

**ADVANCED OIL BURNER FOR RESIDENTIAL HEATING--
DEVELOPMENT REPORT**

Thomas A. Butcher

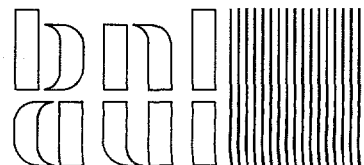
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Energy Efficiency and Conservation Division

DEPARTMENT OF APPLIED SCIENCE

**BROOKHAVEN NATIONAL LABORATORY
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Mr. Len Fisher, consultant to BNL and former Vice-President of Engineering at The Carlin Company has had a major role in this project. Len designed and built the burner head components for the "commercial" versions of the burner prototype which uses the low-pressure air atomizing nozzle. BNL Engineers Yusuf Celebi and George Wei were responsible for all combustion testing, component testing, and the construction of the prototype burners.

ABSTRACT

The development of advanced oil burner concepts has long been a part of Brookhaven National Laboratory's (BNL) oil heat research program. Generally, goals of this work include: increased system efficiency, reduced emissions of soot and NO_x , and the practical extension of the firing rate range of current burners to lower input rates. The report describes the results of a project at BNL aimed at the development of air atomized burners. Two concepts are discussed. The first is an air atomizer which uses air supplied at pressures ranging from 10 to 20 psi and requiring the integration of an air compressor in the system. The second, more novel, approach involves the use of a low-pressure air atomizing nozzle which requires only 8-14 inches of water air pressure for fuel atomization. This second approach requires the use of a fan in the burner instead of a compressor although the fan pressure is higher than with conventional, pressure atomized retention head burners.

In testing the first concept, high pressure air atomization, a conventional retention head burner was modified to accept the new nozzle. In addition, the burner head was modified to reduce the flow area to maintain roughly 1 inch of water pressure drop across the head at a firing rate of 0.25 gallons of oil per hour. The burner ignited easily and could be operated at low excess air levels without smoke. The major disadvantage of this burner approach is the need for the air compressor as part of the system. In evaluating options, a vane-type compressor was selected although the use of a compressor of this type will lead to increased burner maintenance requirements.

In the second concept evaluated, the low pressure atomizer, the nozzle air flow is much higher as a percentage of the total burner air flow but the pressure is much lower. Several different concepts for the design of a burner head using this atomizer were evaluated. In the final design developed during this project all of the burner combustion air is supplied into the burner air tube from a single, high speed fan. Inside the burner this air is divided into primary (atomizer) air, secondary air which flows through a retention ring, and tertiary air which enters the outer annular area of the flame zone.

In developing a system for the burner an independent, low pressure fuel pump was used driven by a small DC motor. Fuel flow metering was accomplished using a pump, pressure regulator set at 10 psi, and a needle valve. The combination of a small accumulator and time delay solenoid valve were used to improve low firing rate ignition. The burner uses a conventional, interrupted ignitor and a conventional burner primary control.

Combustion tests were done with the prototype burner in a wide variety of applications at firing rates from 0.3 gallons of oil per hour to 1.0 gallons of oil per hour. Performance was found to be very good with NO_x emission levels in the 60 to 70 ppm range and CO generally under 40 ppm and zero smoke number at excess air levels as low as 7%. Electric power consumption of the prototype burner is low -less than 100 watts as compared to over 200 watts with a conventional burner.

A variety of potential applications are seen which can take advantage of the burner. The low firing rate capability of the burner can lead to new products such as low input, high efficiency boilers and furnaces. Two-stage or fully modulating versions of the burner can be used to increase the efficiency of existing equipment without sacrificing capacity. The low excess air operation capability of the burner leads to reduced rates of boiler scale formation - a major factor in efficiency degradation over time.

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1. INTRODUCTION

The residential oil burner market is currently dominated by the pressure-atomized, retention head burner. In these burners oil is delivered to a fuel nozzle at pressures from 100 to 150 psi. In addition to atomizing the fuel the small, carefully controlled size of the nozzle exit orifice serves to control burner firing rate. Burners of this type are currently available at firing rates over 0.5 gph. Nozzles have been made which are sized for lower firing rates but experience has shown that such nozzles suffer rapid fouling of the small passages leading to poor spray patterns and bad combustion performance. Two factors contribute to this fouling. The first is fuel system dirt which might be controlled through better filtration. The second is coke formation on internal passages occurring after normal burner shutdowns when the nozzle is heated by radiation from the combustion chamber. This period after shutdown is more severe than the time when the burner is actually firing because of the cooling effect of the combustion air flow over the nozzle during the on-period.

The pressure-atomized, retention head burner has an excellent reputation for reliability and efficiency. While it is only correct to discuss the efficiency of complete heating systems rather than the efficiency of the burner alone the burner has a very strong influence on system efficiency in several important ways. To achieve high efficiency the burner should be capable of operating with a minimum of excess air. Smoke production in steady state is the factor which sets the lower limit on excess air. Modern pressure-atomized burners can operate at excess air levels as low as 15-20% at a firing rate of 1 gallon per hour, under good conditions [1]. At low firing rates higher excess air levels are generally required because of reduced air velocities and mixing intensities. A second way in which burners can influence system efficiency relates to sooting of heat exchanger surfaces and degradation of efficiency over time. Burners which are operating very badly, possibly because of a fouled nozzle for example, may produce high smoke levels leading to rapid coating of heat exchanger surfaces with carbonaceous soot. In more normal cases, where the burner continues to operate smoke-free, fouling rates will be lower. One field study concluded that the average efficiency degradation rate is 2% per year. Studies at Brookhaven National Laboratory (BNL) have shown that a very important part of the normal fouling deposit is iron sulfate scale resulting from the deposition of sulfuric acid from the flue gas onto the heat exchanger surfaces. The amount of sulfuric acid which is produced in a flame is dependent upon the burner excess air level. Acid production and scaling rate can be controlled by using burners which can operate at very low excess air levels [2,3,4].

Another way in which burners can influence heating system efficiency is through off-cycle losses. After the burner has shut off the system continues to lose heat up the chimney. This heat loss rate depends upon the rate of air flow through the unit which, in turn, depends upon the burner design. A burner which has small open passages will allow lower off cycle air flow rates.

The objective of the work described in this report is the development of an advanced, air-atomized burner which can provide new capabilities not currently

available with pressure atomized, retention head burners. Specifically this includes:

- ability to operate at firing rates as low as 0.25 gph;
- ability to operate with very low excess air levels for high steady state efficiency and to minimize formation of sulfuric acid and iron sulfate fouling;
- low emissions of smoke, CO and NOx;
- potential for modulation - either stage firing or continuous modulation.

In addition, of course, any such advanced burner must have production costs which would be sufficiently attractive to allow commercialization.

The modulation objective has been included to take advantage of the efficiency gains which might be obtained by having multiple firing rate capability. At low firing rates the temperature of the flue gas leaving the boiler is low and efficiency is high. With a modulating burner most of the season's heating load could be met with the low firing rate but the system would still have the capacity to meet rare, very high load needs such as extreme cold and pull-up from thermostat setback.

In the past a number of very interesting designs for achieving some or all of the objectives listed above have been developed to varying degrees, some as part of the BNL program. Air atomization, blue flame (recirculating), and prevaporizing burners have received attention. In 1980 a review of prior work in this area was completed by Battelle Columbus Laboratories for BNL [5]. Some of the more recent work in advanced burners has been described in the proceedings of the annual BNL Oil Heat Conferences. In 1990 BNL completed a study in which the emissions performance of both conventional and advanced burners were compared [1]. This study included air atomization and prevaporizing burners. General options for atomizers for advanced burners were reviewed by Krishna et.al. in 1987 [6]. Reasons why none of the advanced burners developed are currently available commercially vary but generally include: high cost, poor reliability, excessive complexity (difficult to service), and others.

In all residential air-atomized burner concepts which have been developed a small compressor was used to provide a small flow of air at 5 to 20 psi to the nozzle for atomization. A conventional fan was also included to provide the remainder of the air needed to complete combustion. This secondary air is delivered at much lower pressure, 2-3 inches of water. As part of the work described in this report some development work was done with a burner of this type. In this case a conventional pressure atomized, retention-head burner was modified to incorporate a "siphon" type air atomizing nozzle and performance tests were done. The nozzle, burner modifications and test results are all described in Section 2 of this report.

Most of the effort during this program has been placed on another air atomization approach which has become termed the Fan-Atomized Burner. Here, fuel oil is atomized with air supplied by the burner's fan. There is only one air supply in this burner - a fan delivering air at 6 to 12 inches of water pressure. Some of this air goes through the nozzle, atomizing the fuel. The remainder passes

around the nozzle providing air into the flame zone to complete combustion. Outstanding performance has been achieved and advantages include: very low excess air requirements, low soot, low NOx emissions, simple, and low sensitivity to back pressure (important for sidewall venting applications).

In Section 3 of this report the atomizer used is described in detail and results of atomization tests (droplet size distributions) are presented. Section 4 describes the results of efforts to develop the burner head for using this nozzle. In Section 5 results of development work aimed at the entire burner system including air supply, fuel pump, fuel metering, and flame safety controls are described. Section 6 presents results of tests of the burner in a wide variety of practical applications. These tests were done using BNL's rapid fouling test system. Section 7 provides a discussion of the results, recommendations for the future, and an analysis of the energy savings potential with the burner. Finally, conclusions from the work are listed in Section 8.

2. DEVELOPMENT OF AN AIR-ATOMIZED BURNER USING A SIPHON-TYPE NOZZLE

Ideally, the objectives of this project would be most easily met by simply replacing the pressure atomizing nozzle in a conventional burner with an air-atomized nozzle. This might enable continued use of many common burner components such as the motor, fan, fuel pump (with reduced discharge pressure), ignition transformer and electrodes, air tube and retention head. It would, of course, be necessary to add an appropriate air compressor and an orifice or other device to control the flow of fuel to the nozzle. In this section results are described of an effort to do this using a commercially available siphon-type nozzle.

The siphon nozzle is illustrated in Figure 2-1. It is normally used with air pressures ranging from 3 to 40 psig. The arrangement of the air/fuel mixing section causes an eductor-effect by which oil can be pulled up to the nozzle from a distance below the nozzle as much as 48". In some applications this eliminates the need for a fuel pump. Of course it is always possible to use a pump and orifice (or other metering device) to feed oil to the atomizer under pressure. The siphon nozzle is used commercially in several applications. In kerosene-fired, construction site space heaters the nozzle is used without a pump to lift fuel from a tank just below the burner. It is also used on burners firing waste oil for disposal and heat recovery. These burners are most often used in automotive service stations. The British company Davair has developed a residential oil-burner using a nozzle of this type although it has not, to date, been openly marketed.

Based on prior studies at BNL [7], the mass mean diameter of drops produced by conventional pressure nozzles is about 40 microns. These drop size measurements were made using a commercial sizer (Malvern Scientific Instruments Corp.) based on Fraunhofer diffraction. In the experimental arrangement atomized spray is collected in a vented chamber which prevents drifting of the smallest drops back into the measurement beam skewing the measured size distribution. Additional experimental details can be found in Reference 7. Figure 2-2 shows the mass-mean diameter of the spray produced by the siphon nozzle over a range of nozzle air fuel ratios. The siphon nozzle is better than the pressure atomized nozzle over the entire range. Typically, for the pressure atomized nozzles the spray mass mean diameter is about 30-40 microns. At the low end of the range shown for air/fuel ratio the atomizing air pressure is low--about 3 psig. At the high end the atomizing air pressure is about 10 psig. Most of the combustion testing was done with air pressures between these limits.

In modifying a conventional burner to use the siphon nozzle it is necessary to remove the normal pressure nozzle, shorten the fuel feed tube (which becomes the compressed air tube) and add an extra tube used to feed oil to the nozzle. In addition, for this test program, some modifications were made to the burner's diffuser to obtain good performance at very low firing rates. Figure 2-3 shows the location of the diffuser in a conventional burner head and the details of both a conventional diffuser and the modified diffuser. The changes were made to maintain a pressure drop of about 1" of water pressure across the diffuser at a firing rate of 0.25 gph. If an unmodified diffuser were used diffuser pressure drop at the low firing rate would be very low, leading to low air velocity, poor

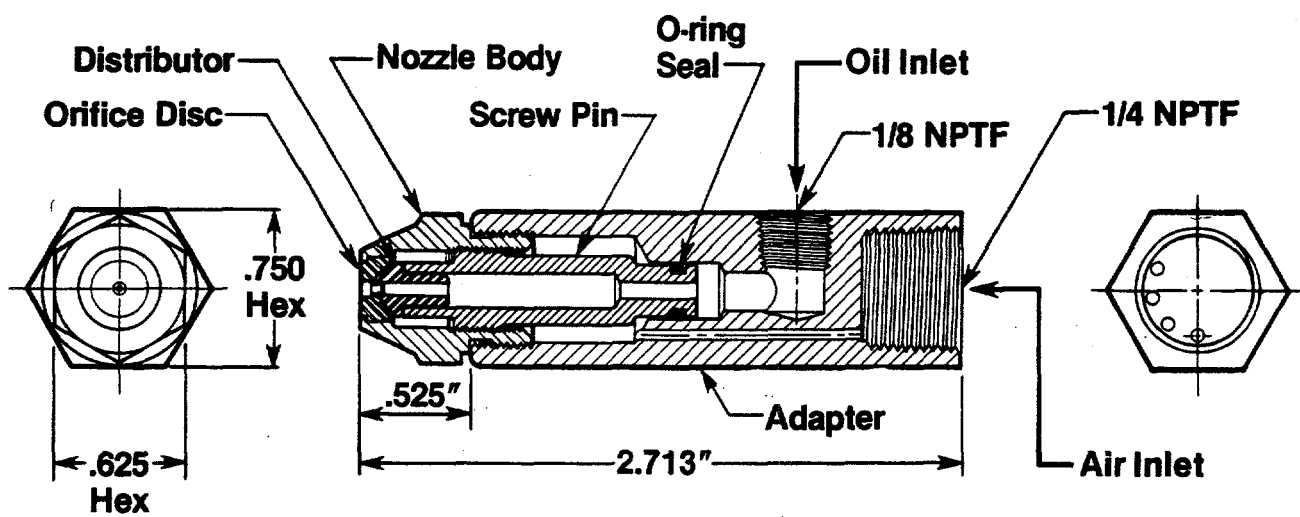
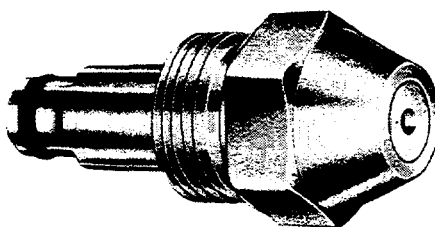
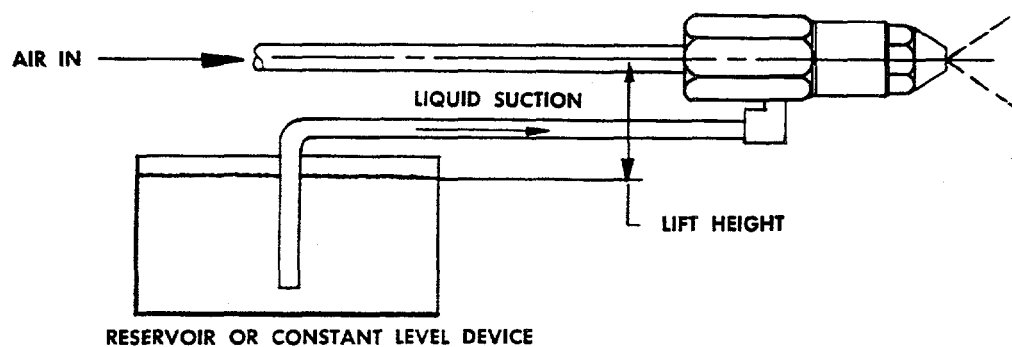


Figure 2-1. Illustrations of siphon-type atomizer. Used with permission - Delavan Fuel Metering Products Operation, Coltec Industries.

air/fuel mixing and poor combustion performance. To reduce to open area of the diffuser a metal ring was added to the front of the diffuser and then covered with a high temperature moldable ceramic. This reduced the diameter of the center hole (to 9/16") and also completely blocked the "slots." These slots are normally used to prevent coke formation on the hot face of the diffuser. In combustion tests with the modified burner no coke formation occurred.

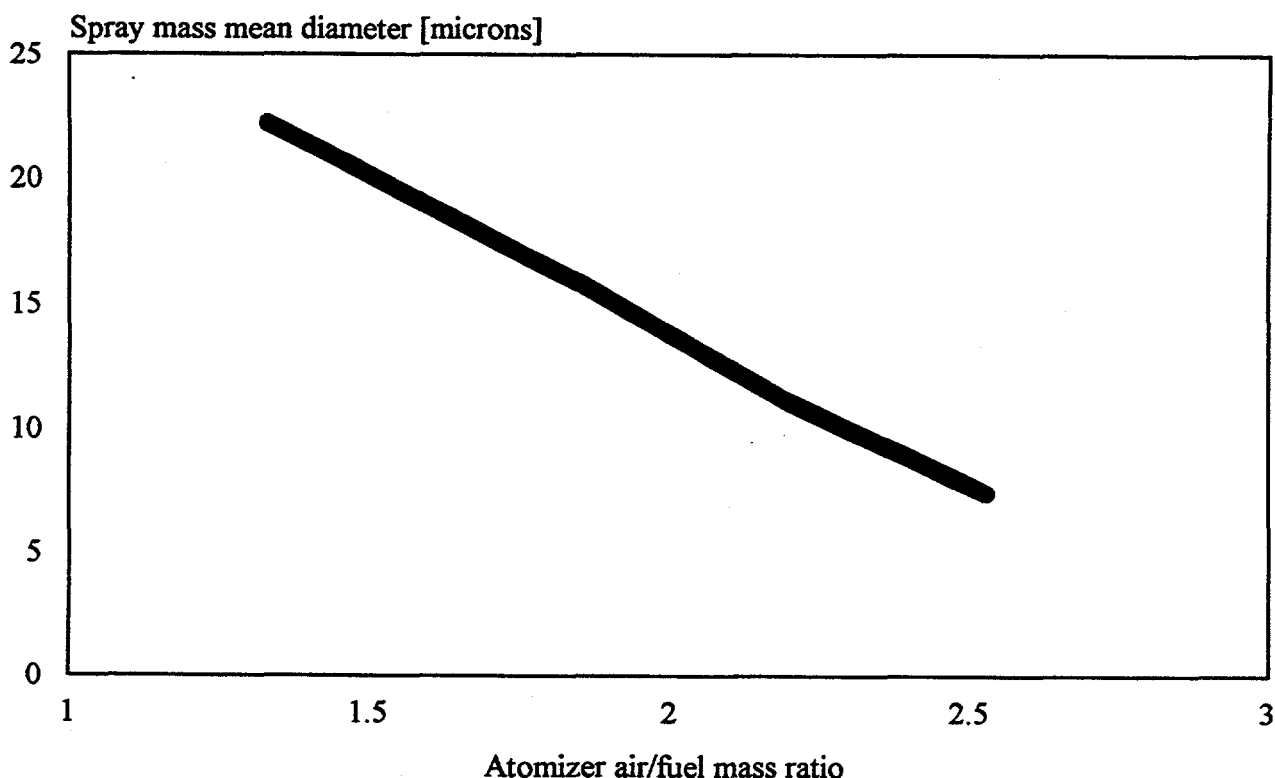


Figure 2-2. Atomization performance of siphon-type nozzle at 0.25 gph firing rate. The mass mean diameter is the diameter for which half of the mass of the spray is both larger and smaller.

Combustion tests with the burner were done in a three section, cast iron boiler with firing rates ranging from 0.25 to 0.60 gph. Generally the burner starts very well and can operate with relatively low excess air--15%. At higher firing rates the flame becomes excessively long. This is mostly a result of a narrow spray angle with this nozzle. At 0.5 gph detailed emission testing was done as part of another project in which emissions from a range of burner types were compared.[1] Figure 2-4 shows smoke, CO, and NO_x emissions. Particulate emission rates were found to be similar to those from pressure-atomized, retention-head burners--about 0.35 lbs of particulates/1000 gallons of fuel oil fired in cyclic operation.

Based upon the results of these tests as well as the commercial availability of products using siphon type nozzles this approach appears attractive. Probably the greatest impediment to commercial use of this nozzle in residential heating applications is the cost and service requirements of the air compressor. In Section 5 of this report these costs are compared with costs of other options.

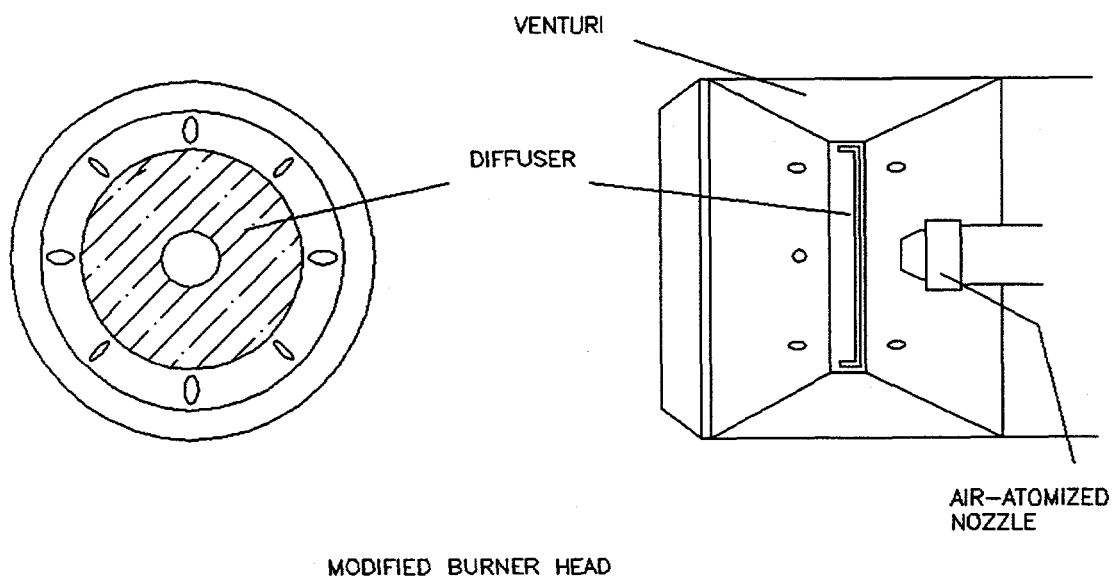
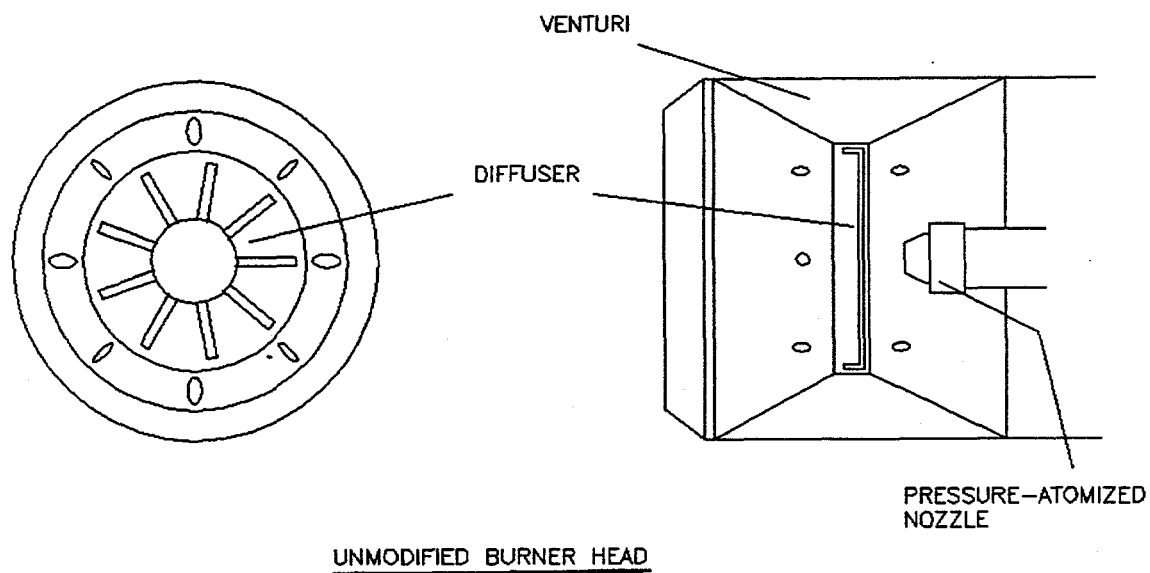


Figure 2-3. Illustration of the changes made in the head of a conventional retention head burner to use the siphon-type air atomizing nozzle.

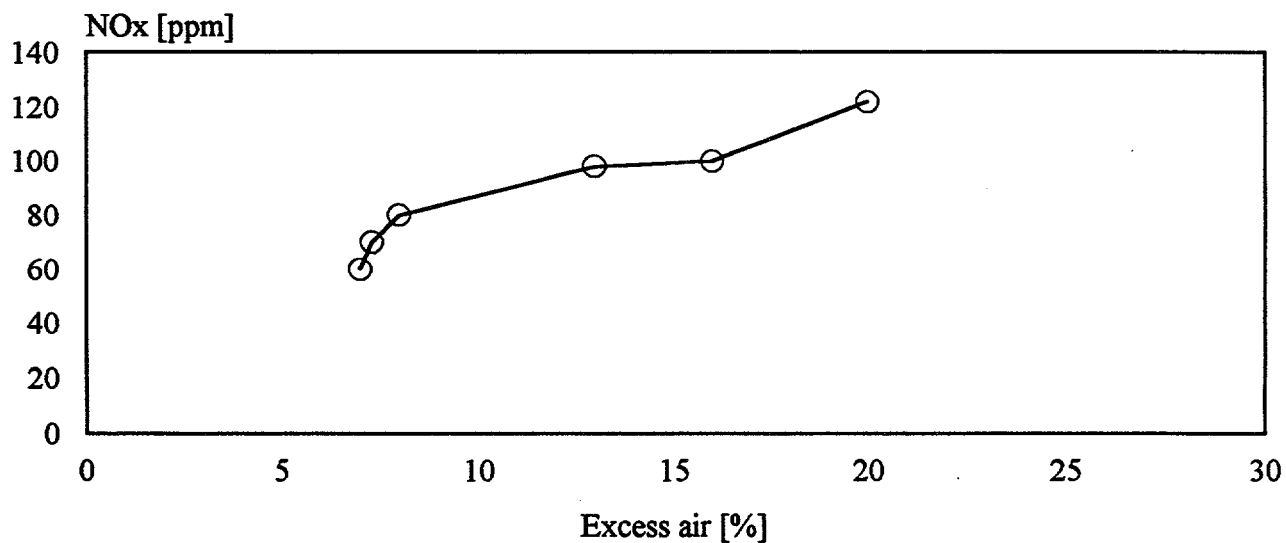
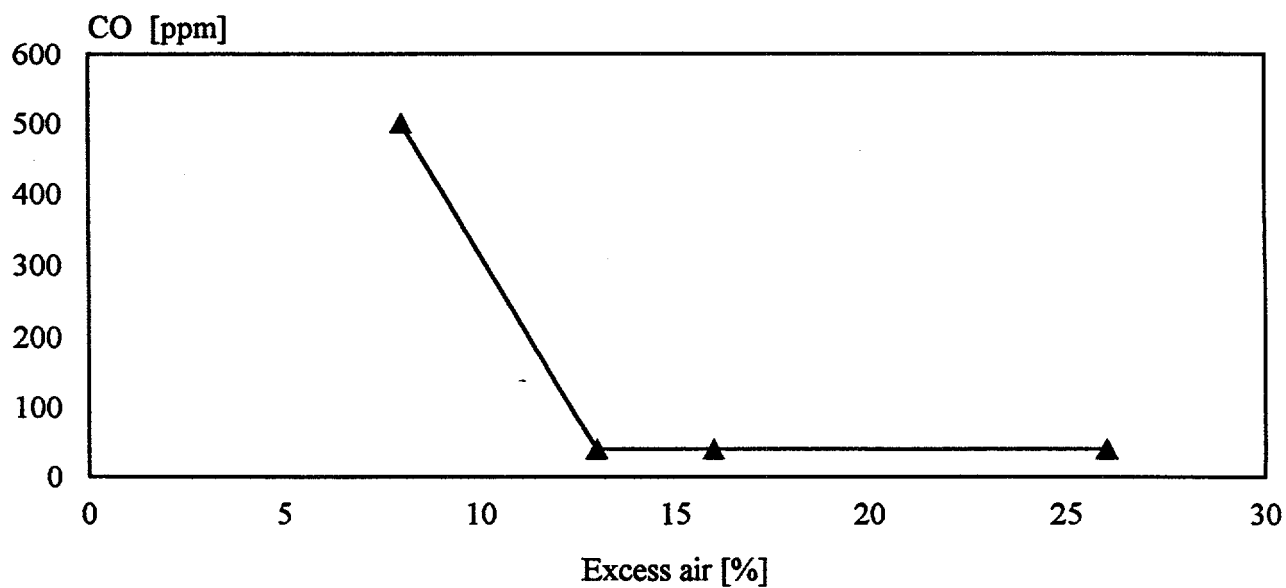
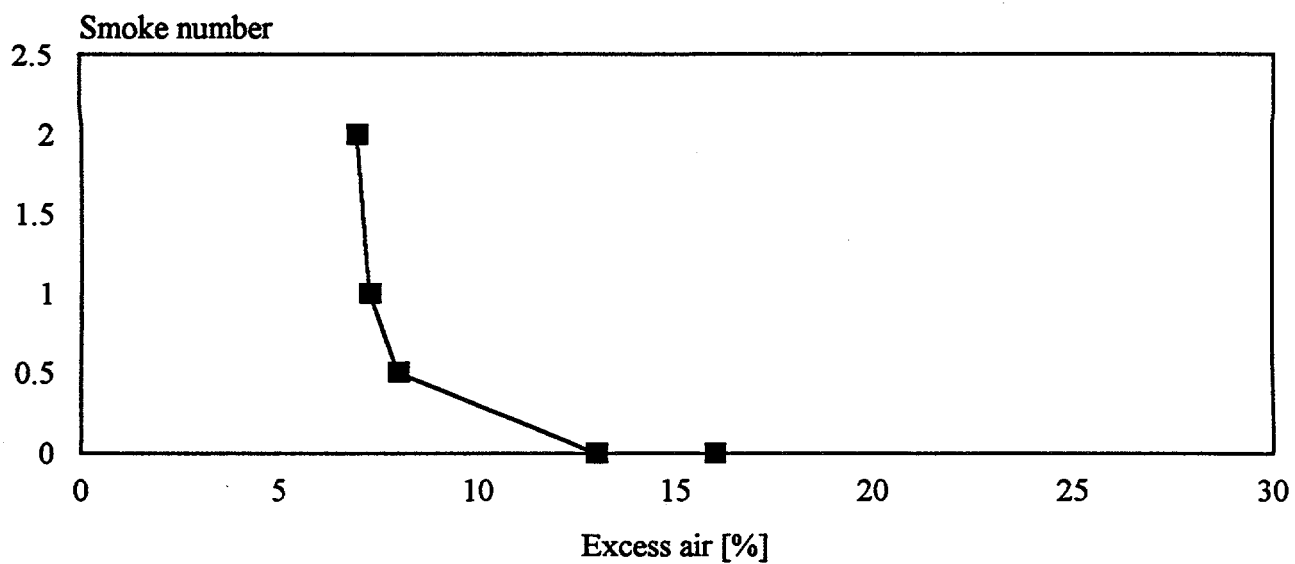


Figure 2-4. Combustion performance of conventional burner modified to have siphon-type, air-atomized nozzle. 0.5 gph firing rate.

