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PACKAGE BOILER FLAME MODIFICATIONS FOR REDUCING NITRIC OXIDE EMISSIONS PHASE II OF III

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PACKAGE BOILER FLAME MODIFICATIONS FOR REDUCING NITRIC OXIDE EMISSIONS -- PHASE II OF III

by

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I. SUMMARY, CONCLUSIONS, AND RECOMMENDATIONS

During Phase II of this three-phase program, tests with a 3.7×10^6 Btu/hr multifuel combustion facility were conducted in order to develop NO_x control techniques for oil-fired package boilers. Both single burner retrofits and suggestions for factory redesign are sought to control emissions from commercial (3 to 30×10^6 Btu/hr) and industrial (30 to 400×10^6 Btu/hr) package boilers, which together contribute 16% of the nitric oxide produced by stationary combustion sources(1). Since NO_x emissions are more readily controlled by flame modifications than by processing stack gases, three control methods were explored: (1) flue gas recirculation, (2) staged combustion, and (3) burner modifications. These categories each contained a wide variety of variables as shown below:

EXPERIMENTAL VARIABLES
(All tests over range of excess air and load)

Flue Gas Recirculation

Level of FGR

Location of FGR Addition:
- Primary Air Stream
- Secondary Air Stream
- Total Combustion Air
- Gas Ports
- Quarl Injectors
- Combinations

Combined FGR and SC

Vary Injection Location
Vary Injection Level

Burner Modifications

Primary/Secondary Air Ratio
Swirl Level
Fuel Type (No. 6, No. 2 Oil; Natural Gas)
Oil Temperature
Atomization Air Pressure
Inert Atomizing Fluid

Staged Combustion

Level of Staging

Location of Staged Air Addition:
- Rear Injection Boom
- Sidewall Injectors

SC w/Refractory Liners for Smoke Control

The experimental tests (approximately 1400) conducted on this laboratory combustor incorporating a modified Ray oil burner allow the following major conclusions to be drawn:

1. Simple burner modifications to atomization air pressure or primary/secondary air ratio can result in 20% reductions in NO.
2. Unlike prior data of Heap et al.[2] and Wasser et al.,[3] a factor of 4 change in swirl affected NO less than 10%.
3. NO reductions of 45% without increased soot were available with FGR, provided the flue gas added to that portion of air which mixes with the fuel in the near vicinity of the burner.
4. Staged combustion was limited as an NO_x control technique in this combustor due to a direct tradeoff in smoke. Approximately 25% reductions in NO could only be realized before smoke emissions became excessive. Inadequate mixing in the near vicinity of the burner is suggested as the limiting factor.
5. The flame in this combustor is "naturally staged" by the secondary air with NO formation principally controlled by the aerodynamics of the flowfield.
6. Combined flue gas recirculation and staged combustion resulted in substantial NO reductions (up to 50%) with only moderate increases in smoke.

An expanded discussion of our results with each of these control techniques follows.

Burner Modifications

Of the various burner modifications tested, the emissions were most sensitive to the primary-to-secondary air ratio, with 20% reductions in NO possible by lowering the P/S ratio. Increasing the atomization air pressure was also found to reduce the NO emissions on the order of 20% with No. 6 oil firing. These reductions were attributed to a retarded rate of fuel/air mixing (essentially "staging" the combustion internally) due to the following aerodynamic changes in the burner flowfield:

- Shifting air from primary to secondary presumably places this air flow near the walls of the combustor where it mixes with the fuel at a slower rate.
- Increased atomization pressure collapses the oil spray pattern from a wide cone to a more cylindrical form. This initial maldistribution, for a given mixing intensity, presumably results in delayed mixing of the fuel initially placed at the jet centerline.

The other burner variables such as swirl and oil temperature were found to have a small effect (less than 10%) on the emissions from the combustor.

Flue Gas Recirculation

The effectiveness of flue gas recirculation as an emission control technique was found to be dependent on the level of recirculation, the location at which the flue gas is added to the burner, and also the type of fuel being burned. The results indicated that the largest increment in NO reduction is accomplished with the first 10% recirculated. This diminishing-returns effect has been observed by other investigators[4,5]. Also, it was found that the flue gas must be added such that it intimately mixes with the air in the near vicinity of the burner in order to be effective. For instance, adding the FGR to the secondary air stream or through quarl injectors had no effect on the NO emissions from the combustor, whereas reductions in NO on the order of 30% could be obtained during No. 6 oil firing when the flue gas was added to the total air, the primary air stream, or through the natural gas manifold. Also, NO reductions with FGR were accomplished with little tradeoff in smoke emissions. Flame instabilities or flame blowout were not observed with any of the configurations tested, with levels of FGR up to 40%*. Regarding fuel type, the greatest percent reductions in NO were obtained with natural gas, then No. 2 oil, and No. 6 oil respectively. The differences due to fuel type are attributed to the nitrogen content of the fuels (the fuel-bound nitrogen conversion to NO is not temperature-sensitive and thus not very controllable by FGR).

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* $$\text{FGR}(\%) = \frac{\dot{m}_{recirc}}{\dot{m}_{fuel} + \dot{m}_{air}}$$

Staged Combustion

Staged combustion was not found to be as attractive as flue gas recirculation in this burner/combustor system due to increases in smoke emissions during staging. Nitric oxide reductions of 25% during No. 6 oil firing could not be obtained without the smoke levels exceeding a number 8 Bacharach smoke number, regardless of the staging configuration. In addition, for a given staging configuration, the NO emissions were not very sensitive to the amount of staged air. It is not certain whether the increase in smoke was predominantly due to the overall fuel-rich condition near the burner during staging or due to prolonged lifetime of local rich regions near the burner due to the reduced velocities and less vigorous mixing.

Combined Methods

Combined staged combustion and flue gas recirculation was found to be an attractive method of reducing NO emissions while still maintaining acceptable smoke performance. With 25% of the air staged through the rear boom and 24% FGR, the NO emissions could be reduced by 45% with only a modest increase in smoke (2 Bacharach smoke numbers above baseline).

A question which still remains is the general applicability of the results in this facility. Since the study was conducted with one particular manufacturer's burner, it remains to be demonstrated that the results are widely applicable to other burner/combustor systems.

Recommendations

It is recommended that a fire tube and a water tube boiler operating in the field be modified and tested in order to assess the effectiveness of the control techniques uncovered during Phase II. The fire tube or scotch marine boiler would represent a lower size range, close in characteristics to the laboratory combustor, and provide the easiest transition from the laboratory to the field. The water tube boiler in the larger size range would provide information as to the scale over which the techniques are applicable.

It is also recommended that further experiments be conducted on the laboratory combustor in direct support of the field testing. For instance, if the fuel delivery system of the fire tube field unit is different than the laboratory combustor (e.g., steam or pressure atomized, different nozzle pattern) then it would be prudent to set up a similar fuel delivery system in the laboratory for preliminary and parallel studies. These tests would aid in rationalizing the field data and would also help in planning modifications to the field units. Similar studies may be warranted with the air delivery system, quarl shape, or perhaps refractory located in portions of the combustor.

II. INTRODUCTION

The effectiveness of flame modifications in reducing NO_x emissions from package boilers is presently being investigated in a three-phase program. During Phase I, a versatile oil-fired facility was designed and constructed to serve as an experimental arena for investigating control techniques[(1)] (a sketch of the facility is shown in Figure 1). The present report documents the work of Phase II which utilized the experimental facility in a systematic investigation of burner modifications, flue gas recirculation, and staged combustion as methods of emission control for oil-fired package boilers. During the testing, control techniques were sought that require both minor modifications (applicable to existing boilers) and extensive modifications (primarily applicable to a redesign of new boilers). Having determined promising emission control techniques during the activities of Phase II, we recommend that these techniques be applied to actual field operating boilers during Phase III.

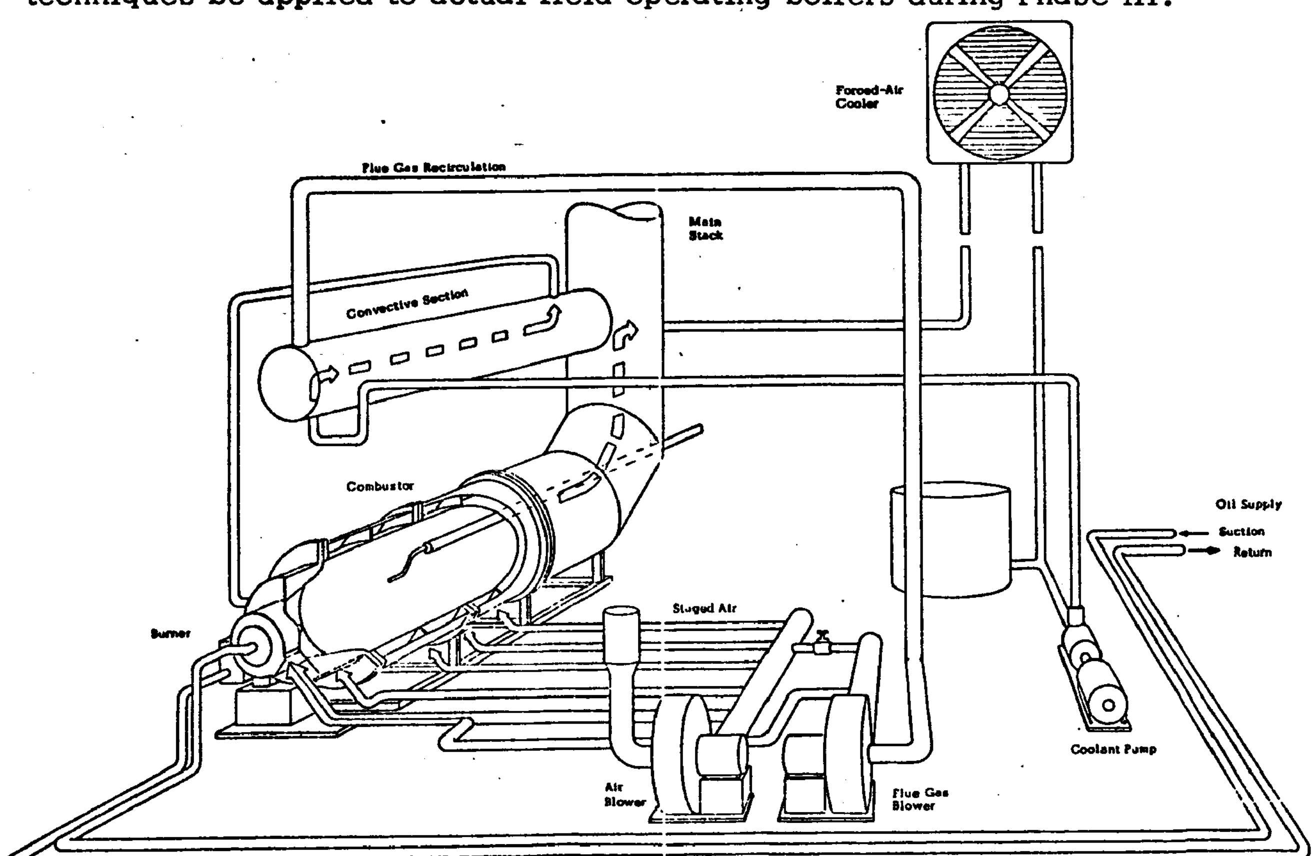

Figure 1

SKETCH OF COMBUSTOR FACILITY

III. EXPERIMENTAL FACILITY

As discussed in detail in References 1 and 6, the experimental facility was designed to provide an authentic package boiler flame, versatile enough for testing, and heavily instrumented. Without moving the baseline design away from that of a typical package boiler, the facility provides the experimenter with as many options as possible both in terms of standard parameters (e.g., fuel type, atomization type, load, excess air), and especially in the modes of external flue gas recirculation and staged combustion. The nominal specifications of the facility are given below:

Load: 3.7×10^6 Btu/hr
Combustion Intensity: 1.7×10^5 Btu/hr-ft^3
Combustor L/D: 3.9
Wall Temperatures: 450°F
Fuel: No. 6 oil, No. 2 oil, natural gas

The furnace is composed of three cylindrical modules, each 30" long; this then allows changes in both combustor volume and L/D simply by removing one of the furnace modules. In addition, an 11" throat module connects the burner to the furnace and provides a portion of the refractory burner cone. Downstream of the throat, the steel walls of the three furnace modules are cooled to 250 to 450°F. Refractory may be added to the furnace as cylindrical inserts.

During burner operation, the coolant distribution system provides a continuous flow of liquid coolant to the flue gas convective section and to the combustor wall cooling jacket. Dowtherm was selected as the cooling fluid to avoid the high pressures associated with 400°F wall temperatures when water is used.

A commercial 90 HP, dual-fired Ray burner was selected as the skeleton for a the research burner. Designed for application in multi-pass scotch marine, water tube or firebox boilers, the burner is set up for low-pressure air atomization of oil and forced-draft operation. The adaptation of this burner to a configuration amenable for research included the following changes:

(i) Physical separation of primary and secondary air supplies for independent metering

(ii) Provision for variable swirl rate

(iii) Provision for variable fuel/air ratio independent of load.

A sketch of the modified burner is shown in Figure 2.

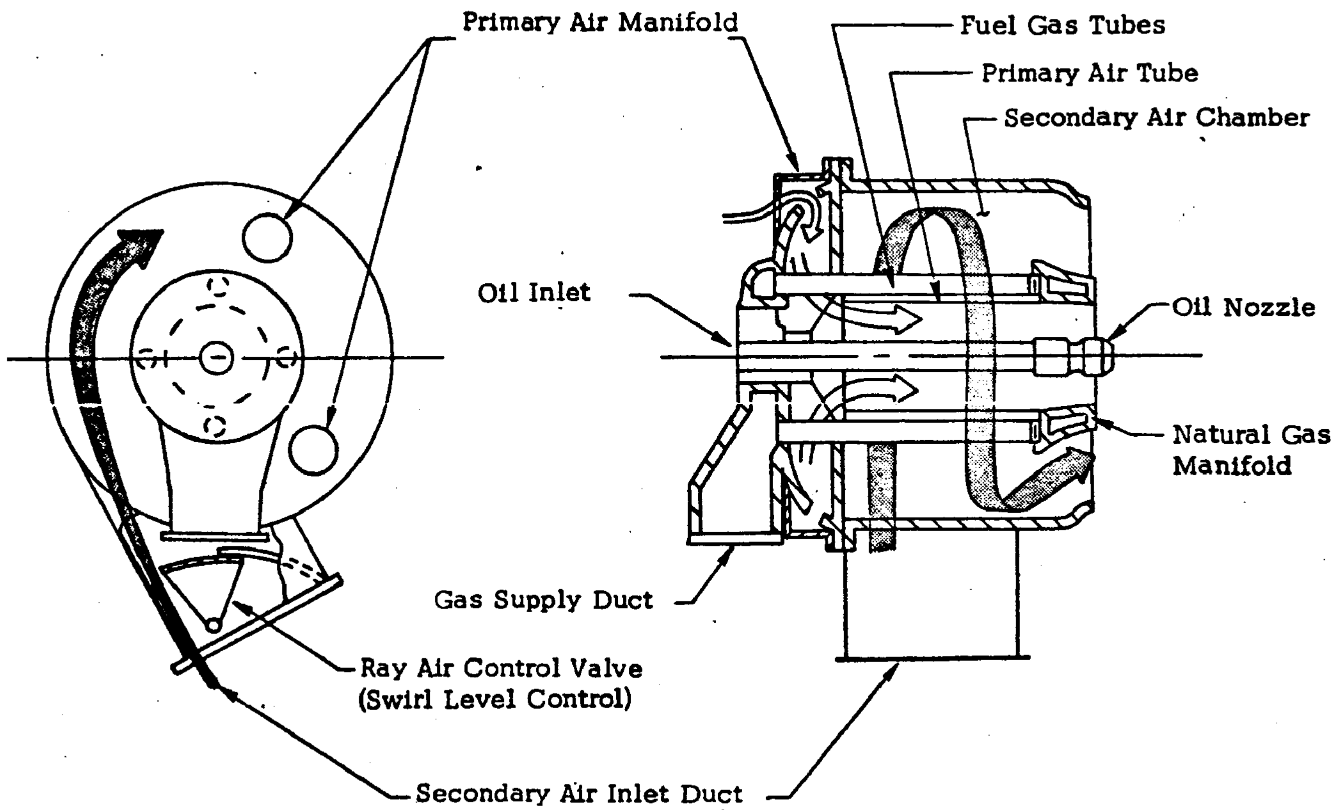

Figure 2

MODIFIED BURNER

The effect of swirl on the emissions from the combustor was investigated in the course of the burner variation studies. The swirl is varied by varying the inlet velocity of the secondary air through the secondary air control valve (Figure 2). Since the secondary air enters the burner tangentially, this velocity determines the angular momentum at the throat which can then be expressed in terms of a swirl parameter for the burner.[1]

In calculating a swirl parameter for the burner, a swirl parameter is calculated independently for the secondary and primary (the swirl parameter for the primary is zero for this burner) and the two numbers are combined linearly using the flow distribution as a weighting factor. (Details of the assignment of a swirl parameter for the burner are given in Reference 6*.)

The overall air and flue gas distribution system is presented in schematic form in Figure 3. Basically, eight supply points can be fed up to 1000 and 500 cfm of air and flue gas, respectively, from centrifugal blowers at elevated pressure (1 psig).

The eight supply points include seven injection points for air, and four for flue gas:

	Injection Point	Provision for Flue Gas	Provision for Air
(i)	Burner primary	X	X
(ii)	Burner fuel gas ports	X	--
(iii)	Burner secondary	X	X
(iv)	Throat injectors	X	X
(v)-(vii)	Sidewall injectors	--	XXX
(viii)	Axial injection from rear	--	X

Staged combustion can be accomplished in two ways in the present combustor:**

Sidewall:
Air addition through downstream sidewall injectors (three locations: 20-3/8", 30-3/8", and 50-3/8" from the oil nozzle, S1, S2, S3 respectively)

Axial Boom:
Air addition through a central boom in direction counter to or at right angles to the main hot gas stream (rear boom position can be continuously varied along the axis of the combustor).

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*This burner swirl parameter is based on an idealization of the inlet angular momentum to the primary and secondary windboxes and should not be interpreted in the classical sense as a swirl number.

**These methods involve explicit downstream air injection. In addition, certain burner modifications probably result in natural "internal" staging of the combustion process which may be simply reduced mixing rates.

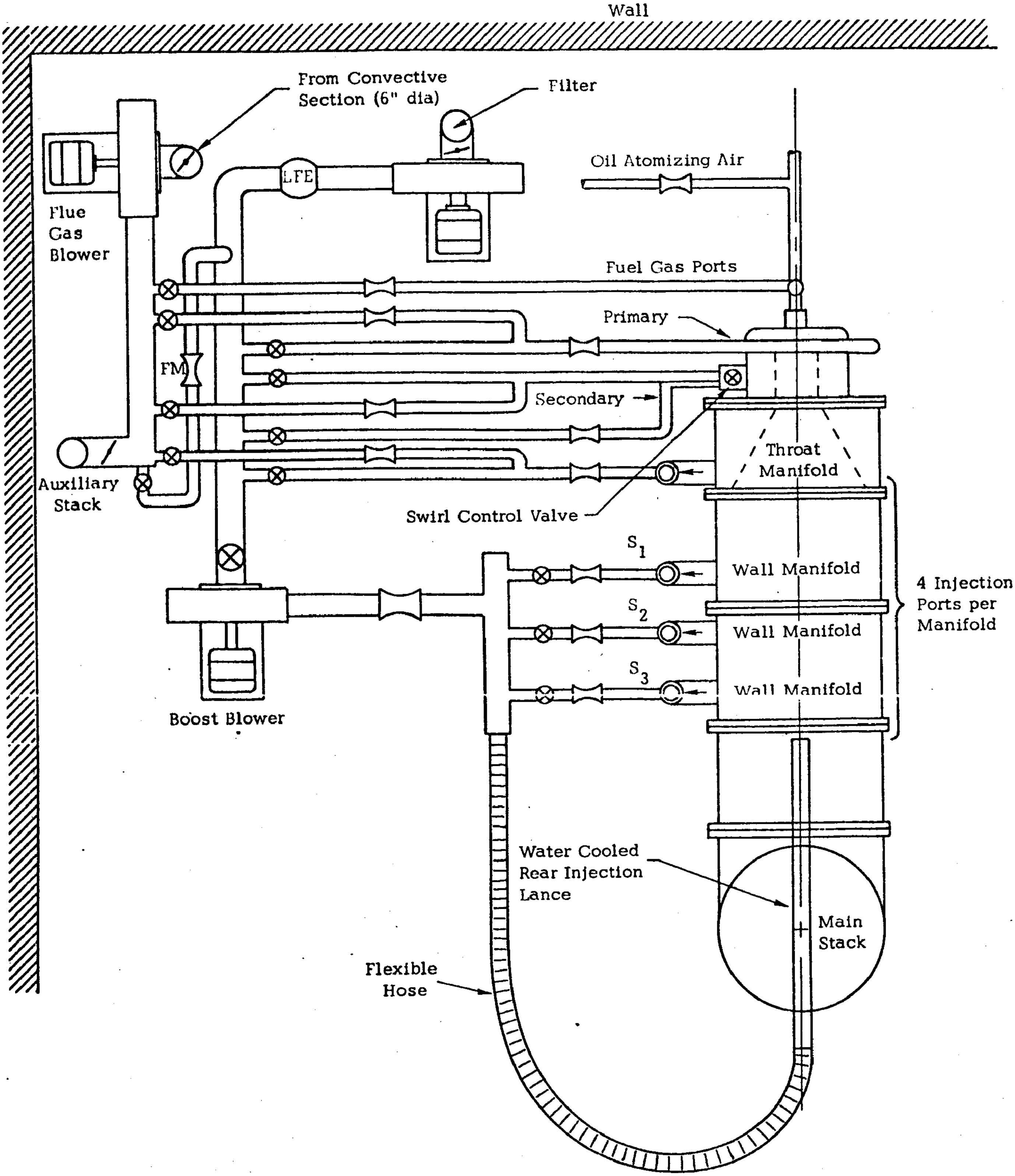

Figure 3

AIR AND FLUE GAS DISTRIBUTION SYSTEM

A deflector is mounted on the end of the boom to disperse the staged air in a radial direction (see Figure 4 below). Visual observations were made of the interaction between the main combustor air and the staged air by adding Boron Trifluoride to the rear boom air. These observations indicated that over the range of flows tested, the staged air was dispersed throughout the cross sectional area of the combustor.

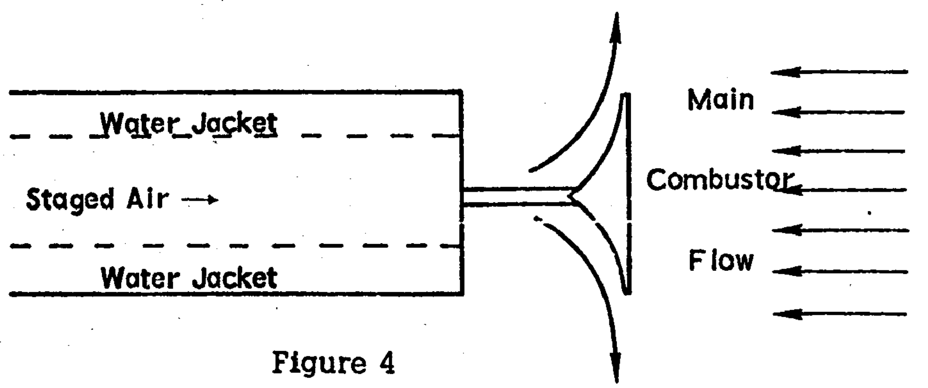

Figure 4

SKETCH OF BOOM TIP

Instrumentation is provided to monitor all air, flue gas, and fuel flow rates and temperatures. Coolant temperatures are also monitored along the length of the combustor to calorimetrically indicate the heat transfer distribution. In addition, the following components of the combustion products are monitored:

NO, NO_x	-	Chemiluminescent analyzer (modified EPA design)
O_2	-	Paramagnetic analyzer (Servomex 0A250)*
CO	-	NDIR (Beckman 715B)
Smoke	-	Bacharach smoke tester

All signals are processed by a specially-designed data acquisition system for visual display in engineering units. In addition, the acquisition system digitizes and records all signals on magnetic tape.

The unit fires number 6 and number 2 fuel oils and in addition, natural gas. Characteristics of the No. 6 and No. 2 oils used during these tests are shown in Table 1:

*Mention of a specific manufacturer does not constitute endorsement by Ultrasystems, Inc. or the program sponsors.

