8.6 Boiler Control and Optimization

X. CHENG, R. H. MEEKER, JR. (2005)

Inputs by G. Liu (2005)

INTRODUCTION

This section is subdivided into four parts. The first part describes the characteristics of the boiler and its associated equipment (fans, dampers, and so on). The second part is devoted to a description of the conventional boiler controls, including steam pressure and temperature, air/fuel ratio, draft pressure, and feedwater controls. The third part describes the pollution control systems. And the fourth part discusses optimization and describes the methods of steam pressure floating, air/fuel ratio optimization, soot blower optimization, and blowdown controls.

Boilers are available in two basic designs: fire tube and water tube. Fire-tube boilers are generally limited in size to approximately 25,000 lb/hr (11,340 kg/hr) and 250 PSIG (1.7 MPa) saturated steam. Although they are noted for their ability to respond to changing demands, their size and pressure limitations preclude their use in large industrial facilities. Because of thermodynamic considerations, boilers should produce steam at high pressure and temperature to realize a maximum work efficiency. These conditions are achievable only with water-tube boilers—hence, they will be given prime consideration in this section.

Steam boilers are used by the electric utility industry to produce steam for power generation and by manufacturing and process industries (nonelectric utility) to produce steam for both power generation and process heating and energy conversion. Electric utility boilers tend to be larger and operate at higher pressures: A typical coal-fired utility boiler might produce 3 million lb/hr (1.36 million kg/hr) of superheated steam at 2400 PSIG (16.5 MPa). A typical nonelectric utility industrial boiler might produce 400,000 lb/hr (181,000 kg/hr) of steam at 900 PSIG (6.2 MPa). Industrial boilers are commonly referred to as co-generation or combined heat and power (CHP) applications.

In this section, the controls of both the electric utility and the process industry boilers are described. In general, the loads on the electric utility boilers are more stable, and when they change, they change slower than the often drastically varying loads that process industry boilers have to handle. As a consequence of this difference, the utility industry boiler control system shown in Figure 8.6l does not include the type of lead/lag compensation that is described in Figure 8.6s. Therefore, the overall controls shown in Figure 8.6l are for reference only, and the reader should study the discussion of the individual loops and select the appropriate ones for the type of load dynamics at hand.

The basic components of a water-tube steam boiler are the furnace, where air and fuel are combined and burned to produce combustion gases, and a water-tube system, the contents of which are heated by the combustion process. The tubes are connected to the steam drum, where liquid and vapor are separated and the generated water vapor withdrawn. If superheated steam is to be generated, the steam from the drum is...
passed through the superheater tubes, which are exposed to the combustion gases. Supercritical or “once-through” boilers operate above the critical point of water where there is not a distinction between liquid and vapor; these boilers are not equipped with steam drums.

THE BOILER

Efficiency

The thermal efficiency of a steam generator is defined as the ratio of the heat transferred to the water (steam) to the heat input with the fuel. One of the goals associated with the operation, maintenance, and control of a boiler is to maximize its thermal efficiency.

The boiler efficiency is influenced by many factors. A fully loaded large boiler that is clean and properly tuned (with blowdown losses and pump and fan operating costs disregarded) is expected to have the following efficiencies:

- On coal: 88%, with 4% excess oxygen; 89%, with 3% excess oxygen
- On oil: 87%, with 3% excess oxygen; 87.5%, with 2% excess oxygen
- On gas: 82%, with 1.5% excess oxygen; 82.5%, with 1% excess oxygen

Boiler efficiencies seldom exceed 90% or drop below 60%. Efficiencies will tend to vary with individual design and with loading, as shown in Figure 8.6a. Efficiencies will also vary as a function of excess air, flue-gas temperature, and boiler maintenance. A 1% loss in efficiency on a 100,000 lb/hr (45,360 kg/hr) boiler will increase its yearly operating cost by about $20,000. A 1% efficiency loss can result from a 2% increase in excess oxygen or from about a 50°F (28°C) increase in exit flue-gas temperature.

Efficiency can be computed by the direct or the indirect method. The direct method uses the ratio of the rate of heat transferred to the water (outlet steam specific enthalpy × steam mass flow–feedwater specific enthalpy × feedwater mass flow) to the rate of heat input by the fuel (higher heating value × fuel mass feed rate).

The indirect method uses fuel, ash, and stack gas analysis to do a per-unit-basis accounting of all heat losses, subtracting all losses from the higher heating value of the fuel and dividing the result by the higher heating value. The indirect method is more accurate, because it does not rely on the relatively inaccurate steam and fuel flow measurements. The major losses considered by the boiler indirect efficiency calculation equations are:

- Dry gas loss: sensible heat carried out of the stack with the combustion air and combustion products
- Moisture loss: loss due to vaporizing the moisture in the fuel and the moisture produced from combustion of the hydrogen in the fuel
- Incomplete combustion loss: loss due to combustion of carbon that results in carbon monoxide (CO), instead of the complete combustion product, carbon dioxide (CO₂)
- Unburned carbon loss: loss due to carbon that does not get combusted and ends up in the refuse (ash)
- Moisture in the combustion air loss: loss due to heating up water vapor contained in the combustion air
- Radiation loss: heat lost from the external furnace walls to the surrounding air and other surfaces

If variations in only the dry gas loss are of primary interest (normally the largest source of energy loss), then the efficiency can be approximated using Equation 8.6(1), which assumes nominal fixed values for most of the above losses, based upon the type of fuel.

\[
E = 100 \left[ 1 - 10^{-3}(0.22 + \frac{K''y}{1 - y/0.21})(T_i - T_a) - \frac{\Delta H}{H_f} \right]
\]

8.6(1)

where \( y \) is the mole fraction of oxygen in the flue gas, and \( K'' \) is a coefficient assigned to each fuel: 1.01 for coal, 1.03 for oil, and 1.07 for natural gas. The term \( \Delta H/H_f \) is about 0.02 for coal, 0.05 for oil, and 0.09 for gas; the terms \( T_i \) and \( T_a \) are the stack and ambient temperatures (°F).

Where accuracy of the calculated efficiency is important, such as validation against performance guarantees, it is best to refer to the full indirect method calculations; a generally accepted standard for steam generating unit efficiency calculations is the ANSI/ASME Power Test Code (PTC) 4.1.

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FIG. 8.6a
Boiler efficiencies and steam costs vary with both the design of the boiler and its loading.

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Equipment

Steam boilers referred to in this section are drum-type boilers. Very large, supercritical pressure boilers are the “once-through” type and are found only in the largest electric generating plants.

In electric utility applications, a boiler is typically part of a generating unit: one boiler dedicated to one steam turbine. In industrial applications, often two or more steam boilers are connected to a common header supplying process steam users and, commonly, one or more turbine generators. The “load” on a steam boiler refers to the amount of steam demanded by the steam users (including turbines).

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The boiler steaming rate must follow the steam demands arising from process heat or power generation requirements. The equipment and the control system often must be capable of satisfying rapid changes in load. Load changes can be a result of rapidly changing process requirements, power demand changes, or cycling control equipment. Whereas load may be constant and steady over prolonged periods, the boiler must have sufficient “turndown” to stay in operation at reduced capacities as portions of the plant may be shut down. This consideration usually leads to a greater “turndown” requirement for the boilers than for any other portion of the plant. At the same time, it is desirable to maximize boiler efficiency at all loads.

Boiler designs, in terms of air and gas flow configuration, are generally either forced-draft (FD) or balanced-draft boilers. Forced-draft boilers operate at positive pressure with air supplied to the boiler by a forced-draft fan. Balanced-draft boilers usually operate at slightly negative pressure with air supplied by an FD fan and flue gas withdrawn by an induced-draft (ID) fan (larger than the FD, due to the combustion products). Most boiler air and flue-gas fans are centrifugal (typically with backward-curved blades). Axial flow fans are used less often due to the nature of the axial flow static-pressure vs. capacity curve and the possibility of stall conditions.

The Role of Sensors

Figure 8.6b shows a typical boiler arrangement for gas, oil, or solid fuel. It also shows the in-line instruments used on a boiler, together with some advice on the type of sensor to be used.

The normal “on-line” requirements for steam boilers serve to control steam pressure within ±1% of the desired pressure; air/fuel ratio within ±2% of excess air (±0.4% of excess oxygen), based on a desired “load” vs. “excess air” curve; steam drum water level within ±1 in. of desired level; and steam temperature (where provision is made for its control) within ±10°F (5.6°C) of desired temperature. In addition, the efficiency of the boiler should be monitored within ±1%.

In order to reach these performance goals, it is necessary to install accurate sensors and to make sure that the load does not change more than 10–35% of full scale per minute, depending on the size, fuel type, boiler design, and that there are no boiler design problems limiting this ability. The various loops tend to interact, so that integration into an overall system is necessary both during design and when the loops are being “field-tuned.”
Flow Detectors  Important and often disregarded are the flow detectors, which provide the basis for both material and heat balance controls. Outlet steam flow measurement, particularly if it is used as part of the boiler firing rate control strategy, for environmental permit compliance, or for on-line efficiency or energy use calculations, should be pressure- and temperature-corrected to a true mass flow. Most steam flow sensors in use (orifice, flow nozzle, vortex-shedding meters, and so on) measure velocity, which translates directly to volumetric flow.

Multivariable transmitters with integrated pressure and temperature inputs and mass flow computation in the meter are preferred. If an uncompensated velocity or volumetric flow measurement, the process variable, is all that is available, then it should be compensated to a true mass flow measurement in the control system using Equation 8.6(2):

\[
F_c = F_A \left( \frac{P + P_R}{P_R} \right) \left( \frac{T_R}{T + T_0} \right) \left( \frac{X}{X_R} \right) \left( \frac{Q}{Q_R} \right)
\]

where

- \(F_c\) = compensated flow (mass flow)
- \(F_A\) = uncompensated flow (velocity or volumetric measurement)
- \(P\) = actual measured steam pressure
- \(P_R\) = reference pressure (pressure at which primary flow element was specified and sized, converted to same units as \(P + P_0\))
- \(T\) = actual measured steam temperature
- \(T_0\) = conversion to Rankine or Kelvin scale (459.69 for \(^\circ F\) to \(^\circ R\), 273.15 for \(^\circ C\) to \(^\circ K\))
- \(T_R\) = reference temperature (temperature at which primary flow element was specified and sized, converted to same units as \(T + T_0\))
- \(X\) = measured actual steam compressibility
- \(X_R\) = reference steam compressibility (at conditions for which primary flow element was specified and sized)
- \(Q\) = measured actual steam quality
- \(Q_R\) = reference steam quality (at conditions for which primary flow element was specified and sized)

In practice, the compressibility and quality terms are often dropped, lacking a good measure of the actual values for these. In most steam flow applications, the pressure compensation term is the most important one. Note also that the above equation assumes that, for a differential pressure type flow measurement, the process variable, \(F_c\), has already had square root extraction applied to convert it to velocity.

For successful control of the air/fuel ratio, combustion air flow measurement is important. In the past it was impossible to obtain ideal flow detection conditions. Therefore, the practice was to provide some device in the flow path of combustion air or combustion gases and to field-calibrate it by running combustion tests on the boiler.

These field tests, carried out at various boiler loads, used fuel flow measurement (direct or inferred from steam flow) and measurements of percentage of excess air by gas analysis; they also used the combustion equations to determine air flow. Because what is desired is a relative measurement with respect to fuel flow, the air flow measurement under these circumstances has historically been calibrated and presented on a relative basis.

Flow vs. differential pressure characteristics, compensations for normal variations in temperature, and variations in desired excess air as a function of load are all included in the calibration. With this traditional approach, using relative measurements, the desired result is to have the air flow signal match the steam or fuel flow signals when combustion conditions are as desired.

The following sources of pressure differential are normally considered:

- Burner differential (windbox pressure minus furnace pressure)
- Boiler differential (differential across baffle in combustion gas stream)
- Air heater differential (gas side differential)
- Air heater differential (air side differential)
- Venturi section or flow tube (installed in stack)
- Piezometer ring (at forced-draft fan inlet)
- Venturi section (section of forced-draft duct)
- Orifice segments (section of forced-draft duct)
- Airfoil segments (section of forced-draft duct)

Of these, the most desirable are the last four, because they use a primary element designed for the purpose of flow detection and measure flow on the clean-air side. Some of these typical traditional installations are shown in Figure 8.6c; none of these sensors meet the dual requirement of high accuracy and rangeability. In fact, they are of little value at 30% flow or less.

**Flow Sensor Accuracy**  Table 8.6d lists some better flow sensors, such as the multipoint thermal flow probe or the area-averaging pitot stations provided with “hexcel”-type straightening vanes and with membrane-type pressure balancing \(dP\) cells. Area-averaging pitot stations are also available in two-dimensional arrays, in a circular or rectangular form.
mounting, that can average the entire flow-field. Thermal (hot-wire anemometer) sensor arrays can also be fairly accurate, offer good turndown, and produce a signal in direct proportion to mass flow. These represent major advances in combustion air flow detection.

This table shows the measurement errors that can be anticipated. Unfortunately, as the flow is reduced, the error—in percentage of actual measurement—increases in all cases except the first two. With linear flowmeters, the error increases linearly with turndown of 10:1. In case of nonlinear flowmeters, the error increases exponentially with turndown. Therefore, at a turndown of 10:1, the orifice or pitot error increases 100-fold and causes these devices to become useless. This situation can be alleviated somewhat by the use of two d/p cells on the same element or by the use of “smart” d/p cell.

Based on the data in Table 8.6d, if the boiler efficiency is to be monitored on the basis of time-averaged fuel and steam flows, the lowest error that can be hoped for is around 1%.

Similarly, the air/fuel ratio cannot be measured to a greater accuracy than the air flow. At high turndown ratios, this error can be very high. Considering that a 2% reduction in excess oxygen will increase the boiler efficiency by 1%, both the accurate measurement and the precise control of air flows are essential in boiler optimization. If combustion air temperature and pressure vary significantly, then air flow measurement can also be pressure- and temperature-compensated to a true mass flow, again, either with multivariable transmitters, or with Equation 8.6(2), dropping the compressibility and quality terms.

The impact of sensor inaccuracy on performance optimization elsewhere is not as critical. Standard instrumentation allows for the control of steam pressure within ±1%, furnace pressure within ±0.1 in. H2O (25 Pa), water level within ±1 in. of desired level, and steam temperature to within ±10°F (5.6°C).

Inferential Measurements Inferential measurements, or soft sensors, are finding application in boiler measurement and control due to the difficulty or expense in directly measuring many of the important process variables related to operation of a boiler. Variables such as NOx emissions, steam rate, and turbine shaft temperature have been successfully measured inferentially. Two techniques that have successfully been employed are neural networks and principal component analysis. See Section 2.18 in Chapter 2 of this volume for discussion of neural networks. Regulatory agencies have accepted neural network-based emissions measurement and reporting in certain parts of the United States.

### Table 8.6d
Flow Sensor Errors on Boilers

<table>
<thead>
<tr>
<th>Flow Streams Measured</th>
<th>Type of Flowmeter</th>
<th>Inaccuracy (% of flow)</th>
<th>Rangeability Limitation</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>At 10%</td>
<td>33%</td>
</tr>
<tr>
<td>Fuel (oil)</td>
<td>Coriolis mass flow</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td>Fuel (natural gas)</td>
<td>Coriolis</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td></td>
<td>Thermal</td>
<td>5.75</td>
<td>2.27</td>
</tr>
<tr>
<td></td>
<td>Turbine</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td></td>
<td>Ultrasonic</td>
<td>0.1–1.0</td>
<td>0.1–1.0</td>
</tr>
<tr>
<td>Fuel (solid: coal, wood, etc.)</td>
<td>Gravimetric feeders or belt scales</td>
<td>0.25–0.5</td>
<td>0.25–0.5</td>
</tr>
<tr>
<td>Fuel (pulverized coal)</td>
<td>Coriolis</td>
<td>0.5</td>
<td>3:1</td>
</tr>
<tr>
<td></td>
<td>Microwave</td>
<td>5.0</td>
<td>5:1</td>
</tr>
<tr>
<td>Steam and water</td>
<td>Vortex shedding</td>
<td>1–1.5</td>
<td>1–1.5</td>
</tr>
<tr>
<td>Steam</td>
<td></td>
<td>Min. R5 = 20,000</td>
<td></td>
</tr>
<tr>
<td>Water</td>
<td></td>
<td>Max. temp = 750°F (400°C)</td>
<td></td>
</tr>
<tr>
<td>Orifice</td>
<td>NG*</td>
<td>2–5</td>
<td>0.5</td>
</tr>
<tr>
<td>Air</td>
<td>Area averaging pitometer traverse station</td>
<td>NG</td>
<td>2–10</td>
</tr>
<tr>
<td></td>
<td>Multipoint thermal</td>
<td>5–20</td>
<td>2–5</td>
</tr>
<tr>
<td></td>
<td>Dual-range unit required</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Piezometer ring, orifice segment</td>
<td>NG</td>
<td>3–20</td>
</tr>
<tr>
<td></td>
<td>Cannot be used below 25% of max flow</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*NG = Not Good.
Safety Interlocks

Many of the interlocks related to the start-up, shutdown, and operation of a boiler are implemented for the purposes of protecting personnel and equipment. Most of the interlock and safety features directly related to the boiler can be classified as either burner management or combustion control. This delineation is made because boiler safety standards define very specific functions for burner management and require it to be implemented in a dedicated system, separate and apart from other control functions.8

A burner management system (BMS) is primarily concerned with the interlock, sequence, and timing functions required to safely put burners into service and to stop fuel and trip the boiler on detection of potentially unsafe conditions (master fuel trip). Other combustion control interlocks and protection functions, not necessarily a part of BMS, include furnace draft (implosion protection) control, fuel/air cross-limiting, and “runbacks.”

An overview of some of the most common boiler safety interlocks is as follows:

- **PURGE INTERLOCK** Prevents fuel from being admitted to an unfired furnace until the furnace has been thoroughly air purged.
- **LOW AIR FLOW INTERLOCK OR FAN INTERLOCK** Fuel is shut off upon loss of air flow or combustion air fan or blower.
- **LOW FUEL SUPPLY INTERLOCK** Fuel is shut off upon loss of fuel supply that would otherwise result in unstable flame conditions.
- **LOSS FLAME INTERLOCK** All fuel is shut off upon loss of flame in the furnace, or fuel to an individual burner is shut off upon loss of flame to that burner.
- **FAN INTERLOCK** Stops forced draft upon loss of induced-draft fan.
- **LOW WATER INTERLOCK (OPTIONAL)** Shuts off fuel on low water level in boiler drum.
- **HIGH COMBUSTIBLES INTERLOCK (OPTIONAL)** Shuts off fuel on highly combustible content in the flue gases.

Where fans are operated in parallel, an additional interlock is required to close the shut-off dampers of either fan when it is not in operation. This is necessary to prevent air recirculation around the operating fan.

Burner Management Systems BMS interlocks must be implemented with dedicated systems. They can be accomplished by hard-wired relay logic, solid-state logic, or programmable logic controllers (PLCs). The BMS is considered a safety instrumented system (SIS). Therefore, if PLC technology is used, it is often based on 1oo2 (one-out-of-two) or 2oo3d (two-out-of-three with diagnostics) logic.

