a premixed lean system. Smoke emissions were monitored using a Von Brand smokemeter. No smoke was ever detected with this device during these tests. The emissions produced by this lean system contained considerably higher NOx, when burning SRC-II fuel, than the emissions obtained at the "minimum levels" during rich primary combustion. Thus it was realized that the rich primary zone combustion system definitely provided a reduction in NOx when compared to semi-conventional primary zone operation.

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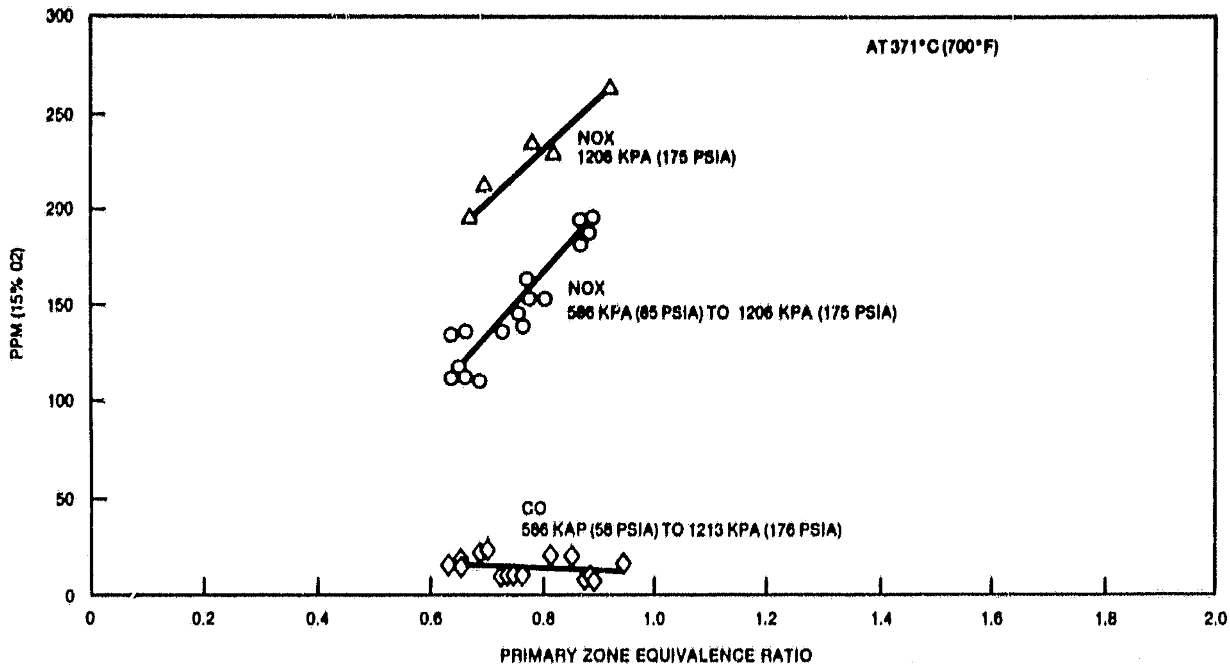

Figure 45. Lean-Lean Emissions Characteristics With ERBS Fuel

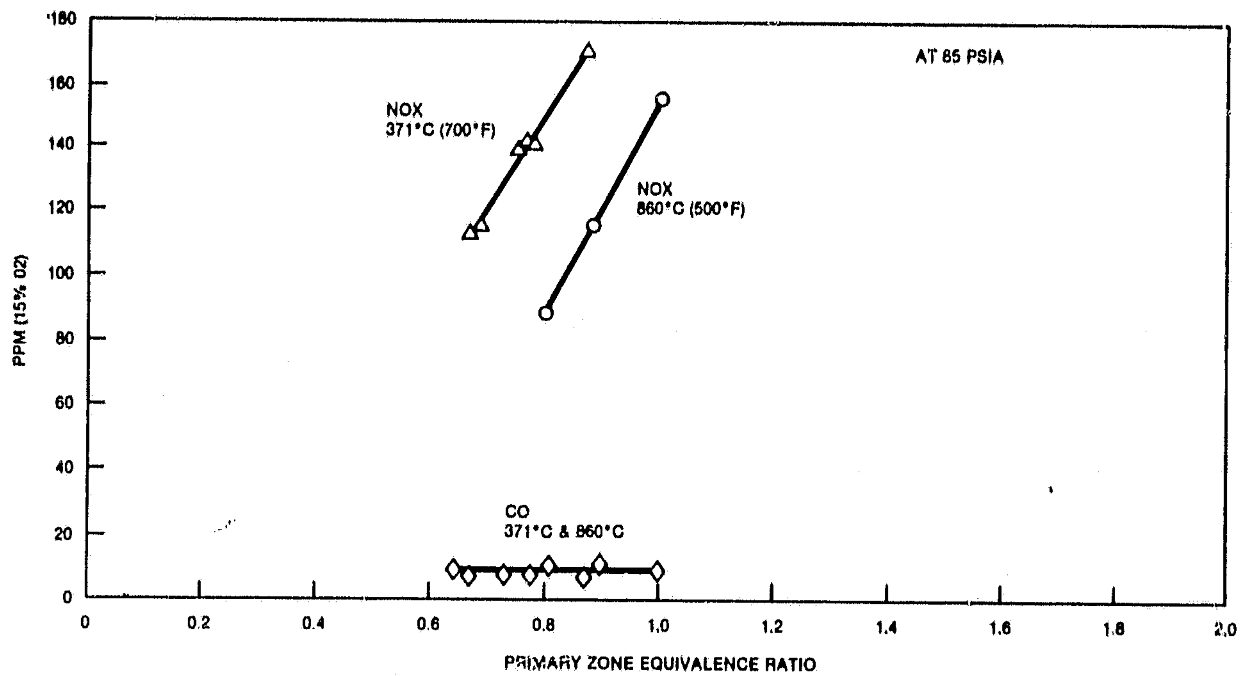

Figure 46. Effect of Temperature on Lean-Lean Emissions Characteristics (ERBS Fuel)

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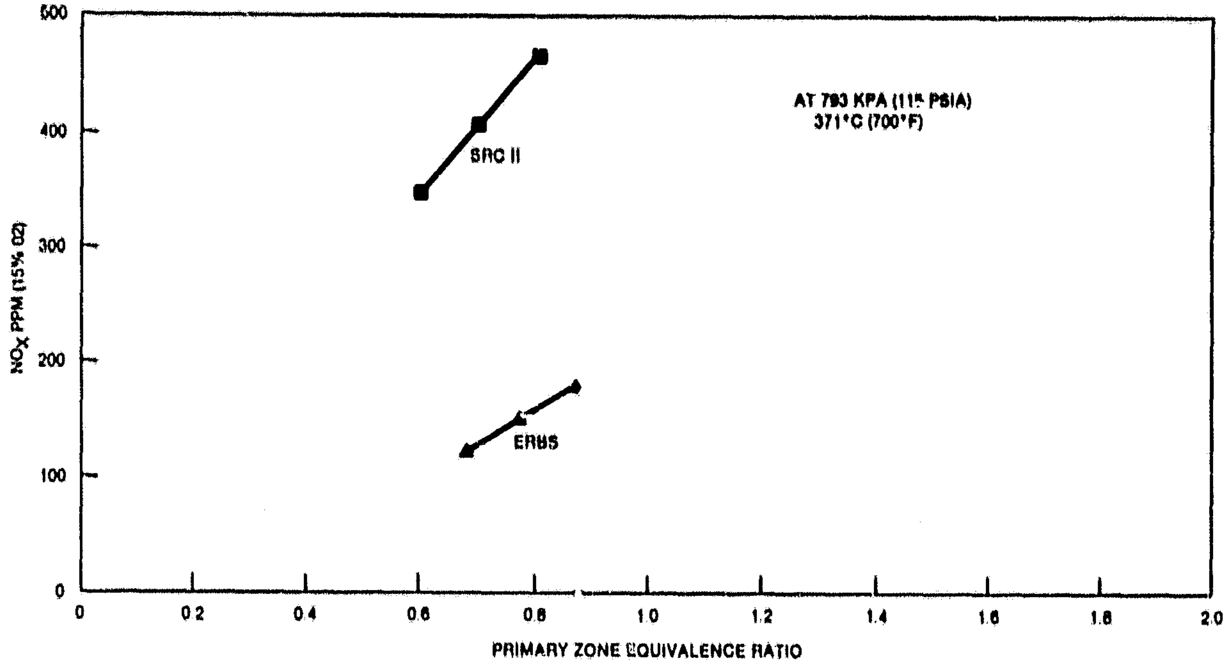

Figure 47 Effect of Fuel Type on Emission Characteristics

The fourth concept which consisted of a single configuration (4.1), was basically a fuel staged version of concept 3. In operation fuel was added to the secondary zone at conditions less than approximately 70 percent power. The fuel was injected through the rear secondary ports where it was air-atomized before entering the secondary zone proper. This concept was not successful mainly because of mechanical integrity problems. Overheating of the secondary zone dome walls was encountered The temperatures measured on the dome were of the order of 1100°C which could not be tolerated for more than a few seconds.