Table 1

FUEL OIL CHARACTERISTICS

	No. 6 Oil	No. 2 Oil
Gravity, °API	16.7	35.0*
Flash Point, PMCC °F	265	170*
Pour Point, °F	80	-15*
Viscosity, SSF at 122°F, sec	97	35*
Heat of combustion, gross, Btu/lb	17,746	19,242*
Water and sediment, %	0.08	0.00*
Ash, %	0.02	--
Sulfur, %	0.42	0.24
Nitrogen, Kjeldahl, %	0.36	0.05
Carbon, %	87.68	86.21
Hydrogen, %	11.61	12.68

*Typical values, Ref. 7

IV. TEST MATRIX AND BASELINE EMISSIONS

A. MATRIX OF FLAME-MODIFICATION TESTS

All combustion modification tests were run on No. 6 fuel oil at two firing rates and two excess air levels, with all other burner parameters held constant. This 4-point operating matrix was useful for the following reason: Since package boilers are fired under widely varying load and excess air, a technique which controls emissions at condition A may produce excessive smoke when applied at condition B. However, since there were so many techniques of applying FGR and SC to the combustor, it was impractical to attempt to test each configuration technique under all boiler operating conditions. It was necessary to select a limited (4-point) set of representative operating conditions, wide enough to establish a reasonable level of confidence that each technique would be applicable over a range.

In addition, a limited number of experiments were performed using the low-nitrogen No. 2 oil. This allowed a rough assessment of the nitric oxide arising from nitrogen fixation and from fuel-bound nitrogen.

The testing was performed in four main blocks according to the test matrix of Table 2: burner modifications, flue gas recirculation (FGR), staged combustion (SC), and combined FGR and SC.

The first tests were run making simple burner changes; variations included:

Swirl level
Primary/secondary air ratio
Oil temperature
Oil atomization air pressure

Table 2

TEST MATRIX

Baseline Conditions:	Load (3.4×10^6 Btu/hr, 2×10^6 Btu/hr) Excess Air (17%, 35%)	Standard 4-point matrix for all tests
	Air Distribution (Primary/Secondary - 50%/50%) Oil Temperature (200°F) Atomization Air Pressure (18 psig) Combustor (L/D = 4) Coolant Inlet Temperature (250°F) Fuel (No. 6 Oil) Swirl (Swirl Parameter = 1.8)[a]	

Focus of Test	Test Conditions
BURNER VARIATIONS	Primary/secondary - 30/70, 40/60, 60/40, 70/30[b] Secondary swirl - low, high (swirl no. .76, 3.8) Oil temperature: 170, 180, 190°F Atomization air pressure: 10 to 36 psig
FLUE GAS RECIRCULATION	3 recycle ratios - 10%, 20%, 30%[c] 5 injection positions - primary, secondary, throat, gas ports, and total
STAGED COMBUSTION	Three Levels of Staging: 17%, 25%, 35% Three Sidewall Injection Stations, each with 4 orientations (upstream, downstream, co-swirl and counterswirl) Rear Boom Injection with variable position
COMBINED STAGING AND RECIRCULATION	External recirculation through the total air stream, combined with 3 types of staged combustion: 2 levels of FGR (10%, 30%) 2 levels of staging (17%, 25%)
INTERSPERSED STANDARDIZATION	Performed at random to detect any systematic drift in combustor behavior and to test No. 2 oil and natural gas.

(a) Secondary air control valve opened 1-3/8" (Ref. 1, also see Fig. 2)

(b) For these tests, the primary/secondary air ratio was also varied (primary = 40%, 50%, 60%) (i.e., 12-point baseline matrix instead of the usual 4-point matrix)

(c) FGR (%) = $100 (\dot{m}_{recirc})/(\dot{m}_{fuel} + \dot{m}_{air})$

In general, these "front-end" modifications call for much less severe hardware changes than the furnace modifications to be described in the following sections, and therefore are logical candidates for quick field modification. The effectiveness of these burner-oriented techniques relies on the well-known sensitivity of emissions to near-burner mixing patterns. They were tried first because of their simplicity. The burner results, in addition to establishing control data for comparison with combustion modification, were repeated periodically throughout the test program to verify standardization of the furnace. Any systematic drift in furnace behavior such as soot accumulation could be readily detected by these repeat calibration tests.

The next combustion modification series utilized flue gas recirculation. Three injection schemes were used, listed in decreasing order of estimated rate of flame dilution and cooling:

1. Conventional annular injection (primary, secondary passages)
2. Injection through fuel gas ports
3. Quarl throat exit injection

The percent of flue gas recirculation was varied for each injection scheme.

Tests of staged combustion followed the FGR testing. Variations in injection schemes included the following five locations of introducing air for delayed combustion:

1. Quarl throat injection

2-4. Downstream wall jets S_1, S_2, S_3 (Figure 3)

3. Downstream axial countershower

These concepts were tested singly and in combinations.

Temperature sensors are located in the coolant stream along the length of the combustor. The incremental temperature rise along the combustor can then be used to monitor the distribution heat transfer from the combustion products along the combustor walls[1]. This then provides a tool for determining whether changes in emissions are due to (1) significant changes in mixing patterns which affect the temperature of the combustion products and thus the heat flux to the walls, or (2) local changes in mixing and combustion patterns which, while significant to NO and smoke formation, do not affect the overall thermal performance of the combustor.

B. BASELINE PERFORMANCE

The combustor was first operated over the baseline conditions outlined in Table 2. Further baseline tests were also conducted at an intermediate load setting of 2.8×10^6 Btu/hr. The emissions of NO, CO, and smoke for these conditions are presented in Table 3 for firing with both No. 6 and No. 2 oil. Throughout the test program the CO emissions were always low and fairly insensitive to the combustor modifications. In seeking low NO configurations, the main tradeoff was in the smoke emissions (the CO levels never increased substantially until after an excessive smoke level was reached). Thus only the smoke and NO emissions results are presented hereafter. Note NO_x reduced to 3% O_2.

Table 3

BASELINE EMISSIONS

Fuel	Excess Air(%)	Load (10^6 Btu/hr)	NO ppm(3%O_2)	Smoke (Bacharach)	CO ppm(3%O_2)	Combustor Exit Gas Temperature (^oF)
No. 6 Oil	17	3.4	273	4	28	1560
	17	2.8	235	7	23	1516
	17	2.0	214	7	27	1359
	35	3.4	288	2	27	1562
	35	2.8	286	3	20	1587
	35	2.0	274	4	24	1397
No. 2 Oil	17	2.5	87	.5	9	1606
	17	3.5	---	-	--	--
	17	2.0	91	0	17	1603
	35	2.5	114	1.5	10	1448
	35	2.0	118	0	10	1510
	35	3.5	---	-	--	--

For comparison with emissions from field-operated boilers, a few data points are plotted in Figure 5 along with a curve of typical commercial boiler emissions obtained during a field study by Battelle.[8] The performance of the experimental facility in terms of emissions of NO and smoke can be seen to be similar to those of actual field boilers.

As seen in Figure 6, with a change in load from 3.4×10^6 to 2.0×10^6 Btu/hr, the heat flux from the combustion products to the coolant only drops by 20%. Further, the heat flux distribution along the combustor changes at the lower loads with a greater percentage of the heat flux taking place near the combustor entrance. This is probably due to the reduced velocity and longer residence times of the gases at the lower load setting.

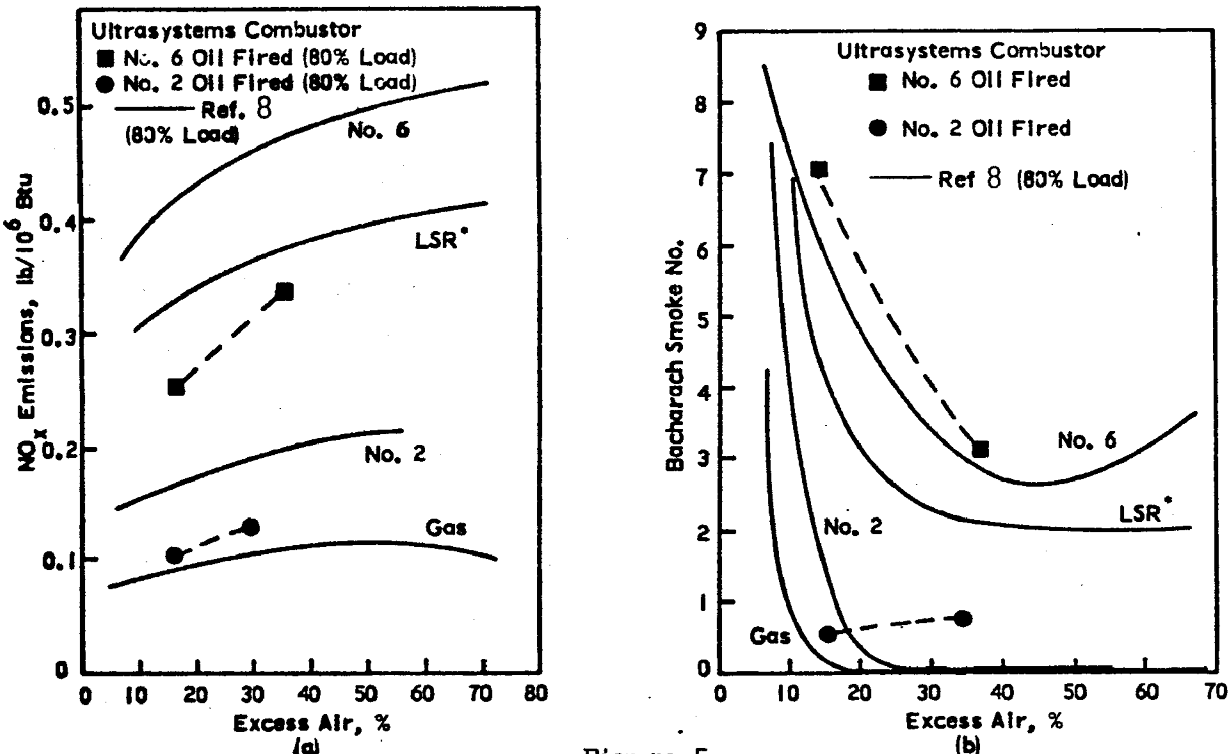

Figure 5

TYPICAL EMISSIONS FROM COMMERCIAL BOILERS

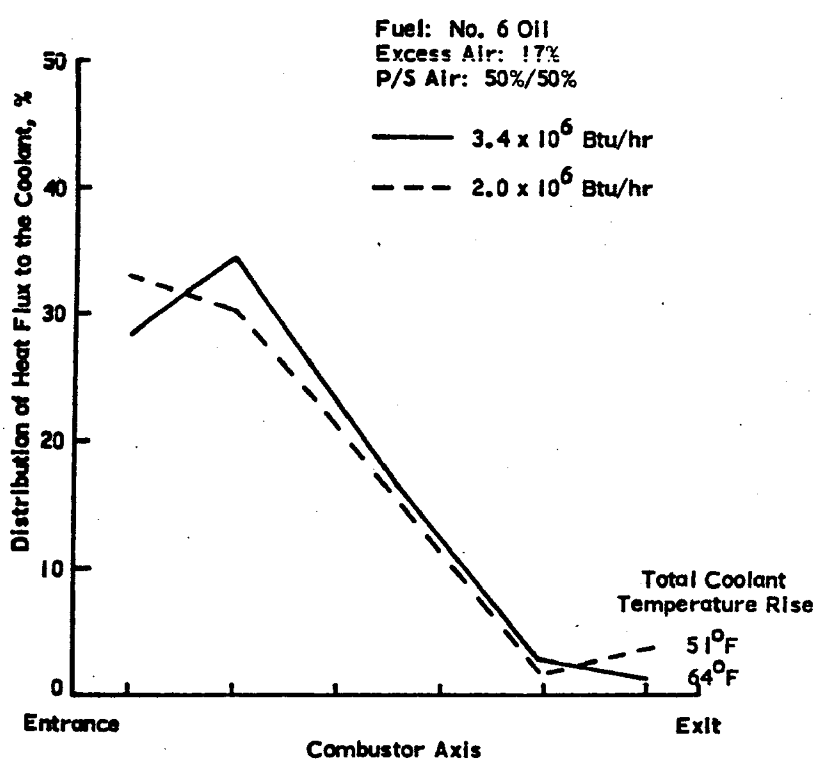

Figure 6

EFFECT OF LOAD ON THE COOLANT TEMPERATURE RISE ALONG THE COMBUSTOR

*Low sulfur residual oil (S = 1.0%, N = 22%, API gravity = 23)

V. EFFECTS OF BURNER MODIFICATIONS

A. SUMMARY

Using the baseline tests as a reference, a series of burner variation experiments were performed. These entailed fairly minor modifications to the system and thus are suitable for field conversion of existing units. Results of the burner variation studies have shown two variables (exclusive of the load to affect emissions by more than 20%: primary-to-secondary air ratio and atomization air pressure. A summary of these tests is presented in Table 4.

Table 4
BURNER VARIATION SUMMARY

Variable	Reduction in NO	Change in Smoke Emission (Bacharach Smoke Numbers
Load (3.4 to 2 x 10^6 Btu/hr)	22%	+2
Excess Air (35% to 17%)	18%	+4
Primary/Secondary Air Ratio (70/30 to 40/60)	28%	+7
Swirl Level (5.6 to 0.7, swirl parameter)	13%	+1
Atomization Air Pressure		
(10 to 20 psi)	4%	+1/2
(20 to 36 psi)	21%	+1/2
Oil Temperature (170°F to 200°F)	2%	+1
Atomization Fluid (Air → Nitrogen)	3%	-1/2

These burner variation tests were performed primarily with No. 6 oil, with a limited number of tests performed with No. 2 distillate oil. In all cases, the burner and combustor variables were held at the baseline condition except for the particular variable under investigation (e.g., primary/secondary air ratio, swirl, oil atomization air pressure, No. 6 oil temperature, oil atomization fluid).

B. PRIMARY/SECONDARY AIR RATIO

The distribution of air between the primary and secondary passageways has a significant effect on the emission of nitric oxide and smoke from the combustor. As shown in Figure 7, increasing the primary air from 40% to 70% increases the NO emissions by 28% and decreases smoke from a number 9+ (off scale) to a 2 Bacharach number. The increased NO and reduced smoke emissions with increasing primary air may result from net higher mean temperatures in the combustor and/or increased mixing between the fuel spray and primary air stream due to the increased primary air velocity.

As the primary-to-secondary air ratio is increased, the overall heat flux to the coolant is decreased 6%, resulting in higher mean temperatures in the combustor; however one sees very similar temperature distributions for the two different settings (see Figure 8). This decrease in heat flux was also reflected in a 3-1/2% increase in the combustor exit temperature of the combustion products (1617°F at P/S = 70%/30% vs. 1560°F at P/S = 50%/50%).

For comparative purposes, the effect of the primary-to-secondary ratio was investigated while firing distillate No. 2 oil. As seen in Figure 9, there is also a marked effect on P/S ratio on both the NO and smoke emissions when firing with No. 2 oil.

C. SWIRL

As is seen in Figure 10, changing the swirl in this burner has a very small effect on the NO and smoke emissions from the combustor. Also, the heat transfer profiles to the coolant for high and low swirl conditions were virtually identical, indicating that the changes in swirl did not influence the thermal behavior of the unit.

One explanation for the lack of sensitivity of the combustor to changes in swirl derives from the fact that the swirl component is contained solely in the secondary air. In the near vicinity of the nozzle the primary air dominates mixing. Hence changing the swirl by increasing the angular momentum of the

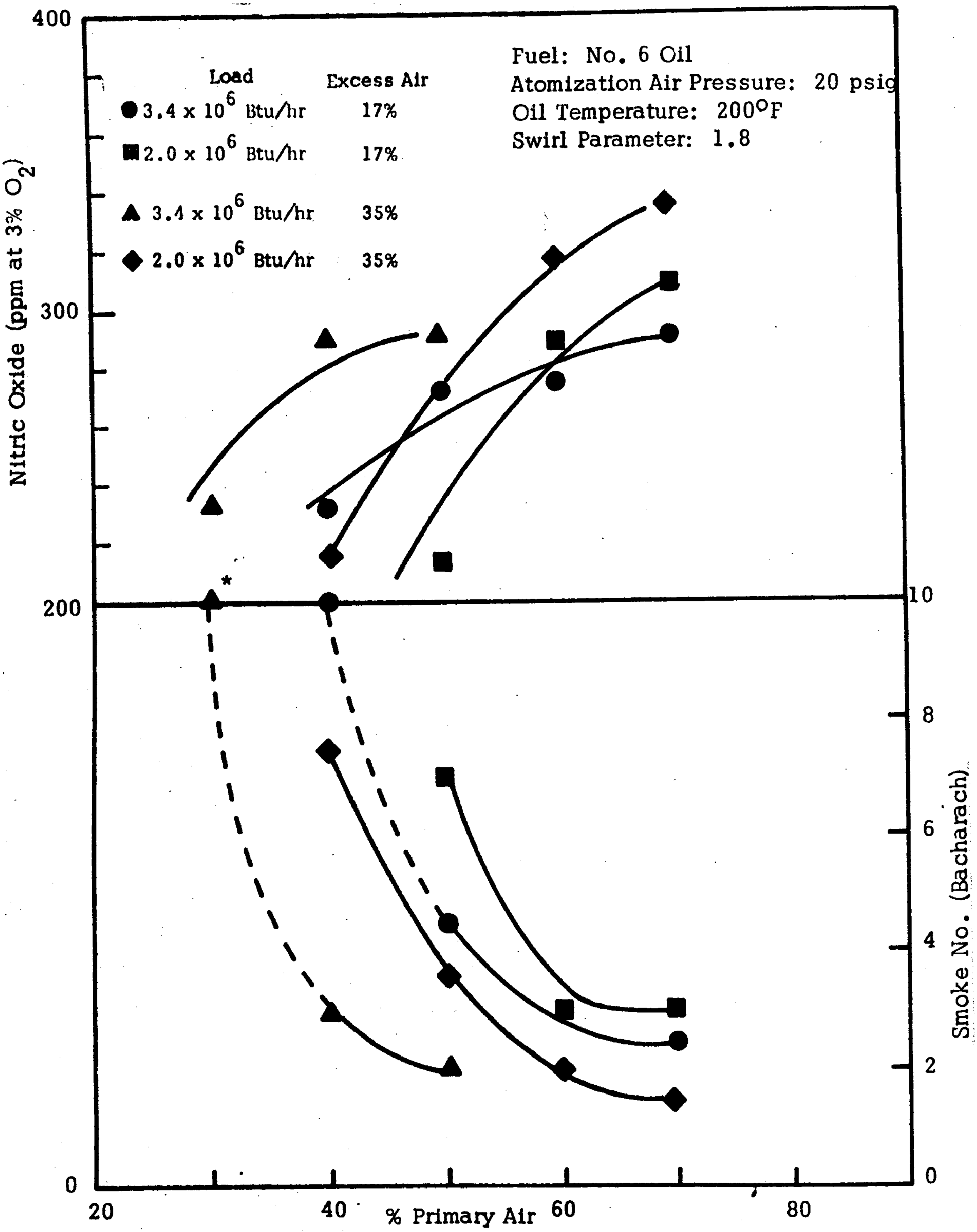

Figure 7
EFFECT OF PRIMARY/SECONDARY AIR RATIO
ON EMISSIONS

*Dashed line indicates that smoke level is off scale. A smoke number of 9 is the maximum on the Bacharach scale.

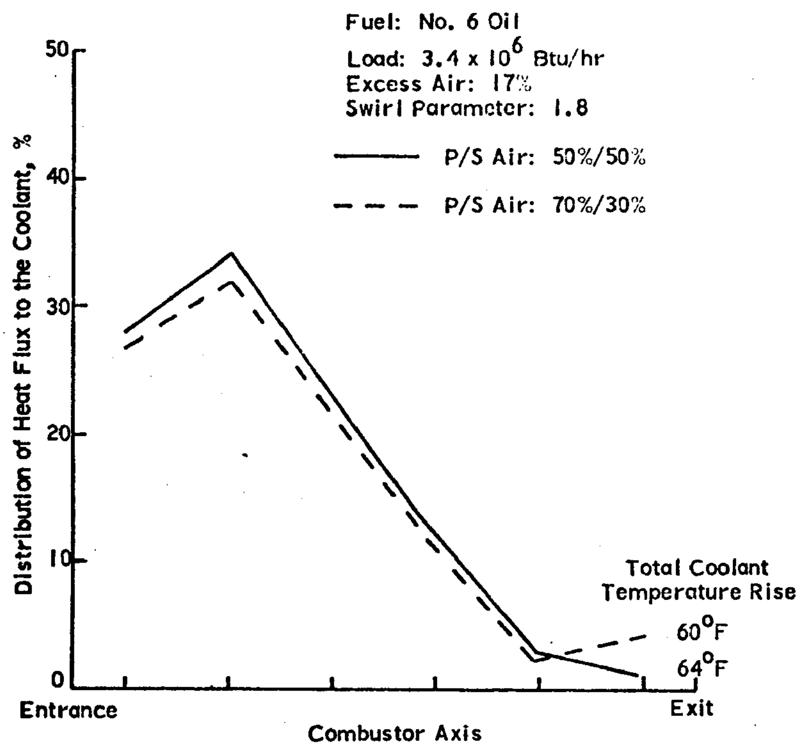

Figure 8

EFFECT OF PRIMARY/SECONDARY AIR DISTRIBUTION
ON THE COOLANT TEMPERATURE RISE ALONG THE COMBUSTOR

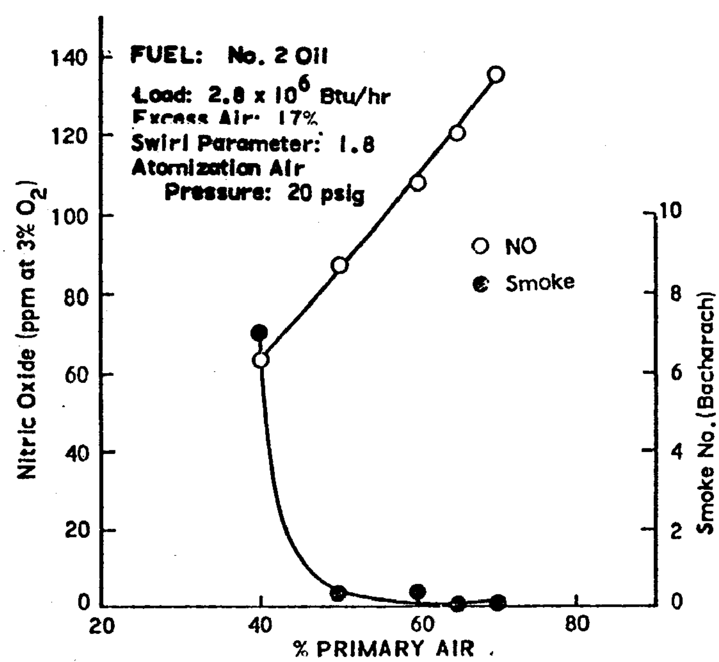

Figure 9

EFFECT OF PRIMARY/SECONDARY AIR RATIO
ON EMISSIONS
(No. 2 Oil Fired)

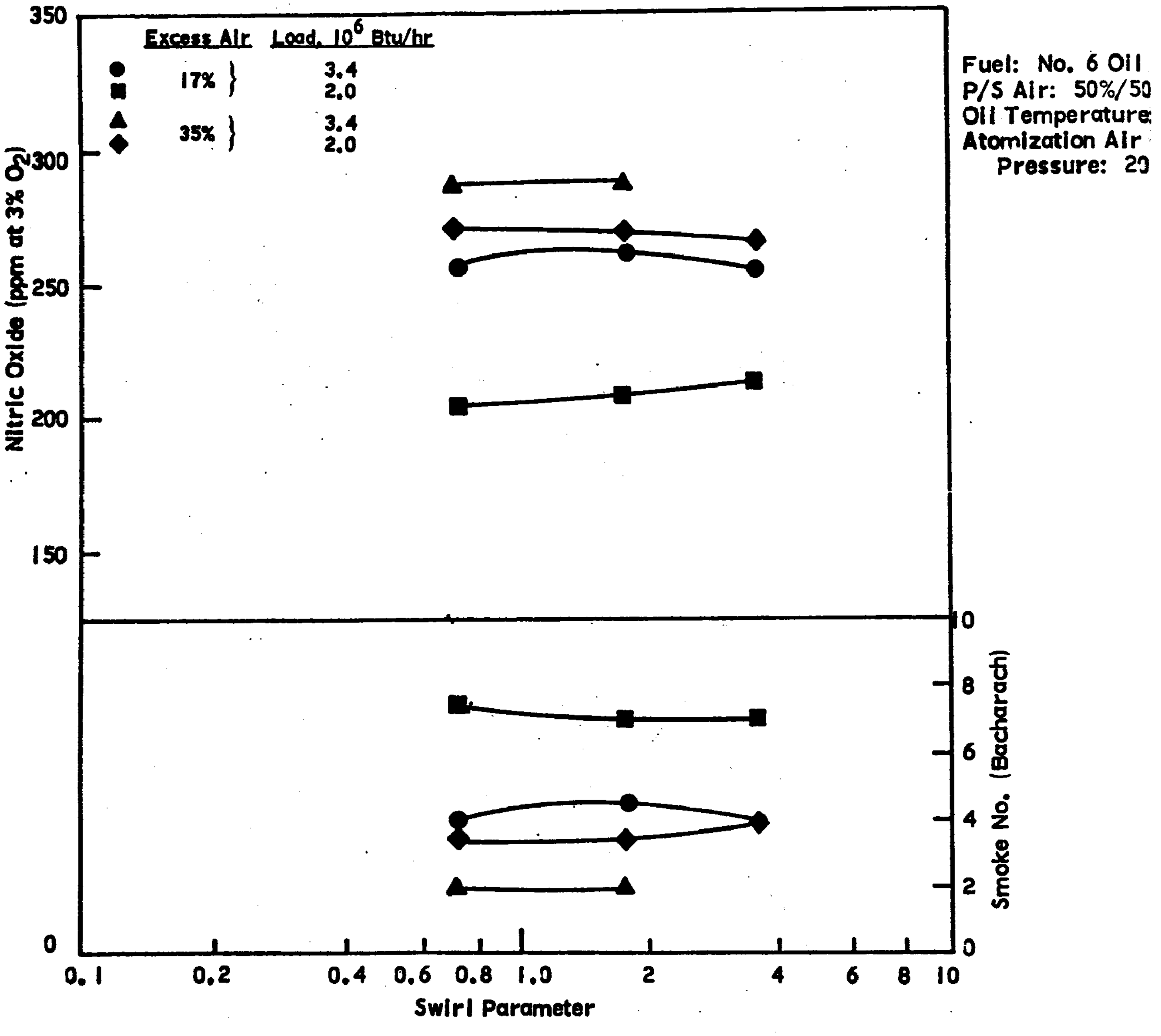

Figure 10

EFFECT OF SWIRL ON EMISSIONS

secondary air may not affect the mixing patterns between the fuel spray and the air. One might expect different results if the same "swirl parameter" had been obtained by varying the angular momentum in the primary air, with an axially directed secondary air flow.

