2005

Quantifying Savings From Improved Boiler Operation

Kevin Carpenter
University of Dayton

J. Kelly Kissock
University of Dayton, jkissock1@udayton.edu

Quantifying Savings From Improved Boiler Operation

Kevin Carpenter
Graduate Research Assistant

Kelly Kissock
Associate Professor

Department of Mechanical and Aerospace Engineering
University of Dayton
Dayton, OH

ABSTRACT
On/off operation and excess combustion air reduce boiler energy efficiency. This paper presents methods to quantify energy savings from switching to modulation control mode and reducing excess air in natural gas fired boilers. The methods include calculation of combustion temperature, calculation of the relationship between internal convection coefficient and gas flow rate, and calculation of overall heat transfer assuming a parallel-flow heat exchanger model. The method for estimating savings from changing from on/off to modulation control accounts for purge and drift losses through the boiler and the improved heat transfer within the boiler due to the reduced combustion gas flow rate. The method for estimating savings from reducing excess combustion air accounts for the increased combustion temperature, reduced internal convection coefficient and increased residence time of combustion gasses in the boiler. Measured boiler data are used to demonstrate the accuracy of the methods.

INTRODUCTION
In our experience, common opportunities for improving boiler efficiency include switching from on/off to modulation control and reducing excess air. The decision about whether to pursue these opportunities often depends on the expected savings. Thus, accurate estimates of savings are vital to efforts to improve energy efficiency. This paper describes methods for estimating the expected savings from these measures in natural gas fired boilers. The methods described here include both calculation of the combustion temperature and heat transfer effects within the boiler, and, as a consequence, provide better estimates of savings than methods that ignore these effects.

CALCULATING BOILER EFFICIENCY
We define boiler efficiency as the ratio of heat transferred to the water/steam to the total fuel energy supplied. A method for calculating boiler efficiency from easily measured input variables is derived below.

The minimum amount of air required for complete combustion is called the "stoichiometric" air. Assuming that natural gas is made up of 100% methane, the equation for the stoichiometric combustion of natural gas is:

$$CH_4 + 2\,(O_2 + 3.76\,N_2) \rightarrow CO_2 + 2\,H_2O + 7.52\,N_2 \quad (1)$$

The molecular weights of natural gas and air are 16.0 lbm/lbmol and 28.9 lbm/lbmol, respectively. Thus, the mass of air needed to completely combust 1 lbmol (16.0 lbm) of natural gas is:

$$[2 \times (1 + 3.76)] \text{ lbmol} \times 28.9 \text{ lbm/lbmol} = 275.1 \text{ lbm}$$

Stoichiometric air/fuel ratio is the mass ratio of combustion air mass, m_a, to fuel mass, m_{fuel}:

$$AFs = m_a / m_{fuel} \quad (2)$$

Thus, the stoichiometric air/fuel ratio, AFs, for natural gas is:

$$AFs = m_a / m_{ng}$$
$$AFs = 275.1 \text{ lbm air} / 16.0 \text{ lbm ng}$$
$$= 17.2 \text{ lbm air} / \text{lbm ng}$$

To ensure that all natural gas is burned, the quantity of air supplied for combustion is typically greater than the minimum air required for stoichiometric combustion. The quantity of air supplied in excess of stoichiometric air is called excess air, EA, where EA is:

$$EA = [(m_a / m_{ng}) / AFs] - 1 \quad (3)$$

The temperature of combustion, Tc, can be calculated from an energy balance on the combustion chamber (Figure 1), where the chemical energy released during combustion is converted into sensible energy gain of the gasses.

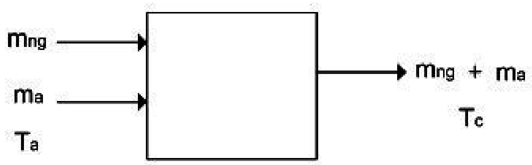

Figure 1. Combustion chamber with mass flows

The energy released during combustion is the product of the natural gas mass flow rate, m_{ng}, and the heating value of natural gas. Because the water in combustion gasses leaves the combustion chamber of boilers as a vapor, the appropriate heating value to use is the lower heating value, LHV, which is about 21,500 Btu/lbm for natural gas. (The water vapor in the combustion gasses may or may not condense as the gasses travel through the rest of the boiler. If they do condense, the latent energy released will be added at that point in the calculations.)

The sensible energy gain of the combustion gasses is the product of the mass flow rate of the product gasses, the specific heat of the gasses, Cp_g (about 0.26 Btu/lbm-F), and the temperature rise of the gasses through the combustion chamber.

