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Laboratory Study of a Low NO_x Hot Water Heater with a Weak-Swirl Burner

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DISCLAIMER

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LABORATORY STUDY OF A LOW NO_X HOT WATER HEATER WITH A WEAK-SWIRL BURNER

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ABSTRACT

Due to environmental concerns and increasing regulation, manufacturers are searching for methods to reduce combustion-generated emissions from their products. Lean-burning is one recognized method for reducing NO_X emissions from gas appliances. This paper describes the performance of a laboratory, low NO_x hot water heater consisting of the premixed, lean-burning Weak-Swirl Burner (WSB) and a heat exchanger from a commercial 15 kW (50,000 Btu/h) Telstar spa heater. In order to identify an optimum design for a WSB water heater, data was gathered on the overall thermal efficiency and emission levels of NO, CO, and O_2, while varying the equivalence ratio, water flow rates, and power inputs. The results of our testing show that the WSB/Telstar system achieved thermal efficiencies of 80% while producing "ultra-low" (< 25 ppm) NO emissions for equivalence ratios below 0.85. Efficiency was essentially independent of equivalence ratio, as well as NO and CO emissions.

INTRODUCTION

For the last two decades, national concerns over airborne pollutants and their environmental consequences have been driving increasingly strict emission legislation. These regulations have been steadily moving from the large-scale, point source emitters, such as the "smokestack" regulations introduced in the 1970s, to the smaller distributed sources that directly affect the individual, such as the corporate automobile fleet efficiency (CAFE) standards. New regulations limiting NO_X emissions from household and small commercial water heaters and furnaces to 50 ppm (corrected to 3% oxygen) will be taking effect in 1997, with the possibility of 25 ppm legislation in the future.

As the conventional rack-style burners in current water heaters utilize diffusion flames, temperatures approaching 2250 K are found in the reaction zone due to an equivalence ratio $\Phi \geq 1.0$. In these fuel-rich conditions, NO_X emissions are primarily attributed to thermal NO_X generation (the Zeldovich mechanism), which increases exponentially with temperature (Bowman, 1975). Once generated, relatively expensive and complex remediation techniques would be needed to reduce high NO_X concentrations below regulation limits.

Since flame temperatures drop with decreasing equivalence ratios, one commonly accepted method to reduce NO_X emissions is to utilize a lean-burning combustion process ($\Phi < 1.0$). With the lower flame temperatures (1800-2000 K) found in lean combustion, total NO_X emissions drop due to decreasing thermal NO_X generation. Premixing the fuel and air allows for precise control of Φ, enabling NO emissions to be reduced below 25 ppm.

The drawback with lean combustion systems is that the flame can be difficult to stabilize on a flame holder. Developed at Lawrence Berkeley National Laboratory (LBNL), the WSB is a new process for stabilizing lean combustion without using a flame holder, thus eliminating the problem of lean flame blow-off. Laboratory studies have shown that the flame generated by a WSB is insensitive to small perturbations in Φ, and its power output is readily scaleable without any modifications to the physical unit. The current WSB model has a 5 cm (2") diameter exit tube and can be run from 8 to 80 kW (30,000 to 300,000 Btu/h). The WSB's original purpose was to further fundamental research in turbulent combustion processes (Chan et al. 1992 and Cheng, 1991), but it has recently been recognized that it should be applicable to commercial products. The work reported in this paper is a joint effort with Teledyne Laars of Moorpark, CA to evaluate the feasibility of using the lean-burning WSB in a commercial water heating application.

WEAK-SWIRL BURNER OPERATION

The first task in this work was to design and test a compact WSB that fits within the Telstar heat exchanger. The burner's base is fabricated from a 2" pipe cross, with two of the four holes sealed shut. The fuel/air mixture enters from the side, travels up through the swirler section, and then out the exit tube. This tube has a 45° tapered rim to allow the premixture to smoothly exit the burner before reaching the reaction zone above (Figure 1).