D. NO. 6 OIL TEMPERATURE

The effect of oil temperature with No. 6 oil firing was investigated over the range of 170°F to 200°F. Over this range, the oil viscosity changed from 260 SUS* to 180 SUS. The results of these tests are plotted in Figure 11. During this test, variations in NO were limited to about 5%.

E. ATOMIZATION AIR PRESSURE

1. Design Range (10 to 20 psig)

The Monarch wide angle (80-70 deg) nozzle which is used in the burner operates with an air pressure range of approximately 10 to 20 psig. Over this range, the atomizing air flow rate varied from 9 to 15 cfm, or approximately 1% to 2% of the total air flow to the burner. Figure 12 presents the results of these tests, and it is seen that there was little effect on the emissions. No. 2 distillate oil results, which are plotted in Figure 13, are similar.

2. Extended Pressure Range (20 to 36 psig)

As can be seen in Figure 14, when the air atomization pressure was increased above 20 psig, the NO emissions dropped essentially linearly such that at an atomization pressure of 36 psig the NO emissions were reduced by 21%. Although the air for these tests was supplied by house air and not from the compressor supplied with the burner, no difference was found in the emissions using either house air or air from the burner compressor for a given atomization air flow. The smoke emissions only increased by 1 Bacharach smoke number as the pressure was raised from 20 to 36 psig. This reduction

*Saybolt Universal Seconds

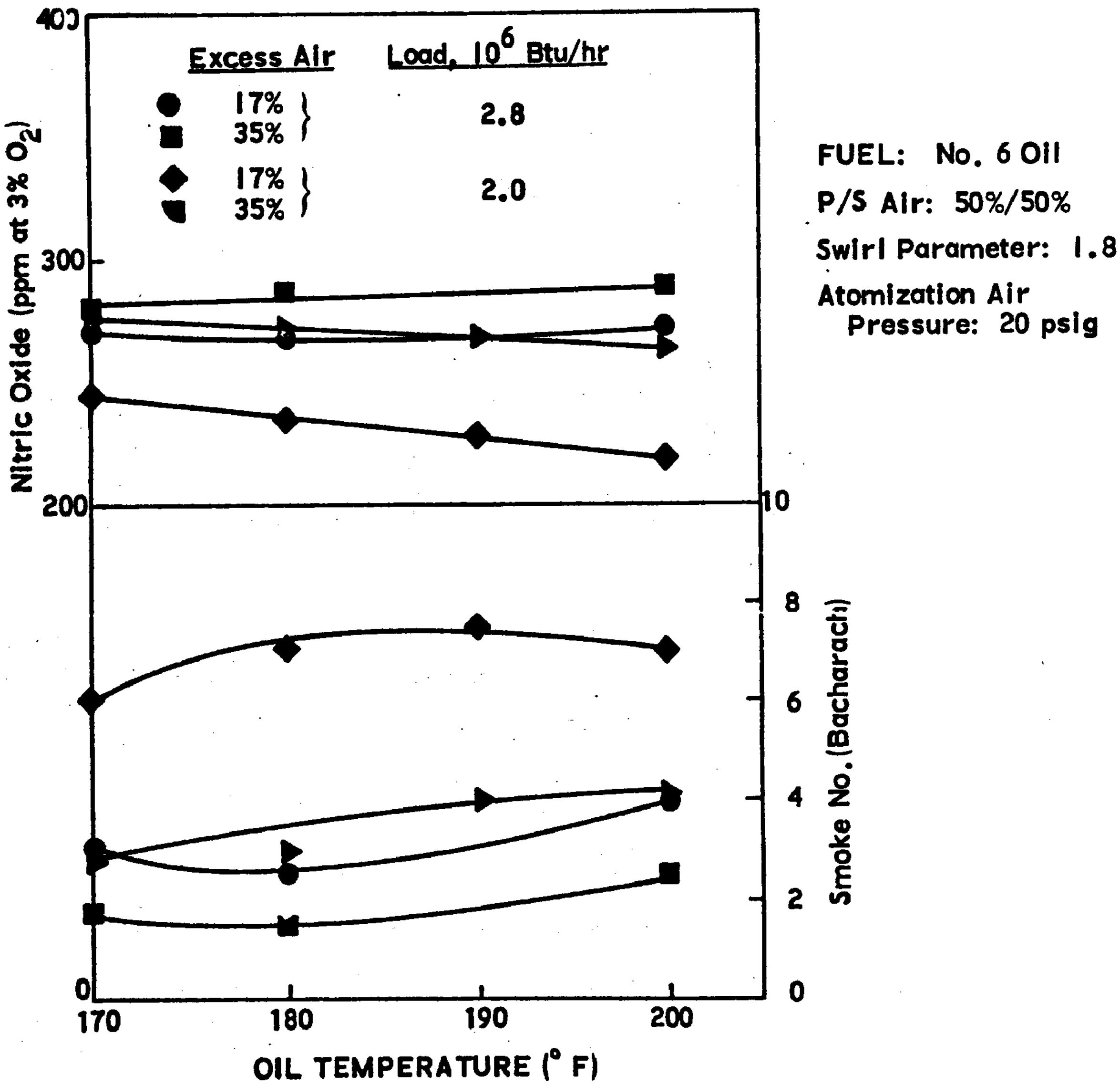

Figure 11

EFFECT OF OIL TEMPERATURE ON EMISSIONS

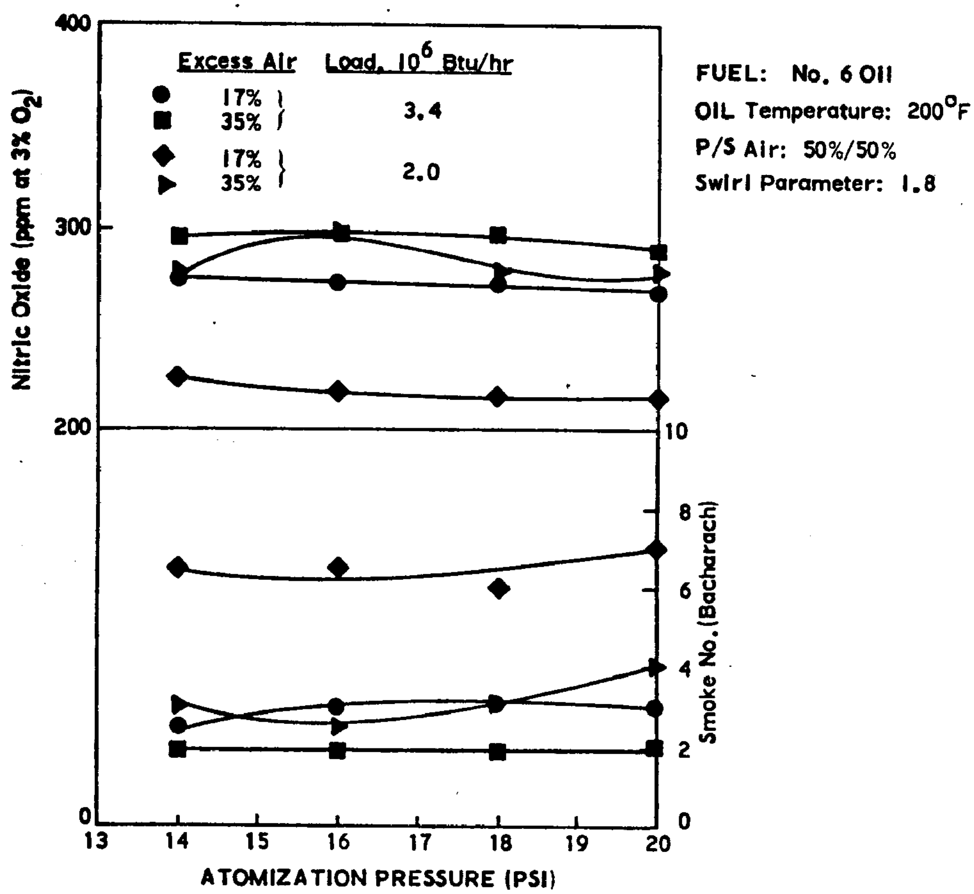

Figure 12

EFFECT OF ATOMIZATION PRESSURE ON EMISSIONS (No. 6 OIL)

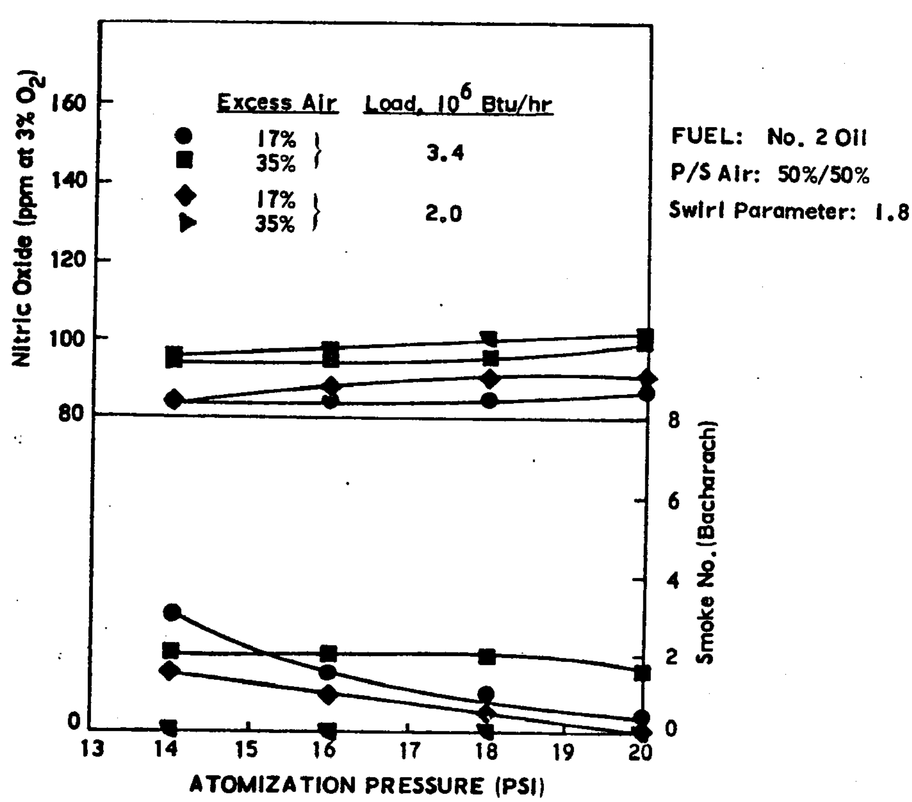

Figure 13

EFFECT OF ATOMIZATION PRESSURE ON EMISSIONS (NO. 2 OIL) in NO with increasing air atomization may be due to smaller droplets or collapsed spray angle:

- Increasing atomization pressure results in a spray with smaller drop sizes. Bowman and Kesten[9] have suggested that NO $\sim d^2$ for drops burning in an environment of constant ambient conditions. Thus the smaller drop sizes may lead to lower NO emissions.

- With increased air atomization pressure, the spray was observed (when sprayed in a stagnant atmosphere) to change from a conical form to a more cylindrical form. This would have the effect of delaying the mixing of fuel and air, and in effect "staging" the combustion.

Illumination of the exact mechanism requires detailed diagnostics of the atomization and flow characteristics which were beyond the scope of the present program.

3. Inert Atomizing Fluid

Also shown in Figure 14 are the results that were obtained using nitrogen as the atomization fluid. The rationale for performing this experiment was that although the atomization air only comprises 1% to 2% of the total air flow, it is intimately mixed with the oil and thus may have a substantially higher potential for NO formation. One might then expect the elimination of oxygen from the atomizing fluid to suppress the conversion of fuel-bound nitrogen to nitric oxide since only 0.2% of the total air flow to the burner is required to oxidize all of the fuel-N to NO. The experiments show that this did not seem to be the case, in that using nitrogen as the atomizing fluid resulted in only about a 10 ppm decrease in NO emissions. This indicates that although the atomizing fluid is initially initimately mixed with the fuel oil, the oxidation step of fuel nitrogen to NO occurs after a delay time such that sufficient mixing of the fuel and burner air has taken place in the combustor. This then diminishes the importance of the oil atomizing air in the fuel-N to NO conversion process.

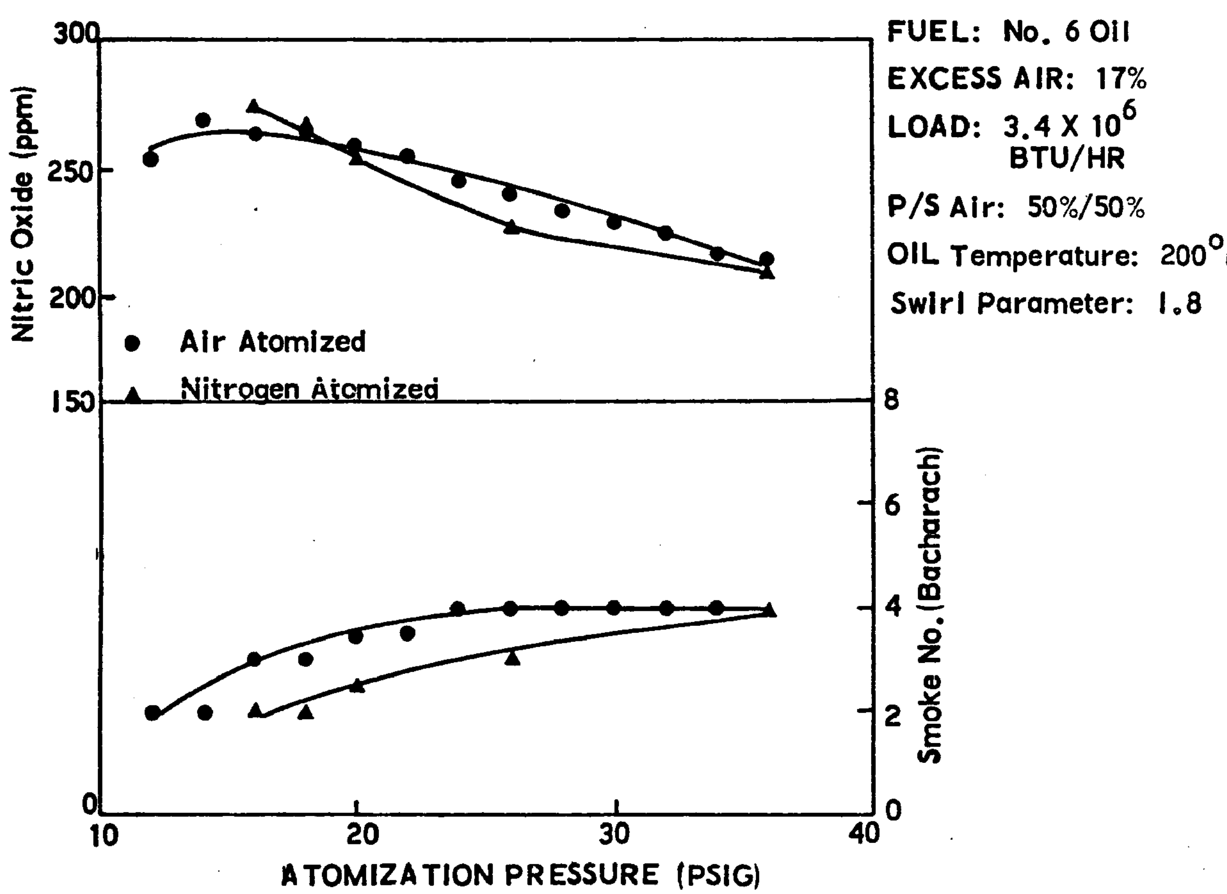

Figure 14

EFFECT OF ATOMIZATION PRESSURE
AND FLUID ON EMISSIONS

VI. EFFECTS OF FLUE GAS RECIRCULATION

A. SUMMARY

A summary of the flue gas recirculation (FGR) results is presented in Table 5. A number of observations may be made from the results presented in this section.

- FGR is most effective in reducing emissions when added so that the FGR comes into immediate contact with the oil spray (e.g., through the primary duct or gas ports).
- The effectiveness of FGR can be compromised if the FGR intensifies the mixing between fuel and air in the near vicinity of the burner.
- The heat transfer from the combustion products to the coolant is markedly reduced in the first part of the combustor due to the presence of FGR.
- FGR reduces NO emissions with relatively minor increases in smoke.
- Nitrogen content of the fuel limits the reductions in NO that can be obtained.

Table 5

FLUE GAS RECIRCULATION SUMMARY*
(No. 6 Oil, 17% Excess Air, 3.4×10^6 Btu/hr)

Location of FGR	NO Reduction with 30% FGR	Smoke Increase (Bacharach Smoke No.)	
		From	To
Gas Ports	42%	4	9
Total Air	25%	4	5
Primary Air	15%	4	5
Secondary Air	0	4	5
Quarl Injectors	0	4	3-1/2

*FGR was added to the particular stream while the primary/secondary air ratio was held constant.

B. EFFECT OF FUEL TYPE ON FGR EFFECTIVENESS

FGR is much more effective with natural gas firing than with oil firing, as shown in Figure 15. In these tests, the flue gas was added to the total air supply, with equal amounts to the primary and secondary air. Thirty percent recirculation results in a 67% reduction in nitric oxide under gas firing, whereas only 25% and 37% reductions are realized with No. 6 and No. 2 fuel oils, respectively.

The most reasonable explanation for this strong effect of fuel type relates to fuel nitrogen. FGR is expected to have a much greater effect on the NO formed via the thermal fixation of nitrogen than on the formation of NO due to the fuel-bound nitrogen, because the recirculated flue gas acts to reduce local temperatures and fixation is more temperature sensitive. Hence the more fuel nitrogen, the less NO reduction expected from FGR. For the fuel oils tested, the fuel-bound nitrogen in the No. 6 oil would contribute a maximum of 512 ppm NO whereas the No. 2 oil would contribute 65 ppm (both at 100% conversion and 3% O_2). The results presented in Figure 15 can be used to speculate that approximately 40% of the fuel-bound nitrogen in the No. 6 oil has been converted to NO, and about 91% of the No. 2 oil's fuel nitrogen is converted. This is based on the expectation of 67% reduction of <u>thermal</u> NO at 30% FGR regardless of fuel type and with no effect of FGR on fuel-N conversion.*

Another conceivable explanation is based on differences in fuel/air mixing with each fuel. The gas/air mixing process may be more sensitive to FGR than the oil-spray combustion. Certainly the heavy oil vaporizes at a slower rate such that FGR is dispersed over the entire air supply by the time ignition of the spray core occurs.

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*The NO comes from two sources: $(NO)_{total} = (NO)_{fuel} + (NO)_{air}$. We expect 67% reduction in $(NO)_{air}$ at 30% FGR from the gas results. Instead, we observe 20% reductions for No. 6 oil. Therefore we infer that $(NO)_{air} = 20/67\ (NO)_{fuel}$ for No. 6 oil, or $(NO)_{fuel} = 67/87\ (NO)_{total}$. At $(NO)_{total} = 271$ ppm this means that $(NO)_{fuel} \approx 210$ ppm or about 40% conversion. For No. 2 oil, similar reasoning leads to $(NO)_{fuel} = 2/3\ (NO)_{total} \approx 62$ ppm, or about 91% conversion. This estimate ignores obvious differences in aerodynamics due to fuel type or FGR.

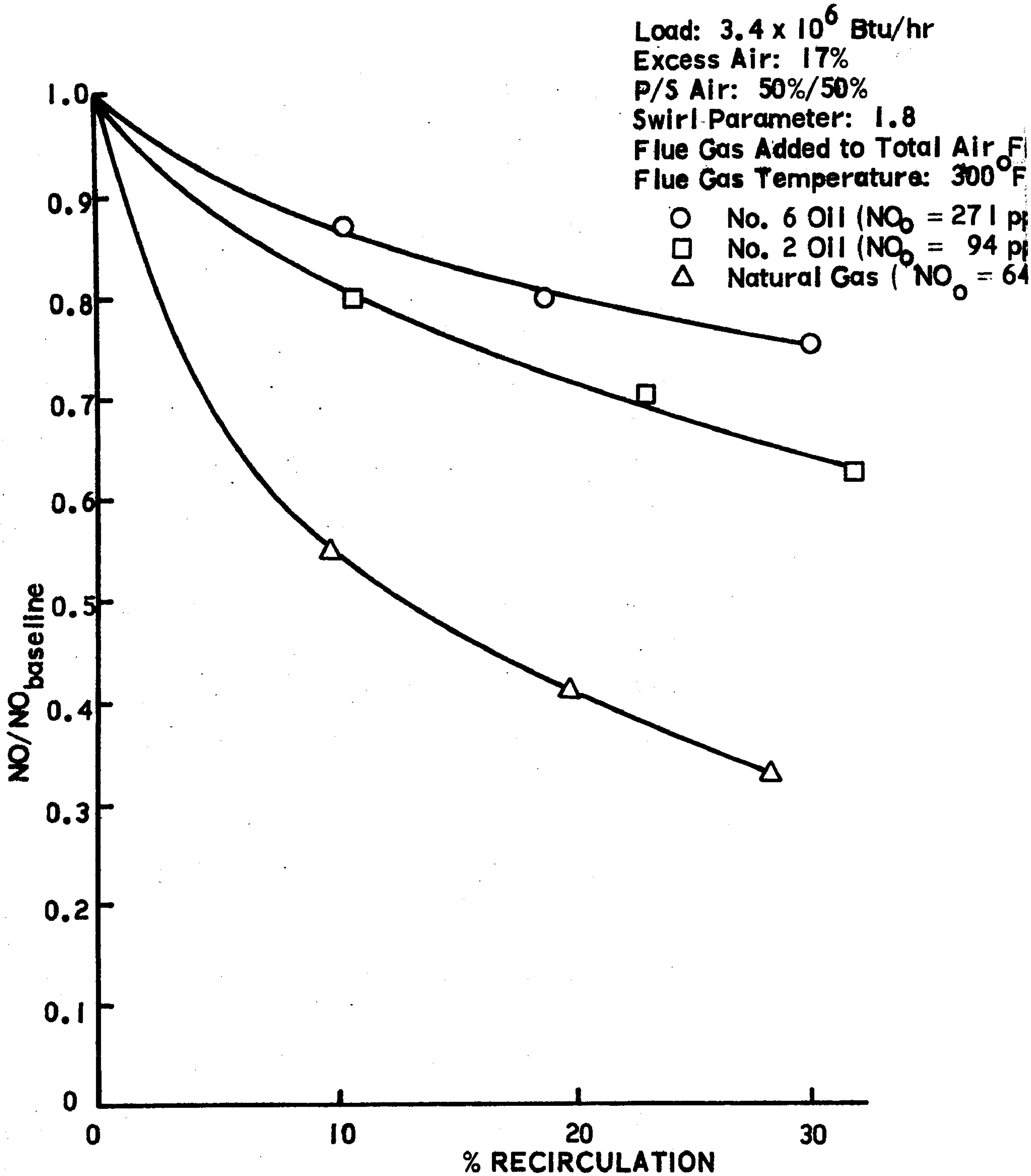

Figure 15

EFFECT OF LEVEL OF FGR AND FUEL TYPE ON NO EMISSIONS

C. EFFECT OF LOCATION OF FGR DELIVERY

The point at which flue gas enters the combustor determines the effectiveness of FGR in reducing nitric oxide. In the present case, flue gas can be introduced into the system at five discrete locations, only three of which proved to be effective as summarized below:

- Most Effective (40% reduction in NO): Addition of FGR through the natural gas ports.
- Moderately Effective (15-25% reduction in NO): Addition of FGR to the primary air, or total air stream or combinations which include either the primary air stream or gas ports.
- Ineffective (no reduction in NO): Addition of FGR through the secondary, quarl throat injectors or the secondary plus throat.

