LBL-37820
UC-400

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October 1995

This work was supported by the U.S. Department of Energy under Contract No. DE-AC03-76SF00098, by a Energy Research Laboratory Technology Transfer through a Cooperative Research and Development Agreement with Teledyne Laars, Moorpark, California.

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ABSTRACT

This paper reports the test results of a collaboration between Lawrence Berkeley National Laboratory and Teledyne Laars to assess the viability of incorporating the Weak-Swirl Burner (WSB) into a 15 kW (50,000 Btu/hour) Telstar spa heater. By stabilizing premixed lean combustion down to equivalence ratios $\phi \approx 0.6$, the WSB greatly reduces NO_X levels by minimizing thermally generated NO_X through the Zeldovich mechanism. The first set of experiments focus on establishing the WSB's minimum and maximum swirl requirement (corresponding to blowoff and flashback) for varying ϕ, power levels, burner size, and enclosure. The second set of experiments evaluates the performance of a laboratory water heater where the WSB is incorporated into a Telstar heat exchanger. It was found that the laboratory test station achieves "low" (< 50 ppm) and "ultra-low" (< 25 ppm) NO_X emissions without compromising the thermal efficiency. The optimum operating condition is for $\phi = 0.8$ at 18 kW (60,000 Btu/hr) where NO_X < 25 ppm and CO < 50 ppm (both corrected to 3% O_2). The results will be used as design guideline for using the low emission WSB in a prototype. The stable operation we found for enclosed WSB demonstrates the potential for its use in other combustion applications as well.

INTRODUCTION

Since the 1970s, public concern over airborne pollutants and their environmental consequences has been driving increasingly strict legislation limiting the permissible levels of toxic emissions. Combustion processes are inherently problematic, producing such pollutants as particulate matter, hydrocarbons, carbon monoxide, sulfur oxides, and nitrogen oxides to name a few, and as such are a prime contender for environmental regulations. Early laws were focused on reducing the amount of pollutants generated by industry, power plants, and automobiles. As it becomes increasingly difficult and expensive to lower emissions from these sources, emissions from household appliances such as wood burning stoves, two-stroke motors, and water heaters are being scrutinized and regulated. Proposed new legislation will limit nitric oxide, NO, and nitrogen dioxide NO_2 (know generically as NO_X) emissions to below 50 PPM (corrected to 3% oxygen), with the possibility of 25 PPM legislation in the future.

As the conventional rack-style burners in current natural gas water heaters utilize rich flames (equivalence ratio $\phi > 1.0$) with temperatures approaching 2250 K, NO_X production is primarily attributed to thermal NO_X generation through the Zeldovich mechanism, with NO_x production increasing exponentially with temperature [1]. Once generated, relatively expensive and complex remediation techniques are required to reduce high NO_X concentrations to acceptable levels. Since flame temperatures drop with decreasing ϕ, one commonly accepted method to reduce NO_X emissions is to utilize a lean burning combustion process ($\phi < 1.0$). With the lower lean flame temperatures (1700-2000 K), total NO_X emissions drop due to decreasing thermal NO_X generation. Premixing the fuel and air allows for precise control of ϕ, enabling reductions of NO emissions to 25 PPM or less.

The drawback with lean combustion systems is that the flame can be difficult to stabilize. Developed at Lawrence Berkeley National Laboratory (LBNL), the Weak-Swirl Burner (WSB) is a new process that solves the problem of lean flame blowoff [2]. Laboratory studies have shown that the operation of WSB is insensitive to small perturbations in ϕ. The power output of a relatively small burner (exit diameter of 50 mm) is readily scaleable from 8 to 80 kW (30,000 to 300,000 Btu/hr.). Developed for fundamental research in turbulent combustion processes [2,3], the WSB is also 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 water heaters. The results described here include an investigation of the swirl requirements for operating WSB in the open and in an enclosure. These experiments provided the necessary guidance for incorporating the burner in a typical water heater configuration. We have chosen the Teledyne Laars Telstar spa heater for evaluating the performance of the WSB. This natural gas heater operates at 15 kW (50,000 Btu/hr) with average thermal efficiency of 82%. In our test station, the heat exchanger and the chimney assembly of the Telstar laboratory are mounted over a WSB. Tests were performed to determine the optimum operating condition for minimum emissions without sacrificing thermal efficiency.

