EPA-600/2-76-256
September 1976

Environmental Protection Technology Series

COMMERCIAL FEASIBILITY OF AN OPTIMUM RESIDENTIAL OIL BURNER HEAD

Industrial Environmental Research Laboratory
Office of Research and Development
U.S. Environmental Protection Agency
Research Triangle Park, North Carolina 27711

RESEARCH REPORTING SERIES

Research reports of the Office of Research and Development, U.S. Environmental Protection Agency, have been grouped into five series. These five broad categories were established to facilitate further development and application of environmental technology. Elimination of traditional grouping was consciously planned to foster technology transfer and a maximum interface in related fields. The five series are:

1. Environmental Health Effects Research
2. Environmental Protection Technology
3. Ecological Research
4. Environmental Monitoring
5. Socioeconomic Environmental Studies

This report has been assigned to the ENVIRONMENTAL PROTECTION TECHNOLOGY series. This series describes research performed to develop and demonstrate instrumentation, equipment, and methodology to repair or prevent environmental degradation from point and non-point sources of pollution. This work provides the new or improved technology required for the control and treatment of pollution sources to meet environmental quality standards.

EPA REVIEW NOTICE

This report has been reviewed by the U.S. Environmental Protection Agency, and approved for publication. Approval does not signify that the contents necessarily reflect the views and policy of the Agency, nor does mention of trade names or commercial products constitute endorsement or recommendation for use.

This document is available to the public through the National Technical Information Service, Springfield, Virginia 22161.

EPA-600/2-76-256
September 1976

COMMERCIAL FEASIBILITY OF AN OPTIMUM RESIDENTIAL OIL BURNER HEAD

by

L. P. Combs and A. S. Okuda

Rocketdyne Division
Rockwell International Corporation
6633 Canoga Avenue
Canoga Park, California 91304

Contract No. 68-02-1888
ROAP No. 21BCC-058
Program Element No. 1AB014

EPA Project Officer: G. Blair Martin

Industrial Environmental Research Laboratory
Office of Energy, Minerals, and Industry
Research Triangle Park, NC 27711

Prepared for

U.S. ENVIRONMENTAL PROTECTION AGENCY
Office of Research and Development
Washington, DC 20460

ABSTRACT

The feasibility of commercial application of an optimum head for distillate oil burners to effect simultaneous reductions in emission of air pollutants and consumption of fuel by residential space heating equipment has been investigated. The optimum head technology was developed under an earlier EPA-sponsored study and was shown to minimize emission of oxides of nitrogen from a variety of research combustors while maintaining both high efficiency and low emissions of other pollutants. The current study was concentrated on selecting the best commercially practiced fabrication method for making optimum heads, on determining that prototype heads made to simulate such production units effectively reproduce the research head's beneficial results, and on extending the data base by testing the prototype heads in two commercial residential furnaces.

Sheet metal stamping was selected as being the best fabrication method. A one-piece stamped and folded optimum head design was evolved, and prototype commercial optimum heads were fabricated. These were shown in research combustion chamber tests to be equivalent to the earlier research head. Tested as retrofit replacements of the stock burner heads in two new warm-air oil furnaces, the prototype optimum heads were found to be operationally satisfactory and potentially durable and long-lived. Measured retrofit effects on furnace thermal efficiency and NO emissions were compared with estimated average characteristics of existing installed residential heating units to project estimates that widespread retrofitting of old existing residential units could increase mean season-averaged thermal efficiency (averaged over those units retrofitted) by about 5 percentage points and simultaneously reduce NO_x emissions from these sources by about 20%. Several issues are noted as being unresolved, including logistics of a retrofit program, service personnel training needs and requirements to ensure meeting codes and standards.

CONTENTS

ILLUSTRATIONS

TABLES

SECTION I

CONCLUSIONS

1. The concept of undertaking commercial production, distribution, and marketing of an optimum distillate oil burner head for retrofit application to residential oil heating equipment was found to be technically feasible.

2. The optimum head technology should be applicable with some benefit to an estimated 50% or more of existing residential oil-fired warm-air furnaces and hydronic boilers in the United States. Inapplicability to some units would result from equipment incompatibilities, operational problems, and/or degradation rather than benefit of efficiency, pollutant emissions, or both.

3. Benefits from retrofitting an individual heating unit would be a modest increase in thermal efficiency and a significant reduction in emissions of oxides of nitrogen (NO_x) air pollutants. Estimated increases in a given furnace's efficiency range from zero (or less) for units already operating with low excess air to 10% or more for units now operating with very high excess air. Judicious application of retrofitting to large numbers of appropriate units should achieve increases in mean steady-state and overall season-averaged thermal efficiencies (averaged over those units retrofitted) of at least 5 percentage points. Simultaneously, NO_x emissions from those same units should be reduced by about 20%. Retrofitting also might reduce significantly the average smoke emissions but would not change appreciably the already low emissions of carbon monoxide and unburned hydrocarbons.

4. Sheet metal stamping was assessed as being the best commercial fabrication method for making optimum heads. Single piece heads with integral swirl vanes, choke plate, and attachment provisions can be made and can satisfy all technical requirements. Sheet metal thicknesses in the range of 0.00087 m to 0.00127 m (0.0344 to

0.0500 inch) are satisfactory, but a relatively refractory stainless steel must be used. While tested for only 500 hours of simulated service, Type 310 stainless steel showed excellent resistance to degradation and should have an indefinitely long lifetime. The less expensive Type 304 stainless steel may also prove to be adequate, but its test exposure duration was not long enough to give a valid indication of potential durability.

5. In volume production, it was estimated that stamped stainless steel sheet metal optimum heads should cost the manufacturer on the order of $1.50 each.

SECTION II

RECOMMENDATIONS

1. If application of the optimum burner head technology is to be pursued, it would be prudent and logical to conduct comparative before and after performance testing of a representative sample of old existing units in actual residential use. Incremental changes in efficiency and NO_x emissions should be measured and correlated to initial burner type, and the requirements for ensuring that retrofitted burners meet applicable codes and standards should be determined.

2. A market survey should be conducted to define as fully as possible the current makeup of the existing oil furnace and boiler population, burner types, firing rates, blast tube diameters, ignition electrode diameters and spacing, etc., and the potential burner retrofit or replacement rate as a function of capital cost to the homeowner and potential annual fuel savings. These data should be combined to define the minimum number of different head designs which can satisfy the market requirements.

3. A manufacturing cost analysis should be performed to project capital and operating expenses for making and distributing the different head designs. The effect of production rate on cost per head, of distribution and markup costs on price to the consumer and of price on replacement rate should be included in an economic analysis leading to estimated production rates for each head design.

4. Because the manufacturing cost per head is closely tied to material costs, ways of minimizing material weight per head should be investigated carefully. Clever interlacing of adjacent stampings should be examined. Also, because the perimeter of each stamping is controlled largely by the size of the swirl vanes, some additional experimental furnace testing might be devoted to investigating the minimum required vane size.

SECTION III

INTRODUCTION

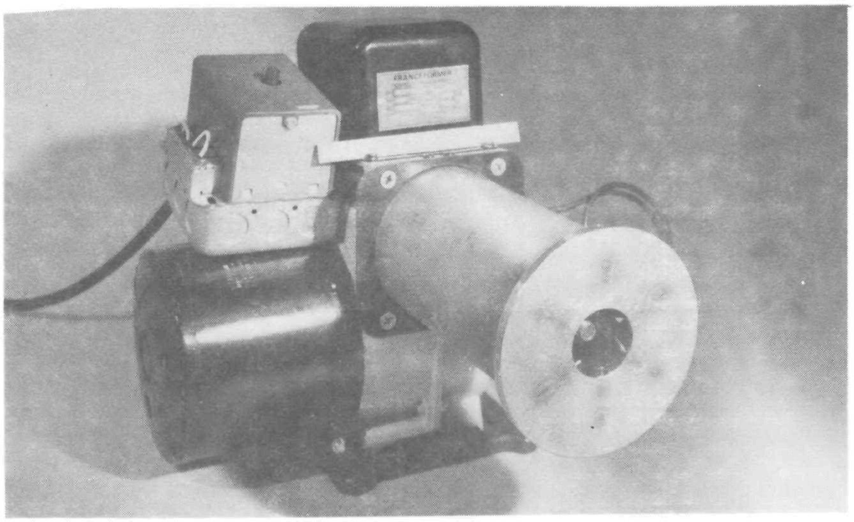
The Environmental Protection Agency has sponsored studies over the past few years to document the emission of air pollutants from existing residential and commercial oil-fired space heating units (Ref. 1 and 2). Concurrently, the EPA has also supported applied research programs to determine the effects of "controllable" parameters on emission levels and to devise strategies for minimizing pollutant emissions (e.g., Ref. 3 through 5). These and related studies have shown that substantial reductions in total emissions can be effected by combustion modifications such as advanced burner designs, flue gas recirculation and two-stage combustion.

In particular, an intensive Rocketdyne investigation of residential and commercial oil burners (Ref. 5) led to criteria for optimizing conventional burner designs with respect to pollutant emissions. For high-pressure atomizing, luminous-flame burners fired into refractory-lined combustion chambers, minimum pollutant emissions were obtained with burners having: (1) no flame-retention device, (2) choke diameter related quantitatively to the firing rate, and (3) oversized internal peripheral swirler vanes oriented at 25 degrees relative to the blast tube axis. This swirler vane angle gave the best compromise between smoke emissions and nitric oxide emissions, while the optimized choke diameter produced minimum nitric oxide emissions. Those burner design attributes were all concerned with the burner "head," i.e., the portion of the burner that admits prepared reactants into the combustion chamber. For that reason, this development was referred to as the "optimum head."

In addition to minimizing the formation of nitric oxide emissions, the optimum head was found to have a potential for increasing overall furnace fuel utilization efficiency. This resulted from its ability to be fired in laboratory testing without producing unacceptable levels of carbonaceous pollutant emissions (carbon monoxide, unburned hydrocarbons, and smoke), with considerably less excess air than is usual in residential furnace installing. Reducing excess air decreases the sensible heat losses in the flue gase and, therefore, results in a net increase in thermal efficiency.

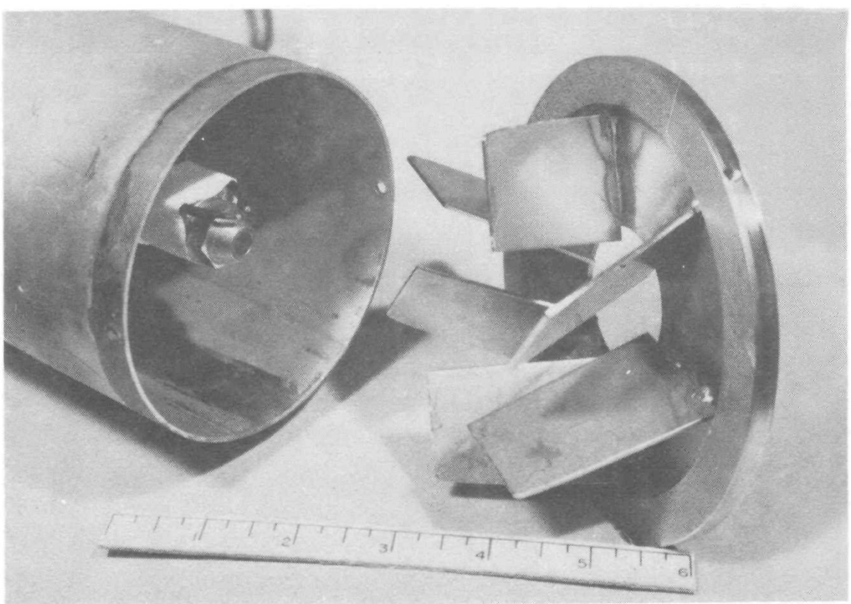
Because the optimum burner's distinguishing design features were confined to the burner head, the optimum head was recognized as presenting a very attractive possibility for helping to reduce simultaneously both pollutant emissions and fuel consumption with a low-cost retrofit device for existing burners in existing furnaces. Previous experience with optimum heads, however, was limited to two research heads (one each in two sizes) fabricated by machining and welding stainless-steel plate and tested predominantly in research combustors (Fig. 1). Additional experience, including testing as a retrofit device in commercially available furnaces, was obviously needed to establish the feasibility of applying the optimum head commercially.

The investigation described in this report was undertaken, therefore, with the objective of evaluating and establishing the technical feasibility of commercial application of the optimum distillate oil burner head to residential furnaces. Several aspects of commercialization were explored in integrated studies described and discussed in the following sections of the report. Predominantly analytical aspects are reported in Section IV. Those include: the analysis of current residential furnace operation and performance to determine design and operating variables affecting fuel economy and the proper firing rate ranges for optimum head experimentation; consideration of commercial fabrication techniques selected as being most applicable for economic commercial production of optimum heads; and the design of prototype heads producible by the selected fabrication technique(s). Aspects that are predominantly experimental are reported in Section V, Experimental Investigation. Included are the fabrication of prototype optimum heads to simulate those that might eventually be made commercially, and the laboratory testing of those heads: (1) in research combustors to establish correspondence with earlier preprototype research head results, and (2) in typical commercial warm-air furnaces to evaluate performance under realistic cyclical operation.



5DZ21-8/6/73-S1B

EXTERNAL VIEW



5DZ21-8/6/73-S1A

OPTIMUM HEAD

Figure 1. 1 ml/s (gph) optimum low-emission residential oil burner (Ref. 5)

SECTION IV

ANALYTICAL INVESTIGATIONS

Brief background studies were made in two different areas preparatory to designing commercially producible prototype optimum heads. The first was concerned with estimating effects conversion to the optimum head might have on typical furnaces' fuel economy and pollutant emissions, and with selecting an appropriate burner firing rate for subsequent prototype head testing. The second was concerned with studying commercial fabrication methods that might be used for producing optimum heads and selecting the one or two judged to be best for high-volume production of low-cost, durable units. Because a fabrication method's applicability depends on having a workable head design, a preliminary design effort was conducted concurrently with that study. The results led naturally to finalized designs for prototype optimum heads.

OIL FURNACE OPERATIONS ANALYSIS

In 1970, approximately 14% of the U.S. energy consumption was used for residential space heating and domestic water heating (Ref. 6). Essentially, all of that energy was derived from fossil fuel combustion, either directly with combustion equipment on the premises or indirectly through the use of electrical resistance heating. That quantity of energy was derived from various energy sources in approximately the following distribution: 65.7% from gaseous fuels, 24.3% from distillate fuel oils, 6.4% from utility electricity, and 3.6% from coal, wood, solar, etc.

At that time, nearly one-half of U.S. residences were heated by forced-draft warm-air furnaces, of which approximately 72% were gas-fired, 22% oil-fired and 5% electrical resistance heated. Almost one-quarter of the U.S. homes were heated by steam or hot water from hydronic boilers, with about 40% gas-fired and 54% oil-fired. The remaining homes (about one-third of the total) were either unheated or employed a broad range of equipment ranging from fireplaces to floor or wall-mounted direct-heaters.

The total population of oil-fired residential furnaces and hydronic boilers in 1970 exceeded 13 million units. Since then, annual sales of such units have averaged about 525,000; additionally, an average of about 154,000 conversion burners have been sold annually. From these figures, an oil furnace replacement rate of about 4 to 5% per year can be estimated; it is evident that the average makeup of the installed population will change only rather slowly over the years. Thus, even though units sold today may offer significantly better fuel economy than the vast majority of older furnaces, their sales rate increases the overall average efficiency only slightly from year to year.

There have been strong market incentives (increased oil prices and uncertain supplies) since late 1973 to replace old inefficient residential heating equipment. Nonetheless, a homeowner who has a unit that is installed, working, and paid for should be expected to move slowly and reluctantly toward deciding to spend several hundred dollars now to reduce future heating costs. A retrofit burner head offering efficiencies comparable with (or even higher than) current new equipment would have considerably greater consumer appeal than would replacement of an entire boiler, furnace, or even burner, particularly if it were easily installed at low cost. Retrofit should, therefore, proceed at a considerably faster pace and contribute substantially to raising the national average efficiency of utilizing heating oil. The magnitude of the consumer appeal would obviously depend on how large an efficiency gain an individual homeowner could realistically expect.

Thermal Efficiencies

Oil furnaces and hydronic boilers are manufactured in conformity to national standards. Current testing and rating standards are ANSI Z91.1-

1972 for oil-fired warm-air furnaces (Ref. 7) and the Hydronics Institute 1975 standard for oil-fired hydronic boilers (Ref. 8). Both of these standards specify that a new unit's steady-state thermal efficiency, based on the fuel's higher heating value, shall equal or exceed 75%. At steady-state, most of the heat that is not recovered is convected up the flue as latent heat of the water vapor formed in the combustion process and as sensible heat of the flue gases, with only minor losses (typically 1/2 to 1-1/2%) conducted through the cabinet, etc., and radiated or convected to the surroundings.

The latent heat of combustion-formed water vapor represents between 6 and 7% of fuel oil's higher heating value. Condensation in the flue system is intentionally avoided in conventional furnace and boiler technology, so losses of this magnitude form an unavoidable baseline. The way condensation is prevented is by maintaining the flue gases above their dew point everwhere in the system; in practice, this is ensured by designing for net unit exhaust gas temperatures* of about 220 C (400 F) or greater. As a direct result, there are substantial thermal losses in the form of flue gas sensible heat. Their minimum value is on the order of 8% when combustion is carried out at stoichiometric conditions (i.e., no excess air) and the flue gases are cooled to 220 C (400 F) net stack temperature. Adding these approximate latent and sensible heat losses suggests that conventional oil furnaces and boilers might have steady-state efficiencies as high as 85%. Usually, however, sensible heat losses are greater than 8% for two reasons: (1) some excess air is required to avoid formation of excessive CO, UHC, and/or smoke, and (2) average net flue gas temperatures are higher than the minimum needed to avoid condensation.

Tabular values of steady-state flue gas thermal efficiency decrements are given in Ref. 7 as functions of net flue gas temperature and flue

*Net flue gas temperature is the difference between actual flue gas temperature and mean heated room temperature.

gas concentration of CO_2 (which is the parameter measured by heating industry personnel, rather than excess air level). These data are plotted in Fig. 2 as a family of curves, with efficiency decrements along the left hand ordinate. The decrements are converted to estimated steady-state efficiencies along the right-hand ordinate by subtracting them from 100% and assuming that 2% of the fuel's higher heating value is transferred ("lost") to the surrounding's through the furnace cabinet. A supplemental scale is given relating exhaust gas stoichiometric ratio (SR) to its CO_2 concentration. Defined as the actual air-to-fuel weight ratio divided by the theoretical stoichiometric weight ratio, SR is related directly to the excess air level. For example, SR = 1.50 corresponds to 150% stoichiometric air and this is equivalent to 50% excess air.

Residential furnaces and hydronic boilers are typically operated in an on-off cyclical manner. The excess air level and flue gas temperature must be controlled to avoid smoke formation and condensation over a range of cycle conditions, and this generally forces them to be higher than would be appropriate for steady-state operation only. This is one of the major reasons for operating at less than optimum efficiency conditions. During cyclical operation, efficiency is further degraded by some transient contributions to a unit's heat losses. When the burner is not being fired (standby), a natural draft flow of air through the burner, firebox, etc., cools furnace components and continues to convect heat up the flue. Typically, this loss may cause cycle-averaged efficiencies to be 3 to 5% lower than steady-state although, in some situations, the decrement may be as large as 15%. External heat losses from the cabinet also continue during standby. With warm air furnaces, additional cyclical cabinet losses are moderately small (~1/2 to 1%) because furnace components are cooled considerably before the warm air flow is turned off. With hydronic boilers, however, they may be substantially larger (~1-1/2 to 3%) because most boiler components are at nearly the same temperatures during standby as during firing.

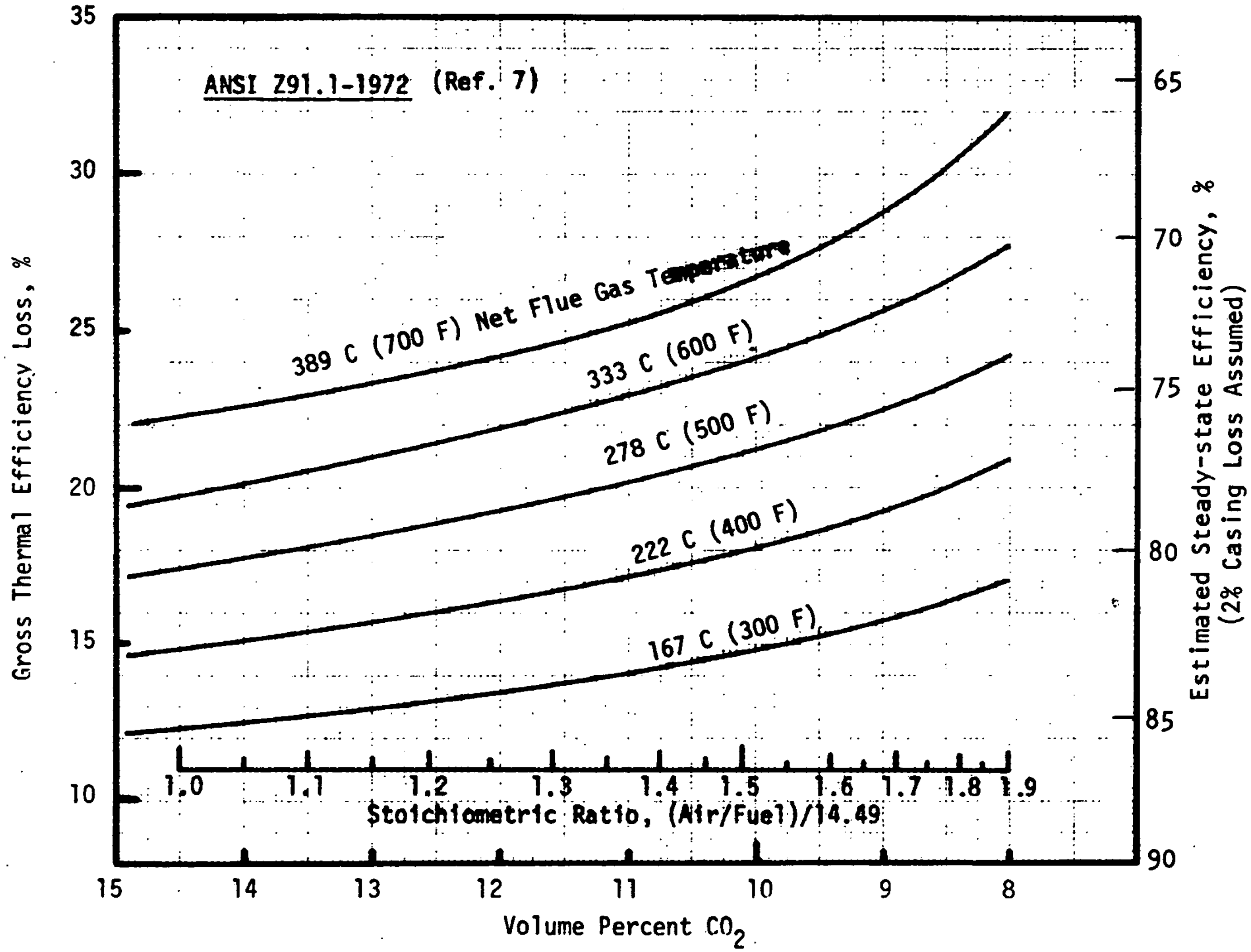


Figure 2. Flue gas gross thermal efficiency losses as a function of net flue gas temperature and composition

The decrements between steady-state efficiencies and cycle-averaged (service) efficiencies also depend on the mean cycle timing, growing larger as standby time increases and vanishing as steady-state operation is approached. This is exemplified by laboratory data for two hydronic boilers reported in Ref. 9 and reproduced in Table 1; general agreement with the ranges stated above is evident.

Over a long period of time, such as an entire heating season, there will be a wide distribution of thermal demand conditions ranging from nearly continuous standby to essentially continuous operation. Season-averaged efficiencies are rarely (if ever) measured, but several investigators have estimated values from shorter term testing. A recent example is given in Ref. 10. Steady-state (absorption) efficiencies were measured in the "as found" and subsequent "tuned" conditions for a representative sample of residential heating systems in northern New Jersey. The results are reproduced in Table 2 and show little influence of unit type or burner tuning. Subsequently, 11 units were instrumented to record performance data, including cycle timing for a long enough period of time (1 to 3 weeks) to correlate cycle behavior and oil consumption to outdoor ambient conditions typically encountered over a heating season. An overall average efficiency (pseudo-season average) of 60% was obtained, indicating that standby losses (predominantly component cooling by draft air flow through the combustor) averaged 15%. It was also shown that these rather large standby losses could be approximately halved by reducing the units' firing rates (by an average of 25%) so that they would fire continuously when the ambient temperature dropped to the local design temperature. Many other investigators' estimates of season-average efficiencies could be discussed but there are so many uncertainties associated with each that this won't be pursued here. Rather, a 60 to 65% range will be assumed to be reasonably valid.

The field survey data reported in Ref. 2 for an entirely different, but similar size, sample of residential oil furnaces and boilers was concerned with emissions rather than thermal performance. Nonetheless,

Table 1. THERMAL AND SERVICE EFFICIENCIES OF TWO RESIDENTIAL HOT WATER BOILERS (REF. 9)

Boiler and Condition	Thermal Efficiency, %	Standby Loss, % of Input	Service Efficiency,% (on/off time, minutes)		
			(20/10)	(15/15)	(10/20)
Boiler A					
New	76.2	2.02	75.50	74.80	71.30
After 6 months	74.8	--	74.10	73.30	70.00
After 10.5 months	73.43	--	72.87	72.06	68.78
Boiler B					
New	72.60	2.40	71.50	70.00	67.10
After 6 months	71.50	--	70.30	68.80	66.00

NOTES: Electric consumption of accessories included in input.
Boiler A differs from most contemporary oil-fired equipment

Table 2. STEADY-STATE ABSORPTION EFFICIENCY OF OIL HEATING UNITS (REF. 10)

Type of Heat	No. of Units	Absorption Efficiency, %	
		As Found	Tuned
Warm Air	12	72.6 ±3	75.4 ±5
Hot Water	11	74.4 ±5	72.3 ±7
Steam	15	75.7 ±3	76.7 ±3
Total (average)	38	75.4	74.6

steady-state efficiencies can be derived from the reported CO_2 and tenth minute stack temperature data. Assuming a uniform 2% casing heat loss, the averages of 33 units' estimated efficiencies were: 71.0%(+10, -21) as found, and 72.1%(+7, -14) tuned. It is possible that there was a consistent bias in the excess air levels between these two surveys, but it seems more likely that any bias was in the instrumentation and measurement methods. The scatter is great enough, particularly in the Ref. 2 data, that the extreme values influence the averages significantly. While this suggests that larger samples would be desirable, for our purposes, the two surveys can be averaged to obtain an approximate characterization of the entire U.S. residential oil furnace and hydronic boiler population:

- Average steady-state conditions somewhere in the ranges:
  90 ±10% excess air; 8 ±0.3% CO_2 } 72 to 75% gross
  500 ±60 F net flue gas temperature } thermal efficiency
- Season-averaged gross thermal efficiency in the range of 60 to 65%

Now we can turn to the question of how much fuel can be saved by replacing or retrofitting an old inefficient unit with equipment amenable to higher efficiency. The range of possibilities is illustrated in Fig. 3. As an example, if a unit averaging 60% thermal efficiency were replaced (or retrofitted) to increase the average efficiency to 85% (a 25% gain), its fuel consumption would be reduced by 29%. It has been seen earlier that 85% represents an approximate upper limit within the conventional furnace and boiler technology. Therefore, a dotted line has been drawn through the locus of points where the old efficiency and the efficiency gain sum to 85%. Portions of the efficiency gain curves to the right of that dotted line are dashed to indicate the impracticality of operating in that region with present-day equipment.

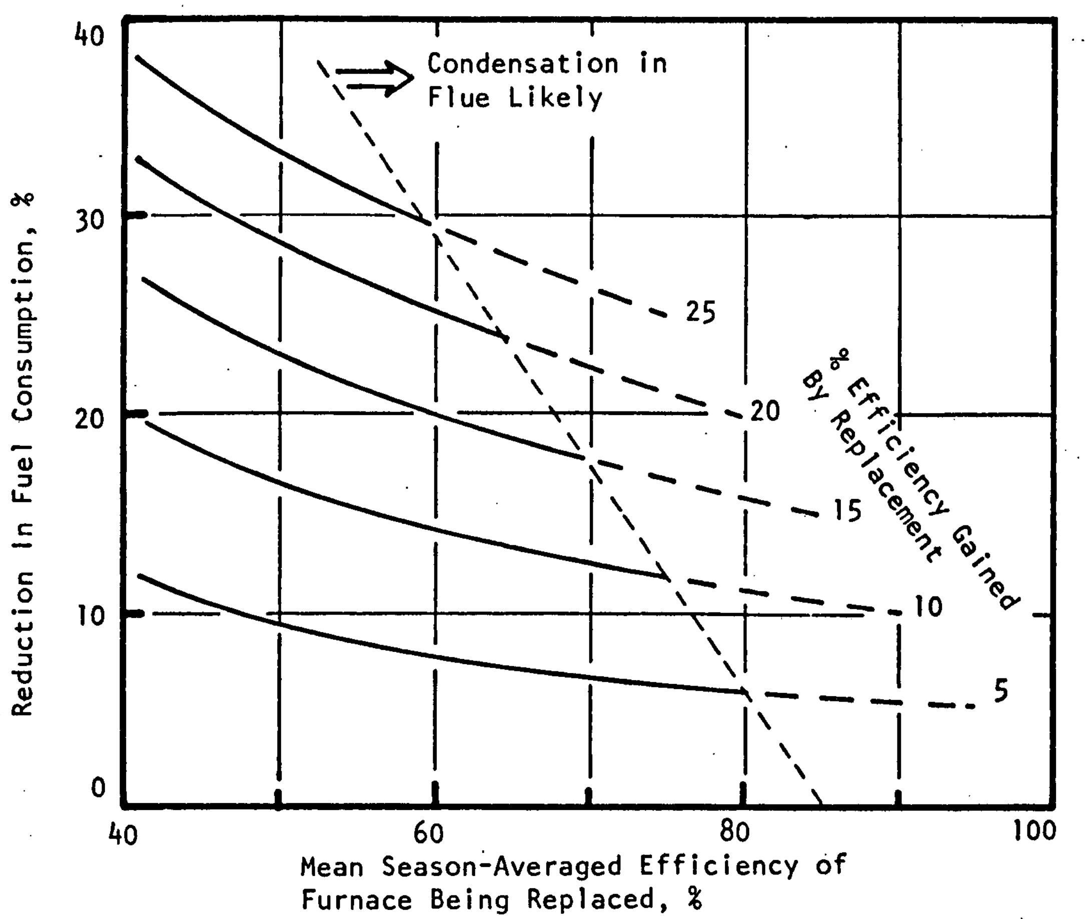


Figure 3. Reduction in fuel consumption when a furnace is replaced by a more efficient unit

If the entire population of residential oil heating equipment were repaired, retrofitted, or replaced with comparable equipment capable of raising the average steady-state efficiency from, say, 74% to 82%, the 8% efficiency gain would reduce national consumption of oil for these purposes by about 10%. If the firing rates were simultaneously decreased to eliminate overfiring, thereby reducing standby losses by another 7%, total estimated reduction in residential oil consumption would be approximately 20%. The 82% steady-state level can be attained by lowering net flue gas temperatures to practical minimum values and lowering excess air to the 20 to 25% range.

Analysis of a Specific Furnace - To obtain more confidence in the validity of the foregoing discussion, an analytical computer model of a warm-air furnace was used to calculate thermal efficiency behavior of a particular furnace as if it were retrofitted with an optimum head and run with 15% excess air instead of earlier representative conditions of 50 and 85% excess air.

The existing WAFURN (warm air furnace) computer program is a transient heat transfer analysis with the capability of accounting for the effects of typical furnace operating variables on efficiency (Ref. 11). A WAFURN program evaluation results in a calculated net furnace cycle thermal efficiency. The cyclic analysis is conducted through iterative calculate/balance loops to ensure cycle-to-cycle continuity. The detailed thermal analysis allows variation of many parameters such as: (1) oil input (firing rate and temperature); (2) stoichiometric ratio; (3) cycle length and profile; (4) flue gas temperature (heat exchanger capability); (5) fuel quality (heat of combustion and water emulsion); and (6) input combustion air conditions (indoor or outdoor supply).