If maximum heat input to the burner, Qmax, and fraction of maximum heat input at which the burner is firing, fi, are known, natural gas mass flow rate, m_{ng}, can be calculated as the product of the two values divided by the higher heating value of natural gas, HHV:

$$m_{ng} = (Qmax \cdot fi) / HHV \quad (4)$$

The mass flow rate of the combustion gasses, m_g, is the sum of the natural gas flow rate, m_{ng}, and the air mass flow rate, m_a.

$$m_g = m_{ng} + m_{air} \quad (5)$$

The temperature rise of the combustion gasses is the difference between the combustion temperature, Tc, and the temperature of the gasses before entering the chamber, Ta. An energy balance on the combustion chamber gives:

$$m_{ng} \cdot LHV = m_g \cdot Cp_g \cdot (Tc - Ta) \quad (6)$$

Substituting Equation 3 and solving for Tc gives:

$$Tc = Ta + LHV / [\{1 + (1 + EA)\ AFs\} \cdot Cp_g] \quad (7)$$

The boiler efficiency, η_b, is the ratio of heat transferred to the water/steam to the total fuel energy supplied. The total fuel energy supplied is the product of the natural gas mass flow rate, m_{ng}, and the higher heating value of the fuel, HHV (about 23,900 Btu/lbm for natural gas). The energy transferred to the water/steam equals the sensible and latent energy loss of the combustion gasses. The sensible energy loss of the combustion gasses is the product of the mass flow rate of the combustion gasses, m_g, the specific heat of the combustion gasses, Cp_g, and the temperature difference between the temperature of the combustion gasses as they enter, Tc, and exit the boiler as exhaust, Tex. The latent energy loss of the combustion gasses is the product of the natural gas flow rate and the latent heat released by the combustion gasses as they pass through the boiler, Q_{lat}. Thus, the boiler efficiency can be written as:

$$\eta_b = [\ (m_{ng} + m_a) \cdot Cp_{ex} \cdot (Tc - Tex) + m_{ng} \cdot Q_{lat}] / \\ / m_{ng} \cdot HHV \quad (8)$$

If the exhaust gas temperature, Tex, is above about 140 F, then the water in combustion gasses leaves the boiler as a vapor. If the exhaust gas temperature is below 140 F, then the water in combustion gasses leaves the boiler as a liquid. The latent heat released by the combustion gasses, Q_{lat}, is the difference between the higher heating and lower heating values. Thus,

$$Q_{lat} = 0 \quad \text{(when Tex > 140 F)}$$
$$Q_{lat} = HHV - LHV \quad \text{(when Tex < 140 F)} \quad (9)$$

Substituting Equation 3 into Equation 8 gives boiler efficiency in terms of easily measured quantities:

$$\eta_b = [\{1 + (1 + EA)\ AFs\}\ Cp_{ex}\ (Tc - Tex) + Q_{lat}] / \\ / HHV \quad (10)$$

Exhaust gas temperature, Tex, and excess air, EA, can be measured using a combustion analyzer.

For example, consider a natural gas boiler using 50% excess combustion air at 70 F and exhausting combustion gasses at 400 F. The results from this method are compared to combustion efficiencies calculated using the PHAST (2) software (Table 1). In general, the results show good agreement. We assume the slight differences are because of different assumptions about the quantity of methane in natural gas. The method presented here assumes that natural gas is 100% methane, although it may be as low 70% (1).

Table 1. Boiler efficiencies calculated using PHAST (2), and using the method developed here assuming the inlet temperature of the combustion air is 70 F.

Excess Air	Exhaust Temperature (F)				
	350	400	450	500	550
0.10	85.1% / 83.9%	83.7% / 82.8%	82.4% / 81.7%	81% / 80.6%	79.7% / 79.6%
0.25	84.3% / 83.1%	82.8% / 81.9%	81.3% / 80.7%	78.8% / 79.4%	78.3% / 78.2%
0.50	82.9% / 81.8%	81% / 80.3%	79.4% / 78.9%	77.6% / 77.4%	75.9% / 76%
0.75	81.5% /80.5%	79.5% / 78.8%	77.5% / 77.1%	75.5% / 75.4%	73.5% / 73.7%
1.00	80.1% / 79.2%	77.8% / 77.2%	75.6% / 75.3%	73.3% / 73.4%	71.1% / 71.5%

MODELING A BOILER AS A HEAT EXCHANGER

The rate of heat transfer to the water/steam, Q, can be calculated as the product of the maximum heat input to burner, Qmax, the fraction of maximum heat input at which the burner is firing, fi, and the boiler efficiency, η_b:

$$Q = Qmax \cdot fi \cdot \eta_b \tag{11}$$

Heat transfer rate, Q, can also be calculated by modeling a boiler as a heat exchanger, in which the hot stream is the combustion gas and the cold stream is the water/steam (Figure 2). For most fire-tube boilers, the parallel-flow heat exchanger model is appropriate since the combustion gasses and feedwater both enter the bottom of the boiler, while the combustion gasses and water/steam both leave near the top of the boiler.