WEAK SWIRL BURNER

The WSB currently used for this work (Figure 1) is fabricated from a 2" pipe cross filled with marbles to generate an even flow field at the entrance to the burner tube (5 cm in diameter by 5.5 cm in length). Two perforated screens (60% blockage ratio) are placed at each end of the burner tube. These act as flashback arrestors as well as generating turbulence in the premixture. Turbulence has the additional purpose of increasing the burning rate and creating a more compact flame. After the burner tube, the premixture enters the swirler section, where four tangentially mounted air jets (3.2 mm in diameter, inclined 20° from horizontal) inject compressed air into the outer edges of the premixture. This arrangement, tangential air injection, is one of the classical swirl generators suggested by Beer and Chigier [4]. The premixture and the swirl air then leave through the exit tube of length ℓ (as measured from the swirl jets). This tube has a 45° tapered rim to allow the premixture to smoothly diverge before reaching the reaction zone. A 1/70 horsepower centrifugal blower with a controllable damper supplies up to 10 liters/sec (0 - 21.2 ft^3/min) for the combustion air. Natural gas is supplied to the system at a household pressure of 11" before being reduced to 1" - 3" of water pressure and injected into the combustion air. Compressed air at 25 psi (reduced to 0 to 10" inches of water pressure) is used to provide the swirl air.

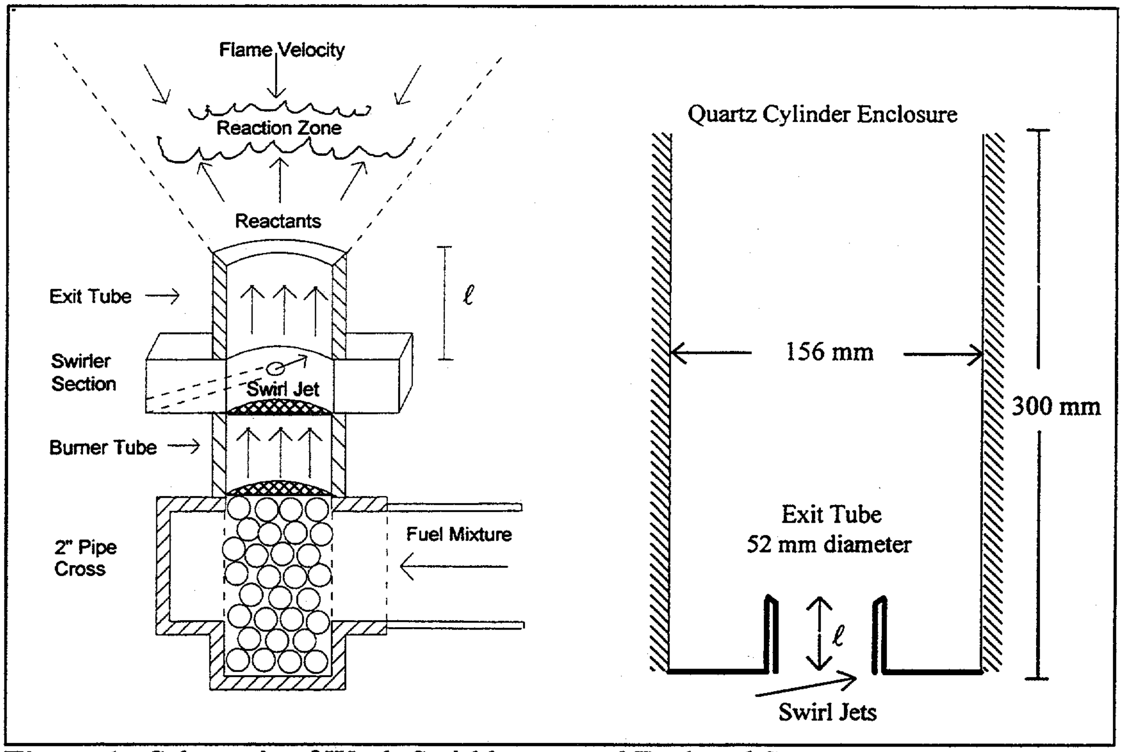

Figure 1: Schematic of Weak-Swirl burner and Enclosed System

The operating principle of the WSB has been described in References 2 and 3. As the premixed fuel exits the system through the 5 cm (2") diameter exit tube, the centrifugal force due to the swirl flow causes the mixture to radially diverge as the fuel proceeds downstream. In order to conserve mass flux, the mixture velocity decreases as the mixture increases in area downstream. Unlike conventional swirl burners, our swirl intensities are not sufficient to generate flow recirculation. When the premixture is ignited, the propagating flame is stabilized at the location where the local turbulent burning velocity matches the local velocity of the reactants. 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; where ϕ decreases, the flame moves downstream and restabilizes at the lower local mixture velocity.