The first one has two channels (two independent CPUs); the second has three channels (three independent CPUs, as in triple modular redundant systems). The criterion for selecting a certain type of SIS equipment is based on a safety integrity level (SIL) assessment that determines the degree of integrity required of the SIS based on probability and impact severity of risks. The Instrumentation, Systems, and Automation Society (ISA) has published detailed standards on SIL assessment.

The minimum interlocks required by the National Fire Protection Association (NFPA) for basic furnace protection for a multiple-burner boiler are illustrated in Figure 8.6e. An important part of the burner logic is the purge system, illustrated at the bottom of Figure 8.6e. Master fuel trip cannot be reset; i.e. the burner light-off sequence cannot be started unless a proper purge sequence has been executed. The purge helps prevent accumulation of unburned fuel in the boiler after trips or failed start sequences. Furnace explosions are often related to accumulation of unburned fuel.

Automatic start-up sequencing for lighting the burners and for sequencing them in and out of operation is common. Timing for a portion of the typical light-off sequence for a 350 MW dual-fired (oil and coal) burner with gas igniter is illustrated in Figure 8.6f.

Combustion Control Safety Features Additional safety features required by code that are considered the responsibility of the combustion control system (as opposed to the BMS) are primarily aimed at:

1. Maintaining proper combustion zone conditions (air and fuel, air/fuel ratio) to support complete and safe combustion
2. Preventing furnace implosion

Fuel demand should never exceed the capability of the combustion air system to supply necessary air for complete combustion. If an ID or FD fan trips in single-fan systems, then a master fuel trip occurs. If there are fans in parallel, as in large electric utility units, then runbacks are employed in the control logic. This technique applies to any of the critical boiler equipment that can be operated in parallel: FD and ID fans, feedwater pumps, and so on. In these cases, interlocks are provided that “run back” the boiler firing rate to an operating point that can safely be supported by only one fan or pump when one of a pair fails. For example, if one of a pair of FD fans fails, the boiler firing rate must, in a rapid controlled fashion, be brought down to a point that the air supplied by the one fan is more than adequate for complete combustion.

Another feature to ensure sufficient air/fuel ratio for complete combustion is air/fuel cross-limiting. This is a safety feature guaranteeing that no change in the firing rate (up or down) can result in a “fuel-rich” mixture. The consequence of this feature is that, on increasing firing rate, air increases before fuel, and, on decreasing firing rate, fuel decreases before air (greater than or equal to the sum of the stoichiometric plus minimum excess air requirement for the fuel being burned).

If the FD and ID fan capacities, combined with the total dynamic head characteristics of the entire air and gas path,
FIG. 8.6e
Interlock system for multiple burner boiler. (Reprinted with permission from NFPA 85-2004, “Boiler and Combustion Systems Hazards Code,” copyright ©2004, National Fire Protection Association, Quincy, MA 02269. This reprinted material is not the complete and official position of the NFPA on the referenced subject, which is represented only by the standard in its entirety.)
can produce a draft or pressure that will exceed the design pressure ratings of the boiler or its associated ductwork, then the furnace pressure control system becomes more critical. In these cases, the furnace pressure control strategy should include features such as redundant furnace pressure transmitters, feedforward action from master fuel trip, and override action or directional blocking on large furnace draft error (Figure 8.6f).

Design and specification of the various safety interlocks are largely guided and governed by insurance company regulations, standards bodies such as the NFPA, and state regulations. NFPA 85 specifically addresses boilers. Insurance company standards
Control and Optimization of Unit Operations

Soot Blowers

Soot blowers are standard equipment on nearly all types of large water-tube boilers. Soot blowing equipment is used to periodically blow off deposits (fouling) that accumulate on the tubes on the inside of the boiler as a result of the combustion process. A clean heat-transfer surface plays a key role in achieving high thermal efficiency.

Steam, compressed air, or water have all been used as a blowing medium, with steam being the most common. Soot blowers installed in the furnace wall area are short fixed-length blowers (so-called IR blowers). Water lances are sometimes used for the furnace wall region, as well. The blowers located in the convection area, economizer area, and air heater area are long and retractable blowers (so-called IK blowers).

Recently, the water cannon has been introduced to improve cleaning on the furnace wall section. Compared to a regular water lance, the water cannon has the following advantages: less maintenance work, higher cleaning efficiency, reduced NOₓ, and longer tube life. Water is never used as a blowing medium, however, in black liquor recovery boilers (used in the pulp and paper industry) due to the potential for explosion from a smelt-water reaction.

Further, proper operation of soot blowers in recovery boilers becomes that much more critical due to the possibility of soot blower steam, directed at a localized area of tubes for too long, causing erosion-induced tube failure and, again, the catastrophic consequences of a smelt-water reaction from the failed water tubes.

In operation, soot blowers are often grouped into sequences based on their physical locations in the boiler. Each sequence consists of multiple steps. Each step may involve several blowers that can run simultaneously. Depending on the blowing medium limitations, normally three or four wall blowers can run at the same time, and one to two retractable blowers can run at the same time.

The operator can run sequences in a fixed schedule, or selectively pick up any individual sequence to run. An individual blow can always be selected whenever needed. Before any soot blower can start to run, the control system logic will first check if all permissives are met. The permissive conditions include whether the requested blower is in service, whether the required blowing medium is available, and whether the medium pressure is adequate.

If all starting permissives are met, the requested blower will start to run. In the retractable blower case, the blower travels forward to the end and reverses back. During the run time, the blower head pressure and the flow of the blowing medium have to be monitored throughout. A significant pressure or flow drop should cause the blower to stop and retract.

The control logic has to be programmed such that whenever abnormal conditions occur, an alarm signal is sent out and the troubled blower stops blowing and, in the IK blower case, fully retracts immediately. The most common failures are the following:

- Fail to start: A blower has been commanded to start, but there is no indication that it actually started within an expected time interval.
- Blow fail: Blowing medium pressure has dropped below an acceptable level for a specified time interval.
- Motor overload or stall: The motor current of a retractable blower has exceeded its normal level by a set amount for a specified time period.
- Elapsed time: A blower remained away from its rest position beyond a specified time.

FIG. 8.6g
System requirements for furnace pressure protection and control when such pressure or vacuum can be applied that exceeds boiler or duct pressure ratings. (Reprinted with permission from NFPA 85-2004, “Boiler and Combustion Systems Hazards Code,” copyright ©2004, National Fire Protection Association, Quincy, MA 02269. This reprinted material is not the complete and official position of the NFPA on the referenced subject, which is represented only by the standard in its entirety.)

may be more stringent than industry trade and professional organization (such as NFPA) standards.
Boiler Dynamics

Boiler response to load changes is usually limited by both equipment design and dead time considerations. Usually the maximum rate of load change that can be handled is from 20 to 100% per minute. This limitation is due to the maximum rates of change in burner flame propagation and to the “shrink/swell” effects on the water level. The period of oscillation of a typical boiler is between 2 and 5 min.

This is the result of a dead time of 30–60 sec and the integrating effect from the storage of energy (similar to tank level). The transportation delay in the boiler is partially due to the displacement volume of the furnace. For example, if the air/fuel ratio is changed, the furnace volume will have to be displaced before the flue-gas composition can reflect that change. The lower the air flow (the lower the load), the longer it will take to displace this fixed volume. Therefore, dead time increases as load is lowered on a boiler.

The transportation delay described above is only one component of the total dead time. The oxygen analyzer also contributes to the total delay, because of its location and its fly ash filter. In addition to the dead time contribution of instruments, control dead time is also created by the fuel/air cross-limiting.

Another way to reduce dead time and thereby increase boiler response is by using feedforward loops. The firing rate demand signal can be made more responsive by feedforward of steam flow, which responds to load changes faster than does steam pressure. Similarly, the induced-draft loop can be made more responsive by adding feedforward off the forced-draft position (damper, inlet vanes, or fan speed). In this system, as soon as the air flow into the furnace changes, the outflow is also modified in the same direction, so that furnace draft pressure is relatively unaffected.

Each of these loops will be discussed in some detail in the following paragraphs.

Air/Fuel Ratio Controls

Performance of air controls on traditional boilers has been limited by inaccurate and unreliable air flow sensors, particularly when air flow rates were less than 25% of maximum rates. This was unfortunate, because it is precisely at low loads that the boiler tends to be the least efficient to start with. Yet, for reasons of equipment inadequacy, some manufacturers will turn off the oxygen trim of the air/fuel ratio at low loads. Contributing to these problems were inaccurate sensors at low flows; leaking, nonlinear dampers with hysteresis and dead band; the use of constant-speed fans; and flame instability at low loads for certain boiler designs and fuels.

Legacy boiler control systems often have workarounds for such historical combustion air measurement and control challenges. In many cases, the firing rate signal itself has been characterized to set the excess air (Figure 8.6h); it has also been used as the oxygen set point on air/fuel ratio trim controls. Frequently, the excess air requirement curve was directly calibrated into the combustion system, and feedback trim based on excess oxygen was not used at all.

The advocates of this open-loop control strategy argue that the predetermined excess oxygen curve is rather permanent, and if an unmeasured effect necessitates a change in it, that change will be the same throughout the full firing range. This view of the process neglects nonlinearities, hysteresis, dead time, the play in linkages, sticking dampers, and other effects that are now understood.

In many traditional systems, the nonlinearity of dampers (Figure 8.6i) was taken into account by characterizing the fuel valve, the linkage, or the signal to the fan damper actuator. This is no easy task, because louver-type dampers are not only nonlinear, they also lack repeatability. In these systems, the air/fuel ratio trim was disabled below 25% load, because the conventional dampers could not be controlled.

It was argued that closed dampers leak as much as 10% of their full capacity and thus need to be opened only to 2% to deliver 25% flow. Therefore, according to this argument, if the oxygen trim signal were not disabled and it did request a 1% change in air/fuel ratio, this would mean a 1% change in the 2% opening of the damper, or a 0.02% change in its stroke or rotation; this the damper positioner could not handle. Naturally, the argument is correct in all respects except in its assumption that such dampers are a necessary limitation.

Other observations include that PID-type controls cannot handle frequent load changes, because the speed of response of the fuel flow control loop is much faster than that...
of the air flow control loop. Suppliers attempted to correct this by making the air actuator twice as fast as the fuel actuator.

In general, the nonlinear nature of the loops once created difficulties for the traditional boiler control designs. The need to change the firing rate in exact proportion to load changes was difficult to meet, because it required characterization in the field of the fuel valve and air damper actuators.

As will be discussed on the following pages, distributed microprocessor-based control systems make it much easier to linearize the nonlinear systems through characterizers and greater use of closed-loop control. It is also possible to memorize actuator dead band and speed of response. Still, the capability of state-of-the-art control equipment should not be used as a justification for installing inferior-quality dampers.

**Dampers** The recommendations for the use of accurate and high-rangeability sensors have already been made in connection with Table 8.6d. Now the desirable damper features will be discussed from a control performance viewpoint.

The reasons why dampers are undesirable as final control elements include their hysteresis, nonlinearity, and leakage. If the flow is accurately measured, the consequences of these undesirable features are easier to handle. For example, assuming that even under proper maintenance the damper displays some hysteresis or dead band, with a reliable flow sensor, the opening and closing characteristics of the damper can be determined.

In most digital control systems, this characterization curve can readily be accommodated, usually input as a series

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**FIG. 8.6i**

Multiple-leaf louvers with dividing partitions were used in many traditional systems.}

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Figure 8.6i shows the nonlinear characteristics of a multiple-blade damper. Curve A on the bottom gives the relationship between damper rotation and open area, while curve B relates damper rotation and flow coefficient. The upper portion of Figure 8.6i describes the installed performance of the same damper, considering its pressure drop relative to the total system pressure drop.

For example, if the damper resistance is 25% of the total, when the damper rotation is 45%, curve #7 will give the damper characteristics, and actual air flow will be about 88% of maximum. From such curves as these, both the damper leakages and the nonlinearities can be estimated before actual installation.

It can be seen from Figure 8.6i that an increase of 1° in damper rotation that occurs at a 10° opening causes a much greater increase in the actual air flow than a 1° increase that occurs at a 60 or 70° opening. This is undesirable. Stable controls would require constant gain in the loop. Ideally, the change in air flow per increment change in damper opening should be uniform from the tightly closed to the wide-open position. Such inherently linear dampers are hard to manufacture, although the design illustrated in Figure 8.6j does hold such potentials. A more often used solution is the use of compensators and characterizer positioners, so that as the damper gain drops off—as it opens—their compensators introduce more gain into the loop, keeping its total gain nearly constant.

While it is possible to compensate for nonlinearity and hysteresis, leakage must be eliminated by selecting the correct design. Figures 8.2h and 8.6j illustrate some of the tight shut-off designs. If these are used, it is no longer necessary to turn off the excess oxygen-based air/fuel ratio trim when the load is below 25%. With such compensated, low-leakage dampers, trim need not be turned off, and the resulting increase in efficiency need not be abandoned, until load drops to 10%.

Nonlinearity and hysteresis, in the form of stiction and backlash, are often directly the result of the actuator itself. Electric damper drives have proven superior to pneumatic and hydraulic cylinder actuators and are becoming the norm for damper modulation.

**Fans**

Even if the best damper is used, damper control will always have a disadvantage, namely that it burns up fan energy in order to control. Therefore, the best method of controlling air flow is at the fan: either by a variable-speed fan or by adjustable inlet vanes. By varying fan speed and eliminating a damper (or running it wide open), it is possible to avoid unnecessary energy loss in the form of damper pressure drop.

Variation of fan speed is accomplished with hydraulic or magnetic couplings, as well as variable-speed AC and DC electric drives. Adjustable inlet vanes, because they are integrated right into the suction area of the fan, are also much more efficient than dampers, though still not as efficient as varying the fan speed. Fan characteristic curves and a comparison of damper vs. inlet-vane vs. variable-speed control are illustrated in Figure 8.6k.

In addition to reducing the overall cost of boiler operation by conserving air transportation energy, variable-speed and variable inlet-vane fans offer better linearity, hysteresis, dead band, and leakage characteristics. For these reasons, the overall control diagram in Figure 8.6l shows fan speed or inlet-vane controls instead of damper controls.

**BASIC BOILER CONTROLS**

Figure 8.6l shows a possible configuration of the basic boiler control loops and the tie-in points for optimization. Although this is a well-designed control system, it is just one of many possible configurations. Boiler size, steam pressure, and number and type of fuel(s) all can vary, necessitating variations in this scheme. The major loops shown in Figure 8.6l are numbered.

**Boiler-Pressure and Firing Rate Controls**

Realizing that the boiler is part of a larger plant system, consideration must be given as to how the boiler output will match the load demand placed upon it by the system. From a control perspective, it is useful to distinguish between electric utility and non-electric utility (i.e., industrial) boiler applications.

The firing rate of industrial boilers is most often manipulated to control to a constant header pressure. The primary controller that accomplishes this, whose output is firing rate, is usually termed the boiler master. In an industrial application,
multiple boilers may connect to one steam header, from which multiple turbine generators may be supplied as well as plant heating loads, through extraction turbines or pressure reducing valves (PRVs). In contrast, an electric utility boiler normally operates paired off as a single generation unit with a single turbine generator. The electrical output desired from the unit can be controlled in the following ways:

- **Boiler-following mode**: load demand is controlled first at the turbine generator, with the boiler supplying (following) whatever the turbine requires (header pressure control on boiler firing rate).
- **Turbine-following mode**: load demand is controlled first by boiler firing rate, with the turbine responding to (following) the boiler (throttle pressure control with turbine valves).
- **Boiler-turbine coordinated control mode**: turbine valves and boiler firing rate are manipulated in concert in response to load demand, while header pressure is maintained.
- **Sliding (variable-pressure) and free-pressure control modes**: control strategies that, by design, allow header pressure variation with load, to improve efficiency, speed of response to load changes, and turbine reliability.

**FIG. 8.6k**
Axial-flow fans with variable-pitch control help eliminate the need to burn up unneeded air transportation energy in the form of damper pressure drops. Fan characteristics are shown with (a) damper control, (b) variable-inlet vane control, and (c) variable-speed control.
This diagram shows good boiler controls without optimization.
Turbine following is the most stable, but slowest mode; boiler following is faster, but limited to a maximum of about 2.5%/min for a controlled load change. The coordinated control mode should be capable of 5%/min or better. Most control systems on large electric utility units are configured to allow selection from among the first two or three of these control modes, and, on some units, all four (some specific variant of the fourth scheme). Additionally, electric utility units usually have provision for the load demand to be generated remotely, referred to as remote dispatch.

A general, simplified structure for load demand control on a typical utility unit is illustrated in Figure 8.6m. The frequency input shown is sometimes necessary as an additional correction to prevent feedback control on turbine generator power (MW) or load demand (e.g., first-stage pressure) from counteracting the turbine generator’s own power regulation response to frequency variations when connected to the electric power grid.

**Firing Rate** The firing rate signal becomes the set point for the fuel and air flow controls that make up the combustion control system. Additionally, air/fuel ratio controls provide for the correct stoichiometric amount of air to combust the fuel plus some percentage of excess air to account for nonideal mixing and imperfect combustion conditions. For safety purposes, fuel addition should be limited by the amount of available combustion air, and combustion air may need minimum limiting for flame stability.

Boiler outlet steam pressure is an indication of the balance between the inflow and the outflow of heat between energy supply and load. Therefore, by controlling the steam pressure, one can establish a balance between the demand for steam (process load) and the supply of steam (firing rate). A change in steam pressure will result from a change in firing rate only after a delay of a few seconds to a minute, depending on the boiler and the load level. Therefore, as will be examined in more detail, feedforward control, from load demand, is frequently employed to improve pressure control by adjusting firing rate (fuel and air) as soon as a load change is detected, instead of waiting for pressure to change first.

When more than one boiler is operated from the same master controller, the ability to individually bias and take control of each boiler should be provided, in addition to the ability to bias the master controller signal up or down when in automatic (Figure 8.6n).
When steam pressure is controlled by other means, steam flow can be the master controller. If variations in fuel heating value are minor, the master flow controller shown might be eliminated and the master load signal generated by a manual loading station.

Situations may arise when it is desirable to have either flow or pressure control. In these cases, a master control arrangement, as shown in Figure 8.6o, can be used. Although it may appear simpler to switch transmitters, it is desirable to transfer the controller outputs so that the controller does not have to be returned each time the measurement is switched, and to make provision for initialization and bumpless transfer.

Typical proportional gain tuning setting for a pressure-controlled “master” is 6.25 (16% proportional band) and the typical integral setting is about 4.0 min (0.25 repeats per min). For a flow control “master” the comparable settings might be 1 for controller gain (100% proportional band) and 0.33 min for integral time (3.0 repeats per min). (See Sections 2.35 through 2.38 in Chapter 2 for details on controller tuning.)

A flow control master can be used for firing rate control on each of multiple boilers connected to a common header, with the flow controllers cascaded to the header pressure controller (Figure 8.6p).