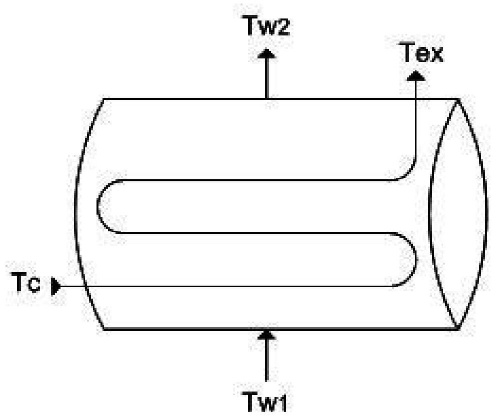

Figure 2. Fire-tubed boiler modeled as a heat exchanger

Using the LMTD method (3), the rate of heat transfer, Q, is the product of the overall heat transfer coefficient times the surface area, UA, and the log mean temperature difference, ΔT_{lm}.

$$Q = UA \cdot \Delta T_{lm} \tag{12}$$

The log mean temperature difference for a parallel-flow heat exchanger is:

$$\Delta T_1 = Tc - Tw_1$$
$$\Delta T_2 = Tex - Tw_2$$
$$\Delta T_{lm} = (\Delta T_2 - \Delta T_1) / \ln(\Delta T_2 / \Delta T_1) \tag{13}$$

A boiler's overall heat transfer coefficient, U, depends on the convection coefficient of combustion gasses, the thermal conductivity and thickness of the boiler heat transfer surfaces, and on the convection coefficient of water/steam. Of these, the convection coefficient of combustion gasses has the most significant effect on U. Using the Dittus-Boelter equation (3), the convection coefficient, h, for turbulent flow through a circular tube is a function of fluid density, ρ, fluid viscosity, μ, fluid velocity, V, tube diameter, D, Prandtl Number, Pr, and fluid thermal conductivity, k.

$$h = [0.023 \cdot (\rho \cdot V / \mu)^{4/5} \cdot Pr^{0.4} \cdot k] / D^{1/5} \tag{14}$$

In the case of a boiler, combustion gas velocity, V, is directly proportional to combustion gas mass flow rate, m_g. Since every other term on the right side of the equation is a constant, the convection coefficient, h, is proportional to combustion gas mass flow rate, m_g, to the 4/5 power.

$$h \sim m_g^{4/5} \tag{15}$$

Since the overall thermal resistance to heat flow from the combustion gasses to the steam is dominated by the thermal resistance from the gas convection coefficient, h, we can assume with minimal error that U is also proportional to combustion gas mass flow rate, m_g, to the 4/5 power. Since surface area, A, remains constant, UA is approximately proportional to combustion gas mass flow rate, m_c, to the 4/5 power.

$$UA \sim m_g^{4/5} \tag{16}$$

When combustion gas mass flow rate changes in the tube, the UA value will change. The ratio of the new value, UA,n, to the initial value, UA, is equal to the 4/5 power of the ratio of new mass flow rate, $m_{g,}n$, to the previous mass flow rate, m_g. Thus, the new overall heat transfer coefficient, UA,n, at flow rates different from the original heat transfer coefficient, UA, is approximately:

$$UA,n = UA \cdot (m_g,n/m_g)^{4/5} \tag{17}$$

VERYIFYING THE VALIDITY OF THE MODEL

To verify the validity of Equation 17, data were collected from six boilers at varying fractions of maximum heat input. Actual and predicted UA values based on the methodology were then calculated at each firing rate and compared.

Calculating Actual UA Value

Excess air, EA, temperature of gasses before entering the chamber, Ta, and exhaust gas temperature, Tex, were measured, and boiler efficiency, η_b, was calculated using Equations 7, 9, and 10. Combustion gas mass flow rate was calculated using Equations 4, 3, and 5 for varying fractions of maximum burner heat input, fi. The rate of heat transfer to the water/steam, Q, was calculated using Equation 11. Water temperature before entering the boiler, Tw_1, and steam/hot water temperature, Tw_2, were measured, and UA was calculated using Equations 12 and 13.

Calculating Predicted UA Value

The predicted UA value at firing rates less than 100% of maximum burner input was calculated as the UA,n term of Equation 17. The m_g,n term used in the equation was the combustion gas mass flow rate at that particular firing rate. The UA and m_g terms used were the actual values at 100% firing rate.

Comparison of Predicted and Actual UA Values

The firing rate of a fire-tube boiler with rated heat input of 5.23 mmBtu/hr boiler when producing 50 psig saturated steam was modulated from about 30% of full fire to 100% of full fire. Figure 3 shows measured boiler efficiency, η_b, and exhaust gas temperature, Tex, at the various firing rates.