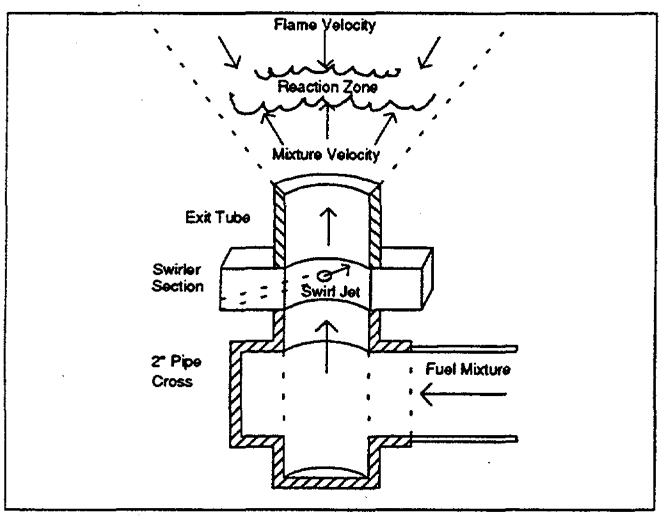

FIGURE 1: THE WEAK-SWIRL BURNER

A 1/70 horsepower centrifugal blower with a controllable damper supplies the desired amount of combustion air of up to 10 liters/sec (0 - 21.2 ft^3/min). Natural gas is supplied to the system at standard household pressure (11" of water pressure) before being reduced to 1" - 3" of water pressure and mixed with the combustion air. Compressed air at 25 psi (reduced to less than 6" of water pressure) provides the necessary swirl air.

The swirl air, delivered by four tangentially mounted jets, imparts a weak swirl to the edges of the fuel/air mixture. As the premixed fuel exits the system through the 5 cm (2") diameter burner tube, the centrifugal force due to the swirl flow on the outer edges of the premixture causes the mixture to radially diverge as it proceeds downstream. In order to conserve mass flux, the mixture velocity decreases linearly as the area increases downstream. When the mixture is ignited, the propagating flame is stabilized at the location where the flame speed matches the local mixture velocity. This creates a stable flame, insensitive to perturbations in Φ. If Φ (and thus the flame speed of the mixture) increases, the flame brush will move upstream. However, the flame will restabilize itself at the new location where the local velocity of the mixture again matches the higher flame speed. The opposite reaction also holds true; when Φ decreases, the flame moves downstream and it restabilizes at the lower local mixture velocity. Due to the absence of a stabilizing recirculation zone, the stabilization method described above was termed "Weak-Swirl" in order to differentiate itself from the high swirl process found in conventional swirl furnaces and turbines (Beer, 1972).

LABORATORY SETUP

A desktop laboratory station has been designed and constructed to test the WSB in conjunction with the Telstar heat exchanger. Figure 2 shows a schematic of the test station with its major components described below.