Concepts 5 and 6 were not tested, however, considerable data is available at Solar Turbines Incorporated on the emissions signatures of these lean premixed systems when burning ERBS type fuels. Data from a combustor of the same dimensions as that planned to be used is shown in Figure 48. These show the extremely low levels of NOx that can be obtained with clean fuels. Additionally Figure 49 has been included which shows the effect of air inlet temperature on the NOx emission levels As shown NOx levels decrease with increasing inlet air temperatures. This is a common occurrence in lean premixed combustor designs and is probably due to improvements in fuel vaporization and mixing. Smaller deviations from the mean fuel-air ratio generally produces lower NOx levels. All the fuels utilized were analyzed in detail at STI, and a comprehensive list of fuel properties is provided in Appendix B. In addition, this Appendix includes the viscosity-temperature relationships for each fuel used.

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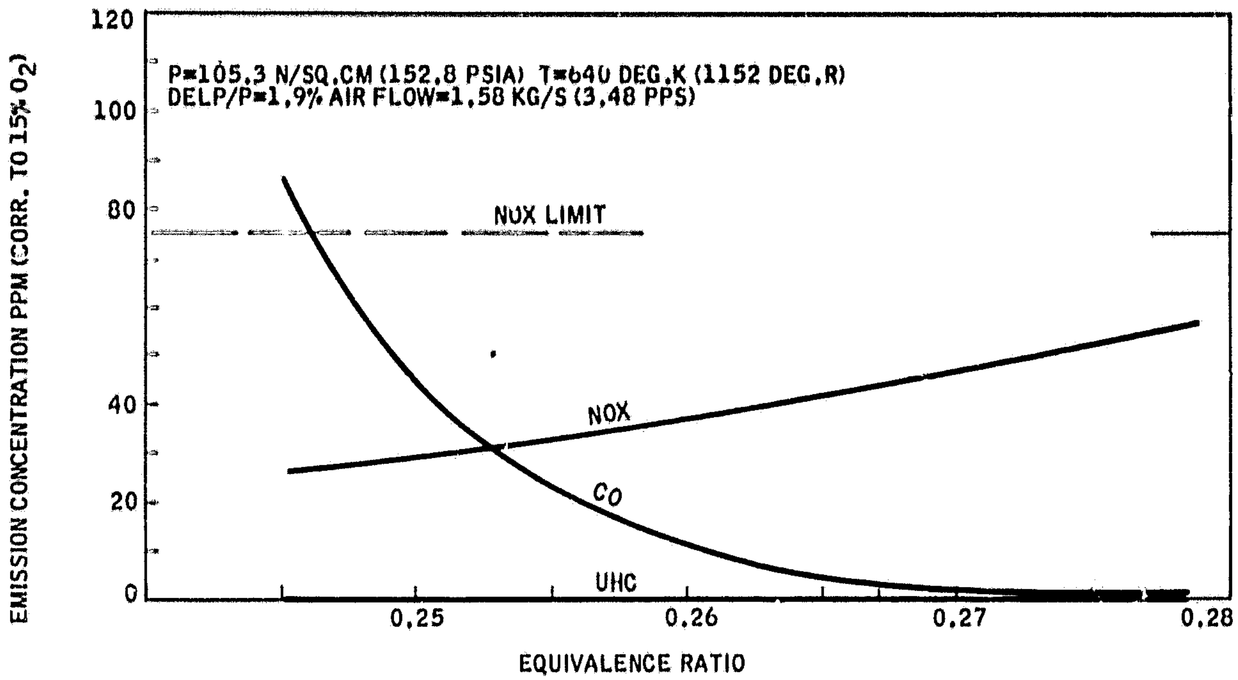

Figure 48. ERBS Type Fuel Emission Characteristics With Lean Premixed Combustors

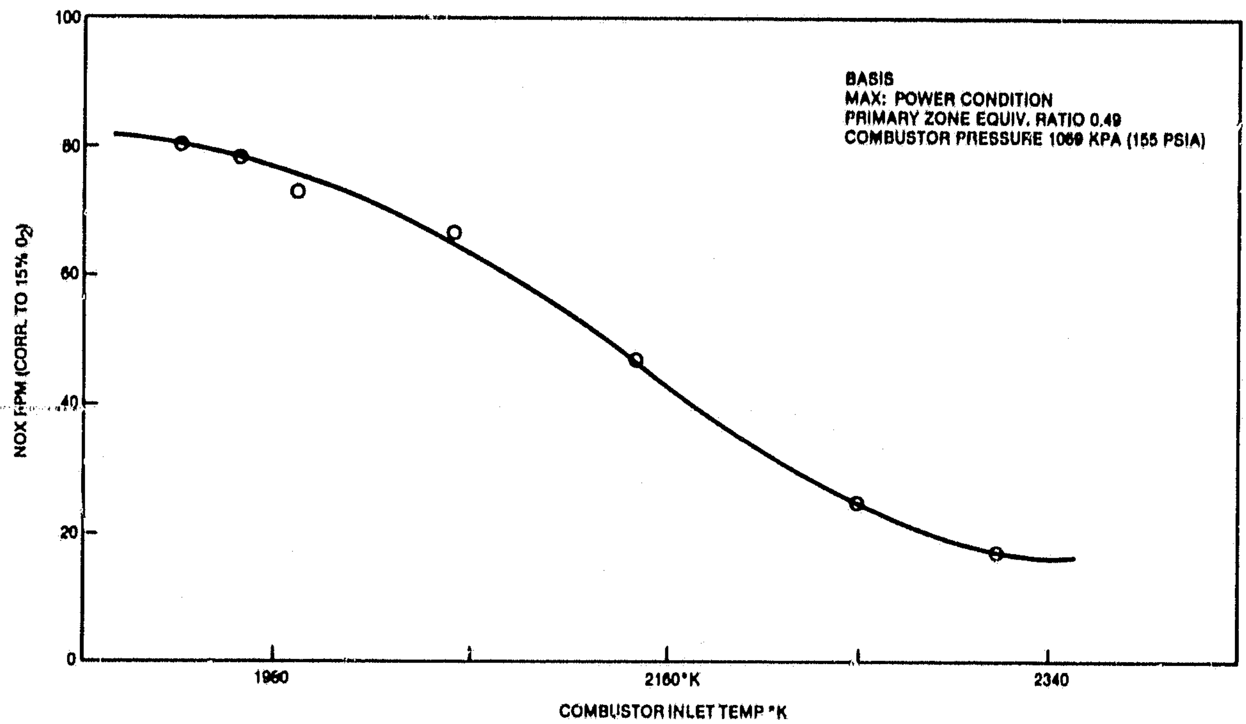

Figure 49. Effect of Air Inlet Temperature on NOx Emissions

6 2 INTERPRETATION OF RESULTS

The NOx emissions produced during combustion of SRC-II and ERBS fuels in the rich-lean combustor show a minimum at an equivalence ratio of approximately 1.6 This latter value is higher than the originally anticipated value of 1.2 for the minimum, which was based on data obtained during atmospheric combustion. It would appear from the data generated that there is a shift in the NOx minimum to higher equivalence ratio values with increasing pressure. Above approximately 414 kPa (60 psia) the minimum NOx level is found at an equivalence ratio of 1.6, below this pressure the minimum point is exhibited at lower values of equivalence ratio Not enough details are available to determine the relationship between the mininum NOx equivalence ratio point and pressure, however, it appears to reduce to a level of 1.2 at near ambient pressure.