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**FIG. 8.6n**
Load sharing controls among several boilers.

**FIG. 8.6o**
Boiler control with alternative pressure or flow master.

**FIG. 8.6p**
On multiple boilers connected to a common steam header, the header pressure controller can be the cascade master of a number of flow controllers, measuring the steam flows from the individual boilers.
Feedforward Control and Load Demand  
To improve the speed of response to load changes, a signal that represents load can be fed forward directly to firing rate. In the case of multiple boilers supplying steam to a header, load can be the total steam flow measurement from all boilers (Figure 8.6q) or the output of a high selector, selecting the highest of the steam flows. Steam flow measurements used in feedforward control should use mass flow detectors, either from a multivariable transmitter that computes mass flow or from a mass flow sensor; see Equation 8.6(2) and the paragraph titled The Role of Sensors.

If a boiler’s own outlet flow is used as the load to feed forward to firing rate, the result will be positive feedback due to the impact of firing rate on flow. Used in this scenario, steam flow is called a regenerative load signal, and it will tend to cause instability and poor control. Even when combined with other steam flow measures (by average or high select) there will remain some amount of positive feedback.

On large electric utility boilers, due to the high temperatures and pressures, and large line sizes, steam flow measurement is often not provided at all. In that case, a good surrogate measure of load is the steam turbine first-stage shell pressure. The flow through the turbine varies almost linearly with first-stage shell pressure (Figure 8.6r).

In some cases (not to include sliding mode or free pressure control), it is also possible to use turbine inlet governor valve positions as an indication of load. However, first-stage shell pressure is generally preferred. These surrogate load parameters will behave more like nonregenerative signals, less coupled to boiler firing rate than boiler outlet steam flow. Therefore, even if steam flow is available it will be preferable in single-boiler arrangements to incorporate a surrogate load measurement for the purpose of feedforward firing rate control.

Feedforward Control and Stability  
The actual steam flow at any point is not necessarily a true indication of demand. For example, an increase in steam flow caused by increased firing should not be interpreted as a load increase—this would create a positive feedback loop, capable of destabilizing the boiler. According to Shinskey, the true load on a boiler can be approximated by $\sqrt{h/p}$, where $h$ is the differential developed by a flow element and $p$ is the steam pressure.

Figure 8.6s describes a boiler-pressure control system using this type of feedforward model. Fuel flow is set proportionally to the estimate of load $\sqrt{h/p}$. Dynamic compensation is applied in the form of a lead-lag function to help overcome the heat capacity in the boiler.

The pressure controller adjusts the ratio of firing rate to estimated load, to correct for inaccuracy of the model and variations in heat of combustion of the fuel. The multiplier also changes the gain of the pressure feedback loop proportional to load. This feature is valuable in that boilers seem more difficult to control at lower loads because of lower velocities.

In the firing rate control system shown in Figure 8.6s, the loops 101 and 102 measure the pressure and flow of the generated steam, respectively. The flow transmitter (DPT-102)
is a high rangeability and linear device, capable of accurate measurements even at low loads. The firing rate demand signal is generated in a feedback manner by the pressure controller PIC-101.

In order to speed up the response of this loop to load changes, a feedforward trim is added. This trim is based on steam flow, because this flow is the first to respond to load changes. Therefore, as soon as the demand for steam changes, FY-102 will trim the firing demand signal, without waiting for the steam pressure to change. The dynamics of FY-102 are adjusted to reflect the time constants of the boiler, recognizing the time displacement between a change in firing rate and the resulting change in the rate of steam generation some time later.

**Fuel Controls**

**Measurable Fuels** The primary boiler fuels are coal, oil, and gas, but there are a large variety of auxiliary fuels, such as waste gases, waste sludges, and waste wood products (bark, sawdust, hogged fuel, and coffee grounds). In many cases, these auxiliary fuels are dumped to the boiler plant on an uncontrolled basis for immediate burning. There are myriads of these combinations, and only the more common fuel control problems will be covered in the discussion to follow.

The control of gas and oil fuels tends to be straightforward, because they are easily measured and can be regulated with a control valve in the fuel line. If closed-loop control of flow is not available, then a valve positioner capable of providing a linear relationship between flow and control signal is desirable (Figure 8.6).
A flow control loop (preferred) is the more usual means of control, providing more precise linearization and immediate response to flow disturbances (Figure 8.6u). In cases in which better than 3:1 turndown and high measurement accuracy is desired, the Coriolis mass flow sensors should be used instead of orifices.

When it is desired to fire the fuels in a predetermined ratio to each other regardless of load, a manually adjustable signal splitter can be used, as shown in Figure 8.6v. The most precise and complex method of rarioing fuel (not shown) is to split the demand signal and send that to individual flow control loops. If the fuel is a gas at variable pressure, a pressure control valve is frequently installed upstream to the flow sensor, as shown in Figure 8.6v.

Both valves affect both variables (pressure and flow), and therefore they will interact. In order to eliminate the resulting oscillations, one should either leave the pressure unregulated and pressure-compensate the flow sensor or assign less pressure drop to the pressure control valve than to the flow control valve (thereby using a larger valve for pressure control than for flow control). Because the burner back-pressure will increase as flow increases, the available pressure differential for the flow control valve will decrease as the flow rises. In order to obtain an approximately linear relationship between fuel flow and valve position, an equal-percentage valve is needed.

In the case of oil fuels, proper atomization at the burner, and therefore complete combustion, will be achieved only if the oil is kept at constant pressure and viscosity. When heavy residual oils (e.g., no. 2 and no. 6 fuel oil) are burned, they must be continuously circulated past the burner and back (Figure 8.6w).

The difference between the readings of two Coriolis mass flowmeters indicates the net flow to the burner. The burner back-pressure is controlled by the control valve in the recirculating line, whereas the flow controller set point is adjusted by the firing rate demand signal. The firing rate is controlled by an alteration in the opening of the burner orifice. Atomizing steam is ratioed to the firing rate, and the heating steam is modulated to keep the fuel viscosity constant.

Figure 8.6x illustrates the controls required when the fuel demand is split between two fuels on a closed-loop (automatic)
basis. The instruments shown ("biasing stations") provide the means of manual control plus the ability of automatic control, with bias of one fuel with respect to the other.

Because one of the requirements ultimately is to have fuel ratioed to combustion air, any totalization of fuel for control purposes should be on an "air required for combustion" basis. If totalization is needed on any other basis, such as BTU for other purposes, a separate totalizer should be used.

**Waste or Auxiliary Fuel Controls** When auxiliary fuel is burned on an uncontrolled-availability basis, the fuel and air control system needs to be able to accommodate sudden changes in auxiliary flow without upsetting the master controller. The master controller should be designed and used to respond to total load demands only and not to correct for fuel upsets. A typical fuel control system for accommodating variations in auxiliary fuel without upsetting the master is shown in Figure 8.6y.

In the basis system without auxiliary fuel, the signal is relayed directly to the control valve. Addition of auxiliary fuel shifts the primary fuel control valve opening to prevent fuel variations from affecting overall boiler performance. A more precise system is shown in Figure 8.6z. Here, the flow controller adjusts the primary fuel control valve to satisfy total fuel demand and prevents auxiliary fuel variations from upsetting the master controller.

Figure 8.6aa describes a slightly more advanced system, in which the allowable maximum percentage of waste fuel that can be burned is set on the ratio relay FY-1. This ratio must be set under 100% if the heating value of the waste is so low that it could cause flameout if not enriched by supplemental fuel. For proper operation, the subtractor FY-2 must be scaled with the flowmeter ranges taken into consideration, and further scaling is required if the heat flow range of the total heat demand does not match that of the waste fuel flow-meter. When waste fuel gas availability becomes limited, waste fuel gas pressure will drop and PY-3 will select it for control, thereby overriding the waste flow controller. FY-2 will respond to this by increasing the supplemental fuel flow.

**FIG. 8.6y**
In this open-loop control configuration, the auxiliary fuel is burned on an uncontrolled-availability basis.

**FIG. 8.6z**
In this closed-loop control configuration the auxiliary fuel is burned on an uncontrolled-availability basis, while the flow controller throttles the primary fuel flow to meet the total demand.

**FIG. 8.6aa**
Automatic control system for burning limited-availability waste fuel up to preset maximum percentage in the total mixture.
Closed-loop control will give greater precision and better linearization, but the performance can be limited by poor hardware selection or poor installation practices. An example of the first case is the problem of transmitter rangeability if orifices are used instead of turbine or mass flowmeters. An example of poor installation practices is the case of long transmission lines being allowed to introduce dead time into the loop because pneumatic leads were installed without boosters.

Whatever type of fuel control is used, the maximum flexibility in design will be present if all flow signals are linear and all control valve characteristics are also linear. In this manner, the various flows and signals can be combined, subtracted, multiplied, or divided to produce the desired control. One optimal condition is to have the total fuel demand signal linear, with fuel totalized on a basis of required combustion air. The other desired end condition is to have the total fuel control capacity maximum at a value approximately 10% greater than that required for maximum boiler-capacity. This excess is necessary for control flexibility at maximum boiler load. Additional excess capacity should not be considered, because it reduces turndown capability.

**Unmeasured Fuels**

Coal can be an unmeasured fuel. In such cases coal control systems are open loop, wherein a control signal positions a coal-feeding device directly. This is the case with a spreader stoker or cyclone furnace or indirectly with pulverized coal.

A spreader stoker consists of a coal hopper on the boiler front with air jets or rotating paddles that flip the coal into the furnace, where a portion burns in suspension and the rest drops to a grate. Combustion air is admitted under the grate. There is no way to control fuel to a spreader stoker except in an open-loop manner by positioning a feeder lever that regulates coal to the paddles.

In pulverized coal-fired boilers, the coal is ground to a fine powder and is carried into the furnace by an air stream. There are normally two or more pulverizers (in parallel) per boiler. Pulverized coal flow is regulated at the pulverizer, and each manufacturer has a different design requiring different controls. One control arrangement is shown in Figure 8.6bb. Here, the primary air comes from a pressure fan that blows through the pulverizer, picking up the coal and transporting it to the furnace.

In addition to the controls shown, an air temperature control is required. In this loop, cold and hot combustion air is mixed ahead of the primary air fan to control the temperature of the coal-air mixture in the pulverizer. This control is necessary to maintain a maximum safe operating temperature in the pulverizer. This is a simple feedback loop, usually involving proportional control only.

A control arrangement for a bowl-type pulverizer is shown in Figure 8.6cc. In this type of pulverizer, the air fan sucks air through the pulverizers with the fan (called an exhauster fan) located between the pulverizer and the burners. The coal-air temperature loop is similar to the one described in Figure 8.6bb.

The control of a ball mill-type pulverizer is again different from a control standpoint. This is shown in Figure 8.6dd, including the application of manual compensation for the number of pulverizers in service.

The ball mill level measurement shown in Figure 8.6dd is normally based on the differential pressure value taken from the dip-tube. This measurement is inaccurate during start-up and during load swings due to large coal particle size. A more delicate control approach maintains mill motor
power levels around operator-selected set points. A sonic signal taken from a microphone is often required to assist the control. The underlying mechanism is that the amount of coal charge in the ball mill is functionally related to the mill motor power level and the mill noise level. The weight of the coal charge can be controlled directly through modulation of the associated mill feeder(s).

A simplified control flow diagram is illustrated in Figure 8.6ee. The system utilizes the mill power level (kilowatts), the mill sound level (decibels), and the operator-selected set point (kilowatts) as the principal inputs to the mill process controller (MIC-150). Boiler fuel demand (or steam flow) can be used by the control system as a feedforward signal to anticipate major changes in mill load demand, and to preposition the mill feeder(s) at the proper steady-state speed.

**Metered Coal Controls** When the flow of coal is controlled, it is done at the inlet to the mill, as illustrated in Figure 8.6ff. The capacity of the pulverizer contributes a delay of a few seconds to several minutes, depending on the design of the mill. Hammer mill delays are the shortest, whereas ball and roller mill delays can reach several minutes. If the delay is less than 1 min, a corresponding delay can be inserted into the coal flow transmitter output (FY-1), which will enable the loop to overcome this problem.

If the mill delay exceeds 1 min, the flow of the primary air that conveys the pulverized coal must be manipulated as a function of coal demand. Because the coal loading of the air is not uniform, Shinskey recommends the improvements noted in Figure 8.6bb. These include the determination of the...
actual heat release based on steam reading (FY-1 in
Figure 8.6bb) and the slow integral correction of the total
flow of primary air until the estimated and actual heat release
are matched.

**Highly Variable Fuels** Solid fuels, such as coal, biomass,
and municipal waste, tend to vary significantly in heating calorific value. This variation can be detected and com-
pen-sated for by comparing the apparent heat release of the boiler
process to the apparent heat input and adjusting the fuel flow
accordingly to achieve the desired heat input. One approach
to quantifying heat release is to sum the load indication (e.g.,
outlet steam flow or turbine first-stage pressure) with the rate
of change of pressure in the drum.

Figure 8.6bb shows the use of steam flow (FT-102) for
load. PT-101 and PY in Figure 8.6bb are the steam drum
pressure and its rate of change. In effect, this is summing the
rate of energy out with the rate of change of energy storage
(FY-1, Figure 8.6bb). This heat-release measure is then com-
pared to the current compensated fuel flow and the difference,
considered to be the error between the two due to heating
value variation in the fuel, is integrated slowly and multiplied
by the fuel flow, producing an adjusted process variable for
the fuel controller.

**Air flow Measurement and Control**

Combustion air for steam boilers may be supplied by induced
draft (suction fan at boiler outlet or stack draft), forced draft
(pressure fan at inlet), or a combination of forced and induced
draft known as balanced-draft fans. With balanced-draft boil-
ers, a slight negative pressure is maintained in the furnace.

In the control of combustion air (if there are both forced-
and induced-draft fans), one fan should be selected for basic
control of air flow and the other assigned to maintaining the
draft pressure in the furnace. The following discussion is
based on a single air flow source (fan) per boiler. Balanced
draft and its effects on air flow control will be covered later
in this section.

For successful control of the air/fuel ratio, combustion
air flow measurement is important; refer back to Boiler
Equipment, The Role of Sensors for detailed discussion on
air flow measurement issues and techniques (Figure 8.6b).

Primary flow elements in the ducts, as well as the furnace
pressure transmitter, will frequently produce noisy signals
because of pulsation from the pumping action of the fans or
from the combustion process. Provision should be made for
dampening the flow signals to facilitate more optimum tuning
of the controllers. Normal differential pressure ranges for
these measurements are between 1 in. (25.4 mm) H₂O and
6 in. (152.4 mm) H₂O for conventional sensors and as low
as 0.1 in. (2.54 mm) H₂O for area-averaging pitot stations.

**Damper and Fan Controls** Control devices for boiler air
flow control on pneumatic installations are double-acting
pistons, but in some cases electric motors are also used. In
either case, a linear relationship is required between control
signal and combustion air flow. Characteristics of most con-
stant-speed fans or dampers approximate those given in
Figure 7.1a in Section 7.1 in Chapter 7, and the desired rela-
tionship is linear.

The relationship between open- and closed-loop control
that was noted in connection with fuel control also applies
to air flow control. Closed-loop air control is more precise
and is self-linearizing, if the integrated system does not com-
pare air flow with fuel flow (or as inferred from steam flow)
directly in a ratio or difference controller.

Open-loop air control variations that may be used depend-
ing on the arrangement of fans are as shown in Figures 8.6gg
through 8.6kk. Figure 8.6gg illustrates a single-damper-con-
trolled open loop.

Figure 8.6hh shows a combination of damper and speed
control. A system of this sort is often necessary to increase
turndown (rangeability) where fan speed is variable. Good
response of air flow based on fan speed adjustment alone is
normally not attainable below approximately 1/3 maximum
speed. Depending on fan design, this may correspond to 50%
of boiler capacity. Use of a damper in combination with speed
adjustment allows further turndown, because fan speed is
blocked at approximately 1/3 speed. Split ranging, as shown
in Figure 8.6hh, conserves steam or fan power.

As a result of inlet damper leakage that is normally
present, it may be necessary for wide-range low-load or start-
up control to parallel inlet and outlet dampers. To save fan

![FIG. 8.6gg](image)

Open-loop air control with single damper.

![FIG. 8.6hh](image)

Combination damper and speed control to increase rangeability.

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power, the inlet damper may be operated over the full 3–15 PSIG (0.2–1.0 bar) range, whereas the discharge damper can be fully open at 3 PSIG (0.2 bar) and closed at 9 PSIG (0.6 bar).

As shown in Figure 8.6ii, when two fans are operated in parallel or on single-fan operation, the idle fan should have its damper closed to prevent recirculation from the operating fan.

When one fan of a two-fan system is switched on or off, considerable manual operation is required to prevent serious air flow upsets. The system shown in Figure 8.6ii eliminates this problem by automatically compensating the operating fan damper position as the parallel fan is started up or shut down. Separate discharge dampers may be used for shut-off purposes, supplementing the interlocks shown.

Closed-loop versions of the loops illustrated in Figures 8.6gg, 8.6hh, and 8.6ii would consist of flow controllers with air flow feedback superimposed on the components shown. For example, Figure 8.6jj shows the closed-loop version of Figure 8.6ii.

**Furnace Draft Control**

Whenever both forced draft and induced draft are used together, at some point in the system the pressure will be the same as that of the atmosphere. Balanced-draft boilers are not normally designed for positive furnace pressure. Therefore, the furnace pressure must be negative to prevent hot gas leakage. Excessive vacuum in the furnace, however, produces heat losses through air infiltration. The most desirable condition is thus one in which there is a very slight (approximately 0.1 in. H₂O, or 0.025 kPa) negative pressure at the top of the furnace.

Pressure taps for measuring furnace pressure may be located some distance below the top. Because of the chimney effect of the hot furnace gases, pressures measured below the top of the furnace will be lower by approximately 0.01 in. H₂O per foot (0.008 kPa/m) of elevation. Thus, if the pressure tap is 20 ft (6 m) below the top of the furnace, the desirable pressure to maintain is approximately 0.3 in. H₂O (0.075 kPa) vacuum.

In the case of a balanced-draft boiler, the maintenance of constant furnace pressure or draft keeps the forced and induced draft in balance. The purpose of this balance is to share properly the duty of providing combustion air and to protect furnaces not designed for positive-pressure operation.

The measurement of furnace draft produces a noisy signal, limiting the loop gain to relatively low values. In order to provide control without undue noise effects, it is desirable to use a full span of approximately 4–5 in. H₂O. This is normally a compound range, such as +1 to −4, or +2 to −3, in. H₂O (+0.25 to −1 kPa, or +0.5 to −0.75 kPa). Even with this span and with a set point of −0.1 to −0.3 in. H₂O (0.025–0.075 kPa), the controller gain can still not exceed 1.0. In some cases, it may be necessary to use integral control only to stabilize the loop.
Air Flow and Furnace Pressure Interaction  Additionally, stability problems and interactions may occur in the overall system because of measurement lags. It is recommended that the pressure transmitter connections to the boiler furnace be made with pipe at least 1 in. (25 mm) in diameter because of the very low pressures involved. If the distance is less than 25 ft (7.5 m), ½ in. (18.75 mm) pipe may also be used.