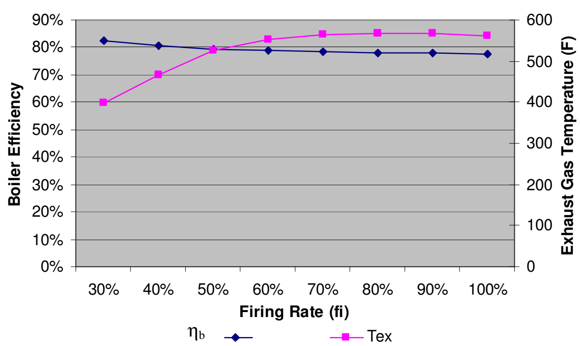

Figure 3. Boiler efficiency and exhaust gas temperature as functions of burner firing rate

Based on this data, actual UA values were calculated and plotted in Figure 4. Figure 4 shows good agreement between actual and predicted UA values over the entire range of firing rates and combustion gas mass flow rates, m_g.

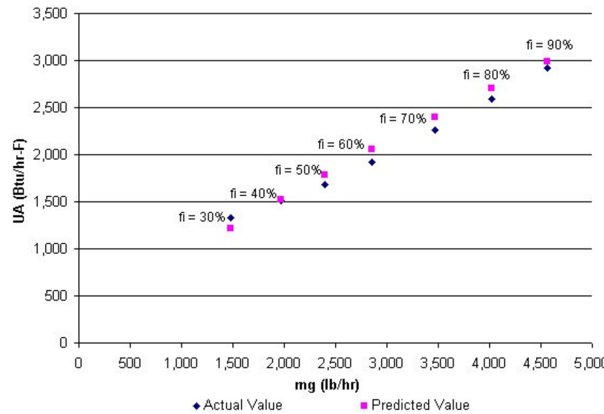

Figure 4. Actual and predicted UA values as a function of combustion gas mass flow rate

Five other boilers were tested, and the results were similar to those shown in Figures 3 and 4. These results increase confidence in the model.

IMPROVING BOILER EFFICIENCY BY OPERATING IN MODULATION MODE

The hot water temperature (or steam pressure) within a boiler is typically controlled by alternately turning the burner on and off or by modulating burner output. In on/off mode, the burner operates at 100% fire and shuts off when the pressure/temperature has reached an upper threshold. In modulation mode, the burner's firing rate is modulated between preset limits to maintain boiler pressure/temperature within a specified range.

In most small and medium sized boilers, mechanical linkages connect the boiler's fuel intake valve and combustion air damper to attempt to maintain a relatively constant air/fuel ratio. Some larger boilers are equipped with electronic controls to independently modulate the air and fuel to maintain the air/fuel ratio within a narrow range.

On/off control mode is less efficient than modulation mode for three reasons. First, flue gas purges occur before and after a boiler's firing cycle, resulting in heat losses. Second, heat is lost from the natural convective draft through a boiler when not firing. Third, boilers run less efficiently in high fire than in low fire, since the ratio of heat transfer area to heat input is lower when burner firing rate increases. Modulation mode is more energy-efficient because it eliminates purge and draft loss, and allows the boiler to operate more efficiently at low fire.

Purge Loss

Before and after each boiler firing cycle, the boiler flue is typically purged for about 10 – 30 seconds to eliminate residual fuel. This purge air draws heat from the boiler, resulting in energy losses.

Because the combustion air fan is used for purging, the purge air mass flow rate is typically equivalent to the combustion air mass flow rate at high fire. Surprisingly, measurements show that the purge air exiting the stack is typically at a higher temperature than the steam or hot water inside the boiler because the purge air passes through the boiler tubes, which are the hottest part of the boiler. However, in the following model, we conservatively assume purge air temperature to be the temperature of the steam/hot water. Boiler purge heat loss, Qp, is the product of air mass flow rate, m_a, specific heat, Cp_a, sum of pre and post-purge cycle time, PreP + PstP, number of cycles, N, and the temperature difference between the air exiting the boiler, Tw_2, and entering the boiler, Ta. Thus, boiler purge heat loss, Qp, can be written as:

$$Qp = m_a \cdot Cp_a \cdot (PreP + PstP) \cdot N \cdot (Tw_2 - T_a) \qquad (18)$$

<u>Draft Loss</u>

In a hot, idle boiler without dampers to stop air flow, cool air is pulled into the boiler through the combustion chamber and exits through the stack. To model air draft through a boiler, consider the force balance in Figure 5.