The above points are illustrated in Figure 16. A data listing for the FGR tests is given in Table 6. For all FGR tests, 50% of the air entered the combustor through the primary and 50% through the secondary, hence the mass flowrates through the primary and secondary were no longer equal when the flue gas was added.

FGR was most effective when introduced through the natural gas ports: 17.5% FGR reduced the NO emissions by 41%. However, the NO reduction was accomplished at the expense of increased smoke emissions (e.g., a 4 Bacharach smoke number with zero FGR, and a greater than 9 smoke number with 17.5% FGR). Qualitatively, when the flue gas was added through the gas ports, visual observations showed the flame to be grossly distorted. The flame was no longer symmetrical, with the FGR jets from the gas ports tearing the flame into isolated sections.

FGR was less effective (up to 25% reductions) when delivered either into the total air stream (primary plus secondary) or into just the primary. However, beyond 14% FGR, flue gas additions to the primary air stream resulted in an upward trend in NO emissions. This counteracting effect of

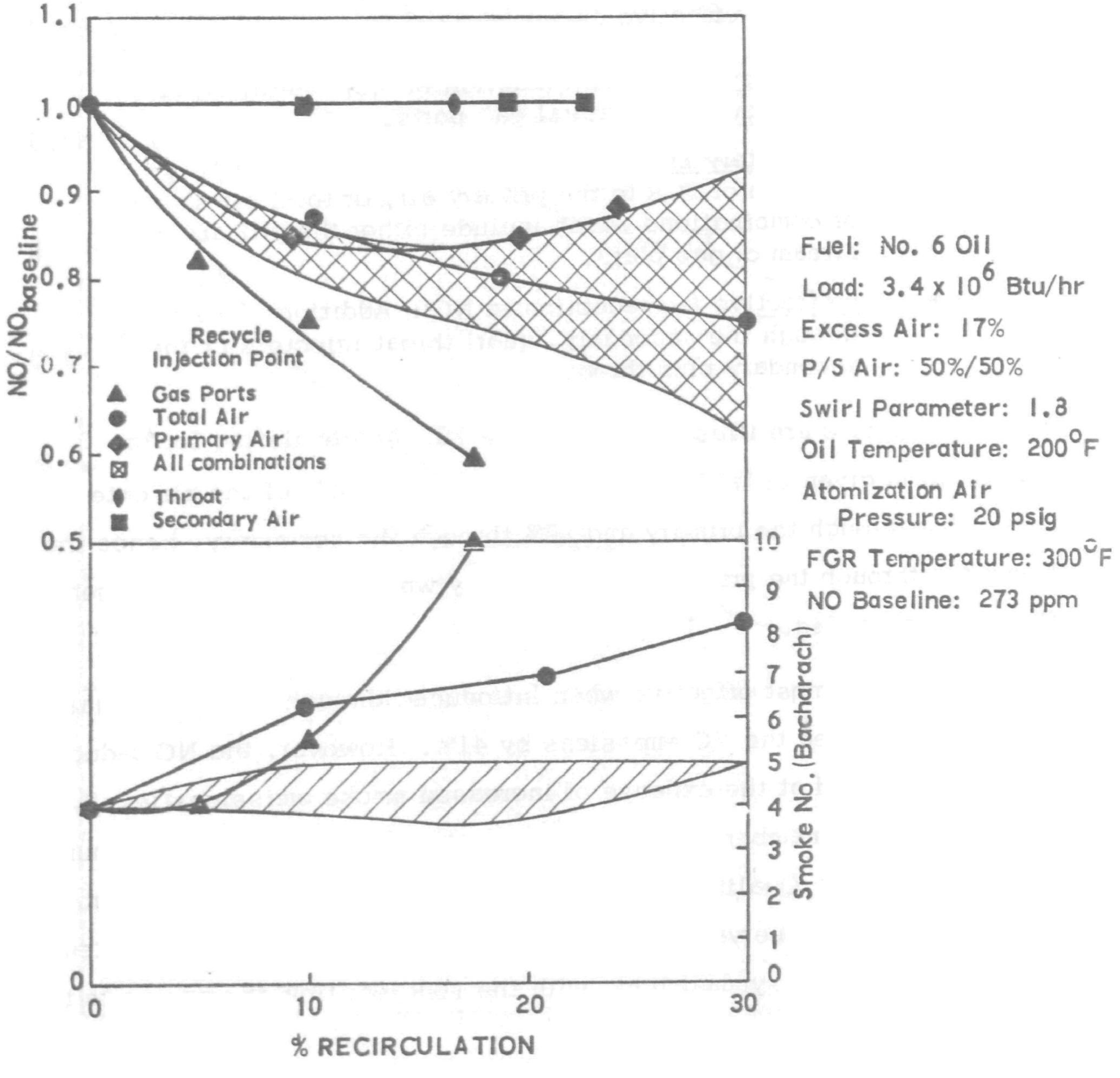

Figure 16

EFFECT OF FGR INJECTION LOCATION ON EMISSIONS

Table 6

Excess Air	FGR Delivery Point	LOAD															
		3.4×10^6 Btu/hr % Recirculation								2.0×10^6 Btu/hr % Recirculation							
		0		10		20		30		0		10		20		30	
		NO (ppm)	Smoke	NO/NO_o	Smoke	NO/NO_o	Smoke	NO/NO_o	Smoke	NO (ppm)	Smoke	NO/NO_o	Smoke	NO/NO_o	Smoke	NO/NO_o	Smoke
	Total	274	4	.87	4	.80	5	.75	5	221	2	.96	6.5	.91	6.5	.89	7.0
	Primary			.85	5	.84	5	.88	5(24%)*	-	-	-	-	-	-	-	-
	Secondary			.99	5	1.03	5.5	1.04	5(22.5%)	-	-	.99	7	1.03	7	.99	8.0
	Gas Port			.75	5.5	.59	> 9	-	-	-	-	-	-	-	-	-	-
	Quarl Injectors			1.0	4	1.0	3.5	-	-	-	-	-	-	1.2	6.5	-	-
	Primary/Gas Port			.83	6	.83	6	.86	6	-							
17%	Secondary/Gas Port	274	4	.81	6	.73	6.5	.62	8								
	Quarl/Gas Port			.83	7	.81	6	.75	7								
	Primary/Quarl			.89	5	.83	5.5	.96	5								
	Secondary/Quarl			.99	4.5	1.07	5.5	1.09	4.5								
	Primary/Secondary			.87	4.5	.79	5	.75	5.5								
	Total			.89	2.5	.87	3	.85	5	245	3.5	.9	4	.86	5.5	.83	6.0
	Primary			.83	2.5	.75	5	.67	5	-	-	.9	4	.93	5	.98	6.5
35%	Secondary	262	2	-	-	-	-	1.01	3(25%)	-	-	-	-	-	-	1.03	6.5
	Gas Port			.87	3	.74	5(16%)		-	-	-	.88	4	.85	6	-	-
	Quarl Injectors			-	-	1.1	3(14%)	-	-	-	-	-	-	1.1	5	-	-

* Numbers in parenthesis indicate FGR levels other than corresponding to co umn.

increased mass flow rates through the primary is probably associated with changes in the rate of fuel/air mixing. The effect disappears at higher (35% vs. 17%) excess air, as shown in Figure 17. Here the nitric oxide emissions decrease monotonically as the flue gas is added through the primary with a 33% reduction occurring at 27% recirculation.

Adding the flue gas to the secondary air stream or through the throat injectors in the quarl had no effect on the nitric oxide emissions from the combustor. This insensitivity reiterates that NO_x formation appears to be controlled by processes occurring in the core close to the axis of the furnace.

As seen in Figure 16, multiple injection configurations are basically no more effective than the single injection schemes. At a load of 3.4×10^6 Btu/hr and 17% excess air, a maximum reduction of 38% is obtained by injecting the flue gas (30% recirculation) in equal quantities into the secondary and gas ports. However, this results in an increase in smoke from a number 4 to a number 8 Bacharach smoke number, which presumably would be an unacceptable penalty in an actual field application.

At reduced load (2×10^6 Btu/hr), the NO reductions are not as substantial (18% as opposed to the 30% reductions obtained at full load).

D. EFFECT OF FGR ON HEAT FLUX DISTRIBUTION

Insight into the differences in relative effectiveness of the various configurations of adding flue gas to the combustor can be obtained by comparing the thermal performance of the combustor. Figure 18 shows the exit temperatures and reduction in heat loss to the walls for various FGR delivery points. The amount of heat transfer to the coolant decreases about 35% as the level of FGR increases up to 20%. Possible explanations for this reduction are as follows:

- Lower mean temperature of the gas mixture after complete combustion.
- Lower achieved temperature due to heat release occurring out of the combustor in the flue.

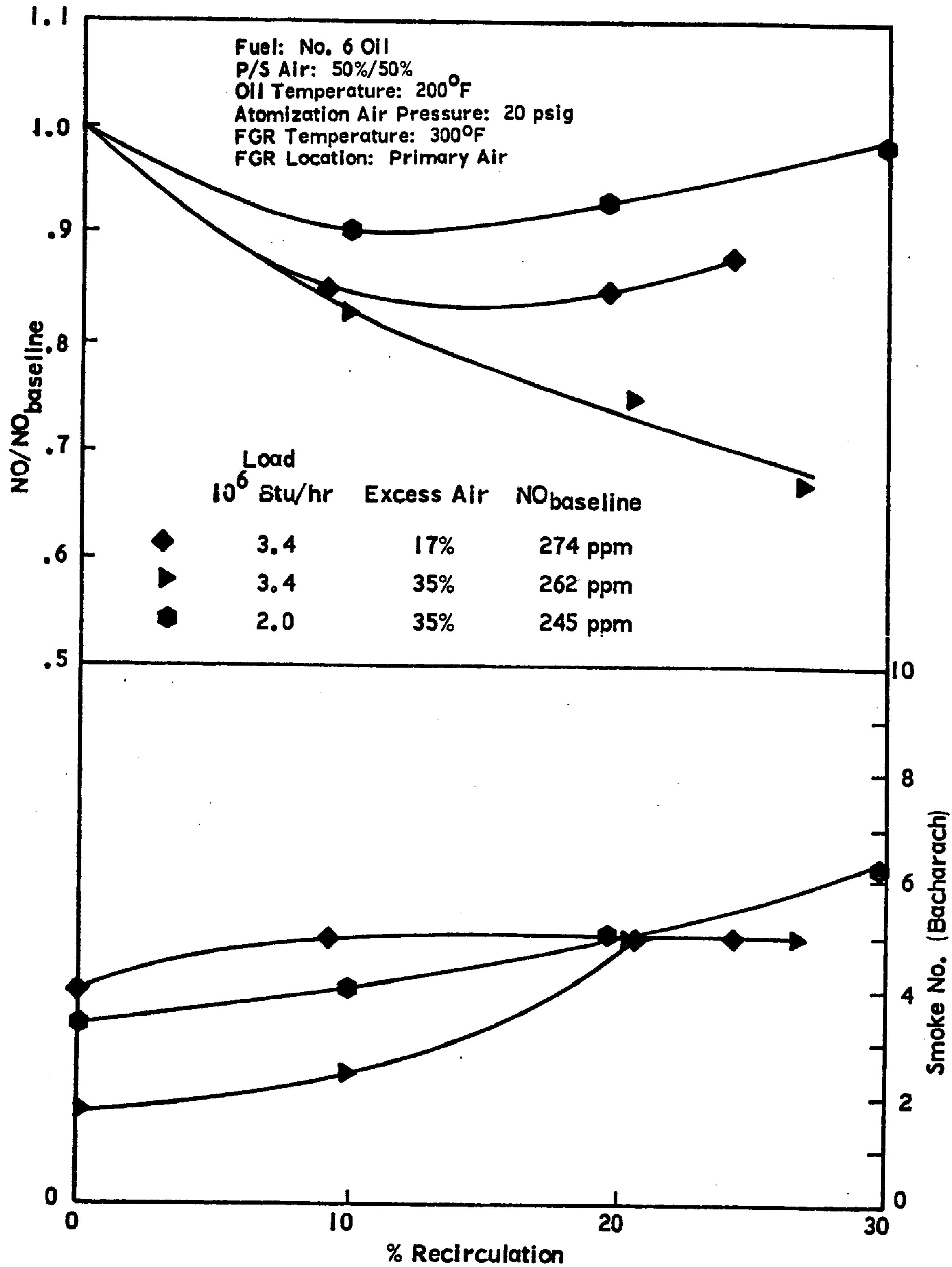

Figure 17

EFFECT OF LOAD AND EXCESS AIR
ON THE EFFECTIVENESS OF FGR

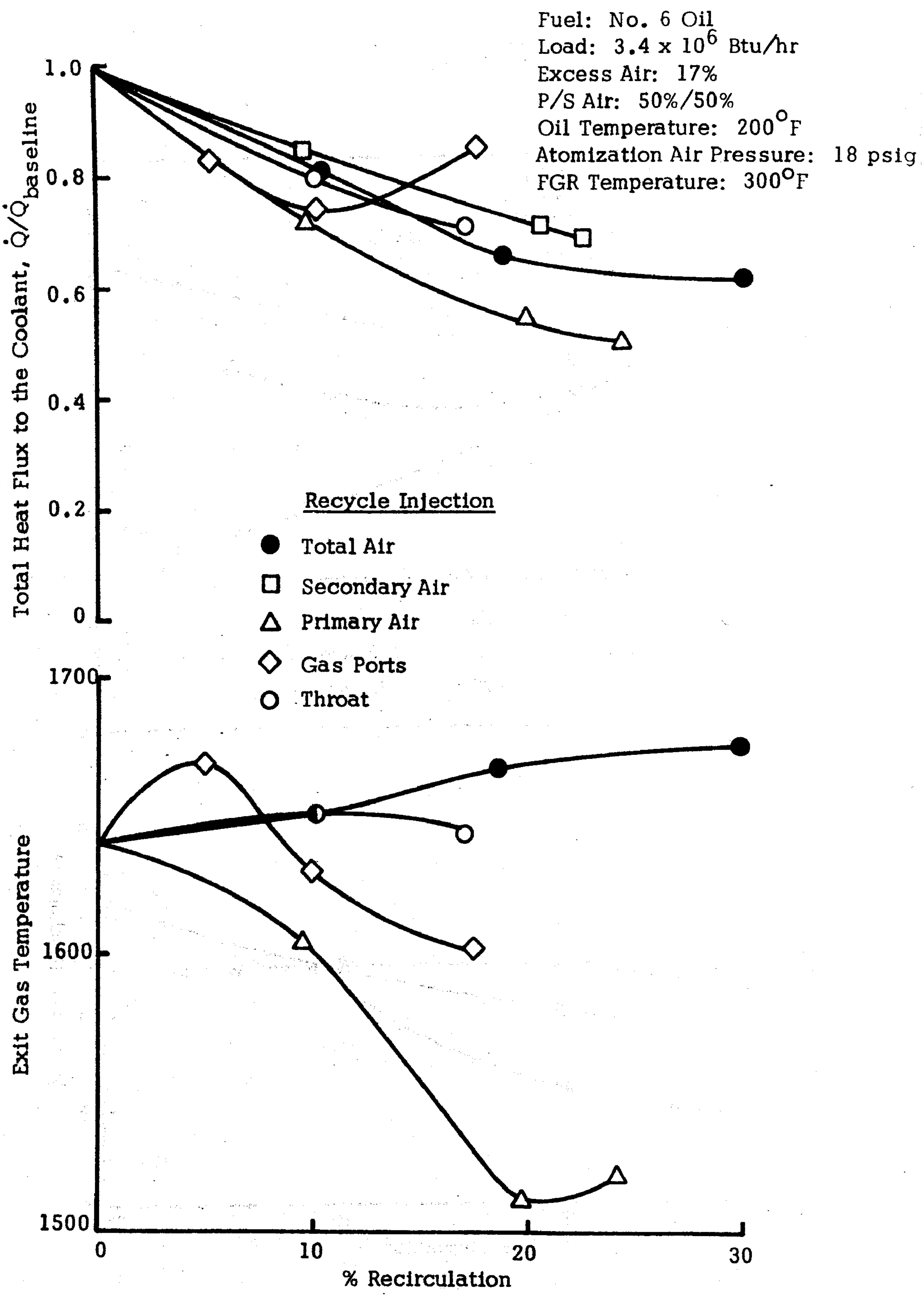

Figure 18

EFFECT OF FGR INJECTION LOCATION
ON COMBUSTOR HEAT LOAD CHARACTERISTICS

- Reduced residence time for heat transfer due to the increased flow through the combustor.
- Cool insulating layer of air or flue gas formed between the wall and hot combustion products.
- Increased blockage of radiative flux from the flame core due to increased soot.

The first two mechanisms above give a reduction in the exit temperature, whereas the last three would predict an increase. Exit temperature is observed to decrease when the FGR is added through the primary stream and the natural gas manifold, supporting the first two hypotheses.

FGR reduces the heat flux to the coolant mainly in the first third of the combustor with fairly minor changes in the distribution occurring in the latter two-thirds of the combustor, as shown in Figures 19 and 20.* This axial distribution is reasonable because the effect of FGR should be greatest where the baseline flame is hottest (the primary).

Reduced heat loss in the first third of the combustor appears to correlate with reduced NO emissions, as shown in Figure 21. Adding the flue gas to the total air supply and the primary air resulted in the greatest reductions of both.

Adding the flue gas through the gas ports had little effect on the heat transfer distribution, but resulted in the greatest increases in smoke emissions. These clues, coupled with the visual observations of a greatly distorted flame with FGR addition through the gas ports, suggest that the changes in emissions in this case are primarily due to severe changes in the aerodynamic patterns in the near vicinity of the oil nozzle and not solely due to a thermal effect of the FGR.

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*In Figures 19 to 21 only the heat flux distribution from the burner to the combustor midpoint is presented in that the distribution in the back half of the combustor is virtually identical.

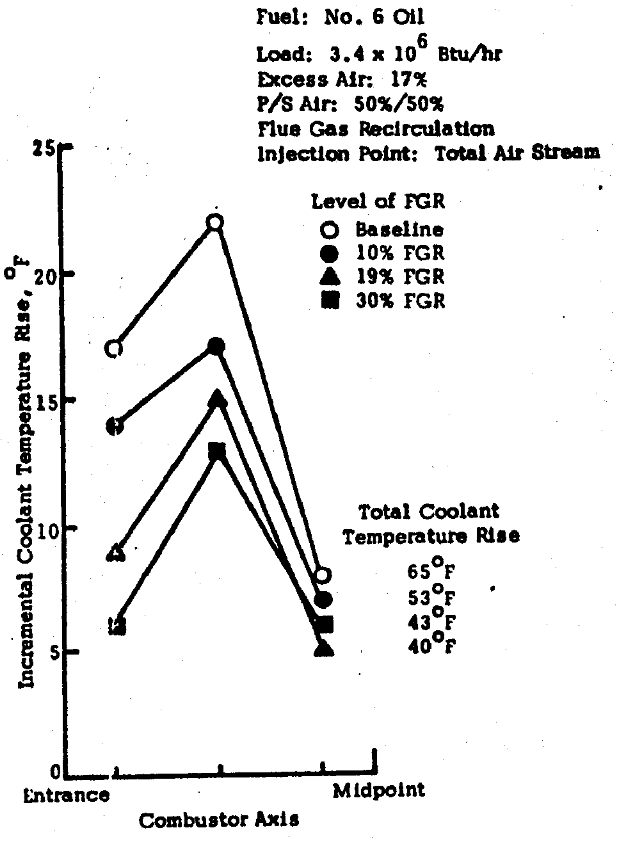


Figure 19

EFFECT OF FGR LEVEL ON COMBUSTOR HEAT FLUX DISTRIBUTION (HIGH LOAD)

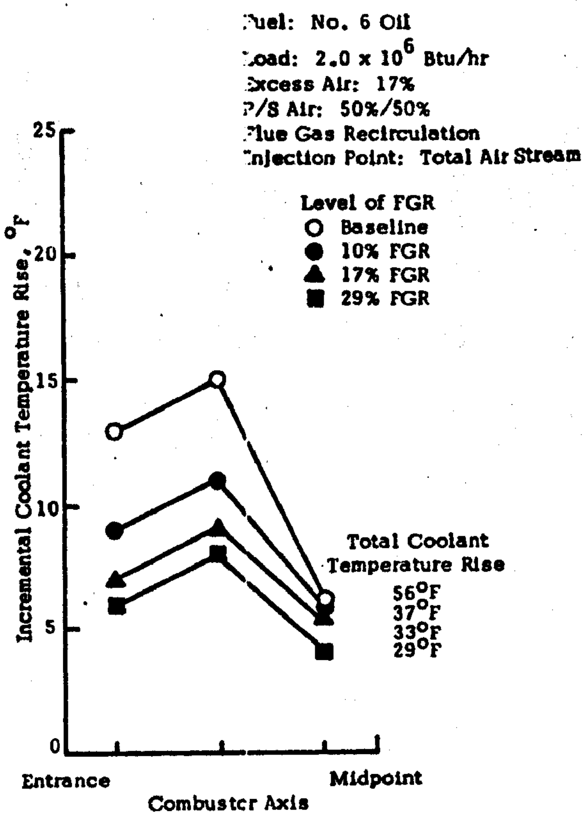


Figure 20

EFFECT OF FGR LEVEL ON COMBUSTOR HEAT FLUX DISTRIBUTION (LOW LOAD)

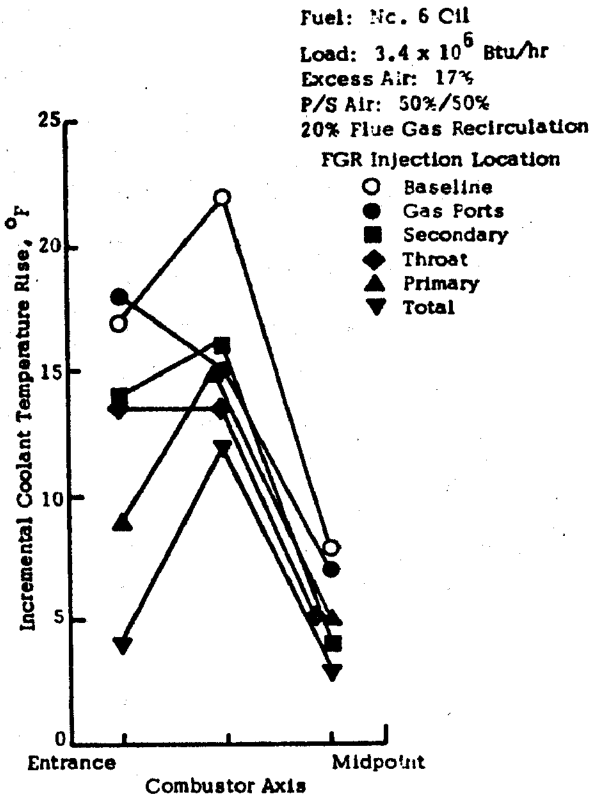


Figure 21

EFFECT OF FGR INJECTION LOCATION ON COMBUSTOR HEAT FLUX DISTRIBUTION

The series of tests reported in Figure 22 were designed to add FGR to the burner while minimizing the flowfield changes. Flue gas was added to the primary or secondary, keeping the mass flow rates between primary and secondary equal. For example, as recirculated flue gas was added to the primary, an equal amount of air flow was rerouted through the secondary. This presumably kept the flowfield and relative mixing rates of primary, secondary, and fuel spray essentially unchanged (although the overall gas velocity through the burner increased due to the addition of the flue gas). Unavoidably, these tests involved not only FGR but also P/S ratio as variables. With a 50%/50% primary/secondary ratio, 26% reduction in NO was obtained by adding 30% flue gas to the primary air stream. This reduction in NO is greater than that obtained when the flue gas was added to the primary without compensating for the air (Figure 16). This is due to the two factors mentioned above: (1) presence of the flue gas, and (2) reduced primary-to-secondary air ratio at the burner. When the flue gas was added to the secondary, the NO emissions increased above the baseline conditions, probably due to the increased air flow through the primary air stream. An increase in the P/S ratio was shown to increase NO (see Figure 7).