The operation of a warm-air furnace with a refractory-lined combustion chamber was modeled. Parameters varied were: (1) cycle timing, (2) cycle length, (3) firing rate, (4) combustion air temperature, and (5) stoichiometric ratio. A summary of calculated results is presented in

Table 3. The thermal efficiencies (η_{furn}) listed include the draft air heat losses but do not include the relatively constant loss through the furnace casing (~1 to 2%). The SR = 1.15 data represent the expected operating condition of the optimum head design while the SR = 1.85 data represent the season-averaged operating condition of existing units in the field. The warm-air flowrate for the 10 runs was fixed at 0.566 m^3/s (1200 cfm) and, except for Case 6, combustion air was brought in from outdoors at 0 C (32 F).

The data from Table 3 are plotted in Fig. 4 and 5. Figure 4 shows the effect of burner on-time on the furnace efficiency. An interesting result is the leveling off of the efficiency curves at about 67% burner on-time; the profiles in Fig. 4 compare well to the behavior of the data in Table 1. Typical burner on-times are more on the order of 33%, showing some room for about a 2% efficiency gain by changing the furnace operational profile. This, of course, would be dependent on the geographical location of the installation and the margin required by the local weather characteristics.

Figure 5 shows the effects of changing the firing rate in a fixed furnace configuration. The graph also shows a comparison of the optimum head against a typical burner/furnace installation. Given a typical furnace operating at 1.0 ml/s with a calculated 77.5% efficiency, replacing the burner head with the optimum head was calculated to increase the furnace efficiency to 84.4%. However, to maintain the same rate of heat output, the firing rate could also be reduced to 0.92 ml/s (i.e., 77.5/84.4) resulting in an overall anticipated increase in efficiency of up to 7.6% (85.1 minus 77.5%) with the installation of an optimum burner head unit.

Cases 1 and 6 in Table 3 provide an efficiency comparison between an outdoor (T = 273 K) and an indoor (T = 293 K) furnace installation (or combustion air supply). The heat transfer analysis shows an improvement of

Table 3. SUMMARY OF "WAFURN" COMPUTER PROGRAM RESULTS OF A REFRACTORY-LINED COMBUSTOR, WARM-AIR FURNACE MODEL

Case No.	Firing Rate, ml/s	Cycle Timing, minutes on/off	Stoichiometric Ratio 1.15 (Optimum Head) η_{furn}, %	1.15 Draft Air Heat Loss, %	1.50 η_{furn}, %	1.50 Draft Air Heat Loss, %	1.85 (Ref. 2 Avg.) η_{furn}, %	1.85 Draft Air Heat Loss, %	Remarks
8	1.0	12/0	86.07	0.0	82.75	0.0	79.77	0.0	Steady-state
9		2/10	81.72	6.23	77.97	8.02	74.17	9.78	17% on, 12-minute cycle
1		4/8	84.40	2.62	81.01*	3.37	77.53	4.11	33% on, 12-minute cycle
6		4/8	85.38	2.17	82.24	2.80	79.17	3.43	Indoor Air (68 F), 12-minute cycle
2		8/4	86.04	0.77	82.76	0.99	79.72	1.20	67% on, 12-minute cycle
10		10/2	86.11						83% on, 12-minute cycle
3		10/20	84.73	2.12	81.43	2.65	78.18	3.14	33% on, 30-minute cycle
4	0.75	5.33/6.67	86.29	2.12	84.09	2.67	80.41	3.27	Lower firing rate, same heat input as Case 1
7	0.75	5.06/6.94	85.56*	2.31	83.38	2.91	80.01	3.58	--
5	1.25	3.2/8.8	82.22*	3.06	78.58	3.91	75.03	4.73	Higher firing rate, same heat input as Case 1

*Cases at different firing rates but having equivalent heat outputs.

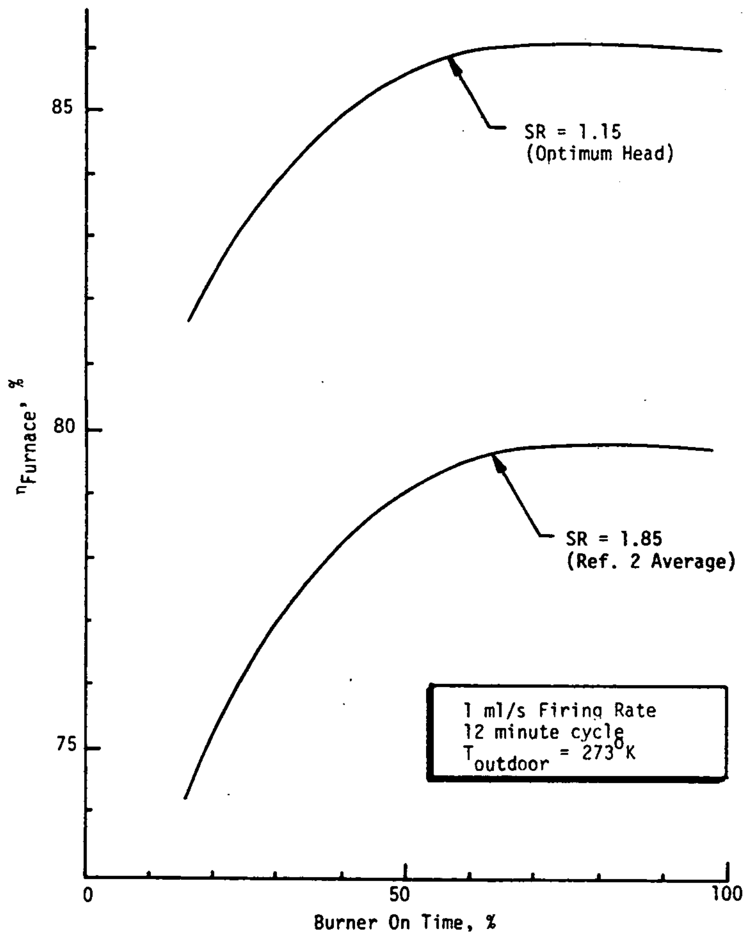


Figure 4. Calculated effect of burner on-time upon furnace efficiency

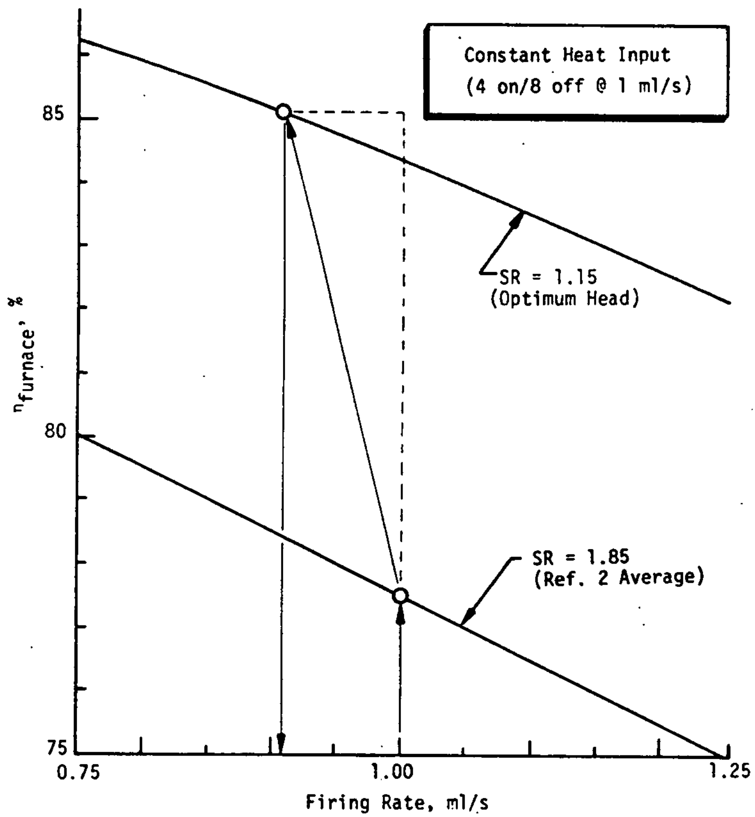


Figure 5. Calculated effect of firing rate upon furnace efficiency

about 1 to 1-1/2% in furnace efficiency by utilizing the warmer indoor air supply. However, the computer model does not account for the source of the heated (and humidified) air and, therefore, it does not show the additional 3 to 4% heat output required to heat the consumed air. The net effect of using living space air for combustion air is more on the order of **negative 2-1/2%**

Pollutant Emissions

Several definitive studies of pollutant emissions from residential oil heat systems have been conducted previously (e.g., Ref. 2, 3, and 5). Typical characteristics are indicated in Fig. 6, reproduced from Ref. 2. It is seen that there is an operating range over which the emissions of incomplete combustion products (smoke, unburned hydrocarbons, and carbon monoxide) are low. Generally, the width of this region is constrained by production of excessive smoke as excess air is reduced (increasing CO_2) and by generation of excessive CO as excess air is increased (decreasing CO_2). Examining Fig. 6, it is apparent why existing oil heating equipment is adjusted to conditions that produce, on the average, about 8% CO_2 flue gases.

Emissions differ from unit to unit. The distributions of levels of pollutants emitted from 33 residential oil heating units reported in Ref. 2 were presented graphically in that report. To ensure their accessibility to interested readers, they are reproduced here as Fig. 7. It is seen that tuning* a burner has a substantial effect on smoke, small but observable effects on CO and HC, and practically no effects on CO_2, NO_x, and filterable particulates.

*Tuning refers to the burner and heating system service procedure of cleaning, adjusting and/or replacing burner components (electrodes, blower wheel, blast tube, oil filter, oil nozzle), finding and sealing easily corrected air leaks, adjusting firebox draft, and setting the combustion air level for maximum CO_2 with minimum smoke from a stable flame.

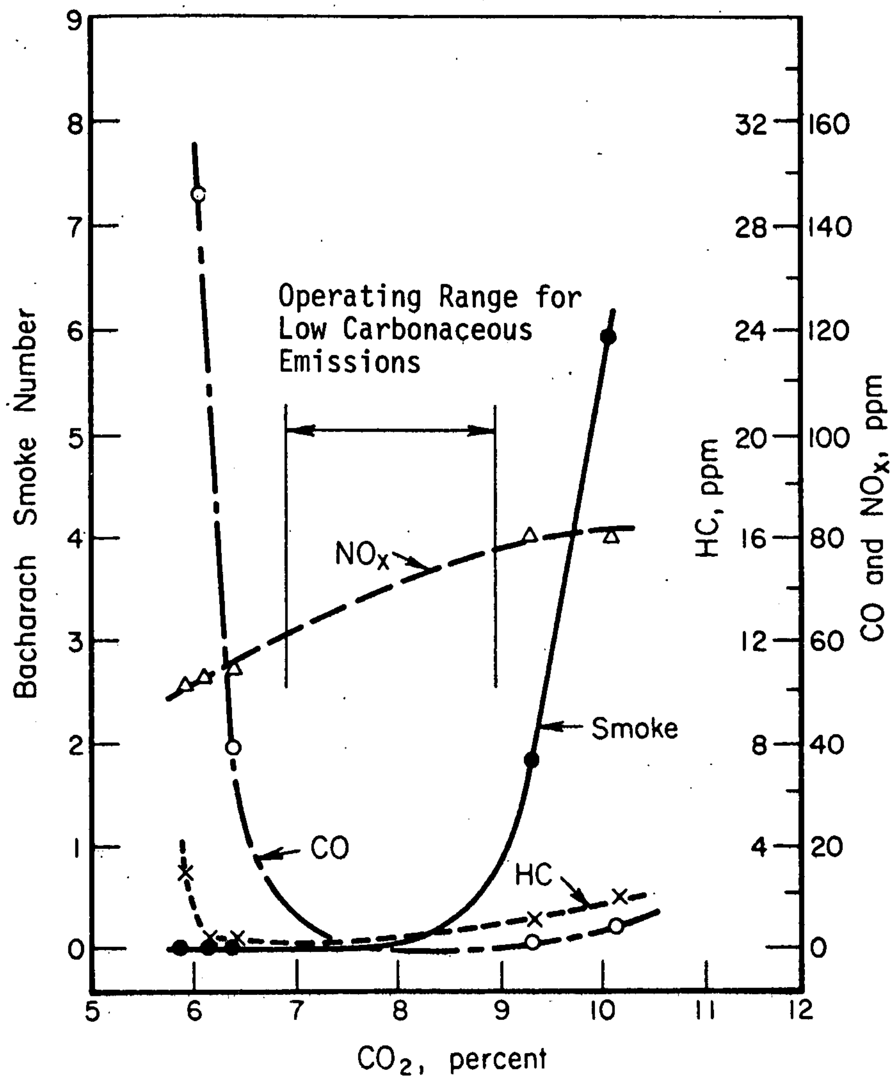


Figure 6. Typical smoke and gaseous emission characteristics for a residential unit in the tuned condition (Ref. 2)

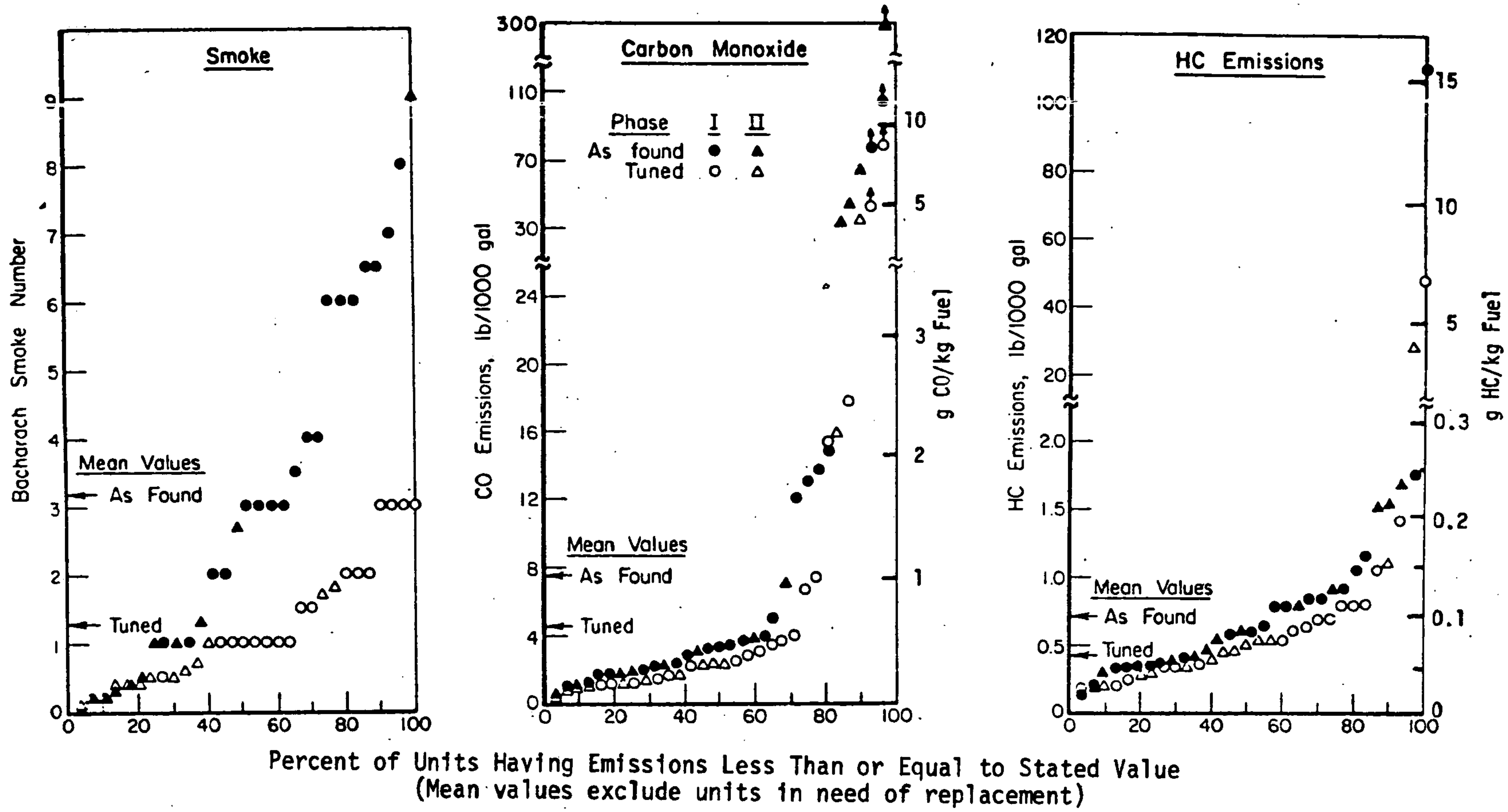


Figure 7. Distribution of smoke, CO, and HC emission for residential units (Ref. 2)

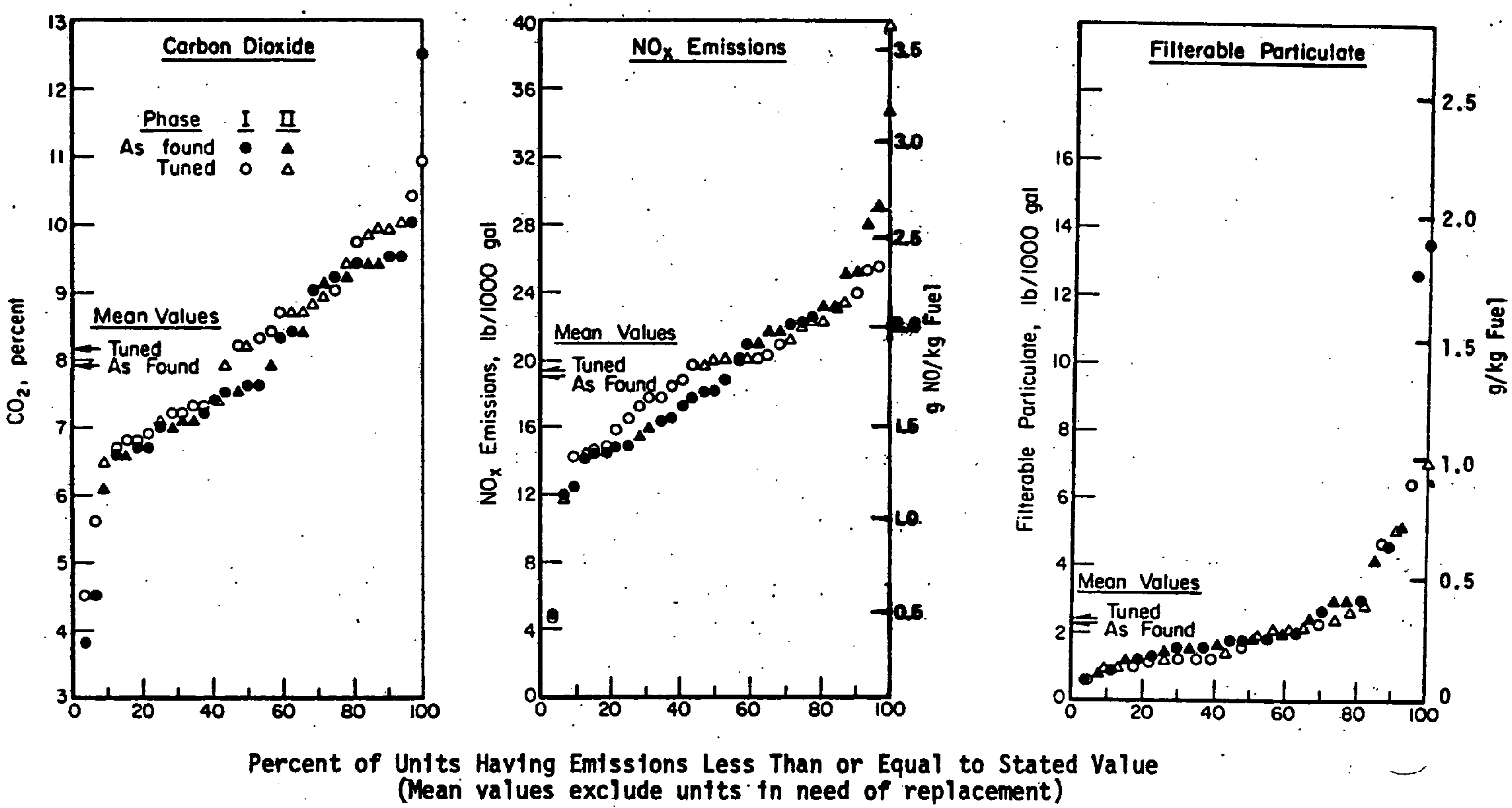


Figure 7 (Continued). Distribution of CO_2, NO_x, and filterable particulate emissions for residential units (Ref. 2)

Flue gas CO_2 concentrations for about 80% of those units field tested in Ref. 2 were linearly distributed between about 6-1/2 and 10% CO_2, which correspond to 130 and 50% excess air, respectively. The research optimum head (Ref. 5) was capable of operating smoke-free in the neighborhood of 10% excess air (13-1/2% CO_2) when fired in research combustors. Allowing a small margin for seasonal degradation, it was anticipated that burners fitted with commercially produced optimum heads could be tuned to operate with as little as 15% excess air (13% CO_2). It was also expected that emissions of CO and UHC would be acceptably low (i.e., below the "as found" mean values of approximately 1 g CO/kg fuel and 0.1 g UHC/kg fuel noted in Fig. 7) at the optimum burner's tuned condition.

In addition to a higher CO_2 level, it was anticipated that use of the optimum head would effect significant reductions in NO_x emissions. When fired continuously at 10 to 20% excess air conditions (13-1/2 to 12-1/4% CO_2) in research combustors, the optimum burner produced approximately 35 to 40% less NO than the average of several stock commercial burners (Ref. 5). More data were taken, in that study, with the tunnel-fired burner orientation than with the side-fired orientation, so the stated NO reduction is weighted toward the former configuration. Side-fired NO emission levels tended to be proportionately increased (by 50 to 100%) over the tunnel-fired levels, although there was wider diversity among the stock burners and their average NO emissions may have increased somewhat less than did the optimum burner's. This means that the reduction in NO emissions to be expected by retrofitting predominantly side-fired existing burners with optimum heads is relatively uncertain but might be as much as 35% on the average.

SELECTION OF COMMERCIAL FABRICATION METHODS

The task of selecting candidate methods for fabricating the commercial prototype heads commenced with a comprehensive assessment of current oil burner head manufacturing techniques. The evaluation included considerations of unit costs (high and low production rates), saleability, manufacturability, and design compromises or advantages, held within the restrictions of a retrofit application to existing burner/furnace systems. It became apparent early in the evaluation that any multiple-piece assembly or tooling would increase the mass production unit cost significantly, and the assessment soon narrowed to considering seriously only the one-piece design options.

The available options were reduced to what seemed to be the best three methods. A summary of these three methods is presented in tabular form in Table 4. The evaluation summarized there is based on a 1-year service life and includes considerations for both the prototype units and the mass production units. The selection criteria were weighed primarily on 100,000 units/year production rate with the higher 1,000,000 units/year figures included as reference values.

The final column of Table 4 ranks the three fabrication methods in a numerical order of preference for commercial production of optimum heads. Recommended order of preference is: (1) stamp forming of sheet metal, (2) cast forming, and (3) injection molding. The stamp-formed, sheet-metal method of fabrication was selected as the first choice primarily on the basis of its design versatility in both fabrication and application. This versatility enhances saleability of stamped sheet-metal heads with respect to those made by the other two methods. In the fabrication phase, the stamp-form tooling will allow changes in the material type and also the material thickness with only minor readjustment of the basic tooling. The design can also be made to incorporate some options that will enable it to: (1) fit a number of blast tube sizes (nominally 0.1 m diameter), and (2) accommodate a wide range of firing rates (0.5 to 3.0 ml/s). In addition, the sheet-metal method was ranked either best or next to best in most all other categories listed in Table 4. These design features are discussed in detail in a following section describing the candidate commercial prototype optimum head.

Table 4. COMPARISON OF COMMERCIAL FABRICATION METHODS FOR OPTIMUM BURNER HEADS

Comparison Items	Fabrication Method		
	Iron Casting	Sheet Metal Stamping	Injection Molding
Material	Heat resistant cast iron	430 stainless steel	Alumina Ceramic
Ease of Changing Material	Once pattern is made a large variety of cast iron materials can be used.	Can be made of any ductile sheet material.	Tooling good for only one material due to shrinkage of part after molding.
Functional Considerations	1. High temperature scaling resistance may be problem with regular grey iron. May require heat resistant alloy. Will run cooler than sheet metal.	1. Holes around edge allow some air leakage. 2. Some warpage may occur during heating.	1. Parts will crack if thermal shock is too great; must be verified during test. 2. Brittle- will break if dropped on hard surface. 3. No problem with oxidation at operating temperature.
Geometry Compromises	1. Spiral vanes to simplify production process. 2. Shorten vanes and remove from center hole to make more castable. 3. Increase thicknesses and provide fillet radii to allow casting. 4. Increase vane angle to 30°.	1. Reduction in vane length to minimize amount of material used. 2. Can be made to original configuration at ~$0.15 extra/part. 3. Increase vane angle to 30°.	1. Reduction in vane length to allow molding. 2. Thicker sections and fillet radii to allow molding. 3. Increase vane angle to 30°.
Ease of Varying Geometry	1. Can machine off vanes or enlarge center hole. 2. May be able to weld or braze additional material on vanes or hole.	1. Vanes can be bent to different angles if required. 2. Material can be welded to vanes or center hole. 3. Vanes or center hole can be trimmed.	1. Part cannot be varied once made.
Fabrication Tolerances	1. Production tolerances ±0.8 mm (0.03 inch) 2. Vane angle ±0.5° 3. May require machining O.D. to obtain indexing step with sharp corner radius.	1. Center hole ±0.25 mm (0.010 inch) 2. Vane angle ±0.5°	1. Center hole ±0.13 mm (0.005 inch) 2. Vane angle ±0.5°. May sag during sintering. Might require special support blocks.
Ease of Changing Center Hole Diameter in Production	Requires minor tooling cost, ~$1000/diameter change	May be able to use knockout ring to adjust on installation ~ $2000 additional tooling or can change hole size in tooling for minor additional cost ~$1000/change	Can change hole size in tooling for minor cost if optional hole is planned for.

Table 4. (continued). COMPARISON OF COMMERCIAL FABRICATION METHODS FOR OPTIMUM BURNER HEADS

Comparison Items	Fabrication Method		
	Iron Casting	Sheet Metal Stamping	Injection Molding
Installation Method	Match drill holes in tube and head and attach with drive screw.	Match drill holes in tube and head and attach with sheet metal screw.	Drill hole in tube to match hole in head, secure with sheet metal screw through tube.
Expected Life	10 year goal, verify during test. Determined by oxidation resistance.	10 year goal, verify during test. Determined by oxidation resistance.	10 year goal, verify during test. Determined by thermal shock resistance.
Sales Features	1. Looks rugged 2. Rough surface finish	1. Good appearance, light and easy to carry. 2. Easy to install. 3. Knockout center hole to adjust air velocity.	1. Good looking white part with good surface finish.
Estimated Costs			
• Two Prototypes	• $2057 • 4 weeks delivery	• $400 • 4 weeks delivery	• $2600 • 4 weeks delivery
• 100,000/Year	$1.50 each, including $10,000 tooling	$1.29 each, including $27,000 tooling	$1.65 each, including $90,000 tooling
• 1,000,000/Year	$1.40 each, including $10,000 tooling	$1.12 each, including $27,000 tooling	$0.72 each, including $245,000 tooling
Comments	1. Poor tolerance on I.D. and O.D. without costly machining		1. Not as easy to install 2. Not fully developed fabrication process, more risk in meeting schedule and more uncertainty in production cost. 3. No existing production facility. Must set up related buildings, etc., to house equipment.
Recommendations	Number 2 choice because of reasonable production cost, and tooling cost.	Number 1 choice because of cost, lightweight, saleability, development versatility, installation ease, relatively low tooling cost, possibility of using knockout ring in center hole to reduce inventory requirements.	Number 3 choice because of risk of part cracking, lack of development versatility, lack of production facility, large capital investment required.

The casting method of fabrication was selected as the second best candidate for a retrofit, commercial prototype head. Its ruggedness and simplicity offer very saleable features to both the serviceman and the customer. It is a well-proved and accepted manufacturing method in the oil burner industry. At a production rate of 100,000 units/year, its estimated $1.50 cost is competitive. However, due to the large amount of material required (~0.5 kg/head) and additional labor costs (minor machining), its estimated cost at a much higher rate of 1,000,000 units/year shows only a slight decrease of $0.10. Additional distribution costs will also be experienced if the heavier cast heads must be shipped over long distances.

The injection molding method of construction was selected as third best candidate for fabrication of commercial optimum heads. It has several advantages, of which the very low estimated unit cost of $0.72/head would be a major consideration for high output production. Many of its features are comparable to the cast-formed head. However, there certainly would be some development effort required to produce a satisfactory final product. This, coupled with the high initial capital investment required for tooling, makes it the least attractive of the three fabrication options.

DESIGN OF COMMERCIAL PROTOTYPE OPTIMUM HEADS

Initial Stamped Sheet-Metal Heads

For the preferred sheet-metal stamping fabrication method, an optimum head design concept was selected so that the entire head can be stamped and formed from flat sheet-metal stock. Figure 8 is a layout drawing illustrating the design concept. The right-hand view is a composite showing the plan view of the initial flat stamping before the six swirl vanes are folded up and a rear view of the prototype head after folding the vanes. This design incorporates "sprung" vanes, folded to 83 degrees rather than a full 90 degrees, so that the OD of the vanes' outer edges

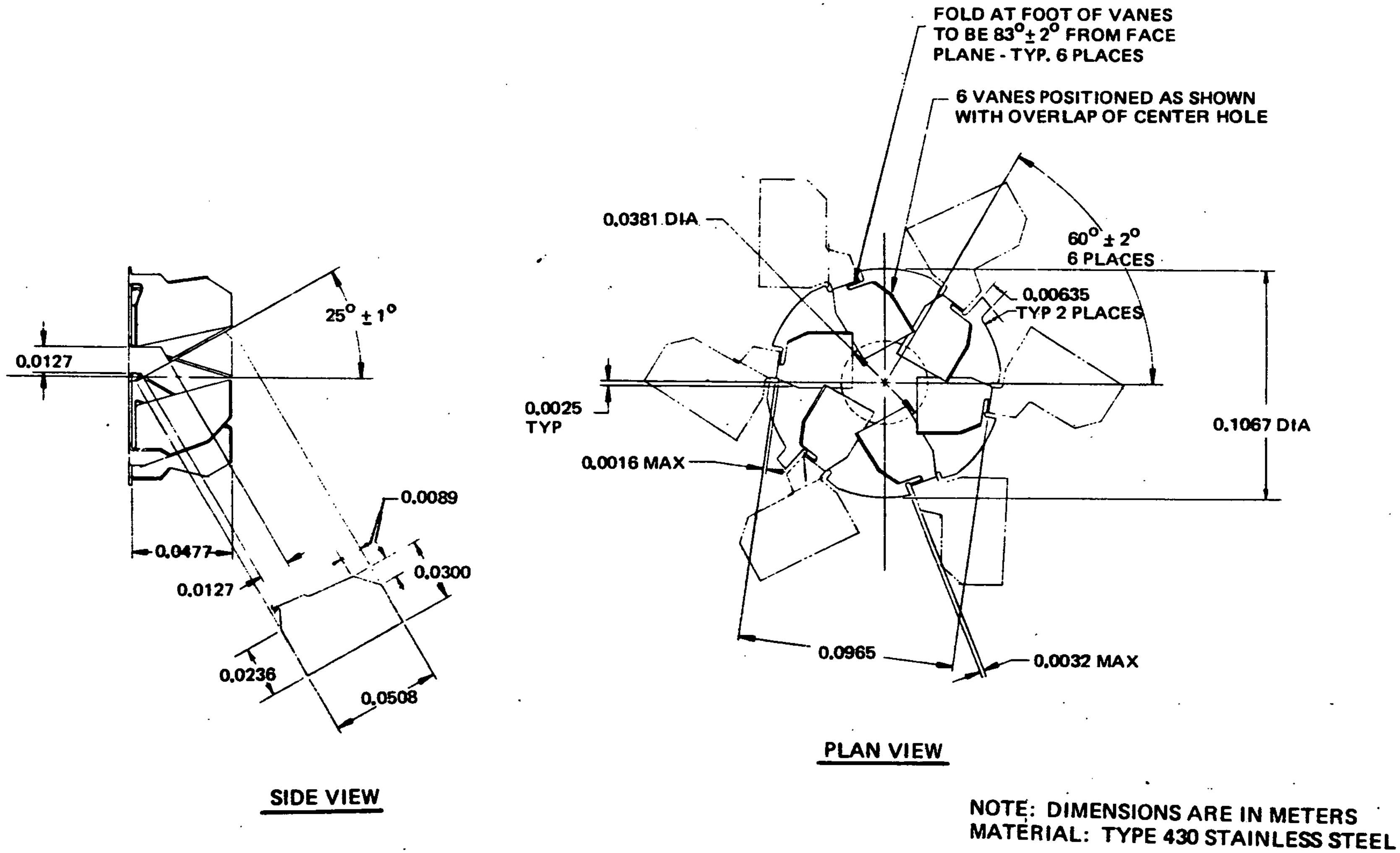


Figure 8. Layout drawing of the stamped-formed, sheet metal prototype optimum burner head

is slightly larger than the ID of a burner's blast tube. This allows snug fitting and self-centering of the head in a variety of commonly found blast tube diameters (0.102 to 0.108 m OD). Fold tabs were added to the outer perimeter of the choke plate for screw attachment to the larger diameter blast tubes.