SWIRL REQUIREMENTS

To provide guidance for commercial product development, our first set of experiments examined the necessary swirl requirements for WSBs with various exit tube lengths (ℓ) and with/without enclosures. To quantify swirl, Claybole and Syred [5] have shown that the swirl number S can be obtained from the burner geometry and the ratio of tangential mass flow rate to the axial flux

$$S = \frac{\pi r_\theta R}{A_\theta}\left(\frac{m_\theta}{m_\theta + m_a}\right)^2 \tag{1}$$

where r_θ (1.6 mm) and R (26 mm) are the radius of the tangential air injectors and burner respectively, A_θ (32.2 mm^2) is the total area of the injectors, and m_θ and m_a are the total mass flows in the tangential and axial directions. Conventional high swirl furnaces operate at $S \geq 1$. We have found that the swirl needed for a larger research burner [2] is $0.05 < S < 0.38$.

The swirl number at flashback and blowoff mark the upper and lower limits of the stable operation zone. The effects of ℓ on blowoff limits were investigated by using five burners with ℓ from 2 to 22 cm. These tests were performed at 15 kW (50,000 Btu/hr) power input. The effect of enclosure on the blowoff and flash-back limits was evaluated by placing a Pyrex cylinder (300 mm long by 152 mm in diameter) over the WSB (Figure 1). The burner for the enclosure test has a exit tube length ℓ of 7 cm and is the standard burner used for all subsequent tests of the laboratory station. The upper and lower limits were obtained at three power levels: 15, 18 and 21 kW.

LABORATORY TEST STATION

Shown in Figure 2 is a schematic of the laboratory test station to evaluate the performance of the WSB in the Telstar Spa Heater. As mentioned earlier, the heat exchanger normally runs at 15 kW (50,000 Btu/h) but it has an allowable power range of up to 25 kW (85,000 Btu/h). It is 23 cm (9") tall and 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. The Telstar chimney mounted on top of the heat exchanger has an exhaust diameter of 10 cm (4") that is blocked by a cap to force the combustion exhaust to flow out radially between heat dissipating fins.

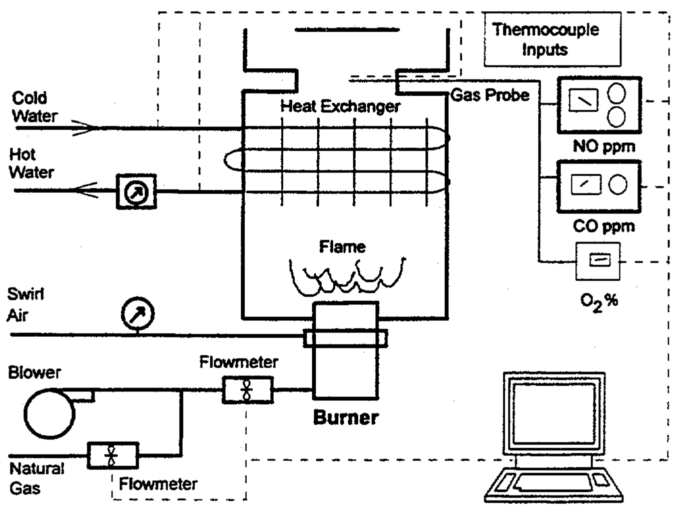

Figure 2: Schematic of Testing Station

In order for the computer to calculate the air flow and equivalence ratio, two turbine meters measure the gas and premixture volumetric flow rates. High pressure water lines supply between 0 to 40 liters/min (0 to 10 gal/min) to the heat exchanger with an accuracy of ± 2%. Calibrated, high precision (± 0.1° C) thermometers measure water temperatures before and after the heat exchanger. These thermometers are used as a references to calibrate six Type T thermocouples that measure temperatures to within ± 0.25 °C. Four thermocouples are used to measure the water temperatures entering and exiting the heat exchanger. A fifth thermocouple placed at the chimney measures the exhaust temperature of the combustion products, and a sixth thermocouple records ambient temperatures. These thermocouples are all interfaced with the computer. Emission analysis of the combustion products is conducted with a chemilumenescent $NO\text{-}NO_2\text{-}NO_x$ 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 chimney.

TEST PROCEDURE & DATA ANALYSIS

The thermal efficiency and emissions were determined by 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 test matrix has three different power input levels. For each power level, while keeping the water flow rate at 16 liters/min, ϕ was adjusted from 0.65 to 0.9 in steps of approximately $\Delta\Phi = 0.05$. Swirl air was adjusted as necessary to stabilize the flame. A second series was conducted with ϕ held constant while the water flow rate adjusted from 8 to 30 liters/min (2 to 8 gal/min) in steps of 4 liters/min (1 gal/min).