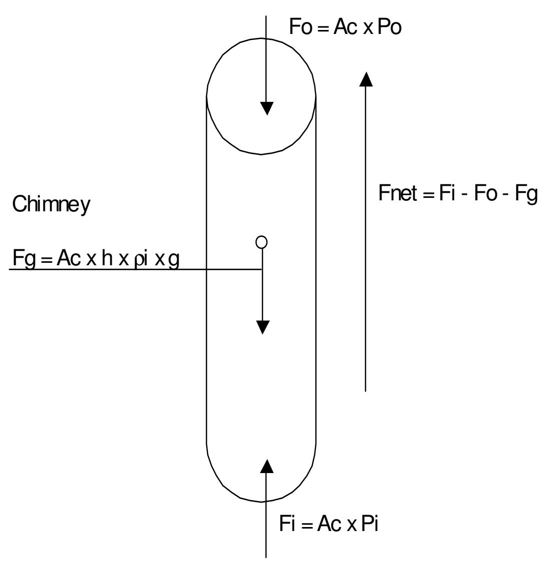

Figure 5. Force balance on air in a boiler stack

The net force upward on the air in the stack, Fnet, is:

$$Fnet = Fi - Fo - Fg \qquad (19)$$

where Fi and Fo are the forces due to air pressure at the inlet and outlet of the wind tower, and Fg is the force of gravity acting on the column of air in the stack as defined below:

$$Fi = Ac \cdot Pi \qquad (20)$$
$$Fo = Ac \cdot Po \qquad (21)$$
$$Fg = Ac \cdot h \cdot \rho i \cdot g \qquad (22)$$

where Ac is the cross sectional area of the stack, Pi and Po are the air pressures at the inlet and outlet of the stack, h is the stack height, ρi is the density of air in the stack and g is the acceleration of gravity. Substituting Equations 20, 21 and 22 in Equation 19 gives:

$$Fnet = Ac \cdot (Pi - Po) - (Ac \cdot h \cdot \rho i \cdot g) \qquad (23)$$

For the stationary air outside the stack, Fnet equals zero, and Equation 23 can be solved for:

$$Ac \cdot (Pi - Po) = Ac \cdot h \cdot \rho o \cdot g \qquad (24)$$

where ρo is the density of air outside the stack. Substituting Equation 24 into Equation 23 gives:

$$Fnet = Ac \cdot h \cdot g \cdot (\rho o - \rho i) \qquad (25)$$

From the ideal gas law, ρo can be substituted for ρi, the density of air outside and inside the stack to give:

$$Fnet = Ac \cdot h \cdot g \cdot [Po/(R \cdot To) - Pi/(R \cdot Ti)] \qquad (26)$$

Dividing by Ac, the net pressure difference is:

$$Pnet = h \cdot g \cdot [(Po/To) - (Pi/Ti)] / R \qquad (27)$$

Assuming Po and Pi are approximately equal to the average atmospheric pressure, Pbar, gives:

$$Pnet = h \cdot g \cdot Pbar \cdot [(1/To) - (1/Ti)] / R \qquad (28)$$

This result is equivalent to the result derived by Jones and West (4) and others for natural draft through a stack caused by a temperature difference. Pnet can be used in Bernoulli's equation to calculate the velocity, V, and volume flow rate, VFR, of air through the stack as:

$$V = \sqrt{2 \ Pnet/\rho} \qquad (29)$$

$$VFR = V \cdot Ac \qquad (30)$$

Draft heat loss, Qd, is the product of volume flow rate, VFR, air density, ρ_a, specific heat, Cp_a, and the temperature difference between the air exiting the boiler, $\approx Tw_2$, and entering the boiler, Ta. This quantity must be multiplied by (1 – Part Load Ratio) to account for the fraction of time the boiler is not firing. Thus, boiler draft heat loss, Qd, can be written as:

$$Qd = VFR \cdot \rho_a \cdot Cp_a \cdot (Tw_2 - T_a) \cdot (1 - PLR) \quad (31)$$

Shell Loss

Shell loss is the convective and radiative heat loss from the warm boiler shell to the surroundings. Since boiler shell temperatures are usually relatively low, radiative losses are minimal and heat loss is dominated by convection. Thus, shell loss, Qc, can be modeled with minimal error as the product of boiler surface area, Ab, convection coefficient of air in contact with the boiler, h, and the temperature difference between boiler shell temperature, Tb, and ambient air temperature, Tia:

$$Qc = Ab \cdot h \cdot (Tb - Tia) \quad (32)$$

The convection coefficient for a warm surface subject to free convection is about 7 Btu/hr-ft^2-F.

On-Cycle Stack Loss

To find on-cycle stack loss, combustion gas exhaust temperature, Tex, and excess air, EA, must be found with a combustion analyzer. Initial combustion temperature, Tc, and boiler efficiency, η_b, can be calculated using these values in Equations 7 and 8. Heat transfer rate to the water/steam when the boiler is firing, Qo, can be substituted into Equation 11 for Q to be calculated for a fraction of firing rate, fi, of 100%. On-cycle stack loss in an on/off boiler, Qs,o, is the product of the part load ratio, PLR, and the difference between heat input to the burner, Qmax, and the heat transfer rate to water/steam when firing, Qo:

$$Qs,o = PLR \cdot (Qmax - Qo) \quad (33)$$

Total Loss in On/Off Boiler

The total loss in an on/off boiler, Qt,o is the sum of purge loss, Qp, draft loss, Qd, shell loss, Qc, and on-cycle stack loss, Qs,o:

