
18 Multi-Burner Boiler Applications

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18.1 BOILER-SPECIFIC BURNER REQUIREMENTS

The requirements for furnaces and burners of gas/oil-fired utilities and industrial boilers are based on long-term operational experience and comprehensive testing of different new designs and retrofits of boiler equipment. These requirements are changing with the times. [Figure 18.1](#) shows typical utility boilers. A typical single-burner industrial boiler is shown in [Figure 18.2](#).



FIGURE 18.1 Typical utility boilers.

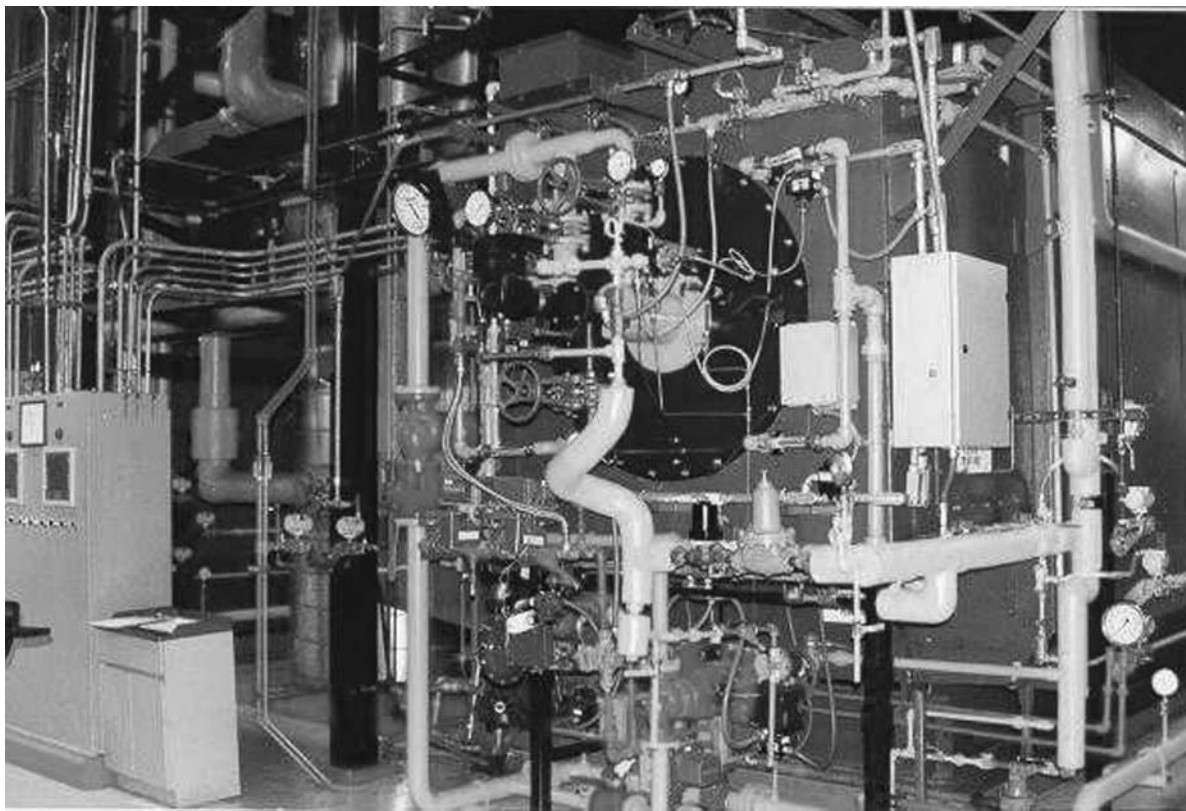


FIGURE 18.2 Typical single-burner industrial boiler.

Prior to the establishment of NO_x emission rate control requirements in the United States and other countries, performance data for furnace and burner design of utility and industrial boilers usually had to match the following common accepted requirements:

- High reliability during long-term operation
- Simplicity and reliability of gas and oil fuel ignition
- High flame stability while firing either gas or oil fuels, even at full turndown
- Provision for designed superheated and reheated steam temperatures while firing either gas or oil fuels at full turndown during long-term operation
- High thermal efficiency and low concentrations of incomplete combustion products (ICPs) while firing either gas or oil fuels using comparatively low excess air, even at full turndown
- Low combustion air system pressure drop, especially the burner register draft loss
- Simplicity of burner/windbox maintenance and adjustment
- Ease of automatic mode operation and fuel change-over between gas and oil
- Provision of restricted flame dimensions to match the dimensions of an existing or newly designed furnace
- Allowing operation with no flame impingement on either the target or side walls of the furnace that promoted reliability of all high-temperature heat exchange surfaces — especially water-wall, superheater, and reheater tubes.

To match these requirements for any boiler application, it was necessary to carefully review how a retrofit burner design should be implemented for each particular retrofit furnace. Prior to the establishment of NO_x emission rate control requirements, burner performance was typically evaluated as a function of ICP concentration levels at low excess air conditions. While firing either oil or gas fuels in preheated air, optimal low excess air burner designs provided carbon monoxide (CO), hydrogen (H₂), and unburned hydrocarbon concentrations of less than 200 ppm at excess oxygen (O₂) levels ranging between 0.2 and 0.6% at full load, as well as O₂ levels less than 1% over a 3:1 turndown range. In many European countries, such as Germany, France, Italy, and Belgium, the low excess air gas/oil burners acceptable condition was considered to be CO less than 1000 ppm and opacity less than 6%. The opacity numbers were measured using the Ringelman or Baharac method.

In some countries, the burner performance was evaluated specific to each boiler design. For example, in Russia, all industrial and relatively small utility boilers with steam flow capacities of up to 220,000 lb/hr (100,000 kg/hr) with preheated air were designed with up to six burners, each with rated heat inputs of up to 170×10^6 Btu/hr (50 MW_l). Larger utility boilers having capacities from 440,000 to 1,800,000 lb/hr (200,000 to 820,000 kg/hr) superheated steam flow were designed with up to eight burners, each with rated heat inputs of up to 400×10^6 Btu/hr (120 MW_l). All ICP concentrations in the flue gas, including both CO and H₂, were required to be less than 1000 ppm at O₂ levels less than 1%. To obtain lower ICP values, burner air velocities were increased to at least 160 ft/sec (50 m/sec), with 300 ft/sec (90 m/sec) preferred.

Until recently, there were two operational parameters used to regulate the combustion processes in boilers to achieve the required operational characteristics: (1) the variation of fuel and air input, and (2) the regulation of reheated/superheated steam temperature. The many years of operating this way resulted in significant difficulties such as:

- Reduction of reliability due to frequent failures of high temperature heat exchange surfaces
- Loss of reheated/superheated steam temperature during full turndown operation
- Overfiring of the unit

Experience has shown that controlling the combustion processes to allow maintenance of key operational parameters within a given range, including superheated and reheated steam temperatures, best solved these problems. Combustion control yielded the most efficient and reliable boiler operation, independent of load, fuel type, or other conditions.

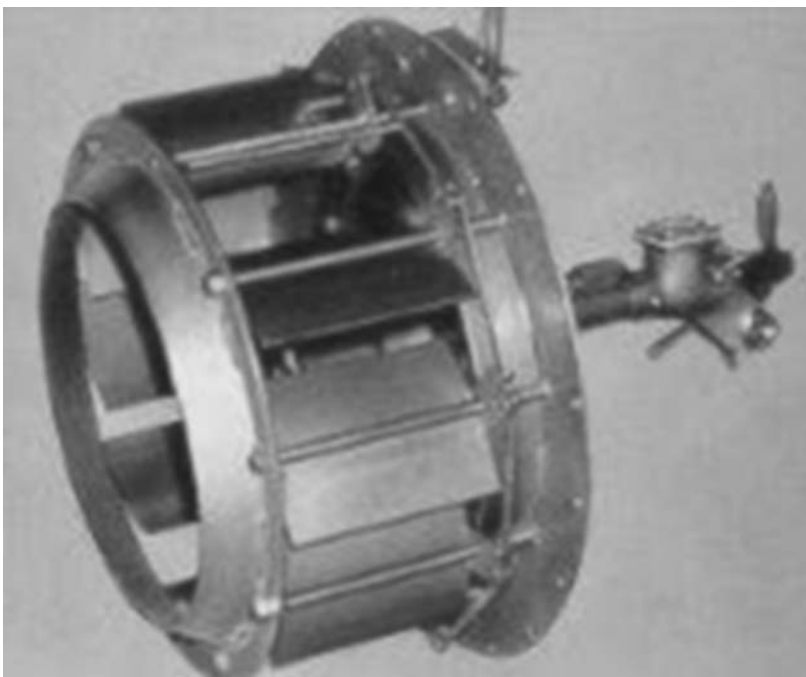


FIGURE 18.3 Swirl burner.

18.1.1 CONVENTIONAL BURNER TECHNOLOGY FOR BOILERS

Older generations of burners, such as the register, or swirl burner shown in [Figure 18.3](#), were considered very reliable and were the staple of the industry for many years. They consisted of several main parts, including a diffuser, air doors, a throat ring, and a fuel supply system. They operated on a principle of precisely controlling the swirl of the burner by adjusting the air doors, either open or closed. In an ideal situation, each burner in a multi-burner windbox should receive an equal amount of air mass flow. However, in the real world, the burner-to-burner mass flow distribution varied up to $\pm 30\%$ of the average mass flow. The typical compensation for deviations in mass flow was to close down on the air doors of the burners receiving too much air. This, in turn, affected the swirl of the burner, taking it away from the optimal single-burner performance setting.

Flame length, which depended largely on heat input, was the critical burner design parameter during this time period. Flame impingement on either the target or side walls was not allowed. [Figure 18.4](#) shows the average flame length for natural gas and #6 oil as a function of burner heat input in an experimental furnace. Other parameters included preheated air of $\sim 570^\circ\text{F}$ (300°C) and O_2 levels less than 1%. The oil was mechanically atomized at ~ 280 to 300 psig (19 to 20 barg) and at 248 to 266°F (120 to 130°C).

Conventional gas and gas/oil burners usually produce flames with fixed parameters. However, the combustion processes can be modified by varying the aerodynamic and chemical structure of flows fed into the furnace via burners. In this case, in addition to the typical requirements listed above, the burner design should provide the following possibilities:

- Changing the direction or reversing the fuel/air mixture swirl to cause interaction of the flame vortices, which simplifies the problem of increasing the furnace wall heat absorption rate when switching from oil to gas while retaining the same O_2 levels in the furnace
- Changing the swirl intensity, flame length, flow density, and static pressure across the furnace cross section, as well as along the length of the combustion zone

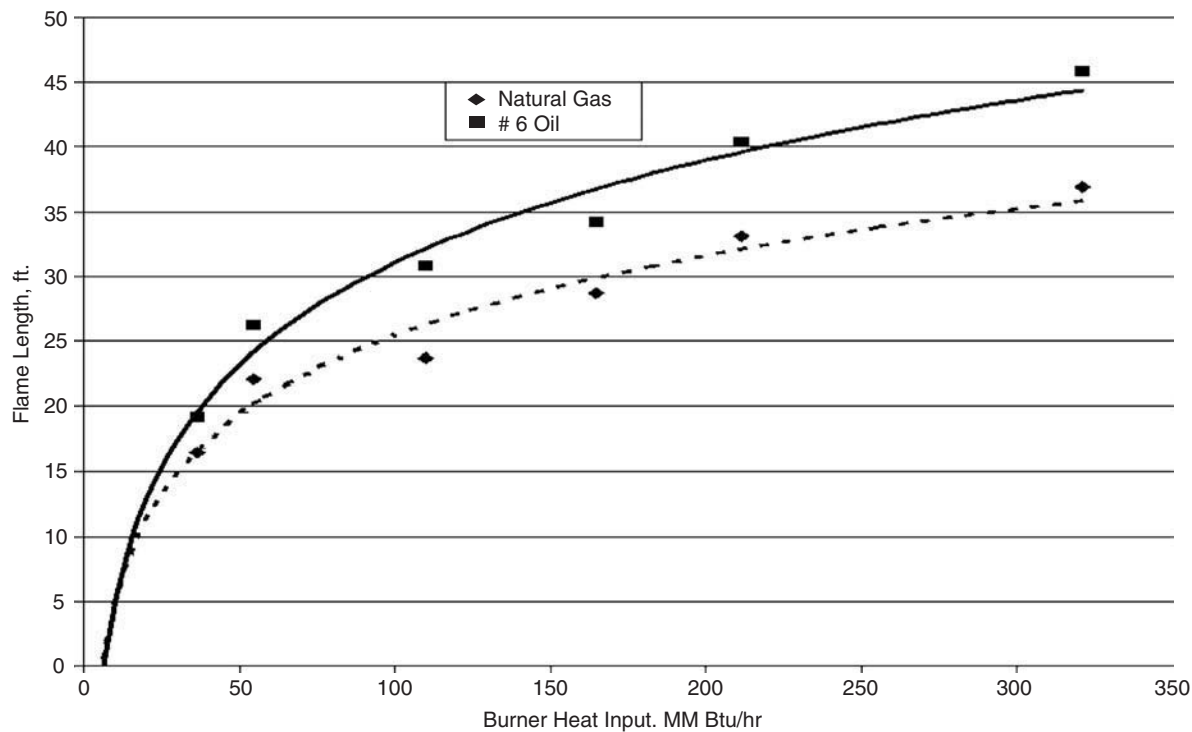


FIGURE 18.4 Average flame length as a function of burner heat input.

- Varying the flue gas recirculation (FGR) rates to control the primary firing zone to enable both stable combustion and low emission formation

Regulating not only the aerodynamic properties of the flame, but also the chemical structure and emission properties of the gas and oil combustion products, can further increase the effect of burners on combustion processes. The chemical structure and emission properties of a flame can be varied using controlled fuel input.

18.1.2 LOW-NO_x BURNER TECHNOLOGY FOR BOILERS

The objective of the modern typical boiler burner retrofit is to reduce NO_x. Two NO_x reduction techniques for boilers are combustion modification and back-end (post-treatment) clean-up. Combustion modifications include low-NO_x burners, low excess air, over-fired air, flue gas recirculation (FGR), fuel-induced recirculation, reburn, and water/steam tempering. Back-end clean-up techniques include selective catalytic reduction (SCR) and selective noncatalytic reduction (SNCR). This chapter focuses on the combustion modification techniques of low-NO_x burners, low excess air, over-fired air, flue gas recirculation, and fuel-induced recirculation.

When the U.S. and other countries mandated NO_x control, in addition to the requirements listed in the previous section, this new requirement became the dictating feature that determined the quality of burner/furnace performance. As a result, many low-NO_x burners, designed for a plethora of different boiler design and heat input capacities, firing gas and/or oil with preheated and ambient air, were developed in the U.S. and many other countries.

The situation is complicated by numerous contradictions between the operational baseline conditions for the existing boiler on the one hand, and the new NO_x requirements on the other hand. For example, all NO_x reduction combustion methods will increase flame length and redistribute the heat flux between radiant and convective heating surfaces. However, the flame length is restricted by the current furnace dimensions. Also, increasing heat flux in the superheater/reheater surfaces is not allowed because it will degrade the lifetime of these surfaces. Decreasing the total area of these surfaces increases their reliability, but reduces the boiler thermal efficiency.

Another contradiction is related to the desire to get simultaneous NO_x reduction and reduction of incomplete combustion products (ICPs). Lowering O₂ levels will decrease NO_x, but it will increase the ICPs. To reduce ICP concentrations, O₂ levels must be increased and fuel/air mixing must be improved. However, these actions will tend to increase NO_x. Solving these problems simultaneously is very difficult for both existing and new design furnaces, especially on utility boilers.

There are other complications on dual-fuel boilers. It is necessary to provide the designed reheated and superheated steam temperatures over long-term operation while firing fuels generating different flame properties over the full turndown load range. The luminescence of an oil flame is significantly higher than that of a gas flame (see Chapter 6). Assuming all other conditions are the same, furnace heat absorption is less and the furnace exit gas temperature is higher while firing gas. Higher exit gas temperatures create conditions for more intense thermal NO_x generation in all industrial boilers and most utility applications for 50 to 150 MW_e power units. On larger-capacity utility boilers, the NO_x is higher while firing gas than while firing oil. A more detailed explanation is given in Chapter 18.3.2.3. Typically, the NO_x while firing oil is higher due to fuel-bound nitrogen (N_p). Fuel-bound nitrogen usually ranges from 0.2 to 0.45% wt., but can be as high as 0.7 to 1% in #6 oil. Because the effectiveness of various NO_x reduction methods is different with gas and oil firing, it makes sense, in principle, to implement different NO_x reduction methods for each fuel. However, doing so will complicate boiler operations so much that such designs would not be acceptable to the customer. Attempts to implement NO_x reduction methods for both fuels create situations, where different methods are selected for the same boiler design installed at different power plants. The method chosen depends on many operational parameters, such as annual consumption and seasonal distribution of each fuel, local climate, annual average load, levels and frequency of load peaks, and stack height.

Today's low-NO_x burner relies on control of the combustion air in several component streams, as well as the controlled injection of fuel into the air streams at selected points for maintaining stable, attached flames with low NO_x generation. A typical venturi-style low-NO_x burner is shown in Figure 18.5 and Figure 18.6. Primary and secondary air enters the burner radially through the venturi and exits the burner axially with a primary air swirl defined by the fixed blade axial swirler. The swirler determines the size and strength of the recirculation zone. A tertiary air stream flows between the venturi base and the burner throat quarl. Tertiary air separates some of the combustion air from the main flame, effectively staging combustion and reducing NO_x. For natural gas firing, fuel can be introduced through internal or external pokers or gas rings and can also be injected through a central gas pipe with multiple orifices at the furnace end. A single, conventional atomized burner (oil gun) located along the burner centerline typically supplies the oil. The oil gun may use dual fluid, mechanical, or rotary cup atomization.

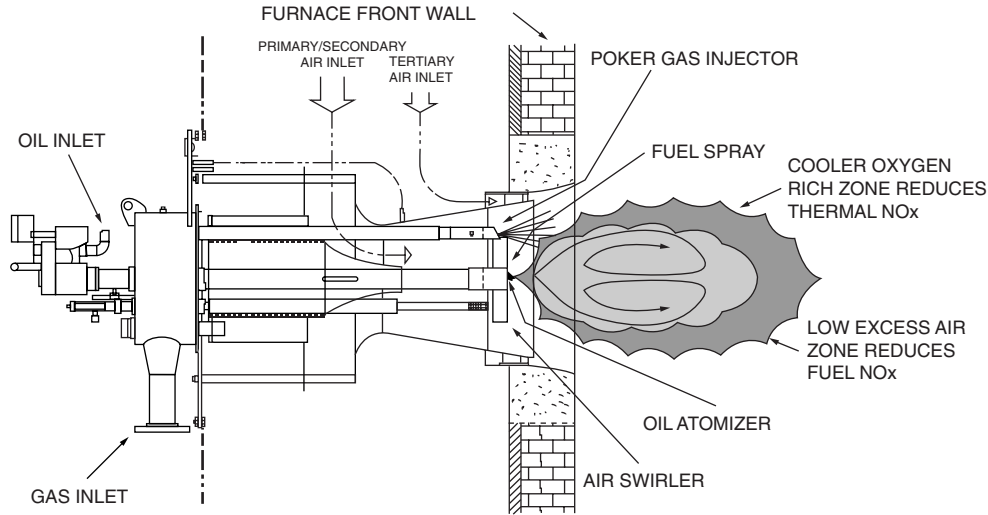


FIGURE 18.5 A typical low NO_x burner: venturi style.