3. CHARACTERISTICS OF THE LOW-PRESSURE AIR-ATOMIZING NOZZLE

As discussed in Section 1, most of the development effort in this project was placed on the use of low pressure air atomization which would enable a burner to use only a single fan rather than a fan/compressor combination. The nozzle used was provided by the Gas Turbine Systems Division of Parker Hannifin Corporation with some help and very useful suggestions provided by Mr. John Gaag, Manager of Advanced Technology in Cleveland, Ohio. It was originally developed by Parker Hannifin in a joint project with the General Motors Corporation to heat air and clean a catalytic filter used to reduce particulate concentration in diesel engine exhaust [8].

A photo showing the parts of the atomizer is included here as Figure 3-1 and a schematic illustrating the principle of operation is shown in Figure 3-2. Air at a pressure of 4 to 16 inches of water enters the back. Most of the air passes through the outer swirler and spins out through the main exit orifice. A smaller amount passes radially inward through four, small offset holes ("A" in Figure 3-2) providing counter-swirling air around the pintle. Fuel entering at the centerline flows radially out through three small holes near the pintle tip where the swirling air distributes and swirls the oil, prefilming it as it leaves the inner orifice (B in Figure 3-2). The two counterswirling air flows shear the sheets and ligaments of fuel into a conical spray.

During the course of this development project atomizers were built with different widths of the outer air swirler slots and different angles on the outer air swirler. Increasing the swirler slot width increases air flow for a given atomizer air pressure. Figure 3-3 serves to define the swirler angle and the swirler slot width referred to below.

For the baseline condition the outer swirler slot width is 0.064". Figure 3-4 shows the relationship between nozzle air inlet pressure and air flow with this slot width. For complete combustion of 1 gallon of No. 2 fuel oil the total amount of air required is 1354 standard cubic feet (scf). This assumes no excess air. Included in Figure 3-4 is the air flow as a percentage of the total flow which would be required at a firing rate of 0.5 gph (71,000 Btu/hr). The intended firing rate range for the burner is 0.25 gph to about 1.0 gph and atomizing air pressures normally range 8-12 inches of water. At the low end of the firing rate range the atomizing air flow becomes a very significant part of the total.

Under cold conditions, as the swirling air (with atomized oil) leaves the nozzle exit orifice it produces a conical jet which entrains air from the surroundings. Radial profiles of axial velocity were measured with the nozzle at selected distances from the exit orifice. These measurements were made using a pitot tube under cold conditions, without fuel flow. Results are shown in Figures 3-5 and 3-6 for two different levels of atomizer air flow. For a given distance from the nozzle the velocity profiles can be integrated to determine total volumetric flow rate. Figure 3-7 shows the results of this for one set of conditions and illustrates the reduction in centerline velocity and also flow entrainment with downstream distance. These cold flow results have important implications for burner design and performance. Because the central jet has such high momentum

and very effectively entrains surrounding air (or combustion products) it is inherently a strongly recirculating burner. Characteristics of ignition and combustion can be influenced by controlling the temperature and composition of the gas which is surrounding the nozzle and which is entrained into the flame zone.

Atomization characteristics of the nozzle were measured with water and oil using the BNL atomization test facility discussed in Section 2. The results with water are of interest because they allow for comparisons with some other types of atomizers for which performance only has been evaluated with water. Figure 3-8 shows the water results and Figure 3-9 shows results with Number 2 fuel oil. Generally, the atomization performance is not as good as that of conventional pressure nozzles (mass mean diameter about 40 microns) except when the atomizing air pressure is very high (over 15 inches of water). At very low liquid flow rates the atomization performance improves significantly.

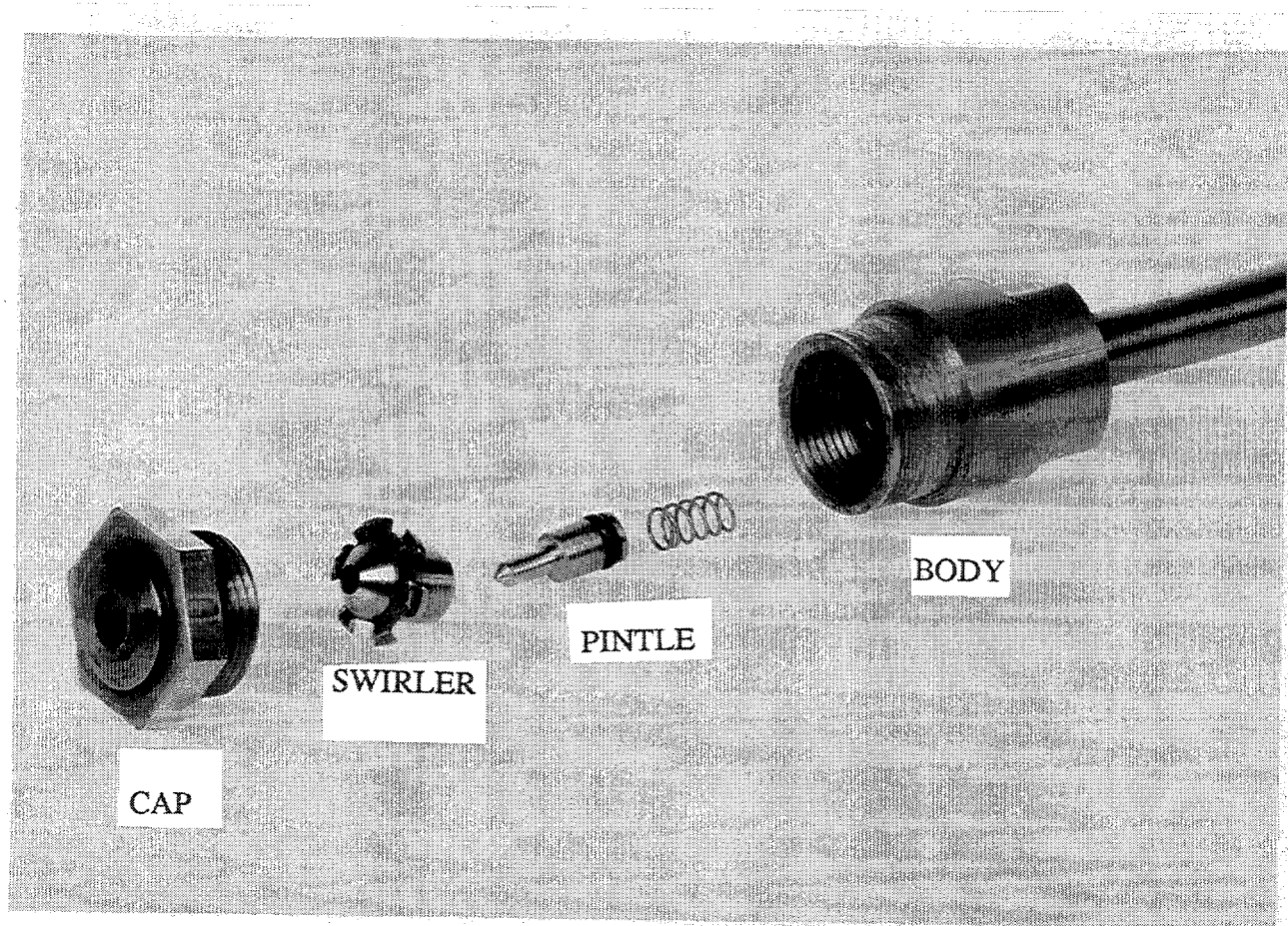


Figure 3-1. Photograph of disassembled low-pressure air-atomizing nozzle.

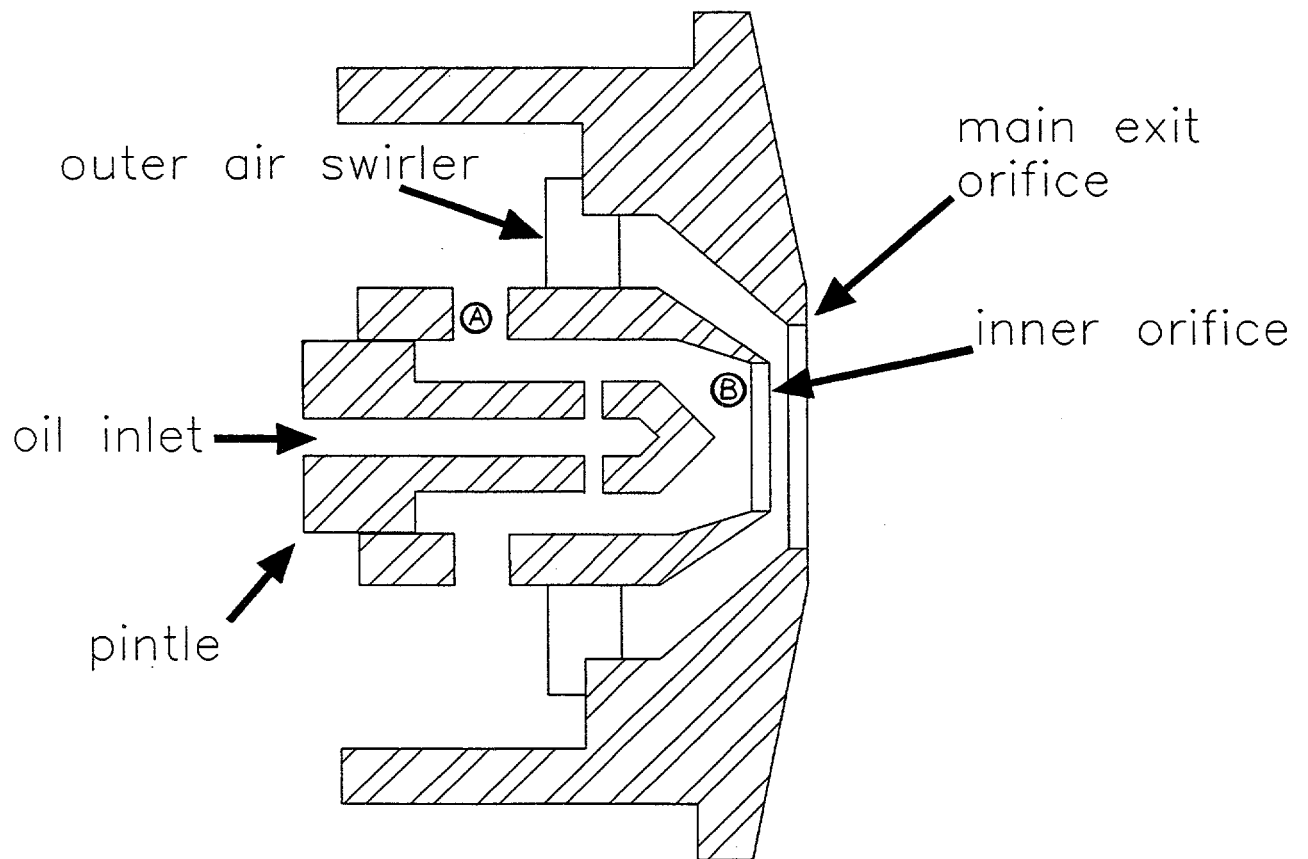


Figure 3-2. General illustration of the low-pressure, air-atomizing nozzle showing air and fuel flow passages.

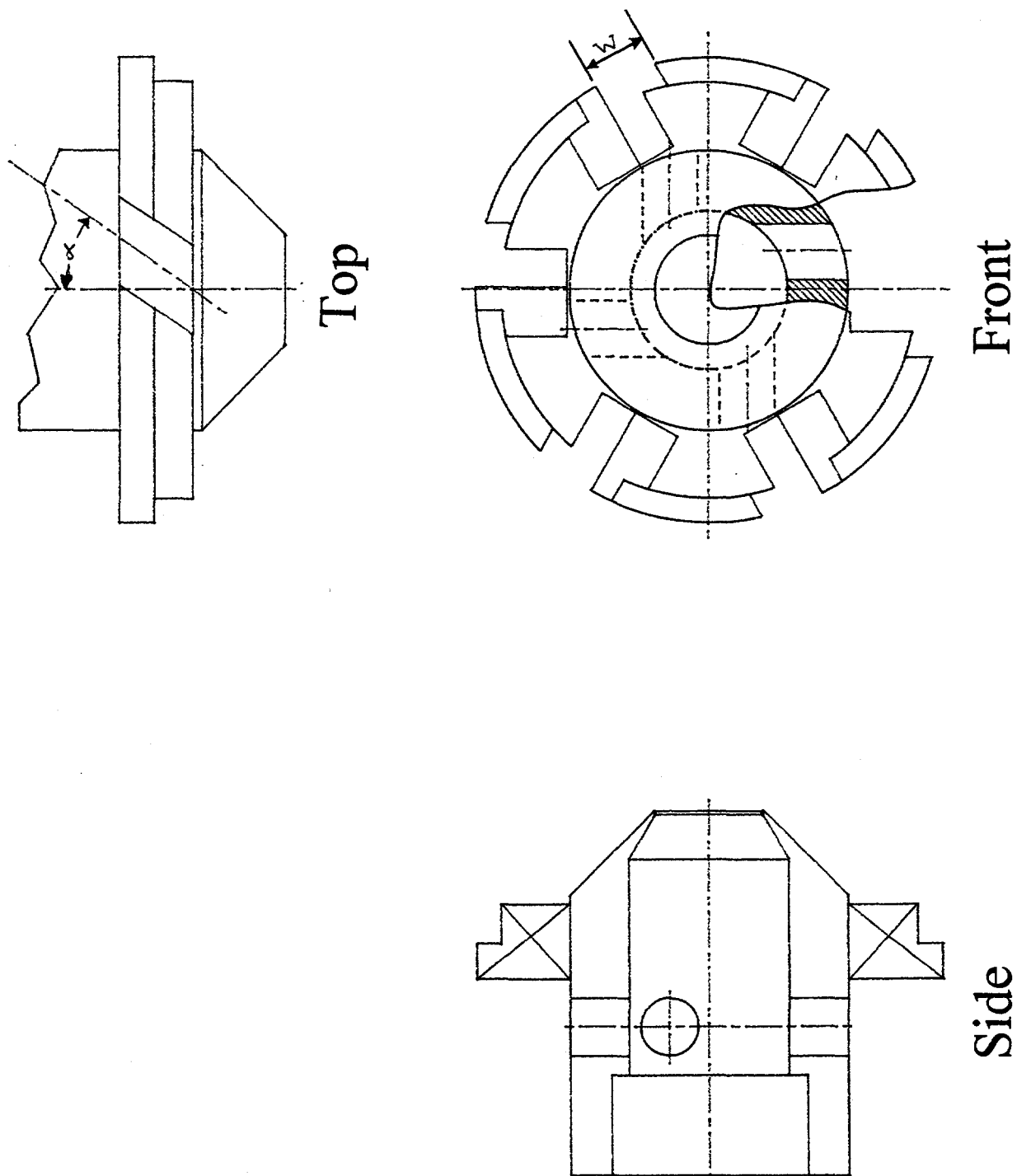


Figure 3-3. Illustration of air swirler defining swirler angle (α) and swirler slot width (w)

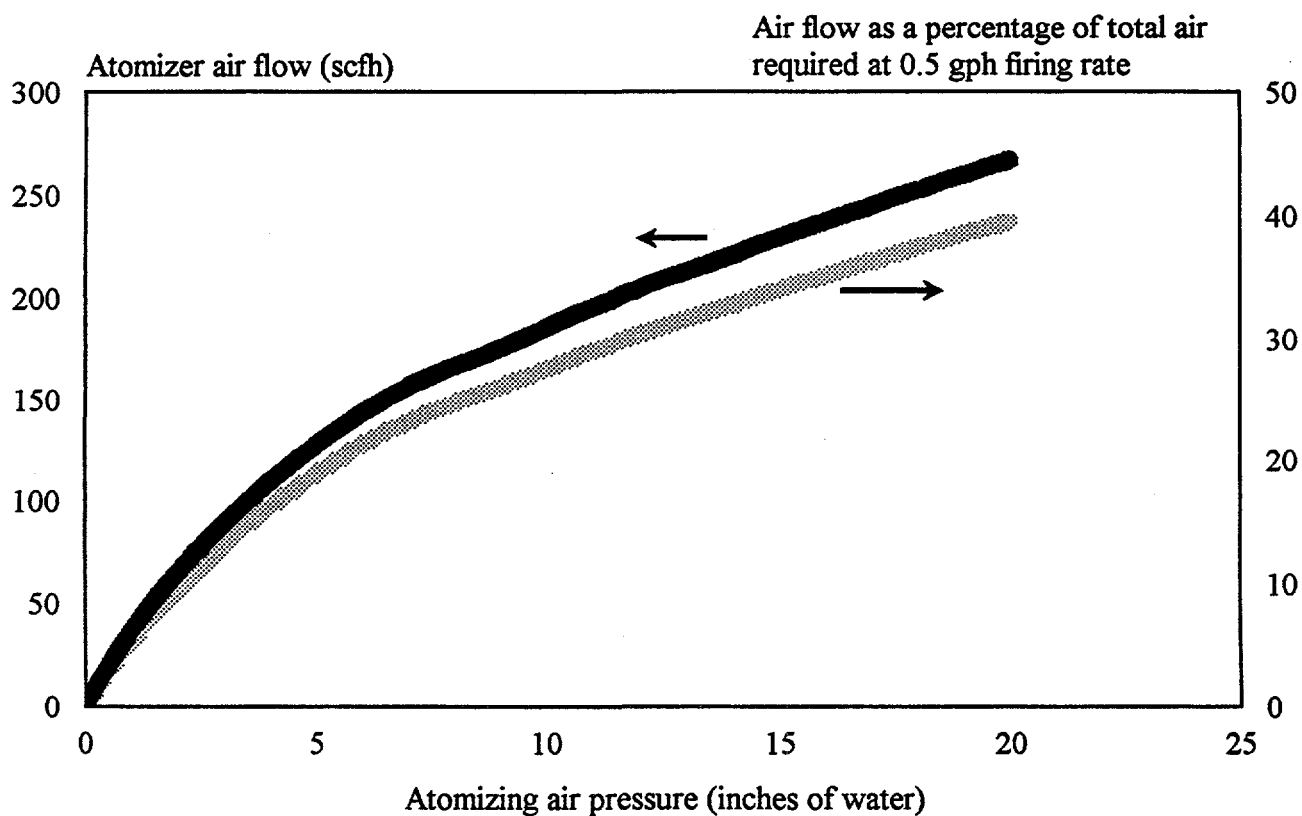


Figure 3-4. Flow/pressure relationship for low pressure atomizer under cold conditions without fuel flow. Shows flow both in standard cubic feet of air per hour and nozzle flow as a percentage of the total air flow required (stoichiometric) at a firing rate of 0.5 gallons per hour.

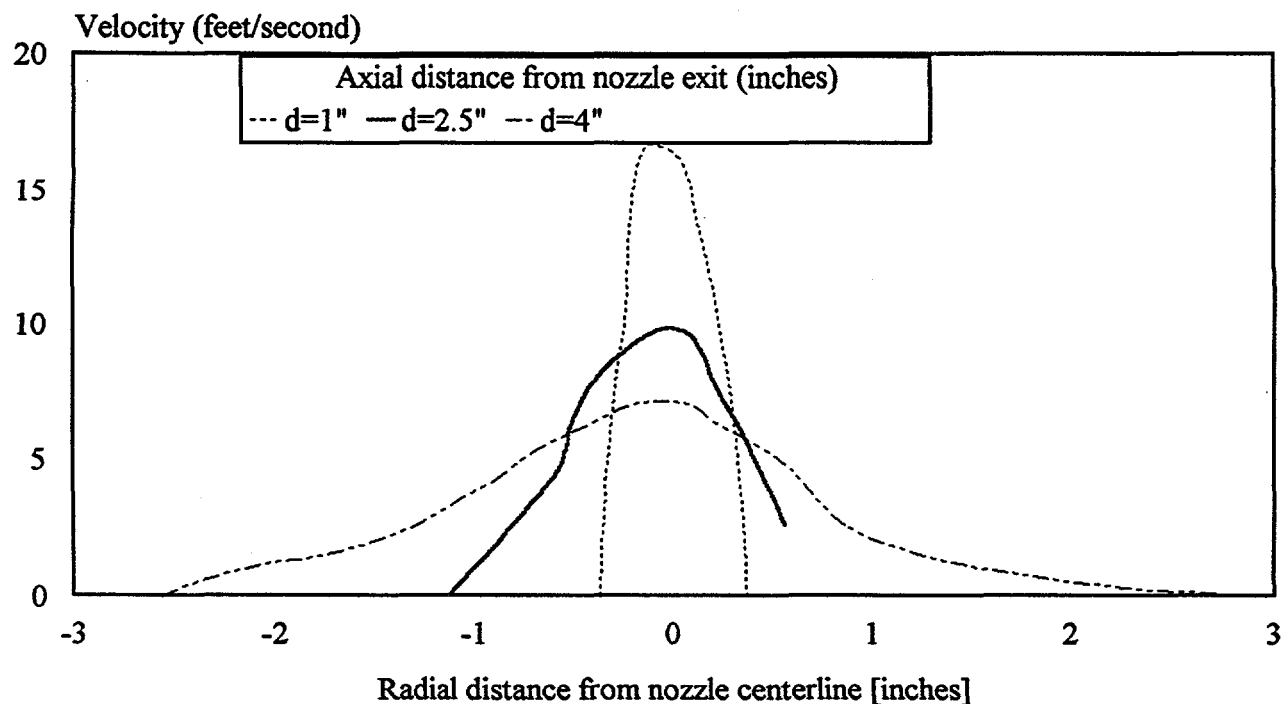


Figure 3-5. Radial profiles of air velocity at three distances downstream from nozzle exit. Atomizer air flow - 150 scfh. Note on x axis - and + numbers indicate different sides of centerline.

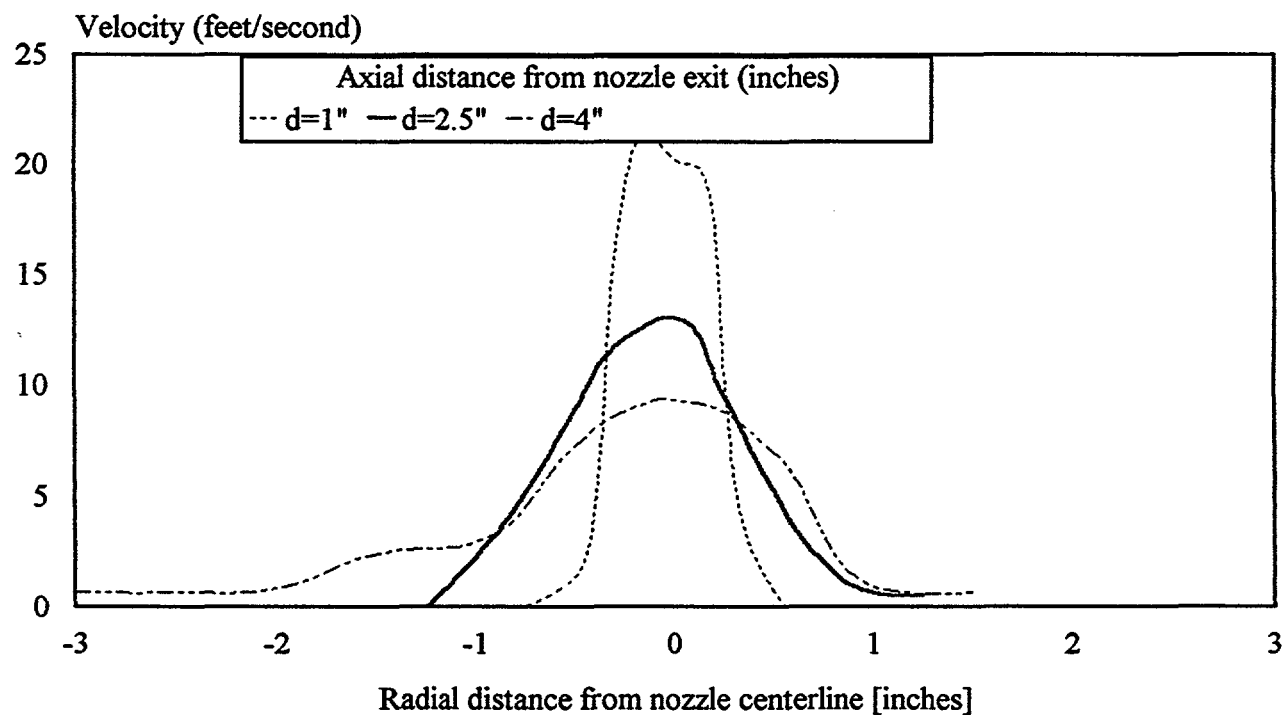


Figure 3-6. Radial profiles of air velocity at three distances downstream from nozzle exit. Atomizer air flow - 185 scfh. Note on x axis - and + numbers indicate different sides of centerline.

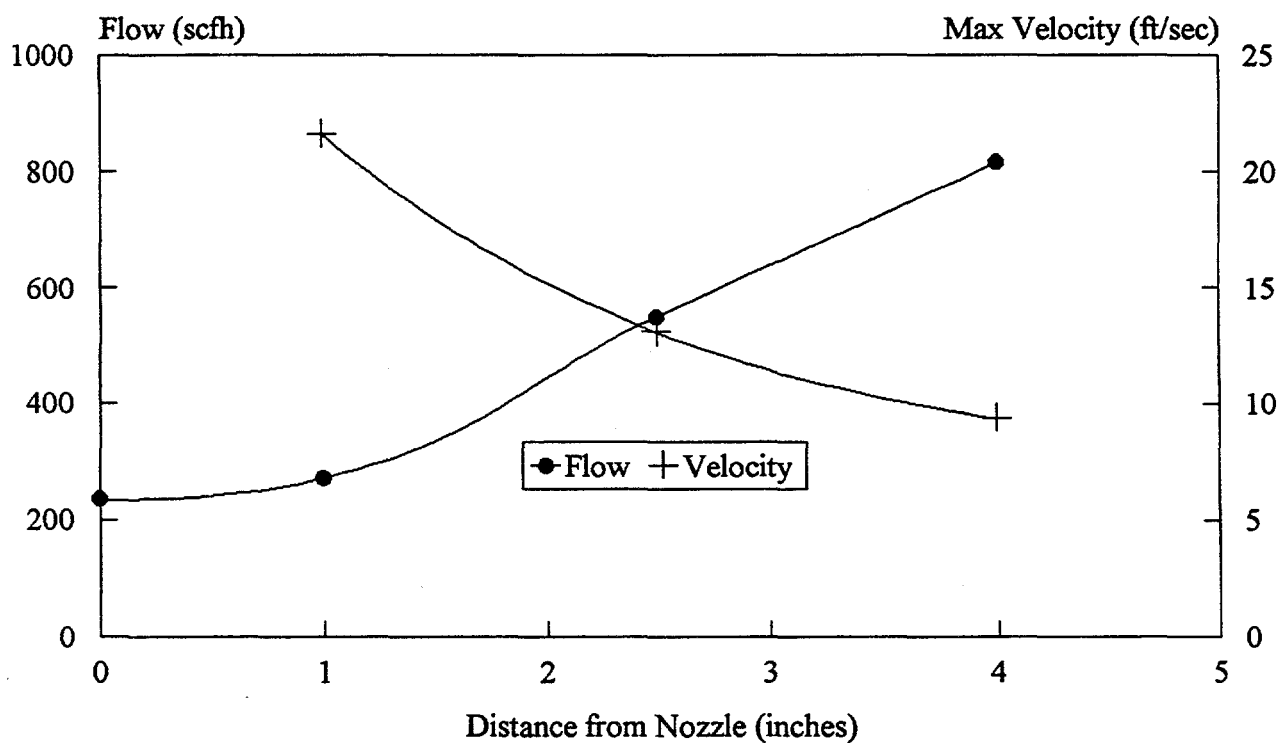


Figure 3-7. Low pressure air-atomizing nozzle. Cold air flow measurements. Decay in maximum velocity and increase in total flow due to entrainment as a function of downstream distance from nozzle exit.