E. FGR WITH NO. 2 FUEL OIL AND NATURAL GAS

The No. 2 fuel oil responded to FGR in a similar manner to the No. 6 oil, with the most effective injection locations being the gas ports, primary air, or total air stream. Some of the results have already been presented in Figure 15, the remainder are shown in Figures 23 and 24. As with No. 6 oil, during all of these FGR tests, the CO emissions were less than 100 ppm.

With natural gas, the most effective location for FGR was the total air stream. This is due to the construction of the gas manifold. The gas manifold is built with slots that inject natural gas into both the primary and secondary streams. Thus one would expect the most effective configuration to be one that injects the flue gas into both the primary and secondary.

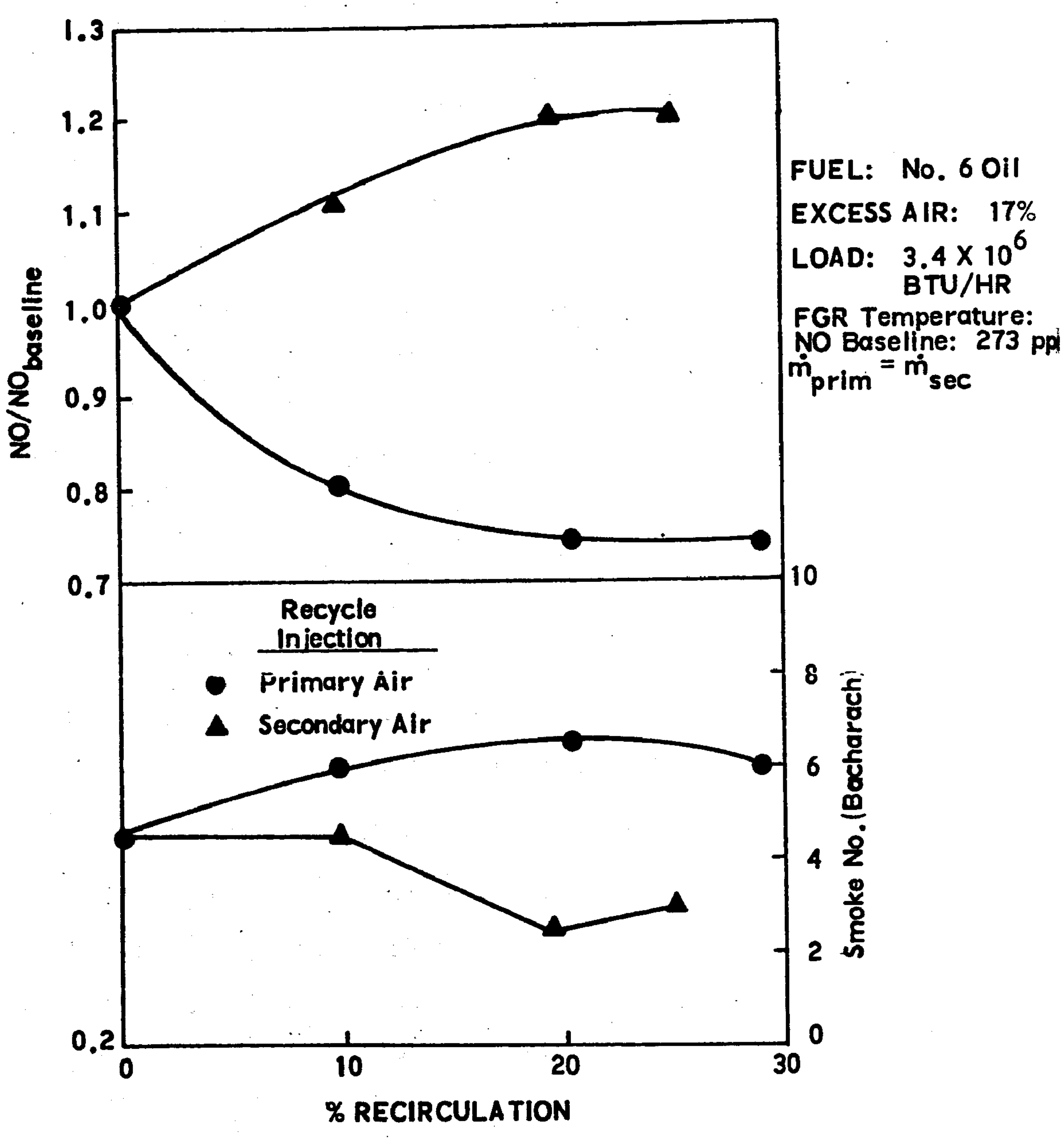


Figure 22

EFFECT OF FGR LEVEL, WITH PRIMARY AND SECONDARY FLOWS CONSTANT, ON EMISSIONS

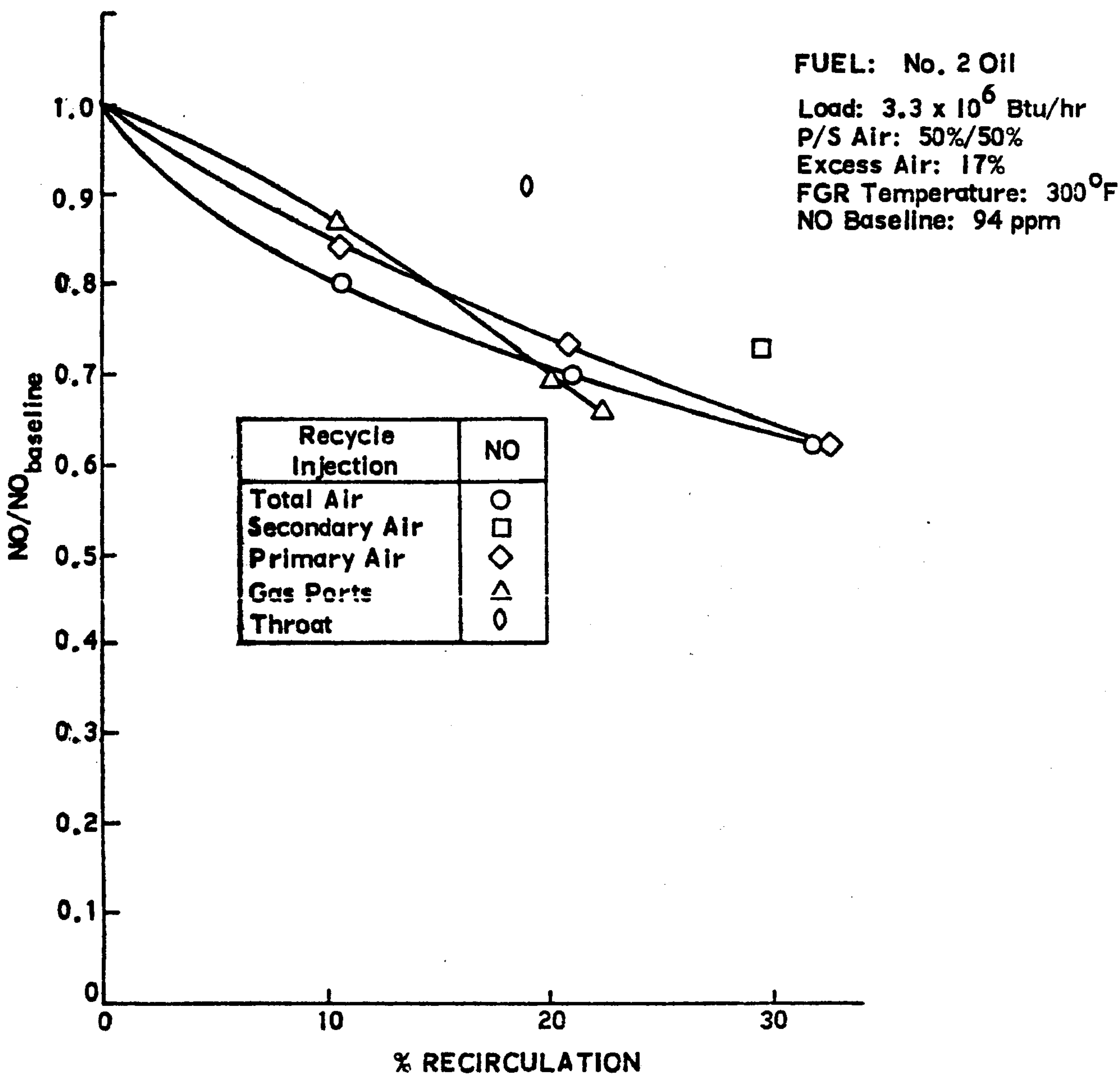


Figure 23

EFFECT OF FGR INJECTION LOCATION
ON EMISSIONS
(No. 2 Oil Fired)

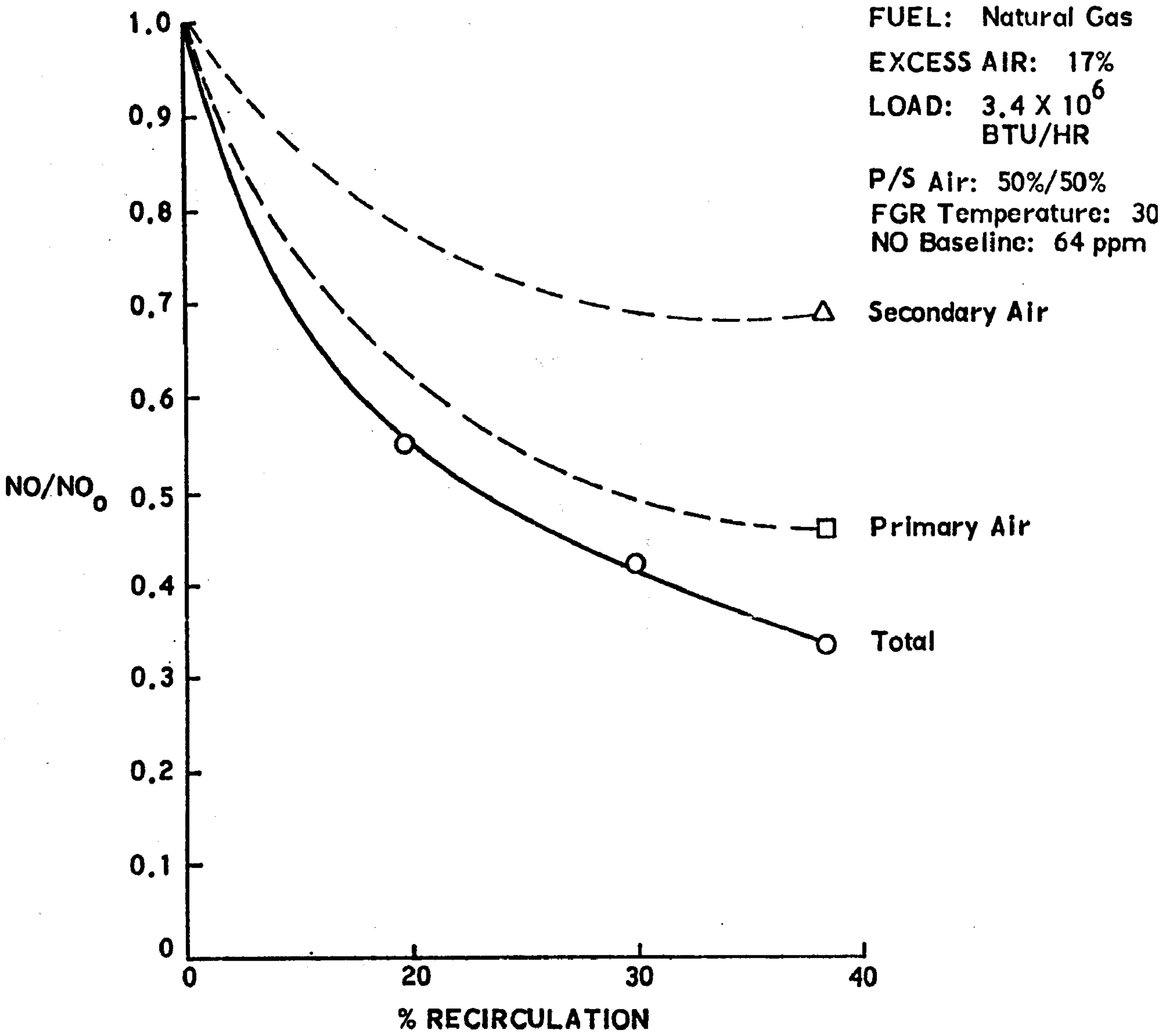


Figure 24

EFFECT OF FGR INJECTION LOCATION ON EMISSIONS (Natural Gas Fired)

VII. EFFECTS OF STAGED COMBUSTION

A. SUMMARY

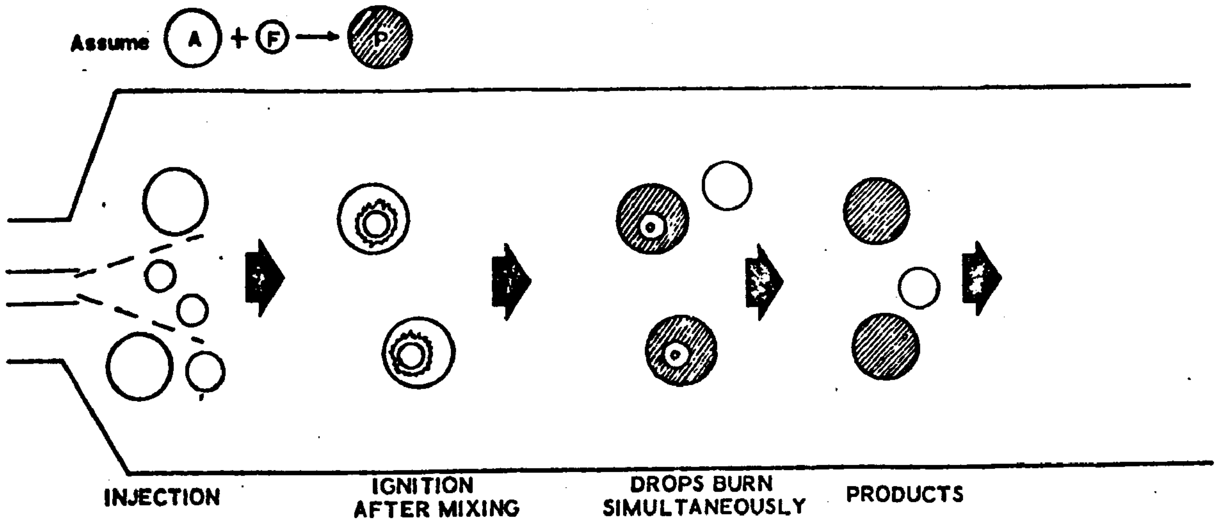
Results of the SC tests indicate the following:

- Sidewall and rear-boom give 24% and 20% NO reductions, respectively, for the present burner configurations.
- Combustion staging has been found more effective in prior investigations of a 50 HP Cleaver Brooks boiler[5] and of utility boilers[10].
- NO reductions are limited by increased smoke.
- The large smoke emissions could be due to either (1) the overall rich fuel/air ratio near the burner, or (2) the reduced fuel/air mixing rate caused by reduced air flow at the burner.
- Staged combustion results in increased heat transfer from the gases to the coolant. This is probably due to changes in flow patterns in the combustor.
- Downstream air injection increases NO unless injected at least 1.5 combustor diameters downstream from the oil nozzle. This result indicates that the near-burner flame is effectively naturally staged without any deliberate SC.
- With rear boom staging, half of the ultimate reduction is obtained with the first 5% of the air staged.

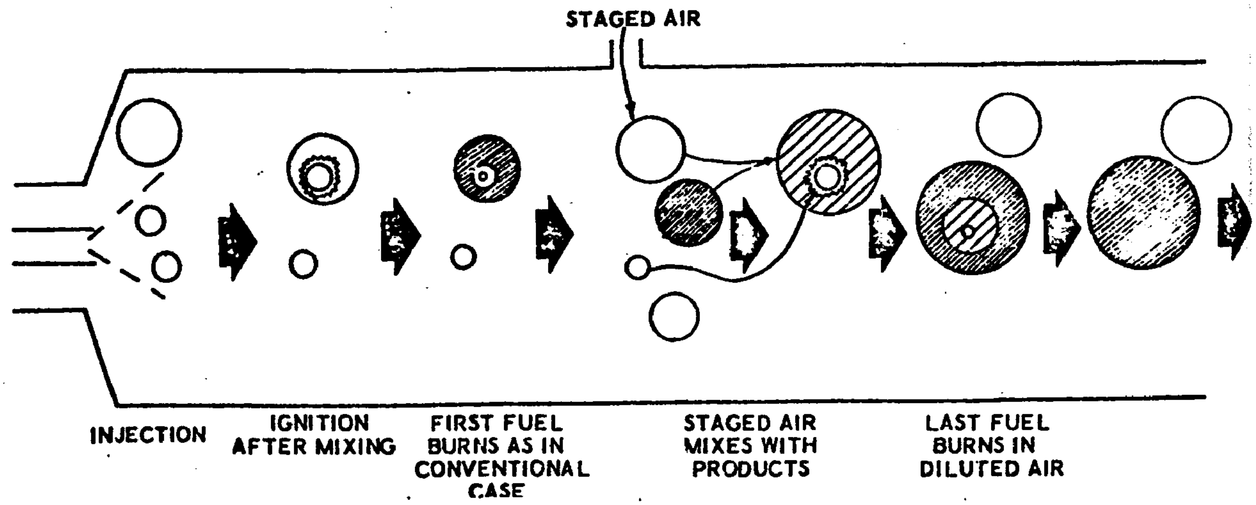
B. INTERPRETATION FOR OIL FIRING

"Staged combustion," in the sense of a fuel rich premixed region near the burner and delayed air addition downstream is a misleading over-simplification in the combustion of No. 6 oil. The mixture produced by an oil-fired burner is a heterogeneous mixture of liquid fuel drops and air burning in a diffusion flame. Realistically, what probably is occurring in this diffusion flame situation is staged burning of drops such that drops are burning in "series," in the products of previously burned drops. This is contrasted to the baseline or non-staging case where there is sufficient air near the burner for all (or at least a large majority) of the drops to burn in "parallel". For this reason SC is not expected to be as effective for oil as for natural gas. In fact, gas turbine manufacturers have found that this limitation can only be circumvented by prevaporizing the fuel.

This is illustrated below in Figure 25. Consider a fuel nozzle which injects two fuel drops of such a size that under stoichiometric conditions each requires one unit of air to burn, thus for overall stoichiometric conditions two units of air are added to the combustor. Figure 25 compares two configurations in which the fuel may be burned: (a) all air and fuel added at the burner, and (b) insufficient air added at the burner, the remainder of the air added downstream.



(a) CONVENTIONAL CASE



(b) STAGED COMBUSTION CASE

Figure 25

PHENOMENOLOGICAL SKETCH OF STAGED COMBUSTION IN OIL FLAMES

In the conventional case (a), there is sufficient air in the local environment of each fuel drop or element near the burner that both fuel elements can burn simultaneously. Thus the environment that each fuel element will see will consist initially of air and as burning proceeds, the environment will progress towards the final level of excess air plus the fuel elements' own products.

The situation is distinctly different with staging [case (b)], where there is not sufficient air at the burner for all of the fuel elements to burn simultaneously. The first element will burn as a diffusion flame in an environment of air and its own combustion products, largely unaware of the presence of the other fuel element (this is due to the self-regulation of a diffusion flame with regard to its stoichiometry).

The second fuel element (drop or vapor pocket) will not have air to burn and must wait until the air is added downstream. When this secondary air is added, the environment which the second fuel element "sees" will no longer consist initially of air; rather it will initially see air plus the products of combustion of the first fuel element as it burns.

Four effects important to NO and smoke formation will occur as follows:

(1) While awaiting the addition of secondary air, the second fuel element will be preheated and vaporized by the combustion products of the first element. This will "initiate" pyrolysis and soot formation.

(2) Also, the presence of the combustion products from the first element around the second will reduce the oxygen mole fraction.

(3) Raising the inital temperature of the reactants will raise the flame temperature of the second fuel element. This effect may be offset by the dilution effect which reduces flame temperatures. Heating cannot occur without dilution.

(4) Heat will be transferred to the coolant prior to the addition of the staged air, thus smoothing the heat release profile in the combustor.

Wilson et al.[11] have shown from a simplified analysis of NO formation in a diffusion flame that the second effect described above dominates, thus lowering the NO formation rate.

C. AXIAL BOOM INJECTION RESULTS

During these tests the rear boom was located at various distances from the burner and the level of staged air varied from 10% to 30% (105% to 82% theoretical air at the burner). The results of these tests are shown in Figure 26. The optimum NO reduction (20%) was attained with 30% of the air staged and with the boom located 2.2 combustor diameters from the oil nozzle. For this condition, smoke increases from 4 to 8 BSN. In fact, the flame turned completely opaque when the boom was moved farther than 2.2 combustor diameters from the nozzle. Figure 26 also shows that delivering the axial air flow closer than 1.5 combustor diameters from the nozzle results in increased NO emissions, presumably because of intensified mixing or higher effective primary air. The unmodified combustor appears to be naturally staged. Most of the NO_x reduction is attained with the first 10% of staged air; beyond 10%, further reductions in NO are very small. Since the burner is at 105% theoretical air at 10% staging, the NO_x reduction is certainly not to be understood in terms of the premixed concept of staging. More likely the mechanism is a shift in the flowfield pattern, or in the drop burning histories.

With 20% rear boom staging, NO and soot were found insensitive to both primary-to-secondary air ratios and also to levels of swirl at the burner, as shown in Figure 27. The observed insensitivity to primary/secondary air distribution is quite different from the no staging case, and was found to persist regardless of boom position. As shown in Figure 28 the smoke emissions under staging were lowered by 2 BSN when the P/S ratio was shifted to 20-80.

D. SIDEWALL STAGING RESULTS--EFFECT OF INJECTION POINT AND ANGLE

Sidewall injectors were used in a series of experiments to investigate not only the effect of the axial location of the staged air delivery (in a manner similar to the rear boom), but also the effect of orientation of the staged air flow angle. The individual injectors are designed such that the air leaves the injector at an angle from the injector axis--see Figure 29.