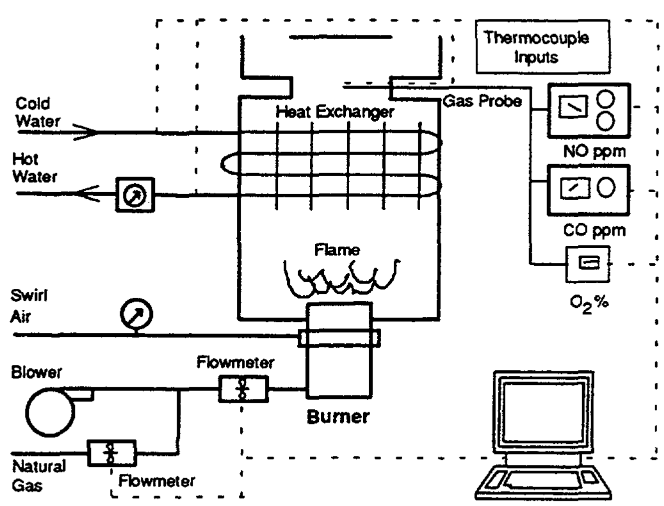

FIGURE 2: SCHEMATIC OF TESTING STATION

Throughout the tests, we use an unmodified heat exchanger from a 15 kW (50,000 Btu/h) Telstar spa heaters manufactured by Teledyne Laars. It has an allowable power range of up to 25 kW (85,000 Btu/h), and achieves overall operating efficiencies of 82 percent. The 23 cm (9") tall heat exchanger has a rectangular interior, 20 x 16.5 cm (8" x 6.5") with six parallel fin-and-tube sections crossing 4 cm (1.5") below the exit of heat exchanger. As this heat exchanger was developed for extracting radiative heat from conventional rack burners, the water tube circuit is wrapped around the outside of the heat exchanger three times before leaving the heat exchanger housing. Two turbine meters measure the actual volumetric flow of both the gas and the total mixture in the system, with the computer calculating the air flow and equivalence ratio in real-time. High pressure water lines supply between 0 to 40 liters/min (0 to 10 gal/min) to the heat exchanger with variations in the flow rate of less than 2% once the valves are set. Calibrated, high precision (± 0.1° C) thermometers measure water temperatures before and after the heat exchanger. By using these highly accurate thermometers as a reference, six Type T thermocouples can provide temperatures to within ±0.25 °C for the electronic interface with the computer. Four thermocouples are used to measure the water temperatures before, during, and after the heat exchanger. A fifth thermocouple measures the exhaust temperature of the combustion products, and a sixth thermocouple records ambient temperatures. Emission analysis of the combustion products is conducted with a chemilumenescent NO-NO_2-NO_x analyzer, an infrared CO analyzer, and a O_2 process monitor. Samples are taken 4 cm (1.5") above the fin-and-tube heat exchanger as the product gas enters the exhaust flue.

TEST CONDITIONS

The first WSB tests were performed without the heat exchanger. These stand-alone tests were used to evaluate the performance of the WSB within cylindrical enclosures and at different orientations.

After these qualitative tests were completed, the WSB was integrated into the entire testing station where quantitative performance measures of thermal efficiency and NO emissions were used for evaluation. Tests were first run to evaluate the performance of the WSB when changes to the configuration of the heat exchanger were made. All subsequent tests were run with the bottom of the heat exchanger completely enclosed and the chimney from a production Telstar unit attached to the top of the heat exchanger. Using this configuration, the thermal efficiency and emissions were evaluated while varying a) equivalence ratios from $0.60 < \Phi < 0.95$; b) input power levels between 12 to 18 kW (40,000 to 60,000 Btu/h); and c) water flow rates from 8 to 30 liters/min (2 to 8 gal/min).

The importance of these three independent variables was measured with a test matrix studying the separate effects of equivalence ratio and water flow rates on the WSB performance. This test matrix has three different power input levels. For each power level, Φ was adjusted in steps of approximately $\Delta\Phi=0.05$ between 0.65 to 0.95 while keeping the water flow rate constant. Swirl air was adjusted as necessary to stabilize the flame. After completing this series, a second series of tests was run where Φ was held constant and the water flow rate was adjusted in steps of 4 liters/min (1 gal/min) between the range of 8 to 30 liters/min (2 to 8 gal/min).

TEST PROCEDURES & DATA ANALYSIS

After setting the water flow rate, real-time data on gas and air flow rates, Φ, temperatures, thermal efficiency, and $NO/CO/O_2$ readings were continuously monitored and recorded every three seconds on the computer. Each run lasted for ten minutes; this was deemed sufficient as water temperatures reached steady-state within one minute. Thermal efficiency, ε, for this system is defined as

$$\varepsilon = \frac{m_f c_p \Delta T_w}{m_f * HHV} = \frac{\text{Energy transferred to water}}{\text{Chemical Energy entering with fuel}} \quad (1)$$

with m_w and m_f referring to the mass flow rates of water and fuel respectively, c_p the specific heat of water, HHV the high heating value of the fuel, and ΔT_w the difference between the water temperature exiting and entering the heat exchanger. For this analysis, c_p and HHV were assumed to be constant at 4.18 kJ/kg•K (0.998 Btu/lbm•° F) and 38,400 kJ/m^3 (1030 Btu/ft^3) respectively. It should be noted that this is neither a combustion nor a Carnot efficiency, but a thermal efficiency for the heat exchanger.

The percent of O_2 in the exhaust product was recorded and used to correct the raw NO ppm readings to 3% O_2 using the following equation.

$$\text{NO (corrected)} = \text{NO (measured)} * \frac{1 - \frac{0.03}{0.21}}{1 - \frac{O_2\text{ (measured)}}{0.21}} \quad (2)$$

Regulations require this scaling in order to accurately compare emission readings with different dilution levels.

RESULTS AND DISCUSSION

Video observations were used to evaluate the performance of the WSB stand-alone tests. These tests showed that tilting the WSB from vertical to 45° inclination created no instability problems. Due to buoyancy effects of the hot reaction zone, the lower edge of the flame brush was lifted, such that the flame was on a 30-35° angle rather than at 45°. These finding suggest that it may be possible to integrate the WSB with a condensing heat exchanger to reach thermal efficiencies of greater than 90%. A tilted burner is necessary with a condensing heat exchanger so that the dripping exhaust gas condensate does not disrupt the combustion region below.