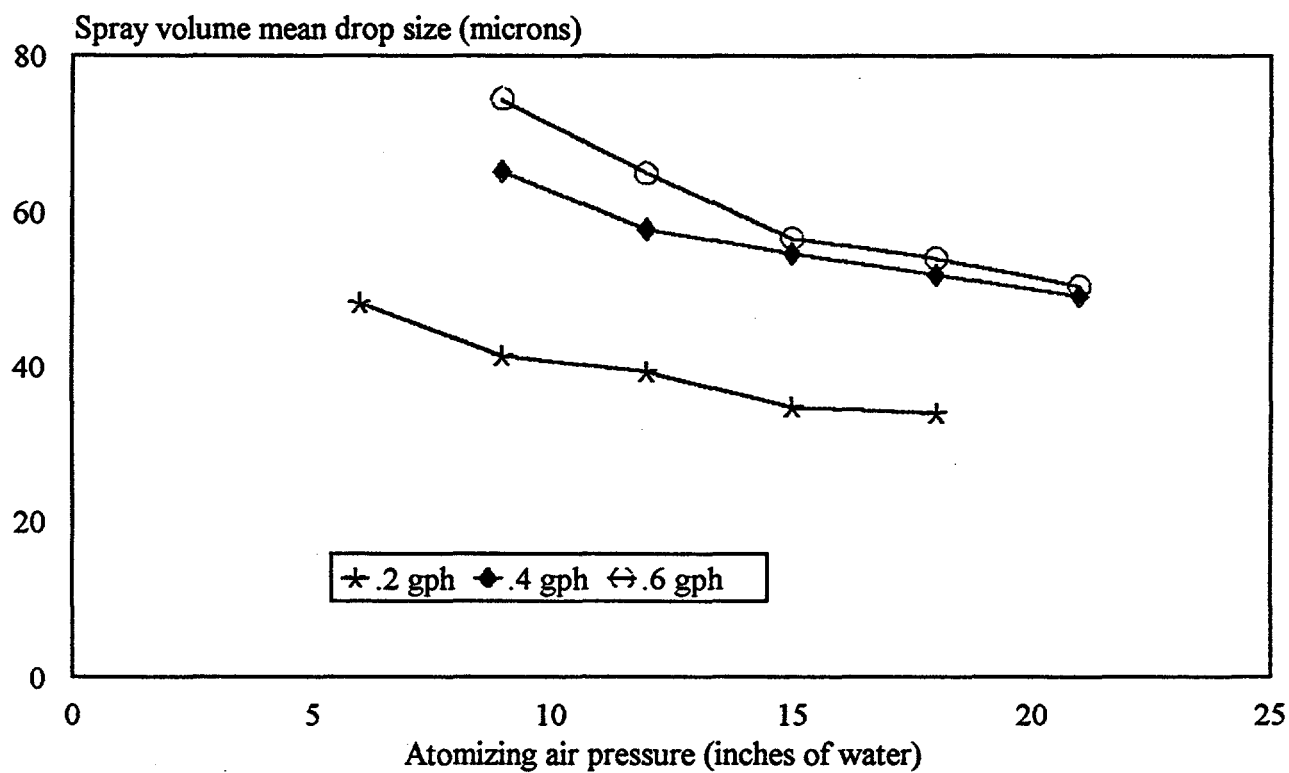


Figure 3-8. Results of atomization tests with low pressure air-atomized nozzle spraying water.

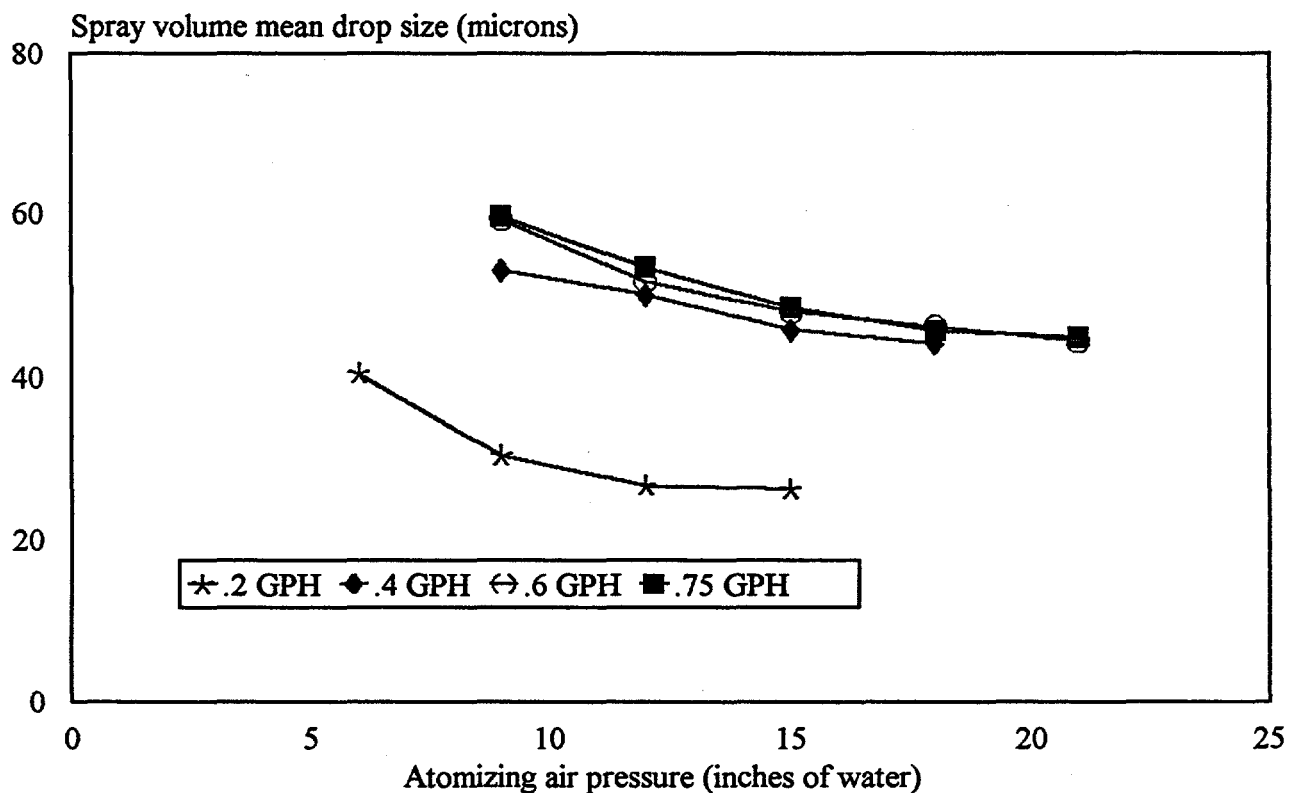


Figure 3-9. Results of atomization tests with low pressure air-atomized nozzle spraying oil.

4. DEVELOPMENT OF A BURNER HEAD USING THE LOW-PRESSURE ATOMIZING NOZZLE

INITIAL DESIGN - SIMPLE, CONCENTRIC AIR FEED

Figure 4-1 generally illustrates the first burner head design approach. This head was configured specifically to allow laboratory evaluation of effects of operating conditions on performance. In this design air is fed in three separate parts. Primary air is delivered to the atomizer through a single, small diameter tube. This air is from a compressor, and is externally metered using a rotameter (not shown). Both secondary and tertiary air are provided by an external blower. The secondary air passes through a metering plate and around the nozzle electrode assembly. The metering plate has a set of holes through which air flows. The plate has two purposes: first, to distribute secondary air evenly and second, to allow the secondary air flow rate to be determined during testing. The flow rate/pressure drop characteristics of the metering plate were established in advance. During a test pressures were measured upstream and downstream of this plate to determine secondary air flow rates. The situation with the tertiary air was similar.

The primary air flow rate is set by the atomizing air pressure. Typically this ranged from 150 to 225 scfh. In testing it was generally found desirable to keep the secondary air flow as low as possible. However, if the secondary air flow was reduced to below 100 scfh recirculation of combustion products behind the nozzle was observed, leading to overheating of the electrodes. The remainder of the air for combustion was provided as tertiary air.

One of the parameters evaluated in the test work was the dimension "L1" in Figure 4-1. Increasing this distance improved ignition and flame stability characteristics at low firing rates. At high firing rates, however, it was found necessary to keep this distance shorter to prevent fuel impingement on the walls of the burner tube. At firing rates below 0.5 gph L1 was about 3.5" in most testing. At higher rates L1 was about 2" for most testing.

Generally, ignition of the burner was found to be good and startup transient smoke numbers were found to be typical of those with conventional retention head burners. Emissions of carbon monoxide were found to be higher during the brief startup period than with conventional burners.

Figure 4-2 shows the trend in NO_x and CO emissions in steady state over a broad range of excess air in one test series at a firing rate of 0.31 gph. Atomizing air flow for these tests was about 55% of the stoichiometric air and atomizing air was about 16 inches of water. The smoke number was zero over the entire range.

Figures 4-3 and 4-4 show results at a firing rate of 0.25 gph with two atomizing air flow rates. In the case of Figure 4-3 the atomizing air flow was 40% of the stoichiometric and air pressure was 10 inches of water. In Figure 4-4 atomizing air flow was 55% of stoichiometric and 15 inches of water. With the lower atomizing air pressure (Figure 4-3) the smoke number was high at the lowest excess air level tested. In all other cases the smoke number was trace or zero.

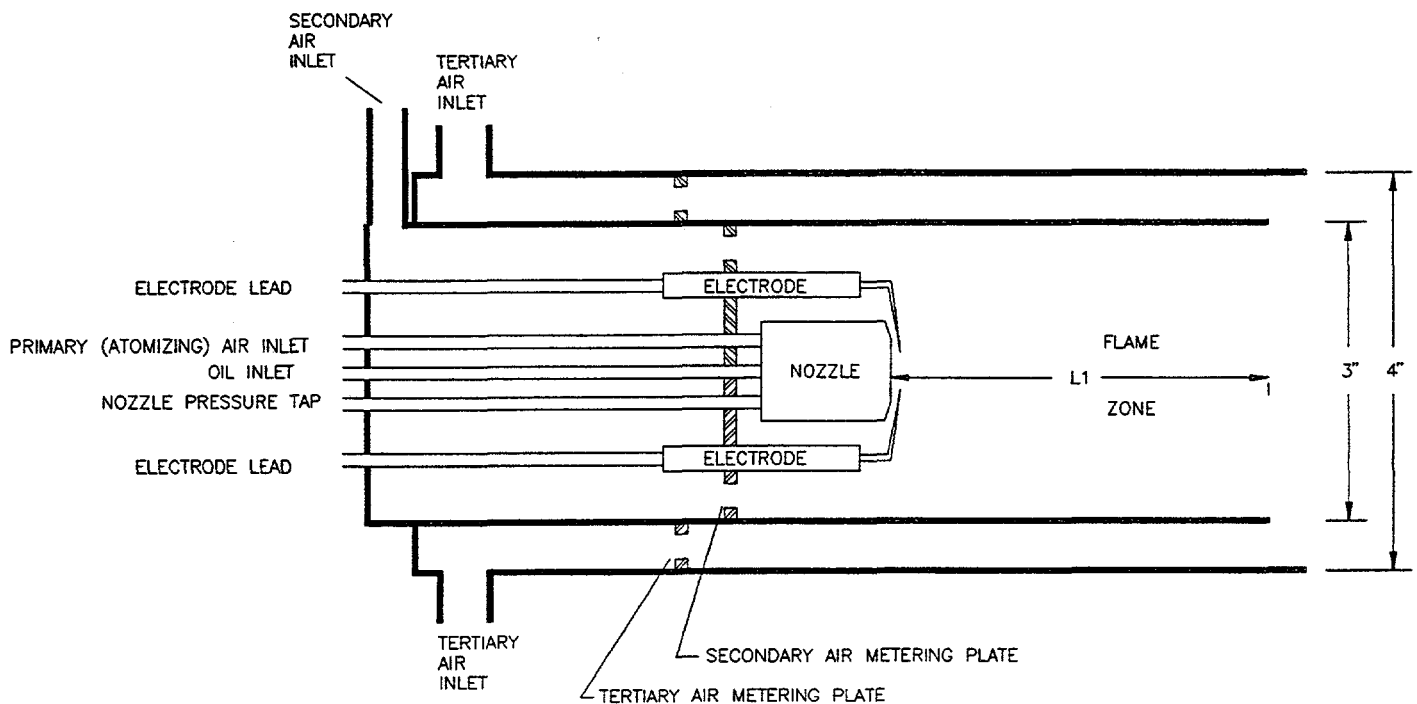


Figure 4-1. Illustration of initial burner head design using the low-pressure atomizing nozzle.

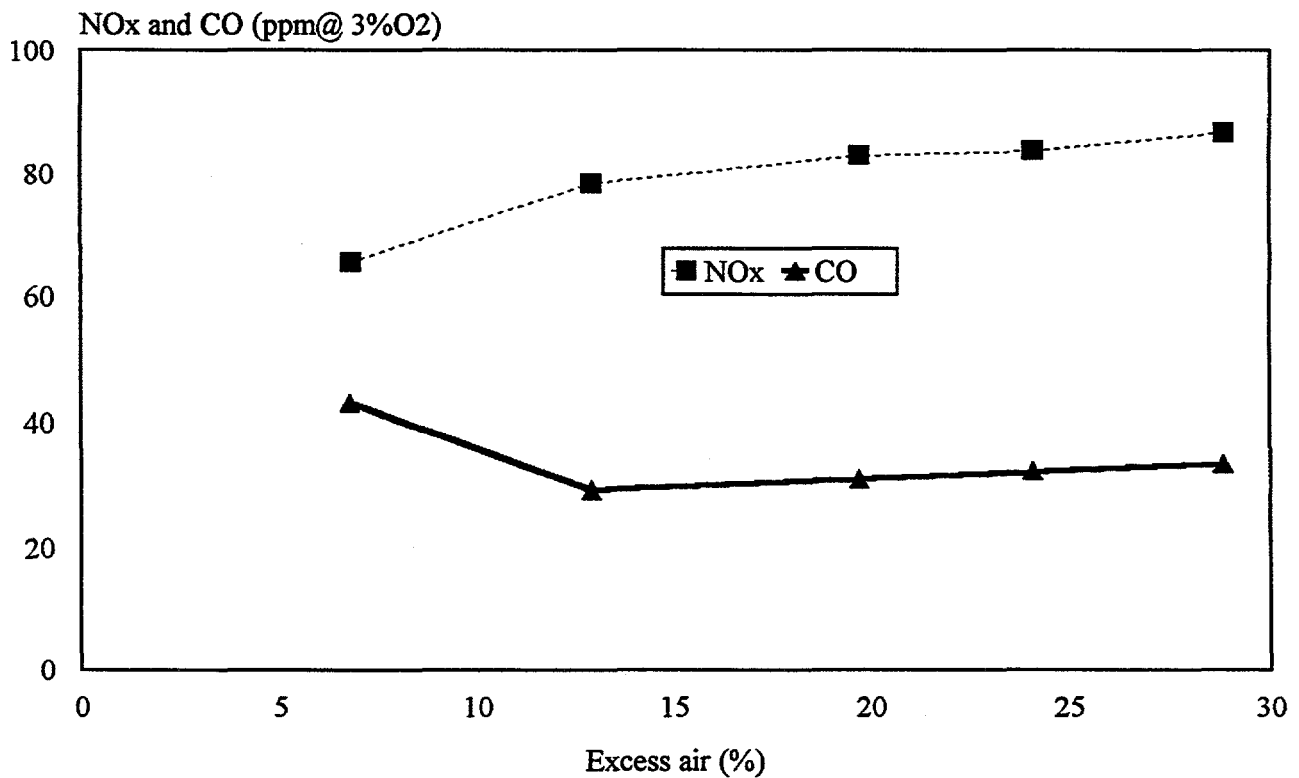


Figure 4-2. Combustion test results - initial design of burner head using low-pressure atomizing nozzle. 0.31 gph firing rate, 16 inches of water atomizing pressure.

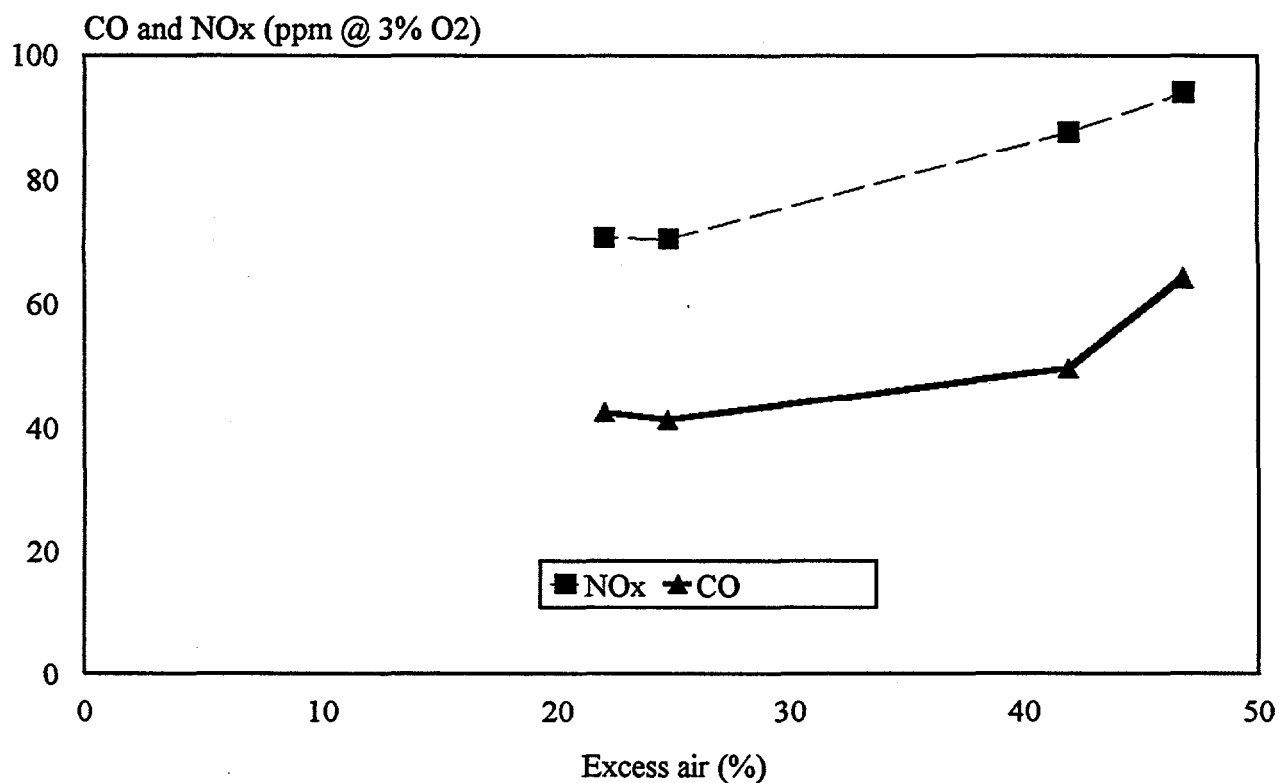


Figure 4-3. Combustion test results - initial design of burner head using low-pressure atomizing nozzle. 0.25 gph firing rate, 10 inches of water atomizing pressure.

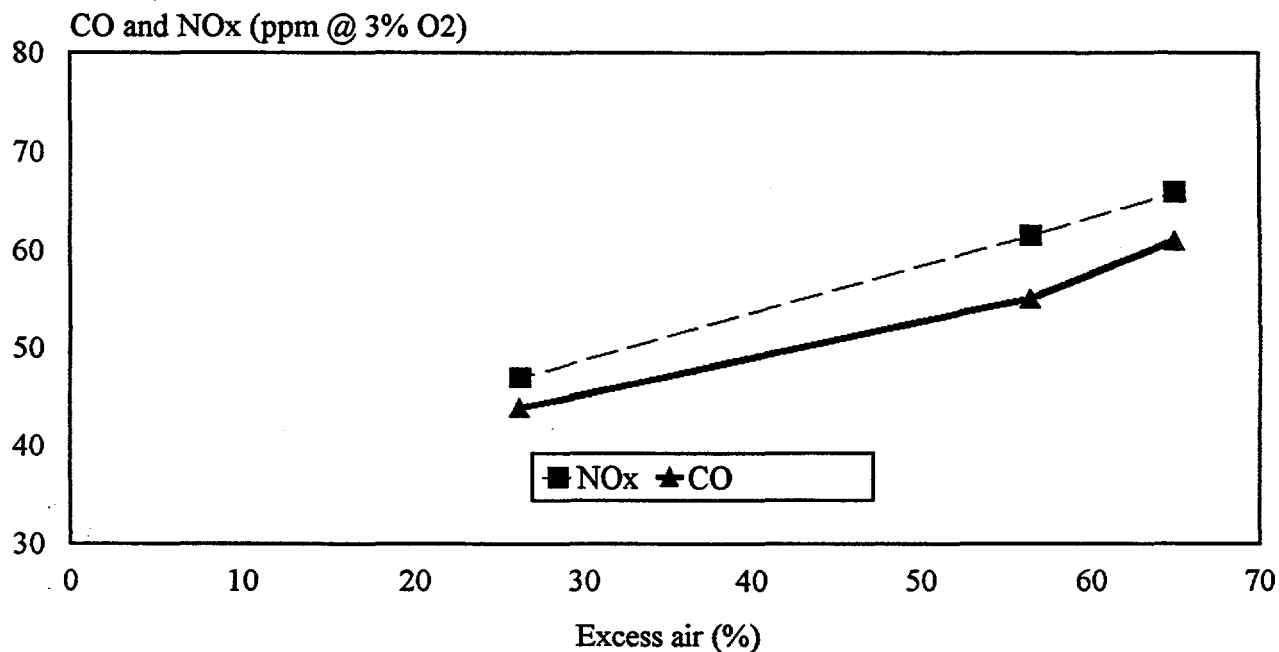


Figure 4-4. Combustion test results - initial design of burner head using low-pressure atomizing nozzle. 0.25 gph firing rate, 15 inches of water atomizing pressure.

DESCRIPTION OF DEVELOPMENT COMBUSTION CHAMBER

Following the initial studies described in the previous section additional development work was done on the burner head to bring it closer to a design which could be commercially acceptable. To assist in this development work a unique combustion chamber was built and is illustrated in Figure 4-5. The main part of the combustion chamber is a horizontal quartz cylinder with an inside diameter of 10 1/2 inches. The burner tested is installed at one end on the axis of the quartz cylinder and combustion products exhaust at the opposite end. The ends of the quartz cylinder are pressed into semi-soft refractory board which effectively provide a seal. Steel plates at the front and rear ends hold the refractory boards rigid and tight against the quartz cylinder. The exhaust products are carried out using conventional flue piping and there is no heat recovery in the system resulting in very high exhaust gas temperatures (1800 F).

A damper in the flue piping was normally adjusted during testing to provide a slight back pressure on the combustion chamber (+.01 inches of water) to assure that air leakage into the combustion chamber which would affect performance and readings did not occur. Gas samples for analysis were extracted from the flue piping upstream of the damper. A simple stainless steel tube was used to extract gas samples from the flue piping. After exiting from the flue piping this sample tube was uninsulated for a length of 4' before entering the gas analyzer conditioning system. This length of exposed tubing allowed the gas to cool about 500 F at which point it could be safely handled by the conditioning system.

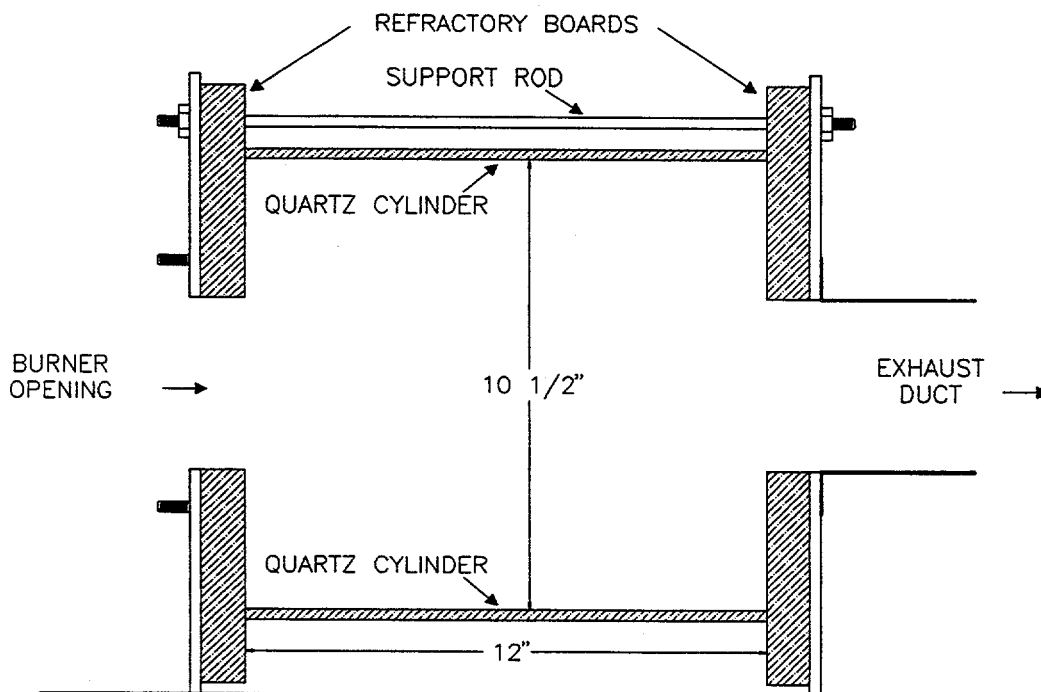


Figure 4-5. Illustration of quartz combustion chamber used for burner development.

DESIGN OF A BURNER HEAD WITH A DIFFUSER AND "SPLIT" AIR SUPPLY

While performance of the burner head described above was generally good and the system provided great flexibility for testing this configuration is certainly not appropriate for use in a commercial burner. In developing a burner head which might be used commercially it is necessary to consider the type of air supply which might be used to provide the needed pressure for the atomizer. In developing this burner head it was assumed that air would be provided by a two-stage centrifugal fan with part of the air extracted after the first stage at a lower pressure. This provides a "split" air supply with one part provided at low pressure (about 3 inches of water pressure) and the other part at high pressure (to 15 inches of water pressure). The high pressure air would be used for atomization and the low pressure air simply to complete combustion. The alternative to the use of such a split supply would be to provide all air at high pressure and throttle part of the air down to the lower pressure. The split supply was initially selected with the intention of reducing fan total power requirements and cost.

In addition to the split air supply this version of the burner head also included a diffuser plate just behind the nozzle. The purpose of this plate is to improve ignition, provide some swirl to the secondary air, to allow operation at lower excess air levels, and to assure that the nozzle assembly is rigidly positioned.

The burner head is shown in Figure 4-6. Air at the higher pressure level (primary air) enters the back of the burner through port 012 and passes through tube 011 to the atomizer assembly, 007. Fuel enters the atomizer assembly through tube 013. The remainder of the combustion air, secondary air and tertiary air, enters the burner through tube 020. The air exits this tube from a side slot and the tube can be rotated to provide some control of the level of swirl in this part of the flow.

At the front end of the burner the low pressure part of the air is divided into two flows. A small part of the air passes through a metering plate [part 009] and then through the diffuser [part 008]. This part of the air flow is termed the secondary air. Small holes in the metering plate control the rate of secondary air flow. In the diffuser plate air passes through swirl slots providing some air swirl in the flame zone. The remainder of the combustion air, the tertiary air, enters the flame zone by passing around the short cylinder, part 016. To adjust the burner excess air level the axial position of the nozzle assembly, diffuser, and connected parts is changed using the hand nut, part 014. This controls the spacing between the front end of cylinder 016 and outer cone-shaped part 015.

Figure 4-6 shows an extension tube, part 023, which confines the flame zone. The configuration of this tube varied considerably during the development effort. Much of the combustion testing was done without this tube and in steady state the performance was very good. The extension tube was added primarily to improve performance immediately after burner startup. During this time some larger fuel oil droplets traveled out of the combustion zone. With the addition of the extension tube the drops hit the tube surface and are burned during this startup period. A consequence of the addition of the extension tube is a small increase in NO_x emissions.