An attractive feature of this design is that prototype heads can be made that duplicate the essential features of the machined and welded research optimum head tested before (Ref. 5). Thus, the design of Fig. 8 has the same number of swirl vanes having the same length, width, and orientation as the research head had. Similarly, the choke plate and its simple central circular opening simulate those of the research heads very well. The only basic discrepancy between this design and the earlier head is the less-than-complete closure of the joint between the head and the blast tube; perhaps as much as 15% of the combustion air could leak through the small openings where the swirl vanes are folded away from the choke plate. They were left as large as they are to accommodate making the first few prototype heads by manual shearing and folding simulations of stamping operations. For actual commercial production, it was believed that careful attention to stress considerations and tolerances would allow substantial reduction of these openings. Similarly, the outer edge of the choke plate was recognized as being rather jagged and unattractive; in a commercial stamping operation, thus undoubtedly could be finished in a way that would both strengthen and beautify the head as well as provide for attachment to the blast tube.

Production design could also incorporate a series of partially cut concentric rings around a minimum size center hole, allowing the serviceman to "knock out" rings to adapt the head to any firing rate from 0.5 to 3.0 ml/s, requiring stocking of only one "universal" size head in his inventory for residential heating units.

Revised Stamped Sheet-Metal Heads

After testing prototype heads of the foregoing design (Section V), design modifications were made so that the heads would be less susceptible to metal scaling and dimensional distortion caused by exposure to intense thermal loads and temperature gradients. One principal design modification was provision of a recessed channel section in the previously flat choke plate (Fig. 9). The strengthening channel design was selected because it offered a minimum of compromises over the goals of the original prototype head design. The channel design provided rigidity at both the perimeter and near the center of the choke plate. The required tooling was simple and amenable to mass production.

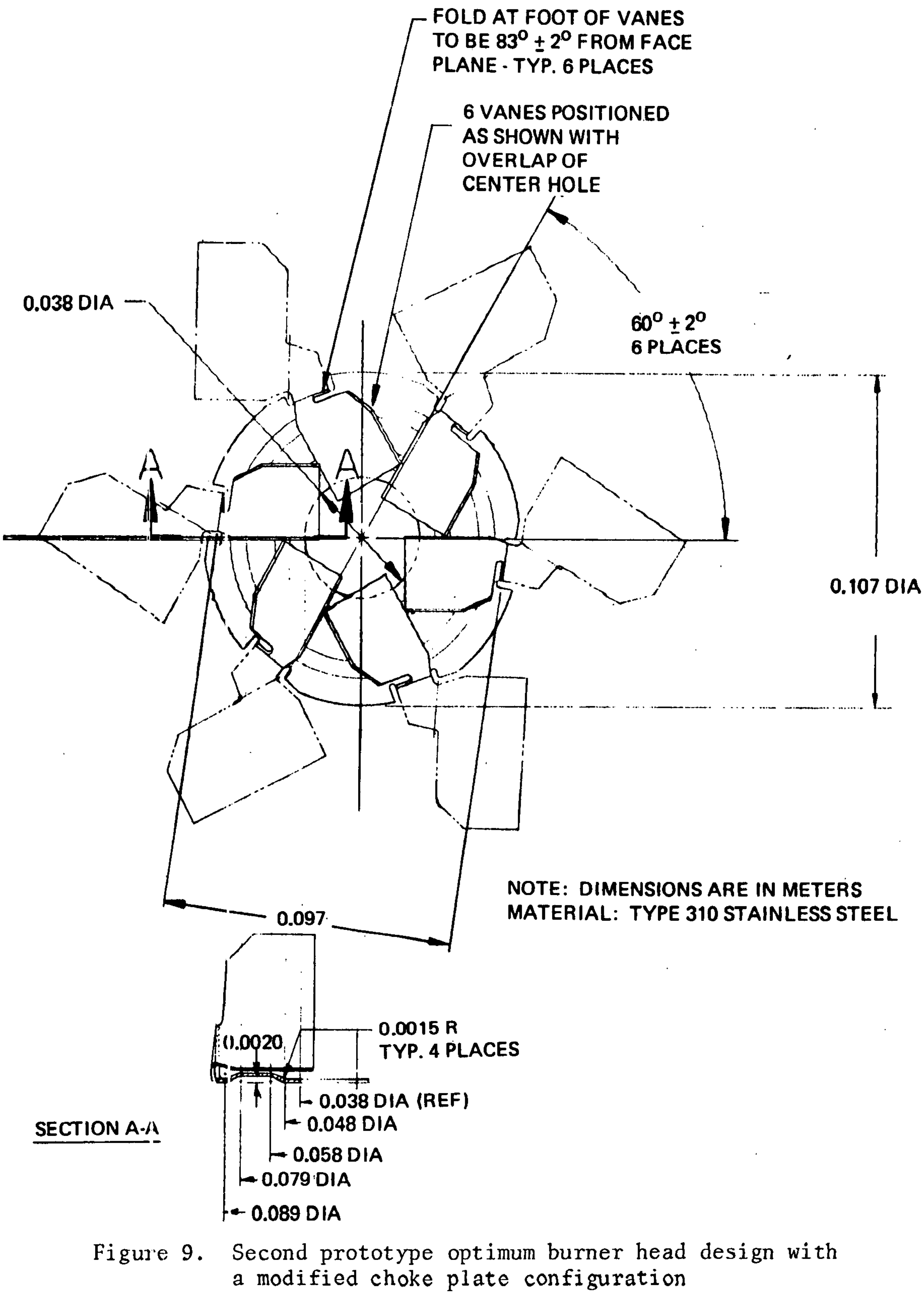


Figure 9. Second prototype optimum burner head design with a modified choke plate configuration

SECTION V

EXPERIMENTAL INVESTIGATIONS

FABRICATION OF PROTOTYPE COMMERCIAL HEADS

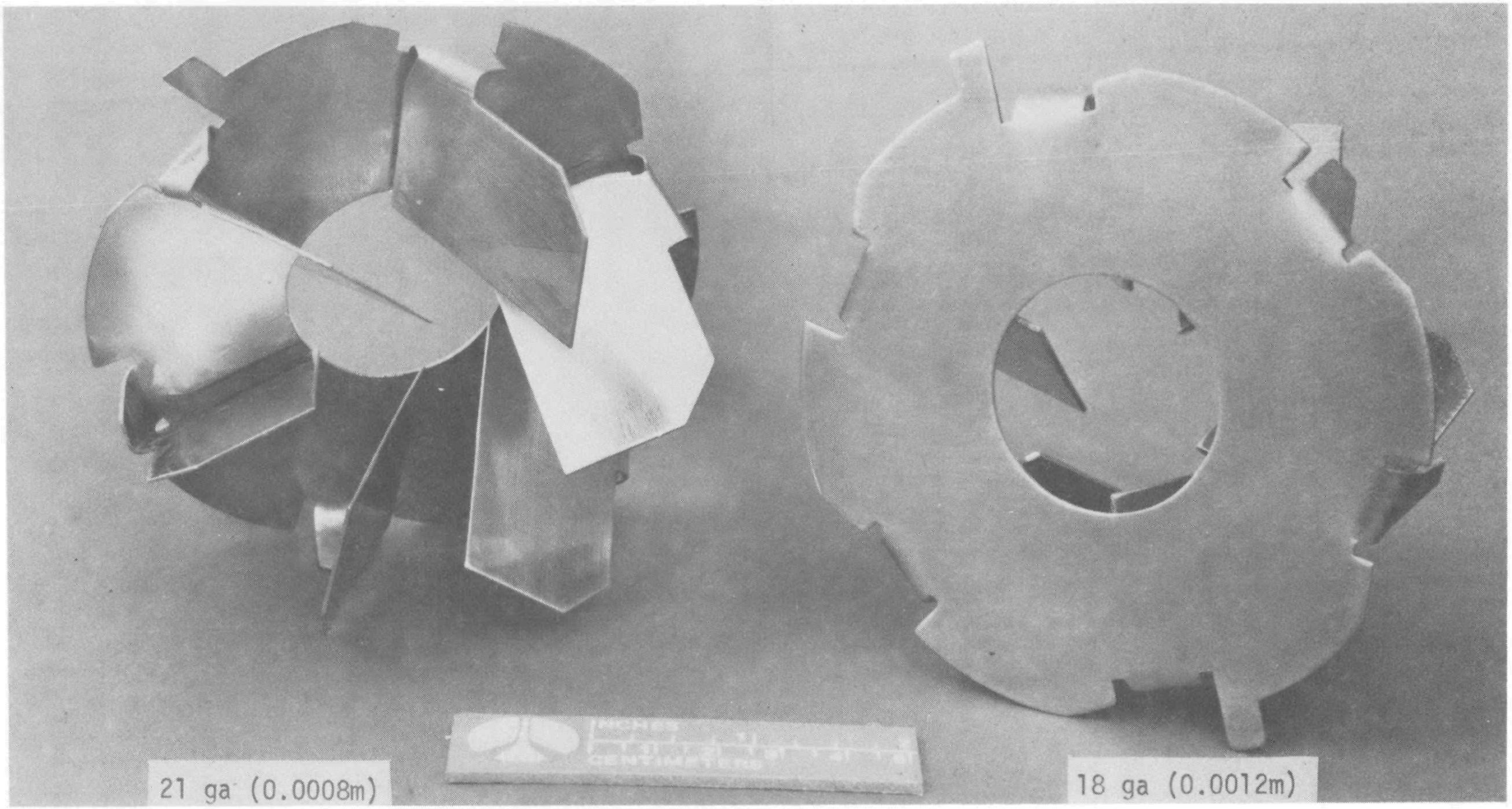
Initial Design

A fabrication bid package for prototype sheet metal optimum heads was submitted to several local commercial shops having sheet metal fabrication capabilities. The low bidder was selected to fabricate two prototype optimum heads using commercial shop practices with minimum tooling to simulate the product which would eventually result from volume stamping operations.

The initial prototype design of Fig. 8 was used. One head was made of 18 gage [0.00127 m (0.050 inch)] and another of 21 gage [0.00087 m (0.034 inch)] Type 430 stainless steel sheet. The thicker (18 gage) prototype optimum head was the primary design choice and was expected to endure the testing schedule with little or no degradation. The second (21 gage) unit was built to explore the effects (and limits) of thinner (i.e., lower cost) stock material construction. A photograph of these two initial heads is shown in Fig. 10.

Revised Design

As described and discussed in the next subsection, the initial sheet metal prototype optimum heads experienced substantial thermal distortion and exhibited inadequate resistance to scaling of the metal. Therefore, the design was modified to strengthen the choke plate, and a more refractory grade of stainless steel was selected.



50P44-9/2/75-S1B

Figure 10. Photograph of two initial design sheet metal prototype commercial optimum oil burner heads constructed from type 430 stainless steel

The research optimum head was made of Type 321 stainless steel. The composition and some other characteristics of Types 430, 321 and some other candidate stainless steels are listed in Table 5.

Table 5. COMPOSITION OF VARIOUS TYPES OF STAINLESS STEELS*

AISI TYPE Stainless Steels	Cr, %	Ni, %	C, %	Other, %	Scaling Temperature, °C	Approximate Price $/kg
304	19	10	0.08 max	-	900	2.20
310	25	20	0.25 max	-	1125	3.30
321	18	10	0.08 max	~0.4 Ti	900	2.20
430	16	-	0.12 max	-	800	1.75

*Base metal - iron

Type 304 stainless steel has composition and scaling resistance very close to those of Type 321, so it may be a good candidate for stamped sheet metal heads. However, the 18 gage (0.00127 m) sheet metal prototype head's choke plate is only about half as thick as that of the research head (0.0025 m) so the warping characteristics may be inadequate. Therefore, it was decided to use Type 310 stainless steel for the revised design prototype heads, even though this material costs about 50% more than Type 304.

Two heads, one made of 18 gage and one of 21 gage Type 310 stainless steel sheet according to the revised design of Fig. 9, were procured subsequently from the same commercial shop which had made the initial prototype heads. Also, because the cost of additional test units was very low once the vendor's patterns and jigs were established, a comparable pair of heads was made from Type 304 stainless steel. The Type 310 heads were considered to be the primary set. The Type 304 heads were kept for backup and, if the Type 310 heads were found to be satisfactory, were to be exposed to cyclical furnace firing at some convenient time to gain at least a preliminary assessment as to whether the less costly Type 304 stainless steel might also be satisfactory. Figure 11 is a fire-side face-view photograph of one of these revised design prototype commercial heads.

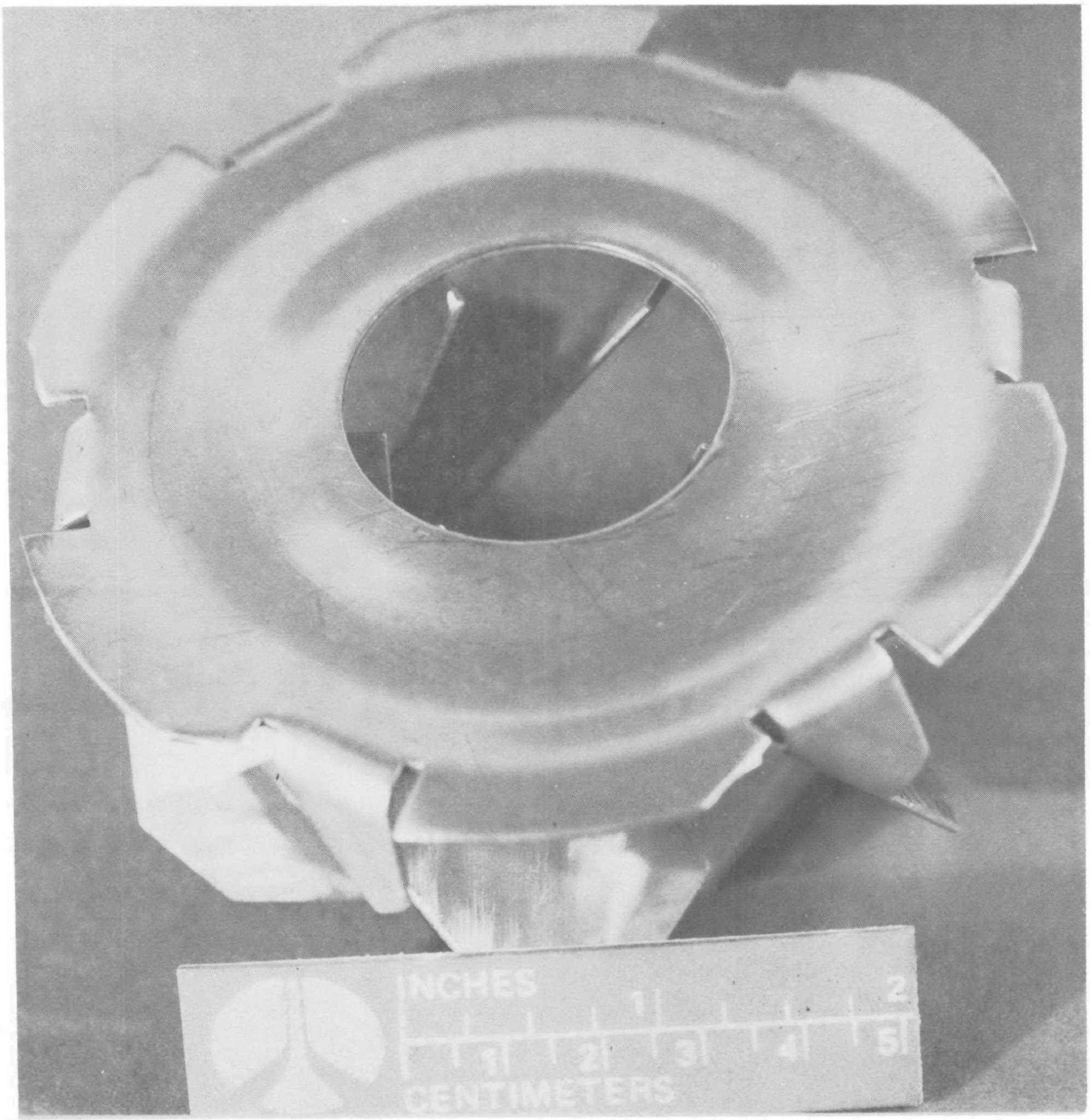
PERFORMANCE OF PROTOTYPE COMMERCIAL OPTIMUM HEADS IN RESEARCH COMBUSTION CHAMBERS

The first experimental tests of the prototype commercial optimum heads were carried out in laboratory research combustion chambers, rather than in residential furnaces. An early comparison was desired between their pollutant emission performance and that of the prior research optimum head. Most of the prior experience with the research head had been in research combustion chambers, so that was the most appropriate vehicle for such a comparison.

Experimental Apparatus

The most common combustion chamber design in residential oil heating units is an uncooled, refractory-lined cylindrical chamber, approximately 8 to 10 inches inside diameter, with a vertically disposed axis and a horizontally disposed, side-fired burner orientation. Therefore, the testing was begun with the 1 ml/s (gph) optimum burner side-fired in an uncooled 0.22 m (8.75 inch) inside diameter cylindrical chamber lined with 0.03 m (1.2 inch) thick refractory fibre (Pyroflex) insulation. Later, tests were conducted with the burner tunnel-fired in the same chamber.

Figure 12 depicts schematically the tunnel-fired combustion chamber arrangement, with a fibre refractory liner in one end of the chamber and a movable, water-cooled heat exchanger inserted in the other end. The side-fired configuration was achieved simply by turning the combustor end-for-end, with the refractory liner, heat exchanger, blank flange, and burner-port flange appropriately relocated.



50P34-11/24/75-S1

Figure 11. Photograph of modified prototype commercial head showing the 0.0020 m deep reinforcement channel (18 gage, type 310 stainless steel)

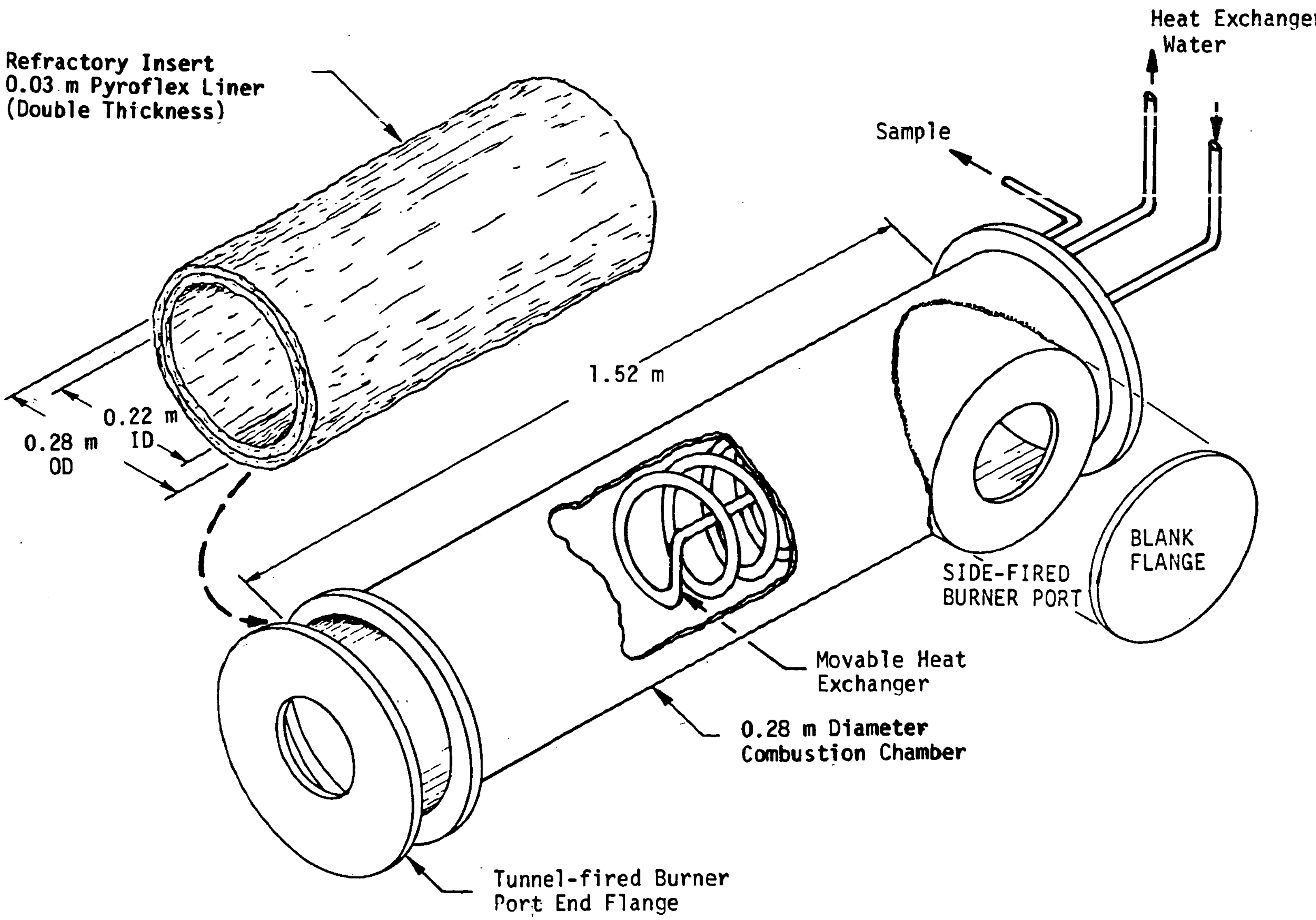


Figure 12. Experimental combustion chamber and heat exchanger arrangement

The water-cooled heat exchanger was used as a convenient means of rapidly quenching the combustion product temperature and was made movable so that heat exchanger position (i.e., firebox length) could be readily varied to observe its effects on pollutant emissions. It consisted of a nested double coil of 0.013 m (0.50 inch) copper tubing and had outside dimensions of approximately 0.15 m (6 inch) diameter and 0.76 m (30 inch) length. Several semicircular baffle plates were cut from 21 gage stainless steel sheet and were slipped between coils at regular intervals, from alternate sides, to ensure that combustion products passed repetitively over the coils and did not bypass around the outside of the coils.

The research combustor was tested at an outdoor facility depicted schematically in Fig. 13. The principal components were attached to a waist-high steel table as shown. Not shown is a Unistrut superstructure at the right-hand end of that table to support the vertically mounted combustion chamber and to suspend the spiral-wound heat exchanger within it. The facility was organized for rapid and easy changing of combustion chambers, burner orientation, and heat exchanger position. Minimum protection from inclement weather was provided by a simple sheet metal roof over the test apparatus.

Experimental data requirements were primarily concerned with flue gas pollutant concentrations. Concentrations of most pollutant species were measured by conducting a continuous flue gas sample to a train of analysis instruments located indoors in a nearby laboratory. Flue gas smoke content was measured intermittently at the flue with a manual smoke meter. The instruments used, analyses performed, and types of data obtained are described and discussed in Appendix A. In addition, the firing rate was monitored regularly by measuring the fuel oil flowrate, the flue gas temperature was indicated by inserting a thermocouple downstream of the heat exchanger, and the temperature rise of the heat exchanger coolant water was measured. Miscellaneous data taken less

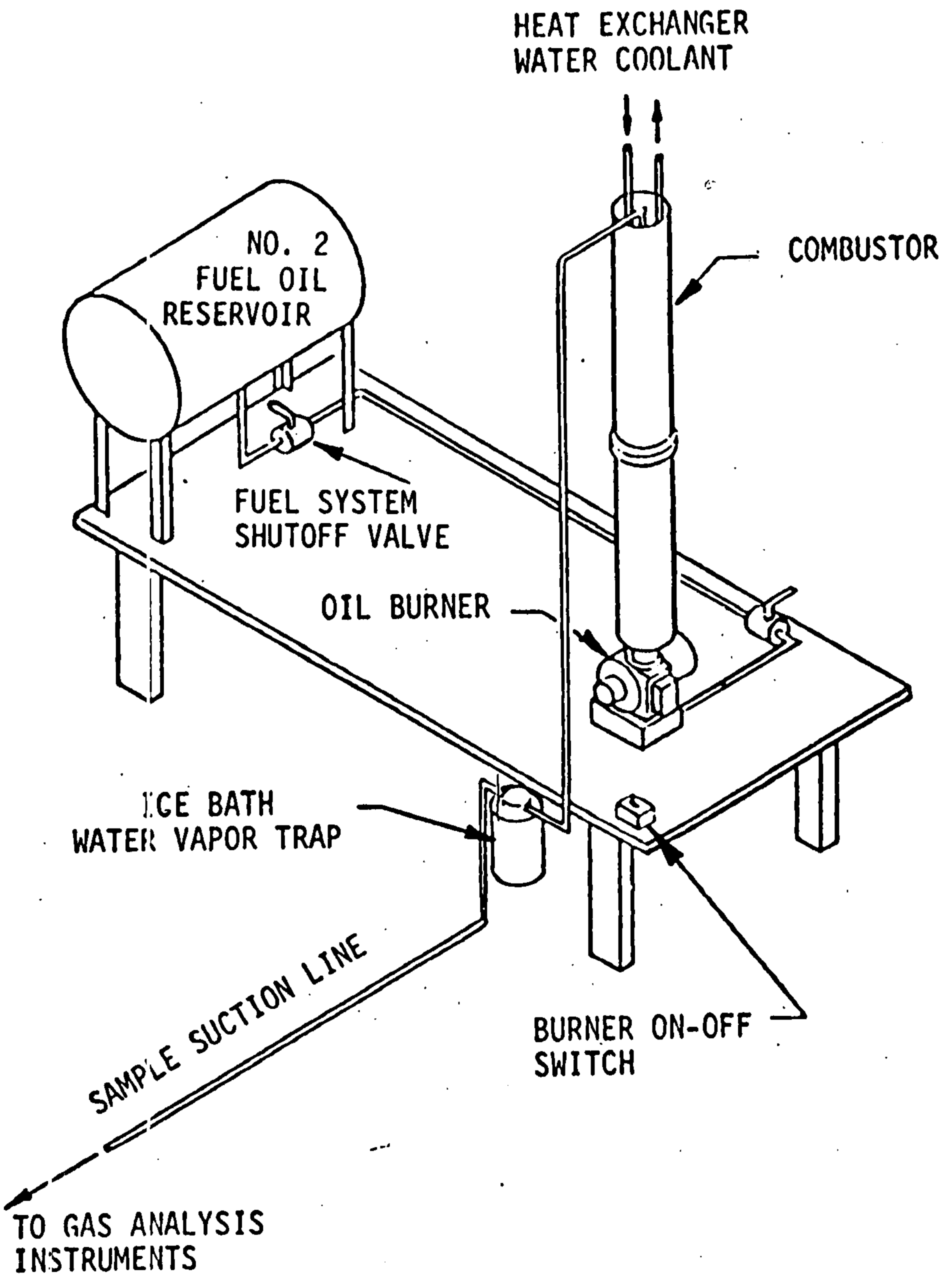


Figure 13. Schematic of oil burner and research combustion chamber test installation

regularly were firebox draft conditions, firebox shell metal temperatures, and combustion air fan characteristics.

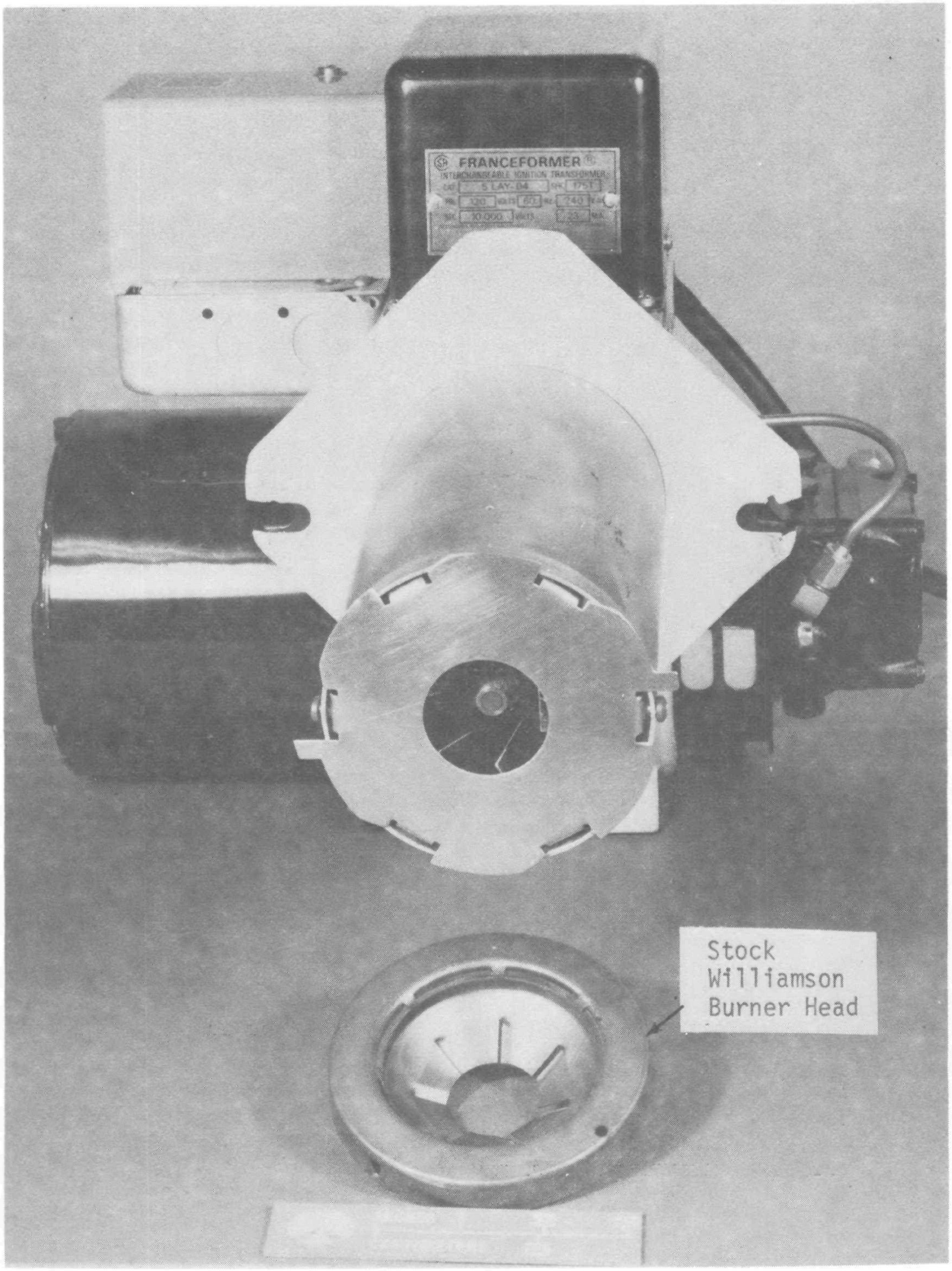
Experimental Results

The cycle-averaged pollutant emission results are tabulated in Appendix B by run number. The operational results are described in this subsection, along with a discussion of both types of results.