Either the forced- or the induced-draft fan can be used to control the furnace draft, with the other fan performing the basic air flow control function. Interaction cannot be completely eliminated between these two loops, but it can be minimized by system designs such as those shown in Figures 8.6kk and 8.6ll.

The common rule is that air flow should be measured and controlled on the same side (air or combustion gas) of the furnace to minimize interaction between the flow and pressure loops.

If the combustion air is preheated, then its temperature will vary substantially and compensation is needed. The mass flow of air is related to \(\sqrt{h/T}\), where \(h\) is a differential across a restriction and \(T\) is absolute temperature. This loop is shown in Figure 8.6ll, together with an excess oxygen trim on the air flow controller. Pressure and temperature compensation in engineering units can be carried out in the control system with Equation 8.6(2) (drop all but the temperature term if only temperature compensation is required).

The air flow and furnace pressures interact similarly to the phenomenon in Figure 8.6v. Because in the case of air flow controls the dampers are the same size and therefore their pressure drops are similar, decoupling needs to be applied. One might reduce interaction by connecting the two dampers (or fans) in parallel and using the furnace pressure as a trimming signal. (This can also be used to overcome problems resulting from noisy furnace pressure signals and slow response caused by the series relationship between flow and pressure loops.) This control system is illustrated in Figure 8.6ll.

In this arrangement the air flow controls move both dampers equally, and the furnace pressure corrects for any mismatch. The furnace pressure might respond faster to a change in the downstream damper opening, and therefore a dynamic lag (FY-104) is provided. In the combined control system shown in Figure 8.6l, basically the same control concept is implemented as shown in Figure 8.6ll. The main difference is that in Figure 8.6l, the furnace draft is throttled not by a discharge damper, but by an induced-draft fan.

The air flow is also detected by a high rangeability flowmeter (FT-104) that is compensated for temperature variations to approximate mass flow.\(^{14}\) The air/fuel ratio is adjusted by applying a gain to the air flow (FY-104). The continuous optimization of this ratio is one of the major tools of boiler optimization.

The air flow controller throttles the speed or inlet vanes of the forced-draft fan and the signal to the induced-draft fan in a feedforward manner. Dynamic compensation is provided by the lag module (FY-104). Thus, as soon as the air inflow to the furnace is changed, the outflow will also start changing. This improves the control of the furnace draft, as the feedback pressure controller (PIC-106) will need only to trim the feedforward signal at PY-106 to account for measurement and other errors.

Air/Fuel Ratio  When the controls for air/fuel ratio are considered, one point is very important. Because of the combustion gas velocity through the boiler, for safety reasons the fuel/air ratio should be maintained on an instant-by-instant rather than a time-averaged basis.

As a general rule (except for the case of very slow-changing boiler loads), fuel and air should be controlled in parallel rather than in series. This is necessary because a lag of only 1 or 2 sec in measurement or transmission will seriously upset combustion conditions in a series system. This can result in alternating periods of excess and deficient combustion air. Consequently, the discussion here will be limited to parallel air and fuel control systems.

The simplest control of air/fuel ratio is with a system calibrated in parallel, with provision for the operator to make manual corrections (Figure 8.6mm). In this system, the operator uses the bias provision of the panel station (FK) to compensate for variations in fuel pressure, temperature, or heating value or for air temperature, humidity, or other factors. A system of this sort should be commissioned with detailed testing at various loads for characterizing and matching fuel
and air control devices. In addition, simple systems of this type should be adjusted for higher excess air, because they have no means of automatically compensating for the fuel and air variations.

The next higher degree of sophistication is a system with simple proportional compensation. This can be done by balancing fuel burner pressure to the differential produced between windbox and furnace pressures.

In the system shown in Figure 8.6nn, the burner fuel apertures are used to measure fuel flow, and the burner air throat is used as a primary element to detect air flow. There is no square-root extraction (such extraction would not show true flows because of the nature of the primary elements), and therefore the actual controller loop gain changes with load (capacity). Rangeability is limited unless there are multiple burners that can be put into or taken out of service.

The arrangement in this control system can also be reversed, with firing rate demand directly adjusting the fuel and the correction control being on air flow. This choice will be considered later in this section.

As boilers become larger, the need for precise control becomes greater, together with the potential for savings. The following series of diagrams represents further degrees of system sophistication.

The system in Figure 8.6oo is quite similar to that shown in Figure 8.6nn, except that here flows are measured as accurately as possible and are used in a flow controller to readjust the primary loop through the combining relay (FY).

In this system fuel and air flow are open-loop controlled, and only secondary use is made of their measurements. The undesirable consequence is that fuel disturbances need master controller action for correction, and the fuel and air loops can interact with each other. The effect of interaction and disturbances in the fuel and air control loops can be minimized by the use of closed-loop fuel and air control. The system shown in Figure 8.6pp (which uses essentially the same equipment) is more desirable from the noninteracting, self-linearizing standpoint but must be provided with high turndown flow sensors and signal transmission that does not contribute a time lag.

In the systems shown in Figures 8.6oo and 8.6pp, fuel flow and air flow signals for proper fuel/air ratio are matched. This is done by matching air to fuel in the field combustion test calibration of the air flow measurement. Field testing is less stringent with the system shown in Figure 8.6pp, because the self-linearizing feature of the closed-loop system reduces the work to characterize the fuel and air control devices.

The systems shown here are for measurable fuels. In burning coal, wood, and refuse, fuel flow is often inferred from steam flow, and the steam-flow/air flow relationship is used as a control index.
## Fuel and Air Limiting

Maintaining the correct fuel/air ratio also contributes to limiting the fuel rate to available air and limiting air minimum and maximum to fuel flow. Air leading fuel on load increase, and air lagging fuel on load decrease. This arrangement also protects against a fan failure or a sticking fuel valve.

Though not normally furnished on smaller boilers, the following limiting actions are desirable for safety purposes:

1. Limiting fuel to available air flow
2. Minimum limiting of air flow to match minimum fuel flow or to other safe minimum limit
3. Limiting minimum fuel flow to maintain stable flame

These limiting features are simple to apply with the basic noninteracting, self-linearizing system in Figure 8.6pp. Figure 8.6rr shows the necessary modifications that can provide these features without upsetting the set point of the fuel/air ratio.

The following is accomplished by the illustrated system:

1. If actual air flow decreases below firing rate demand, then the actual air flow signal is selected to become the fuel demand by low selector (FY-103).
2. If fuel flow is at minimum and firing rate demand further decreases, actual fuel flow becomes the air flow demand, because FY-104A will select the fuel signal if it is greater than firing rate demand signal. A manual air flow minimum is also available to come into use through FY-104B, such that if fuel flow signal drops below the HIC setting, this manual setting will become the air flow set point.
3. Fuel is minimum limited by separate direct-acting pressure of flow regulator (FCV).

The combined control system in Figure 8.6l also shows air and fuel limiting on its air/fuel ratio system, but it is configured differently from the scheme shown in Figure 8.6rr. Figure 8.6l does not show a minimum air setter (HIC), the air/fuel ratio relay (FY-104) is located on the measurement signal to the air flow controller (FIC-104), and its ratio setting is not adjusted manually (HIC) but is optimized by “tie-in #1.” Otherwise, the two loops are quite similar.

In the combined control system of Figure 8.6l, the fuel flow is detected by a high-rangeability, accurate mass flowmeter (FT-103). The set point for the fuel flow controller (FIC-103) is the smaller of either the biased air flow or the firing rate (FY-103). The set point of the air flow controller (FIC-104) is selected as the higher of either the biased fuel flow or the firing rate (FY-104).

These high and low selectors provide the so-called cross-limited parallel metering system. The coefficients in the system are adjusted so that the pairs of signals provided to the high and low selectors are equal under steady-state conditions. When the firing rate demand increases, the high selector would be engaged.
provides it as an air flow set point while the low selector transmits the air flow process variable as the fuel flow set point. Air flow, therefore, starts to increase immediately; fuel flow rises only after the air flow responds.

When the firing rate demand signal decreases, the low selector provides it as the fuel flow set point while the high selector transmits the air flow process variable as the fuel flow set point. Fuel flow starts decreasing immediately; air flow drops only after fuel flow responds. Likewise, if a disturbance causes air flow to drop, the low selector transmits the air flow signal to the fuel flow controller. Fuel flow then decreases regardless of the steam demand, preventing a fuel-rich condition.

The disadvantage of the system is that overall response is constrained by the slower of air flow or fuel flow response to changing demand signals. For example, if the firing rate demand rises slightly, the system first positions the damper to increase air flow; then, as air flow rises, the system opens the fuel valve.

The bias and gain modules FY-103 and 104 were added in Figure 8.6l to improve the system response to small load changes. These reduce the effective fuel flow signal presented to the high selector while raising the air flow variable provided to the low selector.

Under steady-state conditions, the firing rate demand is presented as the set point to both controllers. Likewise, if the firing rate demand changes only slightly, it will still be transmitted by both signal selectors and will cause the fuel and air flows to increase or decrease accordingly. If the changes in firing rate demand exceed the offset introduced by the bias and gain modules, the system will operate like a cross-limited system. As a result, fuel flow can respond to small increases in firing rate demand without raising air flow. Similarly, air flow may be adjusted slightly downward if firing rate demand falls, without having to decrease fuel flow.

In open-loop systems, the fuel and air limits are more difficult to apply. The application of these limits to an open-loop system is shown in Figure 8.6ss. Here, the fuel set point is determined (limited) by actual air flow, and a "fuel cut-back" necessitated by reduced firing rate demand is accomplished at the expense of temporary fuel/air ratio offset.

Limiting combustion air flow to a minimum or to the rate at which fuel is being burned creates special problems because when the limit is in force, provision must be made to block the integral action in the air flow controller.

It may seem that a better way to limit fuel would be to have the firing rate demand directly set the air flow, with fuel being controlled through the combining relay (Figure 8.6tt). In this arrangement final fuel/air ratio correction occurs through the integral mode correction of the fuel/air ratio controller. This system is only partially effective, however, because on a sudden decrease of firing rate demand, the resulting reduction in fuel flow will occur only after the air flow has already been reduced.

A further consideration in setting fuel rather than air directly by the firing rate demand is that the parallel boilers can be more easily kept in balance, because balancing fuel directly balances the heat input without consideration of excess air between boilers.

**Feedwater and Drum-Level Control**

Feedwater control is the regulation of water to the boiler drum. This water is admitted to the steam drum and, after absorbing the heat from the furnace, generates the steam produced by the boiler. On most boilers, makeup water for the feedwater system is filtered, deionized, treated, and deaerated prior to entering the boiler. It is usually preheated through one or more feedwater heaters and an economizer boiler tube section. The final control elements for feedwater are control valves, pump speed, or some combination. Pumps may be driven by electric motor or steam turbine.
Proper boiler operation requires that the level of water in the steam drum be maintained within a certain band. A decrease in this level may uncover boiler tubes, allowing them to become overheated. An increase in this level may interfere with the operation of the internal devices in the drum that separate the moisture from the steam and may cause liquid carryover that can damage the steam turbine.

The water level in the steam drum is related to, but is not a direct indicator of, the quantity of water in the drum. At each boiler load, there is a different volume in the water that is occupied by steam bubbles. Thus, as load is increased there are more steam bubbles, and this causes the water to “swell,” or rise, rather than fall, because of the added water usage (Figure 8.6uu). Therefore, if the drum volume is kept constant, the corresponding mass of water is minimum at high boiler loads and maximum at low boiler loads. The control of feedwater, therefore, needs to respond to load changes and to maintain water by constantly adjusting the mass of water stored in the system.

Feedwater is always colder than the saturated water in the drum. Some steam is then necessarily condensed when contacted by the feedwater. As a consequence, a sudden increase in feedwater flow tends to collapse some bubbles in the drum and temporarily reduce their formation in the evaporating tubes. Then, although the mass of liquid in the system has increased, the apparent liquid level in the drum falls. Equilibrium is restored within seconds, and the level will begin to rise.

Nonetheless, the initial reaction to a change in feedwater flow tends to be in the wrong direction. This behavior, called “inverse response” or “nonminimum phase response,” causes an effective delay in control action, making control more difficult. Liquid level in a vessel lacking these thermal characteristics can typically be controlled with a controller gain of 10 (proportional band of 10%) or less. By contrast, the drum-level controller needs a controller gain closer to 1 (proportional band of 100%) to maintain stability. Integral action is then necessary, whereas it can usually be avoided when very narrow proportional band settings can be used.

Control of feedwater addition based on total drum level alone tends to be self-defeating, because on a load increase it tends to decrease water feed when it should be increasing. Figure 8.6vv shows the response relationship among steam flow, water flow, and drum level that should be present in a properly designed system.

**Single- and Two-Element Feedwater Systems** For small boilers having relatively high storage volumes and slowly changing loads, a simple proportional control may suffice, imprecise as it is. Integral action should not be used, because of resulting instability that is a result of integration of the swell on load changes that must later be removed. Control of this type, therefore, involves the addition of feedwater on straight proportional level control.

For larger boilers and particularly when there is a consistent relationship between valve position and flow, a two-element system (Figure 8.6ww) can do an adequate job under most operating conditions. Two-element control involves adding the steam flow as a feedforward signal to the feedwater valve (or boiler feed-pump speed). Two-element control is primarily used on intermediate-size boilers, in which volumes and capacities of the steam and water system would make the simple “total” level control inadequate because of
“swell.” Total level control is undesirable when it is detected by sensors that are insensitive to density variations, such as the conductivity type. Displacement and d/p cell-type sensors are preferred from this perspective because they respond to hydrostatic head. Smaller boilers, in which load changes may be rapid, frequent, or of large magnitude, will also require the two-element system.

Field testing, characterization, and adjustment of the control valve are required so that the relationship of control signal to feedwater valve flow matches that of the steam flow to the flow transmitter output.

Any deviations in this matching will cause a permanent level offset at the particular capacity and less than optimal control (Figure 8.6xx). The level controller gain should be such that, on a load change, the level controller output step will match the change in the steam flow transmitter signal.

**Three-Element Feedwater Systems** As boilers become greater in capacity, economic considerations make it highly desirable to reduce drum sizes and increase velocities in the water and steam systems. Under these conditions, the boiler is less able to act as an integrator to absorb the results of incorrect or insufficient control. A three-element system is used on such large boilers to arrest disturbances and react to load changes more rapidly, as they occur.

Three-element control is similar to the two-element system, except that the water flow loop is closed rather than open. In this way, pressure disturbances that would affect feedwater flow are handled immediately by the fast response of the feedwater flow loop. There are several ways of connecting a three-element feedwater system, each of which can illustrate the most common way of connecting this system.

In addition to the three primary control variables (three elements)—drum level, steam flow, and feedwater flow—drum vapor-space pressure can be utilized to compensate for density changes. The pressure is passed through a calculator, DY-118 in Figure 8.6xx, that calculates a multiplier to apply to the raw level signal. The multiplier is based on the density change vs. pressure for saturated steam, as taken from the steam tables.

In making gain adjustments on a three-element feedwater system, the first step is to determine the relative gains between level and flow loops. By observing a change in boiler load one can note the particular boiler “swell” characteristics. Maximum system stability results when the negative effect of swell equals the positive effect of flow. For example, if a 20% of maximum flow change produces a 2.4 psi (0.16 bar) change in flow transmitter output and this flow change also produces a 3 in. (75 mm) swell on a 30 in. (750 mm) range transmitter or a 1.2 psi (0.08 bar) transmitter output change, then the gain of the level loop should be double the gain on the flow loop.
Feedforward Control  A feedforward variation is recommended by Shinskey to maintain a steam-water balance, reducing the influence of shrink-swell and inverse-response phenomena. The system shown in Figure 8.6zz causes feedwater flow to match steam flow in absence of action by the level controller. The two flowmeters have identical ranges, and their signals are subtracted. If the two flow rates are identical, the subtractor sends a 50% signal to the flow-difference controller. An increase in steam flow will call for an equal increase in feedwater flow to return the difference signal to 50%.

Errors in the flowmeters and the withdrawal of perhaps 2.5% water as “blowdown” (which is not converted to steam) will prevent the two flow signals from being identical. Any error in the steam–water balance will cause a falling or rising level. Therefore, the level controller must readjust the set point of the flow-difference controller to strike a steady-state balance.

The system assumes orifice-type flow sensors and does not use square-root extractors, because the period of oscillation and dynamic gain of a two-capacity level process varies directly with flow. The gain of the feedwater control loop without square-root extraction seems to compensate correctly for the process gain change.

Figure 8.6zz also shows external feedback from the flow-difference measurement applied to the level controller. This will precondition the level controller during start-up or at other times when feedwater is controlled manually or otherwise limited. Otherwise, an increase in steam or blowdown flow will increase the feedwater flow immediately, without depending on the level controller. This means that the feedback portion of the loop (LC-108) will need only to trim the ΔFC-109 set point to correct for flowmeter errors.

Because the role of the feedback portion is reduced from manipulating feedwater flow across its entire range to adjusting only for flowmeter errors, deviations in level from the set point will be minimized. Controller mode settings are not as critical in this situation, and incorrect actions caused by shrink, swell, and inverse-response are reduced.6

Shrink–Swell Compensation  Another variation of level control proposed by Shinskey to overcome shrink–swell effects involves the use of proportional feedback from a wide-ranging level measurement.17 On large boilers, primary drum level control is often accomplished with a narrow-range transmitter for more accurate control, with a wide-range transmitter present to show large excursions and handle alarms and trips outside the narrow control range.

The wide-range transmitter can be utilized to provide proportional feedback to sum with the output of the narrow-range (NR) level control (Figure 8.6aaa). When rising vapor volume in the water, resulting from an increasing load, causes the water to swell, the wide-range (WR) measurement, sensing much more of the total water inventory, will tend to properly measure lower, while the narrow-range instrument may indicate a rising level. Used with the proper filtering and tuning, the wide-ranging signal can be used to offset the narrow-range control loop’s tendency to initially respond in the wrong direction.

Feedwater Valves  During start-up, when the boiler tubes and drum are being filled, the feedwater control valve must absorb the large pressure drop from the full feedwater pump discharge or feedwater header to the unpressurized drum (less dynamic head and elevation losses). Initial flows can be relatively small. At operating pressure, the pressure drop across

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the feedwater control valve is much smaller and the flow much larger. Due to this drastic variation in service conditions, it is common on large units to split feedwater control into a parallel-piped system with two control valves, one for start-up and one for normal operation.

On small to medium-sized units, it is possible to handle the full range of service conditions with one valve, but sizing, trim design, and material selection are critical to the success of control with a single valve. The start-up valve in a parallel two-valve system, or the regulating/start-up valve in a single-valve system, requires an anticavitation trim design (Figure 6.1y in Section 6.1).\(^6\)

The duty required of a feedwater control valve is quite heavy, not only because of the large energy dissipation as the water passes through the valve, but also because the high-purity water normally used in boilers tends to be “metal-hungry.” This, combined with high water velocity, produces corrosion, erosion, and cavitation effects that call for a chrome-moly or 400 series stainless steel valve body and trim (or other suitably resistant materials).