After setting the water flow rate and waiting for the water temperatures to reach steady-state (usually within one minute), the computer recorded gas and air flow rates, ϕ, temperatures, thermal efficiency, and $NO/CO/O_2$ concentrations every three seconds and refreshed the readings on the video display. Each run lasted for ten minutes. Thermal efficiency, ε, for this system is defined as

$$\varepsilon = \frac{m_f c_p \Delta T_w}{m_f * HHV} = \frac{\text{Energy transferred to water}}{\text{Chemical energy entering with fuel}} \qquad (2)$$

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.

$$NO_{(corrected)} = NO_{(measured)} * \left(1 - \frac{0.03}{0.21}\right) / \left(1 - \frac{O_{2(measured)}}{0.21}\right) \qquad (3)$$

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

RESULTS AND DISCUSSION

The minimum swirl requirements (i.e. blowoff limits) for five open burners with ℓ varied from 2 to 22 cm are shown in Figure 3. Also shown are the maximum and minimum swirl (flashback and blowoff) for the ℓ =7 cm burner at three power levels. The blowoff limits for the five burners clearly show an increase in swirl requirement with increasing ℓ. The likely cause of this trend is that with increasing ℓ, swirl air added at the outside edges of the burner is allowed to diffuse into the core of the premixture and loses some of its tangential momentum. This leads to a decrease in the centrifugal force and thus the divergence of the premixture as it leaves the exit tube. Without adequate divergence, the local premixture velocity is higher than the turbulent burning velocity and the flame is blown away.

Irregardless of the length, there is a steady increase in the swirl necessary to stabilize the flame as ϕ decrease from stochiometry (ϕ = 1.0). As flame speeds diminish with decreasing ϕ, a greater divergence of the premixture is needed to lower the reactant velocity and balance the lower flame speeds. Thus, more swirl is needed at lower equivalence ratios. Therefore, in order to maximize the operating range (i.e. have the lowest blowoff limits) of the WSB, the shortest possible tube length should be used. However, this does not imply that the ℓ = 2 cm burner is the optimum design. Repeated testing showed that its flame tended to be unsteady and could not operate for long periods of time. The ℓ = 7 cm burner is much more stable and thus was chosen for the test station.

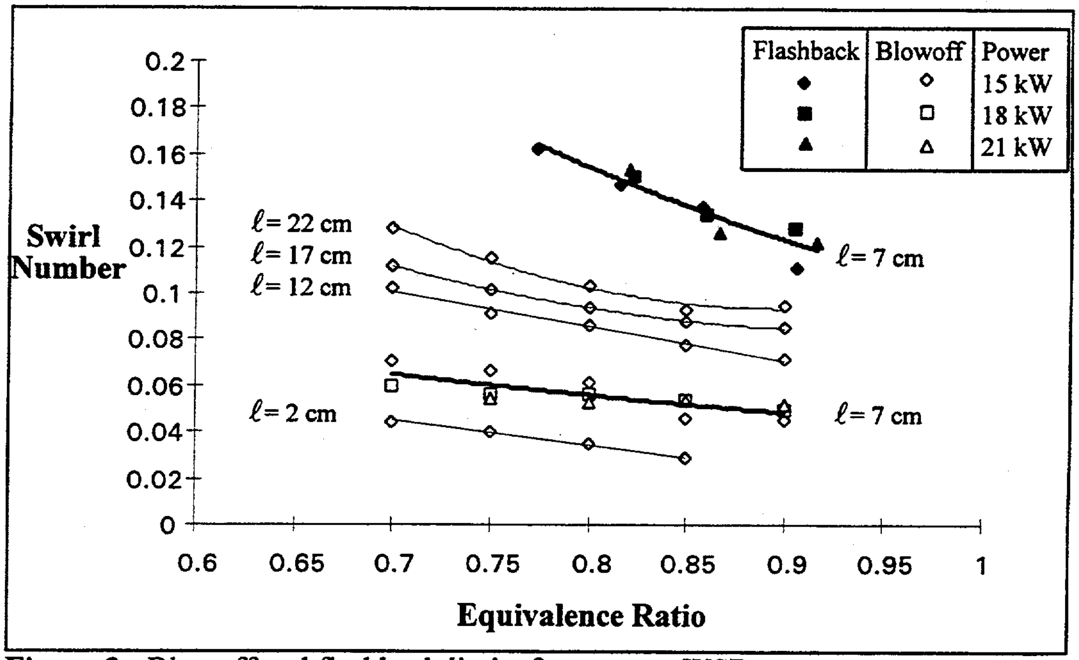

Figure 3: Blowoff and flashback limits for an open WSB

The effect of different power on swirl requirements is shown by the blowoff and flashback limits of the ℓ = 7 cm burner at 15, 18 and 21 kW. Comparison of the operating range at different ϕ also shows a broadening of the operating zone towards lean mixtures. Neither limit shows any effects due to changing power levels.