The first series of tests (runs 463 to 469) was made with the initial 21 gage sheet metal prototype optimum head on a burner (Fig. 14) side-fired in the refractory-lined research combustor. The emission results were entirely satisfactory, but the head did not stand up very well to the thermal load to which it was exposed. The photograph in Fig. 15 **(a) shows the condition of the head after approximately 15 hours of** simulated furnace operation.* The view is from the bottom side of the burner's blast tube, where the scaling and warpage were the most severe. The heat-induced scaling showed a vertically oriented pattern with the greatest scaling at the bottom of the burner head. Maximum warpage of the 21 gage head was approximately 0.0064 m (0.25 inch) from the original face plane. The distortion was apparently caused by the flame during the burner-on period rather than by overheating during the standby period, since there was very little evidence of a matching high temperature discoloration pattern on the back side of the choke plate. Although the warpage was estimated to increase the air leakage around the head's perimeter by about 12% of the total air flow, the nitric oxide emission results were quite comparable with the approximately 2 g NO/kg fuel burned observed in earlier tests of the research optimum head (Ref. 11).

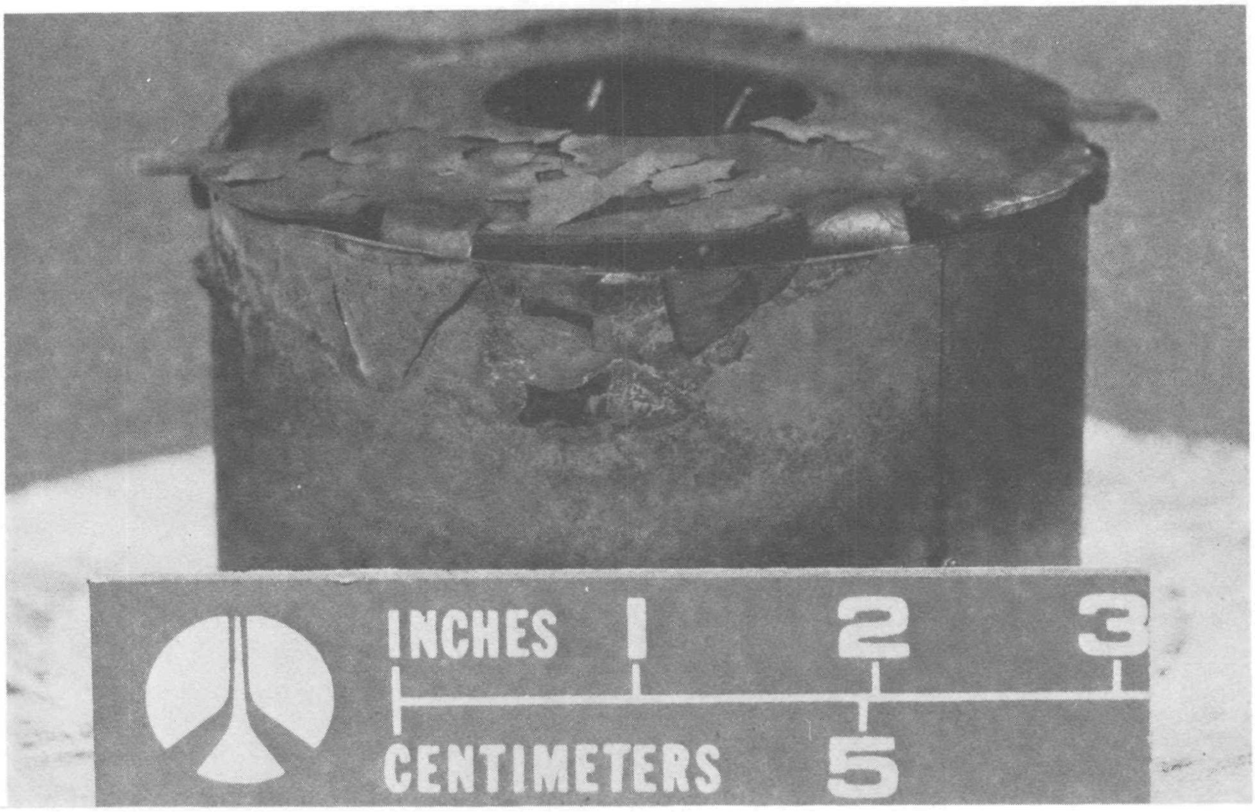
Because of the thermal distortion experienced, the 21 gage head was replaced with the 18 gage head. Also, in an attempt to relieve the

*The 21 gage prototype optimum head was fired for about 8 hours prior to run 463 to cure a new refractory fiber lining in the combustion chamber and that time is included in the stated 15 hours.



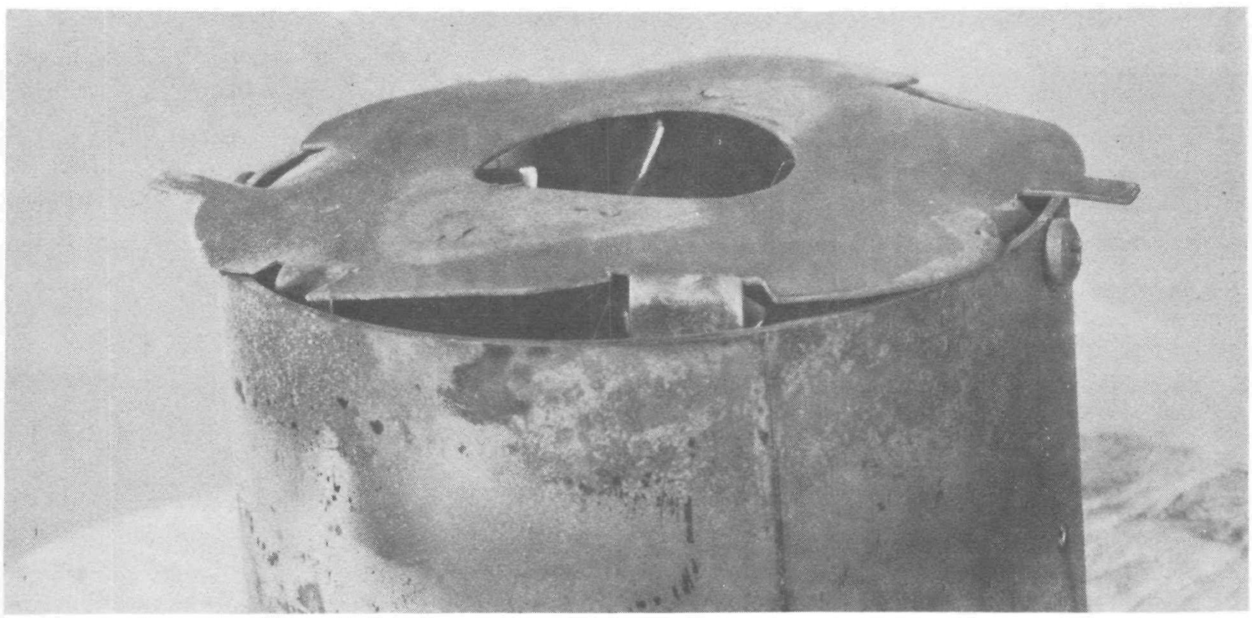
5ZZ31-9/5/75-S1

Figure 14. Photograph of the initial 21 gage sheet metal prototype commercial head installed on the Williamson burner body



5ZZ36-9/15/75-S1A

(b) 18 gage (0.00127 m), type 430 stainless steel head



5ZZ36-9/9/75-S1

(a) 21 gage (0.00087 m), type 430 stainless steel head

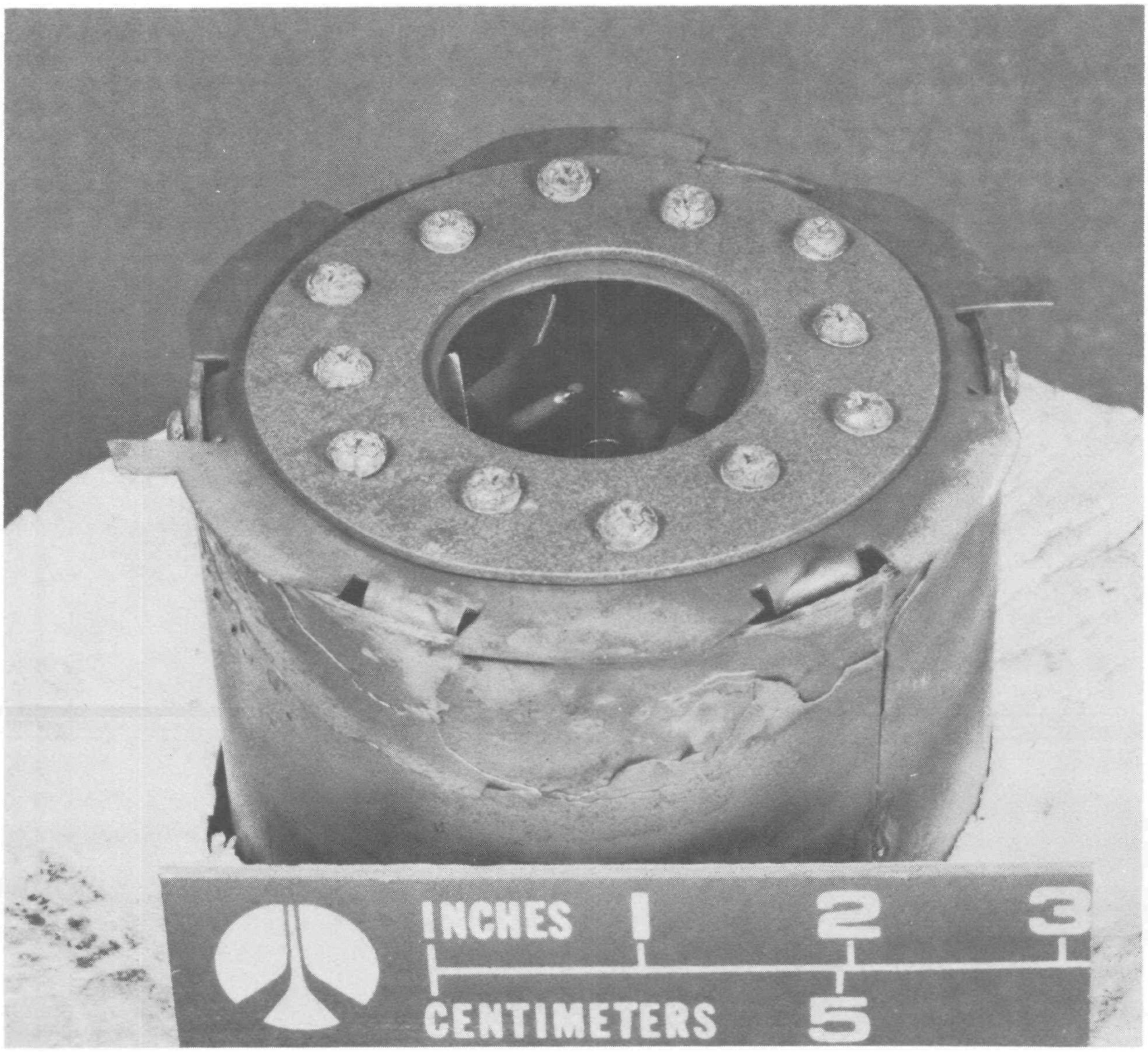
Figure 15. Photographs of the initial sheet metal design, prototype commercial heads after approximately 15 hours of cyclic service in a 0.22 m I.D. insulated side-fired combustor

thermal load on the head somewhat, the 60 degree spray angle nozzle was replaced with a 30 degree nozzle. This combination was fired in runs 470 through 481 with two different heat exchanger positions. Again, the pollutant emission results were quite comparable with earlier experience, but the heavier 18 gage head also suffered metal scaling and distortion. Figure 15 (b) is a photograph of this 18 gage head after approximately 15 hours of service and it shows somewhat more scaling, but less distortion (~0.003 m) than the 21 gage head.

Also evident in Fig. 15 is substantial scaling of the burner blast tubes to which the prototype sheet metal heads were attached. This phenomenon was not studied to establish whether or not it was related to the head distortion and scaling. However, circumstantial evidence suggests that it was: the blast tubes had not shown evidence of scaling in earlier tests of the burners' stock heads, and further scaling was not observed in subsequent testing after the head geometry was stabilized.

To continue with the proof-of-concept firings, a 0.0020 m (0.080 inch) 300 series stainless steel reinforcement plate was added to the 21 gage head to combat the scaling and warping problem. The plate was sized so as not to change any air flow characteristics of the original sheet metal design, especially the peripheral air leakage. Testing was then resumed with frequent inspections between firings. Figure 16 is a photograph of the reinforced 21 gage head after approximately 15 hours of hot-fire service, showing only slight oxidation and no distortion of the flat choke plate. Therefore, testing in the research combustor was carried through to completion using this reinforced prototype head.

The nature of the test matrix conducted is shown in Appendix B. Several variations were made, for both side-fired and tunnel-fired burner orientations, in burner firing level, oil nozzle spray angle, and operating stoichiometric ratio. Two heat exchanger positions were used,



5ZZ31-9/18/75-S1

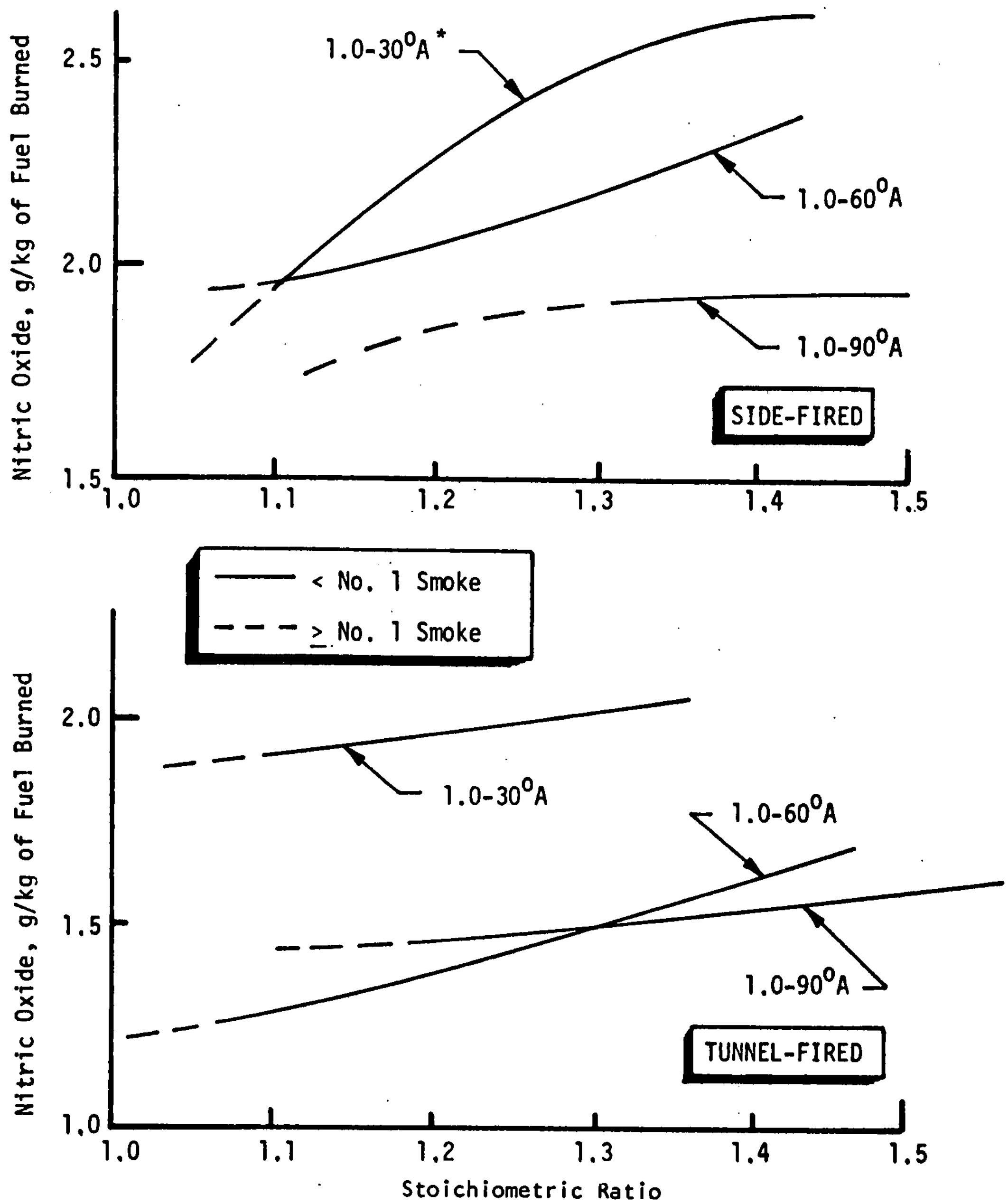
Figure 16. Photograph of the 21 gage type 430 stainless steel prototype commercial optimum head with a 0.0020 m 300 series stainless steel reinforcement plate after approximately 15 hours of service

with a greater number of tests conducted at the 0.75 m position. Additionally, short test series were made to investigate the effects of turning the spark igniter off immediately after ignition and of shortening the cycle time from 30 minutes to 12 minutes.

Operationally, the burner with the prototype optimum head behaved the same in this combustion chamber as it had earlier with the research optimum head. Smooth burning was experienced over the entire operating ranges where emissions were satisfactory, although some noisy combustion did occur on startup when the tunnel-fired burner was fired at low-excess air conditions. In agreement with the earlier results, many combinations of design and operating variables permitted operation with as low as 10% excess air without producing smoke exceeding No. 1 on the Bacharach scale.

The emissions of nitric oxide conformed to those from earlier research optimum head testing in several ways. Variations of NO with oil spray cone angle and stoichiometric ratio are plotted in Fig. 17 for both side-fired and tunnel-fired arrangements. As before, the 60-degree spray angle gave the best overall results (i.e., operation at low stoichiometric ratio with low NO and low smoke) for both burner orientations. Direct comparisons of NO emissions for the two heads in the two burner orientations are presented in Fig. 18 and show some differences in the tunnel-fired configuration, but a very close correspondence between the results with the research and prototype optimum heads in the side-fired configuration.

The prototype head demonstrated good firing rate flexibility in the 3/4 to 1-1/4 ml/s (gph) range (runs 502 to 512 and 542 to 550). Also, the interrupted igniter tests, in which the spark was turned off immediately after ignition, showed that only slightly lower cycle-averaged NO emissions (0 to 14 ppm) might result from adopting this change for the burner firing sequence in the side-fired orientation. This corresponds



*Spray nozzle callouts designate Firing Rate (gph) and Spray Angle (degrees). The "A" signifies a hollow-cone spray.

Figure 17. Effect of oil nozzle spray cone angle upon flue gas nitric oxide concentrations using prototype commercial heads in the 0.22 m I.D. insulated cylindrical research combustion chamber

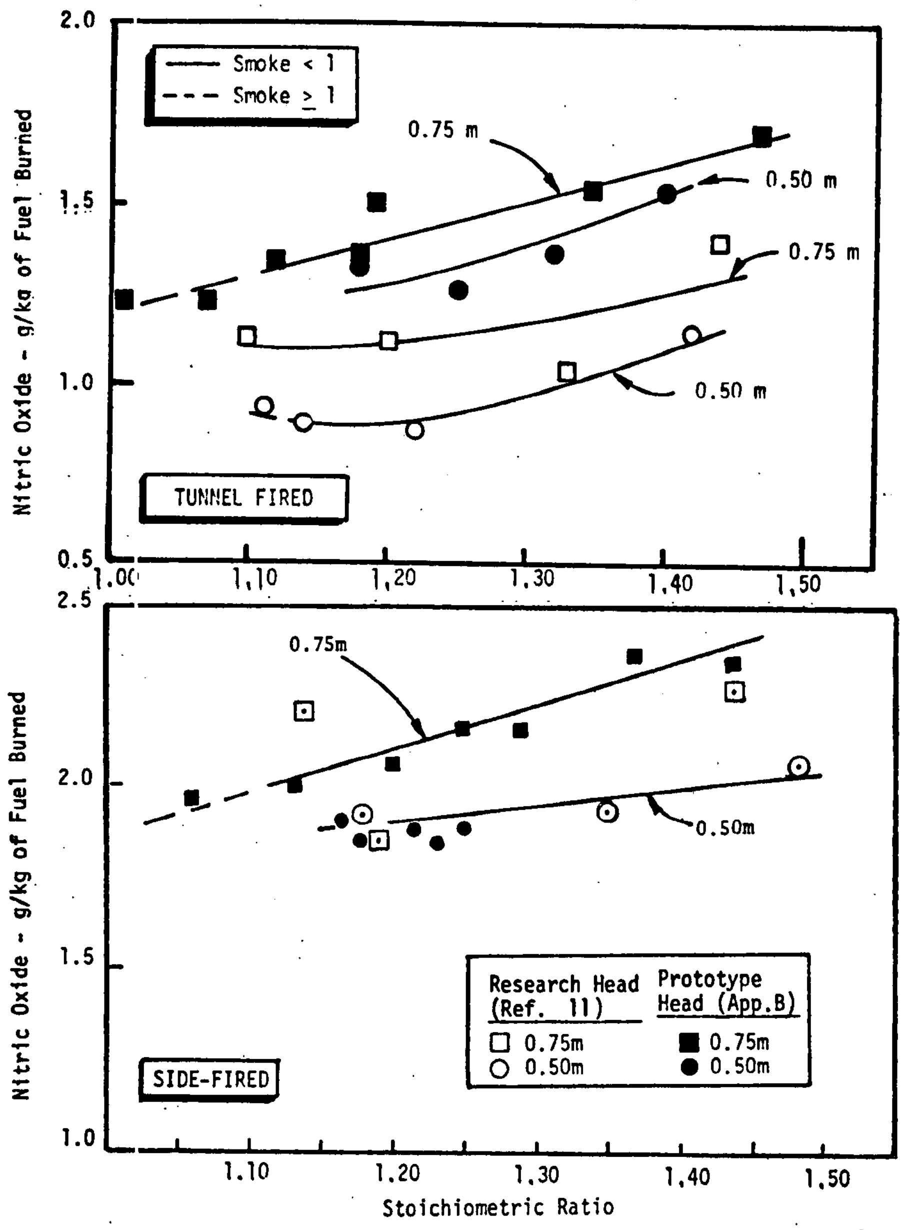


Figure 18. Comparison of flue gas nitric oxide concentrations between the experimental optimum burner head and the commercial prototype optimum burner head in a 0.22 m I.D. insulated cylindrical combustor

with the conclusion from earlier research head testing that an interrupted spark would have no appreciable effect on NO emission levels.

It was concluded that the operational and emissions characteristics of the initial prototype sheet metal head were close enough to those of the research head to support proceeding with its evaluation in commercial residential furnaces. However, it was decided to do that with the design revised to eliminate the untenable metal scaling and thermal distortion experienced with the initial single-piece design. Rather than re-testing the revised design heads in the research combustion chamber, however, it was decided to conduct some preliminary furnace evaluations with the strengthened initial-design prototype head (Fig. 16) to provide data for a comparison basis.

PERFORMANCE OF PROTOTYPE COMMERCIAL OPTIMUM HEADS IN RESIDENTIAL FURNACES

Suitability of the prototype commercial optimum heads as retrofit devices for existing residential furnaces was investigated experimentally. Two commercially available furnaces were selected as being representative of a large fraction of the designs in the existing population of residential space heating systems. New units were acquired and were tested in a laboratory simulation of field operation. The stock furnaces' thermal and pollutant emission performances were characterized before their burners were retrofitted with the prototype optimum heads. This provided a comparison basis for evaluating the subsequent data on the furnaces' behavior with the prototype heads.

Selection of Furnaces

A variety of information was considered in attempting to select just two furnaces as being reasonably representative of the breadth of oil furnace and boiler types, manufacturers, sizes, ages, burners, etc., existing in the United States. It was decided that both units should be of the warm-air type. For the most part, the combustors in hydronic boilers are similar (refractory-lined, side-fired) to those in a majority of warm-air furnaces, so the additional cost of a hydronic unit and greater complexity of installing and instrumenting it in the laboratory were unwarranted for this investigation.

The most prevalent basic combustor design, used in perhaps 75 to 80% of existing units, is a refractory-lined cylindrical steel shell with the side-fired burner orientation. The refractories used in current construction are primarily light-weight monolithic refractory-fiber structures, but those in existing units are still predominantly hard castables and firebrick. To match this "most common" firebox design, the Williamson Model 1167-15 with a hard cast-refractory, side-fired combustion chamber was selected. This furnace also has a relatively common type of heat exchanger: a central, cylindrical, unlined-steel extension above the firebox from which the gases flow out one side to the inside of an annular, double-walled, welded-steel heat exchanger. Its burner, a Beckett Model AF with a flame-retention head, is not so common, however, since about 80% of existing burners are of non-flame-retention types.

The second most prevalent combustor design in warm air furnaces is that manufactured by Duquesne and supplied to many name-brand furnace manufacturers. On the order of 15% of existing warm-air oil furnaces have these combustors, which are characterized by two concentric horizontally disposed uninsulated metal chambers. A pseudo-tunnel-fired burner orientation is used; it is fired along the axis of the inner cylindrical chamber. Combustion products are typically discharged through a narrow slot along the length of the top of the inner chamber into the second (outer) chamber, which is integral with the welded steel heat exchanger. The inner chamber usually is made of a high-temperature stainless steel. It is cooled, partially by convection to a small fraction of the combustion air which is bypassed around its outside, but principally by radiation to the surrounding outer chamber. The warm-air furnace coolant passes over the outside of the outer chamber before being admitted to the main heat exchanger. Thus, the combustion chamber accomplishes an early part of the furnace's heat exchange. The second furnace selected, a Carrier Model 53HV-156, has this type of firebox. It is fitted with a Wayne burner having a conventional burner head.

Taken together, the combustors and heat exchangers in these two furnaces are believed to be characteristic of well over half of the warm-air furnace configurations found in U.S. residences. There are, of course, a number of furnace and installation variables which make it improbable that a sample size of two could truly represent a population that exceeds 13 million units. For comparable furnace designs, the burners and their firing levels, discussed in later paragraphs, are undoubtedly more influential variables than are combustors and heat exchangers. Concerning furnace components, effects of aging (e.g., deterioration of refractories, scaling of heat transfer surfaces and development of air leaks) are probably the most distinguishing differences between units in the field and those tested in this study. Conclusions based upon comparison of experimental results obtained before and after retrofitting the burner heads should not be negated by such differences.

Any given model of furnace will exhibit some variations in performance and pollutant emissions as a result of installation and operational differences among many residences. Probably the most influential variable is the firebox pressure or draft. For operational safety, nearly all residential furnaces and boilers are designed to operate with a slight negative pressure over the fire, typically in the range of 5 to 10 Pa (0.02 to 0.04 inch of water). Excessive firebox draft may degrade combustion efficiency and increase emission levels of carbonaceous air pollutants, presumably by drawing the flame out of the firebox and into the heat exchanger where combustion reactions are quenched a bit too soon. Draft dampers and barometric control devices should be adjusted to minimize the effects of installation differences, but they can not be eliminated entirely. Transient effects associated with starting and stopping a unit, prevailing and gusting wind conditions, and rapidly changing barometric pressure are almost impossible to normalize among different installations. Ultimately, these differences are anticipated to some degree when an experienced serviceman tunes an oil furnace's burner (Ref. 12). Presumably, then, special conditions applicable to a burner in an existing installation would also be applicable if it were retrofitted with an optimum low-emission head. This reasoning is probably valid for a majority of burners, but, because some burners are less sensitive than others to variations in firebox draft, it is not universally applicable.

A burner which is retrofitted with an optimum head becomes a "conventional" type of high-pressure atomizing oil burner. There are two other principal types of high-pressure atomizing burners in use in the United States: (1) shell-head burners, and (2) flame-retention-head burners. These types have been developed more recently than the conventional head burners and, although some manufacturers' designs do not perform significantly better than do many conventional burners, they have the general reputation of achieving higher efficiencies and of being more forgiving of operational peculiarities than do conventional burners. Retrofitting relatively new burners, particularly shell-head or flame-retention-head designs, may not be justifiable. As discussed in Section IV, to the extent that new burners approach the limits of the current technology, it will be difficult to demonstrate advantages to retrofitting that equipment with the optimum head. Thus, comparison of the results which follow represents a severe test of the optimum head technology.

Acquisition of Furnaces - The selected furnaces were ordered directly from the manufacturers. In each instance, questions were voiced by the manufacturers concerning suitability of the ordered furnaces for use in southern California, availability of the ordered and alternate equipment, etc. Thereupon, the intended use for the furnaces was divulged to and discussed with the suppliers. The engineering and distribution personnel of both Williamson and Carrier were very interested, cooperative, and helpful in ensuring timely delivery of precisely the units selected. In fact, the Williamson Company cooperated to the extent of supplying their unit cost-free to Rocketdyne for this investigation, requesting only that they be informed of the published results.

The firing rates for about 2/3 of the existing oil furnaces fall in the range of $0.79 < \dot{w}_{oil} \leq 1.42$ ml/s ($0.75 < \dot{w}_{oil} \leq 1.35$ gph). The manufacturers' nominal firing rates for both selected furnaces fall within this range: 0.85 to 1.00 gph for the Williamson and 1.10 gph for the Carrier. Because the prototype optimum heads were designed for a nominal 1.05 ml/s (1.00 gph) firing rate, both furnaces were tested at that nominal firing-level condition.

Test Facility

The test facility configuration used in the research combustor testing effort primarily provided for measurement of flue gas pollutant emissions, with estimation of thermal performance characteristics more of a secondary nature. However, the furnace testing objectives required quantitative evaluation of the furnaces' thermal efficiencies and some indications of projected long-term degradation of burner heads. Therefore, in addition to the flue gas sampling instrumentation system, which is described in Appendix A, the test facility was configured for more complete measurement of gas flows and their properties. Figure 19 is a schematic of the expanded furnace evaluation system. Shown are the installation of necessary gas and air-flow ducting and a variety

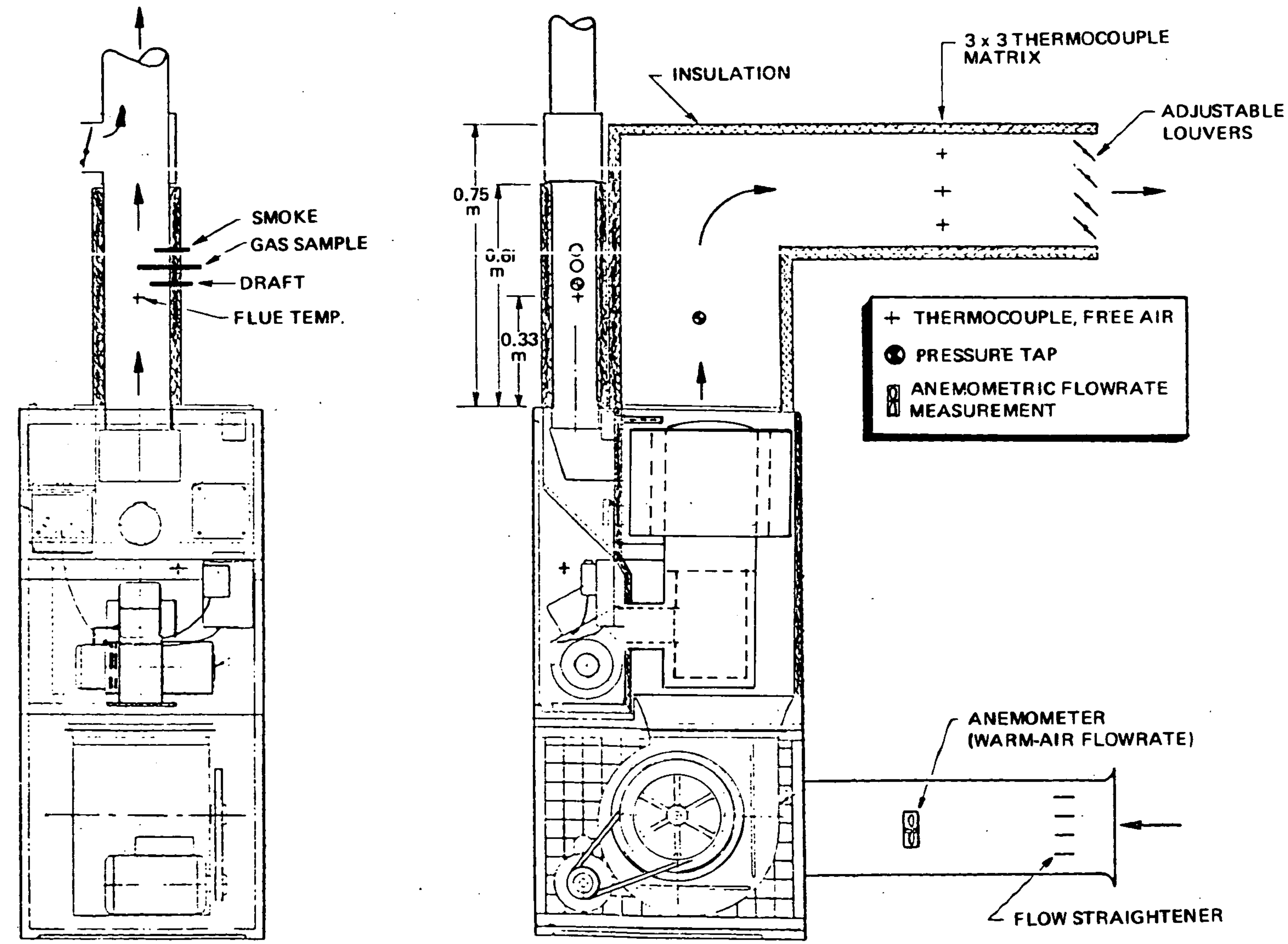


Figure 19. Schematic of the furnace performance evaluation system

of instrumentation. Basic thermal performance measurement techniques conformed with requirements of ANSI Z91.1-1972 (Ref. 7). Other instrumentation were added to provide enlarged understanding of furnace behavior and data for calculating cycle-averaged thermal efficiency.