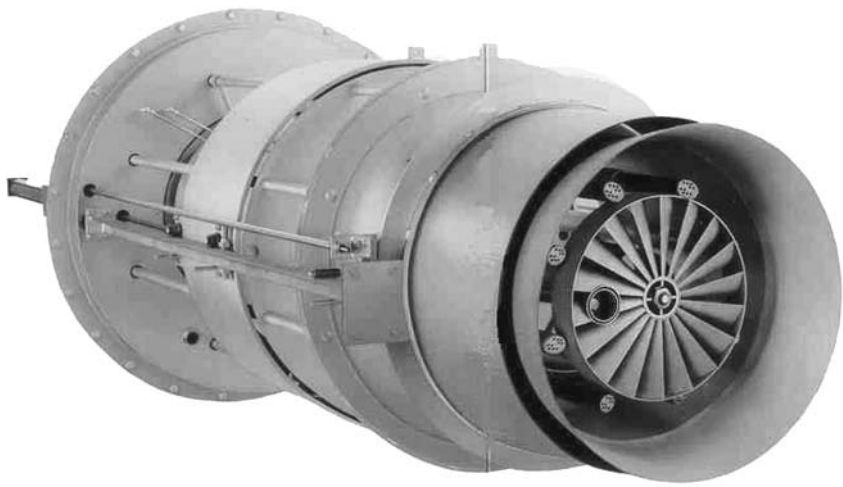


FIGURE 18.6 A typical low-NO_x burner: a venturi style (second example).

18.1.3 STAGED BURNER DESIGN PHILOSOPHY

Most of today's low-NO_x burner designs implement staged combustion principles as an effective way to reduce NO_x. The staged burner should also be designed to provide the maximum degree of flexibility in achieving high burner turndown, low NO_x, and improved flame shaping capability. The design basis of a staged burner is to develop a stratified flame structure with specific sections of the flame operating fuel-rich and other sections operating fuel-lean. The burner design thus provides for the internal staging of the flame to achieve NO_x reductions while maintaining a stable flame.

Controlling combustion stoichiometry to fuel-rich conditions inhibits NO_x production, especially in the burner's flame front. Operating the flame fuel-rich also reduces the burner NO_x dependence on the burner zone heat release (BZHR) rate, which is discussed in Section 18.2.1. This is especially important in applications where a very high BZHR near full load can result in an exponential increase in NO_x.

In addition to controlling NO_x formation, operating under fuel-rich conditions will aid in the production of combustion intermediates that can result in the destruction of previously formed NO_x. In a reducing environment, NO can act as an oxidizer to react with these combustion intermediates, resulting in the reduction of NO to N₂. Therefore, NO, necessarily formed to satisfy the requirements of establishing a strong flame front, can be scavenged by the process and reduced to N₂.

To achieve complete fuel burnout at minimum O₂, the burner design must provide direct interaction of fuel-lean zones with the center fuel-rich sections. This ensures that the "rich" products of combustion from the center flame pass through the oxidizing zone for complete fuel burnout. The burner design allows control of the stoichiometry of the oxidizing zone to range from being pure air to having varying degrees of excess oxygen. This controls NO_x formation by causing the fuel burnout to occur in the form of a premixed flame rather than a diffusion flame. The operational flexibility is provided in the burner design to control stoichiometry to achieve minimum NO_x.

Flame stability should also be a design focus for a staged burner. The stability criterion is to maintain a minimum turndown capability on a gas firing ratio of 8:1 over a broad range of excess O₂. To achieve this objective, it is necessary to carefully select the burner throat velocity and the velocity profile.

The burner flame front should be a fuel-rich premixed flame designed to increase flame speed and, in turn, flame stability. Combustion stoichiometry, approaching a perfect premixed flame, is established at the burner flame front. This condition broadens the flammability range of the fuel and ensures the maintenance of a flammable mixture over a broad range of burner throat velocities and fuel injection rates.

Controlling air distribution within the burner throat improves flame-shaping capability. Provisions are made in the burner design to allow the burner to operate as a low excess air burner as well as a low-NO_x burner with an extended flame envelope.

18.1.4 DESIGN FEATURES OF LOW-NO_x BURNERS

Some of the standard design features of low NO_x burners include:

- Flame stability at low excess air rates for reliable, energy-efficient boiler operation.
- High turndown ratios for wide-range boiler operation.
- Well-distributed air flow to control the flame envelope and provide even heat flux.
- Known flame length and diameter to suit furnace firing lane without impinging on boiler tubes or furnace walls.
- Elimination of combustion-induced vibrations of the boiler, due to the aerodynamics of the register design and turbine blade diffuser.

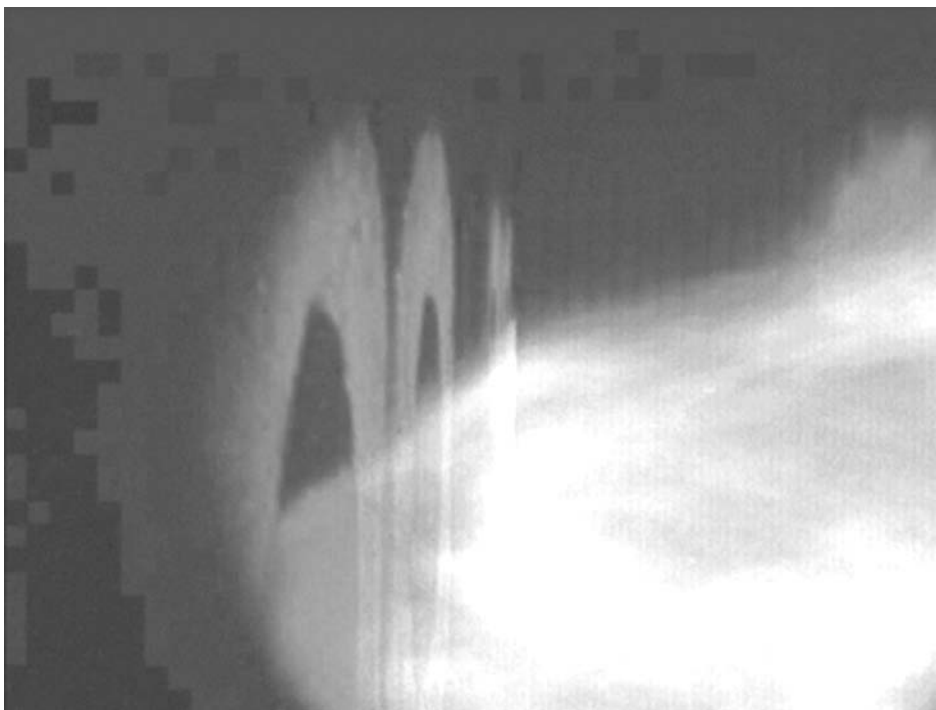


FIGURE 18.7 A strong flame front established within a maximum of 0.5 diffuser diameters off the face of the diffuser.

- A strong flame front established within a maximum of 0.5 diffuser diameters of the face of the diffuser, as depicted in [Figure 18.7](#). The flame front should not move during changes in the firing rate, thus providing a stable flame for scanning, resulting in reliable operation.
- For gaseous fuel firing, a multiple-poker gas burner assembly with “poker shoes” (gas nozzles) oriented to provide internal fuel staging.

18.1.5 EFFECTS OF BURNER RETROFITS ON BOILER PERFORMANCE

Low-NO_x burners have different flame characteristics from their predecessors, the low excess air burners. Therefore, the probability exists that the differences will affect boiler performance. Performance changes would only occur by an alteration of the heat absorption pattern in the furnace, thus affecting the amount of heat going to the boiler backpass. The overall furnace heat absorption could either increase or decrease due to factors such as flame distance to walls, flame temperatures, flame emissivity, and characteristics of ash deposits on furnace walls. The impact on boiler performance will vary by unit and by fuel. In some cases, there will be no impact; but, in most cases, there will be either a positive or negative impact experienced. Historically, however, there has been no net effect on boiler performance that would be considered extreme. Atomizer tip design can drastically impact boiler performance, but total boiler performance is always a balancing act between key variables such as NO_x and water wall temperatures.

To assess the impact on boiler performance, certain information is typically monitored during the testing of low-NO_x burners. Important data includes superheated and reheated steam temperatures, superheater and reheater tube surface temperatures, and any operational parameters that affect or control steam temperatures. The boiler performance variation resulting from a change in overall O₂ levels is not considered in this analysis. The actual impacts on boiler efficiencies are relatively small and there are many complicating factors, including furnace and heat recovery area cleanliness, fuel composition, sootblower availability, operational variability, and others.

18.2 BOILER DESIGN IMPACTS ON NO_x EMISSIONS CORRELATIONS

The two major mechanisms for the formation of NO_x are: (1) NO_x formed by the oxidation of N_f (fuel NO_x) and (2) the thermal fixation of atmospheric nitrogen (thermal NO_x). Separate NO_x correlations have been developed for each. These correlations can be combined to predict the total NO_x emissions for a selected burner design based on the fuel nitrogen content and boiler design.

The conversion of N_f to NO_x depends on oxygen availability and N_f content. Thus, for a given burner system at constant excess air, the key variable controlling fuel NO_x formation is N_f content. Previous studies have shown that the N_f conversion efficiency is inversely proportional to the nitrogen content of the fuel. High conversion efficiencies are observed with low N_f, and low efficiencies are seen with high N_f.

18.2.1 BOILER DESIGN

Based on the strong dependency of thermal NO_x on flame zone temperature, thermal NO_x formation for wall-fired boilers has been correlated with the ratio of heat input to furnace size and the number of firing walls. For this correlation, the ratio of heat input to the furnace burner zone area is defined as the burner zone heat release (BZHR) rate. The BZHR represents the boiler heat release rate divided by the water-cooled surface area in the burner zone (10⁶ Btu/hr-ft²), and is a measure of the “bulk furnace temperature.” The NO_x created within this zone depends on this bulk temperature by an exponential relationship, as discussed in previous chapters. The burner zone is defined as the six-sided surface bounded by the furnace walls and imaginary horizontal planes located one burner row spacing above the top row and below the bottom row of burners. The area of any division walls within this volume is also included in the calculation of the BZHR. A correction is made for re-radiation from any refractory on the floor, walls, or the burner throats. An example of the effects of boiler design on NO_x is shown in [Figure 18.8](#), where the same heat input is placed in two drastically different-sized boilers.

The correlation between thermal NO_x and BZHR rate has been developed using an extensive database of gas- and oil-fired utility boilers. [Figure 18.9](#) shows the NO_x of various boilers included in the database on oil and gas, respectively. NO_x of industrial boilers with comparable degrees of NO_x control were found to be consistent with the BZHR rate correlations.

The correlations between N_f conversion and thermal NO_x can be used to extrapolate burner NO_x from one boiler to another. The increase in NO_x observed with fuel oil compared to natural gas, which contains no nitrogen, is used to estimate fuel NO_x for the selected burner. N_f conversion is then calculated based on the nitrogen content. The correlation of N_f conversion can subsequently be used to project fuel NO_x for fuels containing differing amounts of nitrogen. Thermal NO_x is determined from NO_x with natural gas. The correlation with BZHR is used to project the measured thermal NO_x for the selected burner from one boiler to another. Total projected NO_x emissions for the burner can then be determined by adding the fuel NO_x and thermal NO_x contributions for the fuel and boiler of interest. The amount of air preheat can also have a dramatic effect on NO_x, CO, and particulate emission rates, as well as on flame stability.

18.2.2 EXCESS AIR

18.2.2.1 Theoretical Effect of Excess Air on NO_x

It is well known that thermal NO_x formation primarily depends on the flame temperature, excess air in the flame, and residence time. However, the flame temperature also depends on excess air. Demonstrating the calculated data for natural gas combustion under regular boiler conditions, with

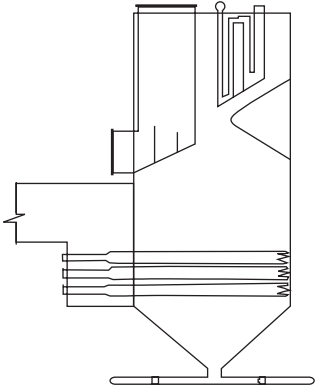
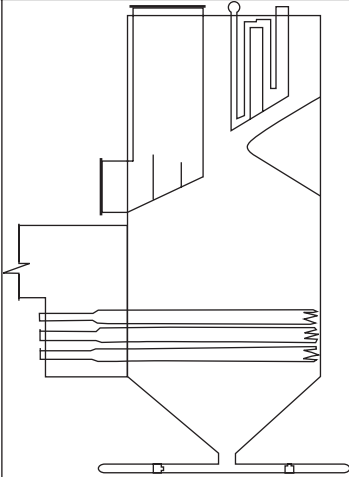
FURNACE SIZE EFFECTS ON NO _x EMISSION RATES		
UNIT DESIGN PARAMETER	HIGH HEAT RELEASE RATE	LOW/MEDIUM HEAT RELEASE RATE
FURNACE ELEVATION (SAME AS SIZE UNIT)		
BURNER ZONE VOLUME	$(3 \times \text{BURNER SPACING}) \times \text{WIDTH} \times \text{DEPTH}$	$1.66 \times (\text{HIGH HRR VOLUME})$
BURNER ZONE HEAT RELEASE RATE	88,000 BTU/Hr/ft ³	53,000 BTU/Hr/ft ³
NO _x @ MCR GAS FIRING*		
LOW EXCESS AIR BURNER	0.66 Lb/Million BTU	0.22 Lb/Million BTU
LOW NO _x BURNER	0.39 Lb/Million BTU	0.16 Lb/Million BTU

FIGURE 18.8 The effects of boiler design on NO_x.

an excess air factor α value close to unity, increased excess air causes NO_x to rise considerably; then, as excess air is further increased, the increase in NO_x slows down, reaches a maximum, and then NO_x is reduced as excess air is further increased.

According to some experimental studies, no maximum is attained and the NO_x = $f(\alpha)$ dependence approaches the stabilized section of the exponential curve. A difference between the mentioned two contradicting results brings different approaches to optimized low-NO_x combustion evaluations.

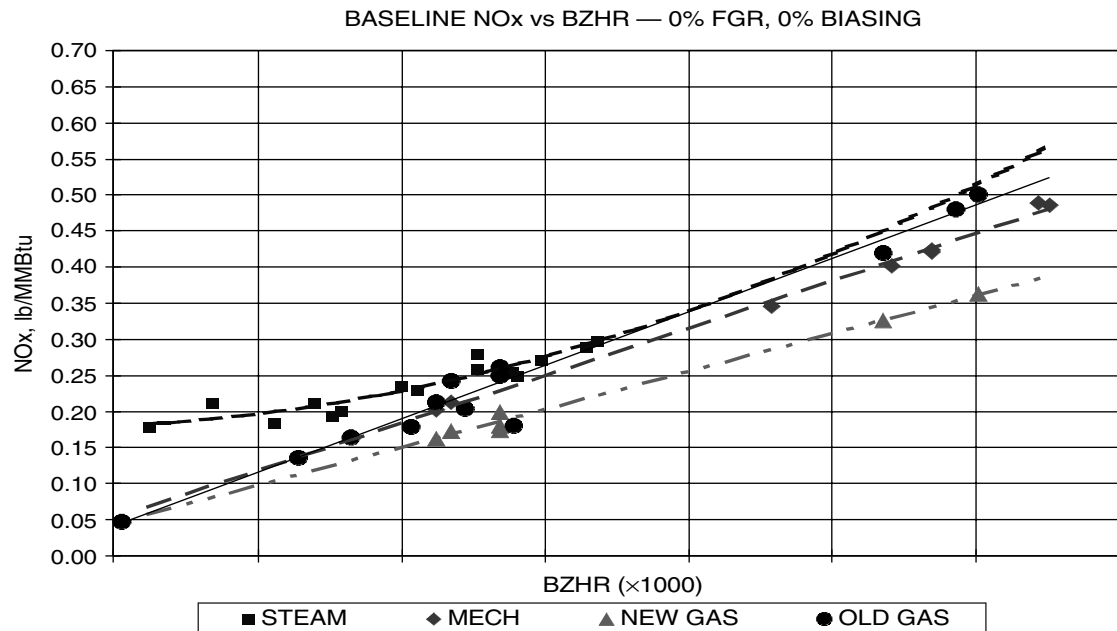


FIGURE 18.9 The NO_x of various boilers included in the database on oil and gas.

That is why it is important to discuss the existence of the NO_x maximum and to consider it in some details. For simplicity, we can consider a natural gas (containing no bound nitrogen) combustion process, as a result of which NO_x is formed only from nitrogen in the combustion air. A deficiency of oxidant, NO_x depends more on α than it does on temperature; and when $\alpha > 1$ (mostly when $\alpha > 1.2$), when the rate of combustion reactions increases, the effect of temperature proves to be the dominant one. Therefore, despite the fact that the theoretical combustion temperature in the region of $\alpha > 1$ systematically falls with increased α , the NO_x increases as long as the thermal NO_x formation is not made more difficult due to a considerable decrease in the temperature level. In this case, there is also a decrease in the NO_x in the flue gas volume due to dilution by the excess air. When firing a fuel oil that contains N_f, it is unlikely that the nature of the above-considered curve NO_x vs. O₂ will change, but in all probability the downward branch of the curve will be flatter.

When comparing the well-known empirical dependency of the ICP component concentrations on α with the empirical NO_x data as a function of α , confirmed with the theoretically calculated dependencies on α , it was found that the excess air factor α value at which a furnace's ICP losses virtually disappear, almost coincides with the values of α corresponding to the NO_x maximum. It follows that the NO_x dependence on excess air has the form of an extreme function with a maximum NO_x value corresponding to that α value at which virtually complete fuel combustion is attained under the given conditions. Hence, if NO_x and α are determined by reliable and adequately accurate methods, the absence of an experimentally established maximum on the test curve NO_x vs. O₂ (or α) leads to an assumption that the fuel is not completely burned, thus indicating the need not only to improve the combustion process, but also to check the validity of the methods used to measure ICP and NO_x in the given α range.

18.2.2.2 Empirical Evidence of the Effect of Excess Air on NO_x

Detailed empirical data sets were obtained under identical operational conditions at approximately full (~94%) load while firing natural gas and #6 oil (containing 0.22 to 0.32% N_f) with preheated air on the two neighboring 165 MW_e utility boilers of the TGM-94 model (~1,100,000 lb/hr or 500,000 kg/hr superheated steam flow), installed at the same power plant. The furnaces were balanced draft and had air-in leakage of 8 to 10%. The two boilers were almost identical except for one difference: the first boiler was equipped with 21 burners, rated at $\sim 100 \times 10^6$ Btu/hr (30 MW_t) heat input each, having a single swirled airflow channel, and the second boiler was equipped with 9 burners, rated at $\sim 230 \times 10^6$ Btu/hr (67 MW_t) heat input each, having two airflow channels, one of which is a swirled portion consisting of ~15% of the total air flow. On both boilers, the burners were installed in three rows on the front wall (3 × 7 and 3 × 3, respectively). The boilers were equipped with an FGR system designed to supply up to 14% FGR flow (isolated from the air flow), supplied directly to the furnace through slots located on the target wall, opposite the lower burner row. The test data while firing either or both fuels, both with and without FGR implementation, shown in [Figure 18.10](#) and [Figure 18.11](#), confirm well-known dependencies of NO_x concentrations on O₂.

Differences can be seen between the maximum NO_x numbers and, accordingly, between the shapes of the curves obtained from these two boilers. These discrepancies are associated with distinctions in the combustion processes related to the differences between the mentioned burner designs, heat inputs, and numbers of burners. The measured flame temperatures while firing gas under full load and all other conditions being equal show that, on the boiler equipped with nine burners, the maximum flame temperature is ~160°F (70°C) higher, resulting in a significant NO_x difference of 100 to 120 ppm (NO_x is ~20 to 24% higher than for the boiler equipped with nine burners). The increased temperature has resulted in a stronger dependency of NO_x on O₂ on the boiler equipped with nine burners, both with and without FGR, as demonstrated in [Figure 18.11](#).