A general criteria has been proposed for an acceptable trim exit kinetic energy at all flows, for single-phase fluid applications.\(^1\) The guideline set forth states that valve trim exit kinetic energy should be limited to 70 ft/s (480 kPa), equivalent to about 100 ft/s (30 m/s), and, for cavitation-prone applications, limited to 40 psi (275 kPa), equivalent to about 77 ft/s (23 m/s). At trim exit velocities of around 150 ft/s (50 m/s), boiler feedwater valves are reported to consistently fail within 3 years. Care in selecting the right valve design can result in much longer service life.

To keep the feedwater flow controller gain independent of load variations, the feedwater control valve should have a linear installed flow characteristic. An equal-percentage or modified equal-percentage inherent characteristic is appropriate for electric utility boilers with dedicated feedwater pumps for each boiler. On systems where multiple feedwater pumps supply a pressure-controlled header, which, in turn, supplies two or more boilers, a linear inherent characteristic may be appropriate. A characterizing positioner can be used to further ensure the linearity of the feedwater control valve. If there is noise in the loop, then dampening may be required, as well.

**Valve Sizing** For control valve sizing, a system “head” curve showing the relationship between system pressures and capacities should be developed. A typical head curve is shown in Figure 8.6bbb. The head curve demonstrates a basic problem in selecting the flow rate and pressure drop when the feedwater control valve is sized. Capacity \(C_2\) and differential \(X\) are the most desirable from a control standpoint. Capacity \(C_3\) and differential \(Y\) or \(Z\) are often used in an attempt to furnish sufficient water to the drum with safety relief valves blowing. It is not necessary to provide all this capacity in the primary feedwater regulating control valve; this will result in an oversized valve and will degrade control performance. It is not uncommon to see a valve that was designed for more than required capacity and for a 30 psi (207 kPa) differential, operating at a 500 psi (3.45 MPa) differential and at a fraction of its design capacity. Capacity for relief valve service can be provided with bypass valves.

In addition to the feedwater control valve(s) regulating water to the boiler, there is a feedwater recirculation valve controlling flow back to the deaerator (or sometimes the turbine condenser). This valve is a very severe service application, with the valve having to take the extreme pressure drop from the feed pump discharge back to the deaerator. In some plants, this may be from 5500 PSIG (38 MPa) down to 150 PSIG (1 MPa).\(^18\) Cavitation abatement trim is required (Figure 6.1y in Section 6.1). The most efficient control strategy for this valve is to modulate it to maintain the net positive suction head (NPSH) available on the boiler feed pump greater than the NPSH required to prevent cavitation in the pump.

**Pump Speed Control** Control of pump speed to regulate feedwater flow can be accomplished if the pump is driven by a steam turbine or is furnished with variable-speed electric drive. This can be used in place of a control valve to save pump power on single-boiler systems. In systems in which several boilers are operating in parallel, the speed control can be used to save pump power by controlling the discharge pressure at a constant differential pressure above boiler pressure.

When pump speed is being used in place of a feedwater valve on a single boiler system, a large part of the speed control range is used in developing pump head at low flow. Characterization of the signal is, thus, necessary for good operation and constant gain throughout the operating range. On large units, there is still usually a start-up feedwater control valve for filling and low load, with pump speed taking over as load rises past some threshold.
Figure 8.6ccc shows typical pump characteristics. Considerable pump speed is necessary just to build pressure in the system, before any significant load is being supplied. (Further discussion of variable-speed pumping is provided in Section 8.34 of this volume and in Section 2.14 of the Process Measurement and Analysis volume.)

Steam Temperature Control

The purpose of steam temperature control is to improve the thermal efficiency of steam turbines. Its most common application is for steam turbine electric power generation. The factors affecting steam temperature in a convection-type superheater are superheater area, flue-gas flow pattern across the superheater, flue-gas mass flow, temperature of flue gases leaving the furnace, and steam flow through the superheater. Additionally, furnace temperature affects radiant superheaters. Some superheaters may be designed for a flat curve, combining radiant and convection surface, but most superheaters are the convection type.

Control-station steam temperatures are limited to approximately 1050°F (566°C), whereas those in industrial units may be considerably less. If these temperatures can be controlled with extreme precision, they can be pushed closer to the allowable limits. Temperature can be controlled by adjustment of the amount of recirculation of the flue gas or by “attemperation,” which is an energy-wasting method of superheat removal through feedwater spraying.

The required positions of burners or recirculating and bypass dampers as a function of load are well-established for any given boiler design. Therefore, it is common practice to program their positions directly from load. Readjustment to correct for inaccuracies in the program and changes in the characteristics of the boiler is accomplished by feedback control of temperature, using proportional and integral action (Figure 8.5ddd). Being applied through a multiplier, the feedback loop gain varies directly with load, and according to Shinskey this tends to cancel the inversely varying process gain.

Desuperheater Spray Controls Temperature control by attemperation is more responsive and can be used to supplement flue-gas manipulation. To minimize water usage, however, and to avoid conflict with flue-gas manipulation, proportional-plus-derivative control should be used for attemperation. The controller may be biased to deliver a nominal amount of feedwater at zero temperature deviation. The control system is described in Figure 8.6ddd.

To use a desuperheater spray for steam temperature control, the boiler would normally be provided with added superheater area. Figure 8.6eee demonstrates the effect of the water spray (which is usually between a primary and a secondary superheater section) for temperature control.

Provision must be made to prevent reset windup when in the uncontrolled load range. Control of this system is shown in Figure 8.5fff.

Large boilers may have burners that tilt up and down approximately 30° for steam temperature control. This effectively changes the furnace heat-transfer area, resulting in temperature changes of flue gases leaving the boiler. Spray is frequently used with these systems as an override control. These systems are used on large power plant boilers and normally require the type of controls illustrated in Figure 8.6ddd.

Work efficiencies are maximized by operating at the highest steam temperature at which the metals are capable of operating. In central stations, this limit is 1050°F (566°C); in industrial applications it is lower. If improved control can elevate the TIC set point from 1000°F (538°C) to 1040°F (560°C), this will increase the available work by 17 BTU/lbm (9.4 cal/kg) in the steam.
Flame temperature does not vary much with load, but the hottest gases do tend to propagate further at higher loads. To increase the steam temperatures at low loads, recirculation blowers or tilting burners are used, which will direct the heat at the superheater sections. At high loads, the rise in the steam temperature is prevented by opening up some of the flue-gas bypass dampers or by desuperheating the steam through attemporation with water. For each boiler, the relationship between load and the required damper, recirculation fan, or burner tilt positions is well-established and, therefore, can be preprogrammed.

Figure 8.6l shows the input signal to the programmer as coming from FT-102, the steam flow transmitter. (Vortex shedding meters are limited to about 750°F, or 400°C.) In other designs, the input to the programmer is taken from the combustion air flow signal (FY-104). In either case, the preprogrammed relationship does need to be adjusted to overcome inaccuracies and changes in boiler characteristics.

TIC-111 throttles the set points of both slave controllers in cascade. Reset windup in TIC-111 is prevented by the external feedback (FB). Without it, windup would occur at low loads, when high temperatures cannot be maintained. The task of temperature control is split between two slaves. The slower of these slaves—the PI controller—modulates burner or damper positions for long-term control. The desuperheater controller is faster; actually, it is faster than its cascade master. In order to use that speed, to minimize water usage (an irreversible waste of available work), and to avoid conflict with the PI controller, only P&D control modes are used.

The desuperheater control loop can also be configured as a cascade loop, as illustrated in Figure 8.6ggg. Here, the...
saturated steam from the steam drum is returned back into the furnace, where it is superheated. If the amount of superheat is excessive, then water is sprayed into the steam, and it is returned once more to the furnace to make sure that all water is vaporized.

The slave temperature controller (TIC-112) is placed right after the spray attemperator, while the cascade master (TIC-111) is located at the steam outlet. In Figure 8.6ggg, the feedforward correction is based on steam flow (FT-102), and the relationship between load and uncontrolled temperature (Figure 8.6eee) is predicted by the FY-102 function relay.

**Cooling by Variable Excess Air** Because mass flow of flue gas affects steam temperature, variation in excess combustion air can be used to regulate steam temperature. This method may be used on a boiler that was not specifically designed for temperature control. Although increasing excess air flow increases boiler stack heat losses, turbine thermal efficiency will also increase, and maintaining steam temperature can provide the greater economic benefit. The control arrangement in Figure 8.6hhhh implements this method.

**Flue-Gas Temperature**

Flue-gas temperature is important for two reasons: first, as an indicator of boiler efficiency, and second, because if it drops below the dew point, the condensate formed will dissolve the oxides of sulfur in the flue gas, and the resulting acid will cause corrosion.

Figure 8.6ll shows the temperature control loop TIC-110. The purpose of TIC-110 is to keep the flue gas dry and above its dew point. The flue-gas temperature at the cold end of the stack is usually arrived at as the average of several sensor outputs, measuring the temperature at various points in the same plane. When this averaged temperature drops near the dew point, TIC-110 will start increasing the set point of the air preheaters (TIC-114). Through the use of steam or glycol coils in the combustion air, the inlet air temperature to the boiler is increased, which in turn raises the flue-gas temperature.

**Integration of Loops**

Loops and subsystems can be combined to create an integrated control system (Figure 8.6ll). Other combinations of the subsystems can similarly be put together to form a coordinated system. When designing, it is advisable to break the overall system down into these subsystems and examine them individually. Only then should the subsystems be put together into the total system.

Major design and operation problems in complex systems are created by inadvertently creating additional interaction, positive feedback, or tracking and initialization problems. In a complex system, as a result of adding, subtracting, multiplying, dividing, and comparing control signals or transmitter signals, it is difficult to get the system on “automatic” control easily and quickly.

The chief points to remember include

1. The systems often interact, e.g., air flow affects steam temperature, feedwater flow affects steam pressure, and fuel flow affects drum level and furnace draft.
2. For flexibility and rangeability, linear flow signals are necessary. Control valves and piston operators need linearizing positioners.
3. Fuels should be totalized on an air-required basis.
4. Tie-back arrangements, which simplify the task of getting quickly on automatic control, are very important in complex systems.
5. The flows of fuel and air should be controlled such that the flow rates reaching the burner always represent a safe combination.

**Pollution Control**

Today, most electric utility-related environmental regulation is directed at reducing emissions of sulfur dioxide (SO₂) and nitrogen oxides (NOₓ) as well as carbon dioxide (CO₂), mercury, volatile organic compound (VOC), particulate, stack opacity, wastewater, and other greenhouse gas pollutants. NOₓ, CO₂, particulate, and opacity are considerably influenced by the boiler design, operation, and control. However, reduction or removal of some of these pollutants from the flue gas often must be accomplished between the boiler and the stack.

Electrostatic precipitators are commonly used for removal of particulate. Scrubbers are commonly used for removal of particulate and SO₂. From a process control viewpoint, two commonly regulated pollutants, NOₓ and SO₂, are
of particular interest. Their formation and control will be discussed in more detail.

NOx Control\textsuperscript{31,32}

Reduction of NOx emissions is a major goal of the Clean Air Act amendments because of its known role in the formation of ground-level ozone and acid rain. A number of NOx control technologies have been successfully applied to utility and industrial boilers. (For a discussion of the various nitrogen oxide sensors and analyzers, refer to Section 8.37 in Chapter 8 of the first volume of this handbook.)

NOx Formation NOx is formed by two primary mechanisms, resulting in thermal NOx and fuel-bound NOx. A third mechanism, “prompt NOx,” also accounts for a minor share of NOx formation. Thermal NOx formation occurs only at high flame temperatures when dissociated nitrogen from combustion air combines with oxygen atoms to produce oxides of nitrogen, mainly NO and NO2. Formation of thermal NOx increases with combustion temperature and the presence of oxygen. However, the reactions are reversible.

Thermal NOx usually comprises 25% of emission for a coal-fired plant, but the majority of emissions for gas-fired combustion. Fuel-bound NOx formation is not limited to high temperature, but is dependent upon the nitrogen content of the fuel. In addition to the high flame temperature, quantity of excess air, nitrogen content in the fuel, the characteristics of the combustion process and the residence time at high temperature also play important roles in NOx formation.

NOx Reduction Strategies The best way to minimize NOx formation is to reduce flame temperature, reduce excess oxygen, or burn low-nitrogen-containing fuels. NOx reduction strategies and technologies for combustion sources can be classified by three major categories: precombustion processing (fuel switching), combustion modifications, and post-combustion processing. Combustion modifications include, but are not limited to, derating, burner system modification, low NOx burners, and diluent injection. Major post-combustion processing techniques include selective catalytic reduction (SCR) and selective noncatalytic reduction (SNCR).

Low NOx Burners These burners effectively reduce the formation of fuel and thermal NOx. These burners generate air-fuel mixing patterns that lower peak flame temperature and oxygen concentrations. Reducing local oxygen concentration can be achieved by introducing flue-gas recirculation zones, air or fuel staging combustion zones, or flameless oxidation burners. Installation of these types of burners can reduce NOx emissions by up to 50%.

Injection of water, steam, or flue gas (diluent injection) also helps to lower the NOx level. By injecting a small amount of water or steam into the immediate vicinity of the flame, the flame will be cooled and the local oxygen concentration reduced. This would result in decreased formation of thermal and fuel-bound NOx. However, this process generally lowers the combustion efficiency of the unit.

Flue gas can also be injected with the influent gas ahead of the burner to reduce the prompt NOx formation. Advanced overfire air, in which air is injected in the combustor, can usually reduce NOx emissions by 10–25%.

Excess air optimization also benefits NOx reduction (see the forthcoming paragraphs on boiler optimization).

Selective Catalytic Reduction Claimed to be the most effective technology currently available for NOx removal, selective catalytic reduction is a post-formation NOx control technology that uses a catalyst to facilitate a chemical reaction between NOx and ammonia to produce nitrogen and water. This process is implemented by injecting ammonia/air or ammonia/steam mixture into the exhaust gas, which then passes through a catalyst. The major NOx reduction reaction is often presented as follows:

$$4\text{NO} + 4\text{NH}_3 + \text{O}_2 \rightarrow 4\text{N}_2 + 6\text{H}_2\text{O}$$

To optimize the reaction, the temperature of the exhaust gas must be in a certain range when it passes through the catalyst bed. Depending on the catalyst type (usually vanadium/titanium) and the location (usually economizer outlet), the typical flue-gas temperature should be in the range of 600–750°F (315–400°C). Removal efficiencies above 80% can be usually achieved, regardless of the combustion process or fuel type used.

Among its disadvantages, SCR requires additional space for the catalyst and reactor vessel, as well as an ammonia storage, distribution, and injection system. In the United States, the SCR control system is designed for operation during the ozone season (May–October), and for bypass operation in the non-ozone season.

The SCR typically consists of an NH3 injection flow control system, an ammonia injection grid, and inlet and outlet NOx monitoring equipment. Precise control of ammonia injection is critical. An insufficient amount of ammonia can result in unacceptable high NOx emission rates, while excess ammonia can lead to ammonia “slip,” or the venting of undesirable ammonia to the atmosphere. NOx reduction efficiency is directly proportional to the NH3:NOx ratio up to about 90%.

Specifically, the stoichiometry of the reaction is such that 1 mole of NH3 reacts with 1 mole of NOx, producing nitrogen (N2) and water (H2O). Adjustments to the molar ratio must be made to account for ammonia slip, i.e., the portion of the injected NH3 that passes through the SCR unreacted.

In practice, as illustrated in Figure 8.6iii, the NH3 flow control system anticipates the ammonia demand for NOx emissions based on the boiler load (or the reactor inlet NOx). This ammonia demand (FY-203) can be used to derive the ammonia feed rate and anticipates demand changes due to load swings. This ammonia demand signal is then trimmed using a feedback controller (XIC-202) that compares the
measured SCR outlet NO\textsubscript{x} to the required outlet NO\textsubscript{x}. Finally, the resulting ammonia demand signal is compared to the measured ammonia flow rate. The difference is conditioned by another slave controller (FIC-201), and the resulting control signal is used to modulate the ammonia flow control valve.

In order to maintain the flue-gas temperature above the minimum required, a set of dampers and a divider are often installed in the economizer, concurrently with the SCR project. The dampers will block only a portion of the economizer, restricting the flue-gas flow that passes through the economizer tubes. As the dampers close, they will force more flow through the open section, effectively reducing the area of economizer tubes that are exposed to the flue-gas flow. This control can be automatic (TIC-204), which compares the economizer outlet temperature (TT-204) to the set point and adjusts the position of the dampers accordingly.

Interlock functions need to be set up to prevent the ammonia system from starting or from continuing operation in abnormal situations. Important failures include, but are not limited to, NH\textsubscript{3} vaporizer outlet temperature too low, flue-gas flow too low, SCR flue-gas inlet temperature too low, dilution air flow too low, or vaporizer ambient NH\textsubscript{3} level too high.

**Selective Noncatalytic Reduction** Utility applications of SNCR processes involve the injection of a nitrogen-based reagent into the upper furnace or convective sections, where the injected chemical reacts with NO\textsubscript{x} to form molecular nitrogen and water vapor. The most common types of reagent used commercially on large utility boilers are urea and ammonia. The optimum injection temperature when using ammonia is 1850°F (1010°C), at which 60% NO\textsubscript{x} removal can be approached. The optimum temperature range is wider when using urea. Below the optimum temperature range, ammonia is formed, and above, NO\textsubscript{x} emission actually increases.

The success of NO\textsubscript{x} removal depends not only on the injection temperature, but also on the ability of the agent to mix sufficiently with flue gas. Compared to SCR, this technology is relatively capital inexpensive and used for smaller boilers. Typical NO\textsubscript{x} reduction efficiency is 30–60%. In principle, the general control philosophy of an SNCR system is similar to the SCR control.

**SO\textsubscript{2} Control**

The formation of sulfur dioxide (SO\textsubscript{2}) is a major concern with coal-fired boilers, due to the relatively high sulfur content in the fuel. For these units, post-combustion removal of SO\textsubscript{2} is realized through the flue-gas desulfurization (FGD) system. FGD systems can be classified into two basic categories: wet scrubbers and dry scrubbers.

Wet SO\textsubscript{2} scrubbers are the most widely used FGD technology for SO\textsubscript{2} control. Calcium-, sodium-, and ammonia-based sorbents are normally used in a slurry mixture, which is injected into a specially designed vessel to react with SO\textsubscript{2} in the flue gas. The most popular sorbent in operating wet scrubber is limestone, followed by lime. They are favored because of their availability and relative low cost. The overall
chemical reaction that occurs with a limestone or lime sorbent can be expressed in a simple form as:

\[ \text{SO}_2 + \text{CaCO}_3 = \text{CaSO}_3 + \text{CO}_2 \]  

A typical FGD system should consist of several major subsystems, i.e., the limestone grinding and supply system, reagent feed system, absorber/reaction tank system, forced oxidation system, process water distribution system, dewatering system, filtrate system, and possibly, flue-gas reheat system. A simplified FGD system is illustrated in Figure 8.6jjj.