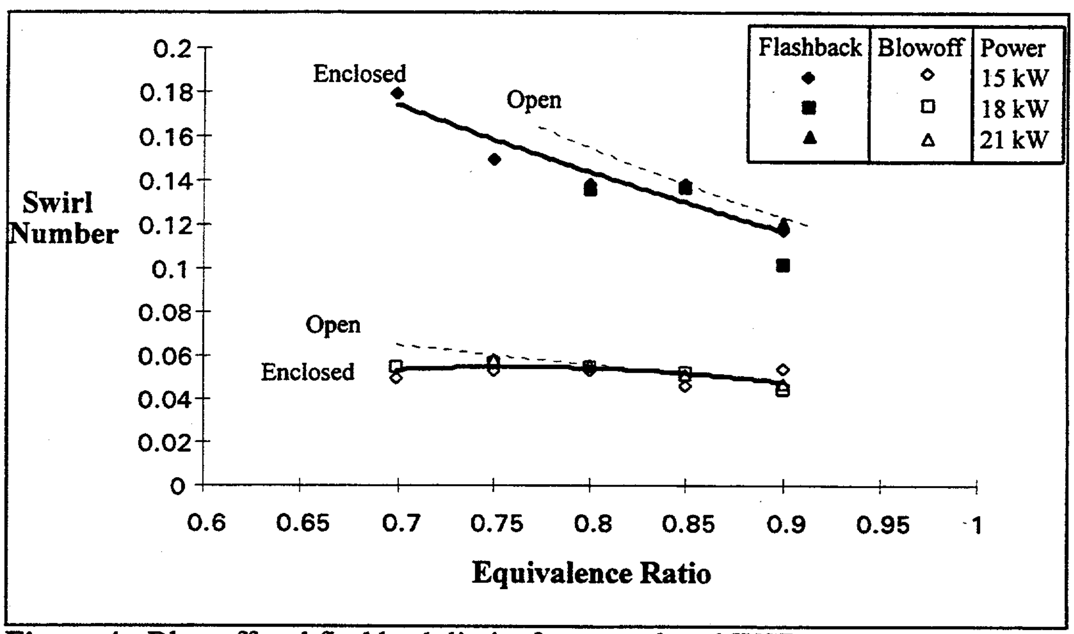

Figure 4: Blowoff and flashback limits for an enclosed WSB

Figure 4 shows the two limits for the enclosed WSB with $\ell = 7$ cm. Also shown are the polynomial fits for the open $\ell = 7$ cm burner results (as in Figure 3). It is clear that the lower limit for the enclosed burner is almost identical to the open case. For the upper limit (flashback) the enclosed burner has a slightly lower swirl number compared to the open case. These data show that the swirl requirement of the WSB is fairly independent of power level and enclosure.

The first set of results obtained from the water heater test station is shown in Figure 5. Thermal efficiencies for $\phi \approx 0.7$ at three power levels with various water flow rates are compared. As can be seen, there is no systematic change in thermal efficiency with water flow rate. For the 15 and 18 kW runs, the thermal efficiency averages about 80%, comparable to the 82% achieved by the current model Telstar heaters. There is, however, a decrease in efficiency when the WSB power input is decreased to 12 kW (40,000 Btu/h). At this low power, the system may be below the optimal performance point for the Telstar heat exchanger, nominally designed for 15 kW.