$$Qt,o = Qp + Qd + Qc + Qs,o \quad (34)$$

The rate of useful heat transfer, Qu, is the difference between maximum heat input to the burner, Qmax, times part load ratio, PLR, and total heat losses, Qt,o:

$$Qu = (Qmax \cdot PLR) - Qt,o \quad (35)$$

Taking all heat transfer effects into account, the total efficiency, η_t, of an on/off boiler is the ratio of useful heat transfer, Qu, to the product of maximum heat input to the burner, Qmax, and part load ratio, PLR:

$$\eta_t = Qu / (Qmax \cdot PLR) \quad (36)$$

The boiler's UA-value and the combustion gas mass flow rate, m_g at 100% fire are needed to determine heat transfer if the boiler were to operate in modulation mode, described in the next subsection. UA-value can be calculated by substituting heat transfer rate, Qo, for Q, in Equations 12 and 13. Natural gas mass flow rate, m_{ng}, can be calculated using Equation 4 for a firing input of 100% of maximum. Air mass flow rate, m_a, can be calculated using Equation 3, and combustion gas mass flow rate, m_g, can be calculated using Equation 5.

Modulating Stack Loss

If the boiler were to operate in modulation mode rather than on/off mode, then UA-value, combustion gas mass flow rate, m_g, combustion gas exhaust temperature, Tex, and heat transfer rate to water/steam when the boiler is firing, Qo, would all decrease to new values (UA,n, m_g,n, Tex,n, Qo,n). Useful heat transfer, Qu, combustion temperature, Tc, and shell loss, Qc, would not change. Purge loss and draft loss would be eliminated.

The new heat transfer rate to water/steam when the boiler is firing, Qo,n, would be the sum of useful heat transfer, Qu, and shell loss, Qc:

$$Qo,n = Qu + Qc \quad (37)$$

Substituting Equation 17 into Equations 12 and 13, the expression for Qo,n becomes:

$$Qo,n = UA \cdot (m_g,n/m_g)^{4/5} \cdot \Delta T_{lm},n \quad (38)$$

$$\Delta T_1,n = Tc - Tw_1$$
$$\Delta T_2,n = Tex,n - Tw_2$$
$$\Delta T_{lm},n = (\Delta T_2,n - \Delta T_1,n) / \ln(\Delta T_2,n / \Delta T_1,n) \quad (39)$$

Heat transfer to the water/steam, Qo,n, can also be calculated as the enthalpy change in combustion gasses between the combustion chamber and the exhaust stack. Assuming the water in the combustion gasses leaves as a vapor, the heat transfer to the water/steam is:

$Qo,n = m_g,n \cdot Cp_g \cdot (Tc - Tex,n)$
(40)

Equations 38, 39, and 40 are three equations with two unknown variables (m_g,n and Tex,n). Thus, the system of equations can be solved by iteration.

With new combustion gas exhaust temperature, Tex,n, known, new boiler efficiency, η_b,n, can be calculated using Equation 10. The fraction of maximum heat input at which the burner would fire, fi, can be calculated using Equation 11.

Modulating stack loss, Qs,m, is the difference between maximum heat input to the burner, Qmax, times fraction of maximum heat input, fi, and the heat transfer rate to the water/steam, Qo,n:

$Qs,m = (Qmax \cdot fi) - Qo,n$
(41)

Total Loss in Modulating Boiler

Total loss in a modulating boiler, Qt,m, is the sum of shell loss, Qc, and modulating stack loss, Qs,m:

$$Qt,m = Qc + Qs,m \tag{42}$$

Taking all heat transfer effects into account, the total efficiency, η_t, of a modulating boiler is the ratio of useful heat transfer, Qu, to the product of maximum heat input to the burner, Qmax, and fraction of maximum heat input, fi:

$$\eta_t = Qu / (Qmax \cdot fi) \tag{43}$$

Total Savings

The energy savings, Qsav, from operating a boiler in modulation mode is the difference between total loss in an on/off boiler, Qt,o, and a modulating boiler, Qt,m:

$$Qsav = Qt,o - Qt,m \tag{44}$$

Example

For example, consider an on/off a boiler rated at 5 mmBtu/hour producing 30 psia steam at 250 F. Table 2 shows the input variables and values to calculate savings from switching from on/off to modulation control mode.

Table 2. Operating parameters for example boiler

Qmax (mmBtu/h) = Rated Burner Input	5.00
Tw_1 (F)	200
Tw_2 (F) = Tsteam or Thotwater	250
EA = Excess Air	0.5
Ta (F) = temperature comb air before burner	70
Tex (F) = temperature exhaust gasses	400
Db (ft) = Diameter Boiler	3
Lb (ft) = Length Boiler	10
Tb (F) = Temp Boiler Skin	110
h (Btu/h-ft2-F) = Conv Coeff	7
Ta (F) = Temp Indoor Air	70
Toa (F) = Temp Outdoor Air	50
Hs (ft) = Stack Height	30
Ds (ft) = Stack Diameter	2
PLR = Frac Time at Full Fire	0.5
N = Num cycles per hour	6
PstP (min) = Post-fire purge time	0.25
PreP (min) = Pre-fire purge time	0.25

Using the method described above, the average overall boiler efficiency, η_t, would increase from 73% to 81% from switching from on/off to modulation control mode, and energy saved, Qsav, would average about 240,997 Btu per hour.