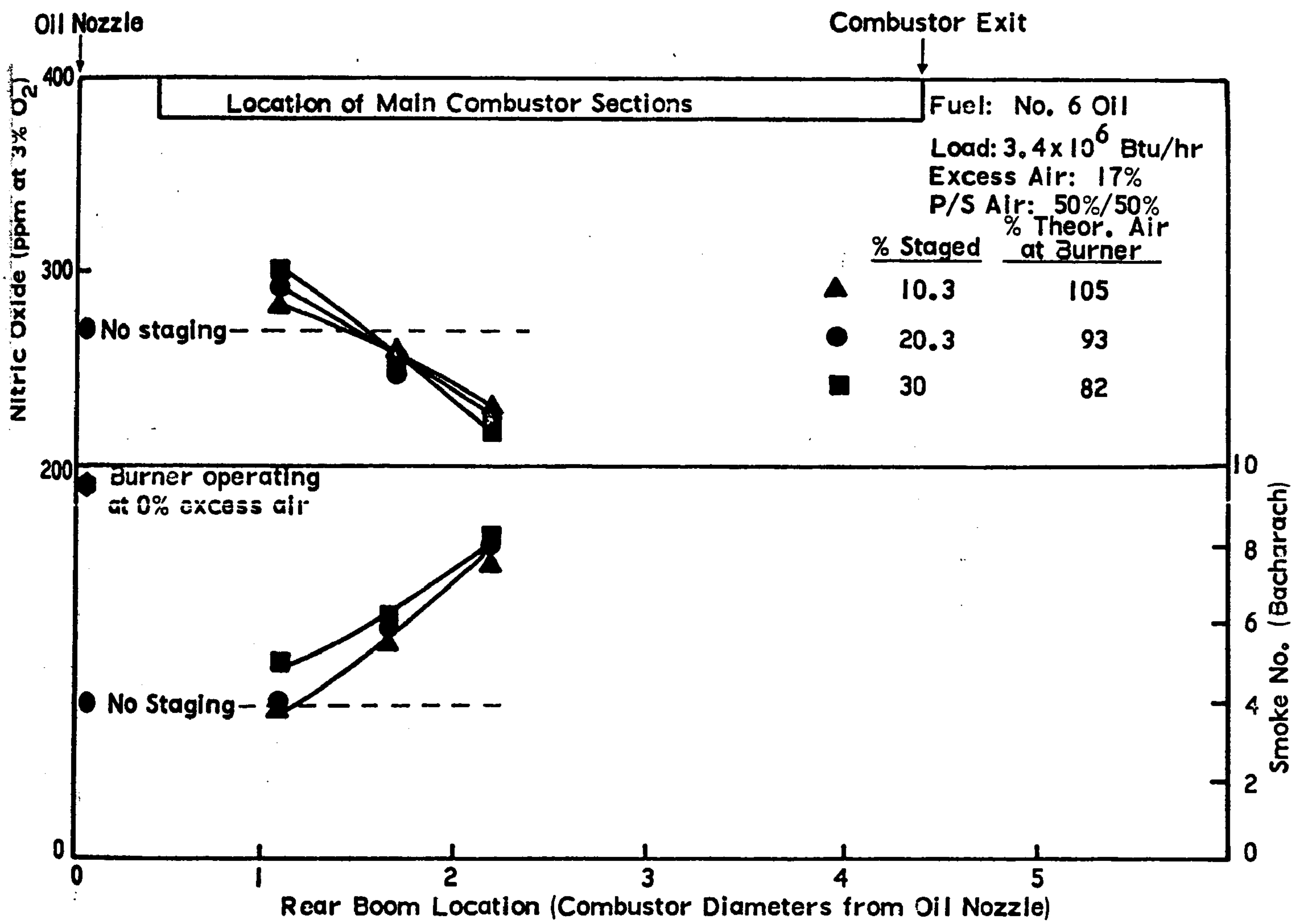


Figure 26

EFFECT OF REAR BOOM POSITION AND LEVEL
OF STAGING ON EMISSIONS

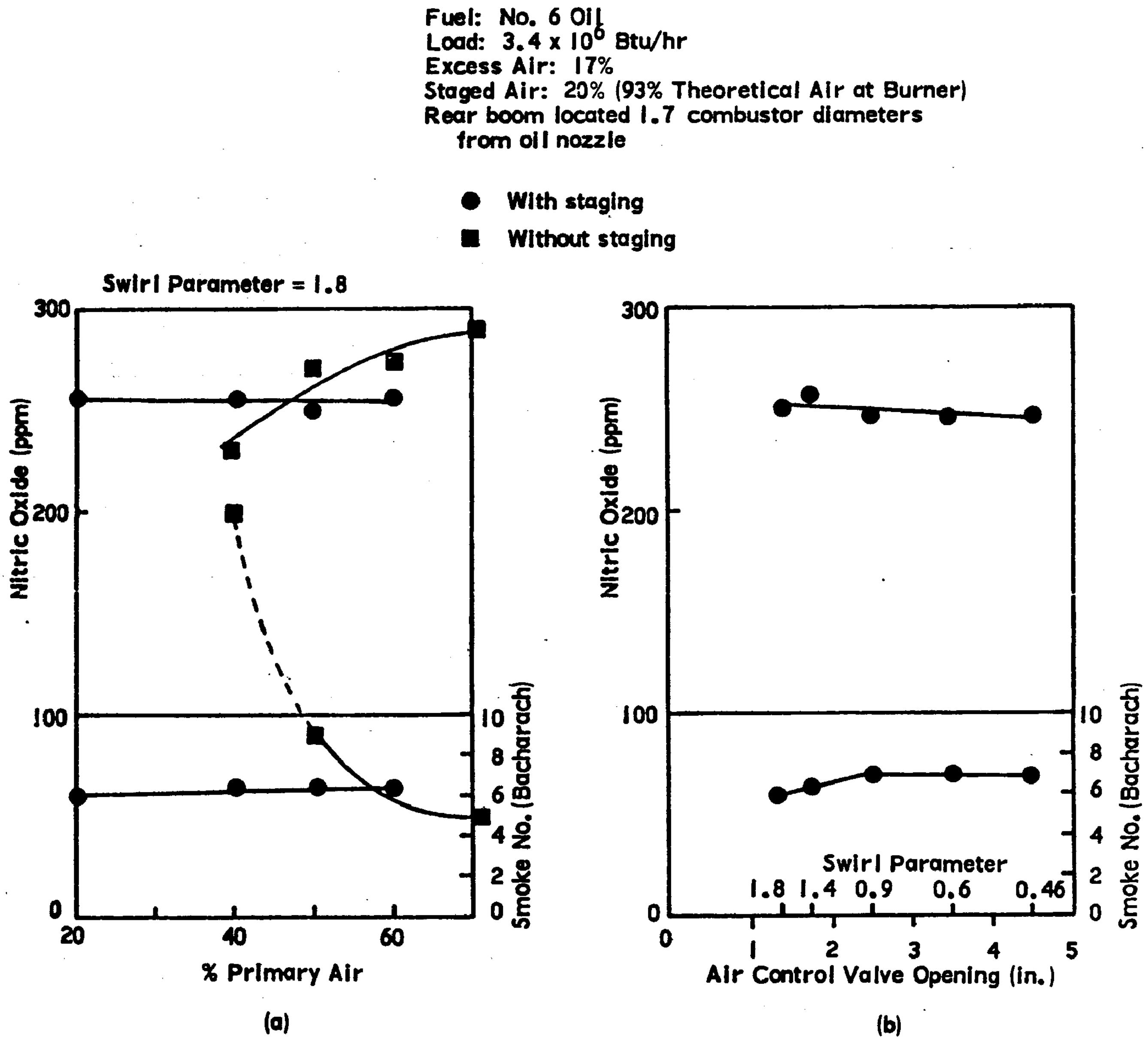


Figure 27

EFFECT OF P/S RATIO AND SWIRL ON EMISSIONS DURING STAGING

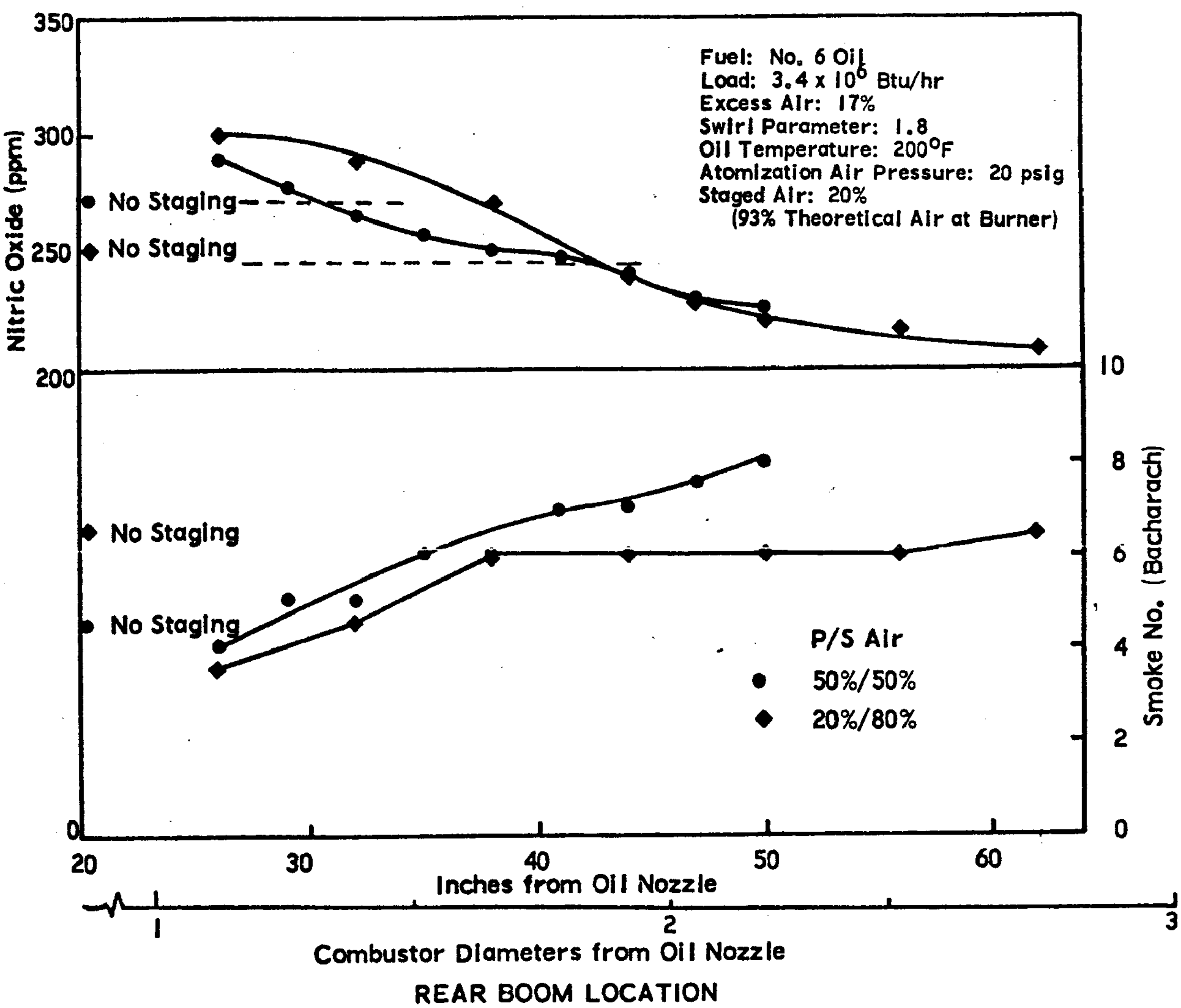


Figure 28

EFFECT OF PRIMARY/SECONDARY AIR RATIO
ON EMISSIONS

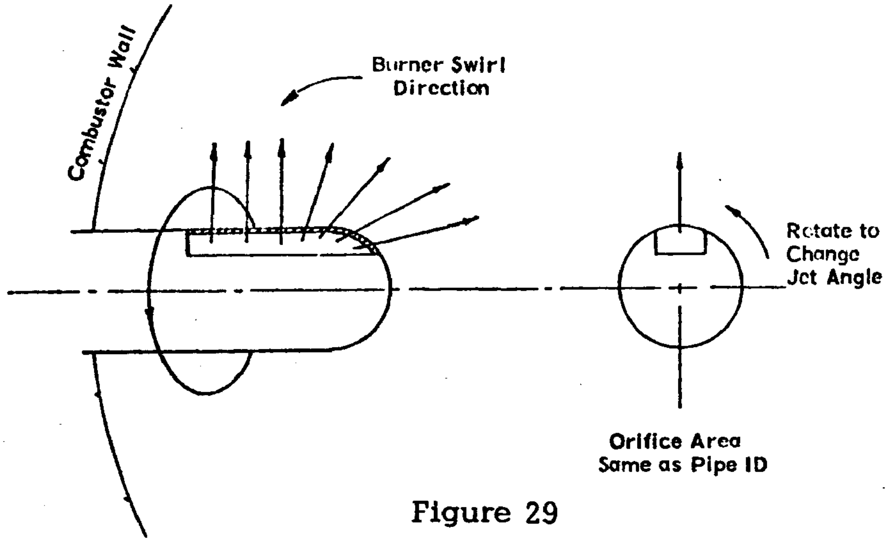


Figure 29

CUTAWAY SKETCH OF INJECTOR TIP

Four orientations relative to the burner flows were tested:

- Co-swirl (in the direction of the burner air swirl)
- Counterswirl (against burner air swirl)
- Upstream (towards burner)
- Downstream (away from burner)

When air was delivered through the first set of injectors (S_1) which are located 20-3/8" (0.9 combustor diameters) from the oil nozzle, it was found that the NO emissions were greater than the case of no staging. This result corroborates previous evidence that the primary mixing field of this burner is readily intensified by air delivered within one combustor diameter.

The second (S_2) and third (S_3) set of injectors were then tested for their effectiveness in adding the staged air, as shown in Figure 30. NO_x reductions were smoke limited to about 24%, with S_3 slightly more effective than S_2. With the S_2 and S_3 injectors, the downstream orientation could not be used due to excessive smoke emissions. In addition, tests could not be performed with more than 17% of the air staged through the S_3 injectors in the co-swirl or the counterswirl orientations due to excessive smoke formation.

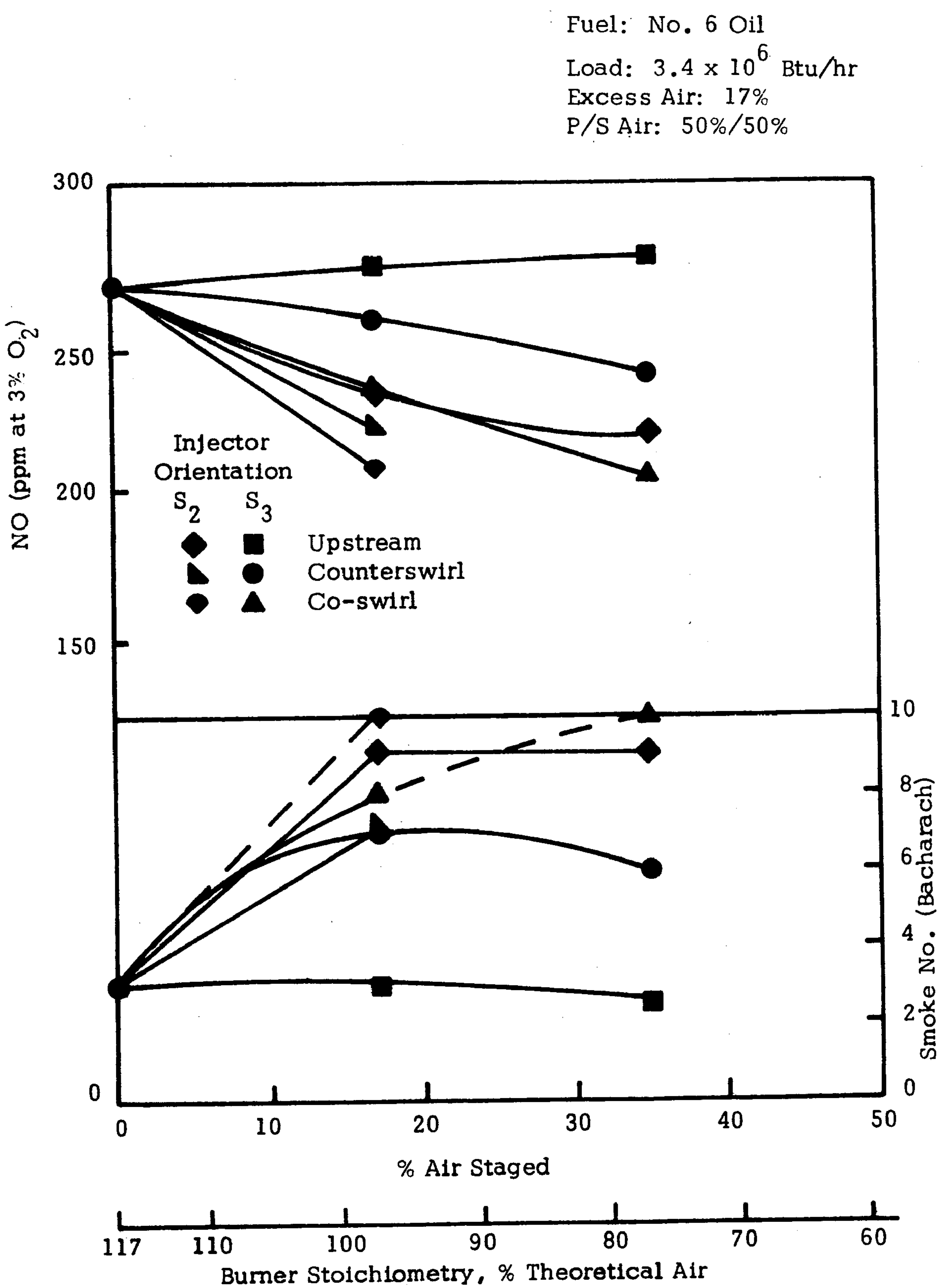


Figure 30

EFFECT OF S_2 AND S_3 INJECTOR ORIENTATION

Various combinations of S_2 and S_3 were also tested with results similar to the tests using the individual injectors. Limited testing was also performed at reduced load (2×10^6 BTU/hr) using the S_2 and S_3 injectors; however, the results were less than promising, with reductions in NO of only 14% before smoke emissions exceeded a 9 Bacharach smoke number.

E. SIDEWALL STAGING RESULTS--HIGH EXCESS AIR

Similar tests were performed with the burner operated at 3.4×10^6 Btu/hr but with 35% overall excess air. These results are reported in Figure 31.

The results showed that:

- The S_3 injectors are more effective in reducing NO emission than the S_2, presumably because they are farther downstream, thus increasing the probability of late-burning drops burning in the products of earlier drops.
- The most effective orientation in terms of NO reductions was when the air was injected in the same direction of the burner swirl. This orientation minimizes the relative velocity between the combustion products and staged air and thus maximizes the staging effect.
- The reductions in NO were attained only by accepting increased smoke. Typically when the NO emissions were reduced to 200 $\pm$ 5 ppm (approximately 25% reduction), the smoke emissions were greater than a 9+ BSN (off scale!).

F. SIDEWALL STAGING RESULTS--HEAT FLUX DISTRIBUTION

In attempting to analyze these results, it is of value to look at the changes in the thermal profile along the combustor with staging through the rear boom (Figure 32), and with the sidewall injectors S_2 and S_3 (Figures 33 and 34). The results show that with staging the heat flux distribution is not more uniform and the total heat transfer to the coolant was greater. In fact the total heat flux increased as the point of staged air addition was moved farther downstream from the burner. The only characteristic difference between the boom configuration (Figure 32) and sidewall configuration (Figures 33 and 34) was some sensitivity to the orientation of the sidewall injectors. Adding the staged air counter to the direction of swirl from the burner resulted in a greater overall heat flux to the coolant than addition of

Fuel: No. 6 Oil
Load: 3.4×10^6 Btu/hr
Excess Air: 35%
P/S Air: 50%/50%

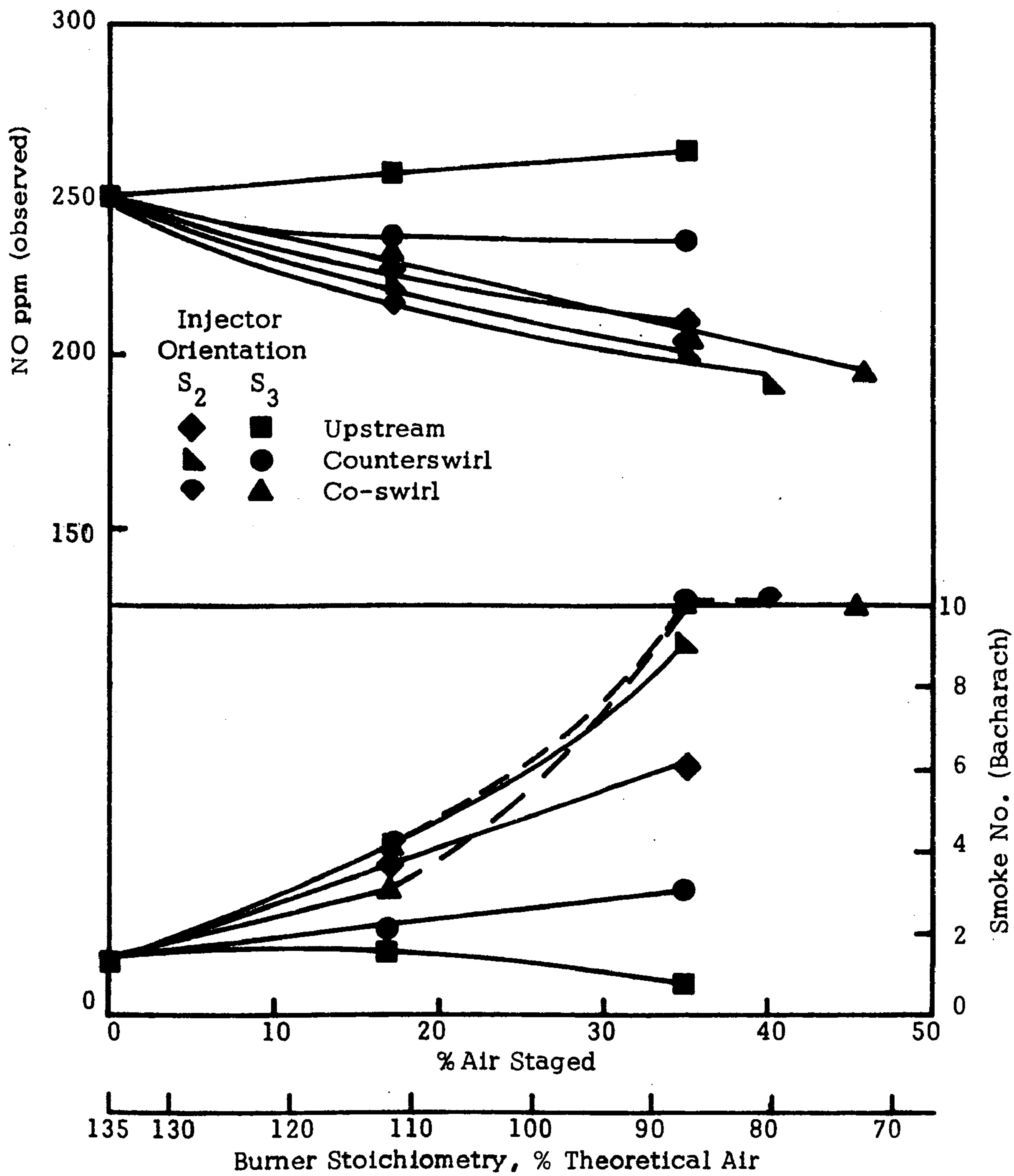


Figure 31

EFFECT OF S_2 AND S_3 INJECTOR ORIENTATION ON EMISSIONS (35% EXCESS AIR)