The second set of qualitative tests investigated the effects of cylindrical enclosures on the WSB flame. Once again, no evidence of flame instability nor changes in the operating characteristics (i.e., no recirculating zones developed) were observed. Flame stability was not compromised when the upstream backpressure was increased, nor when ambient air flow from below the burner was reduced. Once it was determined that the WSB could be successfully operated in an enclosure, the burner was incorporated with the Telstar heat exchanger to investigate the efficiency and emission performance of the WSB in a practical application.

WSB/Telstar Efficiency

The first efficiency tests involved changing the heat exchanger's upstream and downstream conditions. This entailed enclosing the bottom of the heat exchanger and reducing the exhaust area at the top. The motivation for enclosing the bottom of the heat exchanger was two-fold. First, as the flame brush generated by the WSB has a flat disk-shaped appearance, the majority of the heat radiation is directed upwards and downwards rather than to the sides as is the case with tall diffusion flames. It is clear that enclosing the bottom will reflect the downward radiation back into the heat exchanger rather than allowing it to escape from the system. Enclosing the bottom also has the effect of reducing the amount of ambient air from entering the heat exchanger unit from below the burner. The reduction in exhaust area was undertaken to increase the residence time of the exhaust products within the heat exchanger. The effective exhaust area was reduced to 25, 34, and 46% of the unblocked exhaust outlet, with the Telstar chimney reducing the area to 46%.

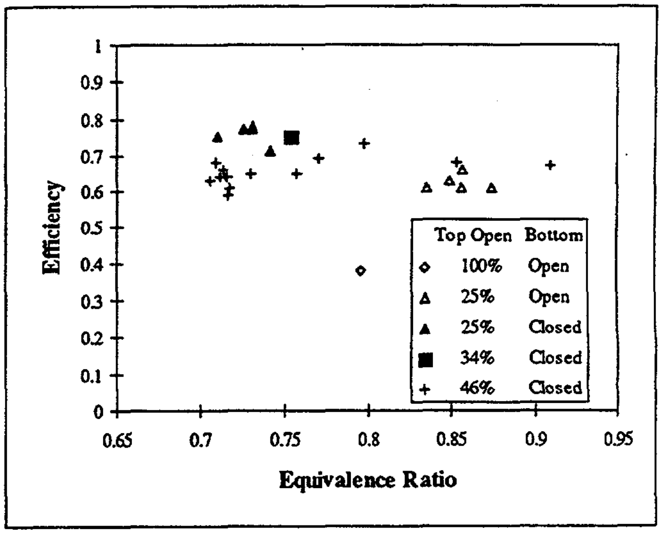

FIGURE 3: EFFICIENCIES FOR VARIOUS CONFIGURATIONS OF THE WSB/TELSTAR UNIT.

As seen in Figure 3, enclosing the bottom and reducing the exhaust area of the heat exchanger had a significant effect on the system's thermal efficiency. The remainder of the tests were run with the bottom of the heat exchanger completely enclosed and the chimney taken from a production Telstar spa heater placed on top of the heat exchanger unit (show in Figure 3 by the cross symbol).

As a preliminary step towards understanding the performance of the WSB/Telstar system under the above configuration, thermal efficiency data was gathered. Figure 4 shows the results when water flow rate was varied while holding the input power and Φ constant.

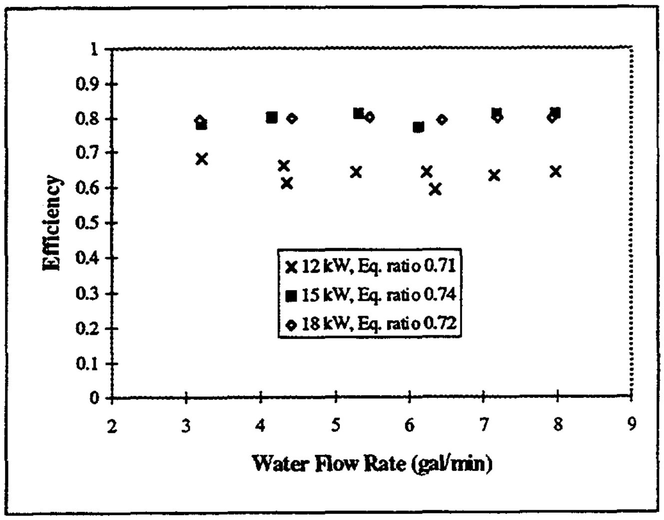

FIGURE 4: EFFICIENCY VERSUS WATER FLOW RATE WITH CONSTANT POWER AND Φ.