The effect of pressure on the absolute value of NOx at the minimum point is complex; it tends to increase the NOx level up to approximately 965 kPa (140 psia) but at pressures above this point a reduction was noticed. This may well be a function of the combustor geometry and fuel injection method rather than a fundamental relationship with pressure.

Combustor inlet air temperature can be seen to have a more significant effect on NOx emissions than pressure (see Section 3). In general as the inlet air temperature increases the overall NOx levels increase. Although the data is too sparse to provide definite conclusions, it would appear that the changes in NOx level at the minimum point (at constant pressure) are less than at stoichiometric. For operating pressures in excess of 965 kPa (140 psia) the relationship between NOx at the minimum point with pressure and temperature is given by:

$$\text{NOx} = 281 \times P^{-0.56} \times \exp\,(T/256)$$

P = atm
T = °K
NOx = ppm @ 15% O_2

This latter relationship suggests that as the pressure increases, NOx will reduce a little, however, as the air temperature increases the NOx will increase significantly.

The NOx value at the minimum point appears to be fairly independent of the level of fuel-bound nitrogen over a wide range. This suggests that the conversion of fuel-bound nitrogen to NOx is low, although the thermal NOx level could be high. It would be difficult to pinpoint the area within the combustor that contributed the most to the thermal NOx level. The most likely areas are the primary fuel-air mixing point and the point of secondary air addition. Improved premixing of the primary fuel and air will, it is felt, lower the NOx somewhat. Penalties will probably have to be paid in the present system if better premixing is to be achieved; these being fouling of the fuel injector and swirler. To provide better premixing of the primary air and fuel with the present fuel injector, the fuel nozzle would be moved

further inside the radial inflow swirler. This however, tends to produce fuel deposits on the outer wall of the swirler annular exit passage and on the fuel injector centerbody It may be possible to redesign the fuel injection system and primary swirler passages to minimize this problem. An axial flowing air film on the surface of the swirler wall could, for example, minimize this problem.

The results in general indicate that if the NOx emission levels were to be met over the entire operating range of a typical 12:1 pressure ratio engine, then some form of variable geometry would have to be utilized. With the system used it would be possible to add a valve to a manifold that fed each of the secondary ports, and an additional valve to a manifold arrangement surrounding the air entry holes to the primary zone. Each manifold in this instance would have to be large enough to minimize variation in port or entry hole discharge coefficients caused by high velocity crossflows. No major problems are anticipated in mechanizing the variable geometry, based on extensive past experience in designing and building variable geometry combustors Both primary and secondary air entry areas would have to be varied in the rich-lean system to avoid excessive pressure-drops. If only the secondary air port areas were varied to affect a change between the primary and secondary air splits then pressure drops well in excess of 6 percent could be encountered.

One of the major problems highlighted by this work is that of mechanical integrity, particularly of the primary zone and the primary to secondary zone transition piece. The high temperatures found toward the rear of the primary zone and in the transition piece could possibly be due to secondary air penetration and subsequent reaction. The temperatures produced in such local hot spots approach stoichiometric levels, creating local overheating of the walls. In addition, the contracting rear walls have a high view-factor; they face the rich flame which has its highest temperature adjacent to these wall, and thus they experience high radiation loads.

It should be possible to minimize these excessive wall temperatures by ensuring the following:

- minimize the secondary air entrainment in the primary zone
- maintain optimum cooling passage geometry
- determine optimum air inlet position
- increase cooling air flows

Secondary air entrainment by the primary zone recirculating flows could be minimized by injecting the throat secondary air as a series of angled jets. These latter jets would be angled downstream toward the secondary zone so that after mutual impingement, the bulk of the flow (in the major desired jet) would be forced into the secondary zone. A small secondary derived jet would still move upstream toward the primary zone proper, however, the mass flow associated with this jet is relatively small and should not provide a major change in the local stoichiometry.

The "trip-strip" cooling system adopted for the primary zone proved to be a very effective convective cooling method, however, it did prove to be sensitive to geometry changes. Thus during operation when the outer wall, being at a lower temperature than the inner, expanded less and closed the flow area, changes in effectiveness were found. Those areas of the combustor experiencing larger changes than others in geometry were found to have locally worsened heat transfer rates. By utilizing heavy wall construction for the primary zone (produced by either casting or by machining it from a solid billet or rolled heavy plate) better control could be achieved through the elimination of buckling. In addition, by carefully selecting the material of the cool outer wall it should be possible to ensure that it will grow the correct amount as it increases in temperature during operation. The correct amount in this instance being the geometry change that provides near constant cooling duct flow areas.

The primary air inlet position could be optimized to provide the highest possible level of cooling. For each position there would be an optimum forward pressure drop for the primary flow and an optimum rearward pressure drop for the throat secondary flow. These pressure drops would be achieved by adjusting the gap between the inner and outer walls

Further improvements in cooling could be obtained by increasing the amount of throat secondary air. There would be limits to the amount of secondary air that could be used in this fashion, these being fixed by the minimum air flow required by the reverse flowing secondary jets. These latter jets enter the secondary zone at the rear through a series of ports and naturally impinge on the centerline, and then interact with the jet exiting from the primary zone to create a local intense mixing zone and a toroidal vortex that provides long zonal residence time. Too little air injected in the rear jets will create a breakdown of the toroidal vortex in the secondary zone. This would decrease the effective residence time of the fluid in the secondary zone, and would probably result in particulate emissions. At present no smoke has been detected at any of the conditions operated at, even when burning SRC-II or residual oil.

The results of the lean-lean combustion system (3.1) were as expected similar to the emission characteristics of a conventional combustor operating in a lean mode. Poor flame stability prevented operation at primary zone equivalence ratios below 0.5, thus very low NOx emissions were not obtained. In comparing the NOx emissions levels of SRC-II and ERBS, it is apparent that a significant portion of the fuel-bound-nitrogen present in the SRC-II is being converted. Generally the conversion level is of the order of 55 percent at the low end of the operating range shown and increasing to 60-70 percent as the equivalence ratio approaches stoichiometric. Thus less of the fuel-bound nitrogen is converted at lean conditions than at stoichiometric; a somewhat surprising result. This could be interpreted as indicating a strong dependence on the critical decomposition reaction temperature. If the reaction temperature is low it is possible that not all the nitrogen is being stripped from the base hydrocarbon molecule. This nitrogen could possibly exist for example as ammonia (NH_2) or cyanide (CN) radicals. At higher reaction temperatures more reactive and more nitrogen rich radicals such as NH can be formed which, when mixed with secondary air, react to

produce high levels of NOx. This mechanism as described is purely speculative and other explanations are possible.

Near complete premixed, lean primary zone combustors exhibit extremely low NOx levels at low primary zone equivalence ratios but the operating range is small since the low NOx point is usually close to the lean limit. Extremely low NOx of the order of 37 ppm @ 15% O_2, can be obtained at primary zone equivalence ratios of the order of 0.25 at an air inlet temperature of approximately 640°K (690°F) and a pressure of 1050 kPa (153 psia) Similar NOx levels can be obtained at somewhat higher equivalence ratios at lower inlet air temperatures and pressures.