The limestone grinding loop is designed to supply limestone slurry containing 40% solids to the reagent feed loop. Usually, pebbled limestone from the limestone storage silo is conveyed by a variable-speed conveyor to the ball mill. Limestone is mixed with water and ground in the ball mill. The resultant slurry flows from the ball mill into the mill slurry sump, where it is pumped to the riser in the classifier. The slurry then flows from the riser into the cyclones that separate the coarse and fine slurry. The overflow (at about 40% solids) goes to the reagent feed tank, and the underflow goes back to the ball mill for regrinding. The reagent feed tank, which stores the slurry, allows for extended periods of operation with the ball mill out of service.

The \( \text{SO}_2 / \text{reagent} \) reaction occurs in the absorber/reaction tank. The major equipment associated with the absorber loop typically are absorber feed tank (reaction tank), absorber spray pumps, wetted film contactor and its associated pump, and demister. The goal of the absorber loop is to absorb the \( \text{SO}_2 \) in the flue gas in the spray tower and wetted film contactor sections and precipitate calcium sulfite in the reaction tank. This is accomplished by controlling the reaction tank at proper pH level and density.

Reaction tank pH is the most important process variable in terms of total tower \( \text{SO}_2 \) removal efficiency and scale control. Operating at a lower pH would decrease the efficiency and increase the tendency of scale formation. Higher pH operation would cause high limestone consumption overall and decrease by-product quality. The pH value can be controlled by adjusting the reagent feed valve. The master control (XIC-302) compares the difference between the desired pH set point and the measured pH in the reaction tank. The output becomes the set point to the inner-loop slave controller (FIC-301), which compares this set point to the actual measured lime slurry flow rate and sends a command signal to adjust the valve opening.

Operating at lower density would remove valuable seed crystals from circulation, making control of the crystal types and the formation location difficult. Operation at a much higher density would increase pump horsepower requirements. Operation within a range of 5–15% solids is tolerable. A density element (DT-303) can be located in the mill slurry sump to measure the density of the mill slurry. Dilution water can be fed into the mill slurry sump to control the density via DIC-303.

Because the lime slurry flow from the reagent feed tank to the reaction tank results in a rising and falling level in the reagent feed tank, the reagent feed tank level control is also important. As shown in Figure 8.8jjj, the level in the reagent feed tank can be controlled by the signals (LT-304 and LY-304) transmitted to the ball mill, limestone conveyors, and ball mill water supply to adjust the limestone feeding and
grinding. The reaction tank level is often monitored. However, it can also be controlled by adjusting the reagent slurry flow and the tank makeup water flow, subject to tank pH and density constraints.

Other important control loops (not shown in the figure) may involve the oxidation air blower control and the flue-gas reheat control. The oxidation air system normally consists of several air compressors and is designed to oxidize the solids in the reaction tank. The purpose of oxidizing the solids is to convert calcium sulfite (CaSO₃) to calcium sulfate (CaSO₄), which dewatered much more easily to achieve a drier waste product. The automatic control can be placed either on the oxidation air blower discharge pressure or the absorber mass flow.

Sometimes, a reheat system is designed to increase the temperature of the gas in the absorber outlet duct. To prevent acid condensation and to allow the gas to have enough buoyancy, the outlet temperature must be maintained at a minimum of around 180°F (82°C). Reheating of the gas is accomplished by passing the gas exiting the tower through reheater coils that are internally heated with steam. Whenever the flue-gas temperature exiting the reheater drops below the desired set point, the appropriate steam flow valve starts to open and effectively raises the temperature of the flue gas as it exits the absorber tower.

In practice, gypsum (CaSO₄), the by-product resulting from this process, can be reused in other applications. Modern post-combustion SO₂ technologies, however, consume as much as 1% of the energy produced in coal-fired power plants.

**OPTIMIZATION OF BOILERS**

The purpose of optimization is to continuously maximize the boiler efficiency, as variations occur in the load, fuel, ambient, and boiler conditions. When the yearly boiler fuel cost is in the millions, even a few percentage points of improved efficiency can justify the costs of added instruments and controls. In the following paragraphs, a number of optimization techniques will be described. The various goals of optimization include:

1. Minimize excess air and flue-gas temperature
2. Minimize steam pressure
   a. Turbines thereby open up turbine governors
   b. Reduce feed pump discharge pressures
   c. Reduce heat loss through pipe walls
3. Minimize blowdown
4. Measure efficiency
   a. Use the most efficient boilers
   b. Know when to perform maintenance
5. Provide accountability
   a. Monitor losses
   b. Recover condensate heat

The first three methods of optimization are achieved by closed-loop process control and can be superimposed upon the overall boiler control system shown in Figure 8.6l. The tie-in points for these optimization strategies are also shown in that figure. The benefits of the last two methods (efficiency and accountability) are not obtained in the form of closed-loop control signals, but they do contribute to better maintenance and better understanding of heat losses and equipment potentials.

**Excess Air Optimization**

If a boiler is operating on a particular fuel at a specific load, it is possible to plot the various boiler losses as a function of air excess or efficiency, as shown in Figure 8.6kkk. The sum total of all the losses is a curve with a minimum point. Any process that has an operating curve of this type is an ideal candidate for instrumental optimization. Such process control systems operate by continuously determining the...
8.6 Boiler Control and Optimization

minimum loss point of the system at that particular load and then shifting the operating conditions until that point is reached.

As shown in Figure 8.6kkk, the radiation and wall losses are relatively constant. Most heat losses in a boiler occur through the stack. Under air-deficient operations, unburned fuel leaves, and when there is an air excess, heat is lost as the unused oxygen and its accompanying nitrogen are heated up and then discharged into the atmosphere. The goal of optimization is to keep the total losses at a minimum. This is accomplished by minimizing excess air and by minimizing the stack temperatures (Figure 8.6lll).

The minimum loss point in Figure 8.6kkk is not where excess oxygen is zero. This is because no burner is capable of providing perfect mixing. Therefore, if only as much oxygen would be admitted into the furnace as is required to convert each carbon molecule into CO₂, some of the fuel would leave unburned, as not all O₂ molecules would find their corresponding carbon molecules. This is why the theoretical minimum loss point shown by the dotted line in Figure 8.6kkk is to the left of the actual one. This actual minimum loss or maximum efficiency point is found by lowering the excess oxygen as far as possible, until opacity or CO readings indicate that the minimum has been reached. At this minimum loss point the flue-gas losses balance the unburned fuel losses.

Assume that for a particular boiler design using a particular fuel at normal loading, the optimum flue-gas temperature is 400°F (204°C) with 2% oxygen. The potential fuel savings through optimization can be estimated by determining the fuel loss using Figure 8.6lll, where the present, unoptimized stack gas conditions are entered.

Flue-Gas Composition Figure 8.6mm shows the composition of the flue gas as a function of the amount of air present. The combustion process is usually operated so that enough air is provided to convert all the fuel into CO₂, but not much more. This percentage of excess oxygen is not a constant. It varies with boiler design, burner characteristics, fuel type, air infiltration rates, ambient conditions, and load.

The top portion of Figure 8.6mm shows that the percentage of excess air must be increased as the load drops off. This is because at low loads the burner velocities drop off and the air flow is reduced, while the furnace volume remains constant. This reduces turbulence and lowers the efficiency of mixing between the fuel and the air. This loss of mixing efficiency is compensated for by the higher percentage of excess oxygen admitted at low loads.

The upper curve shown in Figure 8.6nnn theoretically illustrates the relationship between the excess O₂ requirement and load, and the lower plot provides actual test data for a specific boiler. Because each boiler has its own unique personality, this relationship must be experimentally determined. Once established, it can be used with a fair degree of confidence, although small shifts are still likely to occur as the equipment ages.

A simple demonstration of this law is the union of 1 lb of carbon with oxygen to produce a specific amount of heat (about 14,100 BTU, or 3,553 kcal). The gaseous product of combustion, CO₂, can be formed in one or two steps. If CO is formed first, it produces a lesser amount of heat (about 4,000 BTU, or 1,008 kcal), which, when it is converted to form CO₂, releases an additional 10,100 BTU (2,545 kcal). The sum of the heats released in each of these two steps equals the 14,100 BTU (3,553 kcal) that evolve when carbon is burned in one step to form CO₂ as the final product.

The Effect of the Fuel Used In a boiler furnace (where no mechanical work is done), the heat energy evolved from the...
combustion of the fuel with oxygen depends on the heating value (calorific value) of the fuel, the degree to which the combustion reaction goes to completion.

Because of the weight ratios of oxygen and nitrogen in air (0.2315 and 0.7685, respectively), to supply 1 lb (0.45 kg) of oxygen for combustion it is necessary to supply $1/0.2315 \times 0.7685 = 3.32$ lb (1.5 kg) of air. In this amount of air, there will be $4.32 \times 0.7685 = 3.32$ lb (1.5 kg) nitrogen, which does not enter directly into the combustion process but which nevertheless remains present.

When burning carbon to carbon dioxide, 12 parts by weight of carbon (the approximate molecular weight of C) combine with 32 parts by weight of oxygen (the molecular weight of O₂) to form 44 parts by weight of carbon dioxide (the approximate molecular weight of CO₂). By simple division, 1 lb (0.45 kg) of carbon plus 2.66 lb (1.2 kg) of oxygen will yield 3.66 lb (1.66 kg) of carbon dioxide.

The theoretical amount of air required for the combustion of a unit weight of fuel can be calculated as follows:

$$ A:F = \frac{2.66C + 7.94H_2 + 0.998S - O_2}{0.232} $$  \hspace{1cm} (8.6(5))

where $A:F$ is the air/fuel ratio, and C, H₂, S, and O₂ are the mass fractions (as-burned basis) of carbon, hydrogen, sulfur, and oxygen, respectively, in the fuel.

If theoretical or total air is defined as the amount required on the basis of the above equation, then excess air is the percentage over that quantity. Table 8.6ooo lists the excess air ranges required to burn various fuels. It can be seen that excess air requirement increases with the difficulty to atomize the fuel for maximum mixing.

Figure 8.6ppp also illustrates that gases require the lowest and solid fuels the highest percentage of excess oxygen for complete combustion. The ranges in Table 8.6ooo and the curves in Figure 8.6ppp also illustrate that as the load drops off, the percentage of excess oxygen needs to be increased.

Figure 8.6qqq illustrates the relationship between excess air and excess oxygen for a particular fuel. The optimum

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Type of Furnace or Burners</th>
<th>Excess Air %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pulverized coal</td>
<td>Completely water-cooled furnace—wet or dry-ash-removal</td>
<td>15–20</td>
</tr>
<tr>
<td></td>
<td>Partially water-cooled furnace</td>
<td>15–40</td>
</tr>
<tr>
<td>Crushed coal</td>
<td>Cyclone furnace—pressure or suction</td>
<td>13–20</td>
</tr>
<tr>
<td></td>
<td>Fluidized bed</td>
<td>15–20</td>
</tr>
<tr>
<td>Coal</td>
<td>Spreader, vibrating, and chain grate stokers</td>
<td>25–35</td>
</tr>
<tr>
<td></td>
<td>Underfeed stokers</td>
<td>25–40</td>
</tr>
<tr>
<td>Fuel oil</td>
<td>Register-type burners</td>
<td>3–15</td>
</tr>
<tr>
<td></td>
<td>Multifuel burners and flat flame</td>
<td>10–20</td>
</tr>
<tr>
<td>Acid sludge</td>
<td>Cone and flat flame-type burners, steam-atomized</td>
<td>10–15</td>
</tr>
<tr>
<td>Natural, coke oven,</td>
<td>Register-type burners</td>
<td>3–15</td>
</tr>
<tr>
<td>and refinery gas</td>
<td>Multifuel burners</td>
<td>7–12</td>
</tr>
<tr>
<td>Blast-furnace gas</td>
<td>Register-type burners</td>
<td>15–30</td>
</tr>
<tr>
<td></td>
<td>Intertube nozzle-type burners</td>
<td>15–18</td>
</tr>
<tr>
<td>Wood/bark</td>
<td>Traveling grate, water-cooled vibrating grate</td>
<td>20–25</td>
</tr>
<tr>
<td></td>
<td>Fluidized-bed</td>
<td>5–15</td>
</tr>
<tr>
<td>Refuse-derived fuels</td>
<td>Completely water-cooled furnace—traveling grate</td>
<td>40–60</td>
</tr>
<tr>
<td>Municipal solid waste</td>
<td>Water-cooled refractory covered furnace</td>
<td>80–100</td>
</tr>
<tr>
<td></td>
<td>Reciprocating grate</td>
<td>80–100</td>
</tr>
<tr>
<td>Bagasse</td>
<td>All furnaces</td>
<td>25–35</td>
</tr>
<tr>
<td>Black liquor</td>
<td>Recovery furnaces for kraft pulping processes</td>
<td>15–20</td>
</tr>
</tbody>
</table>

*From Babcock & Wilcox Co., Reference 7.*

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Excess oxygen percentages for gas, oil, and coal are around 1, 2, and 3%, respectively.

**Detectors of Flue-Gas Composition**  The analyzers available for the detection of carbon dioxide, carbon monoxide, and excess oxygen are discussed in Sections 8.9, 8.10, and 8.42, respectively, in Chapter 8 in Volume 1 of this handbook. As shown in Figure 8.6mm, excess air can be correlated to O₂, CO, CO₂, or combustibles present in the flue gas. Combustibles are usually detected either as unburned hydrocarbons or in the form of opacity. These measurements are not well suited as the basis for optimization, because the goal is not to maintain some optimum concentration, but to eliminate combustibles from the flue gas. Therefore, such measurements are usually applied as limit overrides.

The measurement of CO₂ is not a good basis for optimization either, because, as shown in Figure 8.6rrr, its relationship to excess O₂ is very much a function of the type of fuel burned. The CO₂ concentration of the flue gas also varies slightly with the CO₂ content of the ambient air.

It can also be noted from Figure 8.6mm that CO₂ is not a very sensitive measurement. Its rate of change is rather small at the point of optimum excess air. In fact, the CO₂ curve is at its maximum point when the combustion process is optimized.

Excess O₂ as the basis of boiler optimization is also a relatively insensitive measurement, but it is popular. It uses zirconium oxide probes. In order to minimize duct leakage effects, the probe should be installed close to the combustion zone (Figure 8.6ss) but still at a point where the gas temperature is below that of the electrically heated zirconium oxide detector.

The flow should be turbulent at the sensor location, if possible, to ensure that the sample will be well mixed and representative of flue-gas composition. The output signal of these zirconium oxide probes is logarithmic. According to Shinskey[^6] this is desirable. The correct location of the probe will reduce but will not eliminate the bias error caused by air infiltration. Ambient tramp air enters the exhaust ductwork (which is under vacuum) not only through leakage but also to cool unused burners and registers. The O₂ probe cannot distinguish the oxygen that entered through leakage from excess oxygen left over after combustion.

Another limitation of the zirconium-oxide fuel cell sensor is that it measures net oxygen. In other words, if there are combustibles in the flue gas, they will be oxidized on the hot surface of the probe and the instrument will register only that oxygen that remains after this reaction. This error is not
substantial when the total excess oxygen is around 5%, but in optimized boilers in which excess oxygen is only 1%, this difference between total and net $O_2$ can cause a significant error. As infiltration tends to cause an error toward the high side, while the fuel-cell effect results in a low reading, the amount of uncertainty is too high to rely on $O_2$ sensors alone when maximum efficiency is desired.

Other limitations of optimization based on excess oxygen include the fact that local problems at the burners can result in incomplete combustion, even when the excess oxygen in the flue gas is normal. Another limitation is the precision and accuracy of such excess oxygen curves, as shown in Figure 8.6nnn. This precision is a function not only of the resolution at which the curve was prepared but also of changes in fuel composition and boiler conditions.

**CO Measurement**

As shown in Figure 8.6nnn, the most sensitive measurement of flue-gas composition is the detection of carbon monoxide. As can be seen from Figures 8.6rrr and 8.6ttt, optimum boiler efficiency can be obtained when the losses due to incomplete combustion equal the effects of excess air heat loss. These conditions prevail at the “knee” of each curve. While the excess $O_2$ corresponding to these “knee” points varies with the fuel, the corresponding CO concentration is relatively constant.

Theoretically, CO should be zero whenever there is oxygen in the flue gas. In actual practice, maximum boiler efficiency can usually be maintained when the CO is between 100 and 400 ppm. CO is a very sensitive indicator of improperly adjusted burners; if its concentration rises to 1000 ppm, that is a reliable indication of unsafe conditions. Because CO is a direct measure of the completeness of combustion and nothing else, it is also unaffected by air infiltration, other than the dilution effect.

For these reasons, control systems utilizing the measurement of both excess $O_2$ and carbon monoxide can optimize boiler efficiency, even if load, ambient conditions, or fuel characteristics vary. Also, when these systems detect a shift in the characteristic curve of the boiler, that shift can be used to signal a need for maintenance of the burners, heat-transfer units, or air and fuel handling equipment.

Nondispersive infrared (IR) analyzers can be used for simultaneous in situ measurement of CO and other gases or vapors such as that of water. This might signal incipient tube leakage. Most IR sensors use a wavelength of 4.7 $\mu m$ for CO detection, because the absorption of CO peaks at this wavelength, whereas that of $CO_2$ and $H_2O$ does not. $CO_2$ is also measured and is used to determine the dilution compensation factor for CO.

The CO analyzers cannot operate at high temperatures and therefore are usually located downstream of the last heat exchanger or economizer. At these points, the flue-gas dilution due to infiltration is frequently high enough to require compensation. The measurement of $CO_2$ is used to calculate this compensation factor.
Setting the Air/Fuel Ratio  In Figure 8.6l, the set point of the air/fuel ratio relay is designated as the #1 tie-in point for the optimizer controls. This was done to emphasize the importance of this setting and to show that the method used to determine the correct air/fuel ratio will determine if the boiler is optimized or not.

In the early designs of small boilers, the air/fuel ratio was set by mechanical linkages between valve, damper, and the common jackshaft, as illustrated in Figure 8.6uuu. Even at fixed loads, these controls were only as good as the setting of the linkages that had to be readjusted manually as conditions changed.

When the importance of feedback control based on flue-gas analysis was better understood, such mechanically linked boiler controls were retrofitted as shown in Figure 8.6vvv. In this system the excess oxygen content of the flue gas is used to provide a feedback trim on the present relationship between firing rate (ZT) and damper opening (AZ). The influence of this trimming signal is bounded by the high/low limiter (AY-3) as a safety precaution to prevent the formation of fuel-rich mixtures as a result of analyzer or controller failure.

Air/Fuel Ratio with Excess Oxygen Trim  Figure 8.6www shows an example of automatic fuel/air ratio correction based on load and excess air indicated by percentage of oxygen. In this control system, FY-102 provides the relationship between the load (steam flow) and the corresponding excess oxygen set point for optimum performance. Therefore, FY-102 memorizes the characteristic curve of the boiler for the particular fuel being used (see Figures 8.6nnn and 8.6ppp) and generates the excess oxygen set point based on that curve.