IMPROVING BOILER EFFICIENCY BY REDUCING EXCESS AIR

According to the EPA document "Guide to Industrial Assessments for Pollution Prevention and Energy Efficiency" (5) the optimal quantity of excess air to guarantee complete combustion in most natural gas burners is about 10%. Unfortunately, many boilers use more than 10% excess air. This dilutes the combustion products, causing a lower combustion temperature and higher flow rate of combustion gasses.

As a first approximation, the savings from reducing excess air to 10%, the boiler efficiencies at the current level of excess air and at 10% excess air can be calculated, assuming the exhaust temperature remains constant. For example, consider a natural gas boiler using 50% excess combustion air at 70 F and exhaust gas temperature of 400 F. Using the method developed in the CALCULATING BOILER EFFICIENCY section, the boiler efficiency at these conditions would be 80.3% (Table 1). Assuming the exhaust temperature, Tex, remains constant and excess air were reduced to 10%, the boiler efficiency would improve to 82.8% (Table 3).

Table 3. Combustion efficiency results for 10% excess air assuming Tex to be constant

Input Data	
EA = excess air (0=stoch, 0.1 = optimum)	0.10
Ta = temperature combustion air before burner (F)	70
Tex = temperature exhaust gasses (F)	400
Constants (for Natural Gas)	
LHV = lower heating value (Btu/lb)	21,500
HHV = higher heating value (Btu/lb)	23,900
Cp_g = specific heat of combustion gasses (Btu/lb-F)	0.260
Tdpp = dew point temp of H20 in exhaust (F)	140
AFs = air/fuel mass ratio at stochiometric conditions	17.20
Calculated Values	
Tc (F) = temp combustion = Ta+LHV/[(1+(1+EA)(Afs))cpp]	4,221
Q_{lat} (Btu/lb) (Q_{lat} = HHV - LHV if Tex < 140 F, else Q_{lat} = 0)	0
η_b = [{1 + (1+EA)(AFs)}*Cp_g*(Tc-Tex) + Q_{lat}] /HHV	82.8%

Although this method is commonly used, it does not consider the reduction in combustion gas mass flow rate through the boiler. To consider this effect, the heat exchanger model developed in the MODELING A BOILER AS A HEAT EXCHANGER section can be used. When excess air is reduced to EA,n, Equation 3 can be substituted into Equation 17 to yield:

$$UA,n = UA\ [\{1 + (1 + EA,n) \cdot AFs\} / \{1 + (1 + EA) \cdot AFs\}]^{4/5} \quad (45)$$

The steps to calculate the boiler efficiency at reduced levels of excess air are:

1. Calculate the initial combustion temperature, Tc, and boiler efficiency, e_b, using Equations 7 and 10.
2. Calculate the heat transferred to the water/steam, Q, using Equation 11.
3. Calculate boiler heat transfer coefficient, UA, using Equations 12 and 13.
4. Calculate the new combustion temperature, Tc,n, at reduced excess air, EA,n, using Equation 7.
5. Calculate UA,n for the new excess air value using Equation 17.
6. Use the UA,n and Equations 12 and 13 again to calculate the new exhaust temperature at reduced excess air, Tex,n.
7. Use Tex,n and Equation 10 to calculate the new boiler efficiency, η_b,n at reduced excess air.

For example, consider the same boiler in the IMPROVING BOILER EFFICIENCY BY OPERATING IN MODULATION MODE section, except that the boiler runs continuously at 100% fraction of maximum heat input. Table 4 shows the calculations for the new efficiency, η_b,n, using the method described above. The results show that the combustion efficiency would increase from 80.3% to 82.6% if excess air were reduced to 10%.

Note that, in this case, the 82.6% efficiency calculated while considering heat transfer effects within the boiler is very close to the 82.8% efficiency calculated assuming Tex is constant and ignoring heat transfer effects. This appears to be a coincidence. In reality, three independent actions influence the exhaust gas temperature when the excess combustion air is reduced. The combustion temperature increases, which tends to increase the exhaust gas temperature. The residence time of combustion gasses in the boiler increases, which tends to decrease the exhaust gas temperature. And the convection coefficient decreases, which tends to increase the exhaust gas temperature. The net effect of the increased residence time and decreased convection coefficient, it to decrease the exhaust gas temperature and increase the efficiency. However, the effect of the increased combustion temperature is sufficient to increase the exhaust gas temperature to approximately the same temperature as before the excess air was reduced.