The variation in NOx levels with temperature and pressure has been determined The temperature increases there is a reduction in NOx. Pressure, however, has little effect on emissions. The reduction in NOx levels with increasing inlet temperatures is due mainly to the improvement in vaporization and is thus a function of premixer geometry. Lean systems in general are sensitive to local rich spots, and thus every effort has to be taken to maximize the level of premixing As a rule, the main variables that can be modified readily and which also affect premixing quality are the fineness of atomization and the residence time in the premixer. At high inlet air temperatures less residence time would be needed, and larger drop sizes could be tolerated. Conversely at low inlet temperatures (idle through 70% power), residence times should be high and atomization fine. If the premixer is designed for the lower temperature conditions then at higher temperture operation autoignition of the fuel/air ratio charge could occur. Thus the limiting design factor is the autoignition delay time at the maximum temperature point with the particular fuel injection method used. This operating point with the fuel injection method determines the residence time of the premixing section. In turn, this residence time limit fixes the minimum NOx level that can be obtained, particularly at low temperature operating points

6.3 FUTURE REQUIREMENTS

In a general sense the future development of low NOx combustors will involve a dicotomy, in that it is probable that low nitrogen containing fuels may utilize the lean premixed technology whereas fuels containing high levels of fuel-bound nitrogen will use the rich-lean approaches. Both approaches, however, will require considerable development before they could be considered practical. In the lean system an effective means of fuel staging, possibly between two lean zones in series or between multiple cans in the engine or sector parts of an annular system, is needed. This staging of fuel would allow satisfactory engine operation, and provision of a low NOx level over a wide range including full-power. Alternately some form of variable geometry could be developed, so as to allow maintenace of the required primary zone equivalence ratios over the engine operating range. This latter approach is complicated and requires moving parts in high temperature environments to operate in a satisfactory manner. The environment as presently envisioned would be severe enough to seriously impact most conventional lubricating systems. Solid lubricating approaches would have to be developed for both slides and pivot points.

The rich-lean approach as presently developed suffers from mechanical integrity problems and will require considerable development of the cooling system particularly of the primary zone and transition pieces. In operation the rich-lean system could provide a low NOx point at maximum power, for example, and still be capable, without fuel staging or variable geometry of operating over a fuel/air ratio turndown range of 5:1. During operation at conditions other than those close to maximum power (about 70% power) the NOx emissions would be high. These would drop to low values again as ambient light-off conditions are approached.

It is believed that the present STI rich-lean combustor design has several significant advantages over many other approaches. It not only provides low NOx at high power conditions but also zero smoke and low CO and UHC (high efficiency) over the entire engine operating range. These secondary features were obtained mainly through the adoption of the regenerative cooling system, and through the effective high residence time secondary zone.

The regenerative cooling system provides high temperature primary air (all of the primary air is used for cooling the primary zone before it is used in combustion) that enables rapid vaporization and mixing of the injected fuel and air. This heating of the primary air also provides high combustion efficiencies at low inlet air temperature conditions, including ambient light-off.

High primary zone wall temperatures at the rear of the primary zone and in the transition zone are existing problems, possible solutions to which have been disussed above. Future activities should address these problems, to both determine the causes and solutions. If the overheating problem is caused by ingress of secondary air into the primary zone (producing local stoichiometric temperatures) then altering the method of secondary air injection at the transition piece could alleviate this Typical solutions could involve angling the transition piece secondary jets toward the secondary zone. This would minimize secondary air moving upstream and causing burning at the junction of the primary and transition pieces. In addition, the point at which the fuel/air mixture transits the stoichiometric value would be moved further into the secondary zone. Alternately if the overheating were simply due to a lack of cooling air, more secondary air could be diverted for transition throat injection.

For the rich-lean system developed by STI the future development requirements devolve to an in-depth study of the transition area There will be an interplay between NOx emissions and the cooling needs of the rear of the primary zone and the transition piece, involving the quantity and the methods of the transition secondary air injection. The rich-lean combustor as previously envisioned would look in the future something like the arrangement shown in Figure 50.

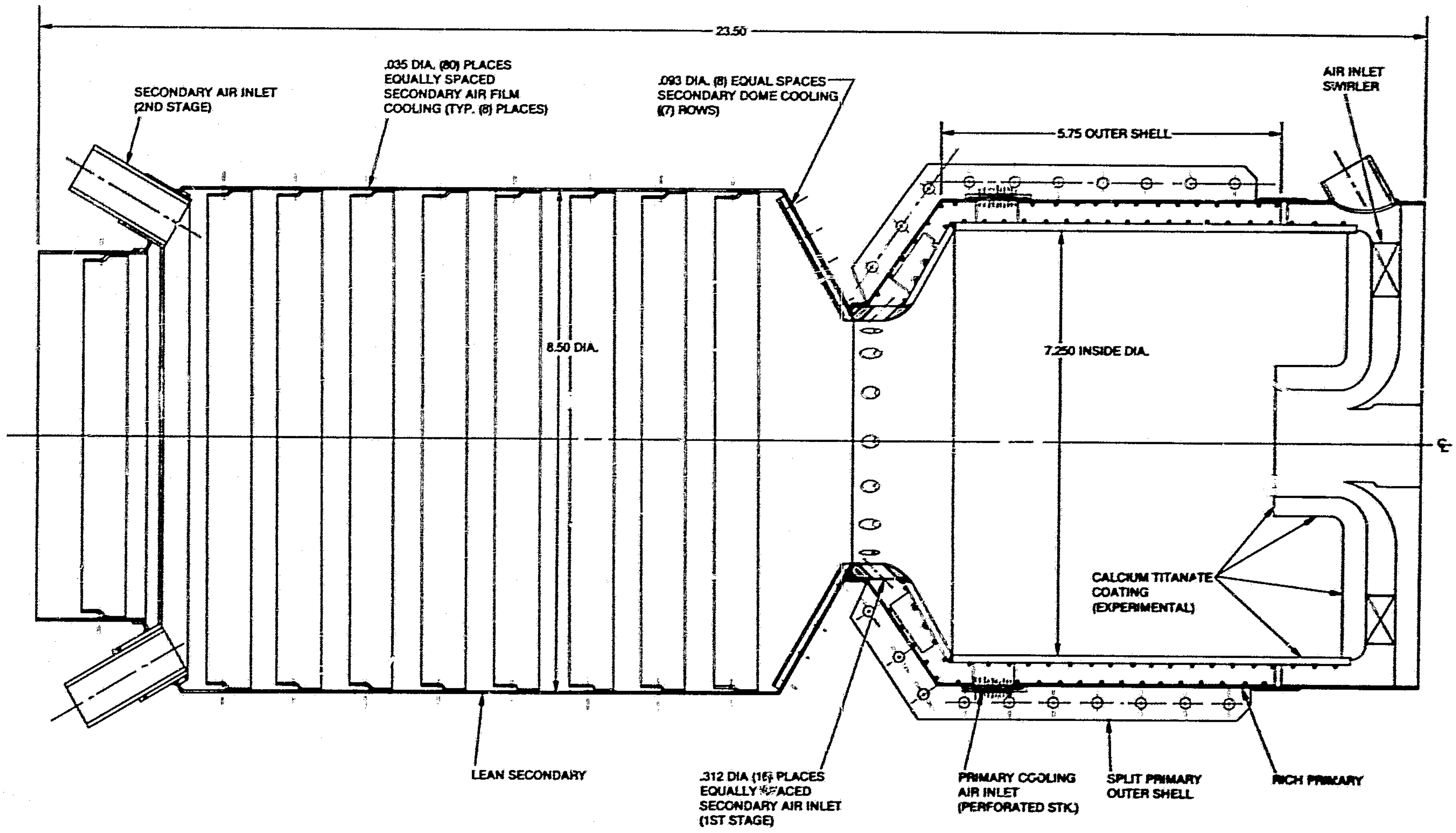

Figure 50. Final Rich-Lean Combustor Design

7
CONCLUSIONS AND RECOMMENDATIONS

Solar Turbines Incorporated has developed a novel rich-lean combustor system that can burn a wide variety of fuels including those with a high nitrogen content with low NOx emissions. In addition, this combustion system provides near zero smoke levels and extremely low CO and UHC levels at all operating conditions; this latter phenomenon being created by the unique application of a convective-regenerative cooling system for the rich primary zone. All the primary air used in combustion is first utilized for primary zone cooling and the resulting high temperature air significantly improves combustion efficiency, particularly at low power conditions. This preheat provides, when mated with a good atomization system, a means of rapidly vaporizing and mixing the fuel with the hot air. Such an approach eliminates the need for an external fuel-air premixing duct with its attendant problems of auto-ignition and flashback. Both autoignition and flashback are problem areas that have plagued the development of effective high temperature, high pressure fuel/air premixing systems for many years. The elimination of these problems is considered to be a major step forward.