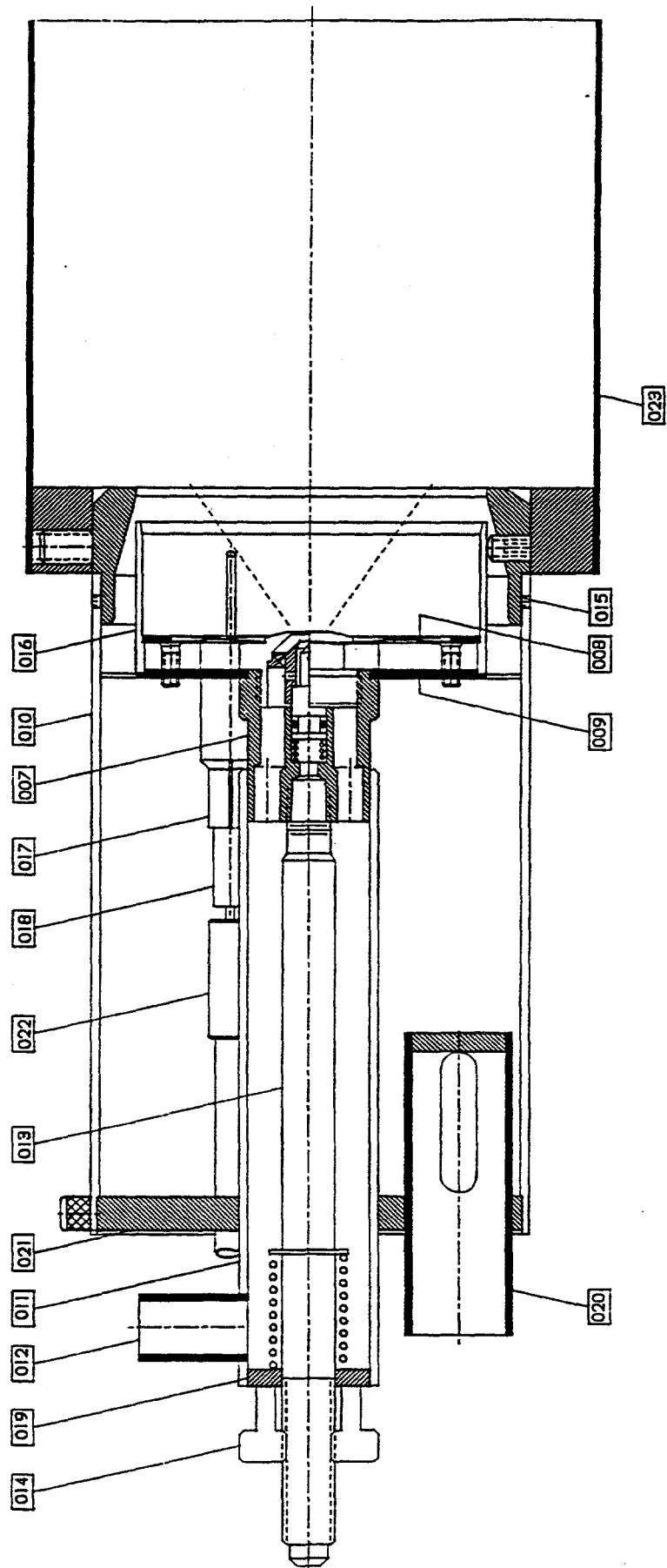


Figure 4-6. Illustration of burner head using diffuser and split air supply.

Combustion testing was done with this burner configuration under a wide variety of conditions. Some example results are shown in Figure 4-7 over a firing rate range from 0.3 to 0.6 gph. All of these tests were done firing in the quartz cylinder combustion chamber without the extension tube discussed above, and with atomizing air pressure set at 15 inches of water. Smoke numbers, which are not reported in this figure, were found to be zero for all tested conditions.

In steady state testing with this version of the burner effects of primary air pressure on performance were explored. No degradation in performance was noted with reduced pressures even as low as 5 inches of water. Due, however, to concerns about startup transients, it was felt that air pressures of 10 inches of water or greater should be used.

DESIGN OF A BURNER HEAD WITH A SINGLE AIR SUPPLY

Combustion performance with the burner head described in the previous section was good. However for the next version of the burner head it was decided to eliminate the split air supply and design for a single air inlet with all of the air at the highest pressure needed for good atomization. This was done for the following reasons:

- It was originally thought that atomizing air pressures may need to be in the range of 15 to 20 inches of water. Based on results with the head designed for the split air supply, however, it was realized that much lower pressures could be used, particularly at low firing rates. As the pressure decreases it becomes more reasonable to provide all air with one fan.
- A fan which provides two pressures would cost more and may be less efficient than a single pressure unit.
- A burner head designed with one air supply would be less expensive and simpler to maintain.
- The air pressure required by the burner could be supplied by an efficient, high speed fan driven with a brushless DC motor. Initially, high electric power consumption due to the increased pressure of this burner were a concern. Using the high speed/brushless motor approach, however, the power consumption could be reduced to levels even lower than with conventional burners.

With the use of a single air supply, all of the air discharged from the fan enters the back end of the burner head. This air is then internally divided into primary, secondary, and tertiary flows. The relative amount which flows into each of these three parts depends upon the details of the internal flow passages for each section. Figure 4-8 shows an artist's illustration of the general flow situation in the burner head. While not correct in detail this sketch shows how all of the air from a single plenum at the back of the burner is divided into three parts. Figure 4-9 shows somewhat more detail and Figure 4-10 is drawn fully to scale. A photograph of the major burner head components is provided in Figure 4-11.

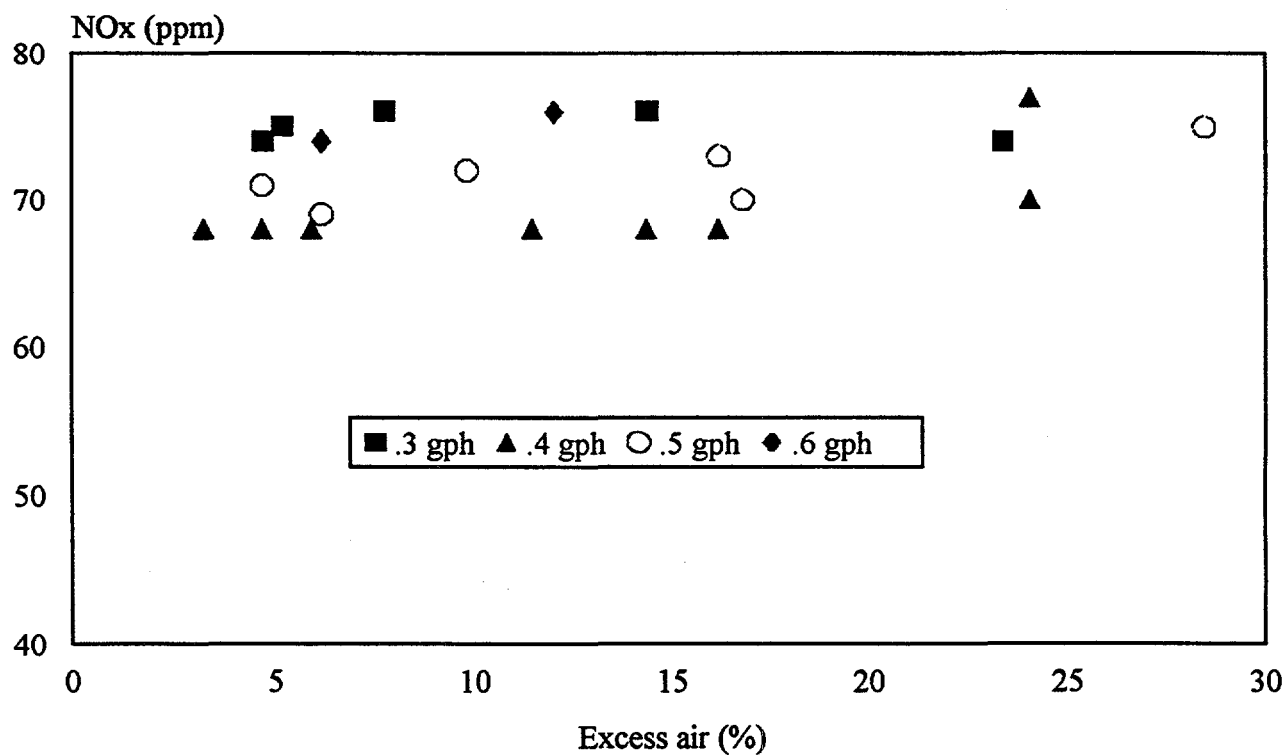
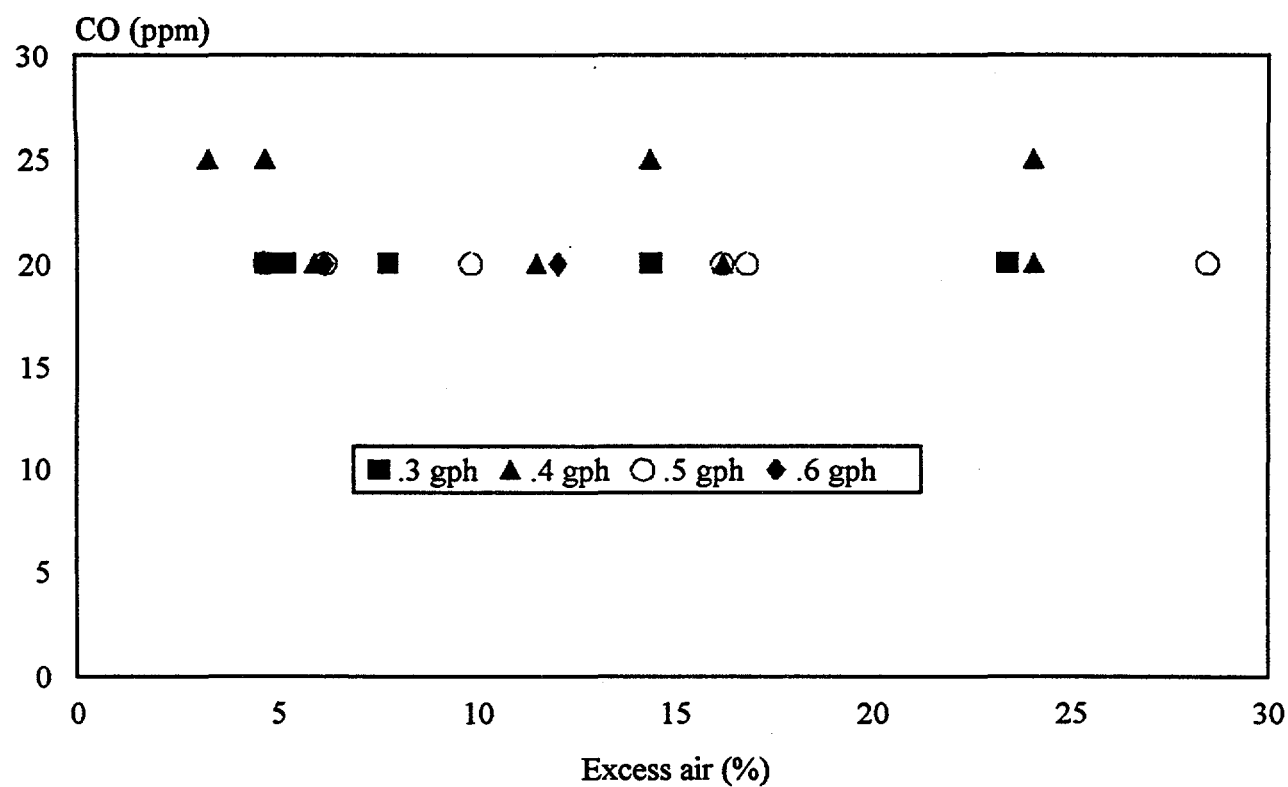


Figure 4-7. Results of combustion tests with burner with split air supply. Tests in quartz cylinder combustion chamber. Atomizing air pressure at 15 inches of water for all tests. For all tests smoke number was zero.

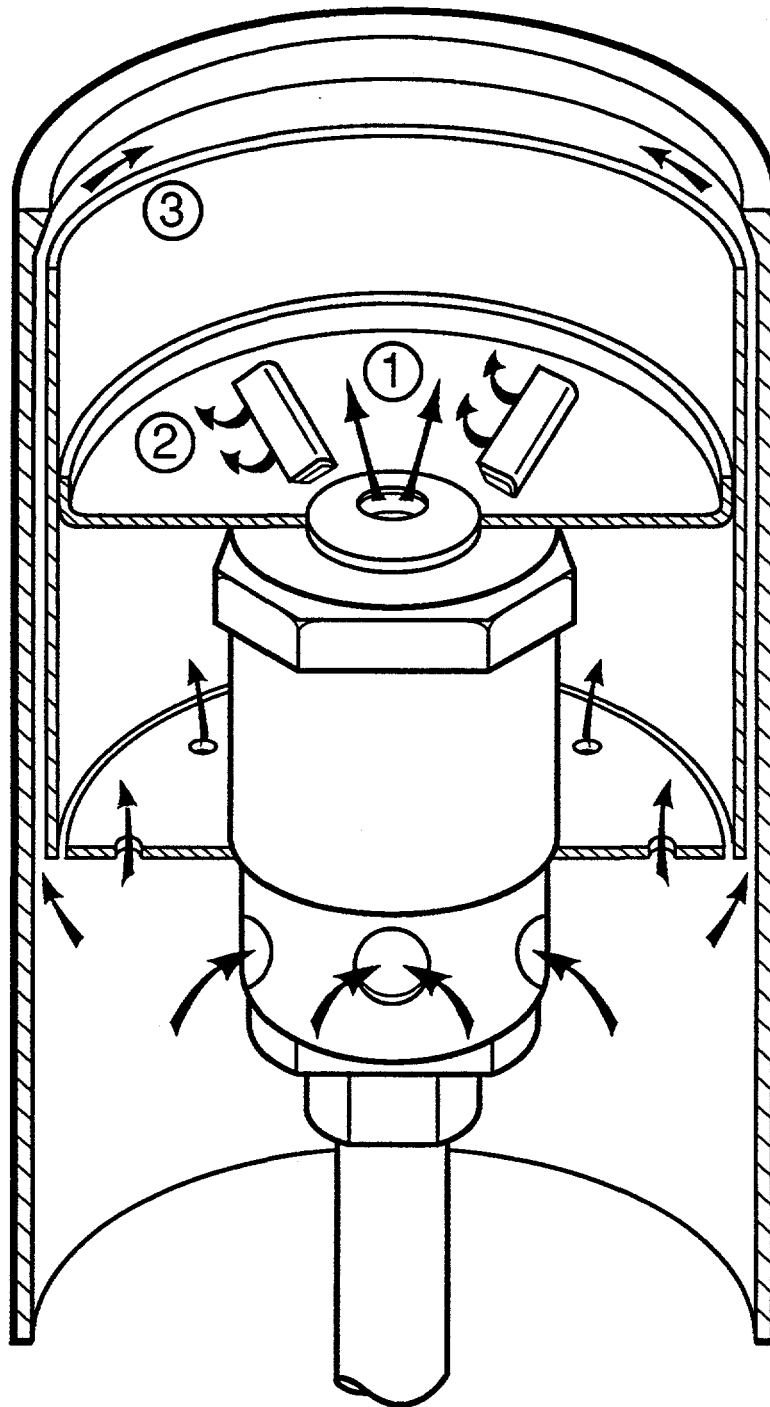


Figure 4-8. Illustration of burner head designed for single air supply

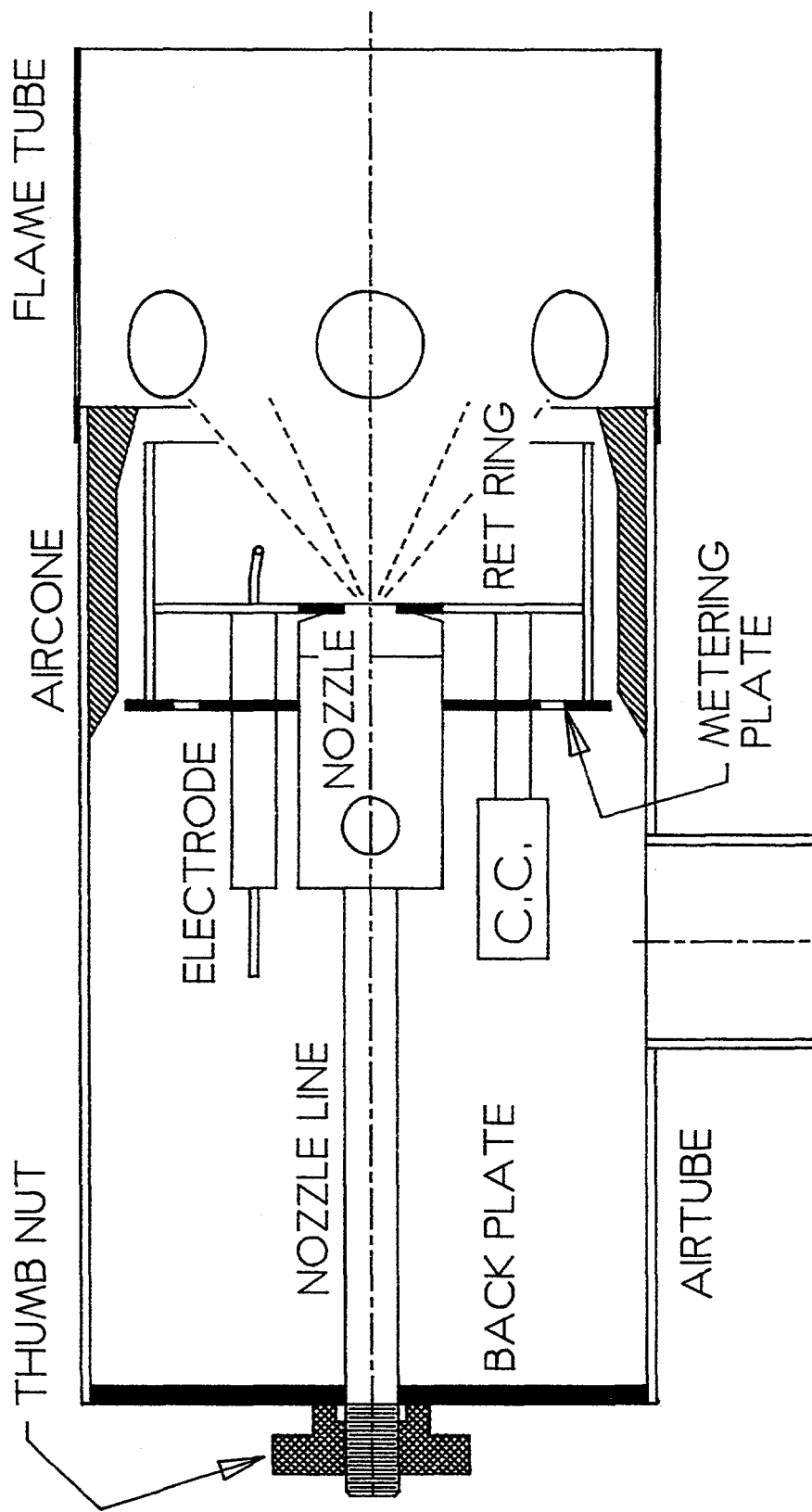


Figure 4-9. Schematic of burner head designed for single air supply.

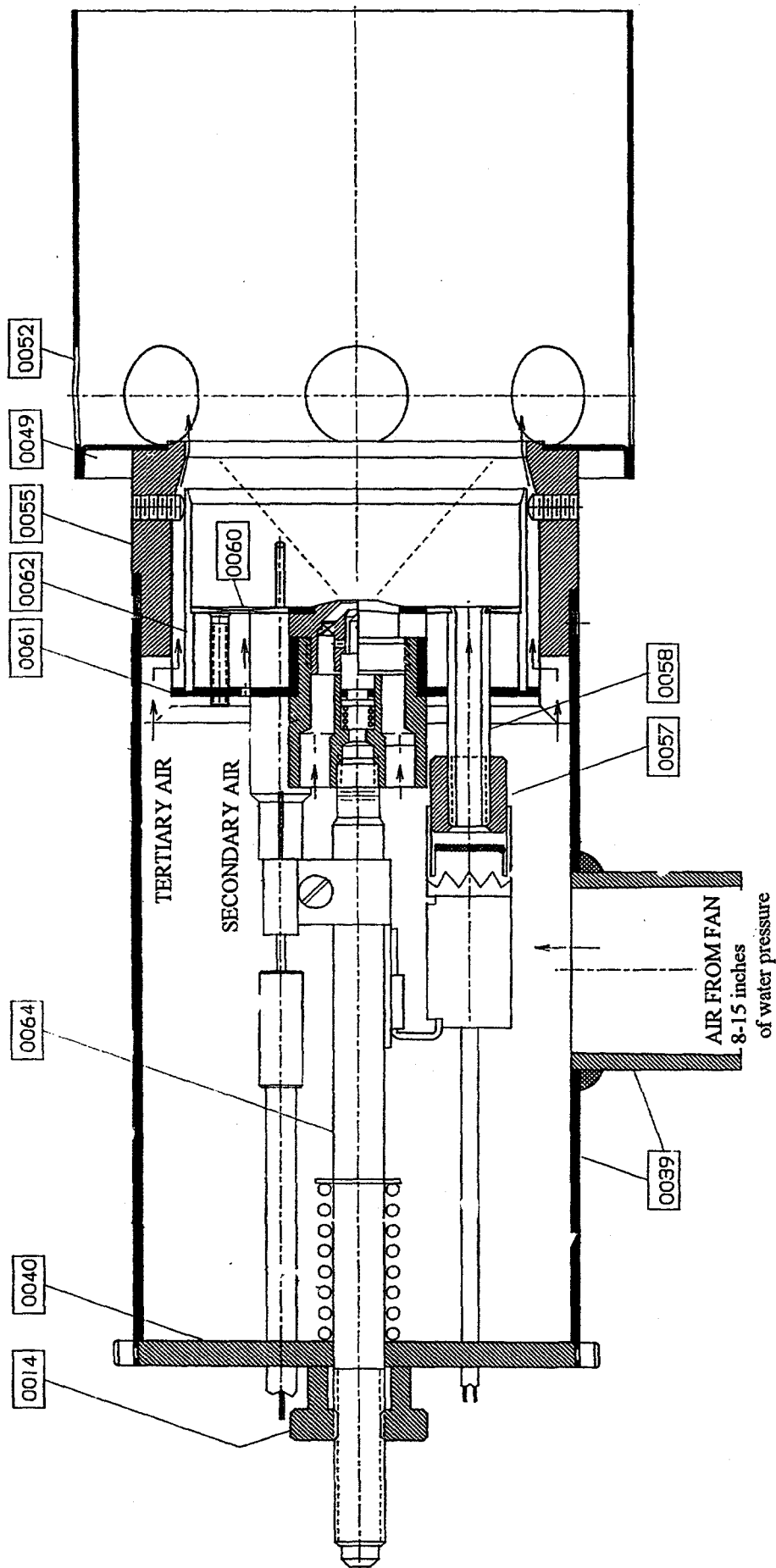


Figure 4-10. Side cross section of burner head designed for single air supply

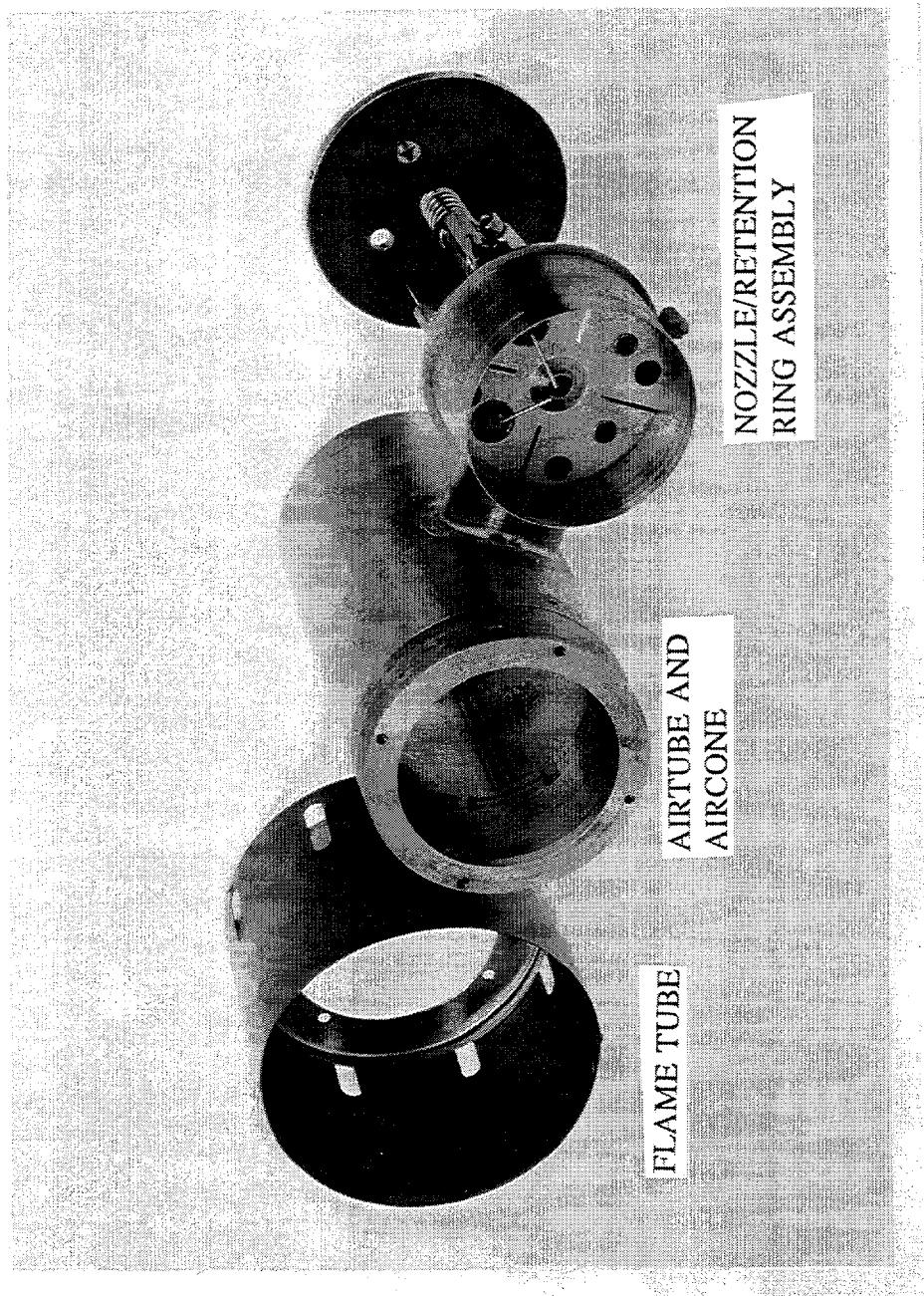


Figure 4-11. Photograph of burner head designed for single air supply. Major components disassembled.

The flow rate of secondary air is controlled by the total area of holes in the metering plate (Fig. 4-9 and part 0061 in Fig. 4-10.) As with the burner head designed for a split air supply and discussed in the previous section, the tertiary air flow rate is controlled by using the thumb nut (Fig. 4-9) to adjust the axial position of the nozzle assembly, diffuser, and connected parts. Referring to Figure 4-10, the tertiary air flow through the annular passage between parts 0062 and 0055. With the single air supply the tertiary air flow is controlled by restrictions both at the front and rear ends of this passage. The pressure drop is about equal between these two restrictions. Attempts to use only one restriction, at the front of the burner resulted in degraded combustion performance due to the high velocity at which the tertiary air enters the combustion zone.

With this version of the burner the added flame tube was used in all testing. A wide variety of flame tube dimensions were tested. Performance of the burner in the quartz combustion chamber was very good and Table 4-1 provides a summary of typical test results at low excess air levels with firing rates ranging from 0.22 to 0.7 gph. For all tests shown smoke number was zero.

Table 4-1. Typical Results of Steady State Tests of Burner Head in Quartz Combustion Chamber

Firing Rate gallons/hour	Air Pressure inches of water	Excess Air (%)	CO (ppm)	NOx (ppm)
.22	7.3	9.3	29	50
.30	7.0	6.2	30	50
.40	10	7.7	30	67
.50	10	7.7	16	69
.60	11	13.0	14	74
.70	14	13.8	14	77

5. DEVELOPMENT OF A BURNER SYSTEM USING THE LOW-PRESSURE ATOMIZING NOZZLE

In planning the configuration for a complete burner system tradeoffs, of course, must be made among component alternatives. The options considered and selected for each of the major components are discussed below. For most of the major components selection cannot be made independently but, instead, interrelationships between components must be considered. To help illustrate these relationships Figure 5-1 provides a schematic of one system configuration considered to be very promising. This configuration along with alternatives are discussed below.

Air Supply

As discussed above and shown in Figure 5-1 a single air supply was sought which could provide all of the air for atomization and combustion. Targeted firing rates for the burner range from 0.25 to 1.0 gallons of oil/hour (gph). Total combustion air flow requirements then range from 6.75 to 27.0 scfm, assuming a maximum excess air level of 20%. The pressure requirement for the air supply was set at 15 inches of water pressure maximum. Initial consideration was given to fans driven with conventional AC motors at 3450 RPM (nominal). Using this approach would allow a conventional fuel pump to be driven from one end of the motor as in current practice. The pressures required could be achieved at 3450 RPM using a two stage fan and system costs with this configuration are attractive (comparison below). However, the fan diameter required is large--10 inches. Such a large diameter would make the burner very difficult to apply in most furnaces and some boilers and this option was rejected for this reason. Regenerative blowers, also operating at 3450 RPM were evaluated and rejected due to cost and noise concerns.