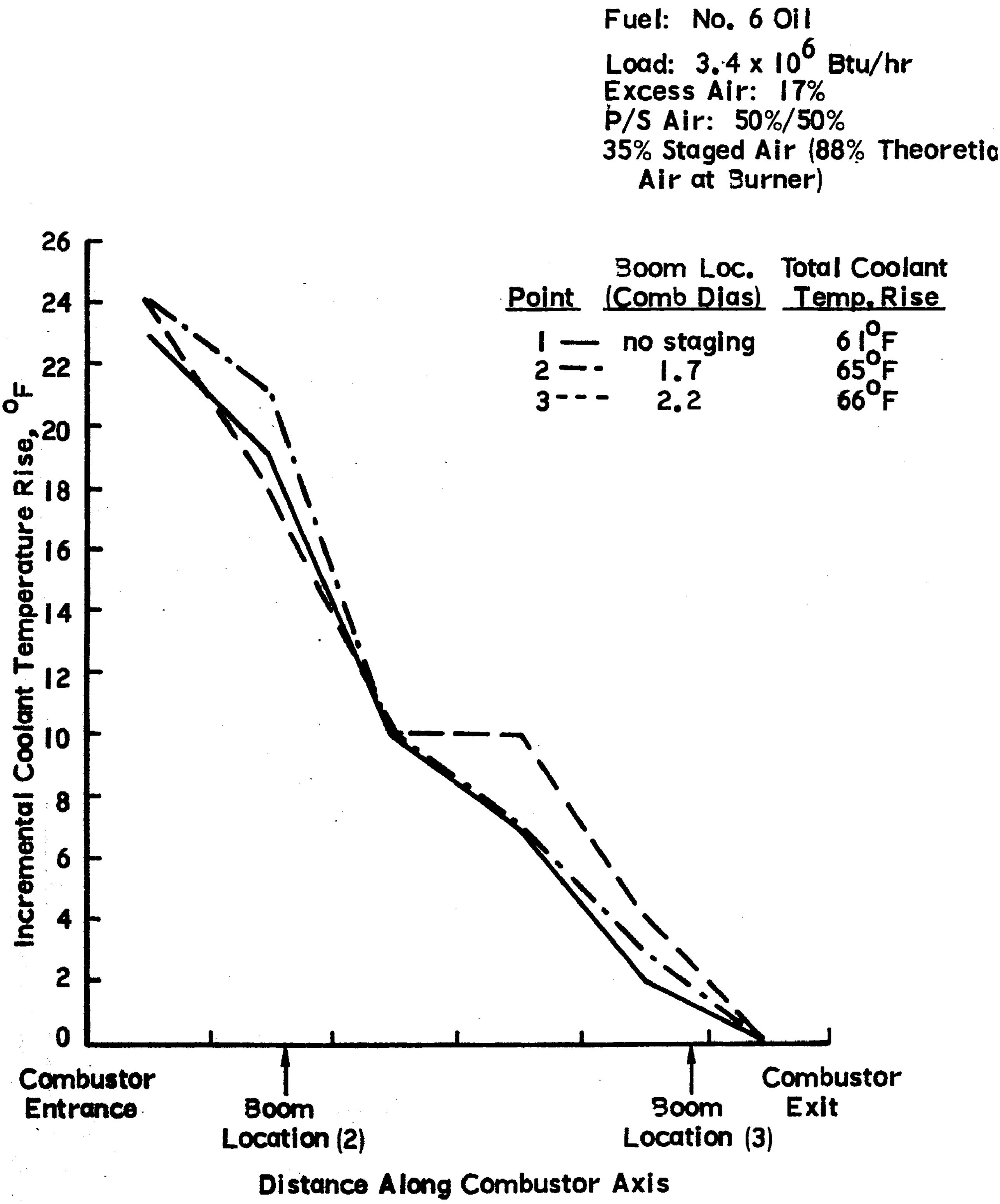


Figure 32

EFFECT OF BOOM LOCATION ON
COMBUSTOR HEAT TRANSFER DISTRIBUTION

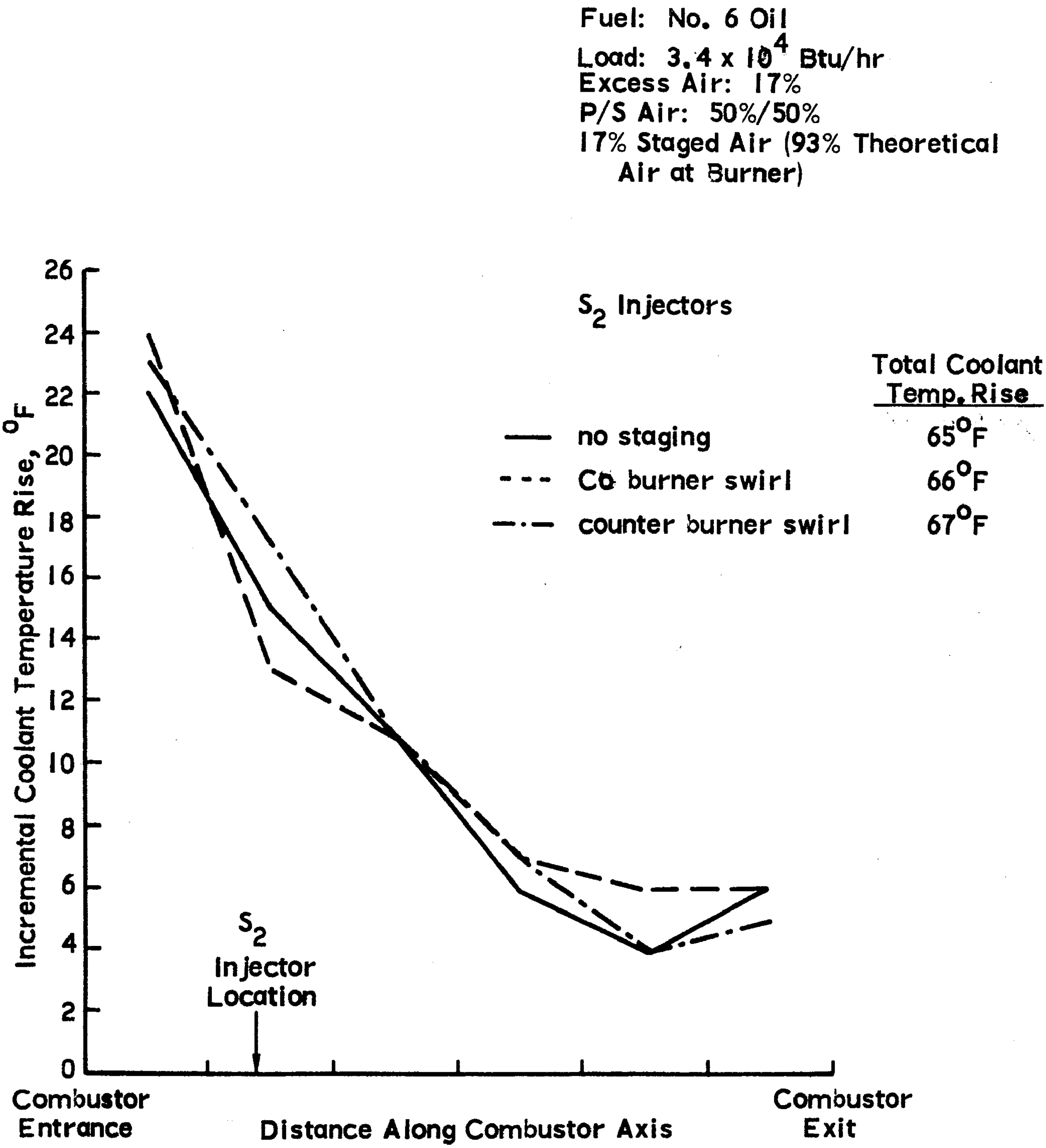


Figure 33

EFFECT OF S_2 INJECTORS ON
COMBUSTOR HEAT TRANSFER DISTRIBUTION

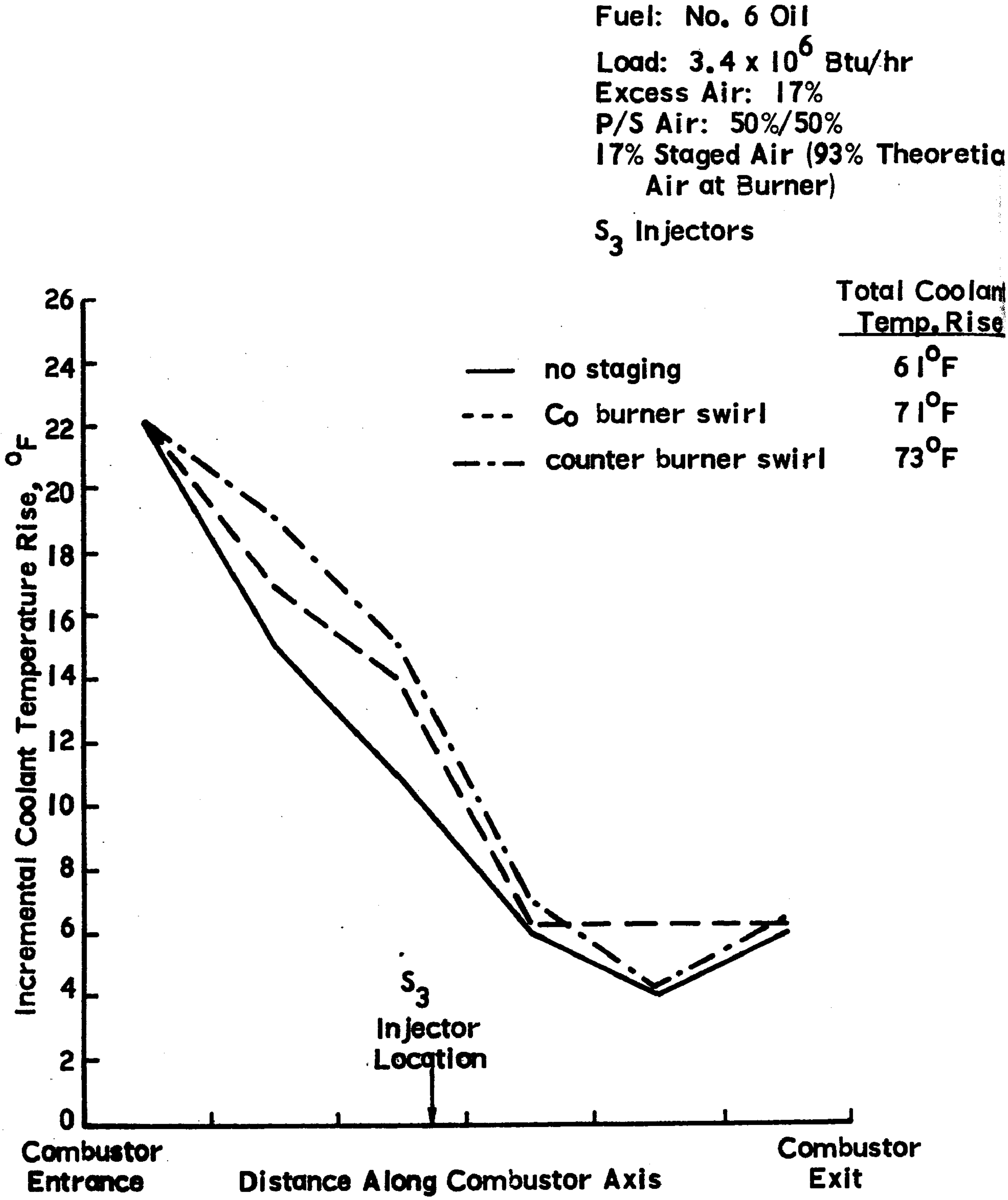


Figure 34

EFFECT OF S_3 INJECTORS ON
COMBUSTOR HEAT TRANSFER DISTRIBUTION

the staged air in the direction of the burner swirl.

One might have expected the opposite result if staged air when injected participated immediately in combustion. Compared to the no-staging case, the rich region upstream of the staged air addition should release less heat with more heat subsequently released downstream of the staged air addition. Staging could conceivably smooth the heat distribution along the boiler, and by "clipping" temperature peaks, reduce the total heat loss. The observed increase in heat flux although unexpected may have at least three plausible explanations: (1) staging may replace the cool, secondary air near the wall by gas which is hot and more turbulent, (2) the fuel spray cone may deliver the combustion zone closer to the walls due to the reduced burner air, or (3) an increase in radiation in the first section of the combustor due to the rich combustion with increased soot formation.

VIII. EFFECTS OF COMBINED FGR AND STAGED COMBUSTION

Tests were conducted with 24% of the total air added through the rear boom (89% theoretical air at the burner) and 25% of the flue gas recirculated through the burner. This arrangement essentially kept the volumetric flowrates through the burner within 5% of the flowrates under conditions of no staging. The results of those tests, shown in Figure 35, show that the NO emissions could be reduced 45% to 150 ppm with smoke up only 2 BSN above baseline. Furthermore, the rear boom could be operated at much greater distances from the burner without producing excessive amounts of smoke as occurred with staging alone.

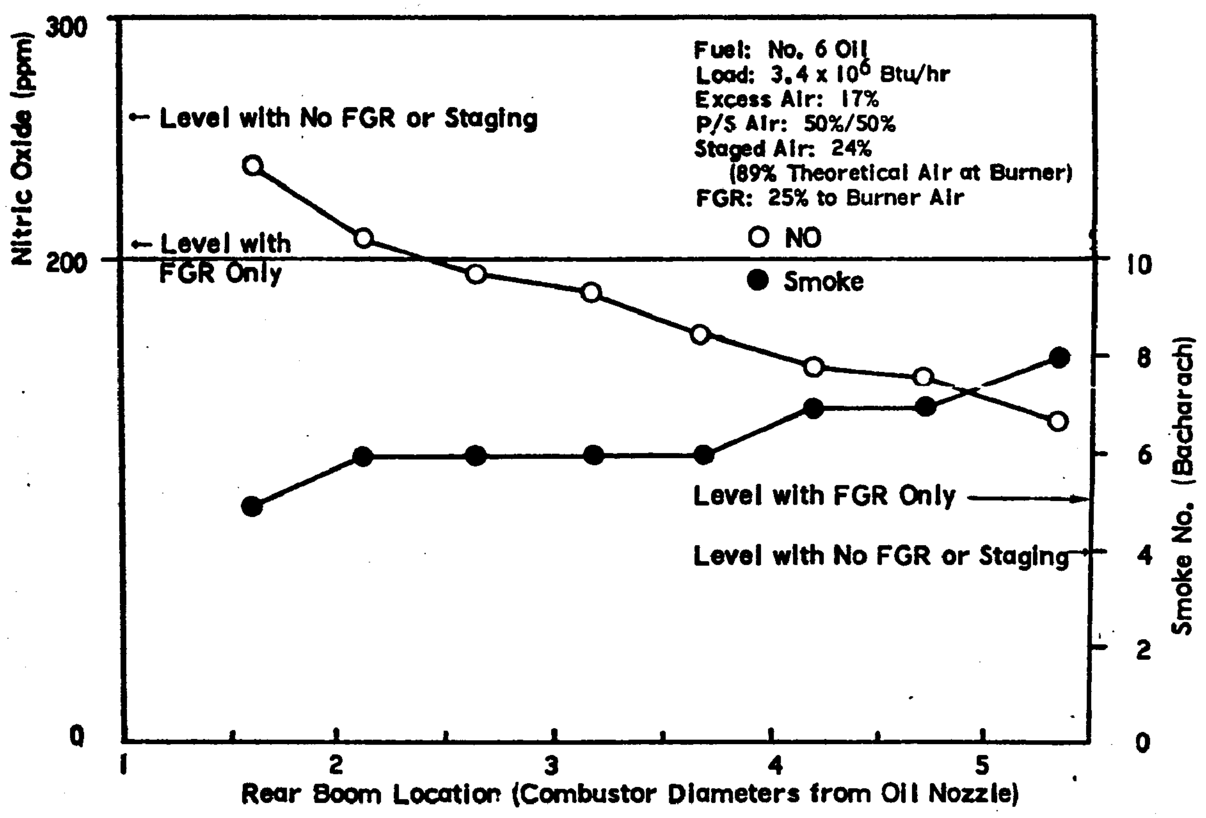


Figure 35

EFFECT OF REAR BOOM LOCATION
ON EMISSIONS DURING COMBINED FGR AND STAGING

Recall that FGR alone gave 30% reduction in NO and staging alone gave 20% reduction (high smoke with staging). The combination gives 45% reduction in NO without smoke penalty. It is possible that the rich smoke-producing regions normally formed with staged air were dispersed by the FGR added through the burner. Another possibility is that the suppression of smoke may have been the result of a chemical effect with the presence of CO_2 or H_2O in the gas [e.g., $CO_2 + C(soot) \rightarrow 2CO$ or $C + H_2O \rightarrow CO + H_2$]. Other workers[4,5,12,13] have obtained similar results and report that with the addition of FGR, the flame has become less luminous indicating a decrease in the soot concentration in the flame.

IX. EFFECT OF REFRACTORY SLEEVES ON NO AND SMOKE CONTROL

Tests were performed with the combustor lined with varying lengths of refractory sleeve in an effort to raise the mean temperature in selected sections of the combustor, stimulating soot burnup. Some increase in thermal NO was also expected.

The refractory was added by casting 2-inch thick, 15"-long rings and sliding them into the combustor (6 complete rings totally lined the combustor), see Figure 36.

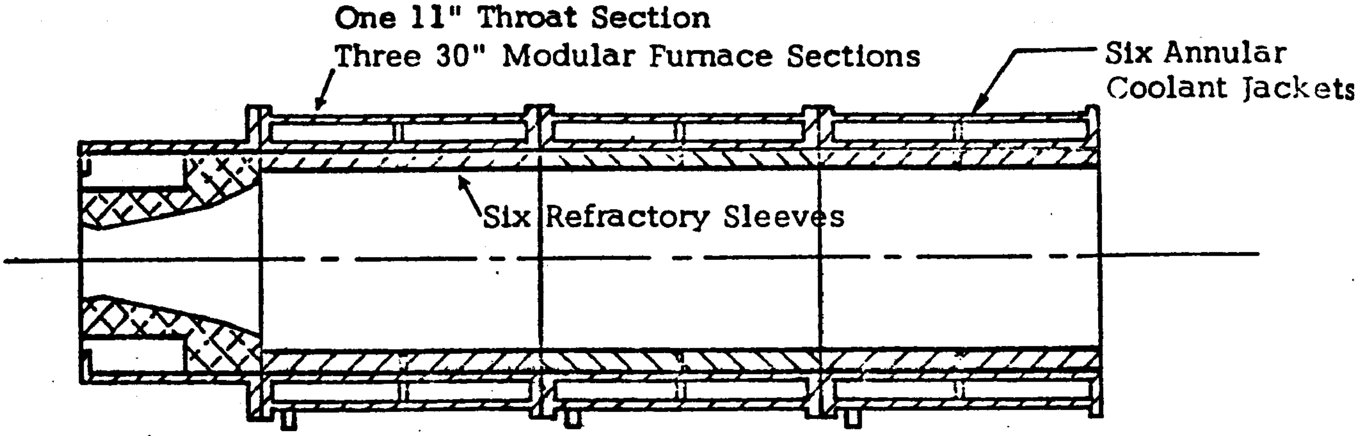


Figure 36

REFRACTORY SLEEVES

Tests were then conducted with pairs of the sleeves located in various 30-inch sections of the combustor. The tests with the refractory liners concentrated on the effectiveness of staged combustion and FGR. During these tests staging was done with the rear injection boom. A summary of the data for staged combustion is presented in Figure 37 for 25% of the air staged through the rear boom. Further data is presented in Figures 38, 39 and 40.* As can be seen in Figure 37, the presence of the refractory raised the baseline NO emissions and lowered the baseline smoke emissions. Surprisingly the most effective location for reducing the baseline smoke emissions was with the center third of the combustor lined with refractory. This may be due to the center refractory raising the temperature of a recirculation zone in the combustor.

*In Figures 37 through 42, the cross-hatched region indicates the section of the combustor that was lined with refractory.

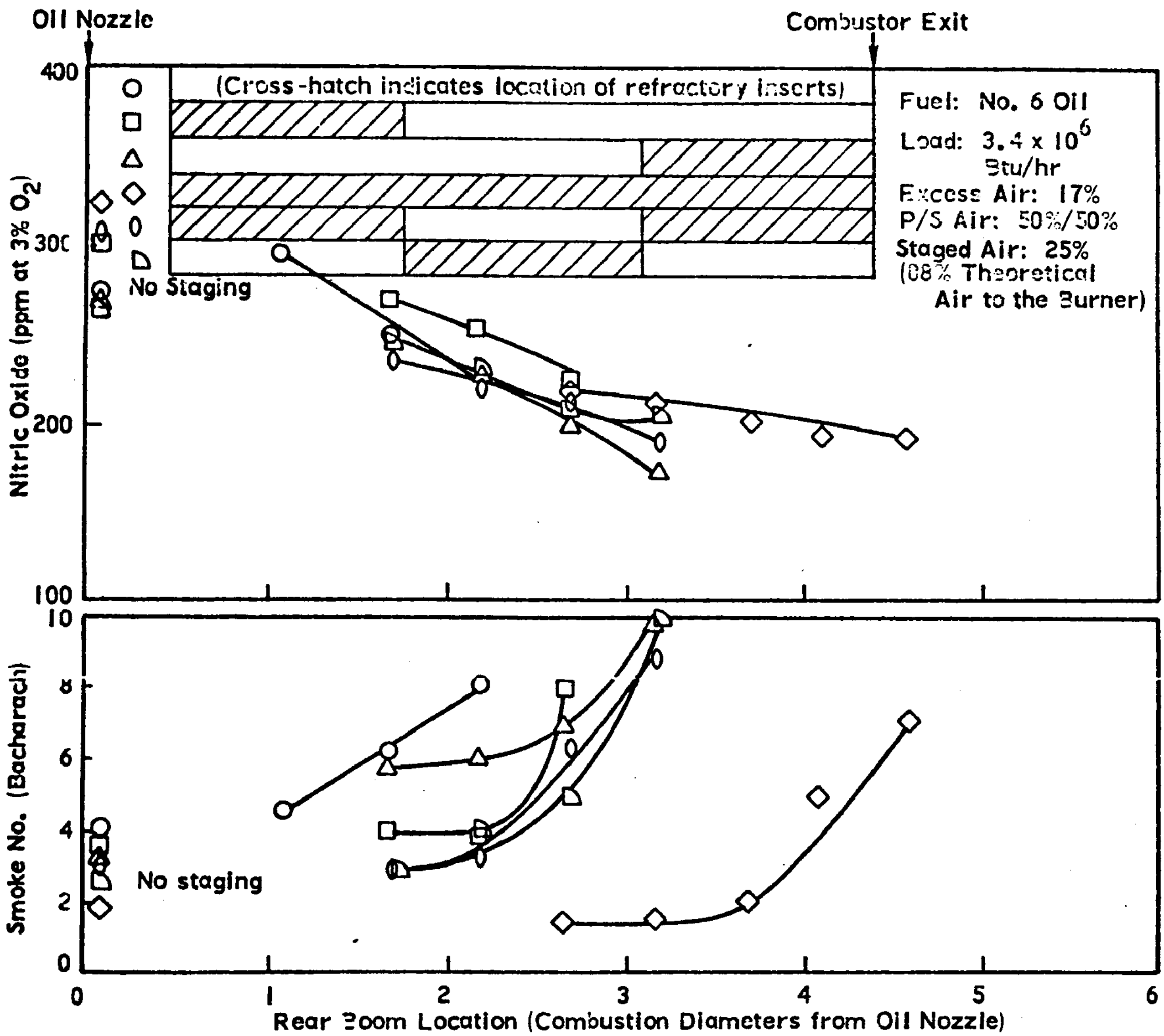


Figure 37

SUMMARY OF THE EFFECT OF REFRACTORY LOCATION ON EMISSIONS

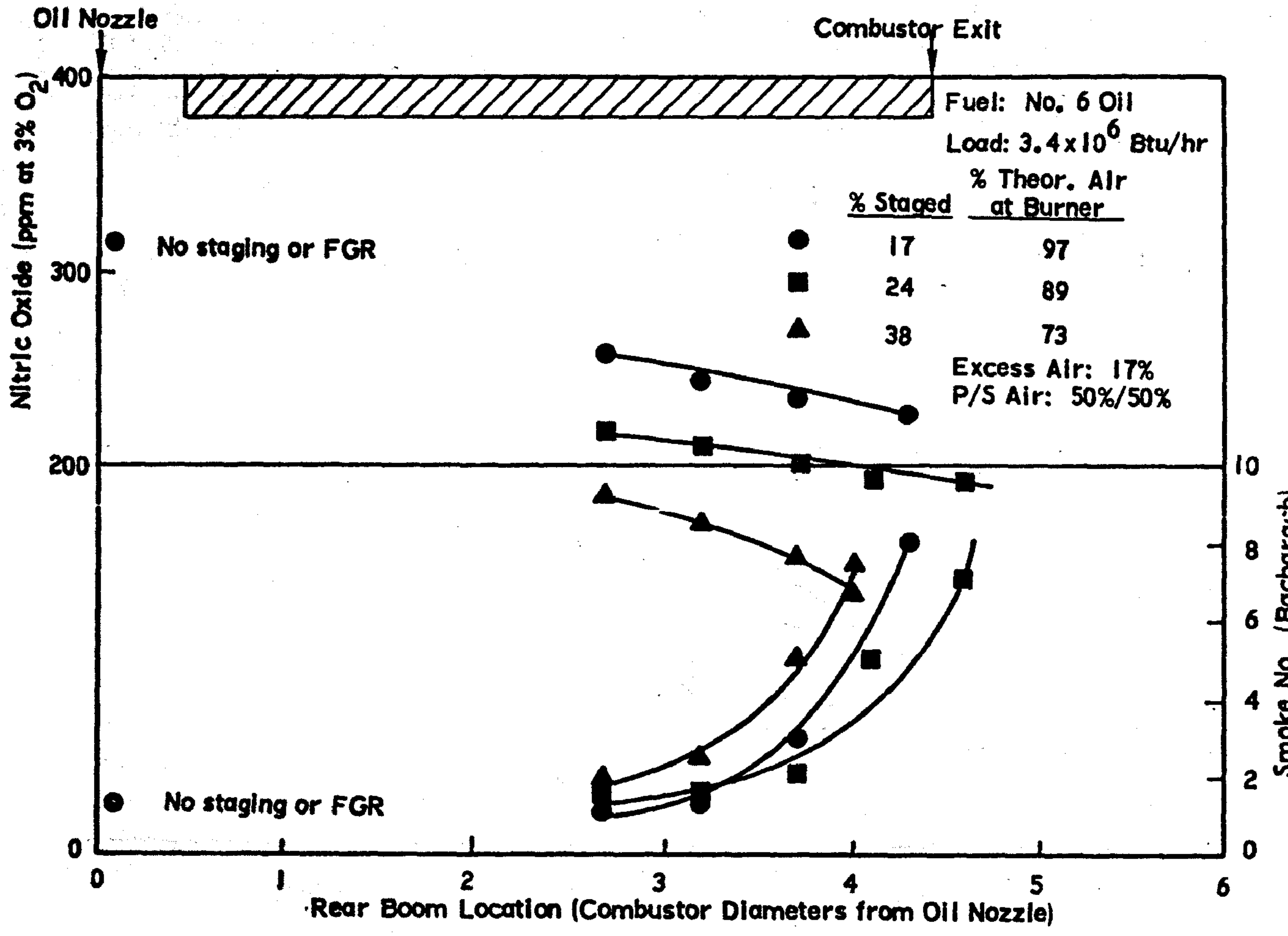


Figure 38

EFFECT OF LEVEL AND LOCATION OF STAGED AIR ADDITION ON EMISSIONS (COMBUSTOR COMPLETELY LINED WITH REFRACTORY)