The near zero smoke levels were obtained by the use of an extremely effective and unique secondary zone. This lean burnout zone has, as designed, a means of providing effective particle residence times much in excess of the plug-flow values. The reverse-flow secondary air jets create a toroidal vortex that has a recirculation ratio much in excess of one, which in turn creates a high level of hold-back. That is, fluid can remain in the primary zone for time periods much greater than the mean values. In addition it has the velocity characteristics of a free vortex. This velocity profile which is inversely proportional to the vortex radius, maintains particles of a particular size in a fixed orbit. This orbit is one in which the centrifugal forces on the soot or smoke particle spinning with the air are balanced by inward directed drag forces. Usually there is a high relative air particle velocity when the particle is so suspended and this aids in the combustion of the particle. As the particle size decreases it spirals toward the center of the vortex, however, and at some critical size it will be able to escape.

Emission levels of NOx below the EPA requirements have been achieved for fuels containing up to one percent (SRC-II) fuel-bound nitrogen. These emissions have been achieved with a combustion staged system, which utilized a rich primary zone and lean secondary zone, with a transition piece between them. Secondary air in this system entered both in the transition piece and at the rear of the secondary zone in the forward facing jets.

Simple lean-lean combustion systems in which a conventional primary zone is made leaner, have little merit in reducing NOx. Lean premixed systems however, with "clean" low nitrogen content fuels such as ERBS can provide ultra-low NOx levels of the order of one-half of the EPA standards. However, the

range of low NOx operation is narrow and some form of fuel staging or variable geometry would be required to allow operation over the full range of engine conditions.

Although the present program has been successful, some problems still remain. Future activity in this area will have to address these problem areas in some detail. Mechanical integrity problems, for example, with the rich-lean system will have to be solved if the approach is ever to be successful, in a general sense. The main development requirement needed is the reduction in wall temperature of the transition piece and the rear of the primary zone proper. This can probably be achieved by a combination of extra cooling air (increased transition secondary air), and by moving the "stoichiometric transition" into the secondary zone. Moving the stoichiometric transition can possibly be accomplished by angling the transition secondary air jets downstream toward the secondary zone. Improved transition cooling techniques could also be applied, such as separate convective or even transpiration cooling if the resulting reaction zone could be displaced from the wall.

Other problems with the particular rich-lean combustor design used by STI were mainly associated with changes in geometry of the convective cooling duct. These latter changes were caused by thermal distortion of the relatively thin sheet metal walls. By incorporating heavy inner walls and thin outer walls, which would have compensation for both the linear and radial expansion of the inner wall, greater control over the cooling duct geometry can be exercised.

In addition, although the rich-lean system can operate over the entire engine operating range it generally will only provide low NOx at conditions near maximum or peak power. If low NOx is desired at all operating conditions, then variable geometry would be required. To accomplish this will require considerable developmental effort. No fundamental obstacles, however, need be overcome to effect the implementation of variable geometry. It essentially involves integrating the variable geometry control system into the engine control.

Lean premixed systems will also require further development before they can be considered for engine use. Either fuel staging or variable geometry systems will have to be developed and integrated. For lean combustors fuel staging or a combination of combustion and fuel staging appears to be the most attractive approach. This approach will probably be more effective than variable geometry and is in general more readily implemented.

It is recommended for the second phase of the Advanced Conversion Technology (ACT) program that a new rich-lean combustor be designed, built and tested that is modelled on the presently developed rich-lean system. The proposed combustor would utilize basically the same primary and secondary zone dimensions, and would incorporate a similar swirler and fuel injector (see Fig. 50). The dilution zone, however, would be much shorter. The main differences would involve increased inner wall thickness, and modified throat or transition piece secondary air injection geometry. Several transition designs would be produced and used in modular fashion. Transition pieces, with secondary air injection holes oriented at different angles, would, for example, be provided. In addition, various cooling systems could be provided on

the interchangeable pieces such as separate convective cooling or transpiration cooling. As before, the rear opposed secondary air ports would have sleeved inserts to enable the air flow characteristics to be readily altered. It may be possible to utilize the existing secondary zone modules as part of the next phase combustor design. The main purpose of the new design and test program would be to develop a new transition piece and primary zone cooling system that minimizes both wall overheating and NOx.

In addition, further work into the effects of the primary air swirler and fuel injector on the primary zone temperature distributions would be undertaken. These temperature distributions seriously impact the wall cooling requirements. It may be possible, for example, to use swirlers with different expansion angles and move high temperature zones away from critical rear wall areas. Changes in the atomization quality should also be investigated, both with regard to NOx production and to wall overheating. Interaction between the position or placement of the fuel injector with respect to the swirler should be evaluated also, especially with respect to NOx formation and carbon fouling of the injector.

Incorporation of variable geometry should be addresssed, although the manner of implementation may depend on the particular engine to which it is applied. The approach generally favored is to separate the valving or variable geometry from the combustor, typically locating it in ducts that form part of the diffuser leading form the compressor. Diffusers in such an arrangment could be annular, although pipe-type diffuser approaches could also be utilized.

For lean premixed systems the recommendation would include developing a serial two stage premixed lean-lean combustor with effective fuel staging between the two combustion zones. Such fuel staging would involve developing valving integral with the fuel injectors to minimize line fill times and consequent combustor flameout. Such an approach should be capable of operating over a 3.5:1 turndown ratio while providing ultra-low NOx with clean fuels at the maximum power point. Control of the fuel staging would be important and new sensors and control systems may have to be developed. For lean premixed systems the NOx is predominately a function of flame temperature and a device that can measure this directly would be very useful. Generally in lean premixed systems the reaction temperatures are maintained circa 1427°C to 1528°C (2600°F to 2800°F). At these conditions it may be possible to utilize noble metal alloy thermocouples mounted in the dome for control purposes.

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

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

2. White, D. J., Roberts, P. B., and Compton, W. A., "Low NOx Emission Combustor for Automobile Gas Turbine Engines", CDR Final Report, Contract No. 68-04-0016 (EPA), Sept. 1973.

3. Heap, M. P., "The Influence of Fuel Characteristics on Nitrogen Oxide Formation - Bench Scale Studies", Proceedings of the Third Stationary Source Combustion Symposium, EPA. 600/7-79-050b (1979) February.

4. Mosler, S. A. and Pierce, R. M., "Advanced Combustion Systems for Stationary Gas Tubine Engines", Proceedings of the Second Symposium on Stationary Source Combustion Symposium, EPA (1977).

5. White, D. J., "Low NOx Heavy Fuel Combustor Concept Program", NASA Lewis Research Center Contract DEN3-145, Solar Turbines Inc. Report No. SR79-R-4671-02 (May 1979).