The furnace flue thermal losses were determined by making measurements to support flue gas heat balances. The combustion gas mass flowrate was backcalculated from measured fuel flowrate and stoichiometric ratio (as determined from flue gas composition measurements). The flue gas exhaust temperature was measured in an insulated flue pipe with a thermocouple located 0.46 m (18 inch) above the centerline of the heat exchanger flue exit, as per ANSI Z91.1-1972. Flue draft, gas composition, and smoke measurements were taken at successive 0.0317 m (1.25 inch) increments downstream of the thermocouple, respectively.

Steady-state thermal efficiency can be calculated, according to the ANSI Z91.1-1972 recommended procedure, from the steady-state flue gas temperature and CO_2 concentration (see Fig. 2). During cyclical operation in which steady state was not reached, values for those parameters just prior to burner cutoff were used in the same manner to get approximations of steady-state efficiencies. Burner firing times of 10 minutes gave such pseudo-steady-state efficiencies which were indistinguishable from those derived from steady-state measurements; those calculated from 4-minute burner firing time data were approximately 1/2 to 1% higher than the steady-state efficiencies.

Determination of furnace thermal performance during cyclical operation is more difficult than during steady-state operation. To avoid the complications of measuring or estimating transient draft air and furnace cabinet heat losses, the method* used to calculate cycle-averaged efficiency was to measure the net heat gained by the warm-air furnace

*Difficulties experienced in actual application of this method led to large uncertainties in the data, as discussed in the next subsection.

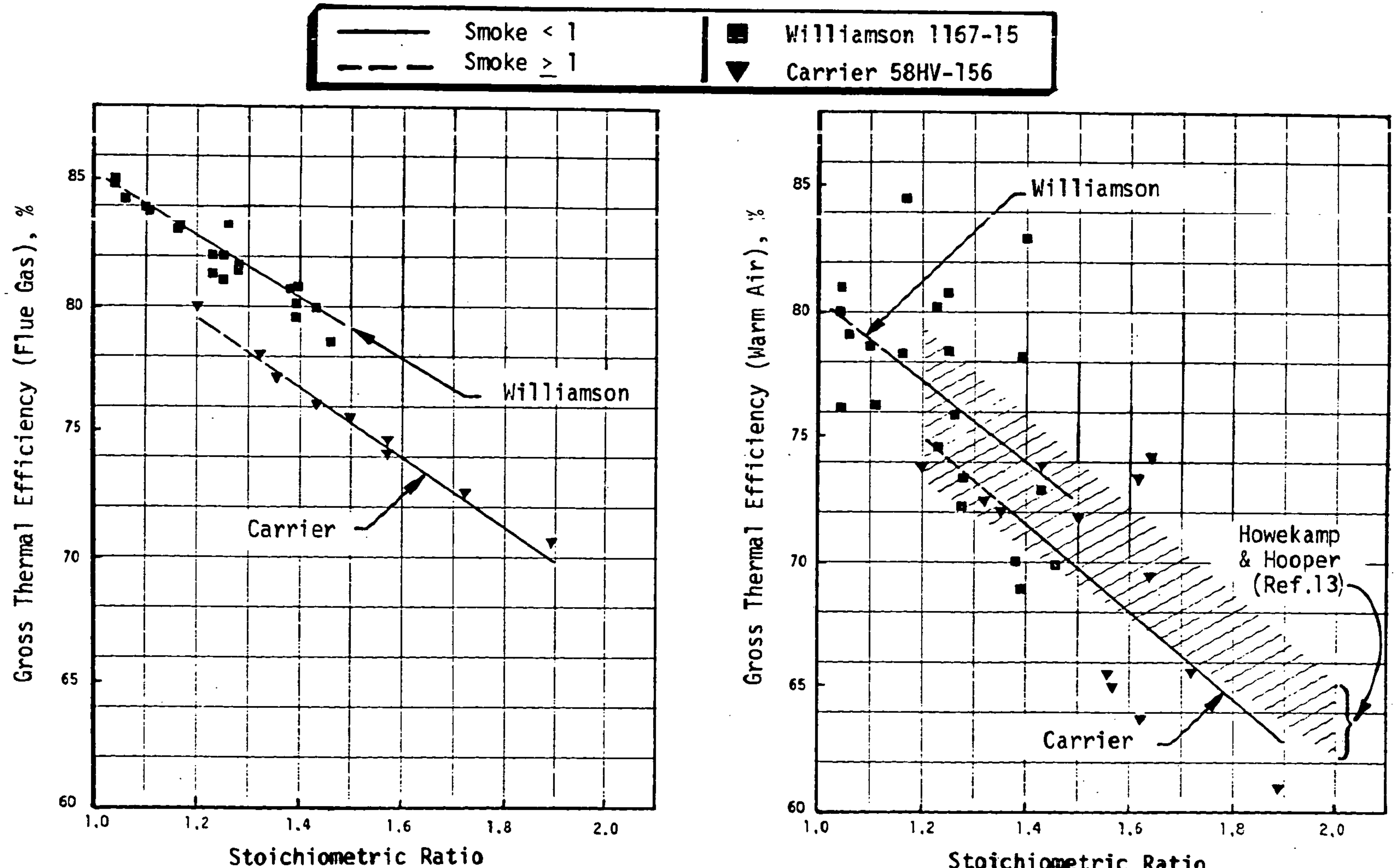
coolant and divide it by the gross heat input with the fuel burned in a cycle. This method required measurements of oil flowrate, oil and combustion air temperatures and, for the warm-air furnace coolant, flowrate and temperatures at the inlet and outlet. The inlet warm air was drawn into the furnace from the ambient outdoor atmosphere through a 0.46 m (18 inch) square duct with an inlet flair and internal egg-crate flow straightener. The volumetric air flow was measured with a cumulative readout, gas flow anemometer (±1%), i.e., it integrated the total furnace-coolant air flow admitted during each complete cycle. Ambient atmospheric pressure, temperature, and relative humidity were recorded continuously at a meteorological data station located approximately 15 meters from the furnace test stand. Furnace coolant air temperatures were measured with a thermometer at the inlet anemometer location and at the warm-air outlet as an average of nine thermocouples in a rectangular grid array. The outlet ducting was wrapped with 0.025 m (1 inch) thick fiberglass matting for thermal insulation. The outlet back pressure was varied by means of a set of adjustable outlet louvers to simulate various installed ducting loads.

Stock Furnace Characterizations

The Williamson Model 1165-15 and Carrier Model 58HV-156 furnaces were tested in their stock configurations to characterize their thermal efficiency and emissions performance. Nearly all the firings were cyclical tests, with the burner fired for 1/3 of the cycle time. Cycle times of 12 minutes were used, primarily. Cycle-averaged data were obtained by: (1) firing the furnace for approximately 15 minutes to warm it up, (2) initiating cyclical operation, (3) waiting for the third cycle before commencing measurements, (4) collecting detailed data during four successive cycles, and (5) taking appropriate arithmetic averages of the resultant data. Data from these tests are tabulated in Appendix C.

Efficiency - The measured gross thermal efficiencies for the two stock furnaces are plotted in Fig. 20. Those in Fig. 20(a) are pseudo-steady-state efficiencies derived from Fig. 2 as functions of flue gas temperature and CO_2 concentration just before burner cutoff, and so are indicated as being "flue gas" derived. Those in Fig. 20(b) are cycle-averaged efficiencies derived from calculation of the cycle-averaged heat transferred to the warm-air furnace coolant stream, and so are indicated as being "warm-air" derived. Also shown in Fig. 20(b) is a shaded band representing the range of efficiencies reported in Ref. 13 for tests of six different burner heads in an earlier model Williamson furnace. (Net efficiencies reported in Ref. 13 were multiplied by the ratio of the lower to the higher heating values of No. 2 fuel oil for the purposes of this graph.)

The correlating lines drawn through the data in Fig. 20 are all least-squares fits. It is evident that there is considerably greater scatter among the "warm air" data than among the "flue gas" data. This is undoubtedly due, in part, to greater uncertainties and experimental errors in measuring the air flowrate and its rather modest (<50 C) temperature rise. The air stream is more voluminous than the flue gas stream, and there are more opportunities for its flow to become striated in both the furnace inlet and outlet measuring sections. Nonetheless, the magnitude of the scatter seen in Fig. 20(b) is surprisingly large; e.g., the five Williamson data points at about 1.4 stoichiometric ratio range from 69 to 83%. It was also found that there were unexplained shifts of 10% or more in the indicated efficiency when one furnace was removed from the facility and another installed. A portion of this (up to ~4%) was found to be related to thermally striated flow in the exit metering section; it was improved by placing a flat baffle upstream of the thermocouple matrix, but the best position and orientation of the baffle had to be determined experimentally for each furnace's tests.



(a) Pseudo-Steady-State Efficiency

(b) Cycle-Averaged Efficiency

Figure 20. Gross thermal efficiency characteristics of furnaces tested in their stock configurations

Some of the data scatter was believed to result from testing the furnaces in an outdoor facility. Both the combustion air and furnace coolant air supply temperature and humidity varied more than if the unit has been tested indoors. The furnace cabinet was exposed to outdoor air currents and winds as well as variations in solar isolation. Resultant variations in heat losses through the cabinet undoubtedly contributed to the data scatter, although they were estimated to be small.

Another factor which contributed to scatter in the cycle-averaged efficiency was variation in the cycle timing. The burner-on, burner-off, and warm-air blower-on times were controlled by a mechanical timer. The firing interval was observed to vary by about ±5%. Cut off of the warm-air blower was effected by a thermoswitch in the warm-air discharge, which was nominally set at 46 C (115 F). For a constant ambient temperature, the burner-off blower-off time was fairly consistent (±10%). However, as the ambient temperature of the outdoor facility went up, that time interval also increased. To keep variations of that interval in reasonable bounds, the cutoff temperature was manually adjusted to higher temperatures--up to 55 C (131 F)--as ambient temperatures rose. Even so, the duration of blower operation following burner cutoff varied between approximately two and three minutes. While this may have been a substantial contributor to the scatter seen in Fig. 20(b), it had little effect on the variation of efficiency from cycle-to-cycle, which also exhibited quite large scatters.

The remainder of the inconsistencies were presumed to arise from metering the air flow. The anemometer was recalibrated repeatedly, replaced once, and its application in the inlet duct was checked on occasion by probing the inlet section with a small hot-wire anemometer. It became apparent that some uncontrolled phenomenon was interfering with the air-metering measurement. The scatter suggests that the degree of influence varied from cycle to cycle; perhaps it was caused by different vortex patterns at the inlet to the warm air blower propagating upstream and altering the flow pattern in the inlet duct.

In short, a substantial amount of effort was expended in attaining the data in Fig. 20(b) and no clear resolution of the apparent instrumental problems was in sight. Comparison of Fig. 20(a) and (b) shows that, for both furnaces, the mean cycle-averaged efficiencies are about 5% lower than the pseudo-steady-state efficiencies at low stoichiometric ratios and about 5-1/2% lower at high stoichiometric ratios. Differences of these magnitudes apparently are characteristic of the furnaces at the burner-firing-time/cycle-time ratio of 1/3, so it was decided to use only the pseudo-steady-state efficiency as the comparison basis for subsequent testing.

<u>Emissions</u> - Cycle-averaged flue gas NO emissions from the stock Williamson and Carrier furnaces are shown as functions of operating stoichiometric ratio in Fig. 21. The NO emissions from the Williamson furnace were comparable with the average values from existing furnaces. Those from the Carrier were about 25% higher than had been expected from its radiation-cooled wall, modified tunnel-fired combustor.

Both Fig. 20 and 21 also show that the stock Williamson furnace could be operated with as little as 10% excess air before its smoke emissions exceeded Bacharach No. 1. Since its burner can already be tuned for normal operation at excess air levels in the target range for burners retrofitted with optimum heads, little or no gain in thermal efficiency should be expected to result from retrofitting the burner head supplied with this furnace. The Carrier furnace, on the other hand, produced greater than No. 1 smoke at excess air settings below about 35%. If retrofitting its burner with an optimum head were to allow tuning for 15% excess air, a modest 3% increase in thermal efficiency would be expected.

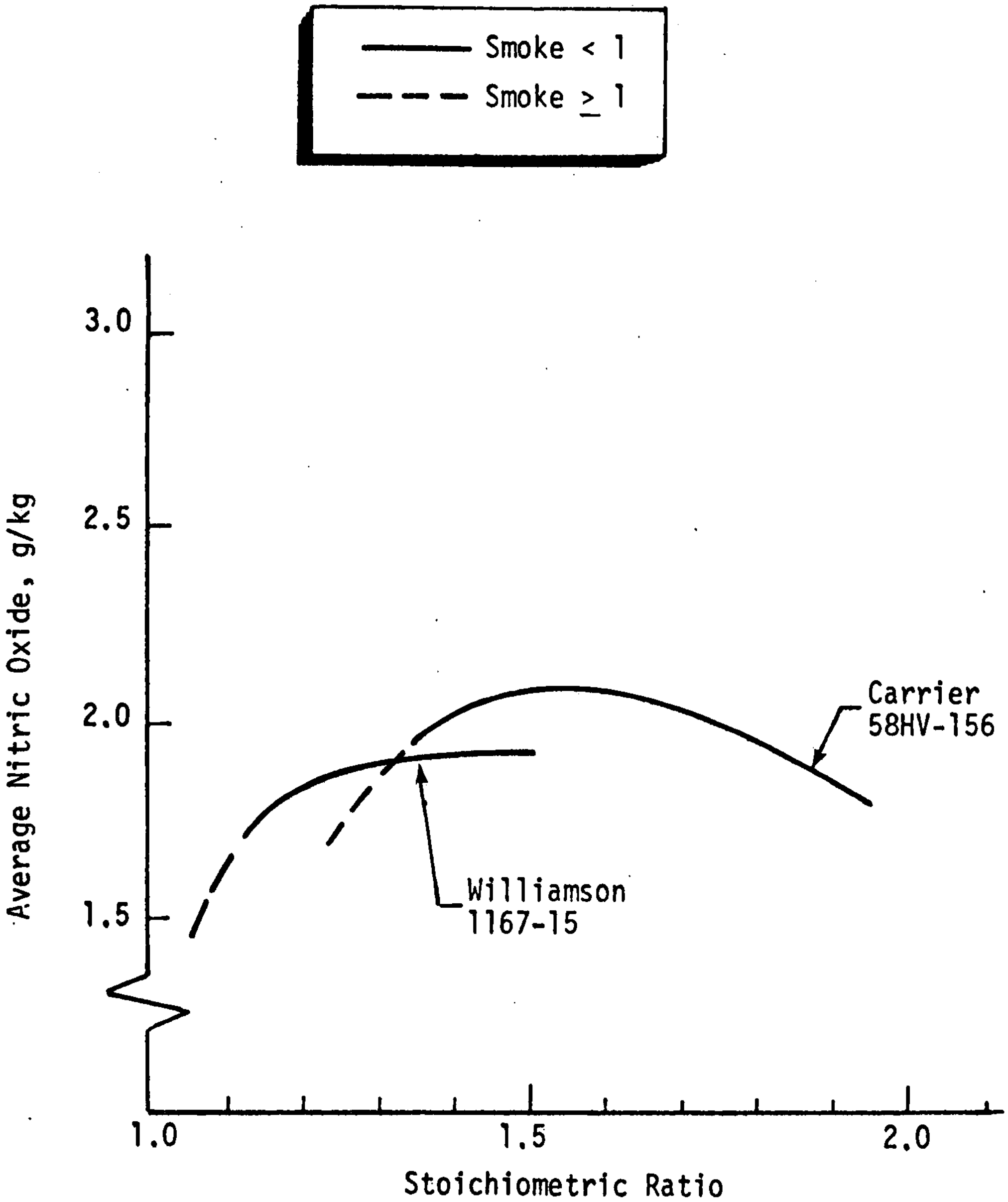


Figure 21. Cycle-averaged flue gas nitric oxide concentrations for furnaces in their stock configurations

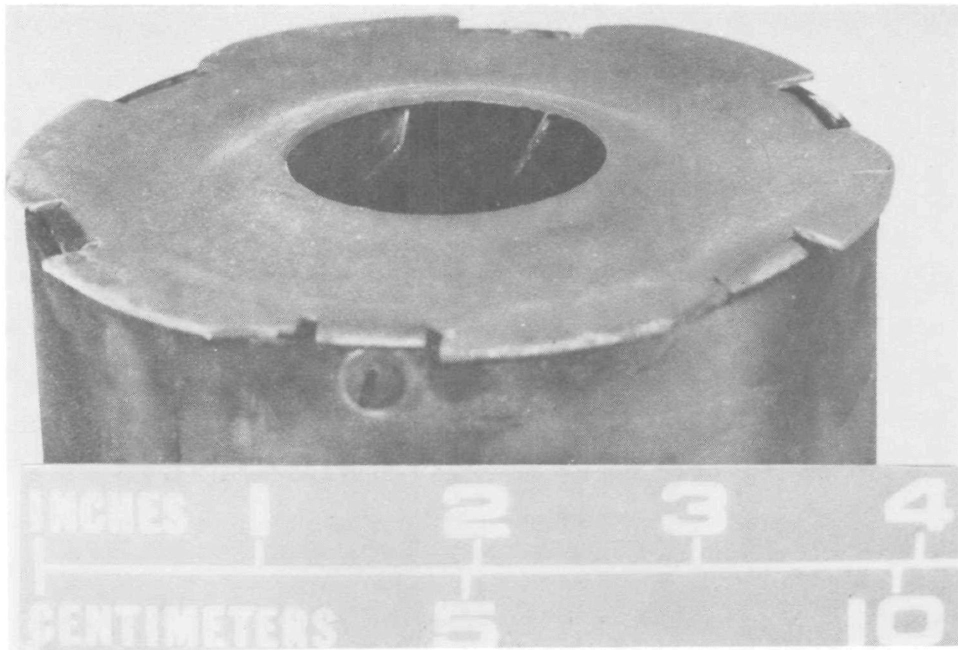
Performance of Furnaces With Prototype Optimum Heads

After the performance of the Williamson furnace in its stock configuration had been characterized, its burner head was replaced by the reinforced prototype optimum head of the initial design (Fig. 16) and tests were made to provide a comparison basis for subsequent tests with the second prototype head design. The data obtained are tabulated in Appendix C, Runs 100-A, B, and C*.

Upon receipt of the Type 310 stainless-steel prototype heads of the revised design (Fig. 11), they were installed on the furnaces' burners, with the 18-gage head in the Williamson furnace and the 21-gage head in the Carrier unit. The furnaces with the prototype optimum heads were checked out, and then each was tested for several days to measure its efficiency and emissions performance. Data acquired are tabulated in Appendix C, Tables C-1 through C-3 for the Williamson and Tables C-4 through C-6 for the Carrier furnaces, respectively. Thereafter, moderately longer-term simulated service testing (4 minutes on/8 minutes off cycles continuously for about 3 weeks) was undertaken, and total test times of approximately 500 hours were accumulated with each head. The modified design Type 310 stainless steel heads were in excellent condition when their testing was completed. Neither the 18 gage nor the 21 gage material showed any signs of either metal scaling or distortion, Fig. 22.**

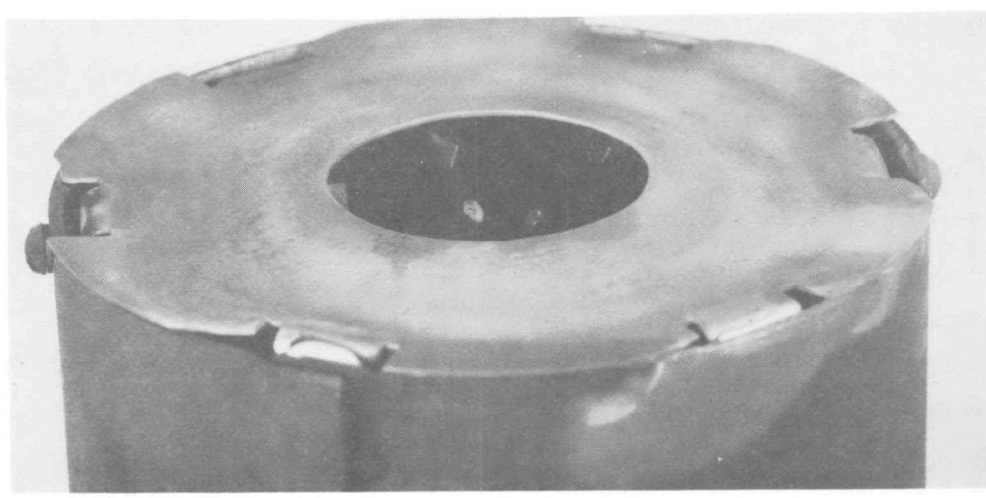
*Comparison of performance and emissions data from these runs with those from Runs 101 through 117 and 135 through 137, made with the revised design prototype head, shows that the two heads behaved essentially the same in the Williamson furnace.

**The revised design prototype commercial head made from 18-gage Type 304 stainless steel was subsequently tested for a total of 20 hours of 4 minutes on/8 minutes off cyclical operation in the Carrier furnace. No indications of any scaling or warping problems were evident after that exposure time.



50P37-1/12/76-S1B

(a) 18 gage (0.00127 m), Type 310 Stainless Steel Head



50P37-1/12/76-S1A

(b) 21 gage (0.00087 m), Type 310 Stainless Steel Head

Figure 22. Photographs of the modified sheet-metal prototype commercial optimum oil burner heads after 500 hours of cyclic service

Efficiencies - Pseudo-steady-state efficiencies, measured for both furnaces retrofitted with prototype optimum heads, are plotted in Fig. 23 together with those obtained with their stock burners. Detailed data are listed in Tables C-2 and C-3 for the Williamson furnace and Table C-5 and C-6 for the Carrier unit.

The efficiency performance of the Carrier furnace with the prototype optimum head was essentially identical to that with its stock burner head. The limit of smoke-free operation occurred at about 35% excess air with both heads, indicating that neither an efficiency gain nor loss would be experienced by retrofitting this furnace with an optimum head.

The efficiency performance curve for the Williamson furnace with the prototype optimum head was about 1% below that for the stock furnace. The drop in efficiency level was attended by an increase of about 17 C/30 F in average of about 17 C/30 F in average. Presumably, this resulted from the burner having been converted from a flame retention burner to a conventional type of burner when its head was replaced by the optimum head. Moreover, the retrofit prototype optimum head produced greater than No. 1 smoke when operated with less than about 30% excess air. Combined, these two effects would force the efficiency of this furnace with a tuned retrofit optimum head to be about 3% lower than that with a tuned stock head.

Emissions - Cycle-averaged NO emissions from the Williamson furnace with the prototype optimum and stock burner heads (Table C-1) are plotted in Fig. 24 and similar results for the Carrier furnace (Table C-4) are shown in Fig. 25. Both furnaces with the optimum heads produced about 1.5 to 1.6 g NO/kg fuel burned, which is substantially below the approximately 2 g NO/kg fuel level experienced in the research combustor experiments (Fig. 17). As a result, it is seen that the prototype optimum heads reduced NO emissions by 15 to 20% from the Williamson furnace and by 20 to 25% from the Carrier furnace.

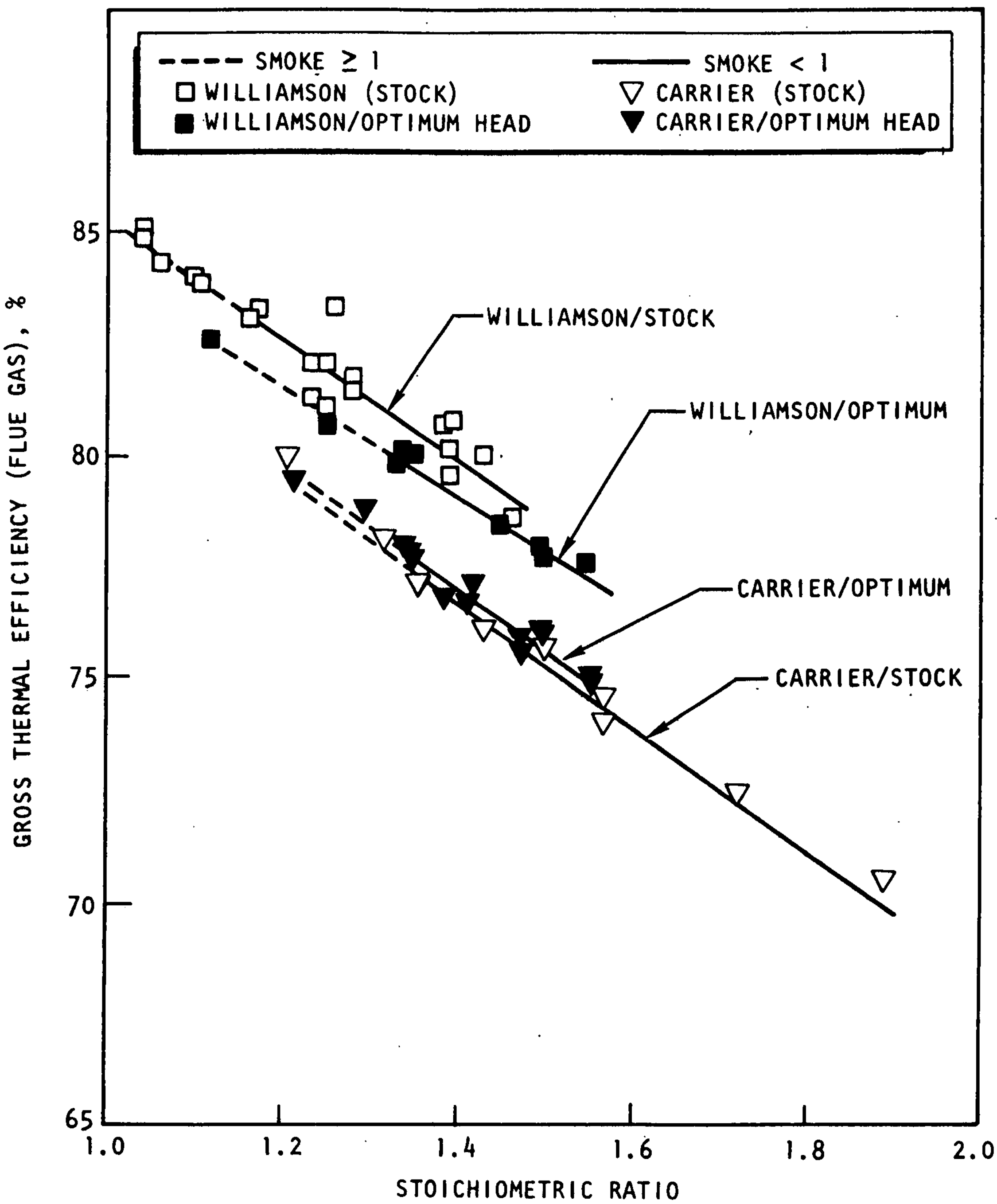


Figure 23. Comparisons of pseudo-steady-state thermal efficiencies from the Williamson 1167-15 and the Carrier 58HV-156 furnaces using their stock burner heads and prototype commercial optimum burner heads

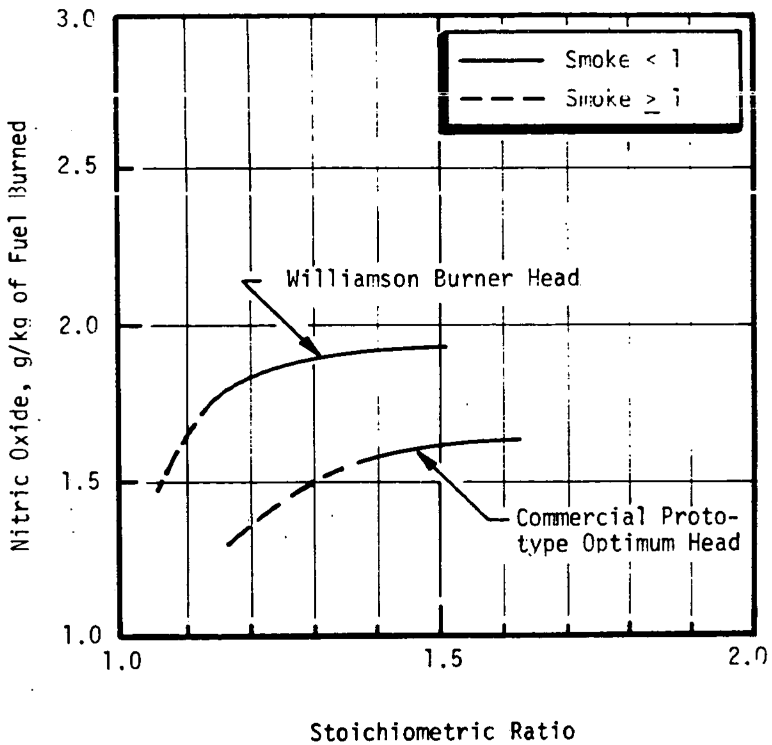


Figure 24. Effect of the commercial prototype optimum head upon cycle-averaged nitric oxide emissions from the Williamson 1167-15 furnace

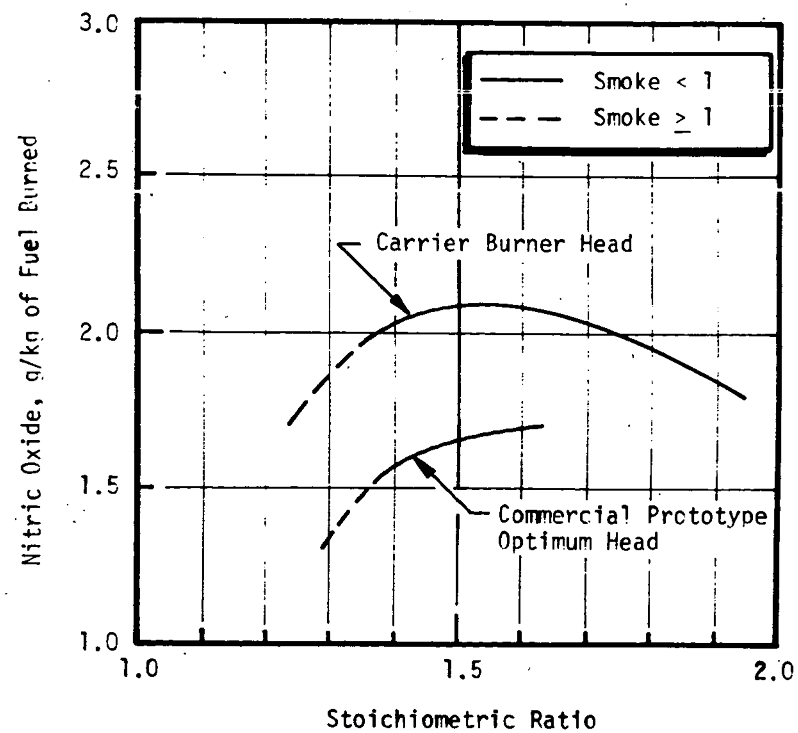


Figure 25. Effect of the commercial prototype optimum head upon cycle-averaged nitric oxide emissions from the Carrier 58HV-156 furnace

Cycle-averaged emissions of CO and UHC are also listed in Tables C-1 and C-4 for the Williamson and Carrier furnaces, respectively. Comparison of the data for the stock units with those from the corresponding optimum head retrofitted units reveals that these emissions were increased somewhat by retrofitting the Williamson unit while they were essentially unchanged by retrofitting the Carrier unit. Broader comparison with the Ref. 2 field survey data, summarized in Fig. 7, shows that the stock Williamson emissions of these pollutants were exceptionally low, while those from both retrofitted furnaces at 35% excess air were lower than the tuned condition averages of those surveyed in the field.