An attractive alternative was found in brushless DC motor driven, high speed fans. Advantages of this approach include: small diameter (6 inches), low electric power consumption, and ability to easily modulate fan speed. These operate, however, at higher RPM and conventional fuel pumps cannot be used, directly driven from the shaft. With a single stage fan the speed is about 10,000 RPM. Using a two stage fan this is reduced to about 6,000 RPM. Units of this type are currently available commercially as complete systems, fan, motor, and all required controls to convert from 120V AC as an integrated package. Costs, however, are a concern. Depending upon the configuration selected the motor and fan system are estimated to cost, in quantity, from \$70 to \$140. Another option which might be considered is a high speed fan connected to a 3450 RPM AC motor through a belt drive. This has not been evaluated to date in this program simply because of concerns about maintenance requirements.

Fuel Supply

The fuel supply system includes the fuel pump and the arrangements made to control the fuel flow rate to the air-atomized nozzle. A very broad range of options were considered here. One class of options includes metering pumps. These are adjustable displacement pumps which can serve the dual function of delivering fuel and metering. Examples of this include rotary nutating disk

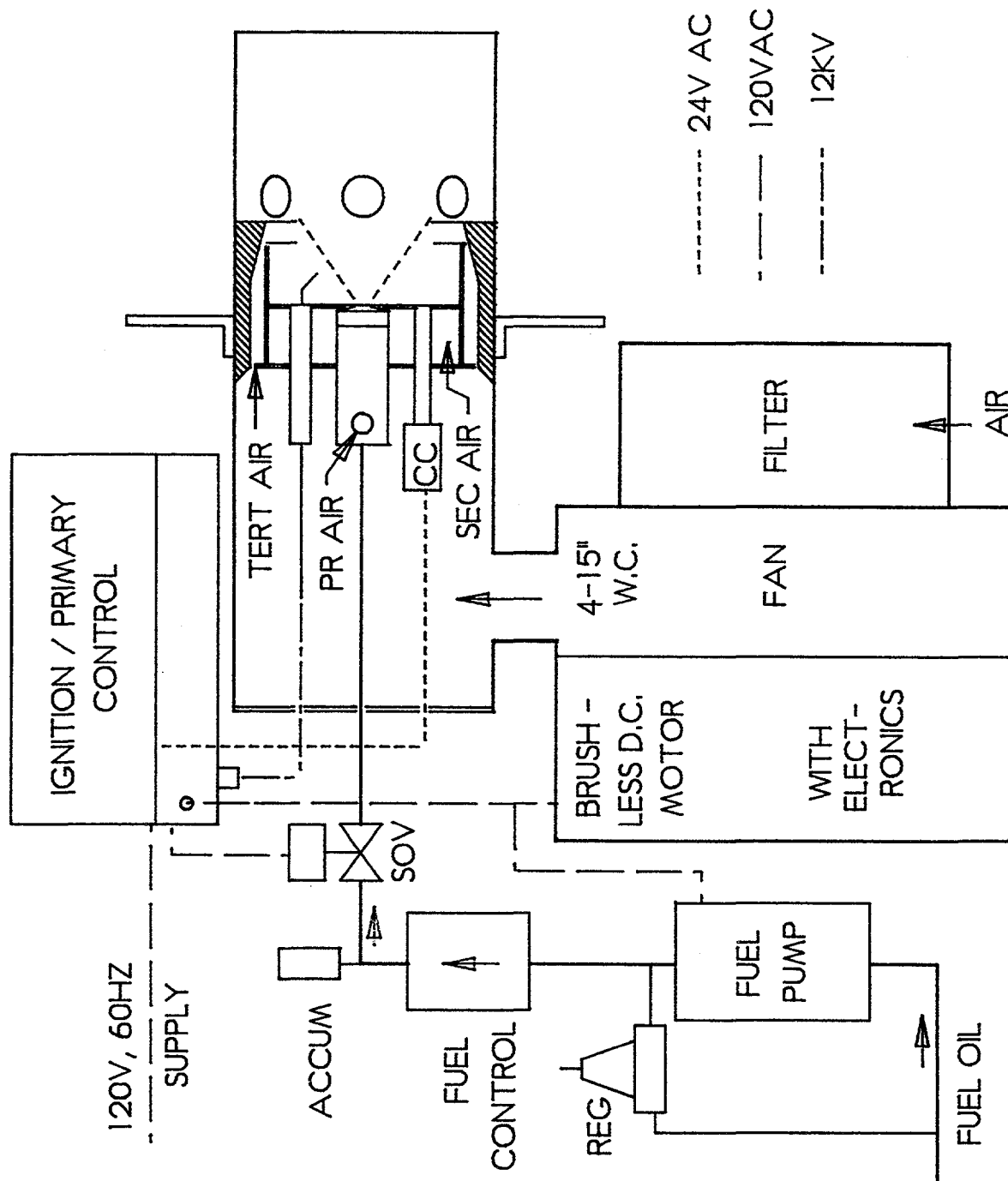


Figure 5-1. General schematic of burner system using low-pressure atomizing nozzle

pumps and variable stroke rate, pulsing piston pumps. The rotating disk pump has the advantages of low power consumption and ability to operate at high RPM. The variable stroke rate pulsing piston pumps would be driven by their own "solenoid" motor. With all types of metering pumps the electric power requirements are very low--20 watts or less and the projected costs are attractive. Perhaps the greatest disadvantage of pumps of this type is their limited displacement. During installation and initial operation it would require an extended time period to purge the air from the fuel supply system making it impractical for underground tank installations with lift requirements.

During the course of this work an interesting fuel pump option was developed by one manufacturer--Suntec Industries. This is a modification of a conventional fuel pump to provide the lift and displacement normally required for residential heating applications and also metering of oil to any low pressure air-atomizer. In consideration of our requirements Suntec developed a sample capable of operating at speeds to 6,000 RPM (at least). Additional future testing of this pump, integrated and balanced with a brushless DC fan system are planned.

While developing a pump which can be integrated with the air supply is considered the most attractive alternative it is difficult to accomplish this on a limited basis. In developing burner samples which can be distributed for manufacturer evaluation it has been decided to use a fuel pump/metering system independent of the fan drive. Presently this includes a small gear pump with pressure regulator providing constant pressure to an adjustable metering valve. The system illustrated in Figure 5-1 also shows a small fuel accumulator ("ACCUM" in Fig. 5-1) and a time delay solenoid valve ("SOL"). These two work together to provide a small "burst" of fuel at startup which assures consistent ignition. When the burner first starts the time delay solenoid valve is closed. During this period pressure builds in the accumulator to the maximum pump discharge pressure as set by the pressure regulator ("REG"). When the solenoid valve opens the stored pressure in the accumulator is released providing a surge of fuel to the nozzle. Typically, the maximum pump discharge pressure is adjusted to 10 psi and the fuel pressure downstream of the fuel control component shown in Figure 5-1 in steady state (after solenoid is open) is less than 1 psi.

The fuel control shown in Figure 5-1 is a simple needle and tapered seat. Testing has also been done using an automotive fuel injector in place of the needle valve. Injectors of this type are pulse width modulated needle valves which open and close at constant frequency (100 Hz) and serve to both meter and atomize gasoline into automotive intake manifolds. At high engine load the needle will stay open for all of its cycle (wide pulse width). At low loads the needle will stay open for only a very small fraction of its cycle (narrow pulse width). Because of its short cycle time (high frequency) the pulses of fuel flow have the same effect as a continuous fuel flow modulation. In the oil burner application these injectors would not atomize the fuel but simply act as metering valves for oil flowing to the air-atomizing nozzle. The primary advantage of this novel application of automotive injectors over needle valves is the ability to electronically control firing rate for modulation. In testing the injector was found to provide controllable fuel flow rates. During startup transients, however, the fuel flow rate was found to vary initially and this is really not acceptable. Additional effort would be required to address this startup concern and this is included in future development plans.

Controls

In conventional, retention head burners, cadmium sulfide photoconductors are used to sense ignition. The primary burner control uses this to prevent continued delivery of fuel into a combustion chamber in the event of an ignition failure. To apply such a conventional control to a burner which uses the low-pressure atomizer the flame must emit visible light (yellow flame). Some prior advanced burners emitted little visible light and required the development of special controls. With a cadmium sulfide photoconductor the resistance of the sensor decreases as the light intensity increases and burner primary controls typically sense a flame when the sensor resistance is below 3000-5000 ohms. In measurements with the developed burner head the cadmium sulfide photoconductor resistance was found to be just over 2000 ohms and the conventional control works well.

A conventional, interrupted high voltage ignition system was found to be acceptable with this burner.

Comparison of System Options

For the purpose of assisting with the design of a complete burner system a set of systems were defined with a broad range of configuration options. For each system estimates were made of the cost premium, relative to a conventional burner first on a component-by-component basis and then for the whole system. In addition estimates of electric power requirements were also made. The systems included in this comparison are listed in Table 5-1. In Table 5-2 results of cost estimates and total system electrical power is presented. Referring to these tables System 1 is a burner which has a siphon type atomizer as described in Section 2 of this report. In this case it is assumed that a conventional combustion air fan is used with a shaft speed of 3450 (nominal) RPM. Electric power consumption assigned to this fan is 70 Watts and this is based on discussions with manufacturers and product data for fan/AC motor packages which have the pressure and capacity required. For this system it has been assumed that the fan motor drives a vane type air compressor instead of a fuel pump as in a conventional, pressure atomized burner. The vane type air compressor is required to provide 1 cfm of compressed air at roughly 10 psi for the siphon nozzle. Based on manufacturer's data the add electric power load imposed by the vane compressor on the motor is 80 watts. It has been assumed for this system that the fuel pump is a separate, gear type pump with an integral AC motor. The specific pump assumed has been made in Japan by Taisan Company for use in small oil-fired heating equipment. A sample of this pump was tested at BNL and based on the results of these tests electric power draw of 28 watts was assigned. It should be noted that this system has some similarities to a burner developed by Davair Heating Limited in England. The Davair burner uses a siphon type atomizer and has a vane type compressor driven from the main fan shaft. Fuel feed, however is by gravity or external pump. In developing cost estimates for System 1 it has been assumed that the price of the fuel pump would be roughly similar to that of a conventional fuel pump and so there is no premium (Table 5-2). This assumption is based on discussions with U.S. burner manufacturers who have used this pump. The cost premium in this burner is for the atomizer and vane pump and estimates here have been based on discussions with component manufacturers. One possible area of concern with this system is the expected life of the vane

System Number	Burner	First Air Supply:				Fuel Pump:		Second Air Supply:	
		Type:	AC or DC	RPM	Power (W)	Type	Power (W)	Type	Power (W)
1	Siphon Nozzle	Conventional Fan	AC	3450	70	Taisan	28	Vane	80
2	Single Air Supply	2 Stage Cent Fan	AC	3450	140	Suntec	60	-	-
3	Single Air Supply	Regenerative Blower	AC	3450	135	Suntec	60	-	-
4	Single Air Supply	2 Stage HS Fan	DC	7000	100	Automotive Gear	24	-	-
5	Single Air Supply	1 Stage HS Fan	DC	9000	100	Automotive Gear	24	-	-

Table 5-1. Comparison of Burner Systems. Definition of systems and electrical power consumption.

System Number	Burner	Cost Premium Relative to a Conventional Oil Burner							Total Power
		First Air Supply and Motor	Fuel Pump	Control	Second Air Supply	Nozzle	Other	Total Cost Premium	Watts
1	siphon	\$0	\$0	\$0	\$50	\$5	\$0	\$55	178
2	single Air Supply	\$60	\$0	\$0	\$0	\$5	\$0	\$65	200
3	single Air Supply	\$135	\$0	\$0	\$0	\$5	\$0	\$140	195
4	single Air Supply	\$100	\$10	\$0	\$0	\$5	\$0	\$115	124
5	single Air Supply	\$100	\$10	\$0	\$0	\$5	\$5	\$120	124

Table 5-2. Comparison of Burner Systems. Costs and Total Power Requirements

air compressor and maintenance requirements. Estimates of expected life vary from 5000 to 10000 hours and, based on this, it seems likely that use of such compressors will increase service needs.

All of the other systems compared in Tables 5-1 and 5-2 include the use of the low-pressure air atomizing nozzle. In System 2 it is assumed that a two-stage centrifugal fan is used to develop the 12 inches of water pressure required by the atomizer. This fan, which provides all required air is driven with a conventional motor which has a double ended shaft. The fuel pump is driven from the second shaft extension, essentially as in a conventional retention head burner. The fuel pump assumed for this system has been developed by Suntec Industries specifically for delivering fuel to low pressure atomizers. Basic specifications for the fan were developed with assistance from a fan manufacturer. A great advantage of this system is the ability to use a conventional motor and nearly-conventional pump. The greatest disadvantage is the large diameter needed for the blower -10 inches. This would prevent such a burner system from being used in many common applications, simply because it could not fit in the space normally available. The total cost premium with this system is modest at \$65. Power requirements have been estimated at 200 watts which is similar to conventional burners.

System 3 is similar to System 2 except that the two stage fan has been replaced with a regenerative blower. This system can also use a conventional motor and fuel pump and the blower diameter is considerably smaller than with System 2. Major disadvantages with this system include system cost and concern about noise. These blowers will require an air inlet silencer.

In System 4 a high speed, two stage fan is used driven with a brushless DC motor. Major advantages of this system include small fan diameter (6 inches) and low electric power consumption. Disadvantages include cost and the difficulty in using a conventional fuel pump. The blower selected for this application and tested during the project operates at about 7000 RPM under expected conditions. The fuel pump which has been developed by Suntec Industries may be capable of operating at this speed but has not been tested. It is expected that the pump/fan combination would need to be balanced as a system which could make field service difficult, in the event, for example, of a pump failure. For this system it has been assumed that a small diameter, automotive type fuel pump would be used driven from the shaft of the DC motor. This pump has lower starting and running torque requirements than the conventional fuel pump but also has limited "lift" capability. Several pumps of this type were tested during this project. If the installation includes a buried fuel tank a pump of this type would not be acceptable because the pump could not self prime during initial startup or after a repair involving the fuel system. This pump was selected here because of its lower electrical power demand, small diameter, and ability to operate at higher RPM. System 5 is essentially the same as System 4 except that the two stage fan has been replaced by a single stage fan. The primary advantage of this approach is a smaller fan length. The primary disadvantage is increased noise.

Selection of Configuration and Description of System Prototype

In addition to the systems listed in Table 5-2 many other configurations certainly could be considered and many other options were considered to some

degree in this project. The system selected for prototype development was System 5 with some changes discussed below. Relative to systems 1 and 2 this approach involves higher cost. Relative to system 1, however, the use of a single fan to supply all of the combustion air represents a simplification and, in addition, the maintenance requirements of the brushless DC fan system will be much lower. It is anticipated that the service needs of a vane compressor make this option unacceptable. The second option was rejected simply because of the large diameter required for the fan. In the future, however, in some specific applications where only lower firing rates and lower air pressures are required or in applications which are not space limited. Advantages of System 5 which most influenced its selection include the small size and low electric power consumption. In the development of the prototype a decision was made not to integrate the fuel pump with the fan but, instead to have the fuel pump driven with a separate motor. This configuration is not considered to be optimal but was used primarily for simplicity. For future systems the use of multistage fans as in System 4 certainly should be considered. The lower speed of such a multistage fan may allow the pump to be directly driven from the fan motor, eliminating a component.

Figure 5-2 is a photograph of the fully assembled prototype burner. A sheet metal case encloses most of the burner components. On the top is mounted a conventional primary control and ignition transformer (R.E. Phelon Co. Model 40100-02). An automotive type air filter extends through the right side (viewed from rear) of the casing. This filters all inlet air to the fan to prevent contaminants from fouling the atomizer. The back panel of the burner can be removed for service and adjustments. The left side panel contains all of the fuel supply system components, except for the fuel solenoid valve, and is also removable. Figure 5-3 is a photo of the prototype burner with the rear panel removed. Figure 5-4 is a photo of the interior of the left side panel only. Figure 5-5 is an illustration of the oil feed system and this figure also generally illustrates the electrical power supply. The burner's electrical system is shown in much greater detail in Figure 5-6.

Beginning with Figure 5-5 and the fuel feed system, the fuel connection to the prototype burner is a standard 1/4 inch flare. An inlet line filter (Delavan Model 60046) is used as a precaution against pump damage. On the prototype a rotameter with an integral inlet needle valve is installed primarily as a convenience for testing at varied firing rates. The inlet needle valve is not used to control fuel flow rate but is left fully open during normal operation. When the burner is being moved between installations, the rotameter inlet needle valve can be closed to prevent oil dripping. The pump being used is a small gear pump driven with a DC motor (Greylor Co. Model PQ-12DC). The pump has plastic (Delrin) body and gears, stainless steel motor and idler shafts, a BUNA-N lip seal and a Teflon gear face diaphragm. The pump is nominally rated for 12 VDC and current draw is 1.8 amps at 12 volts and 10 psi discharge pressure. In the prototype burner, the motor is supplied with 6 volts. This reduced voltage was found to be adequate and produced less pump noise than the full rated voltage. The 6 VDC supply is provided using a 120/6 volt transformer and bridge rectifier (see Figure 5-6). Following the fuel pump is discharge pressure regulator. This is an automotive component normally set to 10 psi. A small needle valve downstream of the regulator is used to adjust burner firing rate. This needle valve can be adjusted while the burner is operating by removing the casing rear

panel. The needle valve is mounted on a copper block which contains a small flow accumulator. This is a 1/8" x 2" hole in the block closed at the top to maintain an air bubble. The needle valve, accumulator, and time delay solenoid valve act together to provide a very small "burst" of fuel during startup which has been found to ensure consistent ignition at the lower firing rates.

The fan used in the prototype burner has an integral brushless DC motor and control to convert 110 V AC input (Amtek Model 116643-E). In this model, fan speed is controlled by an external 0-10 V DC input signal with current demand of about 20 ma. In the prototype burner, this speed control signal is provided using a regulated power supply and digital potentiometer. For the power supply, an integrated circuit component is used (Harris semiconductor HV-2405 F, Figure 5-6). The digital potentiometer is mounted on the casing so that fan speed can be adjusted externally while the burner is operating.

The primary control on the prototype burner is installed to only close the solenoid in the event of a loss of ignition, but the burner fan and fuel pump remain powered. This is simply a convenience, allowing adjustments to be made in fan speed and pump pressure if desired during testing. In normal operation, a firing rate change can be made by adjusting the needle valve, changing fan speed using the digital potentiometer, and also adjusting the burner diffuser position relative to the head using the thumb nut at the back of the burner air tube. These adjustments can all be made with the rear panel of the burner removed. Recommended settings for the rotameter reading, digital potentiometer setting, and head position have been developed to facilitate testing over a firing rate range. The packaged prototype burner has been named the Fan Atomized Burner or "FAB."

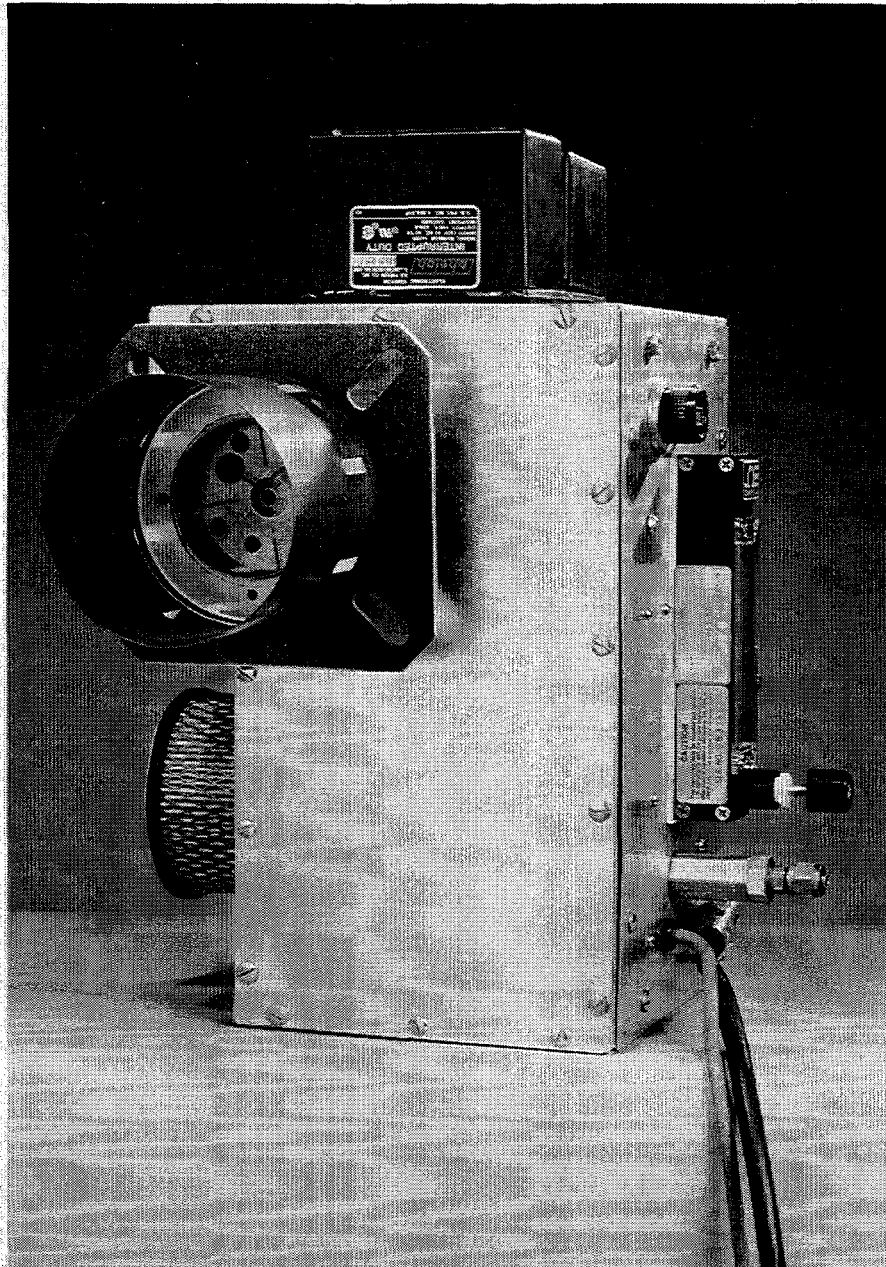


Figure 5-2. Photograph of fully assembled prototype burner.

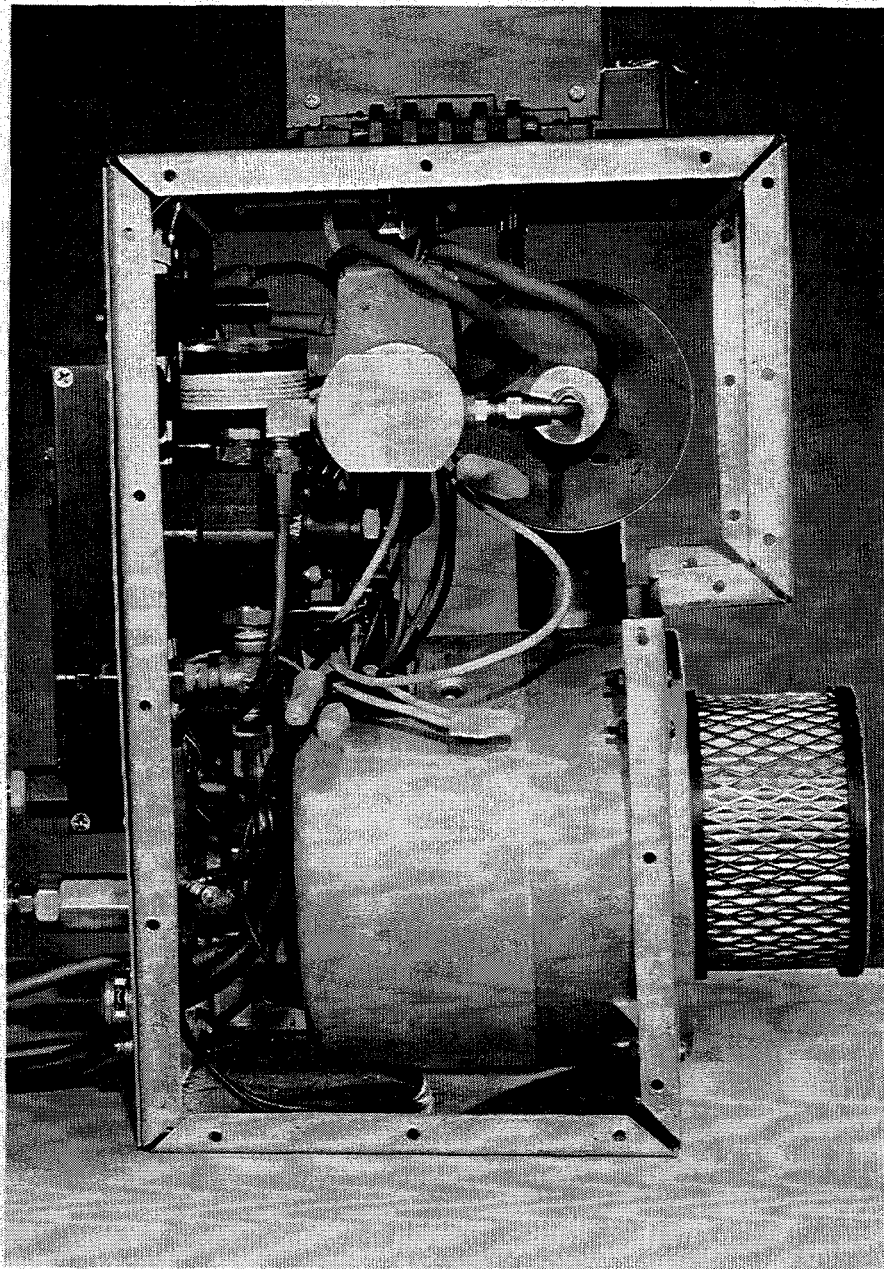


Figure 5-3. Photograph of prototype burner with rear panel removed.

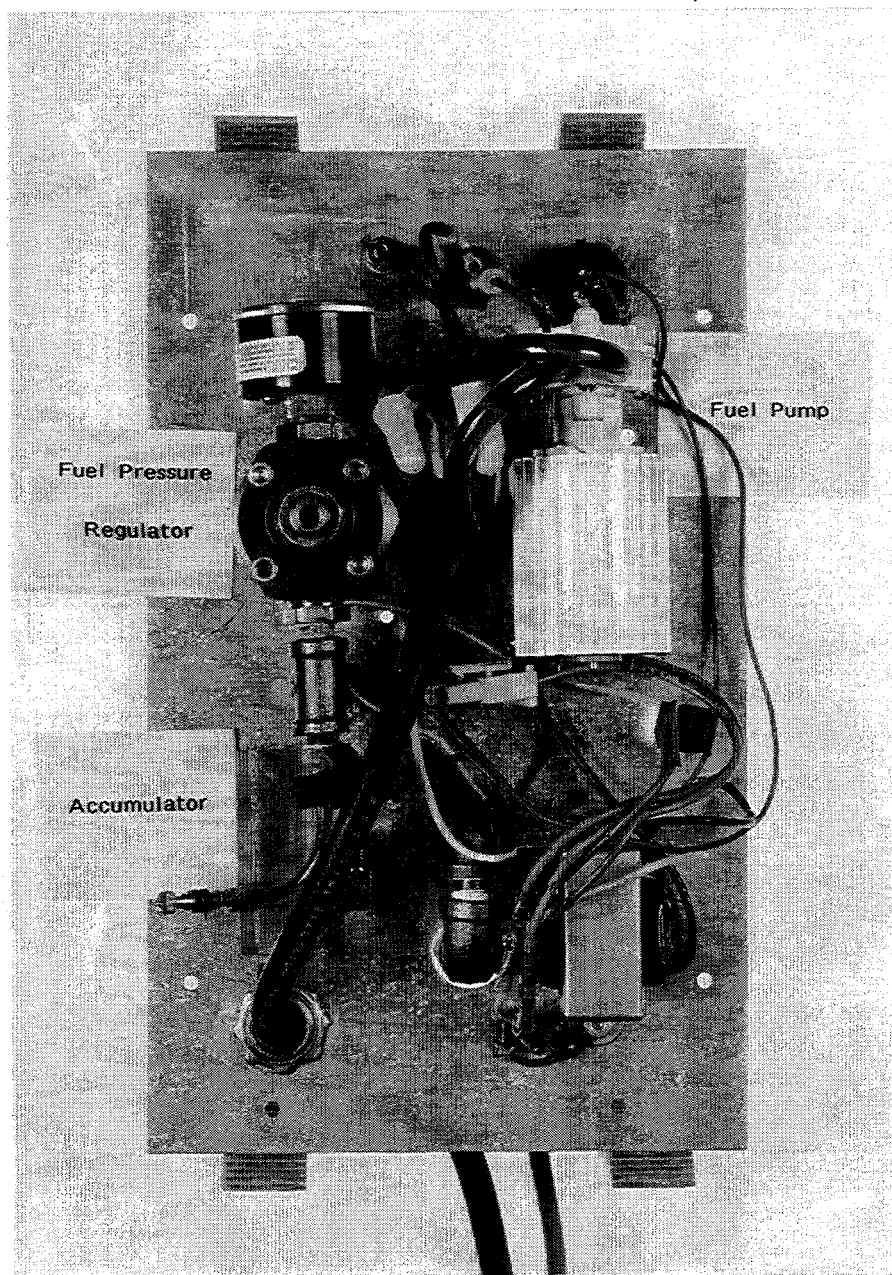


Figure 5-4. Photograph of interior of left side panel.