The above data, as well as other test data obtained on 150- to 800-MW_e utility boilers, under similar operational conditions, illustrate a significantly lower NO_x level on #6 oil (even while containing N_f of up to ~0.7%) in comparison with NO_x numbers on gas. An explanation of this

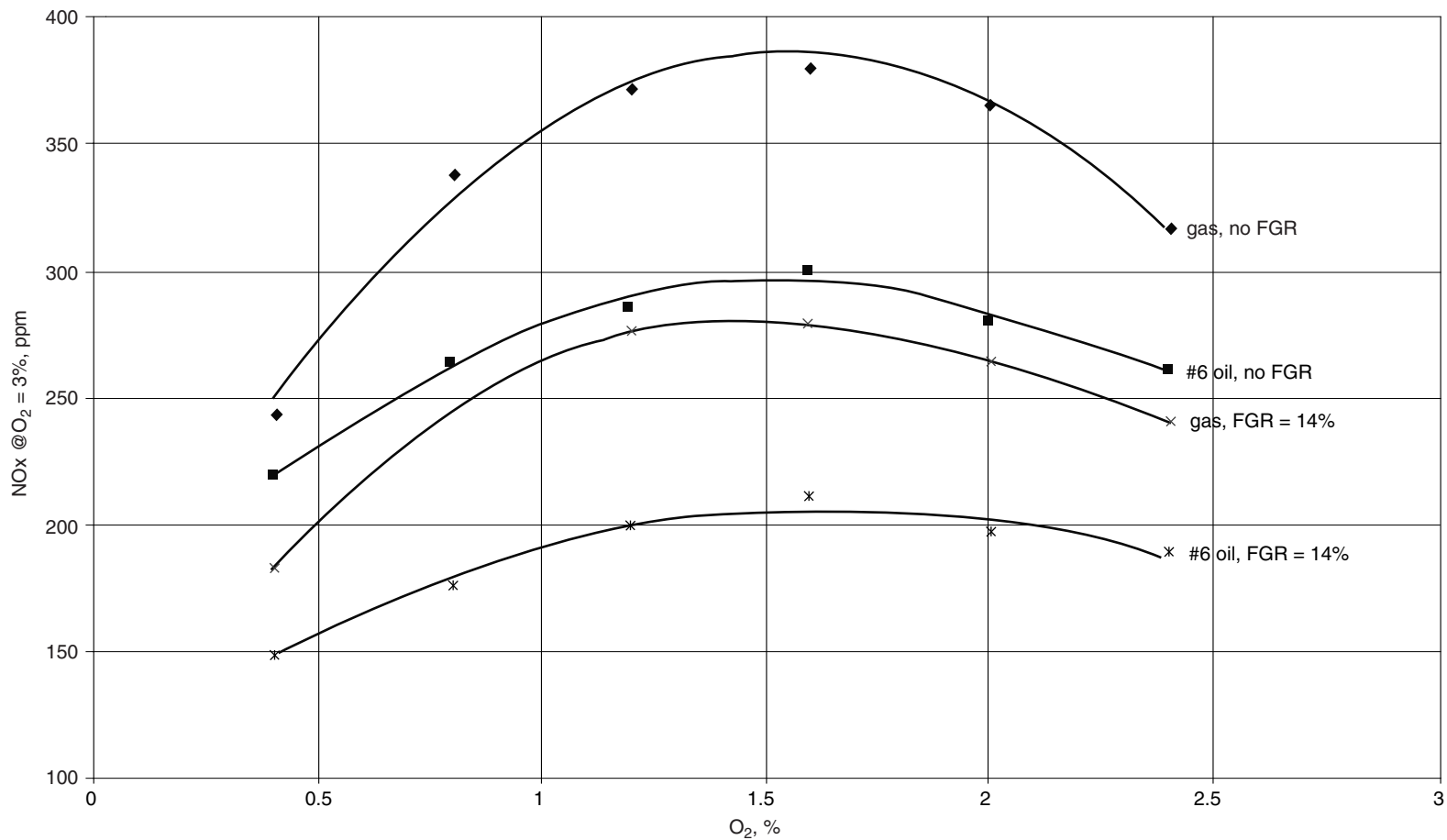


FIGURE 18.10 NO_x dependence on excess O_2 -21 burner boiler.

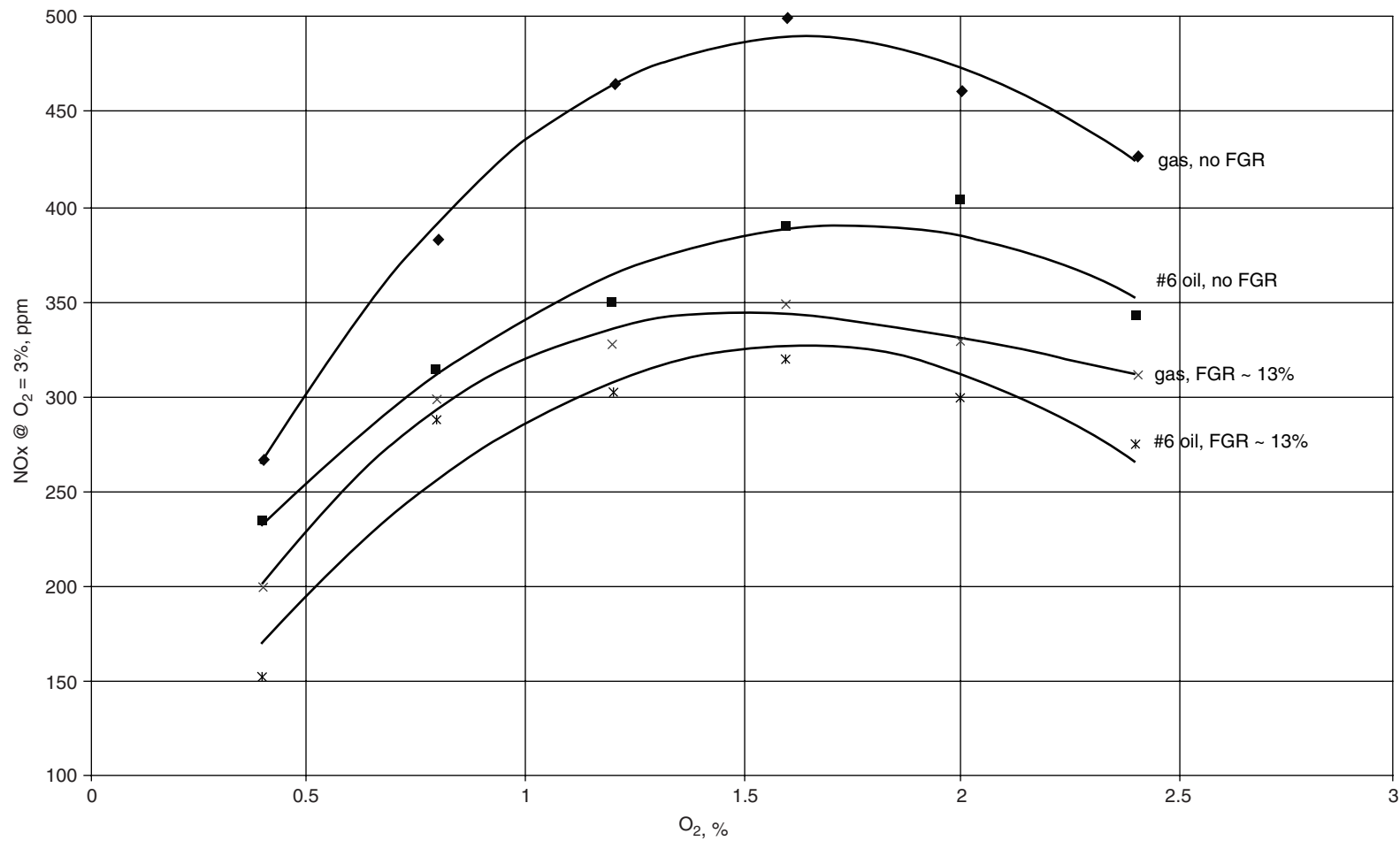


FIGURE 18.11 NOx dependence on excess O₂-9 burner boiler.

fact, contradicting the empirical data obtained on industrial and comparatively smaller utility boilers (where NO_x numbers are higher while firing oil), is presented in the Chapter 17.3.2.3.

18.2.3 BOILER LOAD INFLUENCE ON NO_x

The relationship between NO_x and load was investigated on many utility boilers over their full load ranges, while firing either or both fuels. A detailed investigation was performed on the two TGM-94 boilers described above (see Chapter 17.2.2). The test results obtained at ~94, 75, 50, and ~30% loads, with and without FGR, are shown in Figure 18.12 (gas) and Figure 18.13 (#6 oil). With O₂ levels between 1.2 and 1.6%, over the entire load range CO concentrations were in the 50- to 150-ppm range while firing gas. When firing #6 oil on both boilers, over the entire load range, under the same O₂, the CO and opacity did not exceed 100 ppm and 10%, respectively.

Figure 18.14 shows the data of the combined (load and FGR) influence on NO_x on a 200-MW_e boiler of the TGME-206 model, having no furnace air in-leakage, equipped with 12 low-NO_x venturi-style burners installed in two rows on the rear wall. The boiler was tested at loads ranging from 100 to 53% while firing natural gas at comparatively low O₂ levels (0.8 to 1%), and with 6 to 7% and 12 to 14% FGR flows. The FGR was mixed with the combustion air upstream of the windbox. Under all the above conditions, CO concentrations were less than 20 ppm.

Confirmation of these empirical dependencies on many utility boilers, equipped with various number, design, and arrangement of burners, while firing gas and oil with preheated air (500 to 700°F or 260 to 370°C), is shown in Figure 18.12 to Figure 18.14. All available data obtained on 150- to 800-MW_e boilers, firing natural gas in the load range of 100 to 30%, have been generalized in Figure 18.15 as a dependence of relative NO_x (defined as a ratio between NO_x numbers at current loads and the NO_x number at full load) on relative load. It is seen that relative NO_x is a power function of relative load in the ~1.25 degree. While firing natural gas, significant changes in the O₂ range, FGR flow level, preheated air and FGR temperatures, and other operational conditions provide comparatively small derivations from the mentioned average degree number.

With firing #6 oil under the same other operational conditions on utility boilers, in general, the power function degree number depends on N_f concentration in the oil. The higher the N_f, the lower the degree number. Corresponding test data, obtained at 17 boilers of 150 (three), 165 (eight), 210 (one), and 300 (five) MW_e utility boilers at full load with an O₂ range of 0.6 to 1.2%, are shown in Figure 18.16. With the exception of one 150-MW_e boiler, all boilers were equipped with FGR systems of various designs. Increasing FGR flow from 0 to 14% (on the 165- and 210-MW_e boilers) and to 18% (on the other boilers) influences deviations from the average test data curve but does not change the established dependence.

Similar dependencies have also been established on industrial single-burner and multi-burner boilers, while firing either gas and/or oil, with ambient and preheated air, and without staged combustion. With ambient air, the shapes of the experimental curves, established on the 22,000- to 385,000-lb/hr (10,000- to 175,000-kg/hr) steam flow boilers, with firing both gas and #6 oil (0.22 to 0.49% N_f) in the 100 to 25% load range, are similar to a linear dependence, as indicated in Figure 18.17. On preheated air (300 to 500°F or 150 to 260°C), all curves are power functions with the ~1.25 and (1.1 to 1.2) degree numbers on gas and #6 oil, respectively, as in Figure 18.18.

A clear conclusion was made based on the above empirical test data: as load decreases (as the flame temperature is reduced due to BZHR), the NO_x level also decreases. A reduction in the original NO_x level (full load) limits the potential opportunity to achieve a required NO_x reduction at lower loads, resulting in a reduction in the effectiveness of all NO_x reduction methods. This conclusion was made based on evaluations of data obtained on utility boilers, while using all known NO_x reduction methods. For example, on the TGME-206 boiler with 12 venturi burners, ~14% FGR implementation provides ~70% and ~56% NO_x reduction at 100% and 80% loads, respectively. For industrial boilers with ambient air, this effectiveness reduction is a little more pronounced due to the significantly lower flame temperature level in the industrial boiler furnaces.

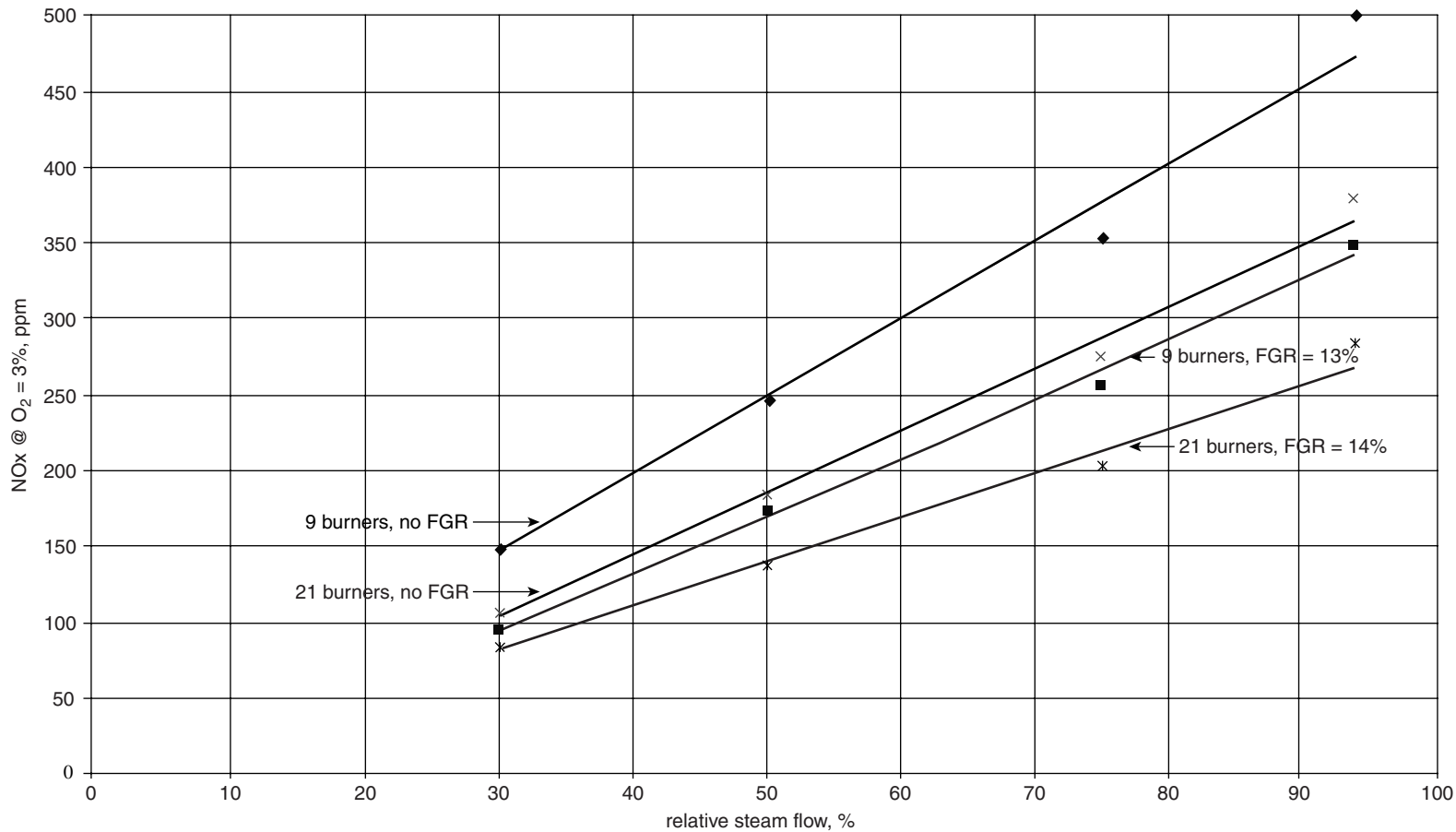


FIGURE 18.12 Test results obtained at ~94, 75, 50, and 30% loads, with and without FGR (gas).

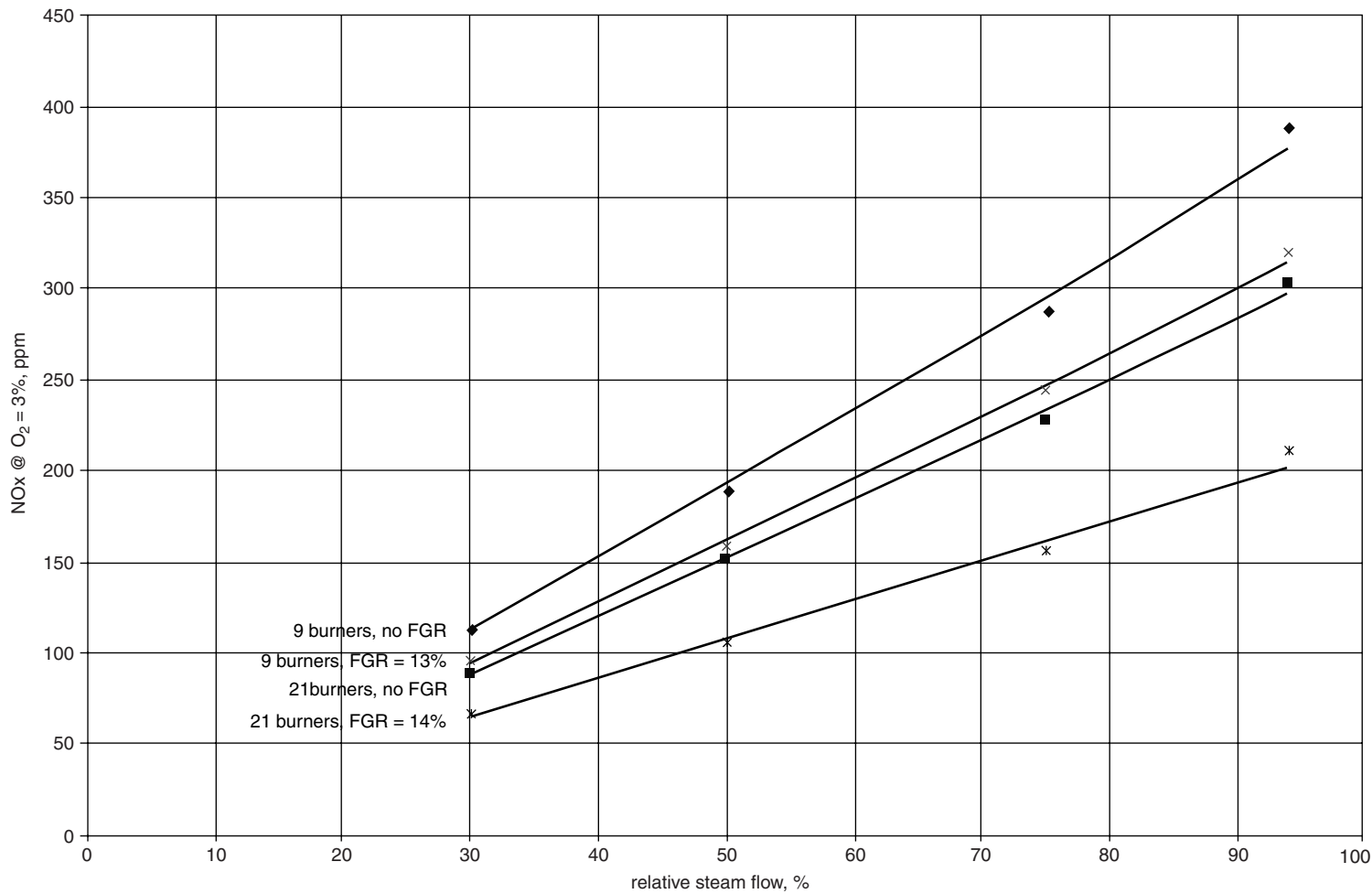


FIGURE 18.13 Test results obtained at ~94, 75, 50, and 30% loads, with and without FGR (#6 oil).