As can be readily seen, there is no systematic change in thermal efficiency due to varying water flow rate. There is, however, a significant decrease in efficiency when the WSB power input is decreased to 12 kW (40,000 Btu/h) at similar Φ. At this low power range, the system may be below the optimal performance range for the Telstar heat exchanger, nominally designed for 15 kW.

The data in Figure 5 address the concern that lean burning flames will not be able to reach the high thermal efficiencies achieved by rich diffusion flames due to the lower combustion temperatures found in lean flames.

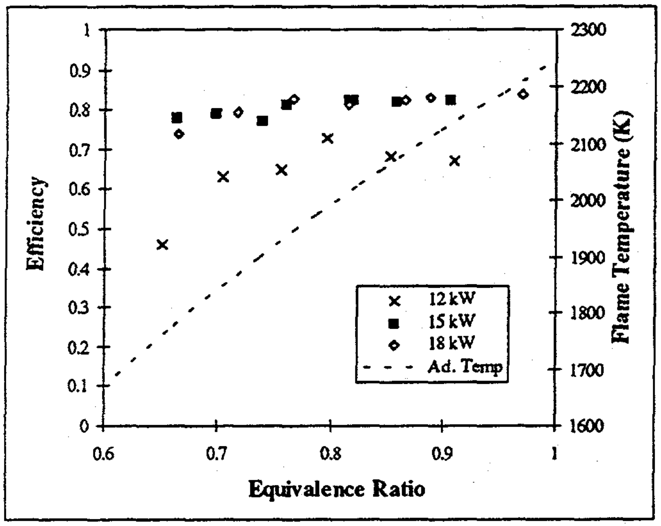

FIGURE 5: EFFICIENCY VERSUS Φ FOR CONSTANT POWER AND WATER FLOW RATES

In these runs, the 15 and 18 kW (50,000 and 60,000 Btu/h) power levels reached thermal efficiencies of 80% for $\Phi > 0.75$. This level is comparable to the recorded efficiency of a current Telstar model operating at $\Phi \geq 1.0$ and correspondingly higher flame temperatures.

For the WSB operating under these conditions, thermal efficiency was only slightly dependent on Φ. The energy transferred to the heat exchanger is linearly dependent upon only two variables. The first variable is the temperature difference between the combustion products and the heat exchanger, with the second variable being the heat transfer rate, *h*, from the combustion products to the water. Therefore, if *h* remains constant, a decrease in the flame temperature (i.e. a drop in Φ) would lead to a linear decrease in efficiency. However, in the WSB/Telstar system this decrease is offset by an increase in *h*. During these tests, Φ is decreased by adding more air to the premixture. This leads to a corresponding increase in the velocity, U, of the fuel/air mixture as it leaves the exit tube, and an increase in the velocity of the combustion products. As $h \propto U^{0.8}$ for convection heat transfer to vertical fins, the decrease in the flame temperature difference caused by decreasing Φ is offset by an increase in *h*. This leads to only a slight dependence of thermal efficiency on Φ, and allows the WSB/Telstar system to maintain the high thermal efficiencies found in current Telstar units.

WSB/Telstar Emissions

As the above figures have shown, the WSB/Telstar system achieves thermal efficiencies comparable to the commercial Telstar spa heater. Unlike the production Telstar, which relies on standard diffusion burners with $\Phi \geq 1.0$, the WSB achieves these efficiencies in the lean-burning range of $\Phi < 1.0$. Due to the lower combustion temperatures, it is expected that the NO emissions from the lean-burning WSB should be significantly less than from the commercial Telstar unit. Figure 6 shows the NO emission concentrations (corrected to 3% O_2) for the WSB at 15 and 18 kW power inputs and a Telstar heater. The proposed regulation level of 50 ppm is shown as reference.