Table 4. LMTD method results from reducing EA to 10%

Input Data	
Ta = temperature combustion air before burner (F)	70
Tex = temperature exhaust gasses (F)	400
EA = excess air (0=stoch, 0.1 = optimum)	0.50
Qmax (mmBtu/h) = rated burner heat input	5.000
fi	1.00
Tw_1 (F)	200
Tw_2 (F) (for either "Steam" or "HotWater")	250
EAn = excess air (0=stoch, 0.1 = optimum)	0.10
"Steam" or "HotWater"	Steam
If "Steam" then enter enthalpy of sat steam leaving boiler, hw2 (Btu/lb)	1164
Constants (for Natural Gas)	
LHV = lower heating value (Btu/lb)	21,500
HHV = higher heating value (Btu/lb)	23,900
Cp_g = specific heat of combustion gasses (Btu/lb-F)	0.260
Tdpp = dew point temp of H20 in exhaust (F)	140
AFs = air/fuel mass ratio at stochiometric conditions	17.20
Calculated Values	
Tc (F) = temp combustion = Ta+LHV/[(1+(1+EA)(Afs))Cp_{ex}]	3,156
Q_{lat} (Btu/lb) (Q_{lat} = HHV - LHV if Tex < 140 F, else Q_{lat} = 0)	0
η_b = [{1 + (1+EA)(AFs)}*Cp_g*(Tc-Tex) + Q_{lat}] /HHV	80.3%
Q (Btu/h) = Useful heat transferred to water = Qmax x fi x 10^6 x η_b	4,016,854
ΔT_1 (F) = Tc - Tw_1	2,956
ΔT_2 (F) = Tex - Tw_2	150
ΔT_{lm} (F) = (ΔT_2- ΔT_1) / ln(ΔT_2 / ΔT_1)	941
UA (Btu/h-F) = Qu / (ΔT_{lm} (F))	4,268
Tcn (F) = new temp combustion = Ta+LHV/[(1+(1+EAn)(Afs))Cp_g]	4,221
UAn (Btu/h-F) = UA [{1 + (1 + EA,n) AFs} / {1 + (1 + EA) AFs}]$^{4/5}$	3,366
η_bn (with EAn but Tex const) = [1 + (1+EAn)(AFs)]*Cp_g*(Tcn-Tex)/HHV	82.8%
ΔT_1n (F) = Tcn - Tw_1	4,021
Guess current ΔT_2n (F) (lower than ΔT_2....)	157.8891
This should equal zero if ΔT_2n is correct	0
Tex,n (F) = ΔT_2n + Tw_2	408
η_bn = [1 + (1+EAn)(AFs)]*Cp_g*(Tcn-Texn)/HHV	82.6%

SUMMARY AND CONCLUSIONS

This paper presented methods for estimating energy savings from improving boiler operation. These methods have been incorporated into a free public-domain software application called BoilerSim, which is available from the University of Dayton Industrial Assessment Center at www.engr.udayton.edu/udiac.

To do so, the paper presented a method to calculate boiler efficiency from easily measured variables. A method to model a boiler as a parallel-flow heat exchanger was developed and verified against measured data. A method for estimating drift losses due to temperature and elevation differences was developed. These methods were then used to develop a method for calculating savings from a boiler operating in modulation mode rather than on/off mode. The results show that changing from on/off to modulation mode can result in significant savings by increasing average boiler efficiency by about 8%.

The methods were also used to develop a procedure for calculating the improvement in boiler efficiency due to reducing excess combustion air. These methods incorporated three related and important effects from reducing excess air; the combustion temperature increases, the gas-side convection coefficient decreases, and the residence time of the combustion gasses in the boiler increases. Interestingly, the results indicate that in some cases these effects tend to cancel each other, and the exhaust gas temperature remains about the same as before the quantity of excess air was reduced. Thus, the overall efficiency determined by simplistically assuming that the exhaust gas temperature remains constant is not much different from the overall efficiency determined by including the relevant combustion and heat transfer effects. However, this may not be true in all cases; thus we recommend use of the extended method, which incorporates the relevant combustion and heat transfer effects for calculating savings from reducing excess air.

REFERENCES

1. Cengel, Y., and M. Boles. 1998. Thermodynamics: An Engineering Approach, 3rd ed. Hightstown, NJ: McGraw-Hill, Inc.

2. Department of Energy, 2003, "PHAST" ver. 1.1.2.

3. Incropera, F., and D. DeWitt. 1996. Fundamentals of Heat and Mass Transfer, 4th ed. New York: John Wiley & Sons, Inc.

4. Jones, J. and A. West. 2001. "Natural Ventilation and Collaborative Design." ASHRAE Journal, November 2001.

5. U.S. Environmental Protection Agency. "Guide to Industrial Assessments for Pollution and Energy Efficiency." EPA/625/R-99/003. Cincinnati: June 2001.