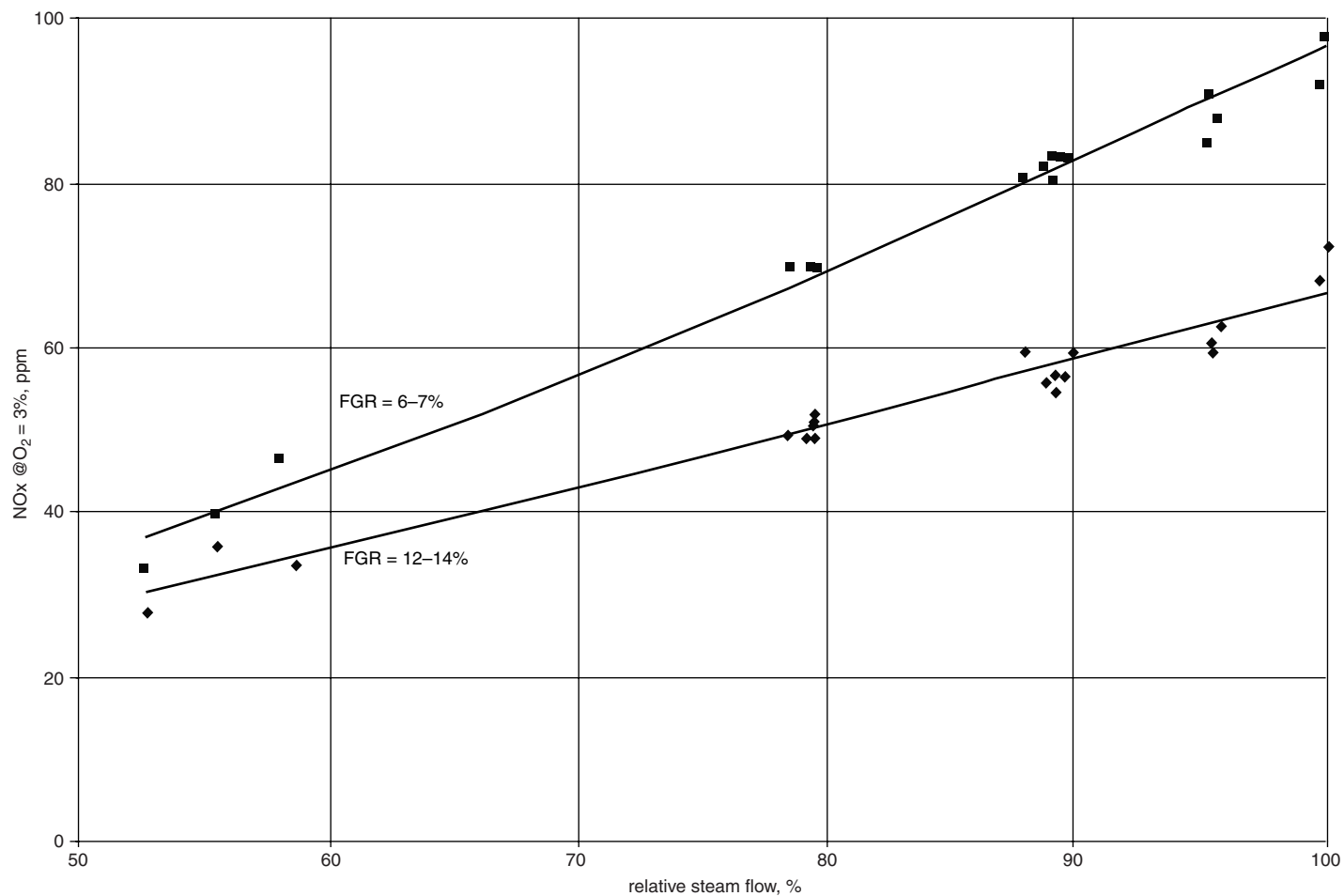


FIGURE 18.14 Combined (load and FGR) influence on NO_x on a 200-MW_e boiler of the TGME-206 model, having no furnace air-in leakage, equipped with 12 low-NO_x venturi-style burners installed in two rows on the rear wall.

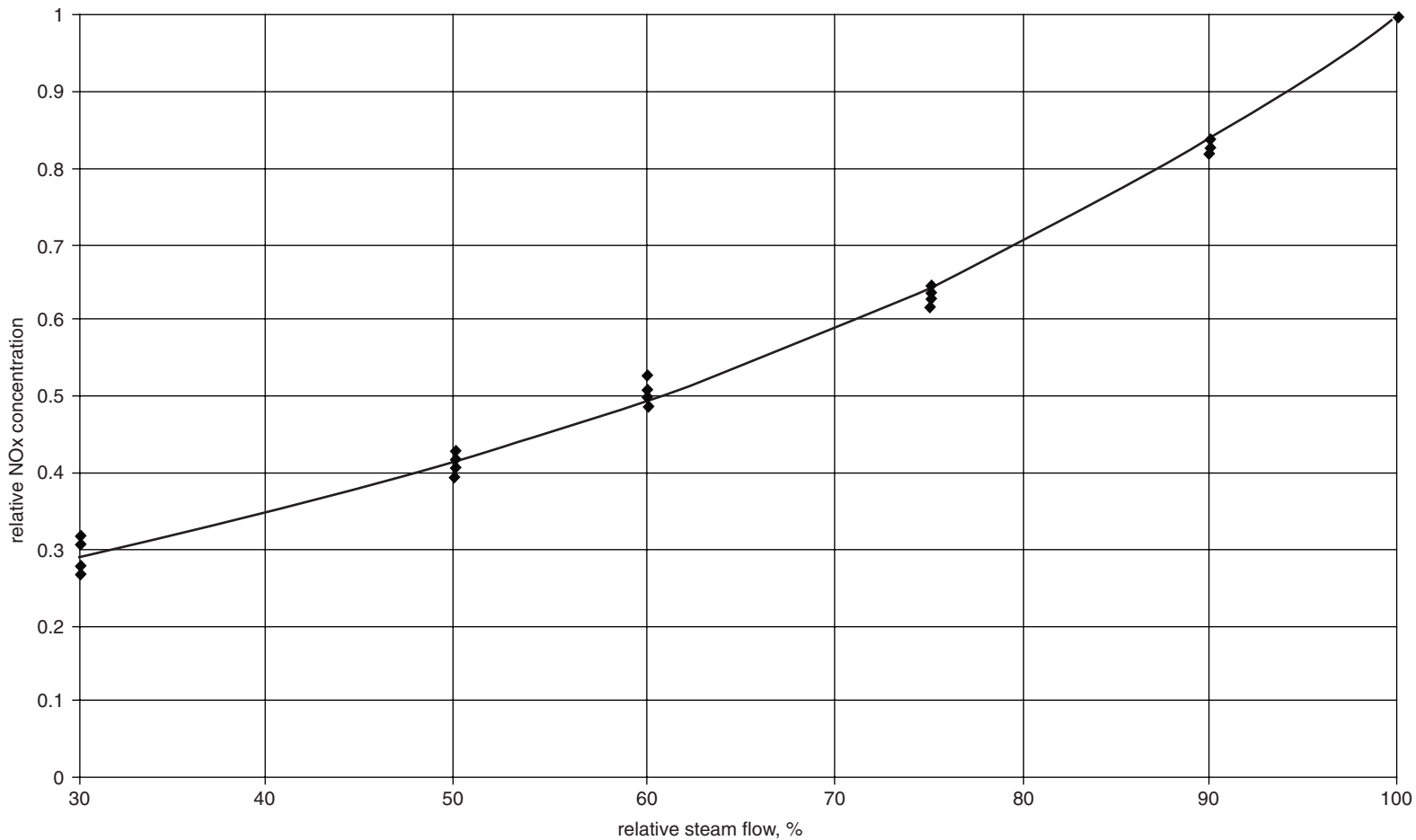


FIGURE 18.15 All available data obtained on 150- to 800-MW_e boilers, firing natural gas in the load range of 100 to 30%.

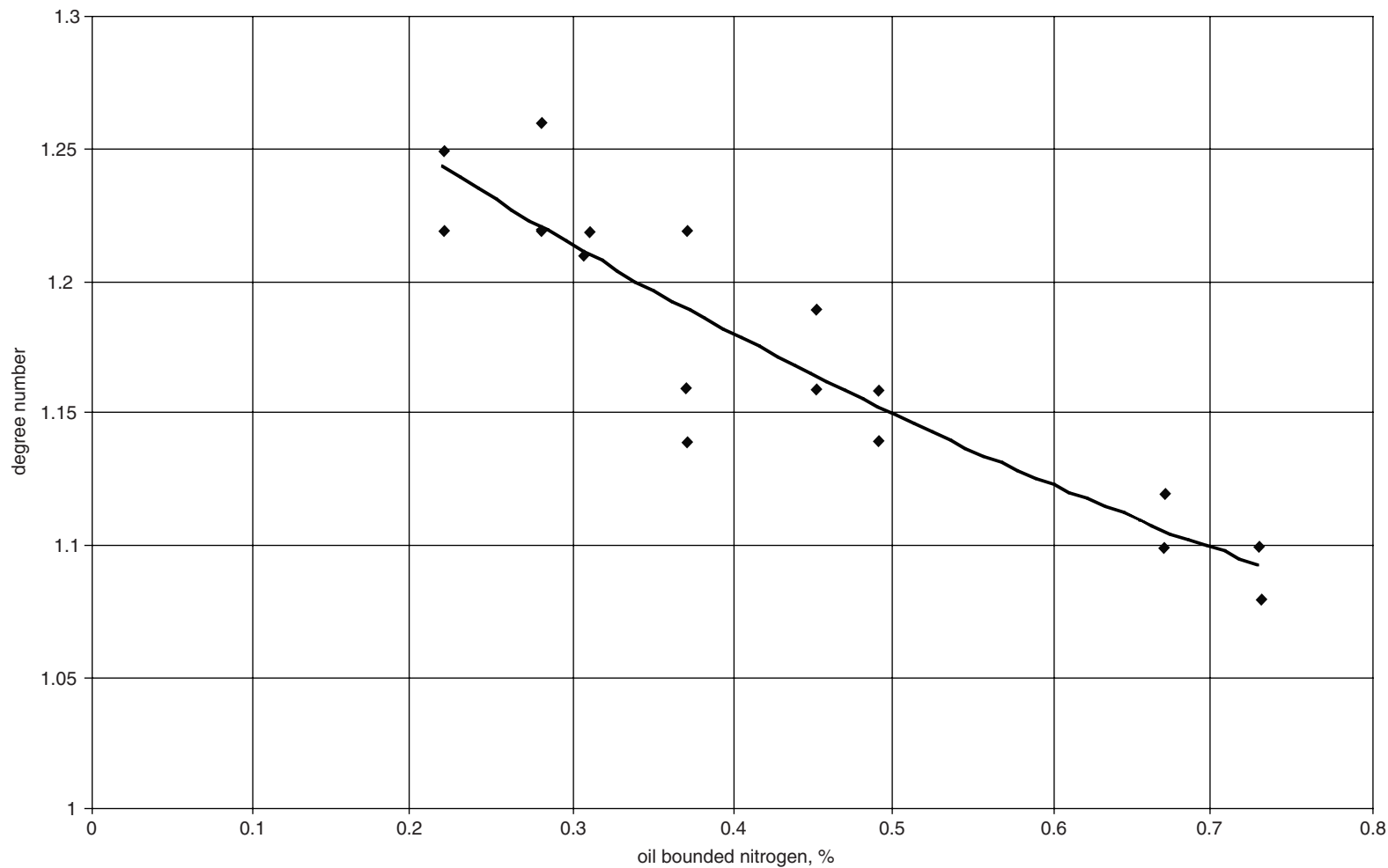


FIGURE 18.16 Seventeen boilers of 150- (three), 165- (eight), 210- (one), and 300- (five) MW_e utility boilers at full load with an O₂ range of 0.6 to 1.2%.

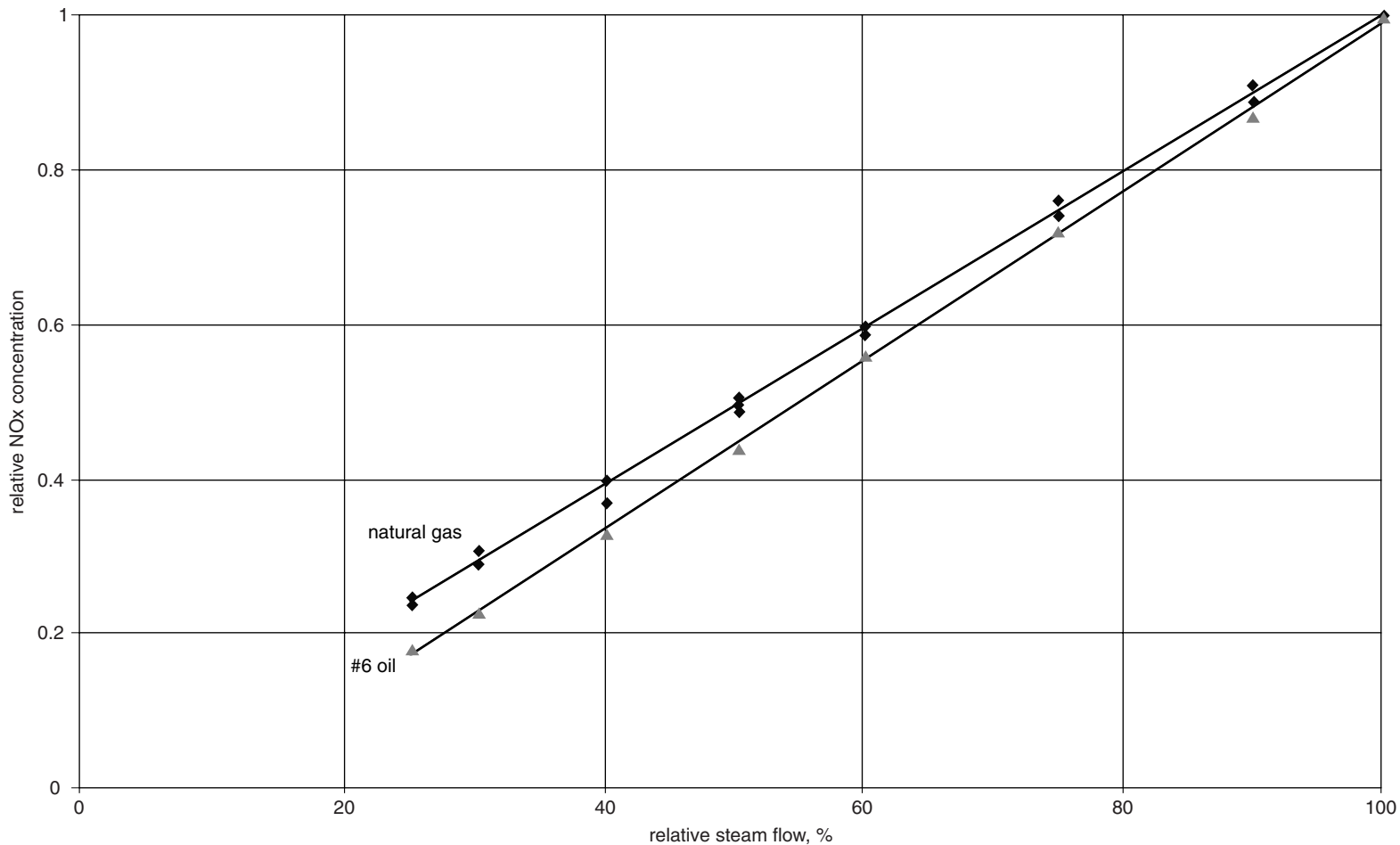


FIGURE 18.17 With ambient air, the shapes of the experimental curves, established on the 22–385,000 lb/hr (175,000 kg/hr) steam flow boilers firing both gas and #6 oil (0.22 to 0.49% N_p) in the 100 to 25% load range, are similar to a linear dependence.

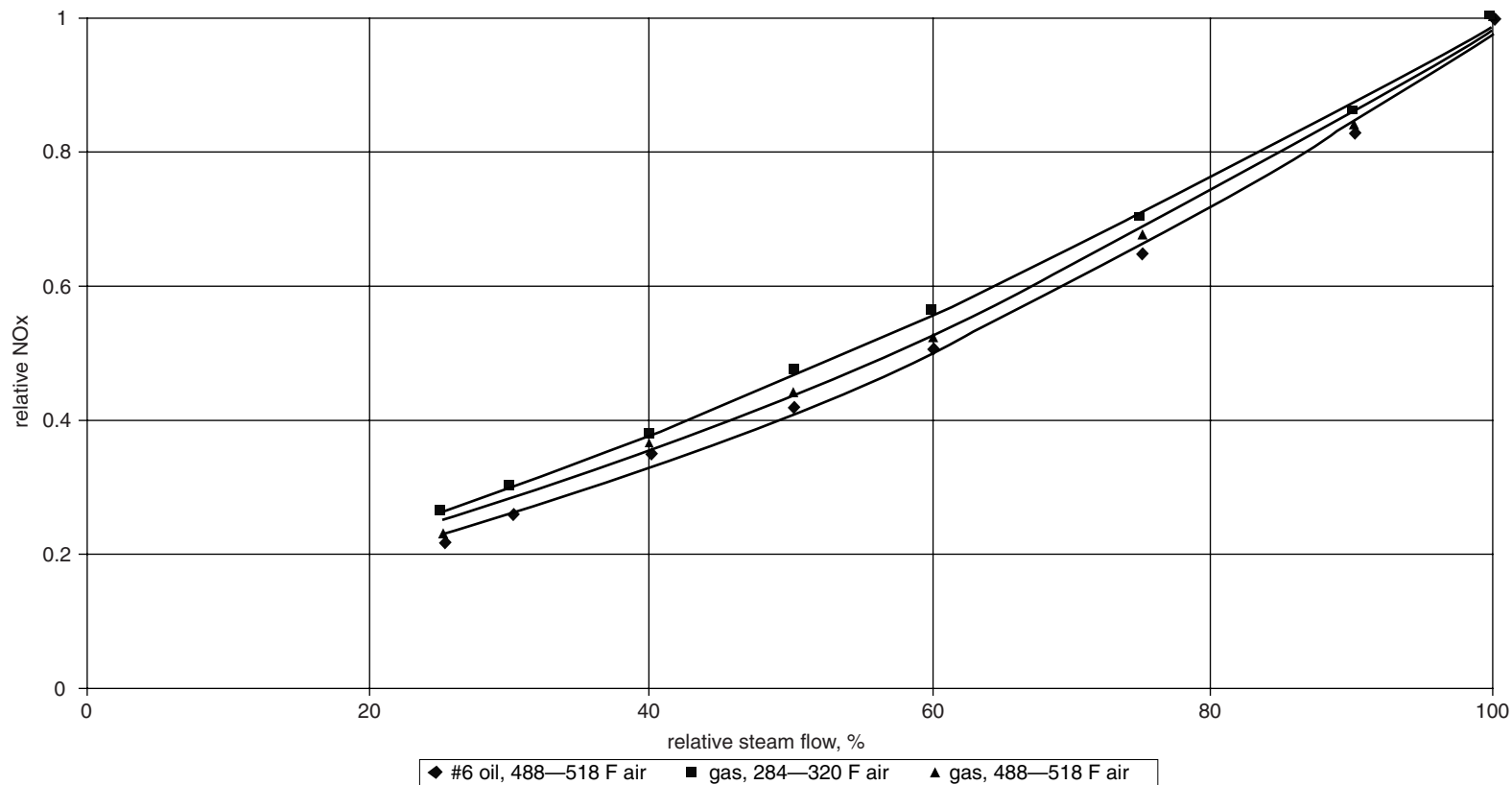


FIGURE 18.18 On preheated air (300 to 500°F), all curves are power functions with the ~1.25 and (1.1 to 1.2) degree numbers on gas and #6 oil, respectively.