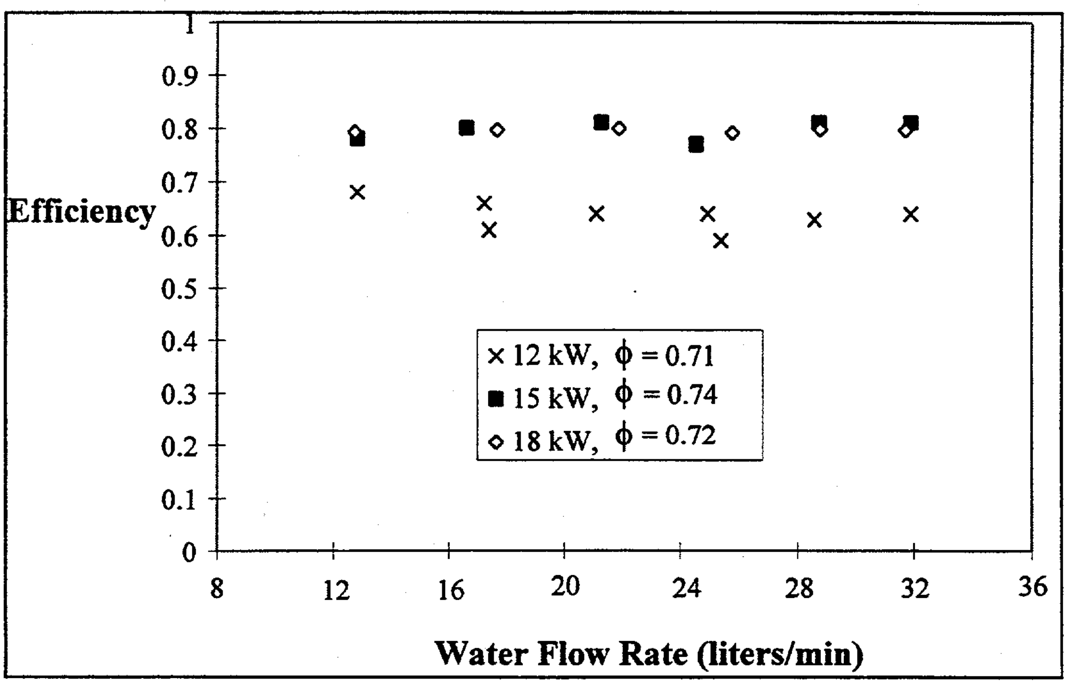

Figure 5: Efficiency versus water flow rate with constant power and ϕ

The data in Figure 6 address whether or not the thermal efficiency is compromised by burning lean. This concern stems from the fact that lean flames have lower flame temperatures that can affect heat transfer. Both the 15 and 18 kW (50,000 and 60,000 Btu/h) runs reached thermal efficiencies of 80% for $\phi > 0.75$ with only a slight decrease towards $\phi < 0.7$. Again, a drop-off in efficiency for the 12 kW runs is found.

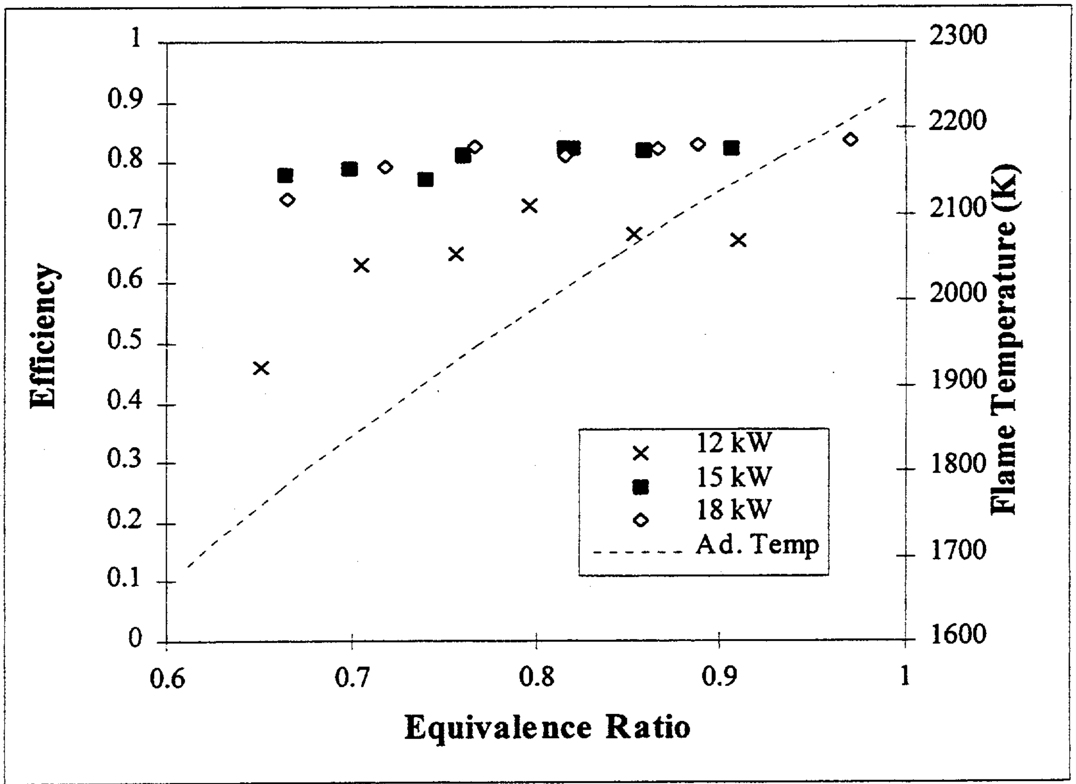

Figure 6: Efficiency versus ϕ for constant power and water flow rates

The weak dependency of efficiency on ϕ can be explained by the fact that 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 present 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 convective 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 this system to maintain the high thermal efficiencies found in current Telstar units.

As the previous figures have shown, the thermal efficiencies at powers of 15 kW and above are comparable to that of the Telstar spa heater. More importantly, the WSB delivers these efficiencies in the lean-burning range of $\phi < 1.0$ where it is expected that the NO emissions would be low. Figure 7 shows the NO concentrations (corrected to 3% O_2) of the WSB/Telstar system for 15 and 18 kW.