APPENDIX A

LOW NOx HEAVY FUEL COMBUSTION CONCEPT PROGRAM

REDUCED DATA

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Data Pt.	Fuel	Inlet Pressure atm.	Inlet Temperature K	Fuel-Air Raio	CO_2 %	O_2 %	NOx ppmv @ 15% O_2	CO ppmv @ 15% O_2
684	E	11.9	589	0.01026	2.35	18.700	215	26.00
690	E	11.9	589	0.01102	2.93	18.425	209	20.50
685	E	11.9	589	0.01225	3.24	17.625	282	16.70
686	E	11.9	589	0.01297	3.59	17.375	225	11.00
687	E	11.9	589	0.01467	3.95	17.075	185	9.76
688	E	11.9	589	0.01480	4.28	16.750	173	8.29
689	E	11.9	589	0.01568	4.65	16.550	155	6.51
675	E	11.9	633	0.00950	2.25	18.250	264	29.20
676	E	11.9	633	0.01059	2.56	17.800	354	19.40
677	E	11.9	633	0.01194	2.71	17.700	335	14.60
678	E	11.9	633	0.01293	2.92	17.375	309	12.50
679	E	11.9	633	0.01385	3.16	17.050	247	9.45
680	E	11.9	633	0.01496	3.49	16.725	179	7.65
681	E	11.9	633	0.01591	3.74	16.375	135	6.41
682	E	11.9	633	0.01698	4.00	16.000	115	6.00
410	E	11.9	633	0.01006	1.42	19.250	122	14.28
411	E	11.9	633	0.01179	1.63	18.925	175	10.44
412	E	11.9	633	0.01373	2.01	18.250	208	2.98
413	E	11.9	633	0.01557	2.42	17.550	190	1.31
414	E	11.9	633	0.01778	2.81	16.875	149	1.15
415	E	11.9	633	0.01871	3.04	16.500	143	5.44
409	E	11.9	533	0.01059	1.38	19.325	90	13.57
408	E	11.9	533	0.01229	1.80	18.750	126	16.67
401	E	11.9	533	0.01279	1.57	19.250	80	8.01
402	E	11.9	533	0.01422	1.80	18.825	104	5.32
403	E	11.9	533	0.01612	2.27	17.875	101	6.33
404	E	11.9	533	0.01816	3.18	16.450	95	3.59
407	E	11.9	533	0.02048	3.59	15.625	90	1.19
406	E	11.9	533	0.02313	4.11	14.800	107	3.94
405	E	11.9	533	0.02550	4.53	14.075	131	3.96
340	E	7.0	416	0.00667	0.71	20.000	31	314.83
339	E	7.0	416	0.00793	0.76	20.000	31	86.14
338	E	7.0	416	0.00966	0.81	19.975	29	36.14
337	E	7.0	416	0.01124	0.91	19.875	30	33.22
336	E	7.0	416	0.01312	0.98	19.750	27	26.52
335	E	7.0	416	0.01484	1.83	18.875	44	19.14
330	E	7.0	416	0.01668	2.30	17.950	42	12.84
331	E	7.0	416	0.01867	3.73	16.075	50	10.90
332	E	7.0	416	0.02088	3.89	15.625	44	6.81
333	E	7.0	416	0.02320	3.71	15.700	44	5.24
334	E	7.0	416	0.02562	5.02	14.125	63	3.94
329	E	3.0	416	0.01093	2.50	19.375	38	65.69
328	E	3.0	416	0.01291	2.60	19.325	33	34.90
327	E	3.0	416	0.01452	3.40	18.825	34	24.22
326	E	3.0	416	0.01671	4.60	18.000	35	13.42
325	E	3.0	416	0.01870	6.20	17.000	38	8.38
324	E	3.0	416	0.02164	7.70	16.000	38	6.56
323	E	3.0	416	0.02479	10.00	14.550	47	5.71

NOTE: Fuel Type E = ERBS

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Data Pt.	Fuel	Inlet Pressure atm.	Inlet Temperature K	Fuel-Air Ratio	CO_2 %	O_2 %	NOx ppmv @ 15% O_2	CO ppmv @ 15% O_2
505	S	11.9	589	0.01118	2.85	17.825	399	34.50
536	S	11.9	589	0.01156	2.64	17.875	351	32.60
506	S	11.9	589	0.01334	3.39	17.250	378	17.80
537	S	11.9	589	0.01379	3.22	17.255	345	18.20
507	S	11.9	589	0.01445	3.57	17.000	366	12.70
538	S	11.9	589	0.01452	3.39	17.125	327	13.50
539	S	11.9	589	0.01524	3.59	16.925	301	11.10
508	S	11.9	589	0.01571	3.90	16.600	313	8.25
540	S	11.9	589	0.01677	3.87	16.550	264	7.46
509	S	11.9	589	0.01684	4.16	16.275	268	7.17
541	S	11.9	589	0.01684	4.14	16.250	245	6.30
510	S	11.9	589	0.01860	4.54	15.750	186	5.10
542	S	11.9	589	0.01964	4.51	15.750	170	4.82
511	S	11.9	589	0.01990	4.71	15.500	142	4.11
512	S	11.9	589	0.02073	4.73	15.325	141	3.01
543	S	11.9	589	0.02111	4.84	15.375	159	3.57
513	S	11.9	589	0.02197	4.81	15.055	147	2.54
544	S	11.9	589	0.02253	5.14	15.000	157	2.76
514	S	11.9	589	0.02400	5.03	14.750	166	1.79
545	S	11.9	589	0.02440	5.59	14.500	181	2.29
515	S	11.9	589	0.02515	5.18	14.450	182	1.75
546	S	11.9	589	0.02663	6.04	13.950	209	2.81
547	S	11.9	633	0.00992	2.53	18.250	422	41.50
554	S	11.9	633	0.01091	2.53	18.000	442	29.80
548	S	11.9	633	0.01156	2.86	17.850	390	19.50
549	S	11.9	633	0.01280	3.10	17.500	389	13.90
550	S	11.9	633	0.01397	3.35	17.200	387	10.20
551	S	11.9	633	0.01506	3.63	16.925	416	8.61
552	S	11.9	633	0.01616	3.69	16.625	377	7.48
553	S	11.9	633	0.01643	3.64	16.575	368	7.36
674	S	8.7	533	0.01079	2.74	18.000	414	48.17
664	S	8.7	533	0.01181	2.88	17.750	410	34.83
673	S	8.7	533	0.01280	3.10	17.525	398	9.63
665	S	8.7	533	0.01399	3.31	17.250	287	15.76
666	S	8.7	533	0.01645	3.79	16.625	153	8.66
667	S	8.7	533	0.01906	4.27	16.000	120	6.10
671	S	8.7	533	0.02013	4.48	15.750	125	5.35
668	S	8.7	533	0.02181	4.77	15.375	138	3.52
670	S	8.7	533	0.02313	5.02	15.050	168	3.25
669	S	8.7	533	0.02395	5.63	14.750	182	2.60
662	S	11.9	533	0.00411	1.02	20.100	218	2591.20
661	S	11.9	533	0.00456	1.21	19.925	237	905.35
660	S	11.9	533	0.00519	1.40	19.700	265	184.24
659	S	11.9	533	0.00577	1.57	19.500	287	50.86
657	S	11.9	533	0.00647	1.76	19.300	302	36.95
658	S	11.9	533	0.00667	1.78	19.250	309	39.11
656	S	11.9	533	0.00730	1.88	19.125	315	35.71
655	S	11.9	533	0.00823	2.03	18.925	332	39.58
654	S	11.9	533	0.00892	2.22	18.625	343	43.76
653	S	11.9	533	0.00995	2.43	18.375	357	41.39
652	S	11.9	533	0.01176	2.81	18.000	373	31.30
663	S	11.9	533	0.01301	3.02	17.500	394	20.94
651	S	11.9	533	0.01884	4.23	16.150	121	6.86
650	S	11.9	533	0.02122	4.63	15.500	127	5.07
649	S	11.9	533	0.02358	5.12	14.950	144	4.92

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Data Pt.	Fuel	Inlet Pressure atm.	Inlet Temperature K	Fuel-Air Ratio	CO_2 %	O_2 %	NOx ppmv @ 15% O_2	CO ppmv @ 15% O_2
557	S	10.3	605	0.01097	2.69	18.00	400	31.60
560	S	10.3	605	0.01166	2.84	17.825	460	27.87
561	S	10.3	605	0.01304	3.10	17.450	440	20.74
562	S	10.3	605	0.01421	3.19	17.125	411	12.93
558	S	10.3	605	0.01528	3.59	16.800	410	8.77
563	S	10.3	605	0.01641	3.56	16.500	381	8.15
559	S	10.3	605	0.01793	4.11	16.175	273	4.21
478	S	11.9	533	0.01233	3.00	17.500	358	29.83
489	S	11.9	533	0.01267	3.29	17.875	335	29.03
466	S	11.9	533	0.01272	2.36	18.200	181	29.42
493	S	11.9	533	0.01281	3.49	17.825	337	28.72
479	S	11.9	533	0.01420	3.38	17.050	333	15.22
490	S	11.9	533	0.01457	3.69	17.300	324	17.79
467	S	11.9	533	0.01524	2.33	17.850	177	18.42
480	S	11.9	533	0.01528	3.59	16.750	279	13.43
491	S	11.9	533	0.01576	4.00	16.950	295	7.84
481	S	11.9	533	0.01643	3.84	16.500	213	8.58
494	S	11.9	533	0.01664	4.08	16.500	237	7.52
492	S	11.9	533	0.01679	4.13	16.700	238	9.83
488	S	11.9	533	0.01755	3.19	17.250	155	12.66
482	S	11.9	533	0.01765	4.09	16.200	159	6.11
495	S	11.9	533	0.01778	4.37	16.275	181	6.06
496	S	11.9	533	0.01894	4.51	16.125	143	6.37
483	S	11.9	533	0.01916	4.29	15.825	132	5.62
469	S	11.9	533	0.01965	3.47	17.450	130	5.81
497	S	11.9	533	0.02029	4.74	15.825	129	4.35
484	S	11.9	533	0.02059	4.59	15.825	121	2.13
498	S	11.9	533	0.02162	5.24	15.450	122	4.41
470	S	11.9	533	0.02206	3.94	16.250	133	5.46
485	S	11.9	533	0.02208	4.83	15.250	122	4.87
471	S	11.9	533	0.02299	4.17	16.000	137	4.95
499	S	11.9	533	0.02330	5.34	15.075	131	2.74
486	S	11.9	533	0.02362	5.12	14.875	136	2.96
500	S	11.9	533	0.02506	5.59	14.750	148	1.76
487	S	11.9	533	0.02557	5.39	14.475	161	2.94
501	S	11.9	533	0.02601	5.86	14.425	164	1.69
488	S	11.9	533	0.02669	5.59	14.200	181	2.61
502	S	11.9	533	0.02757	6.14	14.075	184	2.06
446	S	10.3	533	0.01480	2.69	17.875	246	17.52
447	S	10.3	533	0.01665	2.81	17.700	221	14.25
448	S	10.3	533	0.01869	3.10	17.500	170	8.07
449	S	10.3	533	0.02109	3.51	16.750	158	5.15
450	S	10.3	533	0.02352	4.02	16.300	162	4.52
367	S	7.0	416	0.01501	2.19	18.825	163	33.11
361	S	7.0	416	0.01674	1.64	19.500	103	12.76
366	S	7.0	416	0.01687	2.71	18.125	159	21.74
362	S	7.0	416	0.01922	3.00	18.000	111	9.86
363	S	7.0	416	0.02201	4.28	16.250	127	6.45
364	S	7.0	416	0.02443	4.51	15.900	140	6.28
365	S	7.0	416	0.02745	5.07	15.050	165	3.12