18.2.4 BOILER/SYSTEM CONDITION IMPACTS ON COMBUSTION AND NO_x FORMATION

A new philosophy for NO_x reduction is emerging among multi-burner power boiler users. This philosophy sees the entire boiler as a combustion system where NO_x reduction can be accomplished and maintained in a three-step process:

1. Installation of new low-NO_x burners delivers the required NO_x reduction.
2. Improved maintenance protocols sustain the achieved NO_x reductions.
3. Optimized operational protocols achieve the best overall combustion and boiler performance.

Boiler design parameters such as air preheater (APH) design, superheater (SH) configuration, and the amount of refractory in the furnace also have major impacts on the amount of NO_x reduction achievable on a given boiler.

The cornerstone of this NO_x reduction philosophy is still, as it has been for years, the low-NO_x burner. Proven low-NO_x burners, along with a physical windbox model, provide the required NO_x reduction, but sustaining this NO_x reduction requires the improved maintenance and optimized operational protocols. With a balanced airflow distribution, balanced fuel flow distribution, and a set of proven low-NO_x burners, the combustion system's foundation is established for the desired NO_x reduction.

However, the NO_x reduction provided by the low-NO_x burner is only the first part of the process. The boiler must be properly maintained to sustain the NO_x reduction. Proper maintenance must address boiler cleanliness, the percentage of air in-leakage, and oil heater maintenance. The maintenance protocols affecting boiler cleanliness include proper soot-blowing practices on a daily basis and boiler washes during outages.

18.2.4.1 Boiler Cleanliness

The impact of boiler cleanliness on NO_x became very apparent when the results of NO_x testing, while firing residual fuel oil on the same unit, showed a dramatic 23% increase in NO_x as compared to results of NO_x testing performed just months earlier, with the same atomizer and at the same unit conditions. A visual inspection of the boiler indicated that it was well seasoned or dirty. A waterwall wash was performed during the next available outage, about a week later. After the outage, another NO_x test was performed; the results of this testing revealed a 25% decrease in NO_x as compared to the pre-outage results. The effect of furnace cleanliness on NO_x is shown in [Figure 18.19](#).

The second indication of the impact of boiler cleanliness on NO_x came when a sister unit started up after the low-NO_x burner retrofit. The start-up testing measured residual fuel oil (RFO) NO_x higher than any that had been previously measured with identical low-NO_x burners and atomizers at the first unit.

As the data sets from the units were analyzed, it became apparent that the relative level of furnace cleanliness could be correlated to the flue gas temperature entering the air preheater (APHGIT). It was also found that the second unit had extremely high APHGIT. The unit's soot-blowing practices were analyzed to determine the level of heat recovery area cleanliness, and it was found that many of the soot blowers were out of service. Upon minimizing the number of soot blowers out of service, the RFO NO_x on the sister unit fell to a level equal to those measured with identical low-NO_x burners and atomizers at the first unit. [Figure 18.20](#) exemplifies the impact of heat recovery area cleanliness on NO_x.

18.2.4.2 Furnace Air-In Leakage Influence

The impact of air-in leakage on burner performance is a phenomenon of balanced-draft boilers. There is a significant difference between NO_x generation conditions existing in forced-draft (pressurized)

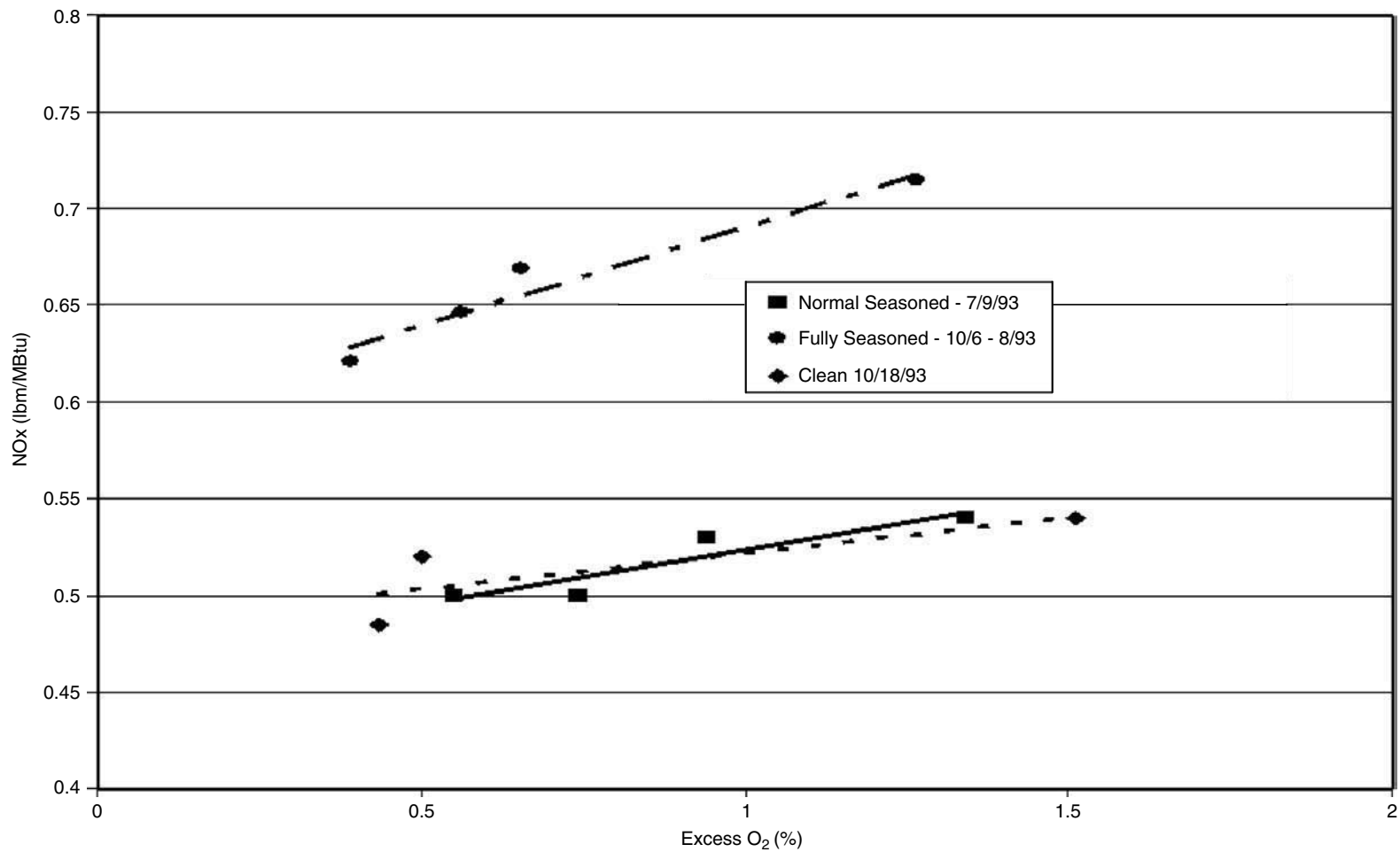


FIGURE 18.19 Effect of furnace cleanliness on NO_x.

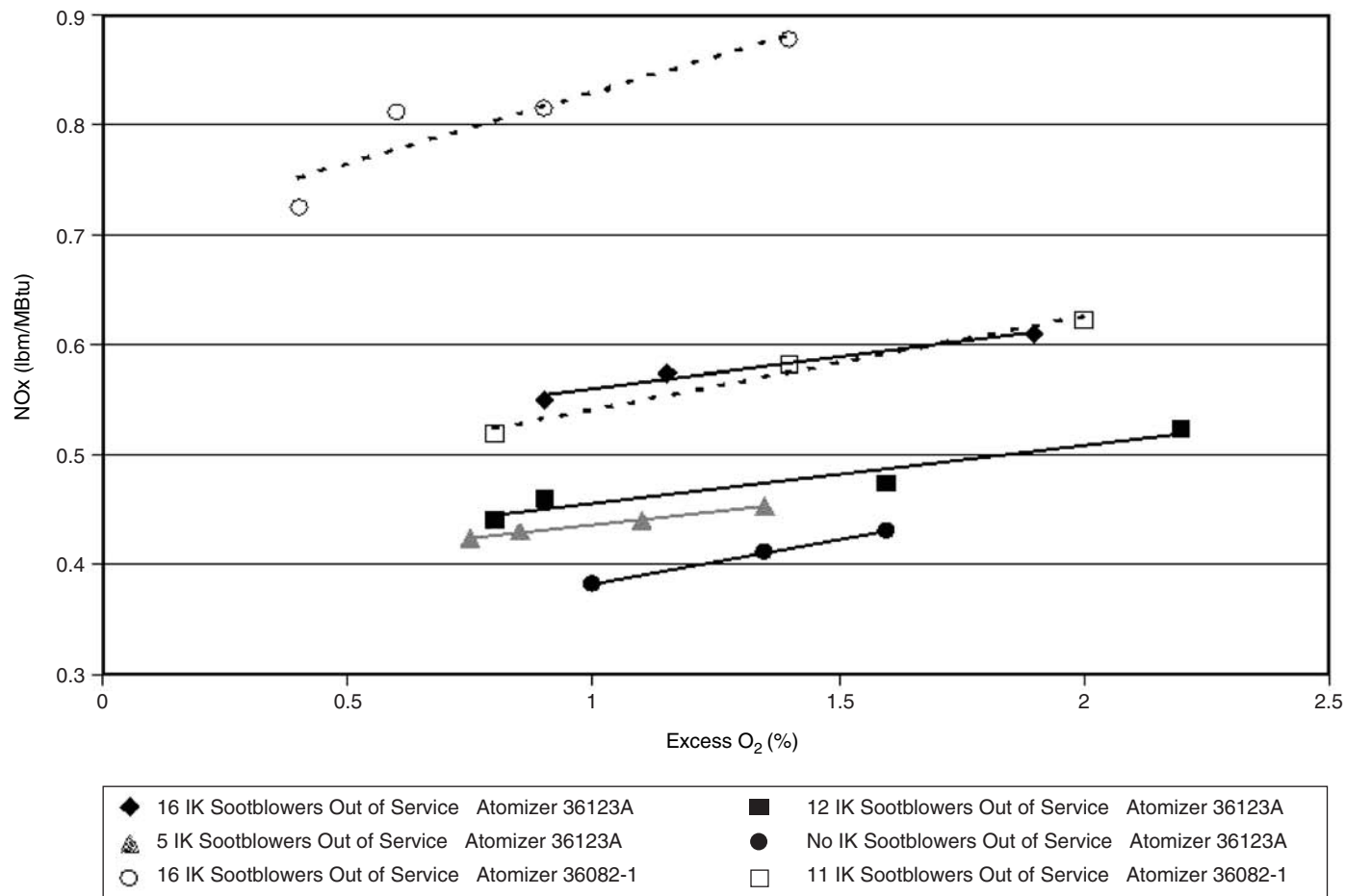


FIGURE 18.20 The impact of heat recovery area (HRA) cleanliness on NO_x.

and balanced furnaces. In pressurized furnaces under typical single-stage (unbiased) conditions, all the combustion air flow enters the furnace through the burners, and the O_2 required for complete combustion can be minimized. In balanced-draft furnaces, there is usually an air in-leakage of at least 3 to 5% (sometimes it exceeds 10%), sometimes leading to sub-stoichiometric combustion air flow coming through the burners. In-leakage air participates in the combustion process as well, providing complete combustion at slightly higher O_2 levels for many applications.

If the in-leakage location is within the furnace, the air deficit in the burners provides a self-stage combustion that complicates ICP burning out on the one hand, but can reduce NOx on the other (again depending on the in-leakage location). With other conditions remaining constant in pressurized and balanced furnaces, an NOx reduction of up to 15 to 18% is available. This was established by testing three sets of identical utility boilers: (1) 200-MW_e boiler with 12 burners on the rear wall, (2) 300-MW_e opposed fired boilers with 16 burners, (3) 150-MW_e opposed boilers with 6 burners. A comparison was made of NOx data measured in these similar utility boiler sets, one operating with a balanced draft and one operating with a forced draft, while firing gas at full load and with ~0% FGR flow.

However, if the in-leakage location is downstream of the furnace exit, in-leakage can have detrimental effects on the perceived performance of the burners. O_2 measurements taken at the top of the furnace can be approximately 0.8% lower than the control room O_2 measurements taken at the economizer outlet. In many cases, the mentioned difference between O_2 measurements is higher. This air in-leakage between the furnace exit and the economizer outlet can shift the NOx vs. O_2 and opacity vs. O_2 curves upward along the O_2 axis, as shown in [Figure 18.21](#).

18.2.4.3 Fuel Oil Temperature

Problems with the main fuel oil heaters can reduce the maximum fuel oil temperature, therefore raising the minimum viscosity attainable. The increased viscosity will increase the Sauter mean diameter of the atomized droplets, thus causing higher opacity. The increased droplet size will increase the excess O_2 required for complete combustion. NOx will increase due to the increased excess oxygen levels. The viscosity required by most burner vendors is 80 to 100 SSU.

18.3 CURRENT STATE-OF-THE-ART CONCEPTS FOR MULTI-BURNER BOILERS

18.3.1 COMBUSTION OPTIMIZATION

18.3.1.1 Equalization of Windbox Air Flow

Considering that air in the combustion process accounts for approximately 94% of the mass flow, numerous observations on boiler combustion systems have shown that correct air distribution and peripheral entry condition are key factors in the achievement of high performance (low NOx, low O_2 , and low CO). The concept of equal stoichiometry at each burner results in the minimal O_2 , NOx, and CO. The most direct way to achieve this is to ensure equal distribution of air and fuel to each burner. Airflow distribution is difficult because it requires a reliable and repeatable flow measuring system in each burner; as well as a means to correct the air flow without disrupting the peripheral inlet distribution or adding swirl to the air flow.

Mass flow deviations should be minimized to enable lower post-combustion O_2 , CO, and NOx concentrations. The lowest post-combustion O_2 concentration possible is constrained by the burner most starved for air. This starved burner will generate a high CO concentration and, consequently, the total O_2 must be raised to minimize the formation of CO in that burner. Minimizing burner-to-burner FGR deviations will even out the flame temperatures and, therefore, minimize the NOx formation from each burner. Equal inlet velocities and elimination of swirl through each burner are also crucial

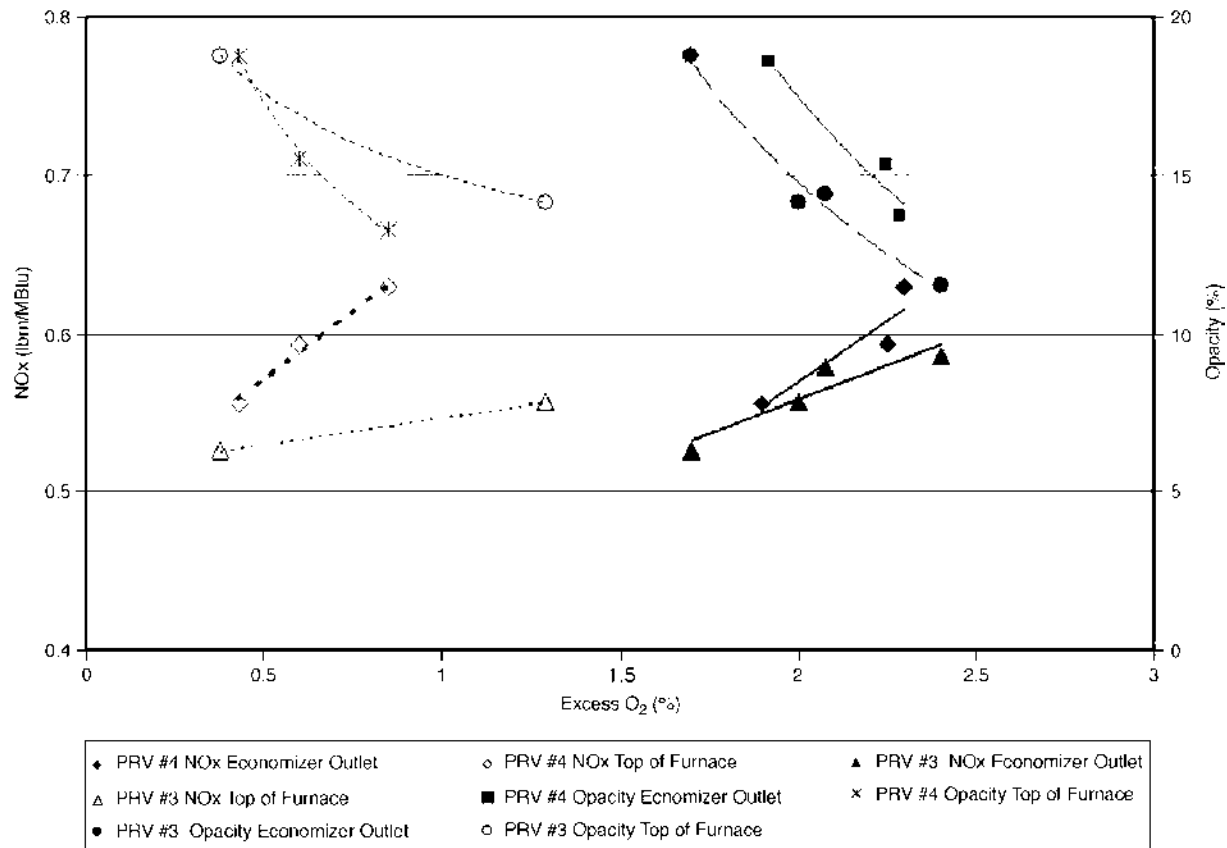


FIGURE 18.21 Air in-leakage between the furnace exit and the economizer outlet can shift the NO_x vs. O₂, and opacity vs. O₂ curves upward along the O₂ axis.

to burner performance. Because low-NO_x burners rely on the injection of fuel at precise locations within burner air flow, it is imperative that the proper air flow be present at these locations. Likewise, for optimum performance, the only swirl present must be that created by the burner itself.

No burner can be expected to simultaneously correct imbalances in the draft system *and* precisely control fuel/air mixing to minimize NO_x formation. Air flow modeling, as discussed in Chapter 11, prepares the air flow for the burner, allowing the burner to precisely control fuel/air mixing for maximum NO_x reduction. This approach has freed burner designers to focus solely on NO_x control, increasing the effectiveness of known control techniques to their maximum extent.

18.3.1.2 Fuel Flow Balancing Techniques

Fuel flow balance is just as important as air flow in reducing O₂ levels. A rough balance of fuel flow distribution is relatively easy to achieve by balancing pressure drops in the fuel header; that is, equalizing the fuel pressure at each burner, thus assuring that each burner is receiving the same amount of fuel.

Once the air flow has been balanced by a windbox model and the fuel pressures at each burner have been equalized, the unit is in the proper starting condition for the concept of fuel balancing to be taken a step further to reduce O₂ and maximize boiler efficiency. This goes back to the concept that there is probably one burner that is limiting the O₂ reduction. Even when the field data indicate that the air mass flow is within $\pm 5\%$ of average, and all the burners indicate equal fuel pressures (assuming that equal fuel pressures means equal fuel flow), the burner that is -5% in air flow will be running at an O₂ level that is approximately 1% lower than the average.

For further balancing, the unit should first be brought to MCR conditions and O₂ should be lowered to a point where the CO is in the 200- 400-ppm range on gaseous fuel, or opacity is in the 12 to 14% by EPA Method 9 for oil. The burner with the highest CO/opacity should be found and the fuel to that burner reduced slightly, resulting in a reduction in the overall CO/opacity level. This is an iterative process and should be repeated until any further fuel adjustments either show no effect, or increase the CO/opacity level.

The trick to this method of burner optimization is to find the burner with the highest CO/opacity. Ideally, on a multi-burner boiler, there should be a measurement grid at the economizer outlet that has been mapped out for burner stratification and measurement probes have been placed along the burner centerlines. CO measurements taken from this grid, at the conditions specified above, typically give a clear indication of the burner with the lowest amount of stoichiometric air. Lacking a mapped out grid, when the boiler is brought to the conditions outlined above, a visual inspection of the flames will typically indicate the “problem burner” as the one that has the “dirtiest” or “sootiest” flame.