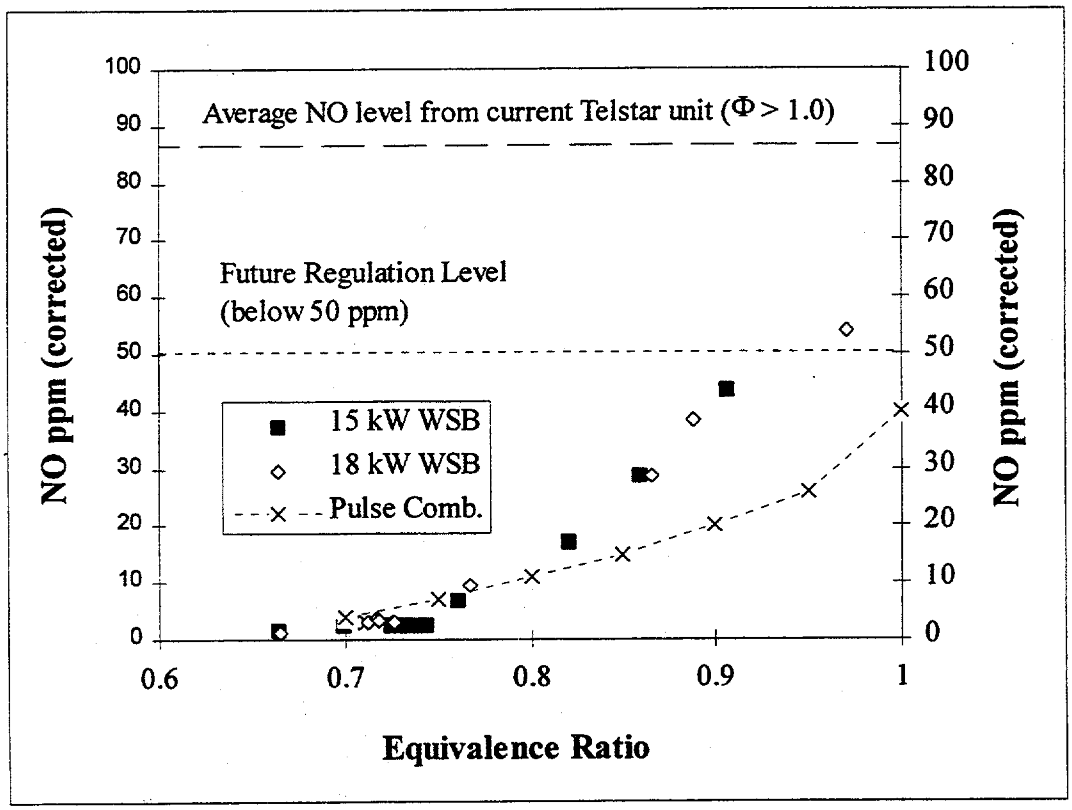

Figure 7: NO concentrations for the WSB/Telstar unit

Also shown in Figure 7 are two references: the proposed 50 ppm regulation and the average NO concentration produced by the unmodified Telstar unit that uses a conventional rack burner. As can be seen, the unmodified Telstar spa heater has NO emissions averaged 86 ppm. The use of the WSB reduces NO emissions down to 55 ppm at $\phi = 0.95$ and even further down to 4 ppm for $\phi < 0.75$. These results demonstrate that the WSB/Telstar unit will meet future NO_x regulations. The "low" (< 50 ppm) NO emissions can be obtained with the WSB for $\phi < 0.95$, and the "ultra-low" (< 25 ppm) levels can be achieved under $\phi < 0.85$. The NO concentrations also compare favorably with the recent results of Keller et al. [6] obtained from a laboratory pulse combustor utilizing exhaust gas recirculation.

Unlike NO concentrations, which decrease with ϕ, CO concentrations increase. Thus, ϕ needs to be chosen to achieve a balance between NO and CO emissions. Figure 8 shows concurrent NO and CO measurements for 15 and 18 kW with water flow rate fixed at 16 liters/min.