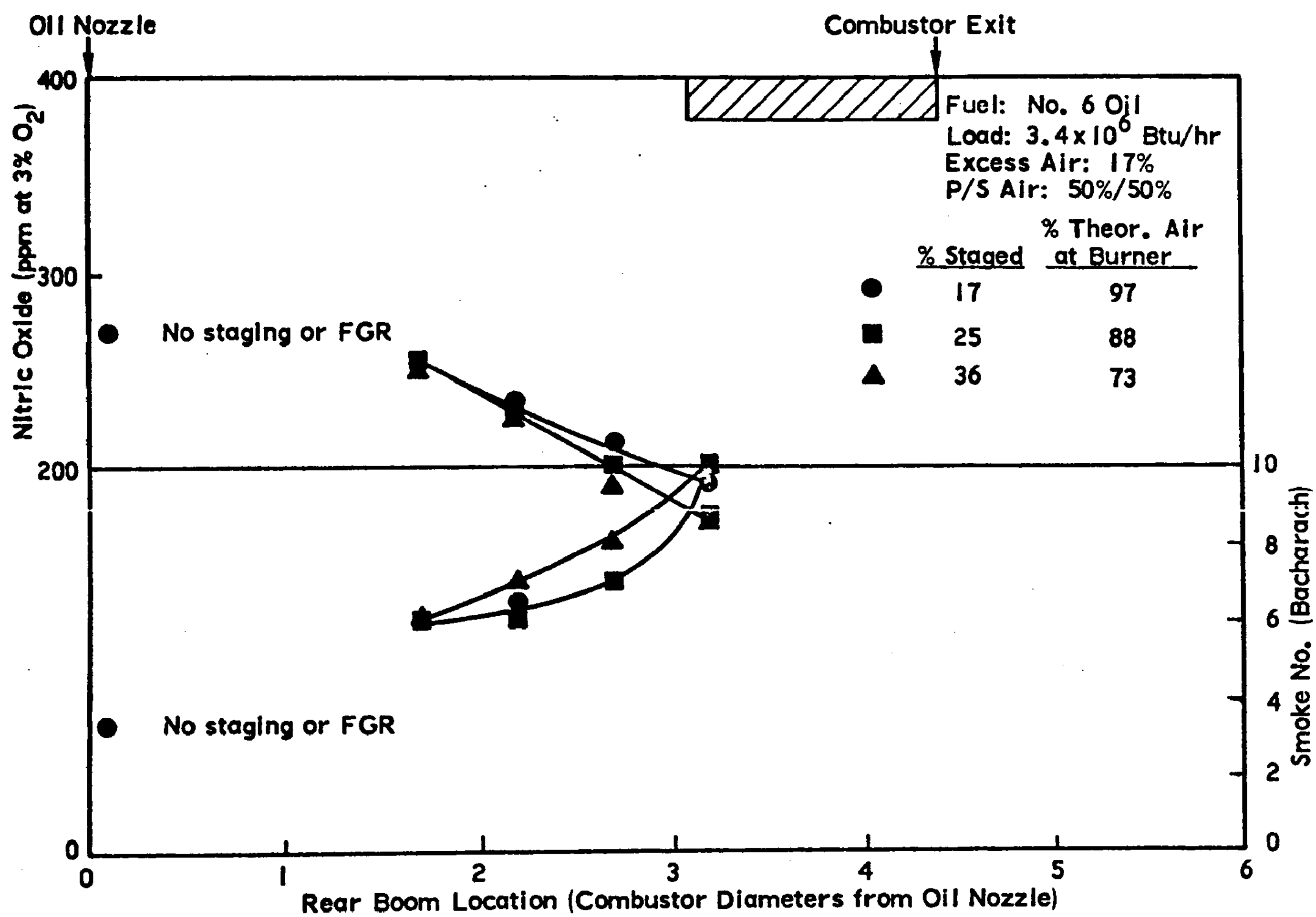


Figure 39

EFFECT OF LEVEL AND LOCATION OF STAGED AIR ADDITION ON EMISSIONS (THIRD COMBUSTOR MODULE REFRACTORY LINED)

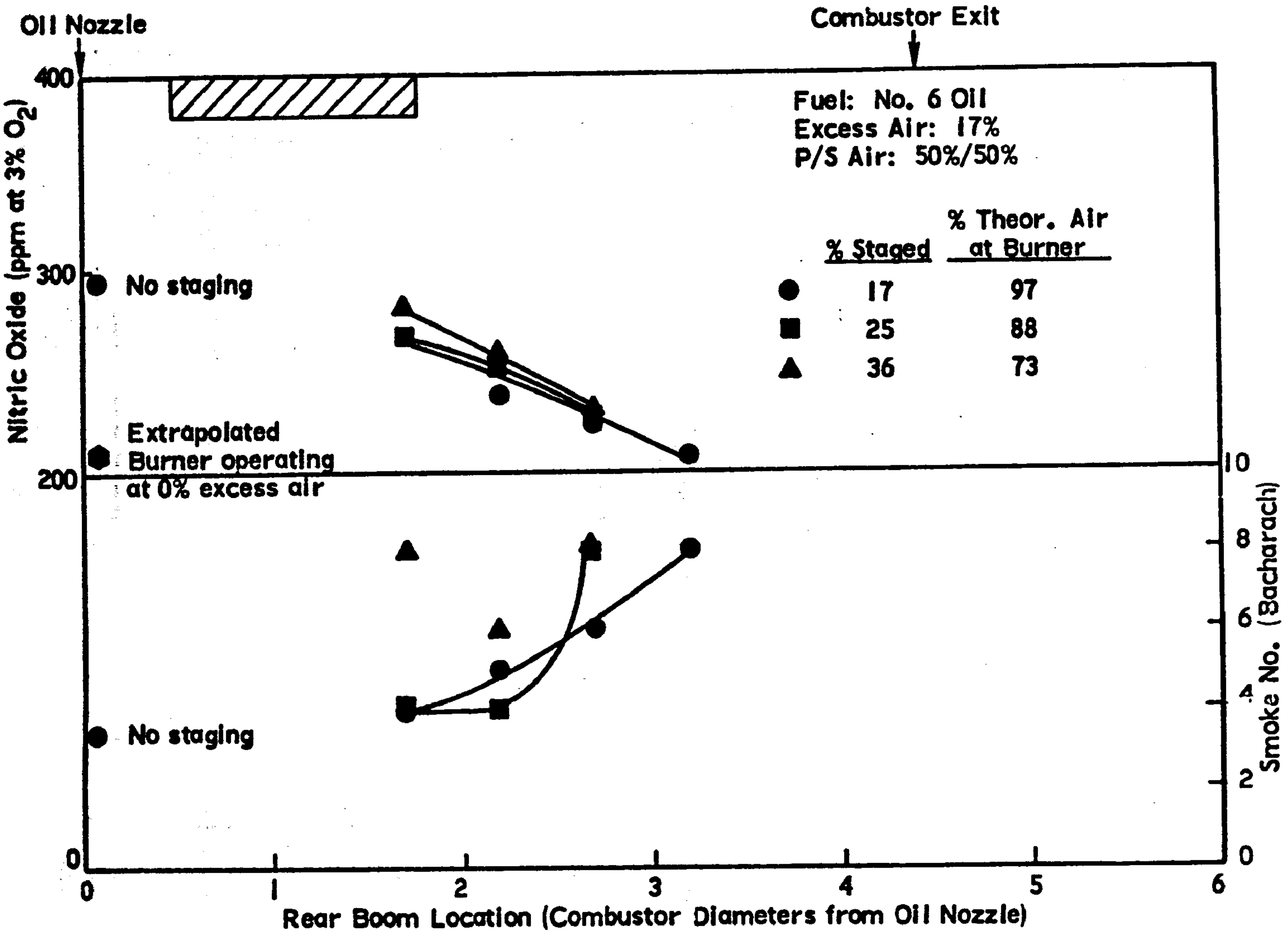


Figure 40

EFFECT OF LEVEL AND LOCATION OF STAGED AIR ADDITION ON EMISSIONS (FIRST COMBUSTOR MODULE REFRACTORY LINED)

It was found that with the partial refractory linings, NO reductions were not very sensitive to the amount of air staged*. In addition, excessive smoke formation limited NO reductions to only 200 ppm as indicated in Table 7.

Table 7

EFFECT OF REFRACTORY POSITION ON MINIMUM NO

Refractory Configuration	NO at Limiting Smoke Level (No. 8 Bacharach)**
No refractory	220
First third	225
Middle third	200
Back third	185
First and last third	195
Full refractory	185

Combined FGR and staged combustion tests were also conducted with two of the partial refractory configurations. The results of these tests are reported in Figures 41 and 42 and compared with the results obtained with staging only. The major difference is again the suppression of smoke emissions from the combustor when flue gas is recirculated; the differences in NO emissions were minor.

*This is not true when the entire combustor is lined with refractory. However, under these conditions, the unit no longer functions as a boiler.

**The limiting smoke level of a No. 8 Bacharach was arbitrarily set, and is high from a practical standpoint. From Figures 41 and 42, it can be seen that if the smoke limit was taken as a No. 6 Bacharach smoke number then the NO levels could only be reduced to approximately 230 ppm, again essentially independent of the level of staged air or the location of the partial refractory.

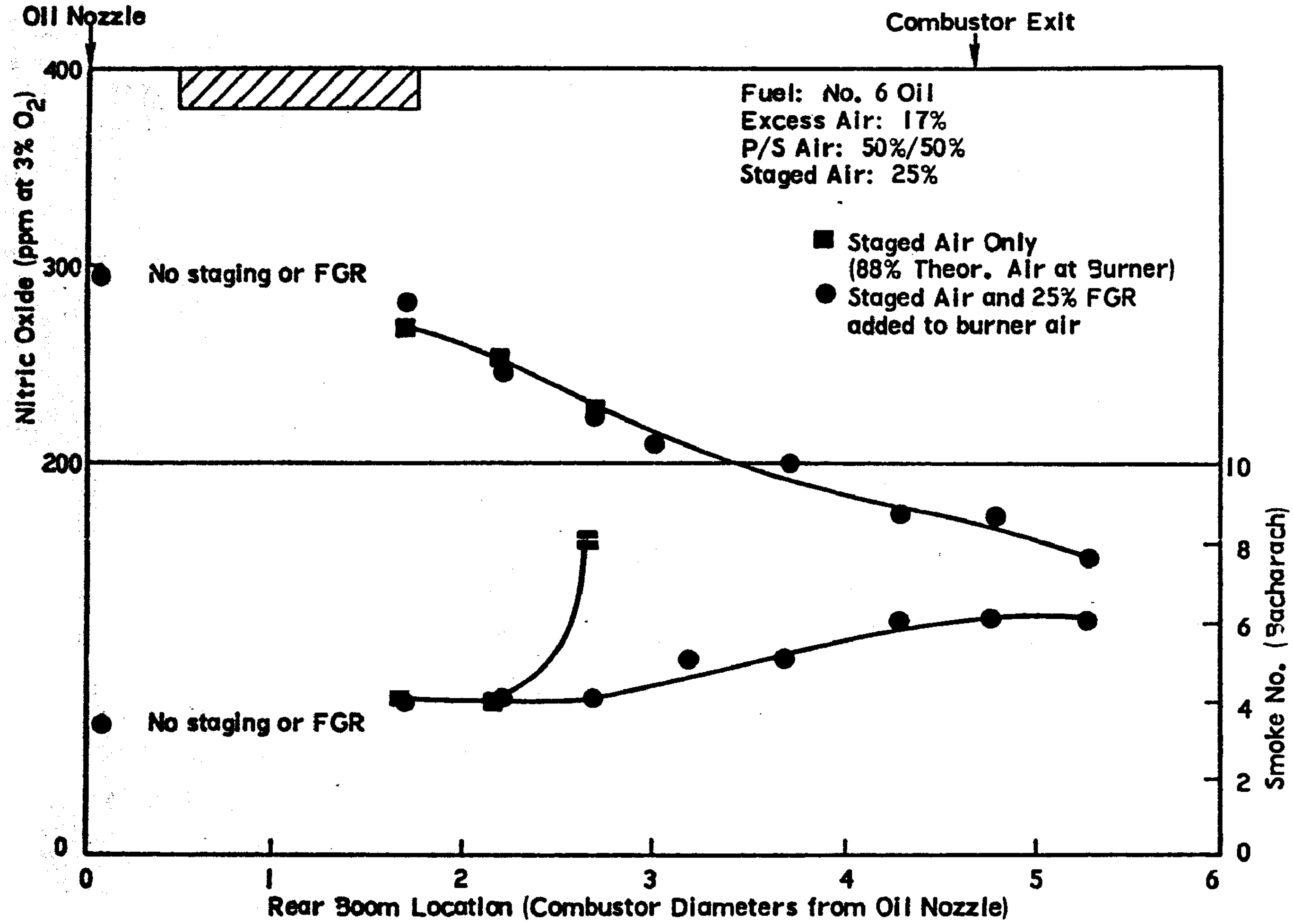


Figure 41

EFFECT OF COMBINED FGR AND STAGING ON EMISSIONS
(FIRST COMBUSTOR MODULE REFRACTORY LINED)

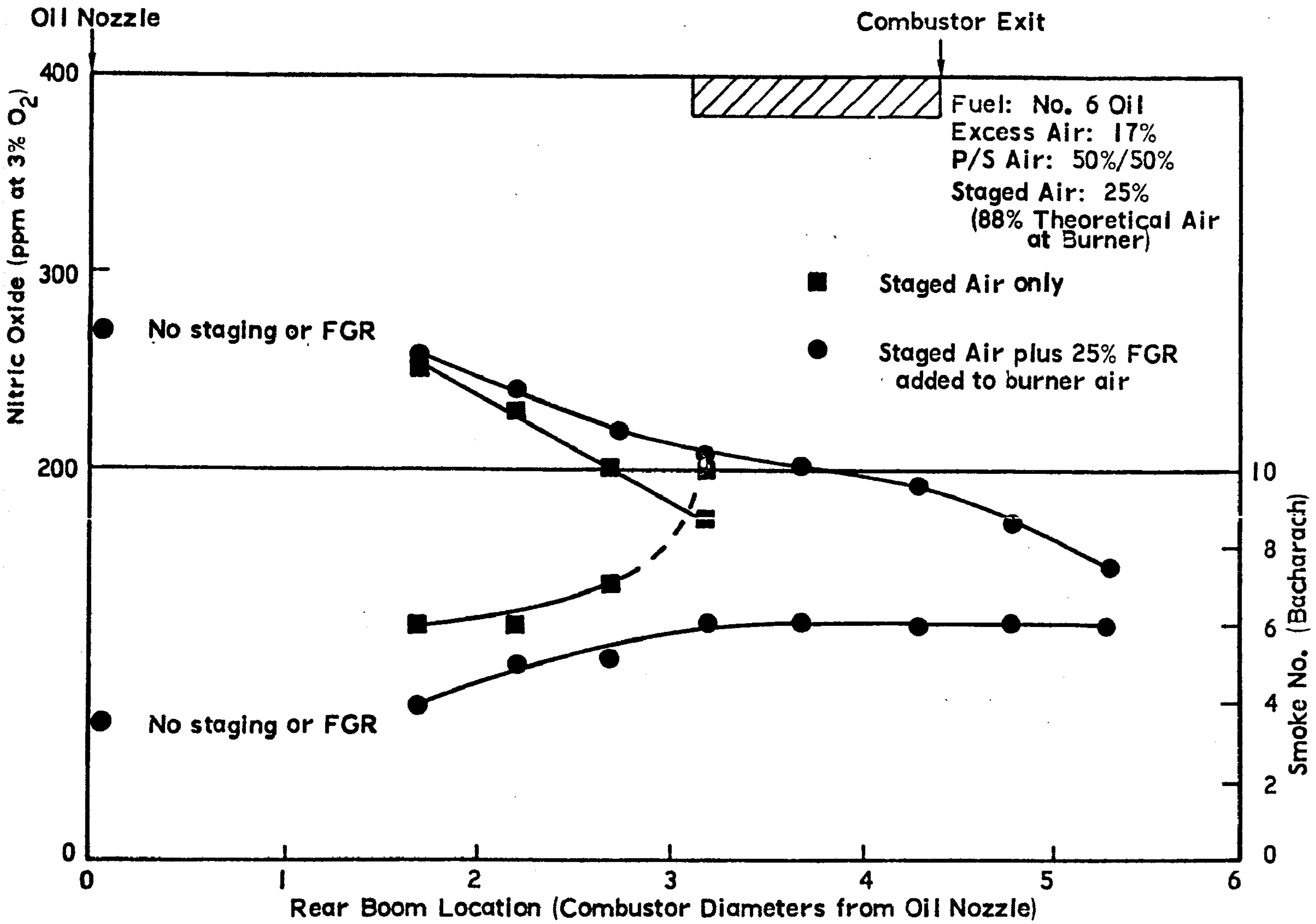


Figure 42

EFFECT OF COMBINED FGR AND STAGING ON EMISSIONS
(THIRD COMBUSTOR MODULE REFRACTORY LINED)

REFERENCES

1. Muzio, L. J. and Wilson, R. P. Jr., "Experimental Combustor for Development of Package Boiler Emission Control Techniques," Phase I of III, EPA Report R2-73-292a, 1973.

2. Heap, M. P. and Lowes, T. M., "Nitric Oxide Production in Large Scale Natural Gas Flames," PR10, EPA Contract No. 68-02-0202, International Flame Research Foundation, December 1972.

3. Wasser, J. H., Berkau, E. E., and Pershing, D. W., "Combustion Intensity Relationship to Air Pollution Emission from a Model Combustor System," EPA, Combustion Research Section, August 1971.

4. Martin, G. B. and Berkau, E. E., "Preliminary Evaluation of Flue Gas Recirculation as a Control Method for Thermal and Fuel Related Nitric Oxide Emissions," presented at WSS/CI Meeting, University of California, Irvine, October 1971.

5. Turner, D. W. and Siegmund, C. W., "Staged Combustion and Flue Gas Recycle: Potential for Minimizing NO_x from Fuel Oil Combustion," presented at The American Flame Research Committee Flame Days, Chicago, Illinois, September 6-7, 1972.

6. Muzio, L. J. and Wilson, R. P. Jr., "A Versatile Combustor for Developing Emission Control Techniques--Reference Report," available from American Petroleum Institute, 1973.

7. Burner Fuel Oils, Mineral Industries Survey, U. S. Department of the Interior, Bureau of Mines, 1972.

8. Barrett, R. E. and Miller, S. E., "Field Investigation of Emissions from Combustion Equipment for Space Heating," API Publication 4180, 1973.

9. Bowman, C. T. and Kesten, A. S., "Kinetic Modeling of Nitric Oxide Formation in Combustion Processes," WSS/CI Paper 71-28, Fall Meeting, Combustion Institute, 1971.

10. Thompson, R. E. and Teixeira, D. P., "The Effectiveness of Combustion Control in Reducing NO_x for Oil Fired Power Plant Boilers," Central States Section/Combustion Institute, Champaign, Illinois, March 1973.

11. Wilson, R. P. Jr., Waldman, C. H. and Muzio, L. J., "Foundation for Modeling NO_x and Smoke Formation in Diesel Flames," Final Report for Phase I, Contract No. EPA 68-01-0436, January 1974.

12. Andrews, R. L. et al., "Effects of Flue Gas Recirculation on Emissions from Heating Oil Combustion," 61st Annual Meeting, Air Pollution Control Association, St. Paul, Minnesota, June 1968.

13. Siegmund, C. W. and Turner, D. W., "NO_x Emissions from Industrial Boilers: Potential Control Methods," ASME Paper 73 1PWR10, 1973.

TABLE OF CONVERSION FACTORS

To Convert		Into	Multiply By
LENGTH	inches	meters	2.54×10^{-2}
	feet	meters	3.048×10^{-1}
	meters	centimeters	1.0×10^{2}
AREA	square inches	square meters	6.452
	square feet	square meters	9.29×10^{-2}
	square meters	square centimeters	1.0×10^{4}
VOLUME	cubic inches	cubic centimeters	1.639×10^{1}
	cubic feet	cubic meters	2.832×10^{-2}
	liters	cubic meters	1.0×10^{-3}
	gallons	liters	3.785
	cubic centimeters	cubic meters	1.0×10^{-6}
MASS	pounds	grams	4.54×10^{2}
	grams	kilograms	1.0×10^{3}
PRESSURE	pounds/in^2	dynes/cm^2	6.9×10^{4}
	pounds/in^2	feet of water	1.60×10^{-2}
ENERGY	Btu	gram-calories	2.52×10^{2}
POWER	Btu/hr	gram-calories/second	7.0×10^{-2}
	kilowatts	gram-calories/hour	8.6×10^{5}
	horsepower	kilowatts	7.46×10^{-1}
	horsepower (boiler)	Btu/hr	3.35×10^{1}
	horsepower (boiler)	kilowatts	9.803
TEMPERATURE	degrees Farenheit (°F)	degrees Celsius (°C)	$(°F - 32) \times 5.56 \times 10^{-1}$
	degrees Rankine (°R)	degrees Kelvin (°K)	5.56×10^{-1}

TECHNICAL REPORT DATA *(Please read Instructions on the reverse before completing)*		
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16. ABSTRACT The report describes Phase II of a 3-phase program to develop nitrogen oxides (NOx) control techniques for small boilers. A 3.7 million Btu per hour oil combustor was used to investigate burner variations, flue gas recirculation (FGR), staged combustion (SC), and combined FGR/SC while firing No. 6 oil. For burners tested, NOx was most sensitive to primary/secondary (P/S) air ratio and atomization air pressure (20% reductions with reduced P/S air ratio and 20% reductions with increased atomization air pressure). FGR was effective if intimately mixed with the air near the burner. NOx reductions of 30%, with little tradeoff in smoke, were realized by adding FGR through the total and primary air or gas ports. FGR had no effect through the secondary air or quarl injectors. SC was not as attractive as FGR in this burner/combustor system due to increases in smoke emissions. NOx reductions greater than 25% could not be obtained without excessive smoke. Combined 25% SC and 24% FGR resulted in 45% NOx reductions with little increase in smoke. The report recommends the application of the most promising methods to two operating boilers during Phase III.

17. KEY WORDS AND DOCUMENT ANALYSIS

a. DESCRIPTORS		b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
Air Pollution	Fuel Oil	Air Pollution Control	13B, 21D
Boilers	Oxygen	Stationary Sources	13A
Combustion	Smoke	Package Boilers	21B
Emission	Test Equipment	Flame Modification	14B
Nitrogen Oxides		Flue Gas Recirculation	07B
Nitrogen Oxide (NO)		Staged Combustion	
		No. 6 Oil	

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