18.3.2 METHODS TO REDUCE NO_x EMISSIONS

Combustion modification techniques developed to reduce NO_x emissions include low excess air operation, fuel- and air-staged burner design, staged combustion (off-stoichiometric or biased firing), reduced air preheat, flue gas recirculation, fuel-induced recirculation, reburn, and water tempering. As discussed in Section 18.2.2, low excess air operation limits the oxygen availability in the combustion zone and is highly effective in controlling fuel NO_x formation and, to a lesser extent, thermal NO_x.

The NO_x problem forced the development of various completely different furnace designs, usually implementing low-NO_x burners and FGR systems, sometimes containing over-fired air ports for supply of air and/or air/FGR mixtures to the furnace space located above the burners. There are single wall burner utility boilers where air ports are located on the target wall. Also, there are low-NO_x industrial and utility boilers containing special furnace devices for steam/water injection in the combustion zone and for fuel injection in the post-combustion zone (a kind of reburning).

Actually, any of the mentioned methods is capable of reducing NO_x by up to 35 to 40%, but it cannot satisfy the strictest of today's NO_x requirements. These applications require implementation of at least two NO_x reduction methods. It is important to note that, as with the example of boiler load and FGR in Section 18.2.3, the efficiency of the implemented methods depends on their sequence. The first NO_x reduction method is much more efficient than the second one. This conclusion has been confirmed on many utility boilers retrofits where up to four NO_x reduction methods were used. That is why a combination using more than three methods on the same boiler is typically inefficient.

18.3.2.1 NO_x Reduction by FGR Implementation

It is well known that thermal NO_x can be effectively controlled by reducing the flame temperature. The most effective technique is the use of FGR, which when mixed with the combustion air acts as a diluent to decrease the flame temperature, and sometimes (at high FGR rates) even increases the flame luminescence. FGR with lower temperatures will be more effective in flame temperature reduction.

Usually, FGR is taken after an economizer where temperatures range between 500 to 800°F (260 to 430°C), the O₂ range is typically between 1 to 3%, and actual ICP concentrations are negligible. A FGR fan is usually required to supply FGR flow to the combustion air flow or to the furnace.

There are also applications where a comparatively cold exit flue gas (240 to 320°F; 120 to 160°C) is taken from ID fan outlet and supplied, for example, to FD fan inlet or to windbox. This flue gas usually contains much more excess O₂, especially on the boilers equipped with air preheaters having comparatively large air leakage characteristics. Empirical data show that, if this air leakage can be minimized, this "cool" FGR gas can provide 25 to 30% higher relative NO_x reduction as compared to the "hot" FGR, but it has a more severe impact on boiler thermal efficiency reduction. FGR flow can also be induced into the combustion air forced draft fan inlet, allowing the use of FGR without requiring a separate FGR fan.

FGR can also be taken from the furnace zone adjacent to the burner exit, and can be induced back into the combustion air flow, also allowing the use of FGR without requiring a separate FGR fan. Unlike regular, comparatively cold FGR flows, this flue gas has temperatures of at least 2000°F and 2200°F (1100 and 1200°C) with gas and oil firing, respectively, it consists mostly of unreacted air (O₂ ~ 16 to 19%) and ICP. This method is completely different from all others above because its effectiveness cannot be associated with significant changes in the flame temperature conditions or lowering the O₂ concentration in the combustion air. The effectiveness of this method relies on interactions between reagents participating in NO formation and both radicals and ICP that are present in high concentrations, which slow the NO formation reactions rates, thereby reducing NO_x output.

Flue gas can be recirculated in a number of ways: directly to the furnace through slots located under, above, around, or between the burners; through over-fired air ports or ports located on the target wall; premixing the FGR flow with the combustion air upstream of the windbox (called premix FGR) (this is typically accomplished via a sparger section as depicted in [Figure 18.22](#), mixing the FGR at the burner exit (called plenum FGR), and premixing the FGR with the gaseous fuel (on oil, the flue gas would pass through the fuel gas piping). This technique of flue gas entrainment is called fuel dilution and uses no additional fan power because it is typically induced into the fuel stream by the fuel pressure itself, or by pressure energy supplied via an intermediate media such as steam if the available fuel pressure is not high enough. Also, FGR can be supplied to the boiler hopper; this method is typically used only for steam temperature control.

Various levels of NO_x reductions can be achieved at the same FGR rates using these different techniques. [Figure 18.23](#) (illustrating test data from 150-, 165-, 200-, and 300-MW_e boilers) and [Figure 18.24](#) (800-MW_e boiler) show full load gas firing test data received on utility boilers equipped with different burner designs, numbers, and arrangements, with O₂ levels of 0.8 to 1.4%.

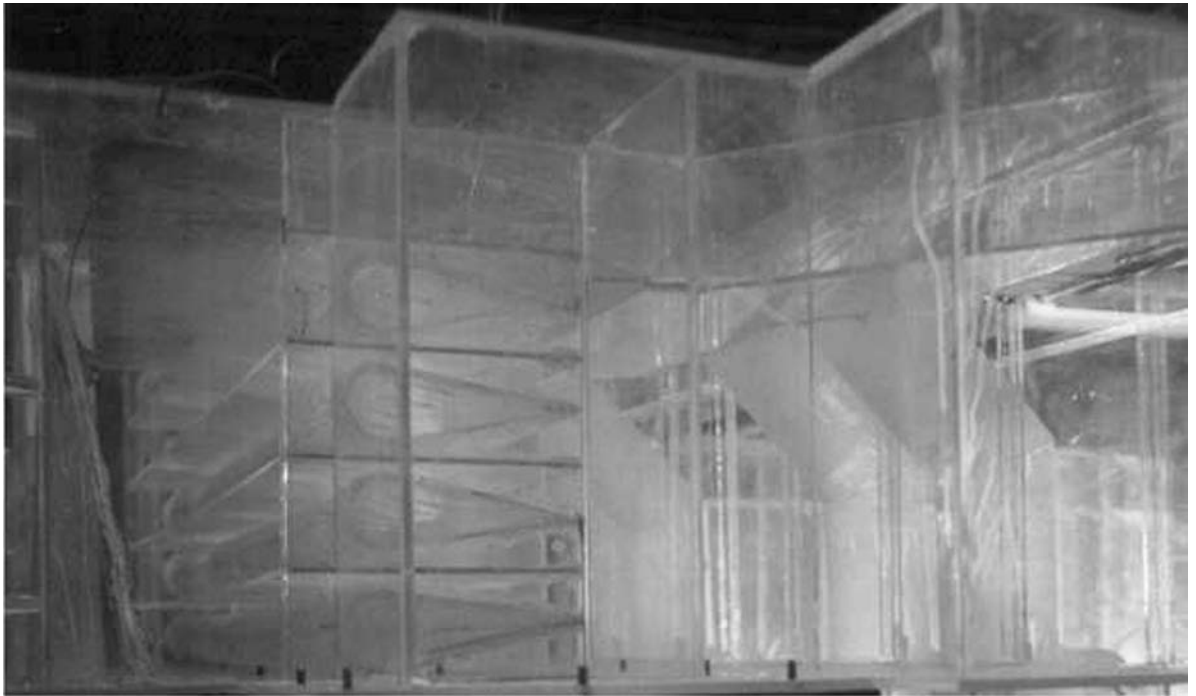


FIGURE 18.22 Premixing the FGR flow with the combustion air upstream of the windbox (called premix FGR, this is typically accomplished via a sparger).

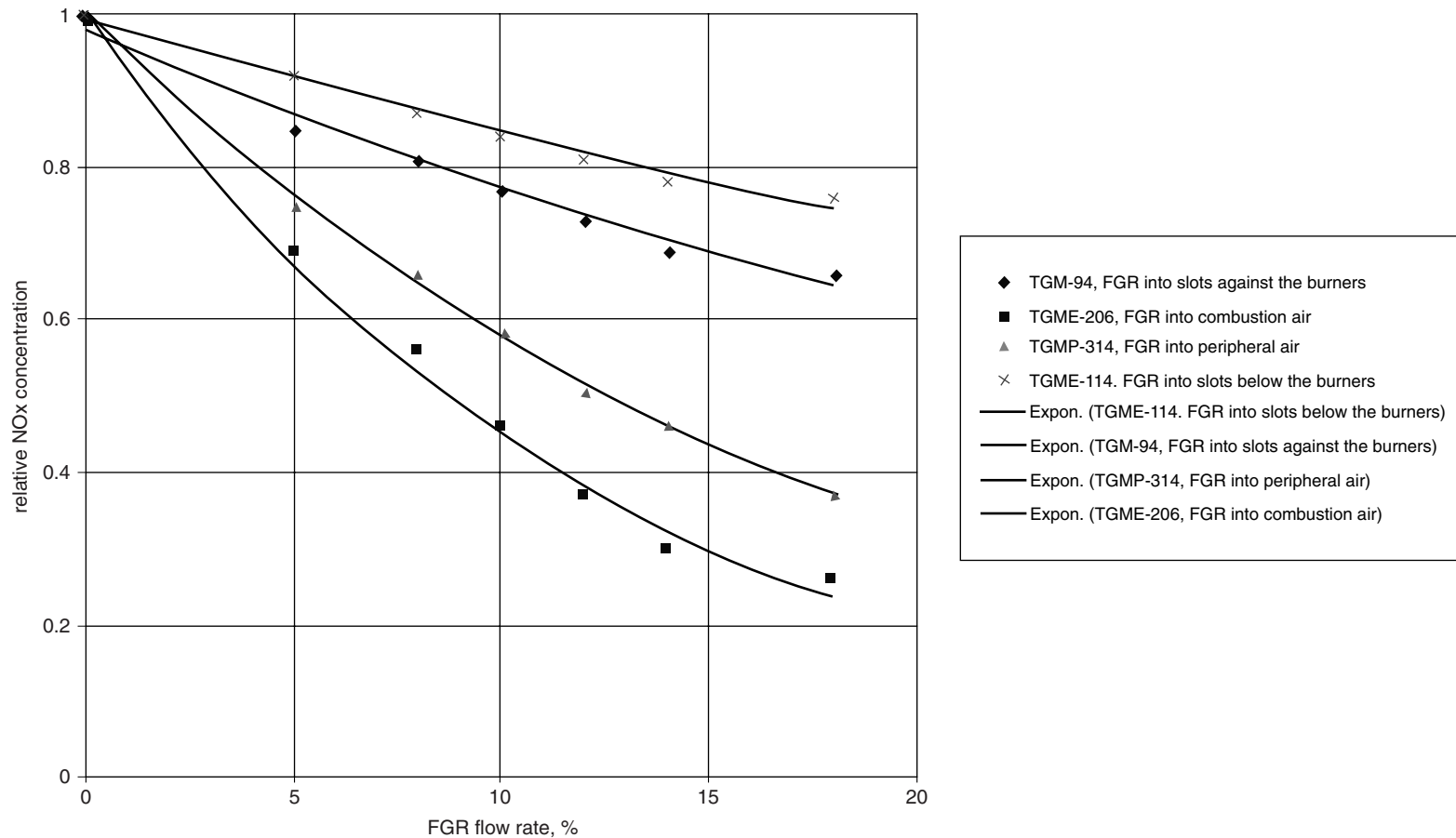


FIGURE 18.23 Test data from 150, 165, 200, and 300 MW_e boilers.

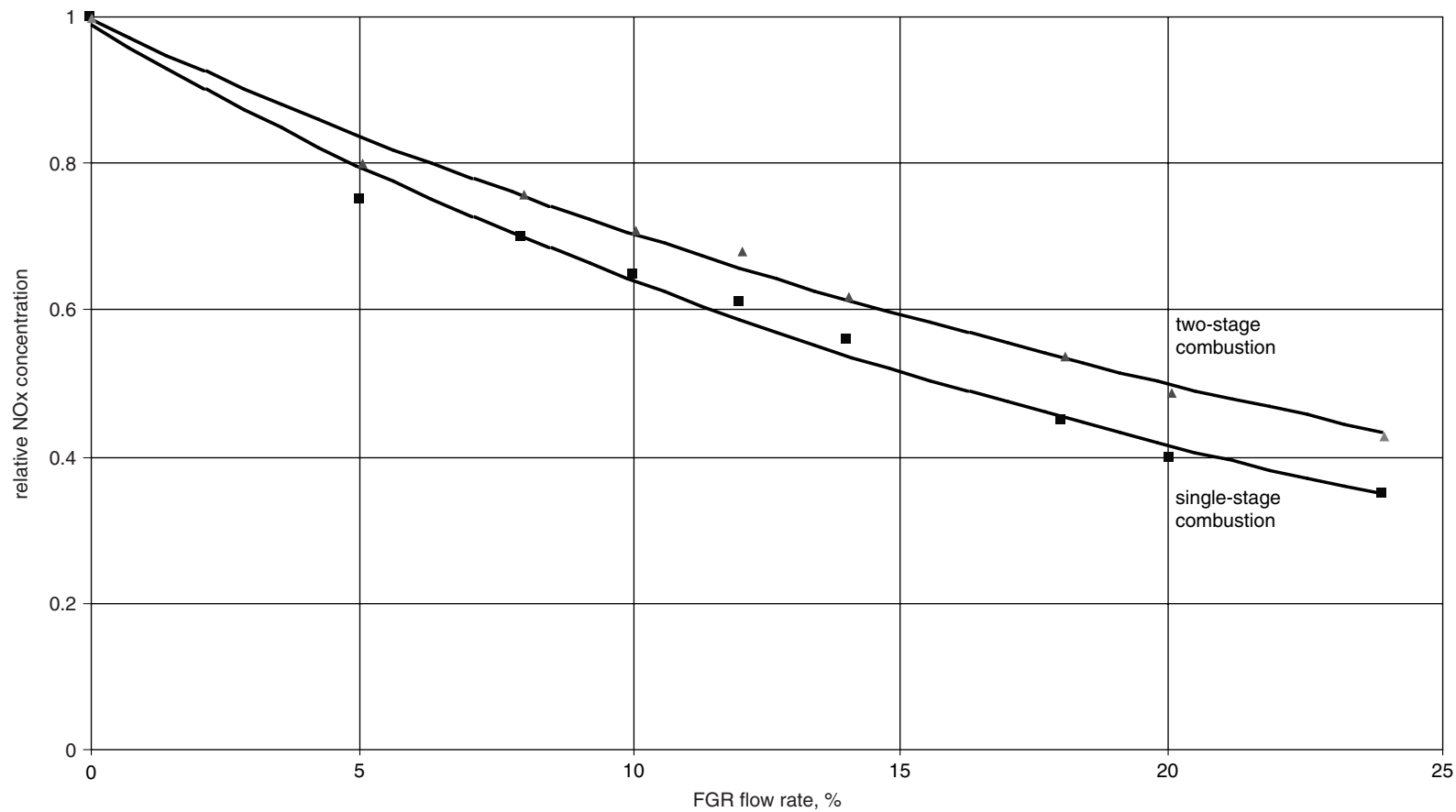


FIGURE 18.24 800-MW_e boiler.

It should be noted that all curves shown in these figures are exponential, meaning that it would make sense to determine an optimal operational FGR rate. This optimal limit will be different for each boiler application, because it depends on the boiler design (and accordingly on FGR impact on the boiler operational parameters). The data presented show that the typical optimal FGR rate range for NO_x reduction is 14 to 18%. The NO_x reductions experienced through FGR implementation, as described above, explain why it is the most frequently implemented NO_x reduction technique on newly designed and low-NO_x retrofit boilers.

Figure 18.23 shows that the most effective conventional FGR supply system for low NO_x is to provide a high-quality mixture of the FGR and combustion air, as with the venturi-style, low-NO_x burners installed at the 200-MW_e TGME-206 boiler (single wall burner installation), where 10, 12, and 18% FGR flows provided 54, 63, and 74% NO_x reduction, respectively. A comparatively high NO_x reduction efficiency is also achieved on the 300-MW_e boiler of TGME-314 model (16 burners installed in two rows on opposed firing walls), where straight FGR flow is mixed with the burner air flow. Methods used to supply FGR directly into the furnace, implemented on the TGM-114 (opposite wall burner installation, 150 MW_e) and the TGM-94 (single wall burner installation, 165 MW_e), are dramatically less effective.

Figure 18.24 demonstrates the test data obtained on the 800-MW_e boiler of model TGME-204 (36 low-NO_x burners installed in three rows on two opposed firing walls) with full load single- and two-stage gas combustion (with 33% air supplied through over-fired air ports located above the top burners) at O₂ levels of 0.8 to 1.2%. As previously discussed, it is only natural that the NO_x reduction provided by premixing FGR with combustion air flow is less effective with two-stage combustion, when the flame temperature level is significantly lower. However, the 46 and 57% NO_x reduction achieved with 18 and 24% FGR flow rates, respectively, during two-stage gas combustion, can still be considered a comparatively high efficiency because it is the second NO_x reduction method implemented (see Chapter 17.3.2). At full load on this boiler, the total NO_x reduction exceeds 80% (from 600 to 620, to 90 to 95 ppm), and CO levels less than 200 ppm at O₂ levels of 0.8 to 1.2%. This NO_x reduction was achieved utilizing a combination of: low-NO_x burner retrofit, two-stage gas combustion, premixed FGR, and water injection in the primary and secondary combustion air flows. The same scope of work was repeated on three neighboring 800-MW_e boilers of the same design installed at this power plant. On all four units, total NO_x reductions of 81 to 85% were achieved.

Similar full load NO_x reduction data have been obtained on packaged industrial boilers equipped with a single venturi-style low-NO_x burner, firing gas with ambient air. The data are presented in Figure 18.25.

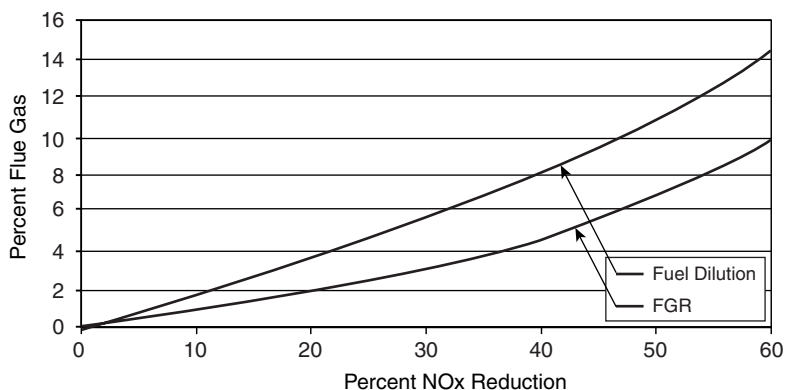


FIGURE 18.25 Full load NO_x reduction data obtained on packaged industrial boilers equipped with a single venturi-style low-NO_x burner, firing gas with ambient air.

Recent developments have shown that, when firing gaseous fuels, using fuel dilution (mentioned above) to induce flue gas into the gaseous fuel is truly the most effective method of FGR supply (~25% more effective, as shown in [Figure 18.25](#)). As per the test data, while natural gas firing at 2.5 to 4.5 O₂, NO_x can be reduced by 66% with 12 to 13% fuel dilution flow.

The reasoning why fuel dilution provides more efficient NO_x reduction as compared to conventional FGR supply methods is as follows. When natural gas (conditionally CH₄) is fired with combustion air, without the use of FGR or fuel dilution, every CH₄ molecule encounters a certain quantity of oxygen molecules, which it is dependent on; the excess O₂ level, the mixing quality of the fuel with the air, and the diffusion rates of the CH₄ and O₂ at the flame front region as shown in [Figure 18.26\(A\)](#). The combustion reaction rates (including reactions for incomplete combustion products burnout, prompt NO_x and thermal NO_x generation) are dependent on the actual excess O₂, the combustion air temperature, and the fuel/air mixture quality. These variables also determine, along with the given burner/furnace design, the heat release and heat distribution within the furnace, which in turn determines the shape of the temperature profile curve, as well as the maximum temperature value and its location within the given furnace. Concurrently, these specific features of the temperature conditions have a major impact on the determination of the above combustion reaction rates.