DISCUSSION

The foregoing results from tests of the prototype optimum head as a retrofit device for existing residential oil furnaces are discussed in this section in terms of potential impact on thermal efficiency and air pollutant emissions of existing installed space heating units.

Efficiency

A convenient method of comparing furnace efficiencies is to superimpose general furnace population behavior and individual furnace operating lines on the efficiency decrement curves of Fig. 2. This is done in Fig. 26. The general behavior of a large percentage of existing oil-fired residential heating units (estimated to be 80%, from data in Ref. 2 and 10) is indicated as a shaded zone. The average of all existing units is estimated to be in a smaller crosshatched zone imbedded in the shaded zone. Boundaries of the crosshatched zone conform to the estimated average operating conditions for all existing residential furnaces, discussed in Section IV. Old oil-fueled equipment, including units converted from coal, tend to operate toward the upper and right-hand regions of the shaded zone, while newer equipment tends to perform toward the lower and left-hand portions of that zone. Obviously, a great many units operate outside of the shaded zone, and they are distributed around it on all sides.

Operating curves for the test furnaces are also shown on Fig. 26. (Because of the different plotting basis, the efficiencies versus stoichiometric ratio indicated by these curves differ slightly from the correlating lines in Fig. 23.) As might be expected with new furnaces conforming to contemporary design practices, the burners in both stock furnaces could be tuned for normal operation (e.g., the point corresponding to a No. 1 cycle-averaged smoke reading) at significantly lower excess air levels than can most existing residential oil furnaces.

The Williamson unit was especially impressive in that regard, being capable of operating satisfactorily with as little as 15% excess air (13% CO_2). This capability is undoubtedly attributable to its flame retention head burner and a good match between the burner and firebox. Further, the net temperatures of the stock Williamson's flue gases was on the low-side of the shaded band in Fig. 26, so that the unit could achieve an estimated steady-state efficiency of nearly 84%. Obviously, the performance capability of this stock furnace left no margin for efficiency improvement via retrofitting with an optimum head.

Indeed, retrofitting the Williamson furnace with the prototype optimum head resulted in both a significantly higher excess air requirement and somewhat higher flue gas temperatures, so that achievable steady-state efficiency was lowered by about 3 percentage points. Approximately 2/3 of that decrement was caused by the higher excess air requirement and the other 1/3 by the increased exhaust temperature. Both of those components of the total effect upon efficiency were undoubtedly caused by replacing an effective, well-designed flame retention head matched to the combustor with a conventional type head of universal application design.

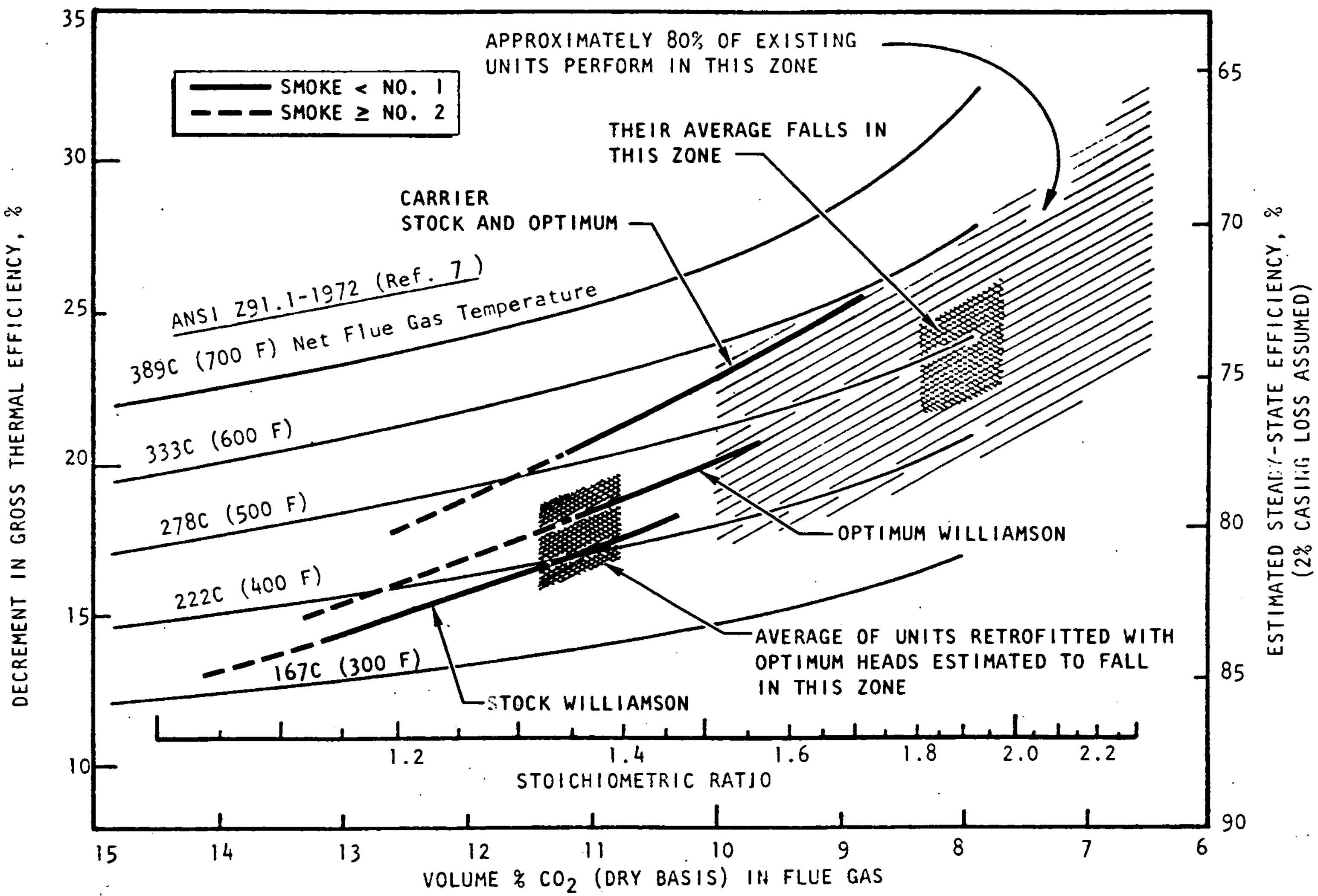


Figure 26. Steady-state thermal efficiency relationships for residential furnaces

It is informative to review the data reported in Ref, 3, wherein six different burner heads were tested on a single burner in another model of Williamson furnace. The stock burner head was of the conventional type, as were four of the other heads. Their measured cycle-averaged efficiencies ranged from 70.5 to 76.6% and averaged 73.9%. Concurrently, their average operating stoichiometric ratio, set by tuning for No. 1 tenth-minute smoke, was 1.6 (±0.20). By contrast, the sixth head (a flame-retention type) could be tuned to operate at 1.19 stoichiometric ratio, where it achieved 83.0% cycle-averaged efficiency. That retention head appears to have capabilities comparable with those of the stock flame-retention head Williamson burner tested in the current program. The 9% decrement between it and the average of five conventional heads was three times as large as that between the stock Williamson burner and the retrofitted prototype optimum head. Two conclusions may be drawn from this. First, a decrease in performance definitely should be expected if one of the better flame-retention heads is replaced by a conventional burner head. Second, the magnitude of that efficiency decrease may be substantially smaller if such a retention head is replaced by an optimum low-emission head than if it were replaced by any of those five conventional heads. By inference, a corollary to the latter conclusion is that the efficiency of a furnace which now has a conventional head might be increased by retrofitting it with an optimum head and a simultaneous reduction in NO_x emissions achieved.

How large such efficiency gains from retrofitting conventional burner heads might be is the next question to address. There are few quantitative data to consider. With the Carrier furnace, essentially identical performance was observed with the stock and optimized low-emission heads (Fig. 26). Steady-state efficiency with each head was approximately 77% when the burner was tuned to operate at 1.35 stoichiometric ratio. That operating condition coincides with the achievable stoichiometric ratio for the optimum low-emission head in the Williamson furnace. In the earlier development of the optimum head, as a part of an optimum burner, it could be tuned to stoichiometric ratios of 1.15 or lower. What can be achieved in this regard also depends upon the designs of the furnace's firebox, heat exchanger, and the transition between them, as well as upon the burner firing level and firebox draft condition. Thus, it should be expected that the optimum head could be tuned to a range of excess air levels corresponding to variations among many residential installations. In light of its known tunability to 10 to 15% excess air in some cases and 35% in others, a conservative estimate is that the 35% excess air level represents a reasonable average retrofit optimum low-emission head operating condition.

A second crosshatched zone is plotted on Fig. 26 to designate the probable location of the average operating conditions for a large number of retrofitted existing units. This zone represents a projection of the preretrofit crosshatched zone from an average of 90% excess air to an average of 35% excess air by following the slope treads of individual furnace operating lines, rather than following the flue gas isotherms. The steady-state efficiency level of this second crosshatched zone is about 6 percentage points higher than that of the first, existing furnace population zone.

Air Pollutant Emissions

Smoke emission data for the test furnaces, tubulated in Appendix C and indicated in Fig. 23 through 26 as being less than (solid curves) or greater than (dashed curves) No. 1 on the Bacharach scale, are all cycle-averaged values. It is known (Ref. 3) that burners tuned to a No. 1 smoke reading at steady-state according to recommended practice (Ref. 12), typically have cycle-averaged smoke readings between No. 2 and No. 3. Since cycle-averaged rather than steady-state, smoke readings of No. 1 or less were used above to select 35% excess air as an average condition to which furnaces retrofitted with optimum burner heads could be tuned, this choice is conservatively high. Cycle-averaged emission levels for the other carbonaceous air pollutants, from both test furnace tuned to that condition, were less than the tuned average levels reported from the field survey of Ref. 2.

The emission levels of NO from the test furnaces, tuned to 1.35 stoichiometric ratios, were reduced an average of approximately 20% when they were retrofitted with optimum heads. That is only about one-half of the reduction anticipated from the earlier tests of the research optimum head in research combustors (Ref. 5). It is, nonetheless, an appreciable reduction which, together with potential efficiency gains, makes commercialization of the optimum head attractive as a retrofit device.

Potential Applicability

The major point in favor of developing the optimum head for retrofitting existing burners is its potential for increasing thermal efficiency and lowering fuel consumption. There are undoubtedly other existing burner heads, particularly some of the better flame-retention heads that could also be used as efficiency-improving retrofit devices. However, none of them is known to offer the other potential benefit of simultaneously lowering the emissions of oxides of nitrogen. Thus, the optimum head investigated here is singularly unique as a candidate retrofit device for simultaneously reducing fuel consumption and air pollution.

The incentive for a particular homeowner to retrofit his oil heating system's burner with an optimum head will be monetary, i.e., the savings which can be realized because fuel consumption is reduced. An average system, operating at 62% season-averaged efficiency, might burn 1300 gallons of No. 2 oil in a season at a cost of nearly $600. If retrofitting were to increase the unit's efficiency by 5 percentage points, fuel consumption would be reduced by 7% (Fig. 3), corresponding to an annual savings of about $42. Thus, a retrofit head cost and installation expense totalling as much as $38 could be recovered in a single heating season by the "average" homeowner (a 10% "cost of money" charge has been deducted).

The same unit could recover a $38 total installation cost in three heating seasons if the efficiency increase due to retrofitting were as small as 1.6%. A lower installation cost could be justified by even lower efficiency gains. However, it seems unlikely that a homeowner would be satisfied that it was a good investment if the payback were prolonged so that it was obscured by year-to-year climatic variations. It is probably inadvisable to retrofit any burner unless an efficiency increase of 1-1/2 to 2 percentage points or more can be assured. Referring to the furnace operating lines on Fig. 26, a 1-1/2% gain is indicated, on the average, by reducing the excess air level from 45% to 35%. Thus, any burner capable of being tuned to a stoichiometric ratio of 1.45 or lower (≥10-1/4% CO_2) probably should not be retrofitted.

Most existing burners capable of being tuned to stoichiometric ratios ≤1.45 probably have flame retention type burner heads. However, not all flame retention heads can be tuned so low*, so some are candidates for being retrofitted. On the other hand, not all burners with conventional heads should automatically be considered to be retrofit candidates. In particular, as exemplified by the Carrier furnace's stock burner, those used in current construction and in relatively new units may be exempted by their performance capabilities. Retrofitting of any burner less than 5 years old probably should be approached with caution.

Additionally, it is anticipated that it will not be possible to retrofit some residential oil burners because of basic equipment incompatabilities. Low pressure atomizing burners and rotary burners are in this category. Also, some high-pressure atomizing burners will probably exhibit poor flame patterns, noisy combustion, and/or an inability to tune for low smoke, etc., at a low enough stoichiometric ratio to be beneficial. As an example, it was attempted to retrofit the prototype

*The average tuned stoichiometric ratio was 1.42 for 10 flame-retention heads and burners tested in Ref. 3.

optimum head to a Lennox Model 011-050-321-4 oil burner acquired as the stock burner in a Lennox 011-140 warm-air oil furnace. That burner has an unusually short blast tube, a slower (1725 rpm fan, and a flame-retention heed. When tested in the laboratory, combustion in the oil furnace with the retrofitted burner was noisy and excessively smoky. Satisfactory operating conditions could not be found, so the tests were terminated without any data being recorded. From data on burner types in Ref. 2, it may be estimated that as many as 25% of the existing installed residential burners may fall in this category. Combining this with an estimated 5% as high-performing retention head burners and another 10% as high-performing conventional head burners leaves a balance of approximately 60% of existing residential oil burners which might appropriately be retrofitted with optimum low-emission heads.

In summary, it should be beneficial to retrofit 50% or more of existing U.S. residential space heating oil burners with optimum low-emission heads. The principal benefit would be modest increases in steady-state thermal efficiencies, and these should translate directly to equivalent increases in season-averaged efficiencies. If the distribution of actual initial efficiencies among the units retrofitted were identical to the distribution among all existing units, an average of about 5 efficiency points should be gained by retrofitting. There are many existing burners, however, for which retrofitting would not improve efficiency appreciably. If these were identified and omitted from the retrofitting program, then the average initial efficiency for those which are modified would be lower than the overall average, and this would provide a margin for achieving greater than a 5% average efficiency gain.

Because of similarities in combustion chamber designs and burner orientations, the optimum head technology is believed to be equally as applicable to hydronic boilers as to warm-air furnaces.

Unresolved Issues

There are several subject areas related to successful commercialization of optimum low-emission burner heads as retrofit devices which have not been considered in this research program. The first is that retrofitting a residential oil burner with an optimum head makes it into a different burner and this may obviate whatever certification it may have had concerning conformance with national, state, or local building, fire, and safety codes and standards. The magnitude and potential solutions for this problem need to be defined as an early part of any serious commercialization effort.

A number of allusions have been made in preceding subsections to restrictions on the applicablity of the optimum heat technology. It cannot, in fact, be applied indiscriminately to all residential oil burners. A corollary is that heating industry service personnel, i.e., those who would actually effect retrofitting of existing furnaces, must be able to discriminate between those burners which should and should not be modified. They will need to be more sophisticated than the average service man now is in utilizing the adjustment guidelines, such as Ref. 12, in determining whether a sufficient potential for higher efficiency exists to justify changing to the optimum head, and in tuning modified burners for minimum pollutant emissions and best efficiency. A commercial manufacturer of optimum heads would need to assemble and supply to the oil heating service industry a range of background information such as recommended retrofit procedures, guidelines concerning burners built by many manufacturers, and guidance in selecting and using adequate instrumentation. Success of a retrofit program might even depend upon providing formal personal training of service personnel.

Commercialization of optimum burner heads will require serious consideration of a number of logistics problems. They range from determining the minimum number of optimum head designs needed for modifying many manufacturer's burners with various firing rates, and determining the appropriate production rate for each design to establishing distribution and marketing systems. Some of these have been included in the recommendations, although they are not discussed further here.

SECTION VI

REFERENCES

1. Hall, R. E., J. H. Wasser, and E. E. Berkau, "NAPCA Combustion Research Programs to Control Pollutant Emissions From Domestic and Commercial Heating Systems," New and Improved Oil Burner Equipment Workshop, NOFI Tech. Publ. 108 ED, National Oil Fuel Institute, Inc., New York, New York, September 1970, pp 83-93.

2. Barrett, R. E., S. E. Miller, and D. W. Locklin, "Field Investigation of Emissions From Combustion Equipment for Space Heating," EPA R2-73-084a (API Publ. 4180), Environmental Protection Agency, Research Triangle Park, N.C., June 1973.

3. Hall, R. E., J. H. Wasser, and E. E. Berkau, "A Study of Air Pollutant Emissions From Residential Heating Systems," EPA-650/2-74-003, Environmental Protection Agency, Research Triangle Park, N.C., January 1974.

4. Martin, G. B., and E. E. Berkau, "Evaluation of Various Combustion Modification Techniques for Control of Thermal and Fuel-Related Nitrogen Oxide Emissions," presented at the Fourteenth Symposium (International) on Combustion, Pennsylvania State University, August 1972.

5. Dickerson, R. A., and A. S. Okuda, "Design of an Optimum Distillate Oil Burner for Control of Pollutant Emissions," EPA-650/2-74-047, Environmental Protection Agency, Research Triangle Park, N.C., June 1974.

6. Stanford Research Institute, "Patterns of Energy Consumption in the United States," Office of Science and Technology, Executive Office of the President, Washington, D.C., January 1972.

7. "American National Standard Performance Requirements for Oil-Powered Central Furnaces," ANSI Z91.1-1972, American National Standards Institute, Inc., New York, N.Y., June 1972.

8. Hydronics Institute, "Testing and Rating Standard for Cast Iron and Steel Heating Boilers," Hydronics Institute, Berkeley Heights, N.J., March 1975.

9. DeWerth, D. W., and J. F. Dicaprio, "Performance Tests of Residential Hot Water Boilers; Gas vs Oil," Catalog No. H00220, American Gas Association, Cleveland, Ohio, January 1966.

10. Turner, D. W., et al., "Efficiency Factors for Domestic Oil Heating Units," Addendum to the Proceedings of the Conference on Improving Efficiency in HVAC Equipment and Components for Residential and Small Commercial Buildings, Purdue University, Lafayette, Ind., October 1974.

11. Combs, L. P., and A. S. Okuda, "Residential Oil Furnace Optimization-Phase I," EPA-600/2-76-038, Environmental Protection Agency, Research Triangle Park, N.C., February 1976.

12. "Guidelines for Residential Oil Burner Adjustments," EPA-600/2-75-069-a, Environmental Protection Agency, Research Triangle Park, N.C., October 1975.

13. Howekamp, D. P., and M. H. Hooper, "Effects of Combustion Improving Devices on Air Pollutant Emissions From Residential Oil-Fired Furnaces," Paper No. APCA 70-75, Annual Meeting of the Air Pollution Control Association, St. Louis, Mo., June 1970.

APPENDIX A

FLUE GAS COMPOSITIONAL ANALYSIS

The sample flow train used for analyzing flue gas composition is illustrated in Fig. A-1. A 0.006 m (1/4 inch) diameter stainless-steel tubing sample probe was inserted near the combustor or flue pipe centerline, downstream of the heat exchanger. Flue gas aspirated into the sample probe flowed through a line to an air-cooled condensibles trap where particulates and heavy oils were separated out. Next, the gas passed into an ice-cooled, stainless-steel condensibles trap where most of the water and any condensible, low-volatility hydrocarbons were removed. After the condenser, the gas passed into a Pyrex wool-filled glass cylinder which served as a final separator for heavy oils and particulates, and provided a visual indication of the cleanliness of the gas being admitted to the analysis instruments. Table A-1 gives a summary of the gas analysis instruments used. The gas leaving the glass-wool filter was split into three parallel paths. One path led directly to the total hydrocarbon analyzer. A second path led through a Drierite bed where water vapor was removed, then into the series-plumbed CO, CO_2, and O_2 analyzers. The third path passed through a combined Drierite and 3 Å molecular sieve bed for total water removal, then into the nitric oxide analyzer. The gas was pumped through the system by three diaphragm pumps located downstream of the nitric oxide analyzer, total hydrocarbon analyzer, and the series of CO, CO_2, and O_2 analyzers.

When the analytical system shown in Fig. A-1 is used to analyze gases which may have been quenched before combustion was completed, there are two factors that must be considered in reducing the data: (1) only burned or partly pyrolyzed fuel is included in the analysis, since minute quantities of liquid or vapor fuel may be removed by the cold trap, and (2) water formed from hydrogen and oxygen during the combustion process is also removed from the analyzed sample by the cold trap.

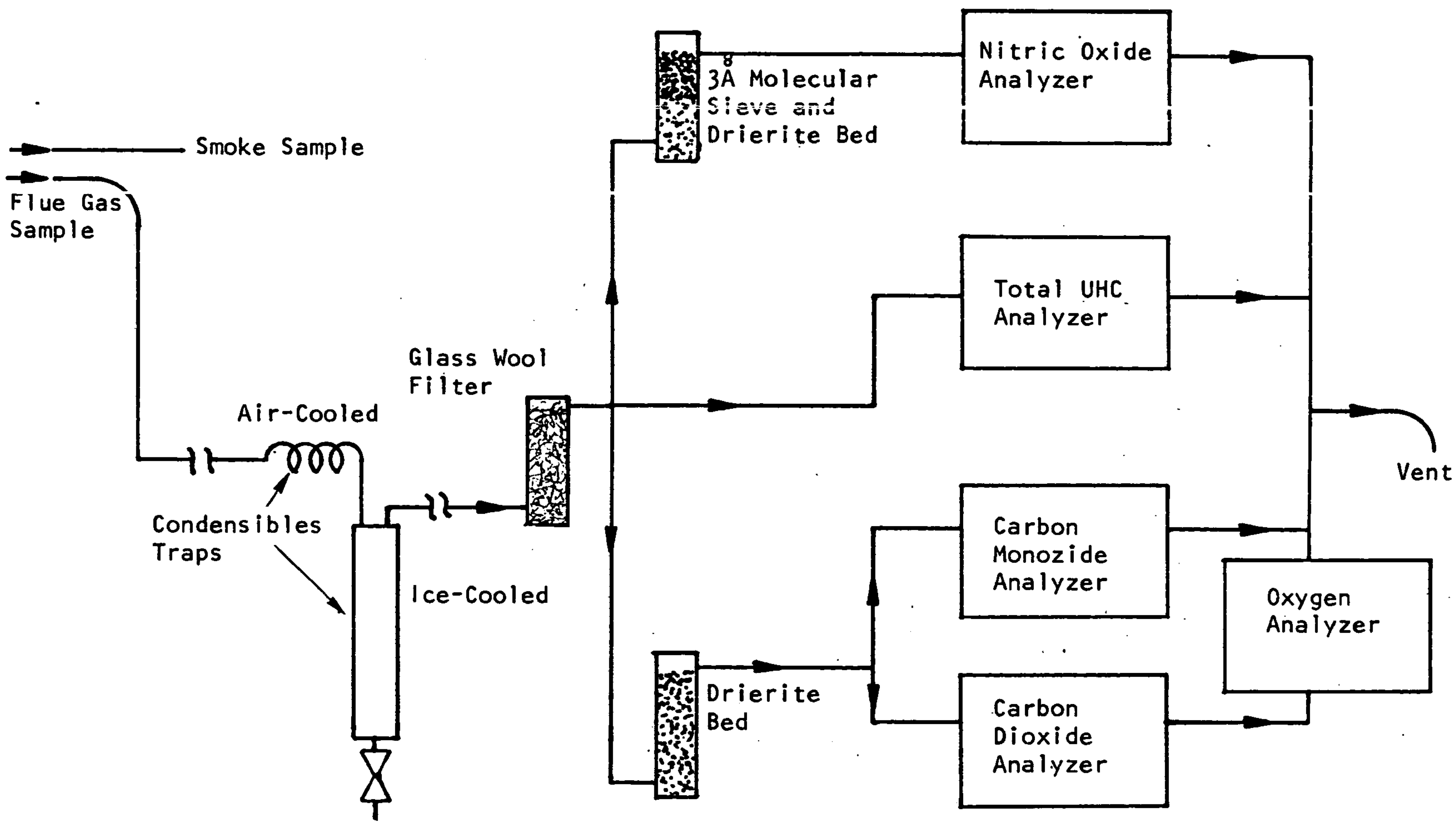


Figure A-1. Analytical system for fuel oil burner emissions analysis

Table A-1. EXHAUST ANALYSIS INSTRUMENTS

	CO	CO_2	NO	Total HC	Oxygen	Smoke
Type	MSA Nondispersive IR LIRA Model 300	MSA Nondispersive IR LIRA Model 300	MSA Nondispersive IR LIRA Model 200	MSA H_2 flame ionization detector	Beckman polarographic	Bacharach (manual)
Range	0 to 1500 ppm (mole)	0 to 20 mole %	0 to 500 ppm (mole)	0.2 to 800 ppm total HC by volume as CH_4	0 to 100%	0 to 9
Sensitivity	30 ppm minimum detectable	0.25% minimum detectable	10 ppm minimum detectable	10 ppm minimum detectable	~0.1%	1
Calibration	1000 ppm CO in N_2 standard gas	14% CO_2 in N_2 standard gas	0.82% C_2H_4 in N_2 used as simulant for 410-ppm NO standard	3% CH_4 in helium used as a standard	Air - 21% N_2 = 0%	Ten spots of monotonically varying darkness

Values calculated from the measured flue gas compositional data included: the overall stoichiometric ratio, the weight of nitric oxide per unit weight of burned fuel, and the weight of carbon monoxide per unit weight of burned fuel. The method of calculation to obtain these values is described below.

The calculations were based on air having the following nominal composition:

Component	Mole %	Wt %
N_2	78.08	75.63
O_2	20.95	23.19
Noble gases (Ar, He and Ne)	0.94	1.13
CO_2	0.03	0.05
	100.00	100.00

The composition of the fuel was assumed to be characterized by the formula CH_x where, for the No. 2 fuel oil burned in this program, x = 1.814. The following symbols were used in the calculations:

AIR = moles of air to produce 100 moles of dry flue gas
FUEL = moles of fuel to produce 100 moles of dry flue gas
CO = moles of carbon monoxide in 100 moles of dry flue gas
CO_2 = moles of carbon dioxide in 100 moles of dry flue gas
NO = moles of nitric oxide in 100 moles of dry flue gas
O_2 = moles of oxygen in 100 moles of dry flue gas
HC = moles of hydrocarbon, as CH_4, in 100 moles of dry flue gas

The values of CO, CO_2, NO, O_2, and HC were obtained directly from the analysis instruments. In the following, it is assumed that all hydrogen is oxidized to water and condensed out of the system at the cold trap, prior to analysis.

An oxygen balance yields:

$$0.2095 \text{ AIR} = CO_2 - 0.0003 \text{ AIR} + 0.5 \text{ CO} + 0.25 \text{ x } (CO_2 + CO - 0.0003 \text{ AIR}) + 0.5 \text{ NO} + O_2 \quad \text{(A-1)}$$

The left hand side of the above equation represents the total free oxygen contributed by the air. The first two items on the right side represent moles of oxygen tied up in CO_2, less the amount of CO_2 originally present in the air. The third term represents moles of oxygen tied up as carbon monoxide. The fourth term represents oxygen consumed to oxidize hydrogen, yielding the water condensed out in the cold trap. The fifth term is the oxygen tied up in nitric oxide. The sixth term is free oxygen remaining in the sample reaching the analysis instruments. Equation A-1 can be arranged to yield:

$$\text{AIR} = \frac{(1 + \frac{x}{4})\ CO_2 + (1/2 + \frac{x}{4})\ CO + 1/2\ NO + O_2}{0.2095 + 0.0003 + 0.0003\ x/4} \quad \text{(A-2)}$$

A carbon balance can be used to calculate the moles of fuel burned per 100 moles of dry flue gas:

$$\text{FUEL} = CO_2 - 0.0003 \text{ AIR} + CO \quad \text{(A-3)}$$

The moles of air available per mole of burned fuel in the sample gas can be obtained by taking the ratio of the values from Eq. A-2 and A-3. AIR must be calculated first, before calculation of FUEL. If the combustion were in stoichiometric proportions, the moles of air would be, by an oxygen demand calculation:

$$\text{AIR}_{stoich} = \frac{(1 + x/4)\ \text{FUEL}}{0.2095} \quad \text{(A-4)}$$

The stoichiometric ratio of the locally sampled burned gases is a parameter frequently used in this report. It is defined as the ratio of AIR to AIR_{stoich}:

$$\text{SR} = \frac{\text{AIR}}{\text{AIR}_{stoich}} \quad \text{(A-5)}$$

Combination of Eq. A-2 through A-5 yields a direct calculation of the burned gas stoichiometric ratio in terms of the measured parameters:

$$SR = \frac{\dfrac{(1+\frac{x}{4})\,CO_2 + (1/2+\frac{x}{4})\,CO + 1/2\,NO + O_2}{0.2095 + 0.0003 + 0.0003\,x/4}}{\dfrac{(1+\frac{x}{4})}{0.2095}\left[CO_2 + CO - 0.0003\,\dfrac{(1+\frac{x}{4})\,CO_2 + (1/2+\frac{x}{4})\,CO + 1/2\,NO + O_2}{0.2095 + 0.0003 + 0.0003\,x/4}\right]} \qquad \text{(A-6)}$$

According to the above definition, when the sample contains just a sufficient amount of air to oxidize all of the fuel in the sample to CO_2 plus condensed-out water, then SR = 1. As a second example, if there is twice the required amount of air for complete oxidation of the fuel, then SR = 2. Note that the stoichiometric ratio, as calculated from Eq. A-6 does not require that the products in the flue gas be in chemical equilibrium.

Note that the accuracy of the stoichiometric ratio calculation would be affected very little if all terms in Eq. A-6 containing the factors 0.0003 and NO were ignored. These factors represent the carbon dioxide originally present in free air, and the oxygen tied up in nitric oxide, respectively.

One partially questionable assumption made in the formulation of Eq. A-6 was that all hydrogen originally present in the fuel becomes oxidized to water and is removed in the cold trap. This was a necessary assumption, since there was no instrument available to measure the actual hydrogen content of the sample gas. The assumption is very good under the combined conditions of air-rich stoichiometric ratios (SR > 1) and chemical equilibrium. To test this assumption, a Rocketdyne thermochemical computer code was used to calculate the species concentrations under conditions of chemical equilibrium for stoichiometric ratios from 0.8 to 2.8. These calculations included the equilibrium presence of free H_2. The actual stoichiometric ratios of these combustion gases, compared to those calculated by Eq. A-6 (which does not recognize the presence of H_2) are given in Table A-2, where it can be seen that Eq. A-6 is quite accurate except for SR < 1. Calculated equilibrium conditions are tabulated in Tables A-3 and A-4.

TABLE A-2. VALIDITY OF STOICHIOMETRIC RATIO CONDITIONS

Actual Stoichiometric Ratio	Stoichiometric Ratio Calculated from Eq. B-6
0.800	0.844
1.000	1.003
1.200	1.197
1.400	1.400
1.600	1.600
2.000	2.002
2.400	2.404
2.800	2.804

The primary cause of the inaccuracy at SR < 1 is the unaccounted for presence of H_2. In nonequilibrium gases, there is likely to be H_2 present even where none would be indicated from equilibrium calculations and, at fuel-rich conditions, there could be more or less than indicated from equilibrium calculations. Because of this likelihood of nonequilibrium, no attempt was made to correct the calculations of Eq. A-6 by means of equilibrium calculations.