To obtain some of the advantages of the closed-loop fuel system, noninteracting oxygen analysis may be used to calibrate continuously the inherently poor fuel flow signal, if it could not otherwise be used with accuracy. An example of how a satisfactory coal flow signal can be obtained by continuously calibrating a summation signal of pulverizer feeder speeds is shown in Figure 8.6xxx.

In both Figure 8.6www and Figure 8.6xxx, the set point of the excess oxygen controller is based on the steam flow. Figure 8.6yyy illustrates a closed-loop control system corrected by oxygen analysis and provided with safety limits to protect against air deficiency. For the dual-selector system to function, air and fuel flows must be scaled on the same heat-equivalent basis.

The dual-selector system forces air flow to lead fuel on an increasing load and to lag on a decreasing load. Then, flue-gas oxygen content tends to deviate above the set point on all load changes. If the oxygen controller were allowed to react proportionally to these deviations, it would tend to defeat the security provided by the selectors. Consequently,
the integral control mode alone is used on the oxygen signal, so that reaction to rapid fluctuations is minimized. The principal function of the controller is to correct for long-term deviations caused by flowmeter errors and variations in fuel quality.

A variation of the previously described control system is shown in Figure 8.6zzz, in which FY-102 represents the relationship between load and excess oxygen. The input to FY-102 is steam flow (in other systems, firing rate is used as the input), and the output is the set point of the excess oxygen controller, AIC-107. The summer (HY) provides a bias so the operator can shift the characterizer curve up or down to compensate for changes in air infiltration rates or in boiler equipment performance.

The oxygen controller compares the measured flue-gas oxygen concentration to the load-programmed set point and applies PI action to correct the offset. Antireset windup and adjustable output limiting are usually also provided. The oxygen controller is direct-acting; the air/fuel ratio adjustment factor, therefore, increases if oxygen concentration in the stack rises because of effects such as reduced fuel heating value at constant flow. Increasing the air/fuel ratio adjustment factor raises the compensated process variable transmitted to the air flow controller, FIC-104 in Figure 8.6l.

The control system in Figure 8.6zzz can be further improved by increasing its speed of response. The transportation lag in a boiler can be as much as a minute. This lag is the time interval that has to pass after a change in firing rate before its effect can be detected in the composition of the flue gas. This dead time varies with flue-gas velocity and usually increases as the load drops. In boilers in which loads are frequently shed or added at irregular intervals, this dead time can cause control problems.

The feedforward strategy shown in Figure 8.6aaaa can substantially lower this dead time. Here, the air flow damper position is controlled in a closed loop based on a set point developed through a characterizer curve from the firing rate command. This loop acts to adjust the excess oxygen in a feedforward manner.

Feedback trim is provided by an oxygen measurement that modifies the set point to the damper position controller. This system anticipates the need for excess oxygen changes by responding to load swings, then correcting oxygen concentration to correspond with the excess air curve. A further refinement can be implemented using the corrections provided by the oxygen controller to adapt the damper characterizer curve for a particular fuel to the current position in a learning mode.

**Multivariable Envelope Control** Multiple measurements of flue-gas composition can be used to eliminate the manual bias in Figure 8.6zzz and to obtain more accurate and faster control than what is possible with excess O₂ control alone.

Figure 8.6bbbb shows a control system in which the manual bias is replaced by the output signal of a CO controller,

---

**FIG. 8.6xxx**
Load vs. excess air correction applied to the air/fuel ratio of a coal-burning boiler.

**FIG. 8.6yyyy**
Feedforward control system that automatically maintains excess air during upsets.
trimming the characterized set point of AIC-107. This trimming corrects the characterized excess O₂ controller set point for changes in the characteristic curve. These changes can be caused by local problems at the burners, resulting in incomplete combustion, or by changes in fuel characteristics, equipment, or ambient conditions.

The control systems shown in Figure 8.6bbbb could be further improved if the CO measurement signal was corrected for dilution effects due to air infiltration, or if the CO controller set point was also characterized as a function of load. As was shown in Figure 8.6rrr, such characteristics can be determined for each fuel and firing rate. The control range for CO tends to remain relatively constant. As CO gives an indication of the completeness of combustion, it is not a feasible basis for control if the boiler is in poor mechanical condition or if the fuel does not combust cleanly.

Figure 8.6cccc provides the added feature of an opacity override to meet environmental regulations. In this system, under normal conditions, the cascade master is CO, just as it was in Figure 8.6aaaa. Similarly, the cascade slave is excess O₂, but when the set point of the opacity controller is reached, it will start biasing the O₂ set point upward until opacity returns to normal.

With microprocessor-based systems, it is possible to configure a control envelope, such as that shown in Figure 8.6dddd. With these control envelopes, several control variables are simultaneously monitored, and control is switched from one to the other, depending on which limit of the envelope is reached.

For example, assuming that the boiler is on CO control, the microprocessor will drive the CO set point toward the maximum efficiency (“knee” point in Figure 8.6rrr), but if in so doing the opacity limit is reached, that will override the CO controller and will prevent the opacity limit from being violated.

Similarly, if the microprocessor-based envelope is configured for excess oxygen control, it will keep increasing
boiler efficiency by lowering excess O$_2$ until one of the envelope limits is reached. When that happens, control is transferred to that constraint parameter (CO, HC, opacity, and so on); through this transfer, the boiler is "herded" to stay within the envelope defined by these constraints. These limits are usually set to keep CO under 400 ppm, opacity below #2 Ringlemann, and HC and NO$_x$ below regulations.

Microprocessor-based envelope control systems usually also include subroutines for correcting the CO readings for dilution effects or for responding to ambient humidity and temperature variations. As a result, these control systems tend to be both more accurate and faster in response than if control was based on a single variable. The performance levels of a gas-burning boiler under both excess O$_2$ and envelope control are shown on the lower part of Figure 8.6ddd.

Envelope control can also be implemented by analog controllers that are configured in a selective manner. This is illustrated in Figure 8.6eee. Each controller measures a different variable and is set to keep that variable under (or over) some limit. The lowest of all the output signals is selected for controlling the air/fuel ratio in Figure 8.6l, which ensures that the controller that is most in need of help is selected for control. Through this herding technique, the boiler process is kept within its control envelope. As shown in Figure 8.6eee, reset windup in the idle controllers is prevented by the use of external reset, which also provides bumpless transfer from one controller to the next.

Operator access is shown to be provided by a single auto/manual (A/M) station. A better solution is to provide each controller with an A/M station. Then, if a measurement is lost, only the defective loop needs to be switched to manual, not the whole system.

**Flue-Gas Temperature**

The amount of energy wasted through the stack is a function of both the amount of excess air and the temperature at which the flue gases leave. The flue-gas temperature is a consequence of load, air infiltration, and the condition of the heat transfer surfaces. Like any other heat exchanger, the boiler will also give its most efficient performance when clean and well maintained.

In optimized boiler controls, a plot of load vs. stack temperature is made when the boiler is in its prime condition, and this plot is used as a reference baseline in evaluating boiler performance. If the stack temperature rises above this reference baseline, this indicates a loss of efficiency. Each 50°F (23°C) increase will lower the boiler efficiency by about 1%.
On a 100,000 lb/hr (45,450 kg/hr) boiler, this is a yearly loss of about $20,000.

The reason for rising flue-gas temperatures can be fouling of heat-transfer surfaces in the air preheater, scale build-up on the inside of the boiler tubes, soot build-up on the outside of the boiler tubes, or deteriorated baffles that allow the hot gases to bypass the tubes. Some microprocessor-based systems will respond to a rise in flue temperature by taking corrective action, such as automatically blowing the soot away, or by giving specific maintenance instructions (see the paragraph on Soot Blowing Optimization).

If the stack temperature drops below the reference baseline, this does not necessarily signal an increase in boiler efficiency. More likely, it can signal the loss of heat due to leakage. Cold air or cold water can leak into the stack gases if the economizer or the regenerator are damaged. The consequence of this is loss of efficiency and the danger of corrosive condensation (sulfuric acid formation), if the temperature drops down to the dew point. Minimum flue-gas temperatures are approximately 250°F (121°C) for natural gas and 300–325°F (149–163°C) for heavy oil and wood. A drop in flue-gas temperature will also lower the stack effect, thus increasing the load on the induced-draft fan.

For the above reasons, the advanced envelope control systems (Figure 8.6dddd) include both high- and low-limit constraints on stack temperature, using the above-described baseline as a reference.

**Fuel Savings through Optimization**

The overall boiler efficiency is the combined result of its heat-transfer efficiency and its combustion efficiency. Heat-transfer efficiency is reflected by stack temperature, which at the hot end should not exceed steam temperature by more than 150°F (65°C) when excess air is near optimum. Combustion efficiency is tied to excess oxygen, which is brought as low as possible without exceeding the limits on CO (usually 400 ppm), opacity (#2 Ringlemann), unburned carbon, and NOx (although reduction in excess O2 is usually accompanied by reduction in NOx).

When optimization reduces the flue-gas losses, the resulting savings can be estimated from the amount of reduction in these losses. In the case of a 100,000 lb/hr (45,450 kg/hr) steam boiler, a 1% reduction in fuel consumption (a 1% increase in efficiency) will lower the yearly operating costs by about $20,000 if the fuel cost is estimated at $2 per million BTU.

With the current average efficiency and the projected or actual efficiency after optimization, fuel savings can be calculated with Equation 8.6(6):

\[
S = \left(1 - \frac{\eta_i}{\eta_b}\right)(F) \quad 8.6(6)
\]

where \( S \) is the fuel savings, \( \eta_b \) is the base efficiency, \( \eta_i \) is the improved efficiency, and \( F \) is the current or base fuel usage.

The fuel savings resulting from the lowering of excess O2 can be estimated from graphs, such as those shown in Figures 8.6eff and 8.6gggg. On these graphs the temperature is the difference between the stack and ambient temperatures. If, in a boiler operating at a 500°F (260°C) stack temperature difference, optimization lowers the excess O2 from 5 to 2%,
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this will result in a fuel savings of 1.5% (2.25% − 0.75%), according to Figure 8.6ffe.

For the same conditions, Figure 8.6gggg gives a savings of 2% (10.5% − 8.5%). Such differences are acceptable, as the size and designs of the equipment do influence the results. Although optimization has lowered the excess O₂ in some boilers to around 0.5%, the value of these last increments of savings tends to diminish in proportion to the cost of accomplishing them. Lowering excess O₂ from 1 to 0.5% will increase boiler efficiency by only 0.25%, a savings of about $5,000/yr in a 100,000 lb/hr (45,000 kg/hr) boiler, and the controls needed to sustain such operation need to be as sophisticated as the ones described in Figure 8.6ddddd. For these reasons, an optimization target of 1% excess O₂ on gas fuel is reasonable.

The fuel savings resulting from improved thermal efficiency can be estimated from Figure 8.6gggg or Figure 8.6hhhhh. Lowering the stack temperature difference from 500°F (278°C) to 400°F (222°C) at an excess O₂ of 2% will result in a savings of 1.7% (8.5% − 6.8%), according to Figure 8.6gggg. Assuming an ambient temperature of 60°F (16°C), these same conditions are marked in Figure 8.6hhhhh at 560°F (293°C) and 460°F (238°C), resulting in a savings of about 2%. Therefore, as a crude approximation, fuel savings of 1% can be estimated for each 50°F (28°C) reduction in stack gas temperature.

As was already discussed, stack gas temperature reductions must be limited to achieving a temperature above dew point that is high enough to provide the required stack effect. Consequently, the total savings from improved thermal and combustion efficiencies can be estimated on the basis of both stack temperature and excess oxygen being lowered to their limits. The resulting total fuel savings can be around 5%.

FIG. 8.6gggg
The fuel-savings potential can also be computed as shown here. (Courtesy of Dynatron, Inc.)

FIG. 8.6hhhhh
Temperature of flue gas leaving a medium-size boiler has a direct effect on combustion efficiency. (Adapted from Reference 2.)
Steam Pressure Optimization

In Figure 8.6l, the second tie-in point for the optimizer is the set point of the pressure controller PIC-101. In basic boiler operation, the steam pressure is maintained at a constant value. Optimization is possible by allowing this pressure to vary.

In co-generating plants, in which the boiler steam is used to generate electricity and the turbine exhaust steam is used as a heat source, optimization is obtained by maximizing the boiler pressure and minimizing the turbine exhaust pressure, so as to maximize the amount of electricity generated (Figure 8.6iii).

In plants that do not generate their own electricity, optimization is achieved by minimizing boiler operating pressure. This reduces the pressure drops in turbine governors by opening them up further, lowers the cost of operating feedwater pumps because their discharge pressure is reduced, and generally lowers radiation and wall losses in the boiler and piping.

In electric utility plants, optimization is achieved through sliding-pressure or free-pressure control strategies that allow the pressure to vary with load, within certain constraints. This reduces temperature fluctuations in the steam turbine, thereby reducing fatigue and improving turbine life, and it improves heat rate at lower loads. In basic sliding pressure control, the turbine governor valves are run wide-open (full-arc admission), and pressure reducing valves are used upstream to control pressure, or the boiler itself is cycled to change header pressure with load.

Pump power is particularly worth saving in high-pressure boilers, because as much as 3% of the gross work produced by a 2400 PSIG (16.56 MPa) boiler is used to pump feedwater. Figure 8.6jjjj illustrates the method of finding the optimum minimum steam pressure, which then becomes the set point for the master controller PIC-101 in Figure 8.6l.

As long as all steam user valves (including all turbine throttle valves) are less than fully open, a lowering in the steam pressure will not restrict steam availability, because the user valves can open further. The high-signal selector (TY-1) selects the most-open valve, and the valve position controller (VPC-2) compares that signal with its set point of, for example, 80%. If even the most-open valve in the plant is less than 80% open, the pressure controller set point is slowly lowered. VPC-2 is an “integral only” controller; its reset time is at least ten times that of PIC-101.

This slow integral action guarantees that only very slow “sliding” of the steam pressure will occur and that noisy valve signals will not upset the system, because VPC-2 responds only to the integrated area under the error curve. The output signal from VPC-2 is limited by PY-4, so that the steam pressure set point cannot be move outside some preset limits. This necessitates the external feedback to VPC-2, so that when its output is overridden by a limit, its reset will not wind up.

This kind of optimization, in which steam pressure follows the load, not only increases boiler efficiency but also does the following:

1. Prevents any steam valve in the plant from fully opening and thereby losing control
2. Opens all steam valves in the plant, thereby moving them away from the unstable (near-closed) zone of operation
3. Reduces valve maintenance and increases valve life by lowering pressure drop
4. Increases turbine drive efficiencies by opening up all steam governors

The total savings in yearly operating costs resulting from optimizing the steam pressure to follow the load is a small percentage of the total cost.

Steam Temperature Optimization

Dynamic Feedforward

Although the feedforward function FY-102 in Figure 8.6ggg is commonly used in superheater steam temperature control, the transient response due to major disturbances (e.g., load swing) is not properly compensated under this design. The reason is that the process dynamics are not taken into account. By incorporating dynamic models in the feedforward control design, the transient error, such as the temperature overshoot, can be effectively eliminated. The dynamic feedforward compensator can be simply chosen in the form of a causal transfer function that is equivalent to a lead-lag type compensator readily available in most control systems’ algorithm library.

Refer to Figure 8.6ggg and let the dynamics from the set point of TIC-112 to TT-111 be modeled by the transfer function \( G(s) \), and the dynamics from FT-102 to TT-111 be represented by the transfer function \( G_d(s) \). This assumes no control action from TIC-111. Let us also assume the steam flow disturbance can be transformed as \( d(s) \), and the static feedforward compensator FY-102 is replaced by a dynamic controller \( G_{ff}(s) \). To completely eliminate the transient error, the following equation should hold if there is no mismatch between the model and the actual process:

\[
G_{ff}(s) \cdot G(s) \cdot d(s) + G_d(s) \cdot d(s) = 0 \tag{8.6(7)}
\]

This immediately yields

\[
G_{ff}(s) = -\frac{G_d(s)}{G(s)} \tag{8.6(8)}
\]

If the feedforward controller \( G_d(s) \) is designed as such, the disturbance effect can be completely eliminated before it creates any transient error. However, partly due to the complexity of data collection and modeling task, this approach has not been widely used in current industrial practice. In fact, for steam temperature dynamics modeling purposes, much of the existing plant data stored for the conventional control design can be utilized.

For example, as part of the unit calibration and tuning process, the relationship between the steam flow rate and the spray water control valve input signal needs to be determined by testing at the design steam temperature while operating at different boiler loads. Much of the test data can be directly used for dynamic modeling purposes, although these data are mainly used to characterize the steady-state relationship in a conventional scheme.

In order to make the dynamic feedforward controller practically useful, the process model \( G(s) \) has to be a minimum-phase transfer function. This means that the process model cannot have inverse dynamics at the beginning of the transient. Details about the dynamic feedforward design can be found in Section 2.9 in Chapter 2. Due to process nonlinearity, multiple linear models should be identified and used for control design at different loads.

It is suggested that the flue-gas temperature at the superheater inlet is a better signal to be used for feedforward compensation. Although the firing rate master signal and its associated variables (e.g., air/steam flow) are often indicators for steam temperature change, the relationship that can be derived is really a coarse estimate. It is the flue-gas temperature and mass flow that directly influence the convection area and the resulting steam temperature. A dynamic feedforward control scheme based on radiation pyrometry is briefly introduced in Reference 35.

Model-Based Multivariable Control

The steam temperature control strategies discussed so far are mainly for superheater sections. In reality, many large utility boilers have at least one reheater section. Although the reheat steam temperature control is similar to the superheater steam temperature control in principle, the subtle differences are often overlooked. First, in the current practice, the flue-gas bypass damper and burner tilting, whenever available, are normally used as the first choice for reheat steam temperature control.

Spray water will be engaged in control action only when other methods are ineffective (possibly due to control output saturation). This measure can significantly reduce spray water usage and help to improve unit heat rate. Second, the reheat outlet steam temperature control is usually more challenging due to the fact that more flue-gas variation is expected in the reheat section.

Given the fact mentioned above, it is obvious that the superheater and reheat steam temperature control is a highly coupled process. Manipulation of flue-gas bypass damper or burner tilts will inevitably affect both superheater and reheat steam temperature, either in the same direction or in the opposite directions. For boilers with split furnace, sometimes the burner tilting and bypass damper movement can have different impact on steam temperatures of different sides. Moreover, as the steam temperature control fights against the interactions, load changes might be called for. The changing firing rate may introduce a number of other disturbances that do not necessarily come in the same fashion.

These include, but are not limited to, air flow, furnace-to-windbox differential pressure, steam flow, and steam pressure. More interactions are identified in Reference 12. A coordinated multivariable control design becomes a natural choice to achieve better performance for the steam temperature regulation. The manipulated variables may include the burner tilting angles, flue-gas damper positions, and spray water flows. The controlled variables will be superheater and reheater outlet steam temperatures. The details of model-based
multivariable control system design can be found in Section 2.13 in Chapter 2.