When CH₄ is fired with combustion air containing typical FGR, with all other combustion conditions being the same as in the previous example, every CH₄ molecule will encounter a lower quantity of oxygen molecules than it would without FGR. The FGR impedes the diffusion of O₂ toward the flame front, as depicted by the increased combustion product content on the air side (slight darkening on the air side) in [Figure 18.26\(B\)](#). The difference between the quantity of oxygen values in these two cases depends on both the FGR composition and the FGR flow rate. For example, for a boiler design operating under a positive furnace pressure (i.e., no air in-leakage in either the furnace or the convective path) and providing an equal distribution of the combustion products in both the furnace exit cross-section as well as in the convective path cross sections, the FGR composition will be similar to the combustion product composition leaving the furnace. Because the combustion air makes up ~94% of the mass flow within the combustion process, by supplying an FGR flow rate of ~11%, the oxygen concentration in the combustion air/FGR mixture is reduced by the same 11% (being only slightly diluted, this is why there is only a slight darkening on the air side in [Figure 18.26 \(B\)](#)). Also, the flame temperature level in the furnace is reduced primarily due to the addition of the FGR mass. The magnitude of the temperature reduction caused by this phenomenon depends on the FGR temperature; relatively high-temperature FGR flows will provide less flame temperature reduction than relatively low-temperature FGR flows. The combination of the decreased O₂ concentration and the lowered flame temperature level results in reduced reaction rates of all the combustion reactions mentioned above, including those reactions that are responsible for NO_x generation in the given furnace.

When CH₄ containing fuel dilution is fired with combustion air in the same furnace, combustion conditions change significantly from the examples previously discussed: because the CH₄ is only ~6% of the mass flow within the combustion process, the fuel dilution has a much greater dilution effect on CH₄. If the fuel dilution rate is the same as the FGR rate previously discussed (i.e., 11%), the dilution is equivalent to a 2 fuel dilution:1 fuel molecular ratio, as depicted with the heavy darkening on the fuel side in [Figure 18.26\(C\)](#). Due to this phenomenon, when a diluted fuel fires, the flame temperature becomes significantly lower in comparison with the second case, related to natural gas firing in the combustion air containing the same FGR flow. The reduced flame temperature influences all of the above combustion reaction rates, including reactions that are responsible for NO_x generation in the given furnace. This explains why the NO_x reduction efficiency of fuel dilution is so much greater than that of FGR. On the other hand, the mentioned flame temperature reduction (associated with the substitution of FGR with fuel dilution of the same percentage and temperature) reduces heat release in the furnace, increases the furnace exit flue gas temperature, and increases heat release in the boiler convective path, especially in the superheater

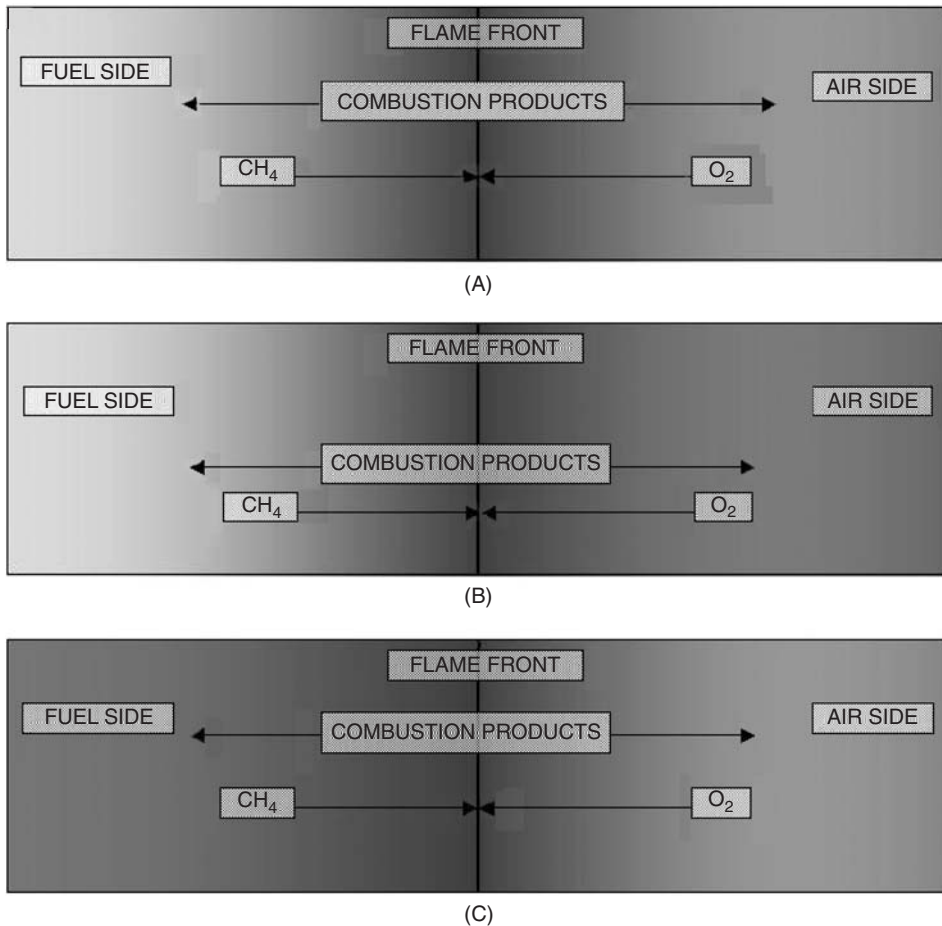


FIGURE 18.26 (A) When natural gas (conditionally CH_4) is fired with combustion air, without the use of FGR or fuel dilution; (B) when CH_4 is fired with combustion air containing typical FGR, with all other combustion conditions being the same as in the previous example; and (C) 1 fuel molecular ratio, as depicted with the heavy darkening on the fuel side.

and reheater sections. Also, the flue gas temperature will be increased throughout the convective path, including the stack flue gas temperature, which reduces the boiler thermal efficiency.

Fuel dilution can also be implemented in combination with FGR premixed in the combustion air and/or steam injection. A combination of fuel dilution and steam injection can provide a total NO_x reduction of up to ~82%.

Low flue gas recirculation rates (<20%) typically have a relatively minor impact on boiler performance (in many cases on utility boilers, it can be compensated for by increasing the turbine's efficiency; on industrial boilers, this loss is unrecoverable). However, high FGR rates (>20%) can significantly impact boiler performance, especially the redistribution of heat flux between radiant surfaces and high-temperature convective heat exchange surfaces of utility boilers and, accordingly, on superheated/reheated steam temperature. Excessive use of flue gas recirculation can result in significantly increased superheated and reheated steam temperatures, or significantly increased superheat and reheat attemperation spray flows, the latter affecting boiler efficiency. In addition, power expenses recalculated for thermal efficiency losses are estimated at 0.03 to 0.05% per 1% FGR. These facts explain why FGR was (before NO_x requirements were established) typically only used to control and regulate reheated and superheated steam temperatures at minimum loads.

A typical older boiler would be designed to provide the required steam temperature and flows without FGR at full load, but at partial loads (usually $<70\%$), FGR was switched on to maintain steam temperatures. FGR flow rates were increased as load continued to decrease. Modern boiler designs incorporate FGR for both NO_x reduction and steam temperature control. At higher loads (100 to 80%) FGR is typically used for NO_x reduction; however, as above, at partial loads, FGR is typically used for steam temperature control.

Very large amounts of premixed FGR, above 45%, can affect flame stability with conventional burner designs, while plenum FGR has been tested on some burners up to 70% with no noticeable effect on flame stability. Depending on the FGR supply method, the FGR/air mixing quality will be different, and, accordingly, flame stability, vibration, and other negative issues of combustion (especially in the case of firing natural gas in ambient air) can be met even at low FGR amounts. However, these problems are not typically experienced when FGR flow rates are below 14 to 18%.

Alternative techniques to reduce flame temperature involve reduced air preheat, and water or steam injection. Water or steam injection will result in efficiency losses that are unrecoverable, while reduced air preheat can be achieved with no loss in efficiency if steam surface changes are made.

18.3.2.2 Multi-Stage Combustion on Utility Boilers

Staged combustion involves delaying the mixing of fuel and air, and is effective for both thermal and fuel NO_x control. Typically, staged combustion creates an initial fuel-rich combustion zone with air added downstream to complete combustion. Staged combustion can be achieved by the use of low-NO_x burners (internal staging), over-fired air ports, operating existing burners with biased fuel firing or with burners-out-of-service (BOOS). Biased fuel firing for a large, wall-fired unit is implemented by operating the lower burners with reduced combustion air and/or increased fuel to produce a fuel-rich zone in the lower portion of the furnace. The upper burners are operated with a corresponding increase in air or decrease in fuel to complete fuel burnout. An extreme form of staging involves operation with an upper row of burners out of service or its equivalent over-fired air ports. In this case, fuel is shut off to these burners and the burners are operated with air only.

The principal of stagewise fuel combustion involves the arbitrary division of the flame into two or more stages. In most cases, in the first, high-temperature stage, combustion takes place with an excess air factor of less than unity and, in the subsequent stages, which have a comparatively low temperature level, secondary combustion of the previously ICPs occurs with relatively high excess air. Thus, NO_x formation is retarded in the first stage, because there is insufficient oxygen in the reaction zone and, in the subsequent stages, due to the relatively low temperature of the flame. This has established the advantages of stagewise combustion as an NO_x reduction method over other methods.

There are various methods of performing staged combustion in furnaces. The most well-known and studied methods are based on the three following options:

1. Gas/oil combustion with a significant air deficit in all burners and supplying a certain part of the total air flow directly to the furnace through ports or slots located above the burners
2. Redistribution of air flows between the burners located at different rows
3. Redistribution of fuel flows between the burners located at different rows

All the mentioned options are directed to get a “vertical” fuel/air unbalance in the furnace: excess fuel and air deficit in the lower part and, on the contrary, excess air and fuel deficit in the top part of the combustion zone. However, when a boiler changes over to stagewise fuel combustion with a “vertical” unbalance in the fuel:air ratio, concentrations of all ICPs — including carcinogenic substances — increase. Much test data exists confirming the above conclusion. An example is shown in [Figure 18.27](#) where test data obtained on a model TGM-94 boiler are presented. It can be seen that a 50% NO_x reduction while firing gas is reached at $\sim 18\%$ over-fire air flow, but CO

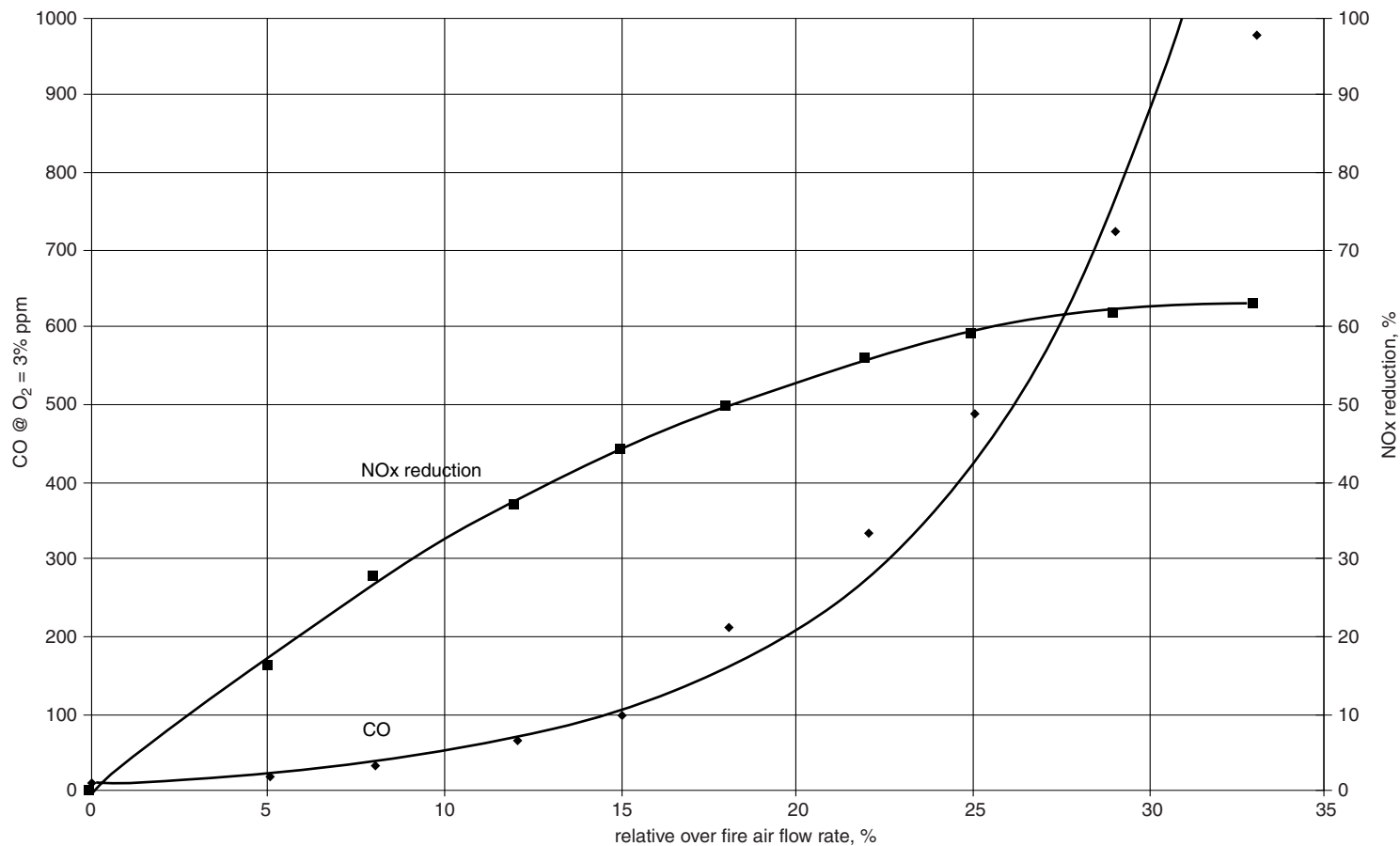


FIGURE 18.27 Test data obtained on a model TGM-94 boiler.

has increased to ~200 ppm. Further increasing the “vertical” unbalance to 25% results in ~60% NO_x reduction; however, CO has increased to ~400 ppm. Increases in CO concentrations and other ICPs are typical of “vertical” stage combustion. The data for this boiler also indicated that staged combustion operation while firing gas at ~18% over-fired air flow reduced the boiler efficiency by ~0.85%.

Another system of stagewise combustion, with a “horizontal” unbalance of the fuel:air ratio, gives rise to substantial reductions of ICP emissions; however, it results in increased NO_x. This method of staged combustion utilizes the interaction of fresh combustion air jets directed from ports or slots located opposite the burners, with fuel radicals and already formed ICP in flames generated by burners receiving sub-stoichiometric air. It intensifies the combustion process and especially burnout while firing oil, as illustrated with [Figure 18.28](#) where test data was obtained on a TGM-84 (~950,000 lb/hr or 430,000 kg/hr steam flow) co-generating boiler firing #6 oil. The data show that this method provides reductions of opacity, carcinogenic substances, and a small CO reduction from ~38 to ~24 ppm. However, NO_x increases very quickly: on average, 1% of the secondary air flow brings a relative NO_x increase of ~1.2%. The boiler data from this testing indicated that this method can be applied for applications requiring increased reheated/superheated steam temperatures. Moreover, this method increases the boiler thermal efficiency and reduces operational power expenses due to decreasing the air resistance of air path. On the above boiler, ~15% secondary air flow increased the total efficiency by 1.1%. These data indicate that optimally designed three-stage fuel combustion systems could be developed with simultaneous “vertical” and “horizontal” imbalances of the fuel:air ratio.

Using calculations explaining the above data as a basis for preliminary design work, a new furnace was developed. The furnace had three burner rows and slots above the upper row located on the furnace front wall. FGR could either be premixed with the combustion air going through the burners and slots, supplied directly into the furnace through slots located on the rear wall, opposite the lower burners, or in both locations. The primary air flow was supplied through the burners; secondary and tertiary air flows were redirected to the slots located on the rear and front walls, respectively.

Tests were performed over a boiler load range of 100 to 40%, α is 1.05 to 1.11 firing gas, and at full load, α is 1.03 to 1.1, firing #6 oil. To obtain reliable data on the NO_x reduction effectiveness of stagewise combustion, tests were carried out in alternate modes, with measurements being made during single-stage combustion under the same load, excess air factor, FGR, and other combustion conditions. Comparative tests when firing oil were carried out with oil supplied from the same tank. All results were compared with the baseline conditions of single-stage combustion, in which the secondary and the tertiary air flow dampers were fully throttled, it being conditionally assumed that their relative flows were ~0.

The test data comparing single- and three-stage gas combustion are shown in [Figure 18.29](#). It is seen that transferring from single- to three-stage combustion at ~10% secondary air flow yields NO_x reductions of ~40 and ~63%, with tertiary air flows of 15 and 25%, respectively. This NO_x reduction is accompanied by a corresponding CO increase from ~135 to 165 ppm during single-stage combustion of 260 to 290 and 350 to 400 ppm, accordingly. Under the above conditions, operation with FGR was not required at any point over the entire load range. The decrease in thermal efficiency was 0.3 to 0.4%, much less dramatic than either two-stage combustion (with a “vertical” imbalance) or FGR fan operation. The same retrofit was performed on six neighboring TGM-94 boilers installed at this power plant.

18.3.2.3 Water and Steam Injection for NO_x Reduction on Utility Boilers

Water and/or steam injection systems have been installed on many multi-burner boiler applications for NO_x reduction. However, due to the associated decrease in boiler thermal efficiency, these systems are only used when required to reach a high level of NO_x reduction, such as at full load conditions,

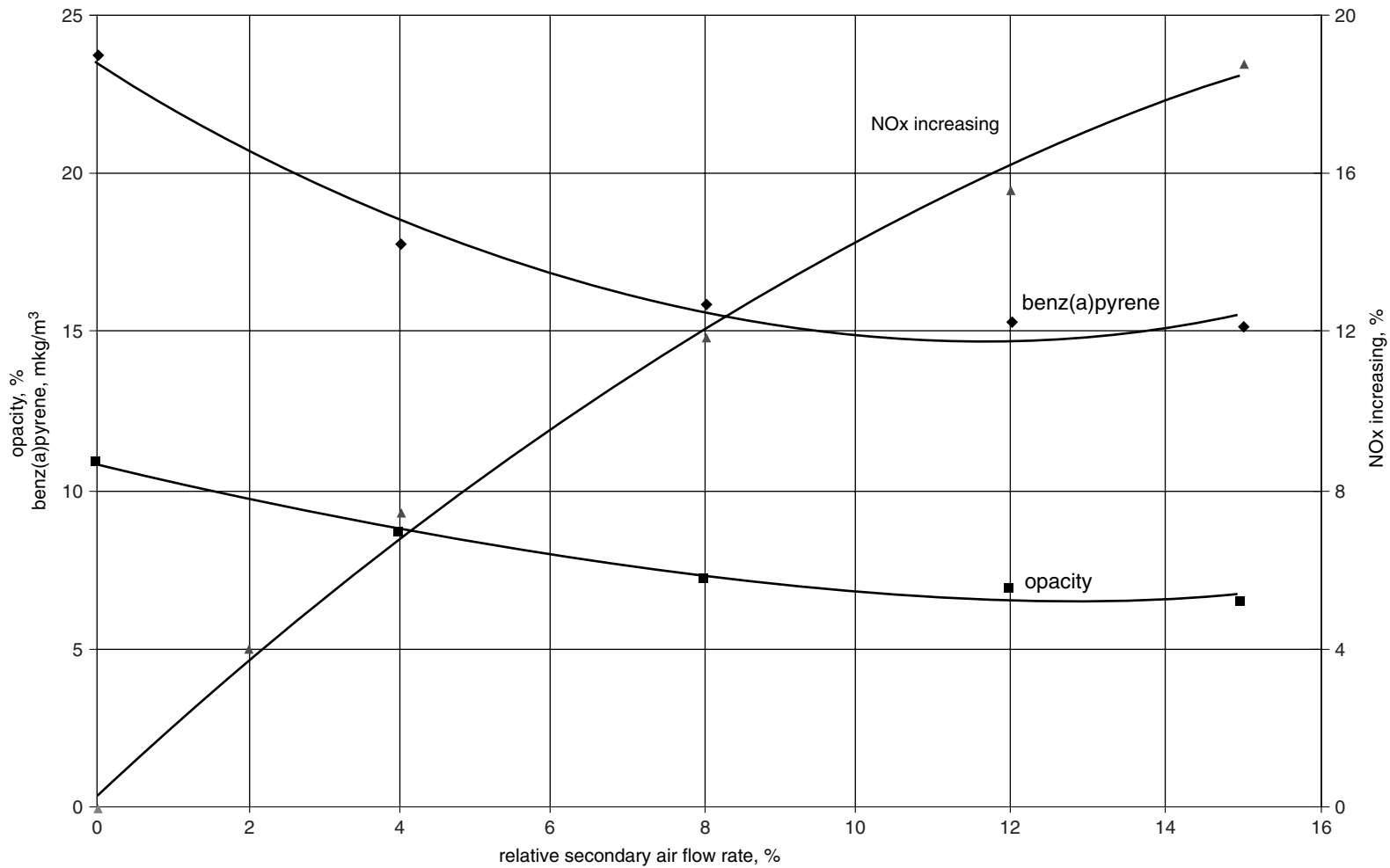


FIGURE 18.28 Stagewise method intensifies the combustion process and especially burnout while firing oil.