The concentration of CO_2 (dry basis) in the flue gas in the parameter most often used in the space heating industry as an indication of combustion conditions. To illustrate the relationship of $\%CO_2$ to the stoichiometric ratio, equilibrium data from Table A-4 were used to calculate the curve shown in Fig. A-2; a calculated $\%O_2$ curve is also shown. A number of values of measured CO_2 concentrations in actual furnace flue gases are also plotted on Fig. A-2. The measured data

Table A-3. EQUILIBRIUM COMBUSTION GAS PROPERTIES FOR NO. 2 DISTILLATE FUEL OIL BURNED WITH AIR

($CH_{1.814}$, 18,443 Btu/lb Net Heat of Combustion With Air at 14.67 psia)

Stoich. Ratio*	Oil + Air Inlet Temp., F	Flame Temperature, F	C_p Frozen, Btu/lb-R	γ Frozen	Viscosity, centipoise	Thermal Conductivity, Btu/hr-ft-F	Prandtl Number	Molecular Weight
0.8	0	3429	0.346	1.261	0.0666	0.0702	0.7946	27.73
1.0		3614	0.341	1.254	0.0687	0.0711	0.7984	28.80
1.2 ↓		3290	0.333	1.260	0.0653	0.0661	0.7954	29.00
1.4		2940	0.324	1.267	0.0615	0.0610	0.7915	29.03
1.6 Air		2649	0.318	1.275	0.0581	0.0567	0.7880	29.03
2.0 Rich		2209	0.307	1.288	0.0527	0.0500	0.7820	29.02
2.4 ↓		1897	0.298	1.298	0.0487	0.0452	0.7771	29.01
2.8		1663	0.291	1.308	0.0456	0.0415	0.7730	29.00
0.8	70	3778	0.347	1.261	0.0671	0.0709	0.7948	27.72
1.0		3649	0.341	1.254	0.0691	0.0715	0.7984	28.77
1.2		3336	0.333	1.259	0.0658	0.0667	0.7956	29.00
1.4		2991	0.325	1.267	0.0621	0.0617	0.7918	29.03
1.6		2703	0.318	1.274	0.0589	0.0574	0.7884	29.03
2.0		2765	0.308	1.286	0.0535	0.0509	0.7825	29.02
2.4		1955	0.299	1.297	0.0495	0.0461	0.7778	29.01
2.8		1722	0.193	1.306	0.0464	0.0425	0.7738	29.00
0.8	200	3867	0.347	1.260	0.0681	0.0720	0.7951	27.71
1.0		3709	0.342	1.257	0.0698	0.0725	0.7983	28.73
1.2		3418	0.334	1.259	0.0668	0.0678	0.7958	28.98
1.4		3085	0.326	1.266	0.0632	0.0629	0.7923	29.02
1.6		2802	0.320	1.273	0.0600	0.0588	0.7890	29.02
2.0		2369	0.309	1.284	0.0548	0.0524	0.7834	29.02
2.4		2061	0.301	1.294	0.0509	0.0477	0.7790	29.01
2.8		1831	0.295	1.303	0.0479	0.0441	0.7751	29.00

*Stoichiometric ratio is unity at 14.49 masses of air per mass of fuel, and proportionately greater than unity for increasing relative mass of air.

Table A-4. CALCULATED EQUILIBRIUM COMBUSTION GAS COMPOSITION, VOLUME OR MOLE PERCENT

Stoich. Ratio	Oil + Air Inlet Temp., F	H	O	Ar	OH	H_2	H_2O	CO	CO_2	NO	N_2	O_2
0.8	0	0.0630	0.0000	0.821	0.0499	2.016	12.263	7.243	8.687	0.000	68.837	0.000
1.0		0.0397	0.0313	0.866	0.2816	0.250	11.690	1.393	12.052	0.253	72.522	0.619
1.2		0.000	0.0217	0.882	0.1862	0.030	10.141	0.161	11.247	0.390	73.784	3.160
1.4		0.000	0.0000	0.890	0.0757	0.000	8.832	0.0203	9.841	0.2955	74.465	5.566
1.6		0.000	0.0000	0.895	0.0790	0.000	7.799	0.000	8.679	0.2080	74.947	7.444
2.0		0.000	0.0000	0.902	0.000	0.000	6.297	0.000	7.000	0.0829	75.603	10.107
2.4		0.000	0.0000	0.907	0.000	0.000	5.276	0.000	5.864	0.0339	76.028	11.888
2.8		0.000	0.0000	0.910	0.000	0.000	4.541	0.000	5.046	0.000	76.326	13.161
0.8	70	0.0737	0.0000	0.821	0.0613	1.996	12.271	7.268	8.659	0.017	68.901	0.000
1.0		0.0455	0.0362	0.866	0.3072	0.269	11.647	1.501	11.934	0.272	72.456	0.666
1.2		0.0000	0.0261	0.882	0.2082	0.036	10.121	0.195	11.210	0.404	73.751	3.159
1.4		0.0000	0.0000	0.890	0.0885	0.000	8.824	0.026	9.835	0.322	74.447	5.553
1.6		0.0000	0.0000	0.895	0.0351	0.000	7.795	0.000	8.678	0.223	74.933	7.432
2.0		0.0000	0.0000	0.902	0.000	0.000	6.297	0.000	7.000	0.096	75.596	10.100
2.4		0.0000	0.0000	0.907	0.000	0.000	5.276	0.000	5.863	0.041	76.023	11.884
2.8		0.0000	0.0000	0.910	0.000	0.000	4.541	0.000	5.046	0.018	76.323	13.159
0.8	200	0.0964	0.0000	0.821	0.0878	1.964	12.273	7.318	8.604	0.027	68.796	0.000
1.0		0.0577	0.0468	0.864	0.3579	0.304	11.562	1.710	11.705	0.310	73.328	0.754
1.2		0.0000	0.0356	0.882	0.2533	0.048	10.078	0.270	11.127	0.451	73.683	3.162
1.4		0.0000	0.0000	0.890	0.1157	0.000	8.806	0.042	9.816	0.373	74.405	5.526
1.6		0.0000	0.0000	0.895	0.0493	0.000	7.787	0.000	8.672	0.268	74.905	7.406
2.0		0.0000	0.0000	0.902	0.0000	0.000	6.295	0.000	7.000	0.125	75.582	10.085
2.4		0.0000	0.0000	0.907	0.0000	0.000	5.276	0.000	5.863	0.059	76.015	11.876
2.8		0.0000	0.0000	0.910	0.0000	0.000	4.541	0.000	5.046	0.028	76.319	13.154

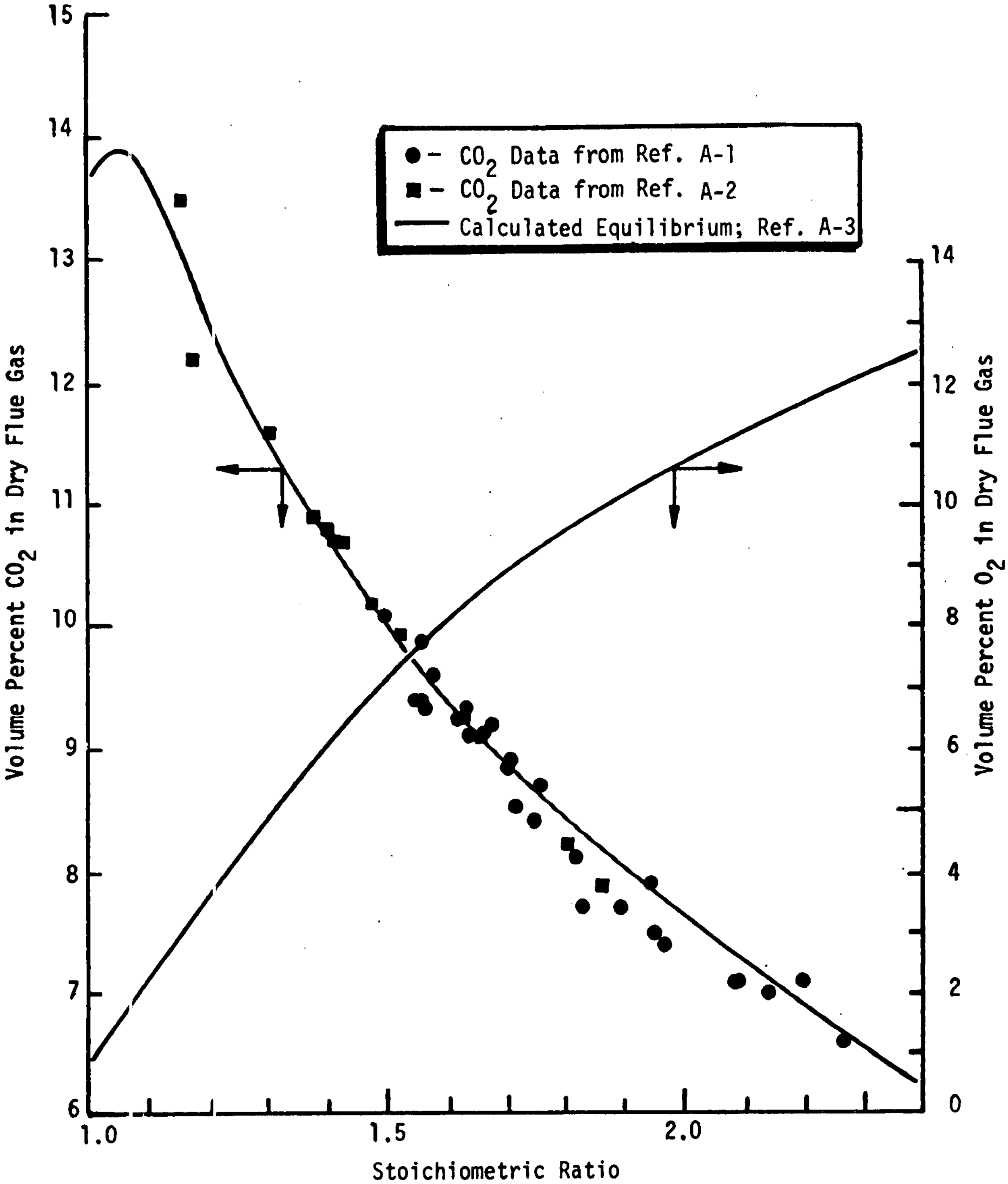


Figure A-2. Flue gas CO_2 and O_2 concentrations for no. 2 fuel oil burned in ambient air at 1 atm are seen to be very well correlated by the calculated equilibrium curve at SR > 1.1 (the calculcated maximum CO_2 concentration as the stoichiometric condition is approached by reducing excess air is not normally observed in furnace testing).

Other parameters of interest for the flue gases are the mass ratio of nitric oxide to burned fuel, the mass ratio of carbon monoxide to burned fuel, and the mass ratio of unburned hydrocarbons (as CH_4) to burned fuel. These ratios are generally expressed herein as grams of nitric oxide per kilogram of burned fuel (g NO/kg fuel), grams of methane per kilogram of fuel (g UHC/kg fuel), and grams of carbon monoxide per kilogram of burned fuel (g CO/kg fuel). These parameters are calculated by aid of Eq. A-2 and A-3 from the following relationships:

$$\frac{\text{g NO}}{\text{kg fuel}} = \frac{(1000)\ (\text{NO})\ (MW_{NO})}{(CO_2 - 0.0003\ \text{AIR} + \text{CO})\ (MW_F)} \qquad \text{(A-7)}$$

$$\frac{\text{g CO}}{\text{kg fuel}} = \frac{(1000)\ (\text{CO})\ MW_{CO}}{(CO_2 - 0.0003\ \text{AIR} + \text{CO})\ (MW_F)} \qquad \text{(A-8)}$$

$$\frac{\text{g UHC}}{\text{kg fuel}} = \frac{(1000)\ (\text{HC})\ (MW_{CH_4})}{(CO_2 - 0.0003\ \text{AIR} + \text{CO})\ (MW_F)} \qquad \text{(A-9)}$$

where

MW_{NO} = molecular weight of NO = 30.01

MW_F = molecular weight of fuel

= 12.01 + 1.008 x = 13.84

MW_{CO} = molecular weight of CO = 28.01

MW_{CH_4} = molecular weight of methane = 16.04

For calculation of the above quantities, the term 0.0003 AIR can be neglected without introducing more than about 0.1% error in the calculations, or AIR can be computed from Eq. A-3 and included in the calculation. The numbers given in this report inlcude the effect of the term. The experimental data were reduced, according to the above equation, by means of a remote terminal timeshare computer program.

In addition to the gaseous pollutants described above, the smoke content of the mixed gases was also measured. The instrument utilized for this purpose was a Bacharach smoke meter. (It is manufactured by the Bacharach Instrument Company, Pittsburgh, Pennsylvania.) This is a hand-held device which, when pumped, sucks flue gases from a 0.006 m (1/4-inch) OD, uncooled sample probe through a piece of white filter paper; 10 strokes of the pump, over a period of about 15 seconds, causes the passage of 57.2 m^3 of flue gas per m^2 of filter paper (2250 $in.^3/in.^2$). The smoke particles deposit out on the filter paper. A reading is taken by comparing the darkness of the smoke deposition spot to a scale of 10 such calibrated spots provided with the instrument. The readings vary from 0 to 9. A reading of zero corresponds to no visually detectable deposit on the filter paper, while a reading of 9 corresponds to a dark black deposit. Intermediate readings are varying shades of black and gray, increasing in darkness with increasing reading numbers. A reading of 1 is generally accepted by the industry as a very acceptable degree of smoke. At the opposite extreme, a reading of 9, which is totally unacceptable, still does not correspond to sufficient smoke to be easily visible from observation of the exhaust stack outlet.

REFERENCES

A-1. Barrett, R. E., S. E. Miller and D. W. Locklin, Field Investigation of Emissions from Combustion Equipment for Space Heating, EPA R2-73-084a (API Publ. 4180), Environmental Protection Agency, Research Triangle Park, N. C., June 1973.

A-2. Hall, R. E., J. H. Wasser, and E. E. Berkau, A Study of Air Pollutant Emissions from Residential Heating Systems, EPA-650/2-74-003, Environmental Protection Agency, Research Triangle Park, N. C., January 1974.

A-3. Dickerson, R. A., and A. S. Okuda, Design of an Optimum Distillate Oil Burner for Control of Pollutant Emissions, EPA-650/2-74-047, Environmental Protection Agency, Research Triangle Park, N. C., June 1974.

APPENDIX B

DATA TABULATION: RESEARCH COMBUSTOR EXPERIMENTS

Cycle-averaged flue gas composition data are tabulated for tests of the 1 ml/s (gph) optimum burner, fitted with various prototype sheet metal optimum heads and fired in a 0.222 m (8.75 inch) ID cylindrical refractory-lined, research combustion chamber. Notations are given in the table to delineate burner orientation with respect to the combustor, combustion chamber length upstream of the water-cooled copper-coil heat exchanger, burner firing level, and spray angle. The latter two pieces of information are contained in the coded designation for a spray nozzle, e.g., "1.0-60°-A" denotes a firing rate of 1.0 gph (1.05 ml/s) and a hollow-cone (A) spray angle of 60 degrees. Cycle timing for all tests was 10 minutes on/20 minutes off except for Runs 516 to 518 which were 4-minutes on/8-minutes off cycles.

SIDE-FIRED

21-GAGE, TYPE 430 STAINLESS-STEEL, PROTOTYPE OPTIMUM HEAD (INITIAL DESIGN)

Configuration	RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
1.0-60°-A, L=0.75m	463	1.25	12.3	4.5	21	93	1	0.37	1.659	0.008	0.2	366
	464	1.46	10.6	7.0	22	93	1	0.45	1.940	0.013	0.2	429
	465	1.39	11.1	6.3	22	96	0	0.43	1.915	0.005	0.3	410
	466	1.12	13.8	2.3	30	116	0	0.44	1.833	0.002	0.9	332
	467	1.07	14.1	1.5	26	110	1	0.38	1.680	0.008	0.7	332
	468	1.20	12.7	3.6	25	117	0	0.40	1.999	0.000	0.0	416
	469	1.16	13.1	3.0	25	121	0	0.38	1.996	0.001	0.3	413

18-GAGE, TYPE 430 STAINLESS-STEEL, PROTOTYPE OPTIMUM HEAD (INITIAL DESIGN

Configuration	RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
1.0-30°-A, L=0.75m	470	1.14	13.5	2.7	10	121	0	0.15	1.961	0.004	0.1	318
	471	1.12	13.8	2.4	10	114	0	0.15	1.812	0.004	0.2	341
	472	1.09	14.0	1.9	21	103	2	0.32	1.596	0.017	0.7	324
	473	1.05	14.5	1.0	139	136	0	1.93	2.026	0.002	2.0	382
	474	1.44	10.6	6.7	11	129	0	0.23	2.648	0.002	0.0	435
	475	1.37	11.1	5.9	10	134	0	0.18	2.620	0.001	0.0	429
	476	1.32	11.5	5.3	10	136	0	0.18	2.560	0.001	0.0	432
	477	1.22	12.7	4.0	10	138	0	0.16	2.388	0.001	0.0	393
1.0-30°-A, L=0.50m	478	1.33	11.6	5.6	10	129	0	0.20	2.451	0.004	0.0	321
	479	1.16	13.3	3.1	10	127	0	0.17	2.093	0.002	0.2	291
	480	1.05	14.5	1.1	130	121	3	1.81	1.814	0.028	4.5	277
	481	1.45	10.6	6.9	10	125	0	0.19	2.596	0.003	0.0	332

SIDE-FIRED

21-GAGE INITIAL DESIGN PROTOTYPE OPTIMUM HEAD WITH 0.00020 m REINFORCEMENT PLATE

Configuration	RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TEG C
1.0-60°A, L=0.5m	482	1.25	12.4	4.5	11	106	1	0.20	1.897	0.006	0.3	285
	483	1.21	12.6	3.9	15	109	0	0.24	1.891	0.004	0.4	279
	484	1.18	12.7	3.4	17	110	0	0.28	1.856	0.004	0.4	279
	485	1.23	12.1	4.1	15	106	0	0.26	1.857	0.004	0.3	279
1.0-60°A, L=0.75m	486	1.06	14.5	1.3	167	130	10	2.35	1.969	0.080	2.0	338
	487	1.13	13.7	2.6	10	125	0	0.15	2.008	0.003	0.2	357
	488	1.29	12.0	5.0	10	118	1	0.17	2.164	0.006	0.0	374
	489	1.25	12.2	4.5	10	121	0	0.18	2.167	0.005	0.0	368
	490	1.20	12.8	3.7	10	121	0	0.16	2.069	0.004	0.0	366
	491	1.37	11.1	6.0	15	121	2	0.27	2.378	0.021	0.0	383
	492	1.44	10.6	6.8	15	114	4	0.31	2.354	0.049	0.0	391
1.0-90°A, L=0.75m	493	1.62	9.5	8.5	6	85	3	0.15	1.980	0.043	0.8	377
	494	1.45	10.6	6.9	8	95	2	0.17	1.971	0.020	0.5	366
	495	1.29	11.9	5.0	12	104	0	0.22	1.927	0.001	1.0	346
	496	1.18	12.8	3.4	20	110	0	0.31	1.847	0.004	1.2	324
	497	1.12	13.6	2.3	40	110	6	0.59	1.748	0.051	2.1	307
1.0-45°A, L=0.75m	498	1.19	12.8	3.6	20	131	1	0.32	2.235	0.010	0.0	388
	499	1.10	13.9	2.1	278	140	300	4.06	2.190	2.500	0.8	382
	500	1.27	12.2	4.8	20	136	1	0.35	2.466	0.014	0.0	407
	501	1.33	11.5	5.5	16	129	2	0.30	2.444	0.018	0.0	407
1.0-60°A, L=0.75m	502	1.18	12.7	0.0	20	126	0	0.33	2.114	0.004	0.0	427
	503	1.25	12.4	4.4	16	121	1	0.28	2.151	0.011	0.0	435
	504	1.13	13.6	2.6	17	129	1	0.27	2.069	0.006	0.2	421
	505	1.06	14.3	1.2	25	132	1	0.36	1.990	0.007	0.5	404
0.75-60°A, L=0.75m	506	1.66	9.3	8.9	13	85	1	0.31	2.033	0.019	0.0	346
	507	1.01	15.0	0.2	620	103	20	8.30	1.475	0.153	5.7	257
	508	1.46	10.9	7.2	10	93	1	0.21	1.942	0.008	0.0	318
	509	1.24	12.7	4.4	10	95	0	0.16	1.674	0.005	0.2	291
	510	1.16	13.6	3.1	15	103	0	0.23	1.703	0.004	0.2	277
	511	1.09	14.5	1.8	30	108	1	0.43	1.671	0.006	0.8	266
	512	1.41	11.0	6.5	2	97	2	0.06	1.953	0.021	0.0	310
1.0-60°A, L=0.75m, Ign. off	513	1.25	12.3	4.5	20	118	1	0.33	2.111	0.008	0.2	302
	514	1.12	13.6	2.4	20	117	0	0.30	1.874	0.003	0.4	341
	515	1.37	11.1	6.0	17	114	2	0.33	2.233	0.024	0.0	374
1.0-60°A, L=0.75m, 4-on/8-off	516	1.37	11.1	6.0	18	112	2	0.35	2.197	0.026	0.0	35[illegible]
	517	1.05	14.5	1.0	315	110	24	4.33	1.640	0.190	3.3	313
	518	1.24	12.2	4.3	22	102	2	0.33	1.807	0.024	0.0	332

TUNNEL-FIRED

21-GAGE INITIAL DESIGN PROTOTYPE OPTIMUM HEAD WITH 0.0020 m REINFORCEMENT PLATE

		RUN NO.	STOIC. RATIO	CO_2 %	O_2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
1.0-60°A	L=0.75m	519	1.29	11.9	4.9	11	81	1	0.20	1.498	0.006	0.0	357
1.0-60°A	L=0.75m	520	1.07	14.1	1.4	25	81	0	0.35	1.234	0.004	0.6	341
1.0-60°A	L=0.75m	521	1.00	15.0	0.1	≥1600	86	11	≥21.24	1.224	0.083	5.5	329
1.0-60°A	L=0.75m	522	1.35	11.3	5.7	10	81	0	0.20	1.557	0.004	0.0	354
1.0-60°A	L=0.75m	523	1.47	10.2	7.0	15	81	1	0.29	1.708	0.009	0.0	371
1.0-60°A	L=0.75m	524	1.18	13.1	3.5	12	81	0	0.20	1.370	0.004	0.0	341
1.0-60°A	L=0.75m	525	1.12	13.8	2.4	16	85	1	0.25	1.352	0.005	0.0	332
1.0-60°A	L=0.75m, Ign. Off	526	1.11	13.9	2.3	456	76	25	6.72	1.212	0.210	0.4	329
1.0-60°A	L=0.75m, Ign. Off	527	1.43	10.7	6.6	16	81	0	0.32	1.669	0.004	0.0	352
1.0-60°A	L=0.75m, Ign. Off	528	1.09	13.4	1.8	1179	75	5	16.98	1.167	0.041	0.1	338
1.0-60°A	L=0.50m	529	1.18	13.1	3.5	46	79	1	0.74	1.328	0.009	2.3	257
1.0-60°A	L=0.50m	530	1.40	11.0	6.4	60	76	1	1.12	1.540	0.016	0.0	285
1.0-60°A	L=0.50m	531	1.32	11.7	5.4	62	72	2	1.10	1.357	0.020	0.0	268
1.0-60°A	L=0.50m	532	1.25	12.2	4.5	47	70	4	0.80	1.257	0.040	0.6	260
1.0-90°A	L=0.75m	533	1.57	9.9	8.1	20	73	1	0.42	1.652	0.014	0.0	341
1.0-90°A	L=0.75m	534	1.11	13.9	2.3	21	93	0	0.32	1.471	0.001	1.0	288
1.0-90°A	L=0.75m	535	1.32	11.7	5.5	15	81	1	0.28	1.538	0.009	0.4	321
1.0-90°A	L=0.75m	536	1.23	12.6	4.2	15	81	0	0.24	1.417	0.005	0.0	313
1.0-30°A	L=0.75m	537	1.04	14.5	1.0	1163	125	0	16.08	1.861	0.002	0.0	330
1.0-30°A	L=0.75m	538	1.18	13.2	3.5	20	117	0	0.31	1.979	0.004	0.0	368
1.0-30°A	L=0.75m	539	1.26	12.3	4.6	11	110	0	0.20	1.975	0.003	0.0	368
1.0-30°A	L=0.75m	540	1.36	11.2	5.9	15	106	0	0.29	2.074	0.003	0.0	399
1.0-30°A	L=0.75m	541	1.02	15.0	0.5	198	134	2	2.69	1.953	0.019	0.8	324
1.25-60°A	L=0.75m	542	1.17	13.3	3.2	20	85	0	0.31	1.409	0.003	0.0	391
1.25-60°A	L=0.75m	543	1.28	12.1	5.0	20	63	1	0.34	1.168	0.006	0.0	404
1.25-60°A	L=0.75m	544	1.03	15.1	0.6	620	85	1	8.44	1.241	0.006	0.0	354
1.25-60°A	L=0.75m	545	1.08	14.3	1.6	20	85	1	0.30	1.301	0.007	0.0	354
0.75-60°A	L=0.75m	546	1.75	8.8	9.5	10	81	1	0.24	2.044	0.011	0.0	313
0.75-60°A	L=0.75m	547	1.41	11.0	6.5	10	93	0	0.21	1.875	0.003	0.0	288
0.75-60°A	L=0.75m	548	1.13	13.5	2.5	17	101	0	0.27	1.617	0.003	0.0	266
0.75-60°A	L=0.75m	549	1.06	14.5	1.3	30	110	2	0.42	1.668	0.016	0.0	246
0.75-60°A	L=0.75m	550	1.18	13.1	3.4	25	109	1	0.39	1.835	0.006	0.0	268

APPENDIX C

DATA TABULATIONS: WARM-AIR FURNACE EXPERIMENTS

Cycle-averaged flue gas composition data and thermal efficiency data are tabulated for tests of two warm air furnaces: a Williamson Model 1167-15 and a Carrier Model 58HV-156. Each furnace was tested in the laboratory with its stock burner, then with the stock head replaced by one or more prototype optimum burner heads. For the Williamson furnace, fired at a nominal rate of 1.0 gph (1.05 ml/s), emissions data are given in Table C-1, and efficiency data are listed in Tables C-2 and C-3. Similarly, Table C-4 lists emissions data, and Tables C-5 and C-6 show efficiency data for the Carrier furnace. The stock Carrier furnace was tested at a nominal 1.1 gph (1.16 ml/s) firing rate; with the optimum head, the firing rate was 1.0 gph (1.05 ml/s). In the tables of efficiency data, values are given for each of four cycles at each test condition; these are followed by efficiency averages for the test. Emissions data tables are shorter because only the averages have been given.

Table C-1. CYCLE-AVERAGED FLUE GAS COMPOSITION DATA FROM TESTS OF THE WILLIAMSON MODEL 1167-15 FURNACE

STOCK BURNER HEAD

RUN NO.	STOIC. RATIO	CO_2 %	O_2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
53	1.41	11.0	6.5	17	84	1	0.34	1.706	0.010	0.0	243
54	1.32	11.8	5.4	15	91	0	0.28	1.717	0.004	0.0	232
55	1.26	12.2	4.6	15	93	0	0.25	1.683	0.005	0.0	227
56	1.28	12.2	5.0	10	109	0	0.17	2.003	0.003	0.0	241
57	1.20	12.9	3.8	10	110	0	0.16	1.887	0.005	0.0	221
58	1.13	13.5	2.6	20	107	1	0.30	1.726	0.011	0.1	218
59	1.06	14.2	1.2	78	97	1	1.11	1.456	0.009	3.5	204
60	1.11	12.0	1.9	25	106	1	0.37	1.666	0.008	3.0	204
61	1.42	11.0	6.6	10	93	1	0.21	1.900	0.008	0.0	243
62	1.49	10.3	7.3	10	86	1	0.22	1.849	0.011	0.0	246
63	1.13	13.4	2.5	18	106	0	0.28	1.707	0.003	2.0	210
64	1.32	11.8	5.4	15	107	0	0.26	2.015	0.003	0.0	229
65	1.47	10.6	7.2	12	94	0	0.25	1.993	0.006	0.0	249
66	1.43	10.9	6.7	12	88	0	0.25	1.809	0.005	0.0	246
67	1.26	12.1	4.6	15	102	0	0.25	1.842	0.003	0.0	227
68	1.20	12.8	3.8	17	106	0	0.29	1.824	0.004	0.0	218
69	1.08	14.1	1.7	21	96	1	0.32	1.478	0.006	3.0	199
70	1.09	14.1	1.8	20	103	1	0.30	1.603	0.006	2.5	199
71	1.44	10.8	6.8	10	93	0	0.19	1.911	0.004	0.0	238
72	1.30	12.0	5.3	14	107	0	0.25	1.990	0.003	0.0	248
73	1.11	14.0	2.3	18	108	1	0.28	1.714	0.006	0.1	218

PROTOTYPE OPTIMUM HEAD

(21-gage, type 430 stainless steel, initial design with 0.0020 m reinforcement plate)

RUN NO.	STOIC. RATIO	CO_2 %	O_2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
*100A	1.13	13.1	3.5	25	79		0.39	1.335		1.5	253
*100B	1.24	12.6	4.4	20	81		0.33	1.431		0.7	257
*100C	1.30	12.0	5.3	15	81		0.26	1.520		0.0	263

PROTOTYPE OPTIMUM HEAD

(18-gage, type 310 stainless-steel, revised design)

RUN NO.	STOIC. RATIO	CO_2 %	O_2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
101	1.37	11.4	6.0	20	74	2	0.38	1.450	0.026	0.0	246
102	1.59	9.8	8.3	48	69	15	1.04	1.579	0.182	0.0	271
103	1.46	10.4	7.3	28	73	4	0.57	1.561	0.044	0.0	271
104	1.15	13.3	2.9	35	80	1	0.53	1.309	0.007	3.0	232
105	1.28	12.0	4.8	15	81	1	0.27	1.486	0.009	1.7	249
106	1.38	11.3	6.3	30	76	2	0.55	1.519	0.021	0.3	254
107	1.55	10.0	8.0	41	72	9	0.87	1.607	0.107	0.0	268
108	1.54	10.1	7.9	35	73	6	0.72	1.614	0.071	0.0	271
109	1.39	11.2	6.4	20	83	1	0.37	1.654	0.012	0.0	254
110	1.35	11.3	5.7	13	78	1	0.25	1.501	0.007	1.5	252
111	1.58	9.8	8.2	38	76	6	0.82	1.723	0.072	0.0	271
112	1.35	11.3	5.8	25	77	1	0.45	1.500	0.010	0.1	249
113	1.30	11.8	5.2	23	79	1	0.42	1.483	0.010	2.0	246
114	1.42	11.0	6.7	30	80	1	0.59	1.639	0.012	0.0	257
115	1.47	10.4	7.4	36	74	1	0.74	1.595	0.014	0.0	260
116	1.54	10.1	7.9	41	72	3	0.87	1.597	0.034	0.0	268
117	1.62	9.6	8.6	61	64	18	1.35	1.511	0.224	0.0	277
135	1.33	11.6	5.6	21	81	1	0.39	1.540	0.010	1.1	260
136	1.50	10.4	7.5	26	77	2	0.54	1.666	0.018	0.0	268
137	1.40	11.1	6.5	22	79	1	0.43	1.591	0.009	1.2	257

*Runs 100A, B, and C were steady-state experiments.