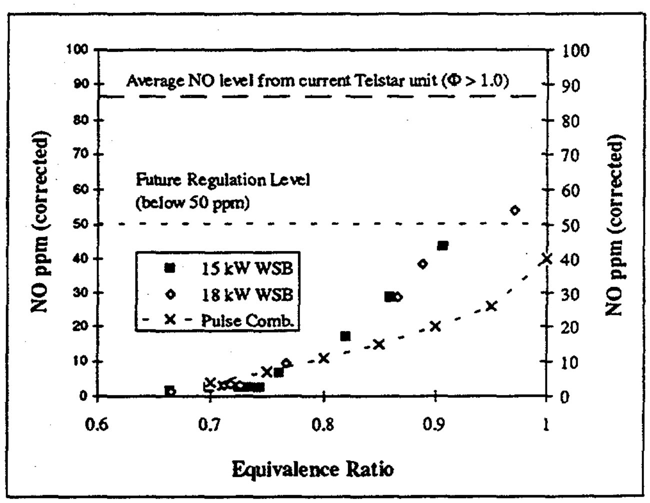

FIGURE 6: NO CONCENTRATIONS FROM WSB AND TELSTAR WITH REGULATION LIMITS.

The line representing the Telstar NO concentration level is an average of data gathered with the unit operating at $\Phi \geq 1.0$ (diffusion flame). As can be seen, the Telstar spa heater has NO emissions over 50 ppm. However, the WSB's NO emissions range from a high of 55 ppm at $\Phi = 0.95$ to a low of 4 ppm for $\Phi < 0.75$. Thus, future NO_x regulations could be easily met by the WSB/Telstar unit. The NO concentrations compare favorably with the recent results of Keller et al. (1994) obtained from a pulse combustor utilizing exhaust gas recirculation.

The results show that "low" (<50 ppm) NO emissions can be obtained with the WSB for $\Phi < 0.95$, and "ultra-low" (<25 ppm) levels can be achieved under $\Phi < 0.85$. However, CO concentrations are known to increase with decreasing equivalence ratios. In order to achieve an optimum balance between NO and CO emission, both of these pollutants should be minimized. Figure 7 gives NO and CO concentrations for two tests where both pollutants were measured concurrently.

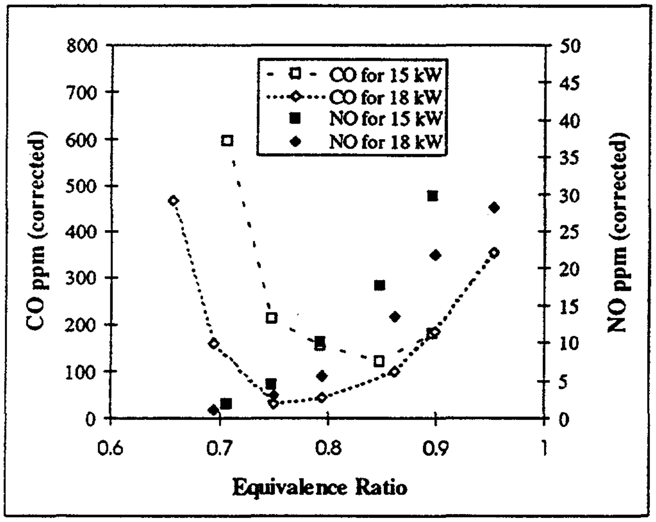

FIGURE 7: NO/CO EMISSIONS FOR THE WSB

As expected, NO emissions follow the same trend as in Figure 6, with all recorded concentrations below 35 ppm in these two tests. CO concentrations fall as Φ decreases from stochiometry (Φ = 1.0), reach a minimum where $0.75 < \Phi < 0.85$, and then begins to increase sharply as Φ drops below 0.75. This upturn may be due to greater incomplete combustion as the lean burning limit is approached. An optimal operating range for the WSB appears to exist between $0.75 < \Phi < 0.85$, where NO concentrations will be below 25 ppm and CO levels of 200 ppm or less can be expected. The higher power level, 18 kW, may be more appropriate for future applications for a WSB with a 5 cm diameter burner tube, as both NO and CO levels are lower than those found during the 15 kW test run.

As Figures 4 -7 have shown, the WSB/Telstar unit avoids the decreased efficiencies normally associated with lean-burning, while gaining the benefits of the lower NO emissions found due to decreased combustion temperatures.

SUMMARY AND CONCLUSIONS

The WSB achieved thermal efficiencies of 80%, comparable to a current model water heater under similar power and water flow rate conditions. Most importantly, "ultra-low" NO concentrations of less than 25 ppm were recorded for a broad range of equivalence ratios and power levels without compromising the system's efficiency. An optimal operating range between $0.75 < \Phi < 0.85$ was identified where both NO and CO concentrations do not exceed foreseeable regulations limits. This demonstration project has shown the feasibility of incorporating the WSB into current water heaters to produce household, low NO_X emitting water heaters within the near future.

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