NOTE: Fuel Type S = SRC-II

Data Pt.	Fuel	Inlet Pressure atm.	Inlet Temperature K	Fuel-Air Ratio	CO_2 %	O_2 %	Nox ppmv @ 15% O_2	CO ppmv @ 15% O_2
394	50E/50S	11.9	533	0.01477	1.89	18.475	157	14.92
395	50E/50S	11.9	533	0.01722	2.35	17.675	145	7.52
396	50E/50S	11.9	533	0.01992	2.94	16.900	149	3.24
397	50E/50S	11.9	533	0.02162	3.28	16.275	143	10.94
399	50E/50S	11.9	533	0.02996	4.53	14.000	152	2.64
400	50E/50S	11.9	533	0.03139	4.75	13.850	168	2.52
392	50E/50S	10.3	533	0.01295	1.90	18.875	142	40.09
387	50E/50S	10.3	533	0.01502	2.64	18.325	140	37.39
391	50E/50S	10.3	533	0.01533	2.34	18.200	143	33.82
388	50E/50S	10.3	533	0.01767	2.52	17.950	115	24.40
389	50E/50S	10.3	533	0.02023	3.50	17.325	109	19.15
390	50E/50S	10.3	533	0.02141	3.23	17.250	100	14.06
377	50E/50S	5.7	583	0.01132	1.42	19.750	129	44.00
378	50E/50S	5.7	583	0.01344	1.87	19.125	149	27.35
379	50E/50S	5.7	583	0.01581	2.45	18.375	155	17.75
380	50E/50S	5.7	583	0.01862	3.00	17.625	150	12.73
381	50E/50S	5.7	583	0.02110	3.59	16.875	130	4.08
382	50E/50S	5.7	583	0.02397	4.25	16.000	129	3.58
383	50E/50S	5.7	583	0.02730	5.07	14.925	129	2.51
384	50E/50S	5.7	583	0.03028	5.64	14.250	158	1.98
376	50E/50S	7.0	527	0.01066	1.16	20.000	101	34.54
375	50E/50S	7.0	527	0.01275	1.61	19.500	121	31.24
369	50E/50S	7.0	527	0.01276	1.87	18.950	131	28.82
370	50E/50S	7.0	527	0.01545	2.16	18.500	129	25.72
371	50E/50S	7.0	527	0.01743	2.71	17.875	130	16.08
372	50E/50S	7.0	527	0.02002	3.06	17.500	121	10.05
373	50E/50S	7.0	527	0.02249	3.49	17.000	118	6.31
374	50E/50S	7.0	527	0.02510	4.32	16.000	106	5.64
347	50E/50S	3.0	416	0.01227	1.99	18.750	108	634.89
354	50E/50S	3.0	416	0.01377	2.16	18.500	149	96.88
353	50E/50S	3.0	416	0.01588	2.33	18.500	124	50.31
352	50E/50S	3.0	416	0.01773	3.29	17.250	143	36.48
348	50E/50S	3.0	416	0.01981	3.55	16.800	136	23.70
351	50E/50S	3.0	416	0.02281	4.28	15.875	126	9.33
350	50E/50S	3.0	416	0.02493	4.67	15.300	116	6.15
349	50E/50S	3.0	416	0.02801	5.14	14.750	138	3.06
355	50E/50S	7.0	416	0.01609	2.90	17.675	87	10.33
360	50E/50S	7.0	416	0.01833	2.59	18.125	106	23.63
356	50E/50S	7.0	416	0.02146	3.09	17.500	80	7.12
359	50E/50S	7.0	416	0.02348	3.48	17.000	79	6.54
357	50E/50S	7.0	416	0.02655	4.38	15.875	92	3.23
358	50E/50S	7.0	416	0.02863	4.77	15.250	99	1.20

NOTE: Fuel Type 50E/50S = 50% ERBS and 50% SRC-II

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Data Pt.	Fuel	Inlet Pressure atm.	Inlet Temperature K	Fuel-Air Ratio	CO_2 %	O_2 %	Nox ppmv @ 15% O_2	CO ppmv @ 15% O_2
534	50E/50S	11.9	589	0.01295	2.23	17.925	296	25.00
533	50E/50S	11.9	589	0.01521	2.58	17.425	295	9.94
532	50E/50S	11.9	589	0.01758	3.01	16.750	267	4.29
531	50E/50S	11.9	589	0.01873	3.24	16.475	233	2.30
530	50E/50S	11.9	589	0.01984	3.49	16.125	198	2.17
529	50E/50S	11.9	589	0.02129	3.77	15.750	152	1.36
528	50E/50S	11.9	589	0.02275	4.03	15.375	134	1.27
527	50E/50S	11.9	589	0.02401	4.27	15.075	132	1.16
526	50E/50S	11.9	589	0.02504	4.50	14.800	139	1.11
517	50E/50S	11.9	533	0.01473	3.13	17.575	248	17.01
518	50E/50S	11.9	533	0.01698	3.57	17.000	226	7.88
519	50E/50S	11.9	533	0.01875	3.92	16.425	207	5.06
520	50E/50S	11.9	533	0.02099	4.32	15.825	138	2.97
525	50E/50S	11.9	533	0.02262	4.21	15.500	107	1.43
521	50E/50S	11.9	533	0.02342	4.73	15.225	119	1.47
524	50E/50S	11.9	533	0.02511	4.90	14.750	120	1.28
522	50E/50S	11.9	533	0.02628	5.26	14.500	135	1.22
523	50E/50S	11.9	533	0.02765	5.72	14.000	151	1.16
453	50E/50S	11.9	533	0.01459	2.72	18.550	131	22.22
454	50E/50S	11.9	533	0.01654	2.27	18.575	114	12.39
455	50E/50S	11.9	533	0.01837	2.24	18.250	94	7.04
456	50E/50S	11.9	533	0.02073	2.77	18.250	100	5.19
457	50E/50S	11.9	533	0.02076	2.11	18.450	83	3.89
464	50E/50S	11.9	533	0.02079	2.30	18.050	92	3.88
463	50E/50S	11.9	533	0.02207	3.49	16.250	128	3.17
458	50E/50S	11.9	533	0.02310	2.67	17.500	87	3.02
462	50E/50S	11.9	533	0.02416	3.48	16.250	118	3.33
459	50E/50S	11.9	533	0.02542	3.48	16.200	119	2.74
461	50E/50S	11.9	533	0.02643	3.95	15.625	137	3.04
460	50E/50S	11.9	533	0.02790	4.36	14.925	157	4.12
436	50E/50S	10.3	533	0.01401	2.45	18.250	172	28.55
426	50E/50S	10.3	533	0.01418	2.59	18.375	160	18.29
435	50E/50S	10.3	533	0.01523	2.77	17.875	163	20.56
427	50E/50S	10.3	533	0.01628	2.71	17.925	160	11.60
434	50E/50S	10.3	533	0.01636	3.09	17.375	182	13.85
433	50E/50S	10.3	533	0.01789	3.48	16.875	184	9.94
430	50E/50S	10.3	533	0.01860	3.55	16.750	184	8.11
431	50E/50S	10.3	533	0.01968	3.74	16.500	184	7.15
429	50E/50S	10.3	533	0.02044	3.55	16.750	165	6.32
432	50E/50S	10.3	533	0.02135	3.65	16.625	164	5.41
428	50E/50S	10.3	533	0.02282	4.11	16.125	160	5.65
437	50E/50S	10.3	533	0.02554	4.61	15.375	158	4.51
438	50E/50S	8.7	533	0.01451	2.67	17.875	192	25.13
439	50E/50S	8.7	533	0.01650	2.99	17.425	173	15.04
440	50E/50S	8.7	533	0.01884	3.38	16.950	167	8.01
441	50E/50S	8.7	533	0.02114	3.75	16.250	153	5.09
442	50E/50S	8.7	533	0.02335	3.79	16.200	139	4.60
444	50E/50S	8.7	533	0.02474	3.89	16.050	129	4.34
443	50E/50S	8.7	533	0.02584	4.17	15.750	140	3.11