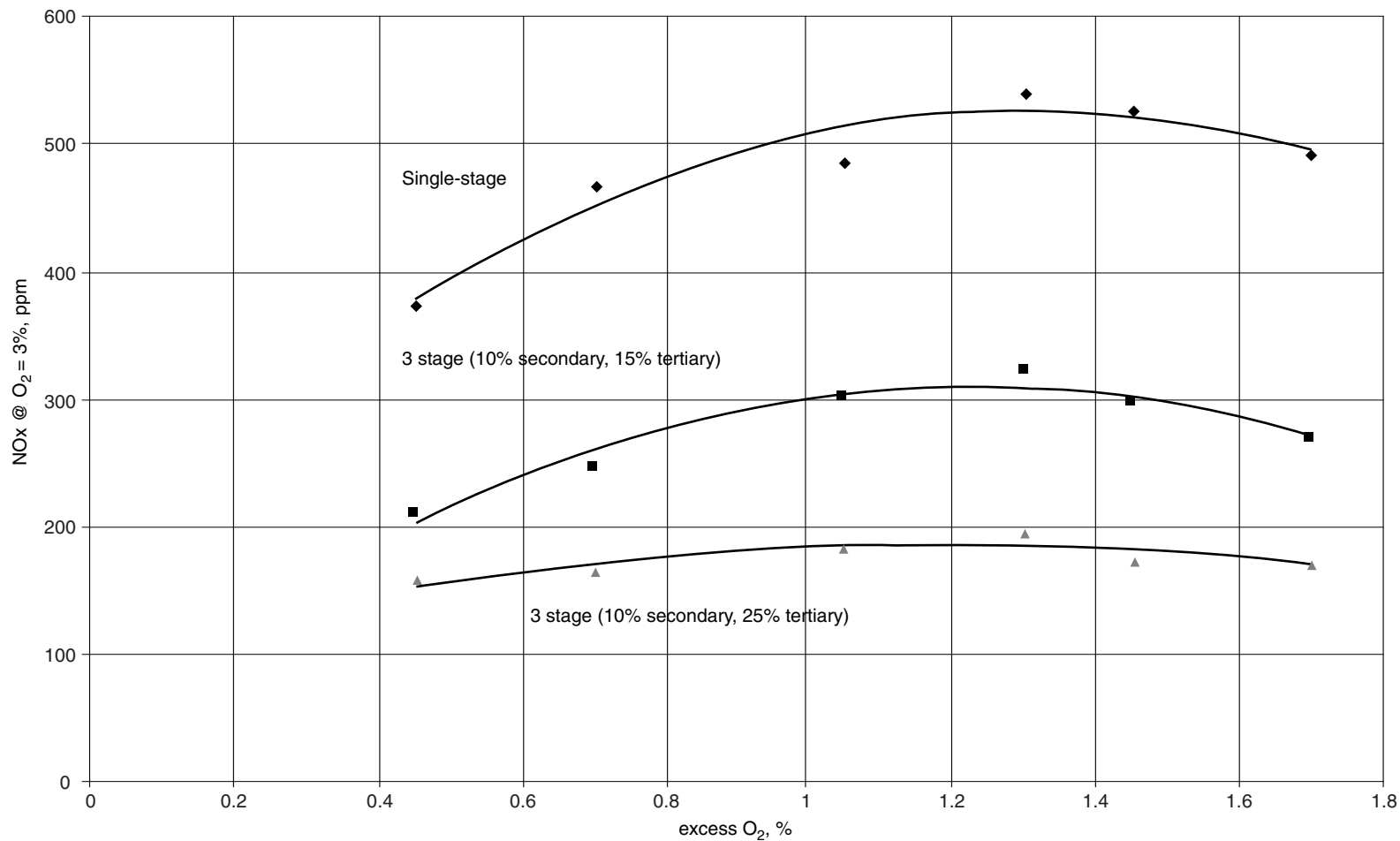


FIGURE 18.29 Data comparing single- and three-stage gas combustion.

and are generally turned off at lower loads when more conventional NO_x reduction methods can maintain the NO_x level requirements.

18.3.2.3.1 Water Injection Effectiveness

Water injection is most effective for high BZHR utility boilers firing natural gas, where a significant NO_x reduction can be achieved with a comparatively low water flow rate injected into the preheated combustion air flow (e.g., water injection reduced NO_x by 35% on a 816-MBtu/sf-hr BZHR boiler).

The impact of water injection on boiler thermal efficiency was studied in detail on two boilers: a 165-MW_e TGM-94 and a 300-MW_e TGMP-314 while firing natural gas at full load. Although initial thermal efficiency values were different for the two boilers (91.9% for TGM-94 and 94.7% for TGMP-314 — both at LHV), thermal efficiency reduction numbers were almost identical, i.e., (~62%).

18.3.2.3.2 Steam Injection Effectiveness

Injecting steam, instead of water, into the combustion air reduces the NO_x reduction effectiveness for a given mass of water injection. As an example, for a given application, NO_x reduction was ~28% with steam as compared to ~46% with water. When the water injection flow rate was increased, the associated NO_x reduction effectiveness was ~51% for water injection and ~41% for steam injection. However, the NO_x reduction efficiency decrease associated with steam injection was accompanied by a lower thermal efficiency reduction due to the added latent heat of the steam. The thermal efficiency decrease was ~0.27% with steam injection, as compared to ~0.62% with water injection (i.e., ~2.3 times lower).

18.3.2.3.3 General Evaluation of Water/Steam Injection Effectiveness

The above discussion demonstrates how high levels of NO_x reduction can be achieved with water/steam injection, and that these reductions are comparable to the NO_x reduction effectiveness reached with FGR described in Chapter 17.3.2.1.

Unlike FGR, however, a water/steam injection does not require significant capital investment, but does have a much larger negative impact on boiler thermal efficiency (as an order of magnitude, percents with water, tenths of percents with steam, as compared to hundredths of percents with FGR). Although water and steam injection are associated with a negative impact on boiler efficiency, the simplicity and high effectiveness of this NO_x reduction method implementation, combined with very low capital expenses, make it very attractive for many multi-burner boiler owners.

18.3.2.4 Special Features of Low NO_x Gas/Oil Combustion on Utility Boilers

Chapters 17.1.2 and 17.2.2.2 show that there are some utility boilers where NO_x is higher while firing natural gas than that experienced while firing #6 oil. For example, data obtained on the TGM-94 boilers presented in [Figures 18.10](#) through [Figure 18.13](#) demonstrate that the NO_x while gas firing is ~24 to 27% higher as compared with NO_x data obtained while firing #6 oil under the same load (~94%), O₂ (1.4 ± 0.2%), and other operational conditions.

This phenomena of the NO_x while firing gas being higher compared with the NO_x data obtained while firing #6 oil has been experienced during testing of many former-U.S.S.R. utility boilers in the size range of 165 to 800 MW_e. In some cases, NO_x while firing gas exceeded the NO_x while firing #6 oil (even containing 0.67 to 0.73% N_p) by ~50%. On the contrary, on utility boilers in the size range of 25 to 80 MW_e, the NO_x while firing #6 oil exceeded the NO_x while firing gas by about 25%; moreover, as N_f values increased (from 0.22 to 0.73%), the greater the margin between the NO_x while firing #6 oil and the NO_x while firing gas. Likewise, on industrial boilers, the NO_x was almost always higher while firing #6 oil, and differences of 24 to 36% and 12 to 18%, respectively, were typically experienced with ambient and preheated air. However, intermediate results were obtained while testing utility boilers in the size range of 100 to 150 MW_e where

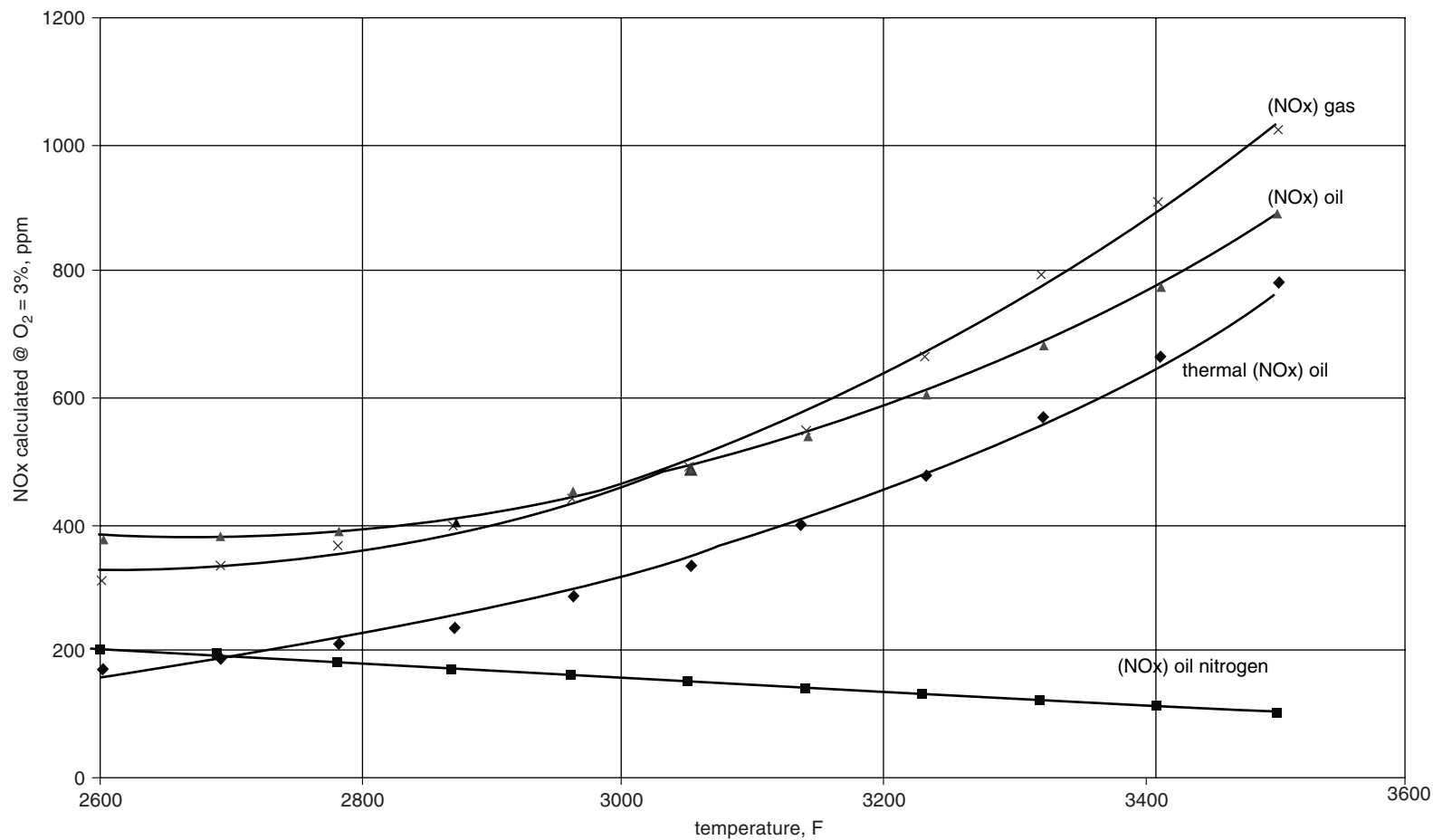


FIGURE 18.30 The corresponding NOx concentration curve, calculated for excess air factor $\alpha = 1$.

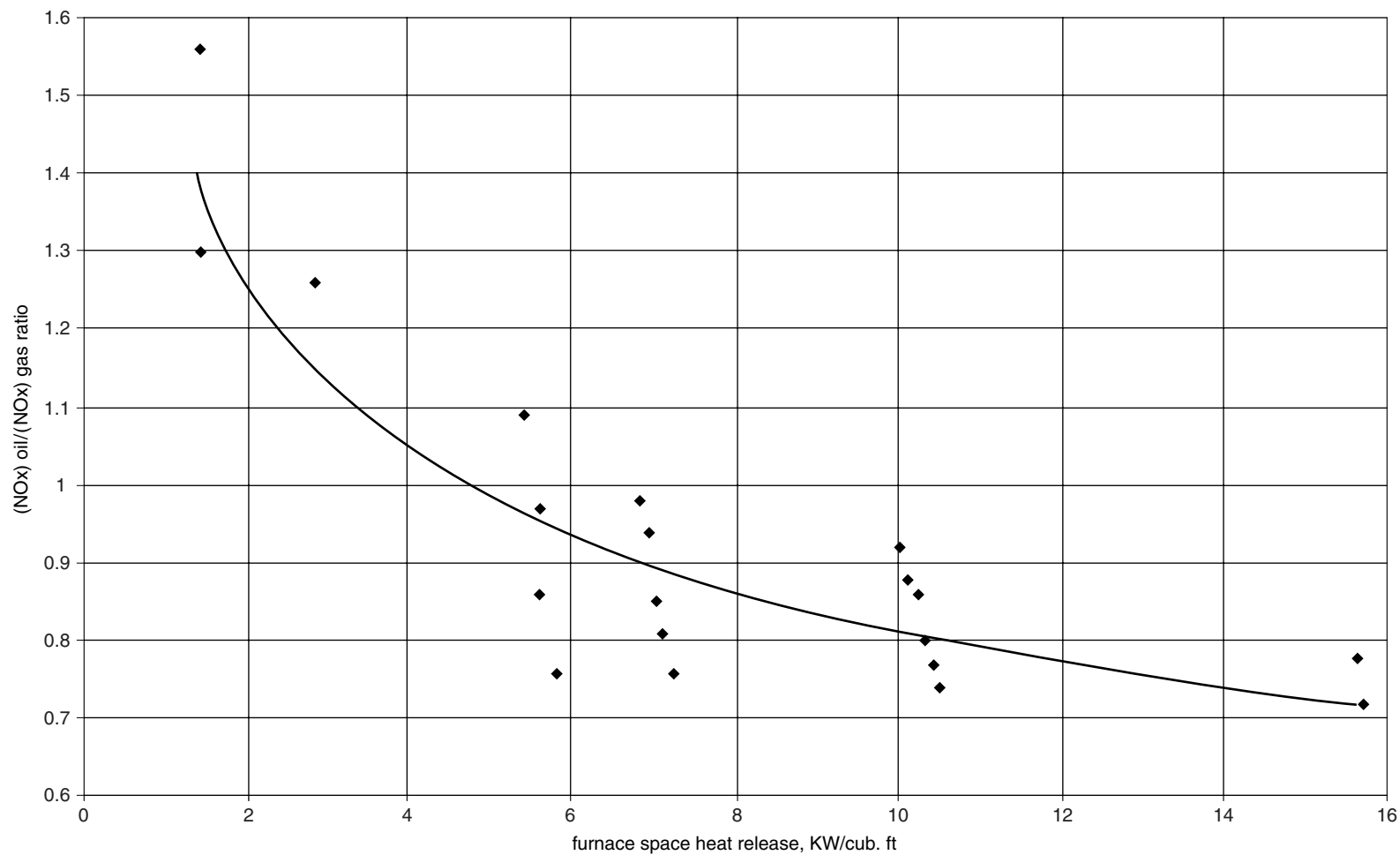


FIGURE 18.31 The ratio between NOx numbers obtained with gas and oil firing are considered as a function of the furnace space heat release.

differences between the NO_x numbers for the two fuels were less than $\pm 10\%$. The controversial data obtained on these boilers of different designs and capacities can be explained based on the well-known influence of the flame temperature on NO_x formation.

While firing natural gas containing no N_f, NO_x can only be generated by the oxidation of nitrogen contained in the combustion air. This NO_x generation can only occur by one of two mechanisms: thermal NO_x formation due to flame temperatures of above 2780°F (1527°C), mostly in the burner zone heat release (BZHR) space, its concentration rises with temperature in an exponential manner; prompt NO_x formation in the beginning of the flame where the temperature level is significantly lower. Under usual combustion conditions, its concentration is much less than the thermal NO_x concentration; prompt NO_x has a slight dependency on temperature. Under certain furnace conditions required for low-NO_x combustion, the thermal NO_x concentration can be reduced to the prompt NO_x concentration level, or even less. However, these conditions have little influence on prompt NO_x.

While firing natural gas (conditionally consisting of methane only), the maximum combustion temperature level depends solely on excess air and the combustion air temperature. The corresponding NO_x concentration curve, calculated for excess air factor $\alpha = 1$, is shown in [Figure 18.30](#).

When firing oil-containing N_f, there are two sources of NO_x formation: air nitrogen and fuel nitrogen. In addition to the two mechanisms available for gas firing, a third mechanism is available as well: oxidation of the partially destructed nitrogen-containing compositions that occur in the beginning of the flame, whose concentrations decrease slightly with increasing temperature.

Figure 18.30 shows calculated NO_x curves for both nitrogen sources, and their summed values. It shows that oxidation of 0.5% N_f can generate NO_x of up to ~200 ppm. When we consider the processes in the comparatively cold furnaces designed for industrial and small utility boilers (50 to 100 MW_e), the above 200 ppm will far outweigh the thermal NO_x formation. Because these processes are occurring at relatively low temperatures, they indicate relatively low NO_x levels from these boilers, and an oil-firing NO_x level much higher than the gas-firing NO_x level. In the 120-150-MW_e boilers, temperatures can reach 2900 to 3100°F (1600 to 1700°C). Figure 18.30 indicates that the curves calculated for gas and oil firing intersect within this temperature interval, indicating relatively equal NO_x levels while firing either gas or oil. For units greater than 165 MWe, the flame temperature can reach between 3200 and 3500°F (1800 and 1900°C). This temperature increase results in a gas-firing NO_x level much higher than the oil-firing NO_x level.

The above explanation can be confirmed with the data presented in [Figure 18.31](#), where the ratio between NO_x numbers obtained with gas and oil firing are considered as a function of the furnace space heat release. Furnace designs for increasingly higher-capacity utility boilers are usually (with very few exceptions) accompanied by a significant increase in heat release as related to the furnace volume. Thus, the design of these boilers allows the analysis of the ratio between NO_x numbers and this designed parameter. Despite experimental data deviations from the average empirically established curve, the strong character of this dependence cannot be doubted.