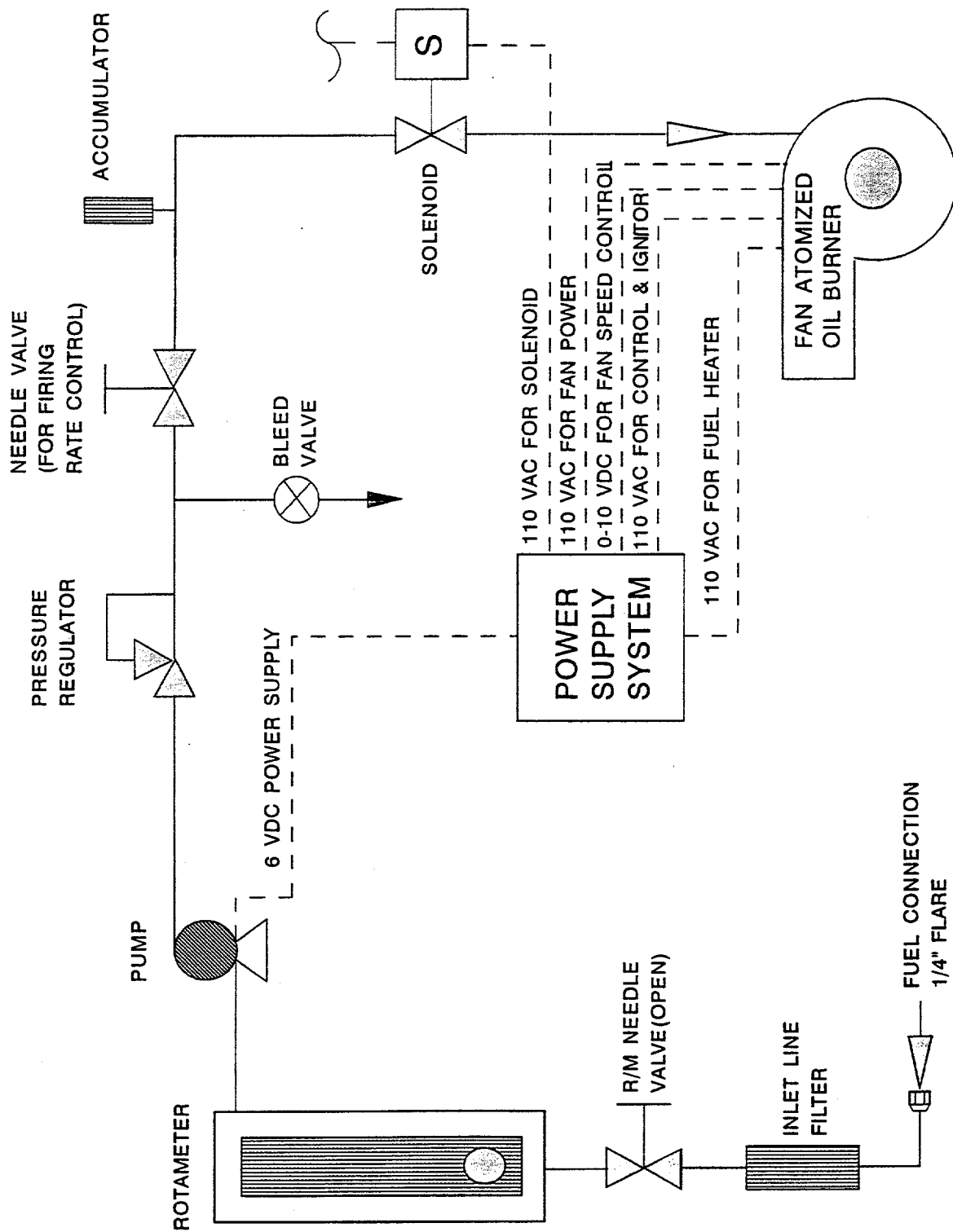


Figure 5-5. Illustration of prototype burner oil feed system.

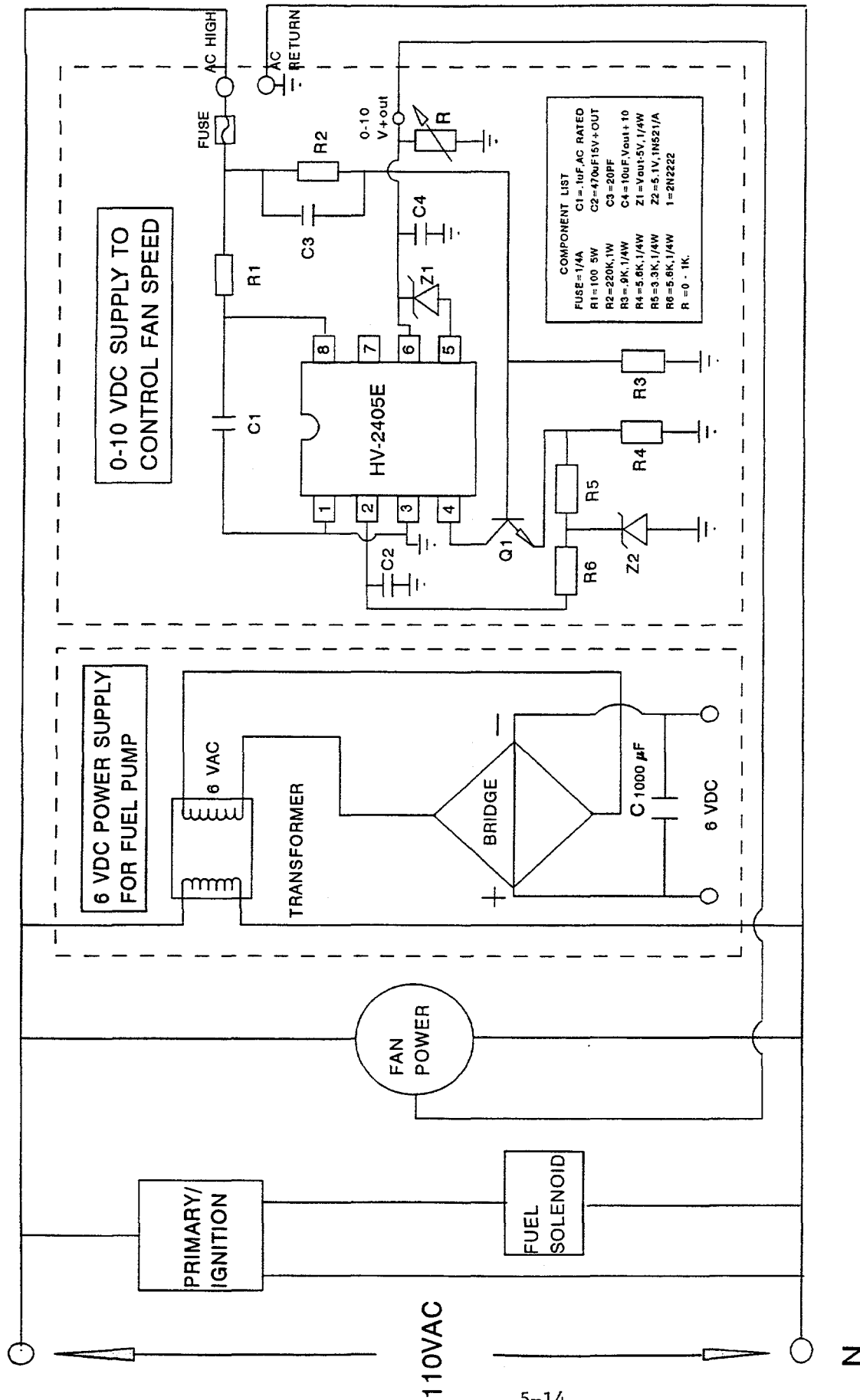


Figure 5-6. Details of prototype burner electrical system.

6. APPLICATION TESTS WITH THE FAN ATOMIZED BURNER

As discussed in Section 5, during most of the burner development work the burner was fired into a test chamber made of a horizontal, uncooled quartz cylinder with refractory front and back faces. To evaluate performance in more practical chambers a series of application tests were done with a selected range of boilers. This includes wet and dry base boilers with chamber environments considered to range from very hot to "cool", based on size and amount of refractory lining used in the chamber. For all boilers the testing was done in steady state. In addition, some testing was also done in selected boilers with a range of on/off cycling patterns. The application tests were done over a period of about 6 months. During this period some details of the burner head were continuing to evolve. Details are discussed below for each specific boiler.

For most of the boilers studied two types of tests were done. First the excess air was fixed at a moderate level, 15% (13.5% CO₂) and the atomizing air pressure was varied by adjusting fan speed. In this case increasing atomizing air pressure requires a reduction of the tertiary air flow to maintain the constant excess air level. In the second type of test the atomizing air pressure was fixed and the excess air was varied over a wide range. Generally the lowest excess air levels are of the most interest. However, these tests were conducted even at very high excess air levels to evaluate possible problems which might occur if the burner were improperly adjusted. It would be best if the burner were "forgiving" i.e. able to operate over a very wide range of conditions without problems. In all tests for which results are presented here the smoke number was zero. In all cases the flue draft was adjusted to be very slightly positive to eliminate effects of air infiltration. Unless otherwise specified the tests were done with a nozzle air swirler which had a slot width of 0.125" and a swirler angle of 35 degrees (see Figure 3-3). For some boilers, as discussed below the tests were done with the burners flame tube removed. In all other cases the flame tube was installed. During the course of these application tests an important change was made in the burner head to increase the maximum size of the tertiary air openings so that higher firing rates could be achieved without using high air pressures. Some of the boilers were tested before this change was made and in these cases higher firing rates were only tested with high atomizing air pressure.

The top part of Figure 6-1 shows the results of the first tests done on a three section, cast iron boiler with a full refractory liner in the combustion chamber. At 0.4 and 0.5 gph the burner operates well with atomizing air pressure over about 9 inches of water. The CO at 0.3 gph was high in these tests and requires over 11 inches of water pressure to come below 50 ppm. At 0.7 gph CO is low although only higher air pressures were evaluated for reasons discussed above. The bottom part of Figure 6-1 shows the NO_x emissions in the same test series. NO_x clearly increases with increasing firing rate, as would be expected and no consistent trend with atomizing air pressure can be seen.

Figure 6-2 shows the effects of excess air on CO and NO_x in the same boiler with atomizing air pressure held fixed at 12 inches of water. At 0.5 gph the burner has a very wide range of excess air over which operation is acceptable. At 0.3

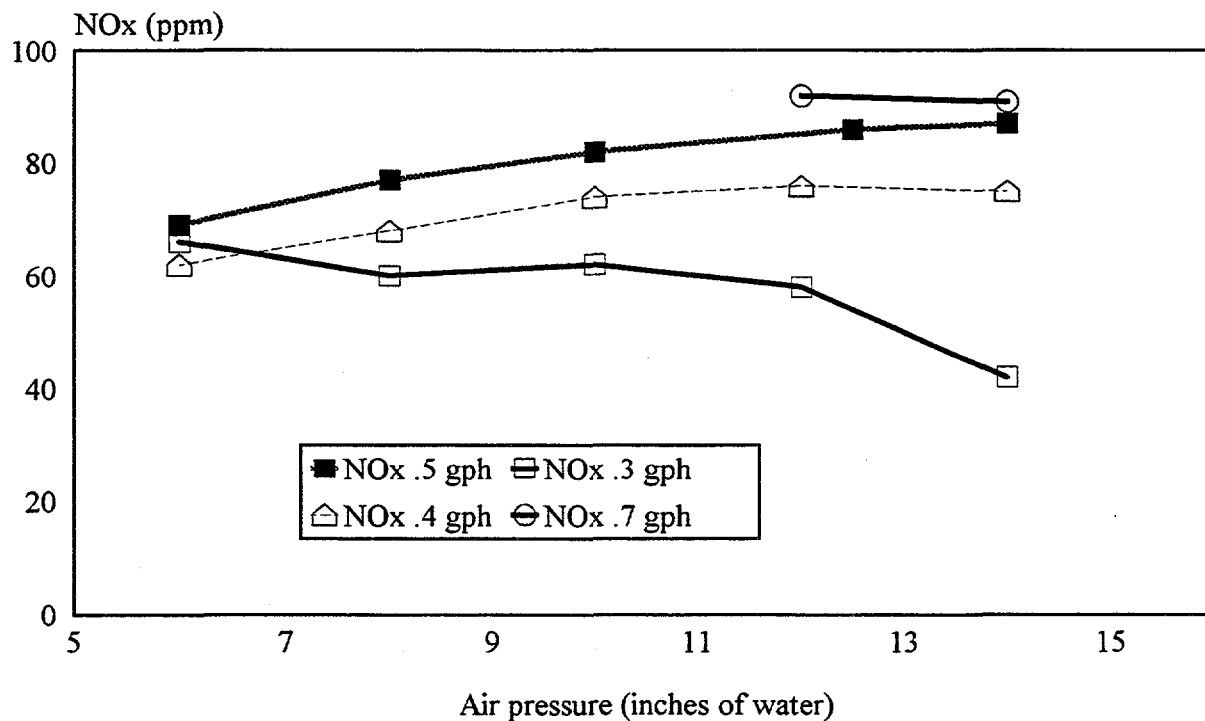
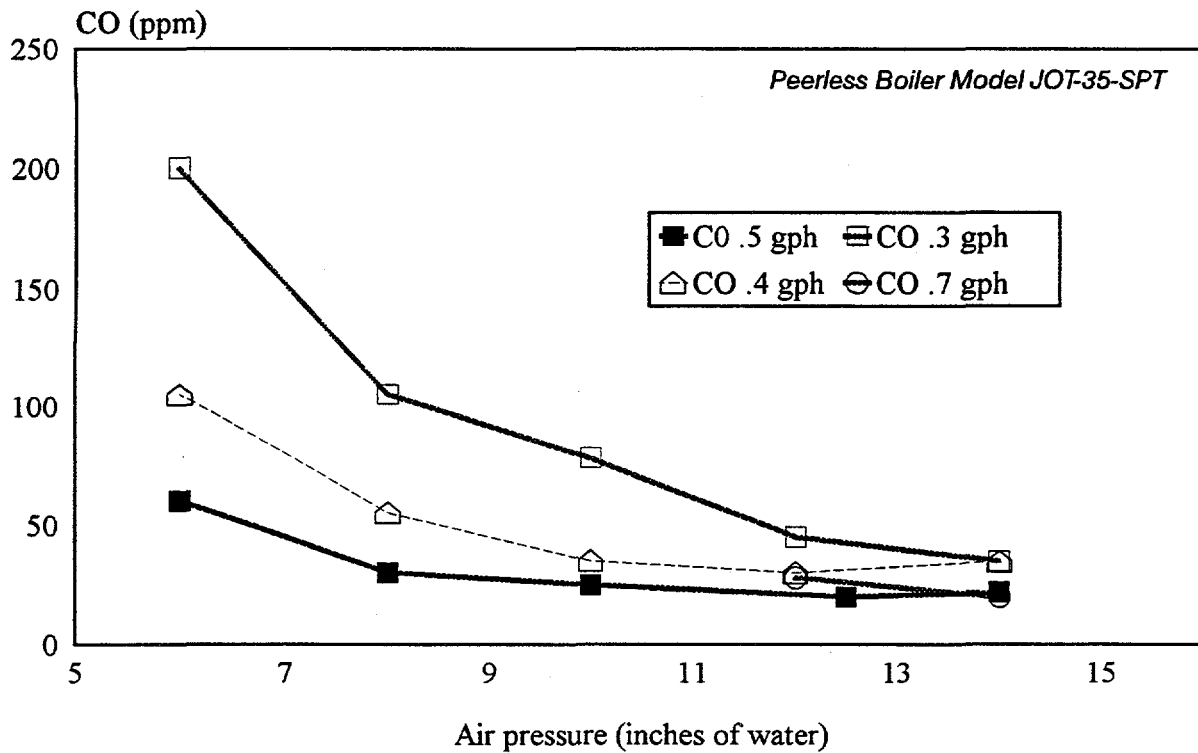


Figure 6-1. Tests with Fan-Atomized Burner in three section, cast iron boiler. Effect of air pressure on CO and NOx at constant excess air level (15%).

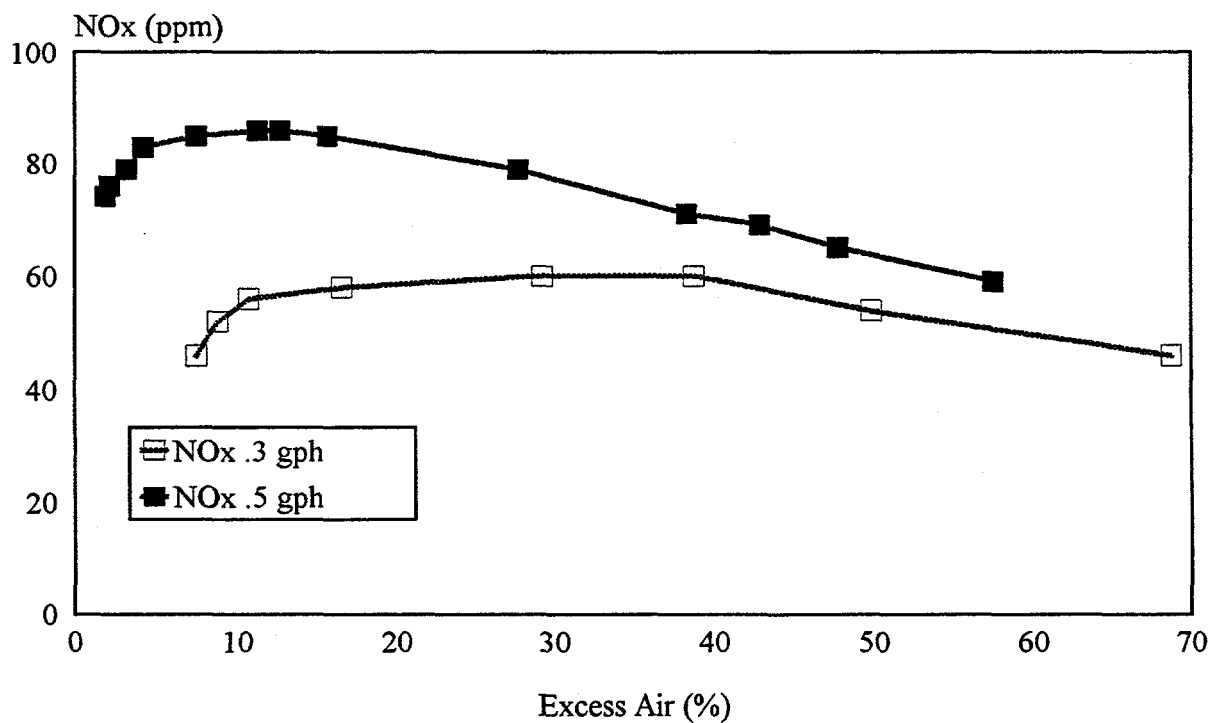
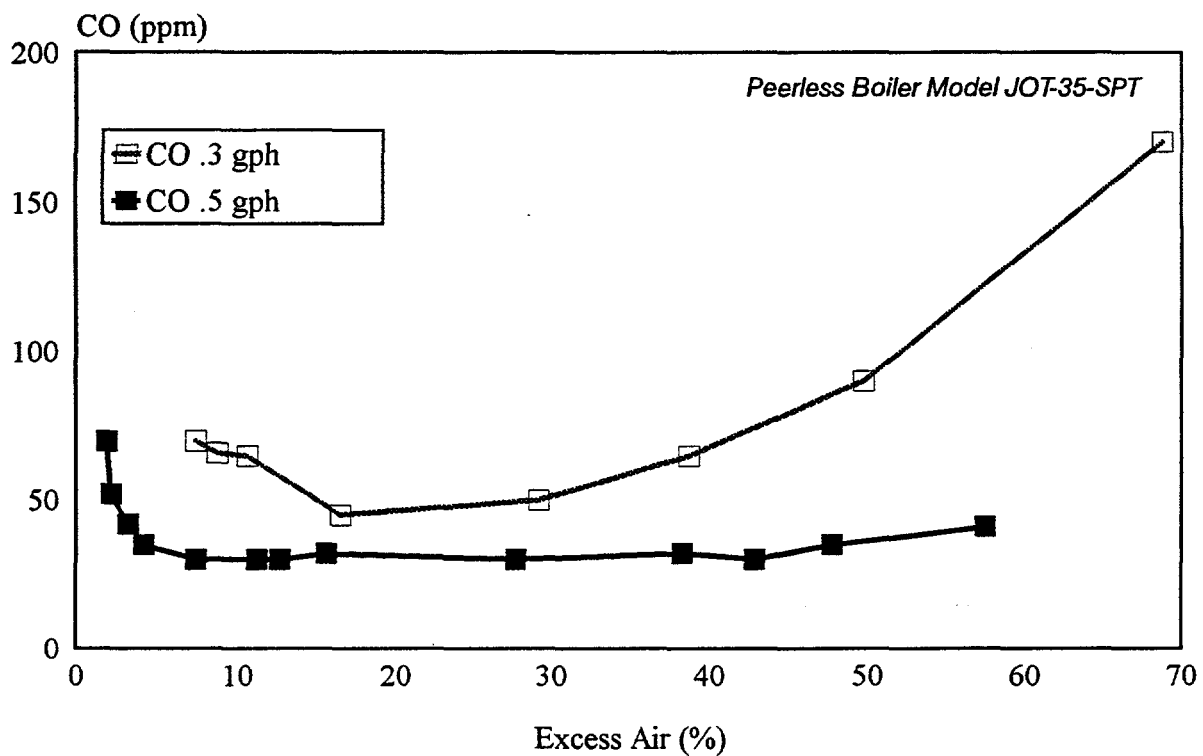


Figure 6-2. Tests with Fan-Atomized Burner in three section, cast iron boiler . Effect of excess air on CO and NOx at constant air pressure (12 inches of water).

gph the range is much narrower. This trend at 0.3 gph is not really acceptable and this firing rate should not be used in this chamber with the burner design tested.

The next boiler tested was a dry-base, steel unit. Figure 6-3 shows the results of the tests on the effects of atomizing air pressure on CO and NOx. Here the CO emissions are consistently low over the entire range tested and NOx is higher (relative to the three section cast iron boiler) as a result of the higher combustion chamber temperature. Figure 6-4 shows the excess air effects in the dry base steel boiler at constant atomizing air pressure. The burner operates with very low CO over a wide range of excess air.

Figure 6-5 shows the results of tests of atomizing air pressure done in the next boiler - a steel, horizontal fire-tube boiler with a cylindrical, horizontal combustion chamber. This cooled chamber has a refractory liner for about half of its length and the burner fires on axis. Figure 6-6 shows the effects of excess air on performance at constant air pressure. The excess air tests at 0.3 gph were done with atomizing air pressure set to 8 inches of water. At pressures much over 10 inches of water flame stability became a concern at this firing rate. For the conditions shown in Figure 6-6 the burner operated very well over a broad excess air range.

Some testing was also done in a two-section, cast iron boiler. Boilers of this type are considered to be particularly difficult to fire in to because of the very short combustion chamber. The boiler tested has a refractory target wall but no refractory liner on the sides or bottom of the cooled combustion chamber. A rather comprehensive test series was done with varied atomizing air pressure, excess air, firing rate, and atomizer swirl angle. Generally good performance could be obtained only at firing rates over 0.5 gph. Below 0.5 gph good performance could only be obtained over a very limited range of conditions. These results are illustrated in Figure 6-7. The tests at 0.7 gph were done with atomizing air pressure at 12 inches of water and a nozzle air swirler which had a slot width of 0.125" and a swirler angle of 30 degrees. At 0.5 gph the nozzle swirler angle was 35 degrees. Some tests were also done with a smaller swirler slot width - 0.095". This smaller slot width would essentially reduce the atomizer air flow and increase the tertiary air flow at a given level of excess air and air pressure. CO was slightly lower with this swirler. To improve performance at the firing rates below 0.5 gph several alternatives were evaluated. The most effective was the addition of a refractory blanket covering the exposed cast iron surfaces on the bottom of the combustion chamber. This change produced a dramatic reduction in the CO emissions and very good performance was observed even at the very lowest firing rates. Figure 6-8 shows CO and NOx as functions of excess air with the refractory blanket. The burner operates very well in this case over a broad excess air range. The addition of the blanket increased NOx emissions by about 20 ppm but was found to have no effect on the flue gas temperature leaving the boiler indicating no significant efficiency penalty. For all of these tests the swirler slot width was 0.125" and the swirler angle was 35 degrees.

Figure 6-9 shows the results of some tests which were done using another wet base steel boiler. This unit has a very heavy refractory liner in the combustion chamber which completely surrounds the flame. Results are shown for 0.3 and 0.5

gph firing rates with constants air pressure and varied excess air. CO emissions are generally low and flame stability was excellent. NO_x emissions are high as a result of the extensive chamber liner. For these tests the burner's flame tube was removed due to concerns about possible overheating and damage to the stainless steel tube. During testing no signs of any heat damage were observed to any burner parts. Testing with the tube to evaluate the possibility that it could be used with this chamber under at least low firing rate conditions was not done.

Figure 6-10 shows the results of combustion tests done in another two-section cast iron boiler. These tests were done after the modifications were made to the burner head which allowed operation at higher firing rates with moderate air pressures. In these tests air pressure ranged from 10 to 12 inches of water. At 0.3 gph the CO is higher than desired but at all other conditions performance is very good. For this two section cast iron boiler Figure 6-11 shows the steady state boiler exit temperature over the range of firing rates and excess air levels tested. At firing rates below about 0.5 gph the stack temperature is too low to vent these combustion products using conventional chimney systems. Condensation problems would result. For this reason it is unlikely that the burner would be used in a boiler of this type at such low firing rates. Trends in stack temperature were not shown for other boilers tested but were generally similar.

Generally, the transient performance of the burner was found to be good in all applications tested although in some cases ignition from cold conditions was found to be difficult at firing rates at and below about 0.3 gph. If the burner is to be used in applications requiring such low firing rates some changes may be required to assure consistent ignition. These may include reducing the flame tube diameter and modifying the chamber dimensions. Generally a smaller chamber is better. These changes might limit the highest firing rate range which could be achieved with the burner.

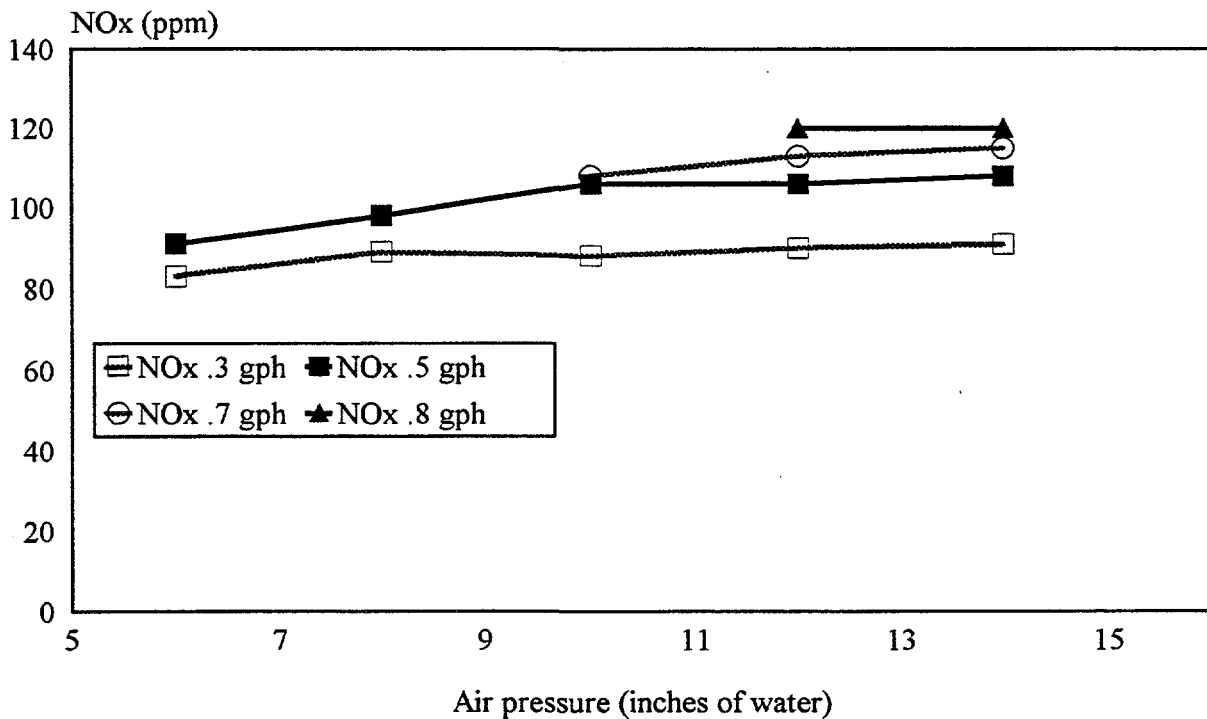
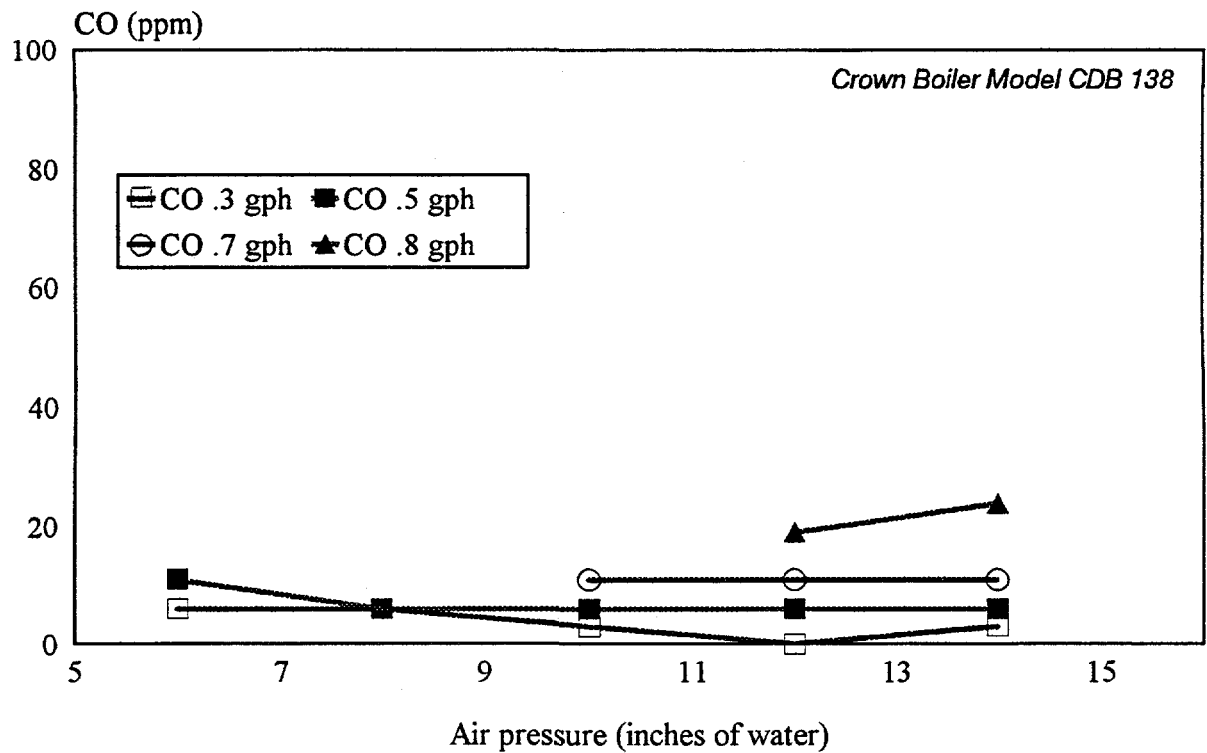


Figure 6-3. Tests with Fan-Atomized Burner in dry base, steel boiler. Effect of air pressure on CO and NOx at constant excess air level (15%).