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Data Pt.	Fuel	Inlet Pressure atm.	Inlet Temperature K	Fuel-Air Ratio	CO_2 %	O_2 %	NOx ppmv @ 15% O_2	CO ppmv @ 15% O_2
647	R	11.9	533	0.01130	2.53	18.00	281	26.84
648	R	11.9	533	0.01232	2.74	17.800	310	20.37
642	R	10.3	533	0.01264	2.81	17.675	274	20.55
643	R	10.3	533	0.01349	3.00	17.425	280	16.03
644	R	10.3	533	0.01372	3.05	17.175	292	14.34
645	R	10.3	533	0.01489	3.31	17.000	230	11.03
646	R	10.3	533	0.01525	3.39	16.750	179	9.92
641	R	10.3	533	0.01642	3.65	17.925	161	24.98
640	R	8.7	533	0.01215	2.70	17.825	168	42.76
635	R	8.7	533	0.01305	2.90	17.325	189	28.19
639	R	8.7	533	0.01417	3.15	17.250	185	22.88
636	R	8.7	533	0.01525	3.39	16.725	186	17.00
638	R	8.7	533	0.01592	3.54	16.700	186	13.56
637	R	8.7	533	0.01682	3.74	16.300	178	12.83
633	R	8.7	416	0.01163	2.58	18.500	114	143.34
632	R	8.7	416	0.01352	3.00	17.825	116	127.11
628	R	8.7	416	0.01482	3.29	16.950	146	55.44
631	R	8.7	416	0.01572	3.49	17.475	122	76.93
629	R	8.7	416	0.01669	3.71	16.500	120	27.28
630	R	8.7	416	0.01883	4.19	15.625	106	21.74
627	R	3.0	416	0.01093	2.41	18.375	129	441.77
626	R	3.0	416	0.01258	2.78	17.750	122	302.68
625	R	3.0	416	0.01423	3.15	17.250	116	208.12
622	R	3.0	416	0.01670	3.71	16.250	123	62.03
623	R	3.0	416	0.01826	4.06	15.800	120	41.31
624	R	3.0	416	0.02254	5.02	14.700	124	31.47

NOTE: Fuel Type R = Residual

Data Pt.	Fuel	Inlet Pressure atm.	Inlet Temperature K	Fuel-Air Ratio	CO_2 %	O_2 %	NOx ppmv @ 15% O_2	CO ppmv @ 15% O_2
24	E	3.0	416	0.01989	3.81	15.650	66	2.96
25	E	3.0	416	0.02178	4.35	14.825	100	3.54
26	E	3.0	416	0.02321	4.59	14.500	110	3.32
27	E	3.0	416	0.01868	3.76	15.625	69	3.16
28	E	3.7	416	0.02254	4.51	14.675	130	2.07
29	E	3.7	416	0.02070	4.10	15.225	99	2.26
30	E	3.7	416	0.01824	3.72	15.775	79	2.13
33	E	3.0	416	0.01914	3.95	15.450	77	9.35
34	E	3.5	416	0.01883	3.81	15.900	76	13.08
35	E	3.5	416	0.02087	4.20	15.175	96	8.66
36	E	3.5	416	0.02264	4.67	14.550	126	6.45
37	E	5.1	416	0.02044	4.15	15.300	99	6.90
38	E	5.1	416	0.02282	4.64	14.575	115	6.16
39	E	4.8	416	0.01839	3.75	15.775	68	7.53
40	E	4.8	416	0.01647	3.37	16.350	43	8.47
44	E	3.0	533	0.02013	4.17	15.250	115	7.67
45	E	3.0	533	0.02208	4.59	14.675	151	6.71
46	E	3.0	533	0.02746	3.58	16.000	93	10.04
47	E	4.4	533	0.01969	3.96	15.575	101	6.51
48	E	4.4	533	0.02170	4.41	14.950	137	5.71
49	E	4.4	533	0.01749	3.56	16.100	69	7.92
50	E	5.8	533	0.01923	3.92	15.675	112	5.50
51	E	5.8	533	0.02116	4.35	15.050	148	4.89
52	E	5.8	533	0.01722	3.51	16.250	87	5.45
53	E	5.8	644	0.01629	3.24	16.625	118	3.51
54	E	5.8	644	0.01672	3.35	16.475	123	3.78
55	E	5.8	644	0.02028	3.77	15.850	142	3.11
56	E	5.8	644	0.01474	2.95	16.950	103	4.44
59	E	5.8	656	0.01673	3.33	16.425	122	4.02
60	E	5.8	644	0.01423	2.88	17.050	103	6.43
61	E	6.8	644	0.01431	2.91	16.975	112	5.72
62	E	6.8	644	0.01634	3.30	16.400	134	4.50
63	E	6.8	644	0.01868	3.75	15.775	162	3.81
64	E	7.1	644	0.01646	3.30	16.400	138	4.34
65	E	7.1	644	0.01438	2.90	17.00	119	4.98
66	E	7.1	644	0.01836	3.70	15.850	167	3.33
68	E	7.1	644	0.01681	3.29	16.250	132	7.40
69	E	7.8	644	0.01693	3.24	16.350	138	5.90
70	E	7.8	644	0.01463	2.87	16.950	119	8.24
71	R	7.8	644	0.01882	3.67	15.775	168	4.43
72	R	3.0	416	0.02348	4.71	14.425	108	7.07
73	S	3.0	416	0.01891	5.06	14.500	377	3.41
74	S	3.0	416	0.01913	5.12	14.450	376	12.34
75	S	3.0	530	0.02074	4.23	15.000	119	2.06
76	S	3.0	530	0.01805	3.88	15.750	217	2.48
77	S	3.0	530	0.01611	3.43	16.575	244	3.29
80	E	3.0	644	0.01882	3.40	16.450	102	7.68
82	E	11.9	644	0.01495	3.18	16.700	191	11.07
83	E	11.9	644	0.01423	3.67	16.000	258	10.14
85	S	5.8	644	0.01458	3.64	16.400	425	14.02
86	S	5.8	644	0.01651	4.09	15.825	471	11.74
87	S	5.8	644	0.01246	3.15	17.000	383	17.43
88	S	8.8	644	0.01462	3.56	16.550	317	12.16
89	S	8.8	644	0.01656	3.99	16.025	350	10.59
90	S	11.9	644	0.01619	3.19	16.750	166	8.88
91	S	11.9	644	0.01809	3.61	16.225	197	7.82

Note: Fuel Type E = ERBS; S = SRC-II; R = Residual