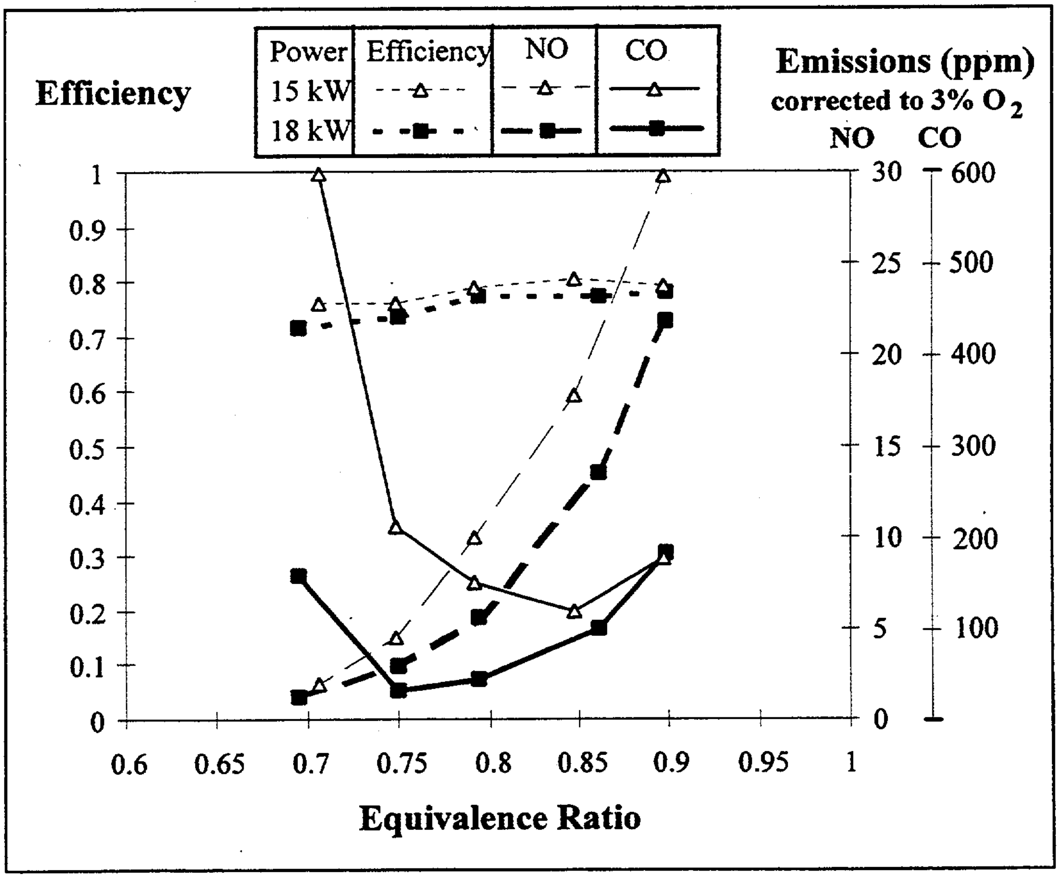

Figure 8: NO/CO emissions from the WSB/Telstar unit.

As expected, NO concentrations follow the same trend as in Figure 7, all falling below 35 ppm. CO concentrations also fall as ϕ decreases from stochiometry ($\phi = 1.0$). They reach a minimum between $0.75 < \phi < 0.85$, and then begins to increase sharply as ϕ drops below 0.75. This upturn is due to incomplete combustion and is often considered as a deterrence to developing lean burners. However, our results show that there is an optimal operating range between $0.75 < \phi < 0.85$ for 18 kW, where NO_x concentrations are below 25 ppm and CO levels are about 50 ppm. At 15 kW, both NO_x and CO levels at $0.75 < \phi < 0.85$ are higher than at 18 kW. These results, and the earlier results showing a drop-off in efficiency at low power, suggest that the current WSB/Telstar system operates better when powers are above 15 kW. Our results also demonstrate that the use of WSB can overcome the problem associated with reduced thermal efficiencies under lean-burn. Therefore the only design criterion important for its application to commercial product is the equivalence ratio at which both NO and CO are minimized.

SUMMARY AND CONCLUSIONS

Our investigations have shown that the maximum and minimum swirl requirements (corresponding to flashback and blowoff) of the WSB are independent of enclosure and power levels. The minimum swirl requirement is found to increase with exit tube length ℓ. A burner with $\ell = 7$ cm was chosen for the laboratory test station. This test station consisted of a Telstar heat exchanger mounted over a WSB with all the flow rates, temperature and emissions monitored by a PC. It achieved thermal efficiencies of 80%, comparable to a current model water heater under similar power (15 - 18 kW) and water flow rate (16 liter/min.). The thermal efficiency is only weakly dependent on equivalence ratio. Most importantly, "ultra-low" NO concentrations of less than 25 ppm were observed for a broad range of equivalence ratios and power levels. The optimal operating range for 18 kW was between $0.75 < \phi < 0.85$ where NO < 25 ppm and CO < 50 ppm (corrected for 3% O_2). These concentrations are well below foreseeable regulations limits. This demonstration project has shown that the WSB is suitable for use in domestic low NO_X emitting water heaters.

This work was supported by the Department of Energy, Energy Research Laboratory Technology Transfer through a Cooperative Research and Development Agreement with Teledyne Laars, Moorpark, California. The authors would like to thank Mr. Gary Hubbard for developing the computer software, Mr. Ken Hom for fabricating the burner, and Mr. Miguel "Alex" Bracchini for assisting in the stability limit experiments.

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