Table C-2. CYCLE-AVERAGED THERMAL EFFICIENCY DATA FROM TESTS OF THE WILLIAMSON MODEL 1167-15 FURNACE WITH ITS STOCK BURNER HEAD

RUN NO.	STOIC RATIO	GROSS EFF. W.A. %	GROSS EFF. F.G. %	BURN TIME SEC	W.A. TIME SEC	Q FUEL KJ	Q W.A. KJ	WARM AIR M3/S	W.A. DEL-T C	T(N) F.G. C	T AMB C
53	1.38	67.93	80.73	240	344	9544	6483	0.5785	27.7	218	19
53	1.38	69.10	80.76	245	380	9746	6734	0.5770	26.1	217	20
53	1.38	69.92	80.43	263	388	10462	7315	0.5727	28.0	223	20
53	1.38	72.85	80.52	258	377	10256	7471	0.5806	29.0	221	18
		[illegible]	[illegible]								
54	1.28	76.25	81.45	241	371	9581	7305	0.5779	29.0	213	18
54	1.29	70.29	81.44	245	356	9743	6848	0.5796	28.2	212	19
54	1.28	72.65	81.45	265	383	10535	7653	0.5915	28.7	213	18
54	1.27	74.22		258	392	10253	7610	0.5925	27.9		18
		73.35	81.44								
55	1.23	74.01	82.06	239	371	9302	6885	0.5733	27.5	208	18
55	1.23	71.34	82.03	240	386	9338	6662	0.5631	26.1	208	17
55	1.23	77.39		244	376	9494	7347	0.5869	28.3		17
55	1.23	75.33	82.03	264	386	10272	7738	0.5853	29.1	208	17
		74.52	82.04								
56	1.22	83.15	81.26	239	398	9281	7717	0.5996	27.5	223	14
56	1.23	79.41	81.06	240	391	9320	7401	0.5821	27.7	226	14
56	1.23	81.06	81.48	244	403	9475	7680	0.5874	27.6	217	14
56	1.29	76.95	80.69	264	424	10252	7888	0.5780	27.4	226	14
		80.14	81.12								
57	1.18	77.72	82.68	241	375	9463	7355	0.5925	28.2	205	15
57	1.15	82.34	83.11	241	457	9470	7797	0.6001	24.2	201	16
57	1.16	76.43	83.01	245	425	9627	7358	0.5899	25.0	201	16
57	1.16	76.51	83.07	263	440	10337	7909	0.5810	26.3	201	17
		78.25	82.96								
58	1.07	79.20	83.84	258	448	10256	8123	0.5795	26.6	199	18
58	1.10	81.64	83.91	240	454	9541	7789	0.5859	24.9	193	18
58	1.10	76.92	84.32	241	436	9581	7369	0.5719	25.2	185	18
58	1.16	76.38	83.46	244	443	9700	7408	0.5768	24.7	193	18
		78.54	83.89								
59	1.05	80.28	84.79	258	469	10151	8149	0.5785	25.6	182	18
59	1.02	80.31	84.77	240	452	9448	7588	0.5760	24.8	187	20
59	1.03	81.42	84.95	241	442	9488	7725	0.5794	25.7	181	20
59	1.04	79.28		245	463	9645	7647	0.5788	24.3		20
		80.32	84.84								
60	1.25	68.16	83.40	259	376	10303	7022	0.5845	27.2	184	20
60	1.25	77.53	83.33	240	394	9541	7397	0.5836	27.4	185	18
60	1.26	78.42	83.21	240	391	9535	7477	0.5818	28.0	186	17
60	1.27	78.88	83.17	245	392	9733	7677	0.5808	28.7	186	17
		75.75	83.28								
61	1.40	66.81	80.16	264	359	10787	7207	0.5972	28.6	226	13
61	1.38	65.94	80.13	260	343	10621	7003	0.5859	29.7	230	13
61	1.38	72.15	80.26	240	343	9804	7073	0.6017	29.2	227	13
61	1.39	70.61	80.10	241	329	9848	6954	0.5943	30.2	229	13
		68.88	80.16								
62	1.45	70.34	79.51	264	377	10680	7513	0.5806	29.2	235	13
62	1.46	70.31	79.63	260	392	10522	7398	0.5758	27.9	231	14
62	1.49	69.75	79.48	241	346	9750	6800	0.5723	29.2	232	13
62	1.45	68.81		241	375	9753	6711	0.5731	26.6		14
		69.81	79.54								
63	1.11	73.51	83.79	265	407	10511	7726	0.5644	28.6	195	15
63	1.13	75.25	83.55	260	388	10316	7762	0.5755	29.6	197	15
63	1.09	76.42	84.09	241	377	9562	7307	0.5800	28.4	191	15
63	1.11	79.49		240	383	9525	7572	0.5807	28.9		16
		76.17	83.81								
64	1.29	69.83	81.72	265	392	10640	7429	0.5697	28.3	208	18
64	1.29	71.04	81.58	260	419	10442	7418	0.5671	26.6	210	18
64	1.27	73.74		241	385	9679	7137	0.5816	27.1		18
64	1.28	74.29	81.62	240	400	9738	7234	0.5659	27.2	210	18
		72.22	81.64								
65	1.42	72.82	79.90	265	407	10752	7829	0.5801	28.2	230	18
65	1.45	69.10	79.75	260	389	10546	7287	0.5682	28.0	230	18
65	1.43	75.02	79.78	240	392	9731	7300	0.5802	27.3	231	17
65	1.41	74.46	80.07	240	379	9636	7175	0.5819	27.7	227	18
		72.85	79.88								
66	1.38	77.62	80.03	262	416	10402	8074	0.5904	28.0	232	16
66	1.40	79.13		240	401	9532	7543	0.5975	26.8		17
66	1.39	77.78	80.28	242	392	9614	7478	0.5906	27.5	225	17
66	1.40	78.07	80.10	246	404	9773	7630	0.5862	27.4	228	17
		78.15	80.14								
67	1.25	78.48	81.94	261	428	10269	8059	0.5823	27.5	207	18
67	1.23	83.81		240	437	9439	7911	0.5896	26.1		18
67	1.24	81.61	82.09	241	440	9488	7743	0.6007	24.9	206	20
67	1.25	78.98	82.08	246	460	9691	7654	0.5870	24.1	205	21
		80.72	82.04								
68	1.16	80.63	83.22	241	395	9093	7332	0.6106	25.9	198	20
68	1.17	88.16	83.09	242	428	9124	8044	0.6073	26.3	199	18
68	1.18	81.67	83.08	245	431	9243	7549	0.5854	25.5	198	20
68	1.17	87.55	83.12	264	464	9957	8717	0.5878	27.2	198	19
		84.50	83.13								
69	1.04	73.46	85.19	242	401	9533	7003	0.5806	25.6	175	21
69	1.04	75.55	85.07	243	436	9573	7232	0.5748	24.6	177	21
69	1.04	80.37	84.95	246	406	9675	7776	0.5910	27.6	180	18
69	1.04	75.12	84.80	265	407	10419	7827	0.5862	27.9	183	17
		76.12	85.00								
70	1.03	79.07	84.90	247	310	9601	7592	0.8172	25.5	182	16
70	1.03	77.73	84.97	266	335	10350	8045	0.7953	25.7	181	17
70	1.04	82.07	84.92	260	340	10116	8302	0.8120	25.6	181	17
70	1.04	84.59	84.92	240	335	9338	7899	0.8029	25.0	181	17
		80.87	84.93								
71	1.40	85.84	80.60	247	334	9617	8255	0.8265	25.5	218	18
71	1.40	80.39		265	356	10324	8299	0.8048	24.6		20
71	1.40	80.69	80.70	260	346	10133	8175	0.7949	25.3	217	20
71	1.40	84.54	80.90	241	347	9395	7943	0.8320	23.4	213	21
		82.86	80.73								
72	1.25	74.84		600	752	23914	17898	0.7692	26.3		23
72	1.27	72.21	80.87	600	707	23899	17256	0.7783	26.7	223	22
72	1.25	83.02	80.92	600	806	21942	18215	0.7700	25.0	224	23
72	1.25	83.54	81.14	600	887	22942	19166	0.7634	24.1	220	24
		78.40	80.98								
73	1.05	81.38	84.35	600	905	22957	18682	0.7627	23.0	192	25
73	1.06	79.66	84.28	600	946	22957	18288	0.7625	21.6	192	25
73	1.06	78.10	84.13	600	757	23406	18280	0.7586	27.1	196	22
73	1.06	76.88	84.13	600	731	23652	18183	0.7949	26.6	196	22
		79.00	84.22								

Table C-3. CYCLE-AVERAGED THERMAL EFFICIENCY DATA FROM TESTS OF THE WILLIAMSON MODEL 1167-15 FURNACE RETROFIT WITH A PROTOTYPE OPTIMUM BURNER HEAD

RUN NØ.	STØIC RATIØ	GRØSS EFF. W.A. %	GRØSS EFF. F.G. %	BURN TIME SEC	W.A. TIME SEC	Q FUEL KJ	Q W.A. KJ	WARM AIR M3/S	W.A. DEL-T C	T(N) F.G. C	T AMB C
101	1.32	70.13	80.18	241	316	10029	7033	0.7909	24.0	235	11
101	1.33	67.98	80.06	255	302	10602	7207	0.7890	25.7	236	9
101	1.32	66.77	80.04	269	343	11191	7473	0.7612	24.4	238	10
101	1.33	70.55	79.82	269	355	11092	7825	0.7752	24.2	242	12
		68.86	80.03								
102	1.53	68.13	77.75	247	331	10185	6939	0.7650	23.3	256	12
102	1.54	62.09	77.66	257	311	10597	6580	0.7668	23.4	256	12
102	1.56	67.80	77.32	274	352	11291	7655	0.7687	24.1	260	11
102	1.56	68.89	77.64	247	326	10212	7035	0.7796	23.5	255	13
		66.73	77.59								
103	1.43	62.71	78.77	225	245	8991	5638	0.7681	25.4	250	10
103	1.45	68.28	78.68	231	298	9245	6312	0.7723	23.3	250	12
103	1.46	63.70	78.02	241	251	9636	6138	0.8182	25.4	260	11
103	1.41	71.09	78.45	244	275	9756	6936	0.8541	25.1	260	11
		66.45	78.48								
104	1.13	81.25	82.43	233	344	9614	7811	0.7552	25.6	218	13
104	1.10	73.12	82.90	237	359	9804	7168	0.8613	19.7	214	17
104	1.13	69.72	82.57	247	407	10228	7131	0.6643	22.4	216	16
104	1.13	64.65	82.46	260	346	10731	6938	0.7622	22.4	218	13
		72.19	82.59								
105	1.25	69.41	80.51	240	317	10386	7208	0.7823	24.7	236	12
105	1.25	74.39	80.61	224	341	9694	7211	0.7572	23.7	233	12
105	1.25	70.64	80.39	245	322	10599	7487	0.7734	25.6	240	11
105	1.23	69.40	80.75	258	341	11161	7746	0.7578	25.5	234	11
		70.96	80.57								
106	1.33	75.97	79.89	265	389	10795	8201	0.6293	28.5	240	8
106	1.33	67.28	79.73	266	367	10846	7297	0.6310	26.8	244	10
106	1.33	65.87	79.57	246	326	10024	6603	0.6248	27.6	248	8
106	1.33	71.56	79.68	240	367	9877	7068	0.6284	26.1	245	8
		70.17	79.72								
107	1.50	65.81	77.63	266	392	10954	7209	0.6323	24.7	261	10
107	1.49	65.96	77.84	265	397	10923	7205	0.6250	24.7	259	11
107	1.51	62.70	77.70	266	368	10954	6868	0.6227	25.5	258	10
107	1.49	0.00	78.07	247	367	10172	0	0.0000	25.5	255	10
		64.83	77.81								
108	1.50	71.96	77.73	254	416	10557	7596	0.5386	28.8	259	8
108	1.49	70.48	77.70	263	419	10935	7707	0.5474	28.6	261	9
108	1.50	73.89	77.63	264	433	10872	8033	0.5528	28.6	261	10
108	1.51	73.40	77.59	244	407	10055	7380	0.5538	27.8	260	11
		72.43	77.66								
109	1.35	81.65	80.14	248	451	9948	8122	0.5911	25.9	232	16
109	1.35	75.31	79.92	261	464	10472	7887	0.5740	25.2	237	17
109	1.33	74.22	80.03	261	475	10686	7931	0.5877	24.2	237	17
109	1.34	73.60	80.12	241	452	9870	7265	0.5709	24.0	233	17
		76.20	80.05								
110	1.31	71.99	80.24	260	397	10642	7661	0.5219	31.5	235	16
110	1.33	73.91	80.01	261	407	10683	7895	0.5237	31.5	237	16
110	1.33	70.89	80.08	241	359	9759	6918	0.5151	31.8	236	15
110	1.33	72.32	80.34	234	385	9479	6855	0.5159	29.4	230	16
		72.28	80.16								
111	1.53	75.43	77.85	235	461	9619	7255	0.4978	26.9	254	16
111	1.56	78.33	77.81	243	538	9959	7800	0.5044	24.5	252	18
111	1.56	75.90	77.74	260	509	10329	7840	0.5081	25.8	253	17
111	1.54	70.95	78.22	240	502	9553	6778	0.5081	22.6	247	21
		75.15	77.91								
112	1.33	73.57	81.15	234	469	9215	6780	0.4955	24.8	214	20
112	1.33	71.55	80.66	245	479	9652	6905	0.4933	24.9	222	21
112	1.33	75.01	80.84	261	509	10282	7713	0.5053	25.5	219	21
112	1.33	79.45	80.27	260	509	10236	8133	0.5031	27.0	231	20
		74.90	80.73								
113	1.28	73.24	80.61	233	377	9265	6786	0.4970	30.8	229	19
113	1.28	78.00	80.70	245	407	9740	7596	0.4996	31.8	227	18
113	1.28	74.56	80.70	259	424	10296	7677	0.5009	30.8	227	18
113	1.28	76.42	80.70	259	427	10296	7868	0.5012	31.3	227	18
		75.56	80.67								
114	1.38	81.93	79.57	258	439	10355	8484	0.5194	31.7	242	17
114	1.38	81.16	79.70	236	433	9472	7688	0.5050	29.9	239	17
114	1.38	79.95	79.98	234	446	9395	7511	0.5015	28.6	233	18
114	1.38	81.97	79.98	242	450	9915	8127	0.5186	29.7	233	18
		81.25	79.81								
115	1.45	80.10	79.08	237	439	9525	7630	0.5272	28.1	242	20
115	1.45	81.28	79.08	233	436	9364	7611	0.5292	28.1	242	20
115	1.45	82.76	79.11	237	442	9528	7885	0.5331	28.5	242	20
115	1.45	82.71	79.11	258	482	10266	8491	0.5132	29.2	242	20
		81.71	79.09								
116	1.50	78.07	78.42	247	451	9927	7749	0.5154	28.4	248	20
116	1.50	78.70	78.59	233	422	9364	7369	0.5185	28.6	245	20
116	1.51	80.81	78.36	242	439	9729	7862	0.5234	29.1	247	20
116	1.50	70.50	78.62	265	475	10654	7510	0.4763	28.3	245	20
		77.02	78.50								
117	1.59	78.99	77.59	234	448	9506	7509	0.5200	27.4	252	21
117	1.57	72.41	77.68	233	422	9466	6854	0.5137	26.9	252	21
117	1.59	79.49	77.35	243	425	9866	7842	0.5364	29.3	256	20
117	1.59	87.04	77.42	256	451	10593	9220	0.5440	32.0	255	18
		79.48	77.51								
135	1.29	69.66	79.87	247	256	9983	6954	0.7324	31.6	247	12
135	1.29	74.92	79.85	247	281	9983	7479	0.7289	31.0	247	12
135	1.31	74.39	79.76	227	241	9180	6829	0.7700	31.3	246	13
135	1.31	67.16	79.76	225	238	9100	6111	0.7209	30.3	246	13
		71.53	79.81								
136	1.45	73.41	78.31	234	244	9470	6951	0.8173	29.7	256	14
136	1.43	68.08	78.71	247	301	9996	6805	0.6697	28.7	251	14
136	1.45	69.37	78.49	248	310	10039	6964	0.6847	27.9	253	15
136	1.45	69.31	78.46	227	272	9186	6367	0.6895	28.8	253	14
		70.04	78.49								
137	1.35	68.20	79.68	248	289	10036	6844	0.6927	29.1	242	14
137	1.35	73.55	79.68	248	311	10036	7381	0.6868	29.4	242	14
137	1.35	72.03	79.68	229	286	9267	6675	0.6888	28.8	242	14
137	1.35	70.90	79.68	225	274	9105	6455	0.6949	28.9	242	14
		71.17	79.68								

Table C-4. CYCLE-AVERAGED FLUE GAS COMPOSITION DATA FROM TESTS OF THE CARRIER MODEL 58HV-156 FURNACE

STOCK BURNER HEAD

RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
88	1.61	9.6	8.5	15	92	1	0.32	2.135	0.014	0.0	329
89	1.64	9.6	8.9	20	89	1	0.46	2.101	0.019	0.0	338
90	1.36	11.4	6.0	16	102	2	0.31	1.994	0.021	0.0	307
91	1.24	12.6	4.3	40	97	1	0.66	1.704	0.009	3.5	282
92	1.95	8.0	11.0	71	66	12	1.90	1.882	0.188	0.0	349
93	1.70	9.3	9.4	32	83	5	0.75	2.031	0.065	0.0	343
94	1.41	11.1	6.7	21	96	2	0.41	1.946	0.020	0.0	318
95	1.50	10.5	7.6	23	96	1	0.48	2.067	0.013	0.0	327
96	1.57	10.1	8.4	23	91	2	0.51	2.070	0.027	0.0	329
97	1.70	9.2	9.4	35	76	4	0.80	1.886	0.055	0.0	332
98	1.78	8.8	9.9	40	72	8	0.96	1.866	0.105	0.0	343

PROTOTYPE OPTIMUM HEAD

(21 gage, type 310 stainless-steel, revised design)

RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
118	1.52	10.2	7.7	21	63	1	0.45	1.377	0.014	0.0	313
119	1.33	11.7	5.6	26	68	1	0.48	1.296	0.012	1.8	291
120	1.60	9.7	8.4	27	72	2	0.60	1.658	0.026	0.0	316
121	1.39	11.2	6.4	21	76	2	0.41	1.529	0.019	2.5	302
122	1.60	9.8	8.5	27	74	4	0.60	1.718	0.050	0.0	318
123	1.51	10.2	7.6	25	74	4	0.51	1.620	0.052	0.0	316
124	1.45	10.6	7.0	17	74	1	0.35	1.555	0.014	0.0	304
125	1.38	11.3	6.3	23	72	2	0.44	1.427	0.019	1.9	299
126	1.27	12.4	4.9	28	76	1	0.49	1.392	0.014	2.8	227
127	1.40	11.3	6.5	21	79	1	0.41	1.591	0.016	1.3	302
128	1.54	10.2	8.0	35	77	6	0.72	1.719	0.071	0.2	307
129	1.47	10.6	7.2	25	81	4	0.51	1.704	0.045	0.3	307
130	1.45	12.2	7.9	35	76	7	0.68	1.592	0.077	0.4	318
131	1.44	10.8	6.9	21	73	2	0.42	1.515	0.018	0.4	310
132	1.49	10.4	7.4	21	81	5	0.44	1.739	0.057	0.0	307
133	1.38	11.1	6.2	25	77	4	0.46	1.527	0.042	1.7	302
134	1.44	10.8	6.9	20	78	2	0.40	1.611	0.027	1.4	

Table C-5. CYCLE-AVERAGED THERMAL EFFICIENCY DATA FROM TESTS OF THE CARRIER MODEL 58 HV-156 FURNACE WITH ITS STOCK BURNER HEAD

RUN NO.	STOIC RATIO	GROSS EFF. W.A. %	GROSS EFF. F.G. %	BURN TIME SEC	W.A. TIME SEC	Q FUEL KJ	Q W.A. KJ	WARM AIR M3/S	W.A. DEL-T C	T(N) F.G. C	T AMB C
88	1.57	64.08	74.32	245	281	9947	6374	0.5542	34.8	312	20
88	1.57	67.49	74.46	258	332	10475	7070	0.5437	33.3	309	20
88	1.57	64.60	74.43	238	296	9659	6240	0.5421	33.0	310	19
88	1.53	65.49	75.00	235	284	9538	6246	0.5432	34.4	307	19
		65.42	74.55								
89	1.56	63.02	74.13	238	266	9949	6270	0.5722	35.0	318	18
89	1.56	66.36	74.27	235	289	9824	6519	0.5351	35.9	316	18
89	1.56	65.07	74.16	245	302	10245	6666	0.5388	34.8	318	19
89	1.57	65.35	73.71	259	292	10618	6939	0.5387	37.5	323	19
		64.95	74.07								
90	1.30	72.96	78.41	234	308	9407	6863	0.5468	34.6	281	20
90	1.31	73.27	78.20	243	317	9772	7160	0.5521	34.8	283	21
90	1.33	71.84	77.94	257	346	10339	7427	0.5411	33.8	285	21
90	1.33	71.45	77.91	254	338	10635	7599	0.5463	35.0	285	21
		72.38	78.12								
91	1.16	75.44	80.24	254	359	10430	7868	0.5468	34.1	260	22
91	1.18	75.14	80.13	232	344	9526	7158	0.5447	32.5	260	22
91	1.23	71.07	79.88	231	332	9485	6741	0.5396	32.0	257	22
91	1.20	73.17	79.95	240	338	9855	7211	0.5541	32.7	260	22
		73.71	80.05								
92	1.91	61.48	70.36	257	289	10764	6617	0.5512	35.4	326	22
92	1.88	60.26	70.58	237	280	9929	5983	0.5571	32.6	326	22
92	1.88	63.02	70.54	232	277	9717	6123	0.5649	33.3	326	22
92	1.88	58.82	70.54	242	284	10235	6020	0.5509	32.7	326	22
		60.89	70.51								
93	1.62	58.09	73.21	233	230	9823	5706	0.5494	38.3	323	16
93	1.62	63.23	73.11	240	241	10124	6401	0.5956	37.9	325	17
93	1.63	65.29	73.11	256	272	10806	7054	0.5810	37.9	324	18
93	1.62	68.04	73.21	257	287	10852	7384	0.5741	38.1	323	19
		63.66	73.16								
94	1.36	69.67	77.01	236	301	9684	6747	0.5506	34.6	297	21
94	1.35	70.40	77.22	244	316	10009	7046	0.5415	35.1	295	20
94	1.35	71.80	77.10	258	317	10587	7601	0.5523	36.9	297	21
94	1.35	76.17	77.07	258	347	10478	7981	0.5659	34.6	297	20
		72.01	77.10								
95	1.45	72.49	75.81	236	311	9397	6811	0.5774	32.2	305	21
95	1.43	71.58	75.95	247	313	9838	7042	0.5576	34.3	304	22
95	1.42	70.78	76.07	259	358	10316	7301	0.5356	32.4	304	22
95	1.42	77.18	76.24	258	379	9856	7606	0.5401	31.6	301	22
		73.01	76.02								
96	1.47	70.75	75.64	236	371	9418	6663	0.5411	28.2	303	25
96	1.49	71.01	75.70	245	416	9783	6946	0.5265	27.0	300	26
96	1.53	71.15	75.44	259	418	10339	7356	0.5370	27.9	297	26
96	1.50	74.38	75.43	259	436	10339	7690	0.5591	26.9	303	26
		71.82	75.55								
97	1.59	67.60	74.35	232	281	9166	6195	0.5618	33.3	308	26
97	1.64	66.22	74.11	240	300	9485	6281	0.5449	32.8	305	26
97	1.64	69.82	74.08	254	313	10035	7006	0.5428	35.1	306	26
97	1.69	73.79	73.69	256	319	9906	7310	0.5379	36.3	305	26
		69.36	74.06								
98	1.72	56.03	72.75	233	271	9394	6203	0.5538	35.2	315	25
98	1.72	65.59	72.53	240	271	9673	6344	0.5537	36.0	318	25
98	1.72	63.26	72.57	255	283	10274	6499	0.5464	35.8	318	24
98	1.74	67.10	72.32	253	283	10298	6909	0.5789	35.9	318	24
		65.49	72.54								

Table C-6. CYCLE-AVERAGED THERMAL EFFICIENCY DATA FROM TESTS OF THE CARRIER MODEL 58 HV-156 FURNACE RETROFIT WITH A PROTOTYPE OPTIMUM BURNER HEAD

RUN NO.	STOIC RATIO	GROSS EFF. W.A. %	GROSS EFF. F.G. %	BURN TIME SEC	W.A. TIME SEC	Q FUEL KJ	Q W.A. KJ	WARM AIR M3/S	W.A. DEL-T C	T(N) F.G. C	T AMB C
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
118	1.50	72.54	75.81	230	313	8949	6492	0.5428	32.5	295	17
118	1.50	72.94	75.76	238	317	9254	6750	0.5425	33.4	296	16
118	1.50	76.29	75.91	253	353	9841	7507	0.5349	33.8	292	17
		74.41	75.84								
119	1.29	78.36	78.95	231	407	8718	6831	0.5626	25.4	270	20
119	1.30	75.19	78.95	229	401	8648	6502	0.5683	24.3	268	21
119	1.29	76.33	79.03	239	419	9026	6859	0.5633	24.8	268	21
119	1.29	76.49	78.98	252	406	9717	7433	0.5712	27.3	270	20
		76.59	78.98								
120	1.59	68.21	75.30	232	388	9047	6171	0.5744	23.6	291	21
120	1.58	65.07	75.26	227	338	8844	5754	0.5891	24.6	292	20
120	1.51	72.22	75.62	236	334	9188	6636	0.5852	28.9	296	18
120	1.53	72.24	75.49	249	368	9899	7151	0.5848	28.3	296	18
		69.44	75.42								
121	1.35	79.95	77.88	241	292	9353	7478	0.6051	36.0	282	13
121	1.34	76.63	77.76	252	314	9789	7501	0.5973	34.0	286	15
121	1.34	75.44	77.79	253	329	9831	7417	0.5828	32.9	286	15
121	1.34	74.13	77.79	233	314	9054	6711	0.5787	31.4	286	15
		76.54	77.80								
122	1.56	69.29	75.00	238	296	9254	6412	0.5845	31.5	301	16
122	1.53	70.80	75.18	254	316	9883	6997	0.5784	32.6	303	17
122	1.54	70.77	75.13	253	319	9838	6961	0.5755	32.3	301	16
122	1.57	72.17	74.84	231	292	8885	6412	0.5784	32.3	302	16
		70.76	75.04								
123	1.47	71.66	75.93	239	311	9103	6524	0.5724	31.1	297	17
123	1.47	73.14	75.95	252	329	9602	7023	0.5791	31.3	297	18
123	1.47	73.57	75.93	253	341	9637	7090	0.5709	31.0	297	17
123	1.47	70.95	76.14	233	311	8692	6167	0.5761	29.3	293	19
		72.33	75.99								
124	1.42	74.84	77.25	248	362	9360	7005	0.5797	28.4	281	20
124	1.41	71.89	77.12	253	365	9536	6855	0.5763	27.7	286	18
124	1.42	76.14	77.13	251	373	9470	7211	0.5760	28.6	284	20
124	1.42	78.28	77.02	231	356	8988	7036	0.5732	29.3	286	17
		75.29	77.13								
125	1.33	73.94	78.40	239	371	9213	6812	0.5774	27.0	276	20
125	1.35	76.12	78.22	252	364	9714	7394	0.5885	29.4	276	20
125	1.33	71.04	78.21	250	331	9631	6842	0.5591	31.5	280	18
125	1.33	67.80	77.88	230	265	9126	6187	0.5816	34.2	286	15
		72.22	78.18								
126	1.20	71.33	79.26	230	256	9114	6501	0.6138	35.2	277	13
126	1.21	76.26	79.22	226	260	8955	6829	0.6299	35.4	277	13
126	1.21	66.81	79.35	230	287	9117	6091	0.5807	31.1	273	13
126	1.21		79.37								
		71.47	79.30								
127	1.33	69.41	77.88	225	283	8651	6005	0.5773	31.3	286	15
127	1.33	72.20	77.82	236	284	9068	6547	0.6069	32.3	287	14
127	1.34	71.90	77.79	247	302	9901	7118	0.6174	32.4	286	15
127	1.35	72.69	77.91	247	317	9908	7202	0.6060	31.9	282	16
		71.55	77.85								
128	1.47	68.29	76.11	225	281	8746	5972	0.6040	29.9	293	16
128	1.49	73.48	76.12	234	287	9096	6683	0.6071	32.6	291	16
128	1.50	70.12	76.02	246	313	9565	6707	0.5917	30.8	290	16
128	1.49	73.47	75.99	246	290	9562	7025	0.6254	32.9	293	16
		71.34	76.06								
129	1.41	75.32	76.86	229	268	8995	6775	0.6291	34.2	291	16
129	1.42	68.28	76.71	224	286	8796	6005	0.5780	30.9	291	15
129	1.42	68.69	76.71	234	274	9189	6312	0.6162	31.8	291	15
129	1.40	71.63	77.10	246	301	9660	6919	0.6070	32.2	288	15
		70.98	76.85								
130	1.47	68.33	75.61	250	251	9792	6691	0.6265	36.1	304	11
130	1.47	65.59	75.48	250	263	9792	6423	0.5856	35.4	307	11
130	1.47	64.78	75.53	230	241	9014	5839	0.5995	34.4	306	12
130	1.47	63.70	75.56	228	242	8846	5635	0.5821	34.0	305	12
		65.60	75.55								
131	1.38	67.23	77.45	249	281	9771	6569	0.5889	33.7	287	14
131	1.40	67.56	76.82	249	266	9762	6595	0.5992	35.1	294	12
131	1.40	65.21	76.70	229	248	8981	5856	0.5840	34.3	296	13
131	1.38	67.51	76.94	225	247	8732	5895	0.5936	34.2	296	13
		66.88	76.97								
132	1.42	66.90	76.77	234	283	9194	6151	0.6106	30.3	290	16
132	1.45	67.11	76.52	246	307	9666	6487	0.5787	31.1	290	16
132	1.46	71.00	76.37	246	292	9663	6860	0.6255	32.0	291	16
132	1.45	65.70	76.49	228	281	9142	6006	0.5931	30.6	291	16
		67.67	76.54								
133	1.35	66.69	77.61	223	287	8762	5843	0.5804	29.8	287	16
133	1.35	63.47	77.76	235	284	9234	5860	0.5744	30.5	285	16
133	1.35	66.99	77.76	246	305	9666	6475	0.5973	30.2	285	16
133	1.35	69.74	77.73	246	311	9864	6879	0.5847	32.1	285	16
		66.72	77.71								
134	1.40	66.71	77.13	213	251	8367	5581	0.6254	30.2	288	16
134	1.40	68.34	77.10	225	263	8835	6037	0.6347	30.7	288	15
134	1.40	65.46	77.25	234	269	9189	6015	0.5895	32.2	286	15
134	1.40	76.52	77.22	246	274	9657	7389	0.6320	36.3	286	15
		69.26	77.18								

TECHNICAL REPORT DATA *(Please read Instructions on the reverse before completing)*		
1. REPORT NO. EPA-600/2-76-256	2.	3. RECIPIENT'S ACCESSION NO.
4. TITLE AND SUBTITLE COMMERCIAL FEASIBILITY OF AN OPTIMUM RESIDENTIAL OIL BURNER HEAD		5. REPORT DATE September 1976 6. PERFORMING ORGANIZATION CODE
7. AUTHOR(S) L. P. Combs and A.S. Okuda		8. PERFORMING ORGANIZATION REPORT NO. R-76-103
9. PERFORMING ORGANIZATION NAME AND ADDRESS Rocketdyne Division Rockwell International Corporation 6633 Canoga Avenue Canoga Park, California 91304		10. PROGRAM ELEMENT NO. 1AB014; ROAP 21BCC-058 11. CONTRACT/GRANT NO. 68-02-1888
12. SPONSORING AGENCY NAME AND ADDRESS EPA, Office of Research and Development Industrial Environmental Research Laboratory Research Triangle Park, NC 27711		13. TYPE OF REPORT AND PERIOD COVERED Final; 6/75-2/76 14. SPONSORING AGENCY CODE EPA-ORD
15. SUPPLEMENTARY NOTES IERL-RTP project officer for this report is G. B. Martin, Mail Drop 65, 919/549-8411, Ext 2235.		

16. ABSTRACT

The report gives results of a study of the feasibility of commercializing optimum oil burner head technology developed earlier for EPA. The study included: selecting the best commercial method for fabricating optimum heads; determining that prototype simulated-production heads could reproduce an earlier research head's beneficial results; and testing prototype heads as retrofit devices in two commercial residential furnaces. Sheetmetal stamping was selected as the best fabrication method. A one-piece stamped and folded design was evolved and prototype commercial heads were fabricated. Research combustion chamber tests showed these to be equivalent to the earlier research head. Tested as retrofit replacements for stock burner heads in two new warm-air oil furnaces, the prototype heads were found to be operationally satisfactory and potentially durable and long-lived. It was estimated that widespread retrofitting of old residential units could increase mean season-averaged thermal efficiency (averaged over those units retrofitted) by about 5% and simultaneously reduce NOx emissions from these sources by about 20%. Logistics of a retrofit program, training for service personnel, and requirements to ensure meeting codes and standards were not resolved.

17. KEY WORDS AND DOCUMENT ANALYSIS

a. DESCRIPTORS	b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
Air Pollution	Air Pollution Control	13B
Combustion	Stationary Sources	21B
Oil Burners	Residential Burners	13A
Nitrogen Oxides	Burner Heads	07B
Thermal Efficiency	Commercialization	20M

18. DISTRIBUTION STATEMENT	19. SECURITY CLASS *(This Report)*	21. NO. OF PAGES
Unlimited	Unclassified	110
	20. SECURITY CLASS *(This page)* Unclassified	22. PRICE

EPA Form 2220-1 (9-73)