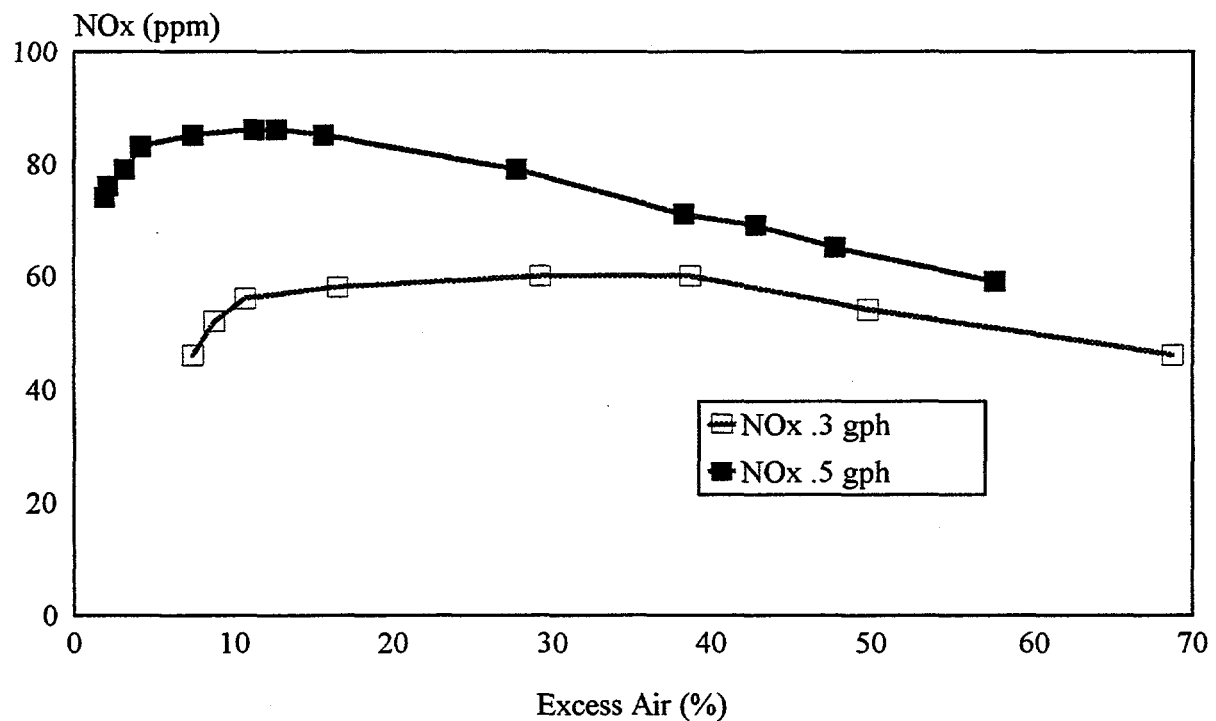
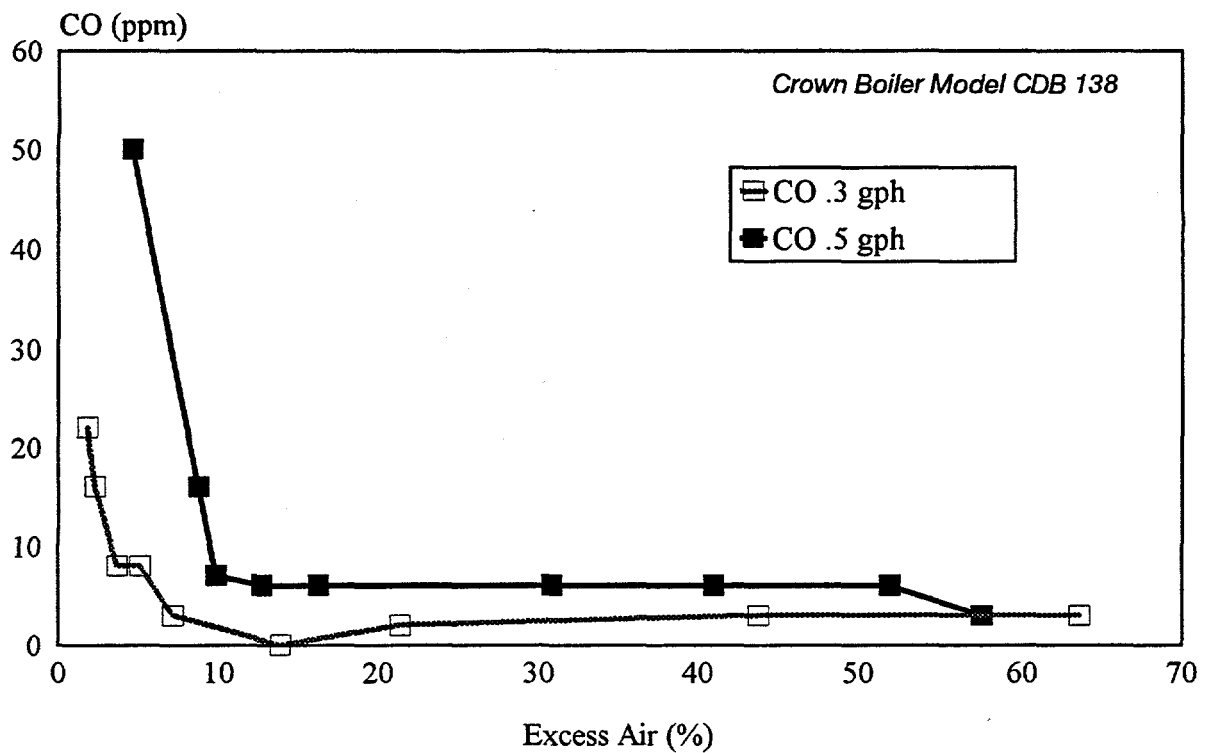


Figure 6-4. Tests with Fan-Atomized Burner in dry base, steel boiler . Effect of excess air on CO and NOx at constant air pressure (12 inches of water).

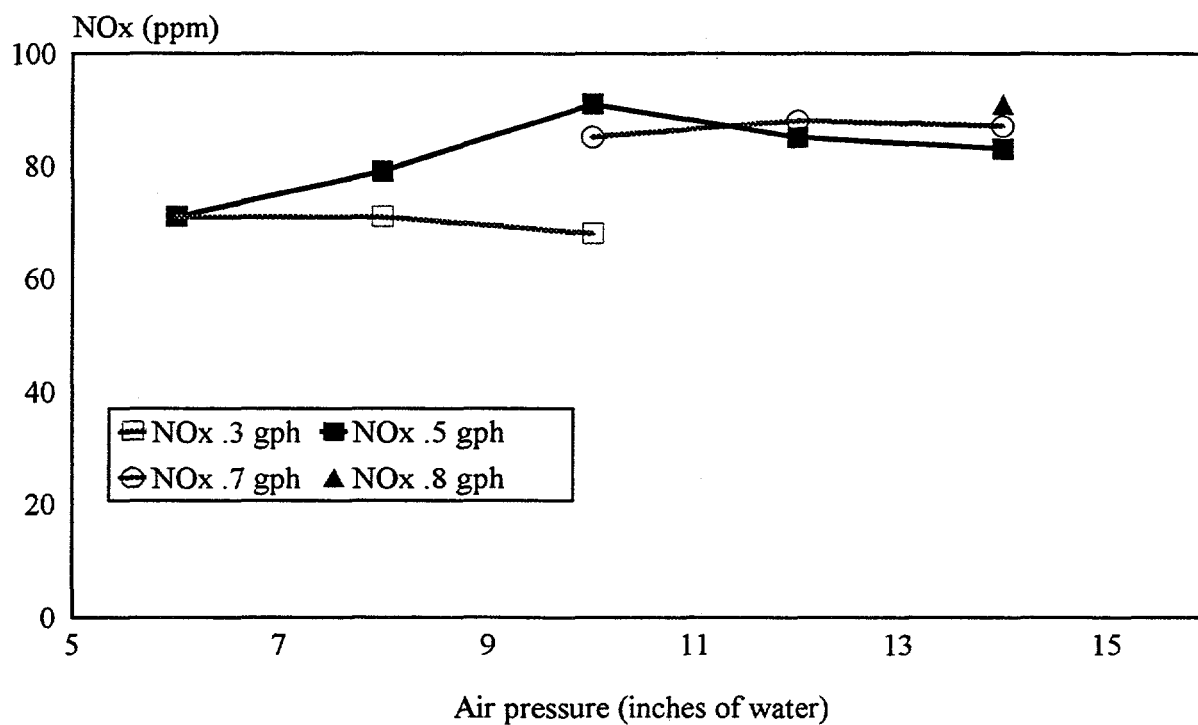
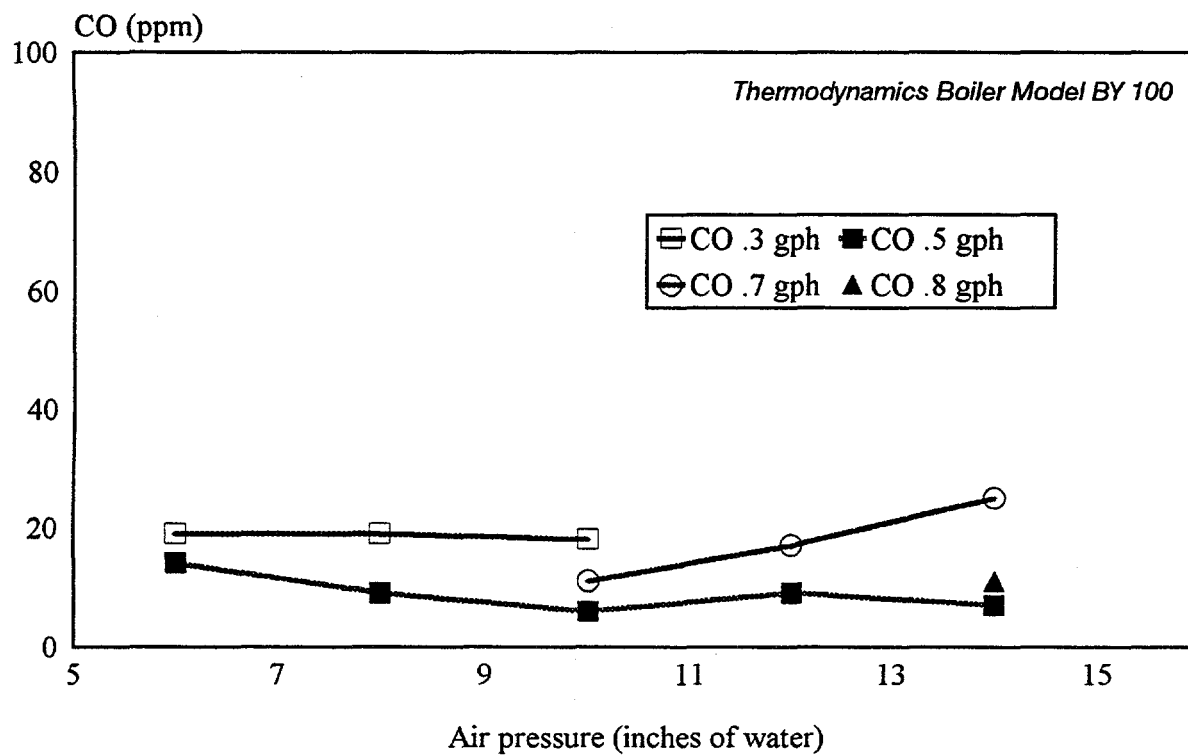


Figure 6-5. Tests with Fan-Atomized Burner in steel, horizontal firetube boiler . Effect of air pressure on CO and NOx at constant excess air level (15%).

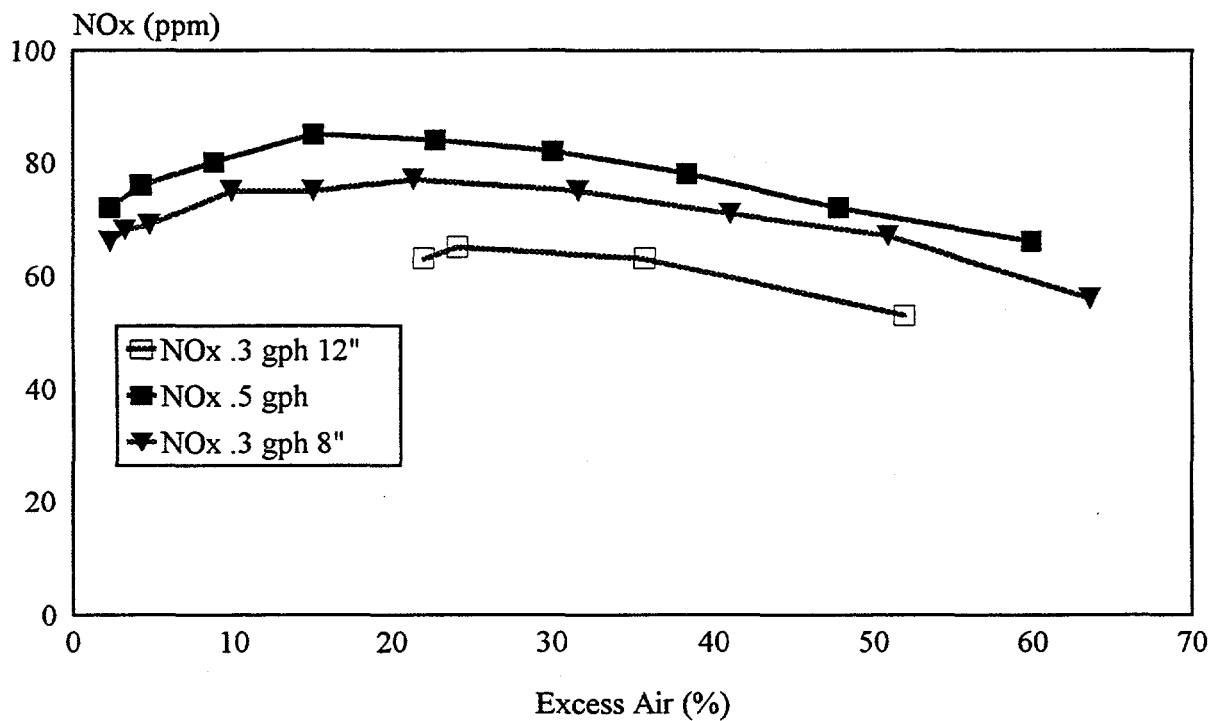
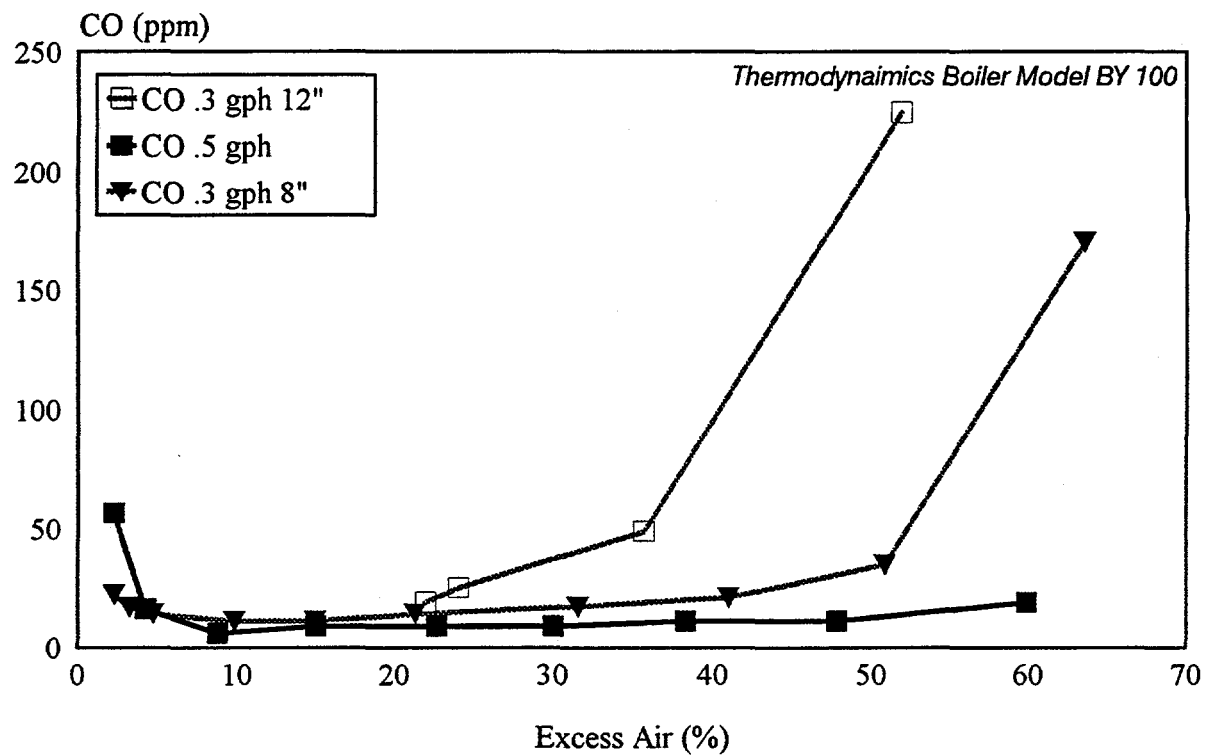


Figure 6-6. Tests with Fan-Atomized Burner in steel, horizontal firetube boiler . Effect of excess air on CO and NOx at constant air pressure . Tests at .5 gph were done with 12 inches of atomizing air pressure. At .3 gph tests were done at both 8 and 12 inches of water.

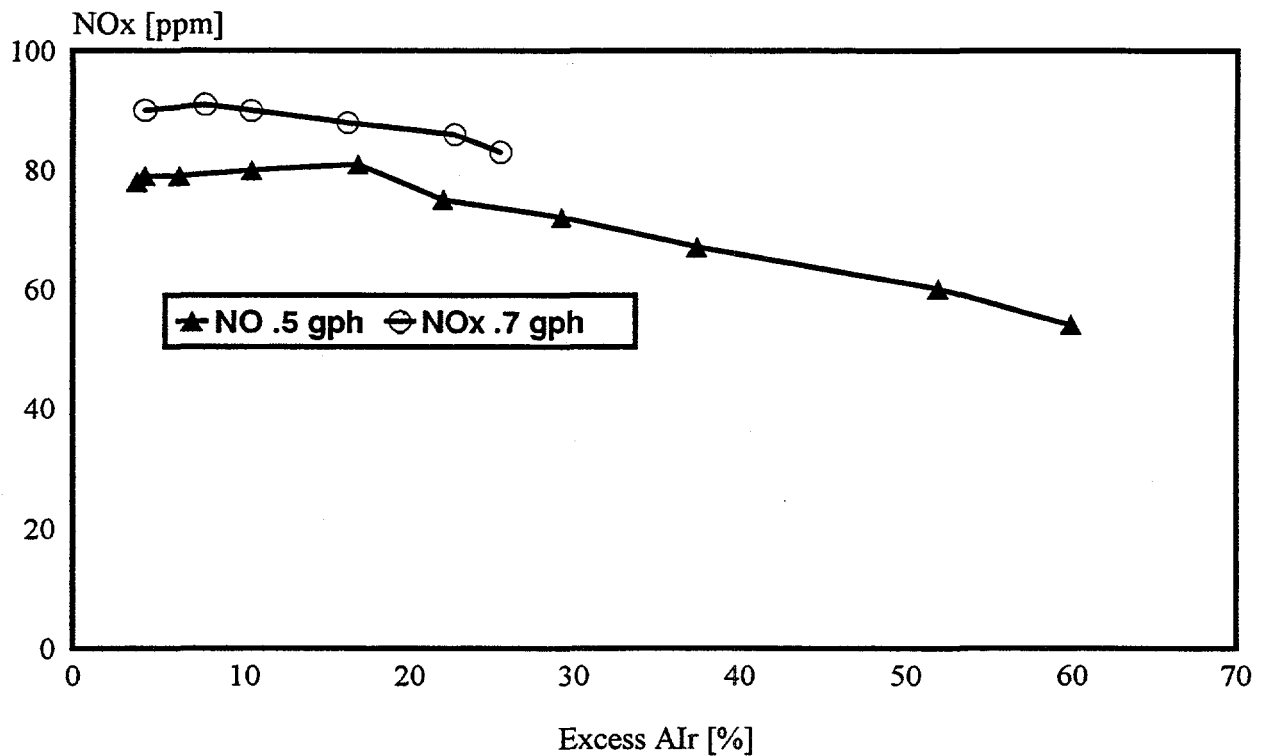
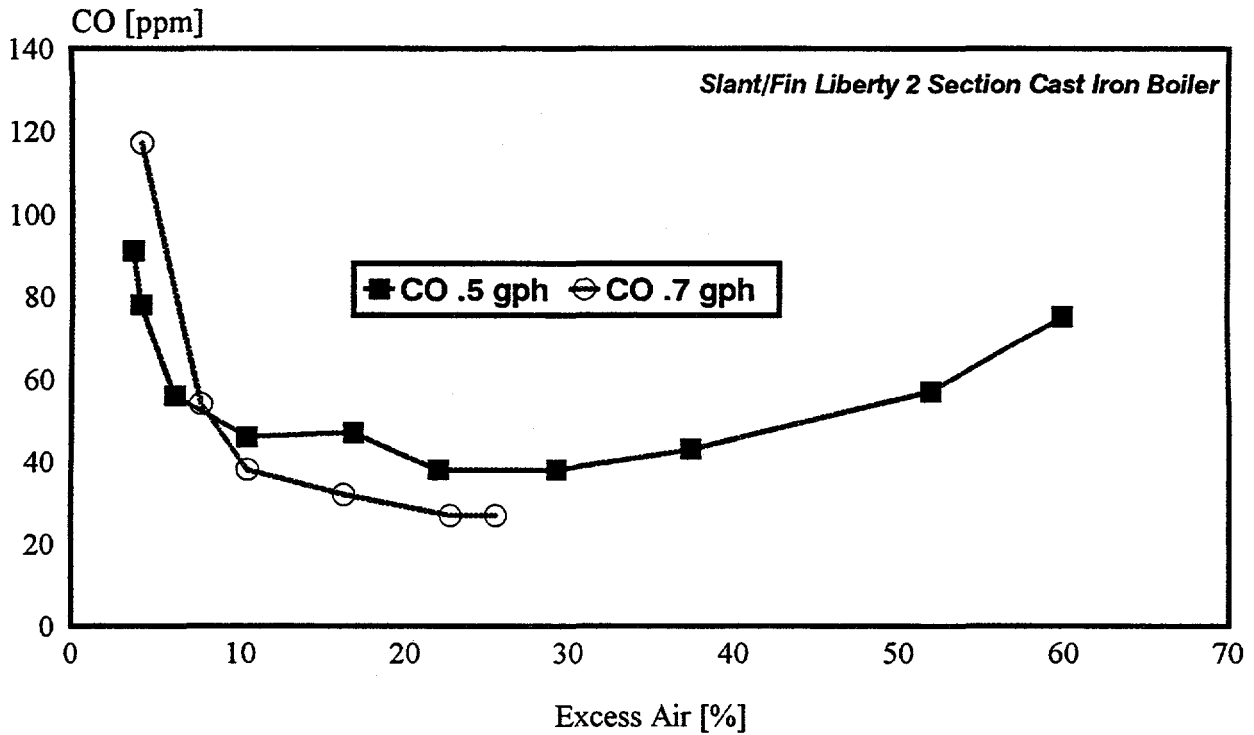


Figure 6-7. Tests with Fan-Atomized Burner in two section cast iron boiler. Effect of excess air on CO and NOx at constant air pressure .

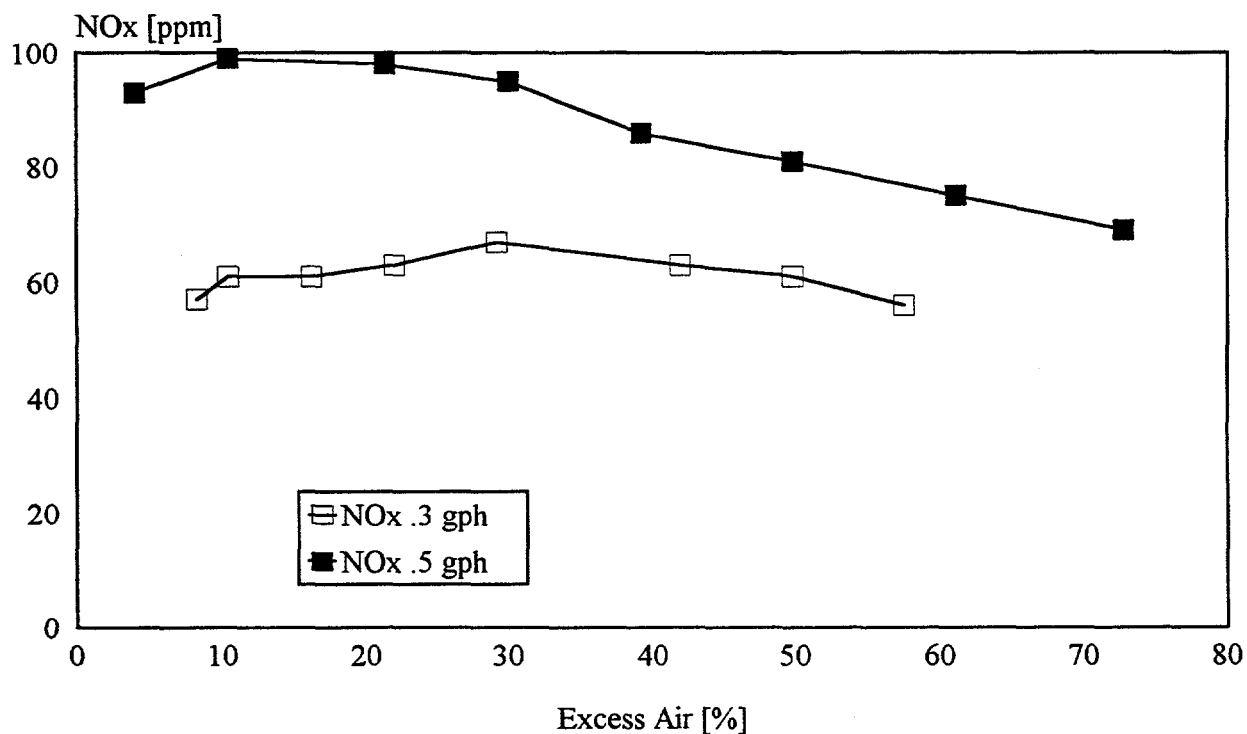
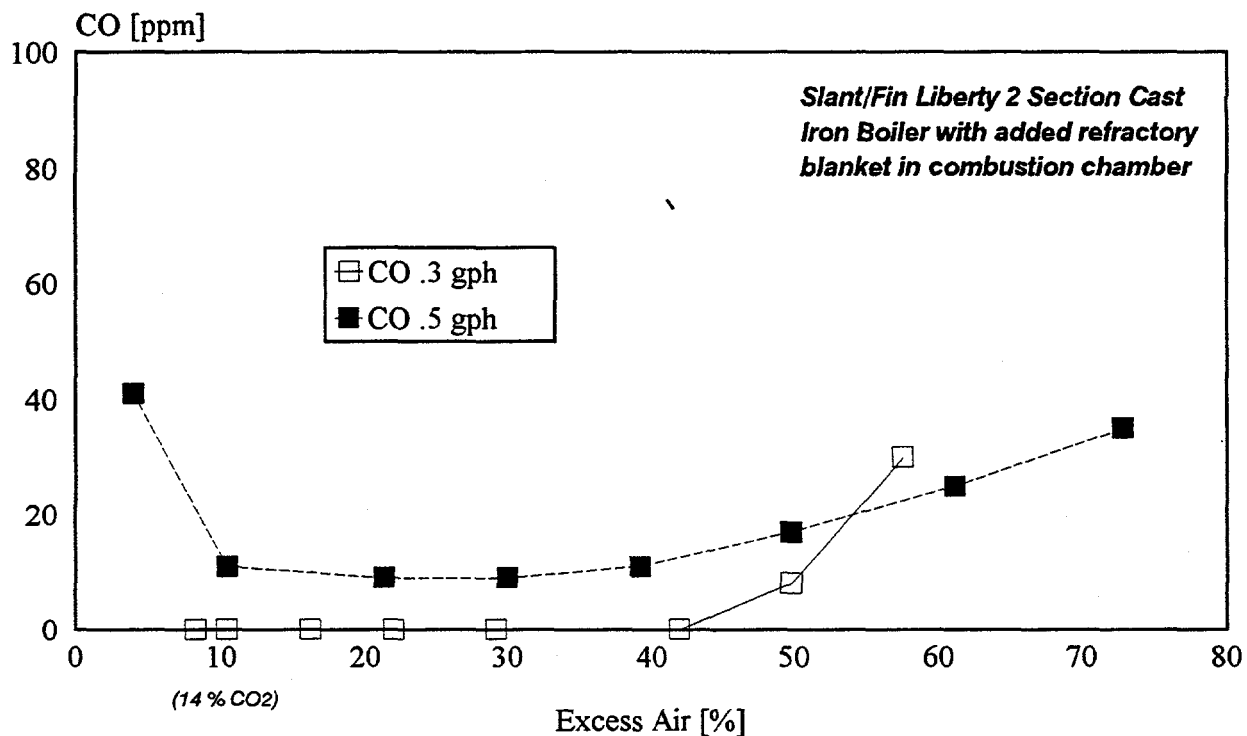


Figure 6-8. Tests with Fan-Atomized Burner in two section cast iron boiler with refractory blanket added on combustion chamber floor. Effect of excess air on CO and NO_x at constant air pressure .