**Water Side Optimization**

The optimization of the water side of a steam generator includes the optimized operation of the feedwater pump at the condensate return system and of the boiler blowdown. As pumping system optimization is the subject of a separate chapter in this book, only the boiler blowdown will be discussed here.

**Blowdown Optimization**

In Figure 8.6l, the third tie-in point for the optimizer is the set point of the blowdown flow controller. The goal of optimization is to minimize blowdown as much as possible without causing excessive sludge or scale build-up on the inside surfaces of the boiler tubes. The benefits of such optimization include the reduction in the need for makeup water and treatment chemicals and the reduction in heat loss as hot water is discharged. About 90% of the blowdown should occur continuously, and 10% would result from the periodic blowing down of the mud drum and of the headers.

Blowdown can be optimized by automatically controlling the chloride and conductance of the boiler water. The neutralized conductivity set point is usually around 2500 micromhos. Automatic control maintains this set point within ±100 micromhos. The required rate of blowdown is a function of the hardness, silica, and total solids of the makeup water and also of the steaming rate and condensate return ratio of the boiler.

The amount of blowdown can be determined as follows:

\[
BD = \frac{S - R}{C - 1} \tag{8.6(9)}
\]

where

- BD = blowdown rate, lb per hour
- R = rate of return condensate, lbs per hour
- S = steam load, lbs per hour
- C = cycles of concentration based on makeup

The value for cycles of concentration is generally determined on the basis of the chloride concentration of the boiler water divided by the chloride content of the makeup water. The value is also given by dividing the average blowdown rate into the average rate of makeup water, assuming no mineral contamination in any returned condensate.

Figure 8.6kkkkk illustrates that the rate of blowdown accelerates as the boiler water conductivity set point is lowered. A reduction of about 20% can result from converting the blowdown controls from manual to automatic. In the case of a 100,000 lb/hr (45,450 kg/hr) boiler, this can mean a reduction of 1,340 lb/hr (600 kg/hr) in the blowdown rate. If the blowdown heat is not recovered, this can lower the yearly operating cost by about $10,000 (depending on the unit cost of the fuel).

Overall boiler efficiency can also be increased if the heat content of the hot condensate is returned to the boiler. Pumping water at high temperatures is difficult; therefore, the best choice is to use pumpless condensate return systems. Figure 8.6llll illustrates the operation of such a system, which uses the steam pressure itself to push back the condensate into the deaeration tank. This approach eliminates not only the maintenance and operating cost of the pump but also the flash and heat losses, resulting in the return of more condensate at a higher temperature.

Alternatively, blowdown heat can be recovered by using a heat exchanger to preheat boiler makeup water. Consider a boiler producing 400,000 lb/hr (180,000 kg/hr) steam at a drum pressure of 900 PSIG (6.2 MPa), with a blowdown rate of 5% (percentage of feedwater); Assuming makeup water at 60°F (16°C), a heat exchanger “approach” ΔT of 2°F (1.1°C), and 90% heat recovery from blowdown, energy savings would be nearly 9 million BTU/hr (9.5 million kJ/hr).

The performance of the steam and condensate piping system in the plant can also be improved if steam flows are metered. Such data is helpful not only in accountability calculations but also in locating problem areas, such as insufficient thermal insulation or leaking traps.

**Load Allocation-Based Optimization**

The purpose of load allocation between several boilers is to distribute the total plant demand in the most efficient and optimized manner. Such optimization will reduce the steam
production cost to a minimum. Such computer-based energy management systems can operate either in an advisory or in a closed-loop mode. The closed-loop systems automatically enforce the load allocation, without the need for operator involvement. The advisory system, on the other hand, provides instructions to the operator but leaves the implementation up to the operator’s judgment.

The load allocation techniques discussed in this section, which are often referred as economic dispatch, only cover the scenario where all units in consideration are located in one plant or in a nearby neighborhood. Load allocation at grid level is more complicated, because the transmission loss, transmission flow constraints, reactive power constraints, fuel transportation, production scheduling, and many other factors need to be taken into account.

In simple load allocation systems, only the starting and stopping of the boilers is optimized. When the load is increasing, the most efficient idle boiler is started (Figure 8.6a); when the load is dropping, the least efficient one is stopped. In more sophisticated systems, the load distribution between operating boilers is also optimized. In such systems, a computer is used to calculate the real-time efficiency of each boiler. This information is used to calculate the incremental steam cost for the next load change for each boiler.

For example, if the load increases, the incremental increase is sent to the set point of the most cost-effective boiler. If the load decreases, the incremental decrease is sent to the least cost-effective boiler (Figure 8.6). The required software packages with proven capabilities for continuous load balancing through the predictions of costs and efficiencies are readily available. With the strategy described in Figure 8.6, the most efficient boiler either will reach its maximum loading or will enter a region of decreasing efficiency and will no longer be the most efficient.

When the loading limit is reached on one boiler, or when a boiler is put on manual, the computer will select another as the most efficient unit for future load increases. On the other hand, the least efficient boiler will accept all decreasing load signals until its minimum limit is reached. Its load will not be increased unless all other boilers are at their maximum capacity. Such units are usually not allowed to be shut down but are given a greater share of the load by a special subroutine.

If all boilers are identical, some will be driven to maximum capacity and others will be shut down by this strategy, and only one boiler will be placed at an intermediate load. Boiler efficiency can be monitored indirectly (by measurement of flue-gas composition, temperature, combustion temperature, and burner firing rate) or directly (through time-averaged steam and fuel flow monitoring).

For the direct efficiency measurement, it is important to select flowmeters with acceptable accuracy and rangeability (Table 8.6). In order to arrive at a reliable boiler efficiency reading, the error contribution of the flowmeters, based on actual reading, must not exceed ±1/2 to ±1/4 %.
Boiler allocation can be based on actual measured efficiency, on projected efficiency based on past performance, or on some combination of the two. The continuous updating and storing of performance data for each boiler is also a valuable tool in operational diagnostics and maintenance.

The load allocation strategy described above is sometimes based upon heat rate curve (particularly for electric utility units). Most plants’ heat rate characteristics are non-linear, in that they have a high value at the low load and are flat out at the high end (Figure 8.6). Usually a unit’s heat rate curve can be approximated by a polynomial function of the load. The incremental heat rate curve can be obtained by taking derivative of the heat rate curve with respect to the load.

Whether the load allocation is based on the incremental steam cost curve or the incremental heat rate curve, the simple method discussed so far suffers from the following deficiencies:

1. The incremental cost curves do not take operating constraints into account. Especially in the past decade, emission constraints are imposed on almost every power plant. Pollution control credits and penalties should all be taken into account when the optimal load allocation is considered.

2. Methods simply based on the change of incremental cost curve do not usually work well with nonsmooth and nonconvex cost functions, as is often the case for plants that have combined cycle units. The overall heat rate curve for multiple combustion turbogenerators is discontinuous.

To overcome the disadvantage of incremental cost curve-based approach, a relatively complete, yet still simplified, economic load allocation method can be formulated as the following optimization (We assume there are total number of $N$ units available for generation):

$$
\text{Minimize } J = \sum_{i=1}^{N} (F_i + H_i - C_i) \quad 8.6(10)
$$

subject to constraints:

- \( \sum_{i=1}^{N} L_i = L_{\text{total}} \) (Total load constraint)
- \( L_{i,\text{min}} \leq L_i \leq L_{i,\text{max}} \) (Single load constraint)
- \( E_i \leq E_{i,\lim} \) (Single emission constraint)

where

- \( L_i \) = the \( i \)th unit load (the decision variable)
- \( L_{i,\text{min}}, L_{i,\text{max}} \) = the low and high limits for the \( i \)th unit
- \( L_{\text{total}} \) = the total load demand
- \( E_i \) = the \( i \)th unit emission (\( E_{i,\lim} \) is the corresponding limit)
- \( F_i \) = the \( i \)th unit fuel cost
- \( H_i \) = the \( i \)th unit emission control cost
- \( C_i \) = the \( i \)th unit emission credit

The fuel cost for each unit is a function of its load level, heat rate, and fuel price. The emission output is also a function of the load. The emission credit \( C_i \) can be the result from the emission credit trading market. A negative \( C_i \) would indicate penalty. This constrained optimization is nonlinear in general and can be directly solved by a state-of-the-art nonlinear programming algorithm. Alternatively, this problem can be tackled by a standard linear programming approach if nonlinear models are piecewise linearized first.

Note that many electric utility units have implemented automatic generation control (AGC). In AGC mode, raised or lowered pulses of varying lengths are transmitted to the unit from a central location. The control logic changes the unit’s load set point up or down in proportion to the pulse length. If the optimal load allocation program is not integrated with the AGC program, then the unit under AGC mode has to be tuned out from the load allocation program.

**Soot Blowing Optimization**

The impact of soot deposit and soot blowing on boiler performance is complicated (Figure 8.6). The complication is not only due to the obvious fact that the fouling reduces the heat transfer, but also because the fouling changed the heat distribution pattern along the flue-gas path. Besides the heat-transfer pattern, soot blowing can also affect steam temperature, thermal NO\(_x\) emission, and stack opacity. For example, one study shows that soot-blowing impact on NO\(_x\) can be as high as 6\%, change in steam temperature can be as much as 40°F (22°C), and change in the heat rate can be up to 110 BTU/kWh (116 kJ/kWh).

Efficient removal of fireside soot deposit has long been a challenging task. Frequent operation of soot blowers wastes
steam, increases blower maintenance cost, and aggravates the tube erosion. Conversely, far less frequent blowing allows too much soot accumulation and, hence, decreases efficiency. It may also cause high stack opacity when a heavily fouling area gets blown. Therefore, intelligent adjustment of the cleaning schedule according to the actual cleaning need becomes the means of achieving our primary goal: efficiency improvement. This is realized through advanced control software involving a cleanliness factor calculation and a rule-based expert system.

**Cleanliness Factor Calculation**  
Boiler section fouling status can be quantified by the section cleanliness factor (CF). By usual definition, the heat-transfer effectiveness $\varepsilon$ is the ratio of actual to design heat-transfer rate, i.e.,

$$\varepsilon = \frac{Q_{actual}}{Q_{ideal}} \tag{8.6(11)}$$

where $Q_{actual}$ and $Q_{ideal}$ are the actual and ideal section heat absorption rate (BTU/hr or kJ/hr), respectively.

Then, the cleanliness factor is defined as the ratio of the actual effectiveness vs. the baseline effectiveness.

$$CF = \frac{\varepsilon_{actual}}{\varepsilon_{baseline}} \tag{8.6(12)}$$

The baseline effectiveness is determined by design and can usually be calibrated in the field. Because the baseline heat-transfer effectiveness is most likely assumed to be a constant in practice, the cleanliness factor CF can be conveniently represented by the $\varepsilon_{actual}$.

A conventional method of calculating the heat absorption rate is the log-mean-temperature-difference approach, which requires steam (water) and flue-gas temperature measurements at each heat-transfer section inlet and outlet. The formula is

$$Q = \mu AT_{lm} \tag{8.6(13)}$$

and

$$T_{lm} = \frac{(T_{s_i} - T_{o_i}) - (T_{s_o} - T_{o_o})}{\log((T_{s_i} - T_{o_i}) / (T_{s_o} - T_{o_o}))} \tag{8.6(14)}$$

where

- $\mu$ = surface heat-transfer coefficient (same as $\varepsilon$)
- $A$ = heat exchange section area
- $T_{s_i}$ = flue-gas temperature measured at section flue-gas inlet
- $T_{s_o}$ = flue-gas temperature measured at section flue-gas outlet
- $T_s$ = steam temperature at section steam inlet
- $T_o$ = steam temperature at section steam outlet

The steam/water temperature can be measured at many places along the boiler heat-transfer path. On the other hand, flue-gas temperature measurements are usually only available around air heater inlet and outlet. At all other locations, flue-gas temperatures have to be obtained by backward computations according to a system of energy balance equations.

Another method for calculating heat absorption and cleanliness factor is empirical model based.\(^5\) Given the steam flow rate, temperature, and pressure at each section inlet and outlet, the actual heat absorption can always be calculated as

$$Q = F_s \cdot (H_o - H_i) \tag{8.6(15)}$$

where

- $F_s$ = steam flow rate (lb/hr)
- $H_i$ = steam enthalpy at section inlet (BTU/lb)
- $H_o$ = steam enthalpy at section outlet (BTU/lb)

Enthalpy $H$, as a function of steam temperature and pressure, can be determined from the standard ASME steam tables.

In order to model the ideal heat absorption for a clean section, fire-side influence (i.e., the flue-gas temperature) needs to be identified. The idea is that the steam temperature at section outlet is largely decided by section inlet flue-gas temperature and section fouling status. So, in the ideal clean section situation, if all required measurements are available, the calculation of ideal heat transfer can be modeled as

$$Q_{ideal} = f(T_s, G_i) \tag{8.6(16)}$$

where $G_i$ generically represents all variables that have influence on the section inlet flue-gas temperature.

This empirical method relies on the system’s ability of identifying all fire-side influencing variables and correctly interpreting the acquired data that represents a clean boiler section. A typical boiler will have cleanliness calculation for

![FIG. 8.6](image-url)  
*Effect of water wall cleanliness on NO\textsubscript{x} emissions in a coal-fired boiler. (Courtesy of Energy Research Center, Lehigh University.)*
the furnace wall section, the economizer section, the air heater section, and each of the superheater and reheater sections.

Figure 8.6pppp shows an example of the cleanliness factor calculation results. Also worth mentioning is that, in order to directly measure heat absorption rate for the furnace wall, in situ heat flux sensors can be installed in the wall area. However, the cost is high for purchasing, installation, and maintenance.

**Rule-Based Expert System** In addition to cleanliness factor considerations, expert systems also play a key role. Because operators are most familiar with the daily operation, their experience in detecting and handling different fouling scenarios is important and, therefore, should be incorporated into the rule base.

Expert rules can be implemented from the following perspectives.

- Determine the desired cleanliness factor for each boiler section. The soot-blowing decision can be made based on the difference between the desired and actual cleanliness factors.
- At low loads, the fouling is built up slowly, and hence longer idle time between running sequences is expected. Therefore, allowing the furnace wall to have relatively low cleanliness factor will leave more heat for the following convection sections. This also increases the opportunity to blow convection sections more, and should raise steam temperatures without having to lift the firing rate.
- At high loads, the fouling is built up rapidly, and hence shorter idle time between running sequences is expected. In this situation, superheat temperature tends to run too hot, requiring attemperating spray water to prevent overheating. Therefore, convection sections should be allowed to have relatively low cleanliness factors and the furnace wall section should be cleaned more often.
- In order to limit stack opacity, the fuel/air ratio, operating status of precipitator, and fuel burner will be frequently checked, and the result will be taken into account by the rule base.
- Regardless of cleanliness factors, there should be a minimum idle time between runs for each blower sequence.
- Regardless of cleanliness factors, each blower sequence will have a maximum allowed off-time so that slag will not be heavily accumulated in one area.

**Model-Based Boiler Optimization**

Once surrounded by skepticism, model- and computer software-based boiler optimization schemes have now been applied and proven successful in many utility and industrial boiler applications. Optimization typically involves O₂ and CO, and is targeted at efficiency improvement and, often, NOₓ reduction as well. NOₓ reduction of 10–30%, and heat rate...
The controlled variables are those performance variables that we would like to drive to the most cost-effective and regulatory compliant region. The manipulated variables are those that affect the CVs and can be directly adjusted by the control strategy. In the boiler optimization context, they are various supervisory set points (or their biases). The disturbance variables are the variables that affect one or more performance variables. They are measurable but cannot be adjusted by the control strategy directly. In the boiler case, it is usually the load (or steam flow) change. Depending on boiler size and configurations, the following controlled and manipulated variables are normally selected for the initial experiment.

Potential CVs are CO, NO\textsubscript{x}, \(O_2\), and boiler efficiency/heat rate.

Potential MVs are feeder speed bias, mill exit primary air temperature bias, \(O_2\) trim bias, FD/ID fan bias, furnace pressure, auxiliary air damper bias, fuel flow bias, flue-gas damper bias, burner tilt bias, and secondary over fire air (SoFA) damper bias.

Step tests are normally carried out at different load levels, e.g., high, medium, and low load. Although it is desirable to exercise most parameters to values beyond those encountered in normal operation, due to operating constraints in most plants, the magnitude of the test signal is normally selected at 5–10\% of the overall operating range. Correlation analysis can be performed to sort through the large number of variables involved and to identify the variables that have significant impact on the performance.

Modeled relationships can take the form of step response, impulse response, state-space representation, or a neural network (a direct nonlinear form). If a linear form is assumed, then the model is linearized about some operating point, or a series of linear models is produced; each represents a specific operating condition (usually load level). The obtained model can be used for solving a static optimization problem to find out the optimal operating point. The “optimal” criterion can be user selectable. For example, the selection can be minimizing NO\textsubscript{x}, minimizing heat rate, or a combination of both. The model can also be used for carrying out closed-loop control, i.e., to use the identified MVs and drive the CVs to the region of optimal performance.

**Closed-Loop Control and Optimization**

Closed-loop multi-variable boiler control has to be planned and performed carefully, because plant operators are not traditionally willing to reduce air/fuel ratios due to concerns about CO and other symptoms associated with oxygen-deficient combustion. Model predictive control (MPC) is by far the most widely used technique for conducting multivariable boiler optimization and control. Forms of MPC that are inherently multi-variable and that include real-time constrained optimization in the design are best suited for boiler application.

For example, when NO\textsubscript{x} and CO are selected as CVs, in a constrained optimization they do not have to be controlled to any specific set point as long as they are all held below specified limits. NO\textsubscript{x} limit can be specified either by regulation.
or by the plant operator. The CO limitation can be specified by the operator as a performance constraint.

Fuel quality, boiler loading, heat exchanger surface fouling, ambient condition, and aging of equipment will all cause process to drift and affect the model accuracy. Adaptive tuning computations can be built in to take care of known quantifiable relations (e.g., variation of dead time with load). Online training, incorporating an adaptive learning algorithm, can be used to automatically train models in real time, combining recent results with the initial and historical training data.

Certain results from the optimization calculation may be very intuitive. For example, it might call for reducing the top elevation mill coal flow whenever feasible, or removing the top elevation mill from service whenever the load can be sustained with the lower level mills. This coincides with our intuition that reducing the fuel input for the upper level mill will result in a lower fire-ball position, and effectively lower the upper furnace flame temperature, which in turn reduces the thermal NOx formation.

CONCLUSIONS

The various goals of boiler optimization include the following.

- To minimize excess air and flue-gas temperature
- To measure efficiency (use the most efficient boilers; know when to perform maintenance)
- To minimize steam pressure (open up turbine governors; reduce feed pump discharge pressures; and reduce heat loss through pipe walls)
- To minimize blowdown
- To provide accountability (monitor losses; recover condensate heat)
- To minimize transportation costs (use variable-speed fans; eliminate condensate pumps; and consider variable speed feedwater pumps)

If the potentials of all of the above optimization strategies are fully exploited, the unit costs of steam generation can usually be lowered by about 10%. In larger boiler houses, this can represent a savings that will pay for the optimization system in a year or less.

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