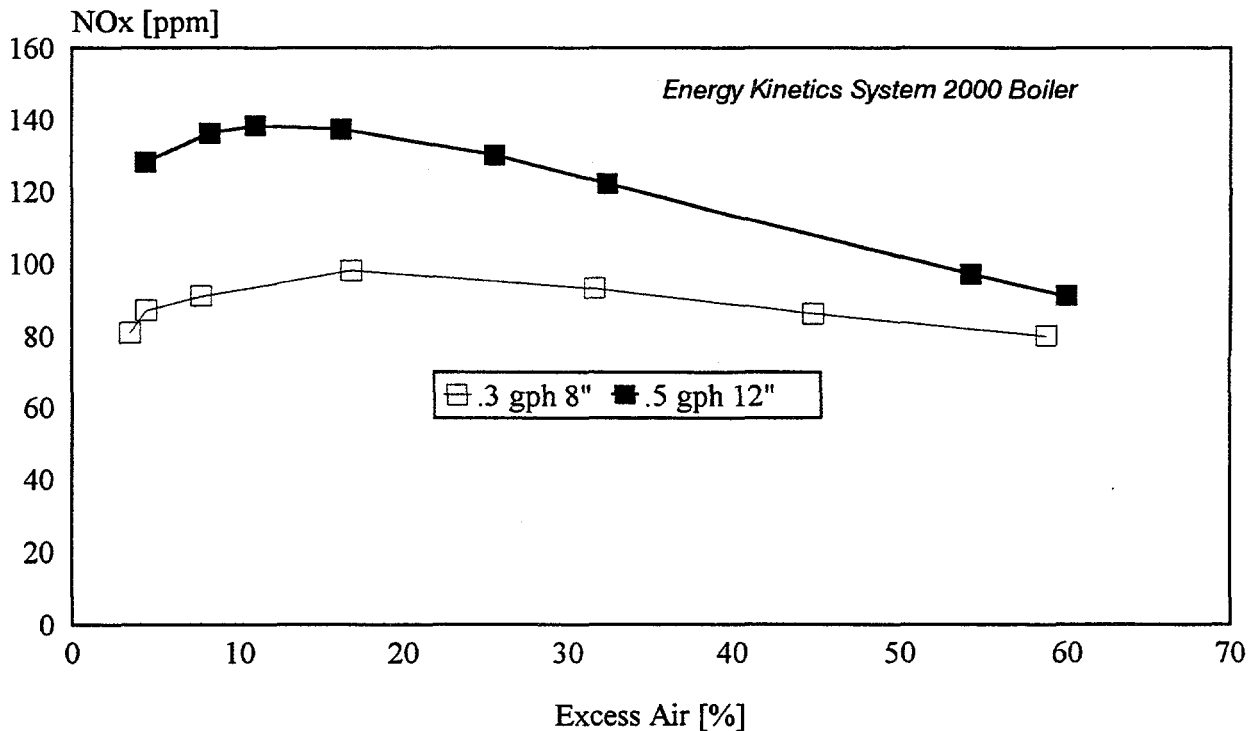
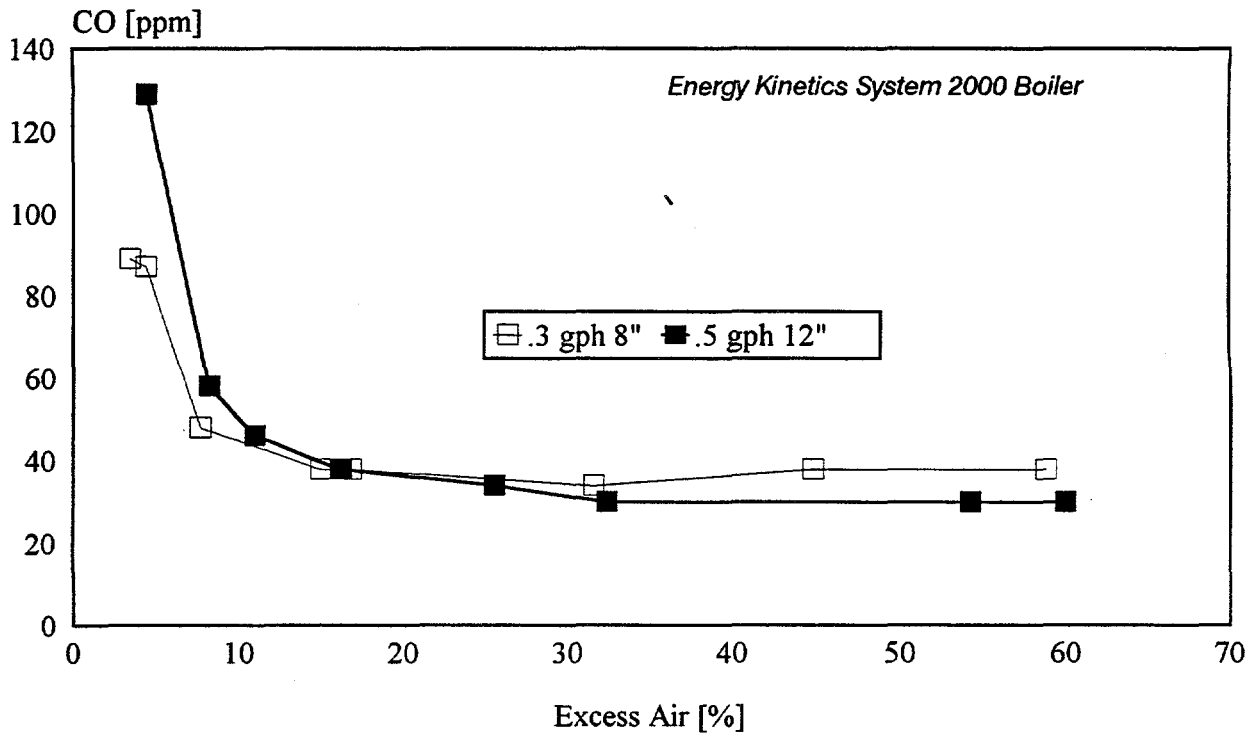


Figure 6-9. Tests with Fan-Atomized Burner in a steel wet base boiler. This boiler has a heavy refractory liner in the combustion chamber. Effect of excess air on CO and NOx at 8 and 12 inches of air pressure .

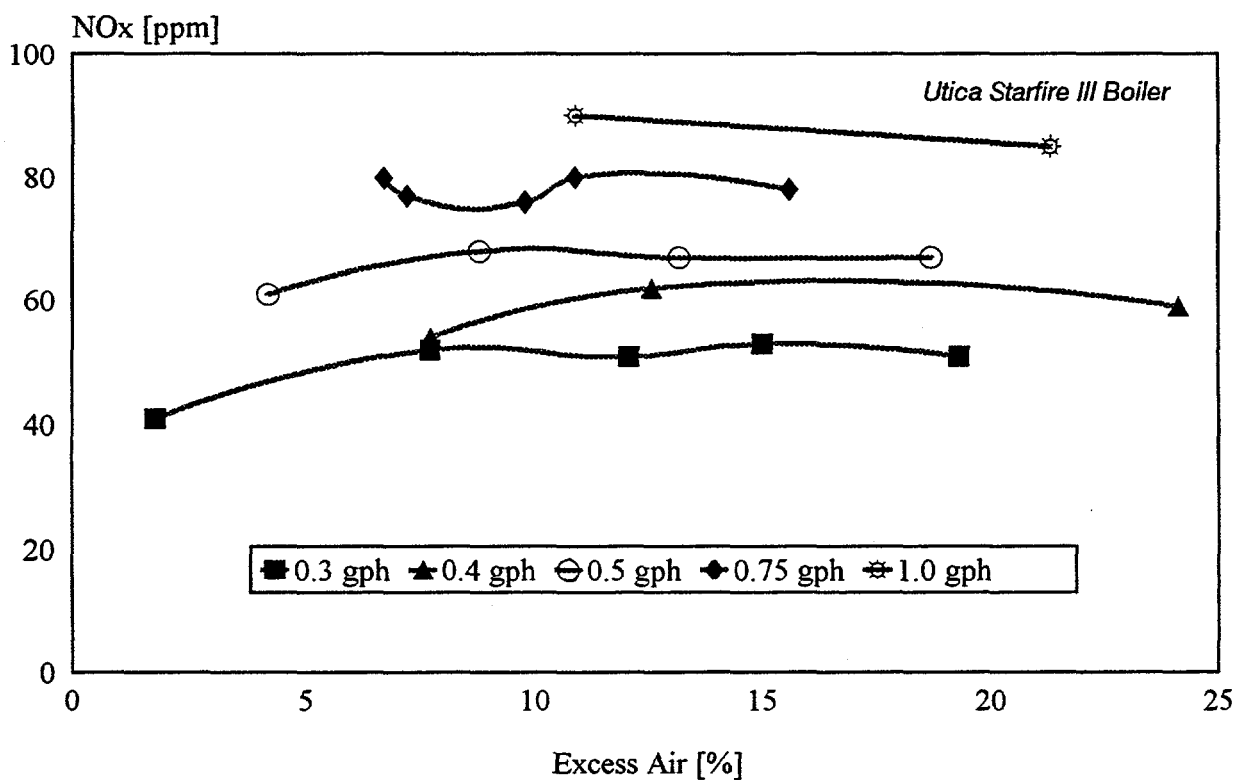
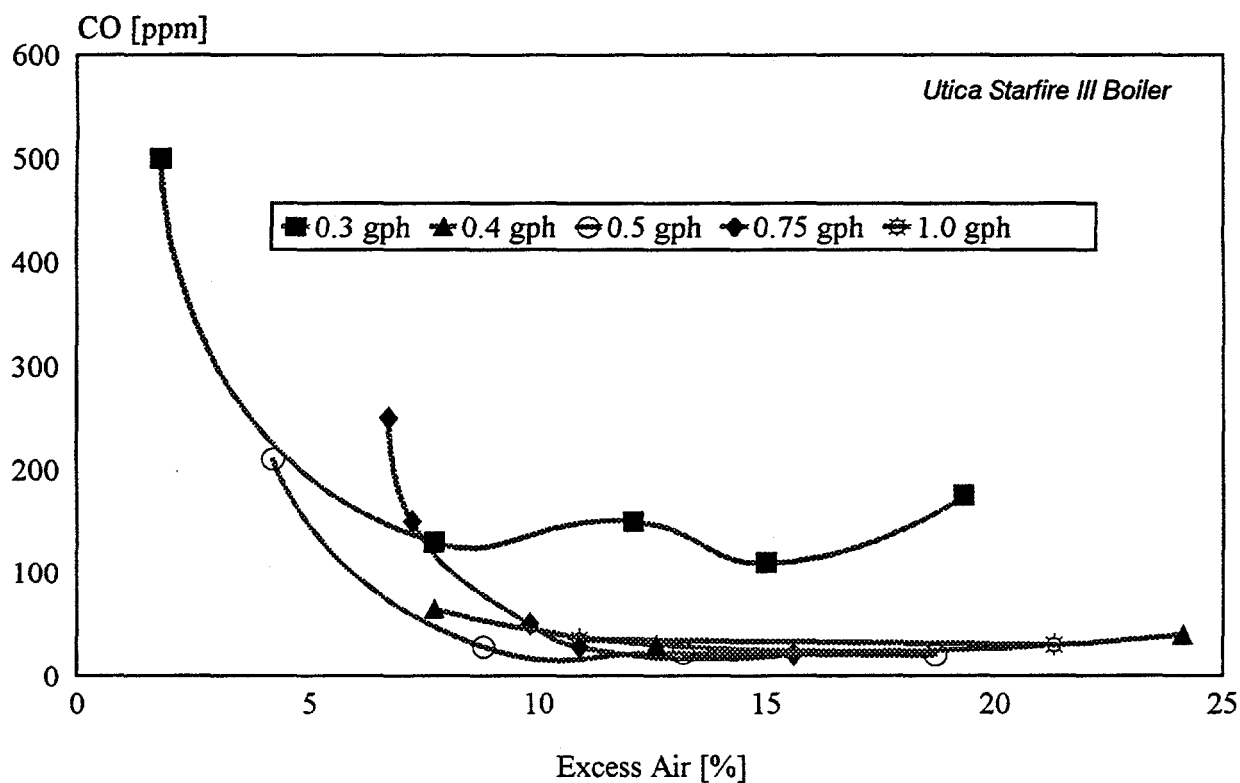


Figure 6-10. Tests with Fan-Atomized Burner in two section cast iron boiler. Effects of excess air on CO and NOx. A refractory blanket has been added on the chamber floor for these tests.

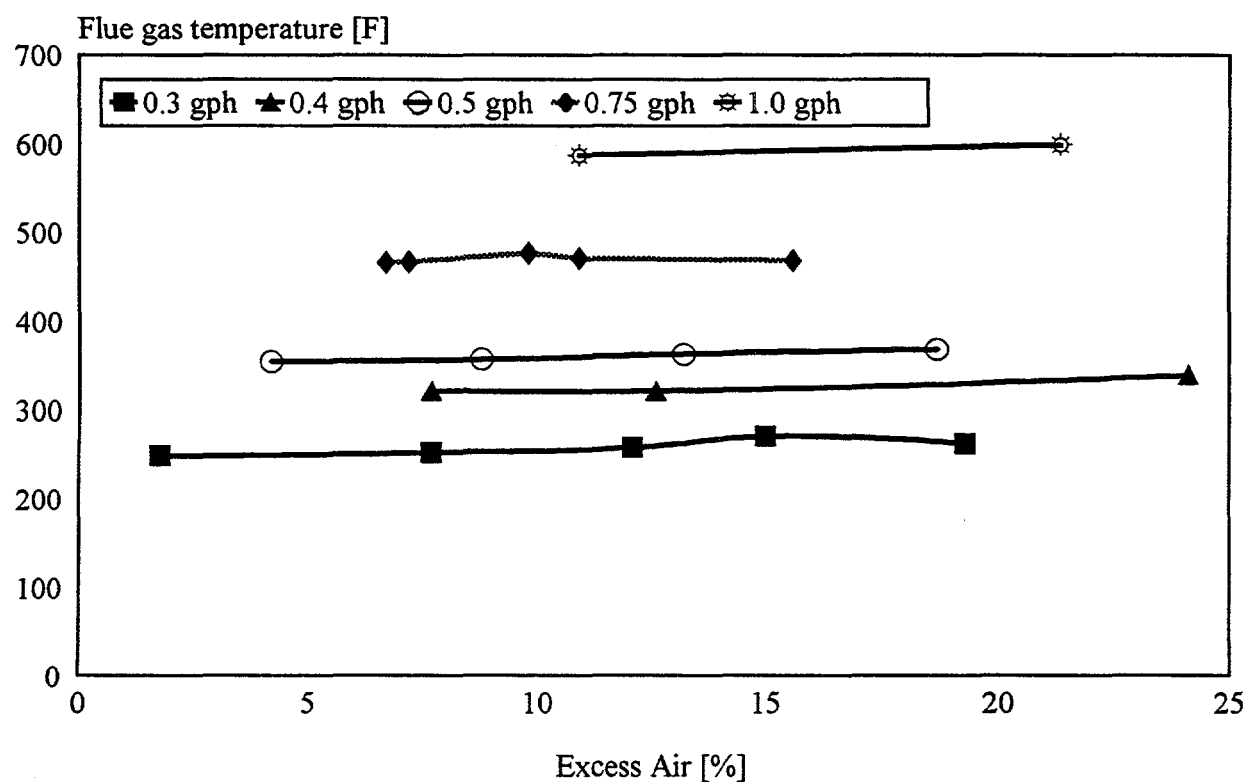


Figure 6-11. Tests with Fan-Atomized Burner in two section cast iron boiler. Gas temperature at boiler exit. A refractory blanket has been added on the chamber floor for these tests.

7. ADVANTAGES OF THE "FAN ATOMIZED BURNER" AND POTENTIAL MARKETS

Based upon the results presented in Sections 5 and 6 the Fan Atomized Burner (FAB) has several very positive features which can make it attractive for future commercialization. These include:

- Ability to operate at lower firing rates than conventional burners;
- Ability to have firing rate adjusted (without changing the nozzle);
- Ability to operate at low excess air levels;
- Low NOx emissions;
- Low electric power consumption.

The low firing rate capability of the burner can lead to new product markets such as small input boilers and furnaces for spaces which are small and/or highly energy efficient and low input, high efficiency water heaters. This burner can go far below the lower practical limit of about 0.6 gph currently provided by pressure atomized, retention head burners. The burner could also be considered for use as a refit unit, increasing the efficiency of existing boilers or furnaces through reduced input and reduced excess air. Considering, for example a current two section cast iron boiler rated for 0.75 gph with a flue gas exit temperature of 500 F and 30% excess air the steady state efficiency of 83%. Reducing the firing rate to 0.5 gph, excess air to 10% and flue gas temperature to 350 F (based on test results in Section 6) leads to a steady state efficiency of 87.8%, an increase of 4.8%.

Conventional, pressure atomized retention head burners can operate with excess air levels of about 20% under good (laboratory) conditions. In the field excess air levels are often set much higher - 30-50%. The prototype burner developed during this program, based on the results presented in Section 6, can operate with excess air levels generally under 10%. At firing rates over 0.5 gph lower excess air levels can be achieved.

At a firing rate of 0.5 gph conventional retention head burners have NOx emissions ranging from 80 to 120 ppm [1]. Based on results in the previous section NOx emissions with the air atomized burner at the same firing rate range from 60 to 80 ppm. NOx emissions are strongly dependent upon combustion chamber environment. To achieve the lowest possible NOx emissions it is necessary to carefully match the burner and chamber together.

Reducing the firing rate as discussed above may not be acceptable because of reduced heating system capacity. This could be overcome by the burners ability to have the firing rate adjustable, for example with the development of an automatic, two-stage version of the burner. During this project the potential benefits of such an approach was evaluated in a warm air furnace. Annual Fuel Utilization Efficiency tests were done with this unit both at fixed firing rate and with two stage firing. The two stage firing was done at 0.3 and 0.6 gph. With single stage firing AFUE was measured at 79%. With two stage firing the AFUE rating increased to 84%.

The ability to operate at low excess air levels even at fixed firing rates certainly leads to improved efficiency due to lower steady state flue gas energy

loss. In addition, however, operating at low excess air levels without smoke can also lead to reduced rates of fouling of heat exchanger surfaces and a lower rate of efficiency degradation over time. As discussed in Section I BNL studies have shown that a very important part of the fouling deposit is iron sulfate scale resulting from the deposition of sulfuric acid from the flue gas onto the heat exchanger surfaces. Acid production and scaling rate can be controlled by using burners which can operate at very low excess air levels [2,3,4]. The magnitude of efficiency degradation due to fouling has been estimated to be 2%/year. Figure 7-1 shows the results of measurements of heat exchanger fouling rates made with both a conventional burner and the FAB prototype. These measurements were made on small, cooled fouling tests sections. The burners were operated for 24 hours in steady state for these tests and additional details of the test arrangement can be found in References 2,3, and 4. Fuel sulfur content for these tests was 0.18%. Because these tests were done in steady state the fouling deposit is primarily iron sulfate scale. In transient operation carbon would also contribute to the total deposition rate (about half the total mass would be carbon in normal cyclic operation). The smoke number was zero for all tests with the exception of the lowest excess air tested with the conventional burner in which case the smoke number was 4. These results demonstrate the great benefit which can be realized from low excess air burner operation.

The fan used in the prototype burner developed during this project has a much higher pressure capability than the fan used in a conventional, pressure atomized retention head burner. The conventional burner fan has a maximum static pressure of about 3 inches of water. The higher pressure capability of the prototype burner has several advantages. In cases where boilers or furnaces are sidewall vented there is concern about effects of direct wind pressures against the side of the building on burner performance. The use of higher burner fan pressures reduced the magnitude of this effect. The prototype burner developed during this program would be expected to have a reliability advantage over conventional burners in sidewall venting applications. In addition the high pressure capability of the prototype burner may lead to the future development of more compact, high efficiency heat exchangers which operate with higher gas side pressure drops than now considered to be acceptable.

The prototype burner developed during this project is considered to have great potential for fully modulating operation and feedback excess air control. Fan speed can be easily controlled to trim air flow and approaches toward electronic fuel flow modulation were discussed in Section 5.

The electric power draw of conventional, pressure-atomized retention head burners is about 230 watts. In Table 5-2 expected power for the prototype burner was presented at 124 watts. Based on test measurements, however, the burner power was typically somewhat lower, about 95 watts.

To evaluate potential market interest and most significant commercial advantages of the burner a limited marketing survey was conducted. Questionnaires were distributed and responses obtained from about 35 representatives of manufacturing and service companies. While the survey size is clearly limited some responses were very strong and lead to important conclusions. Features that an advanced oil burner might offer which are of the most interest commercially include reduced fouling and reduced maintenance requirements. The next areas of interest

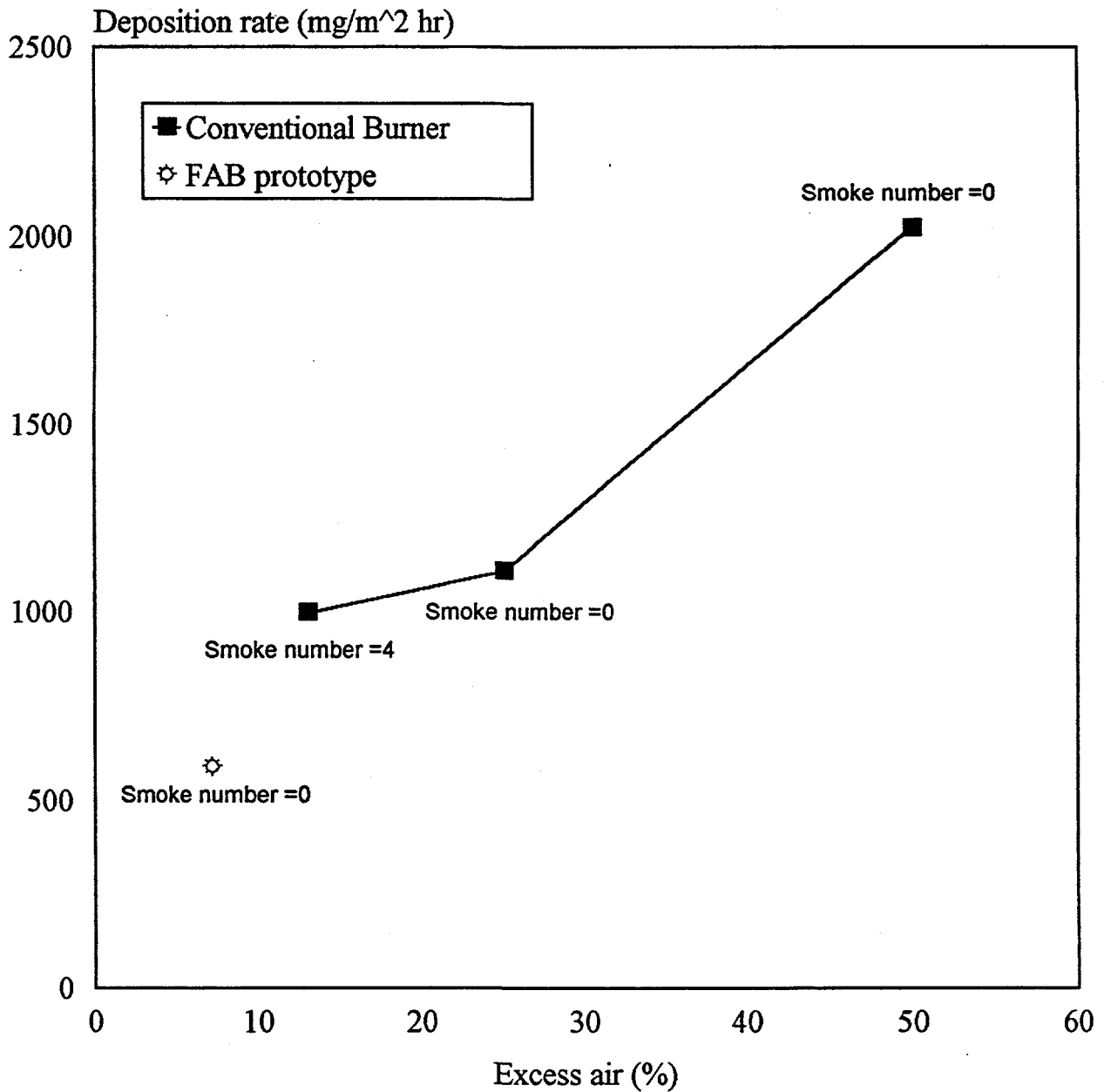


Figure 7-1. BNL Fouling Test - Fouling rate, both pressure atomized retention head burner and prototype advanced burner (FAB). Smoke number = 0 for all tests except the lowest excess air with the retention head burner. At this condition smoke number = 4.

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include low firing rate and capability for variable firing rates. Some interest was also expressed in ability to sidewall vent and low NOx emissions. The lowest feature of interest is reduced electric power draw. Another question asked in the survey is the potential share of the burner market which could be met by burners below 0.5 gph firing rate. Results varied but the mean response was under 25%. The total size of the burner market in the U.S. is about 400,000 units/year. Those surveyed were also asked about interest in using the burner in water heaters and 65% indicated interest in this. The median response to a question about the increase in wholesale price justified by the performance of the BNL burner was \$100.

8. CONCLUSIONS AND RECOMMENDATIONS

The work done during this project has shown that practical options exist for advanced oil burners which will provide advantages including:

- low firing rate (to 0.3 gph)
- low excess air (under 10% vs 20% with a conventional burner)
- high efficiency (system dependent)
- low NO_x emissions (60-70 ppm vs 80-120 ppm with conventional burner)
- reduced electric power consumption (<100 watts vs 230 watts with a conventional burner)
- reduced heat exchanger fouling
- variable firing rate (0.3 to 1.0 gph)

A complete prototype burner using the low-pressure air-atomizing nozzle has been developed and using this burner all of these benefits have been demonstrated. While the performance of the prototype burner is very good additional testing, in cooperation with manufacturers is recommended. This will serve to identify changes needed to meet industry standards for reliability, ease of installation and maintenance, and field performance. Any potential manufacturers will evaluate cost implications of needed changes and market potential.

A second major area which should be given consideration is applications. The advanced burner developed during this program could be used at one fixed firing rate with boilers or furnaces which are currently on the market. For any applications, however, which are chimney vented the lowest firing rate applied (and maximum efficiency achieved) will be set by the need to avoid condensation of water vapor from the flue gas. In this case the overall benefits of using an advanced burner will be limited to reduced excess air and NO_x. With sidewall venting the use of the burner has greater advantage. The system can be force vented as discussed in the previous section and firing rates can be adjusted to give lower gas exit temperatures (higher efficiency). These types of applications either with a single firing rate or multiple firing rates should receive more attention. The fixed firing rate prototype developed during this project should be advanced to a modulating version -- either two stage or fully modulating.

While the applications discussed above will benefit for the use of an advanced burner they will not take full advantage of all of the capabilities of the burner particularly for low and modulating firing rates. The availability of an advanced burner like the one developed during this project opens new possibilities for low input and integrated system options. It is recommended that these be evaluated in the future.

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