Understand Steam Generator Performance

The key performance variables are excess air, fuel type, exit gas temperature, load, and emissions.

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Several variables affect the plant engineers plan their opera tion better.

This article discusses the effects of such variables as excess air, fuel type, exit gas temperature, load, and emissions on generator design and operation. It also discusses some of the potential benefits of customized steam generators over standard, prepackaged designs, which often compromise on overall performance. The focus of the article is limited to gaseous and oil fuels.

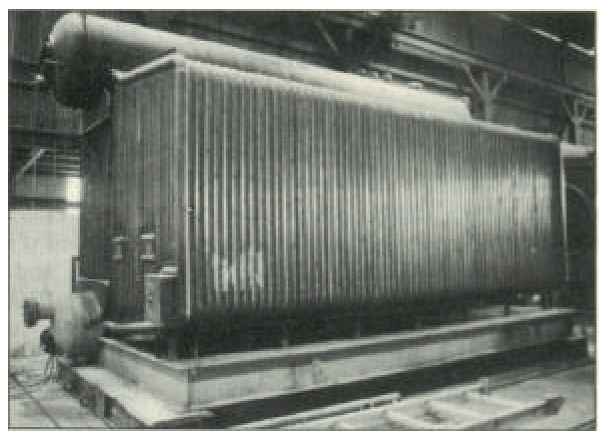
BOILER EFFICIENCY

The single most important variable from a performance standpoint is the steam generator efficiency. This is particularly true for base-load steam generators that will operate most of the time (unlike a standby boiler, which operates for only a few hours per year).

During the design stage, the consultant or the end user specifies a certain efficiency. Efficiency is primarily affected by the fuel composition, unburned carbon losses, excess air, exit gas temperature, and the type of fuel (Table 1).

Fuel composition

The fuel composition is important, as it affects the flue gas composition, which in turn affects the various heat losses. While variations may not be significant between typical natural gases, differences between a low-Btu and a high-Btu gas do matter.



• *Figure 1. Large packaged steam generator.*

increase the fuel moisture loss. Similarly, the percentages of hydrogen and carbon in oil fuels affect the fuel moisture loss and hence the efficiency, as shown in Table 1.

Unburned carbon losses

The various boiler heat losses are evaluated at the design stage using the American Society of Mechanical Engineers' Power Test Code heat-loss method, ASME PTC 4.1.

One of the losses impacted by the combustion process is the unburned carbon loss. Carbon in the fuel is converted to carbon monoxide instead of carbon dioxide, which results in lower carbon utilization. With gaseous fuels this may be insignificant. However, with fuel oils, the amount of CO formed can be very high - on the order of several hundred ppm.

The type of burner used, the amount of excess air, turbulence in the combustion zone, and the type of furnace construction (that is, whether it is a membrane wall or tangent tube design) influence this loss. Leakage of combustion products from the furnace to the convection section via a tangent tube partition wall contributes a great deal to CO formation, because the combustion products do not have the residence time in the furnace to complete combustion before entering the convection section. A high excess-air operation may be required to minimize this loss; however, this decreases the efficiency due to increased heat losses, as shown in Table 1.

The loss L (in Btu/lb) due to CO formation is given by:

$$L = 10{,}160\ C\ [CO/(CO + CO2)]$$

where CO and CO_2 are the volume percentages in the flue gas and C is the weight fraction of carbon in the fuel.

Excess air

Excess air affects efficiency significantly, as indicated in Table l.

The choice of how much excess air to use depends on the type of fuel used and the desired levels of NOx and CO emissions, as well as the degree of flue gas recirculation (FGR). Burner suppliers often recommend the amount of excess air after reviewing the emission levels to be guaranteed, the fuel analysis, and the furnace dimensions. A high excess air (on the order of 10-15%) is often suggested even for natural gas fuels. This is because FGR is used to limit NO, which in turn affects the burnout of CO; higher excess air helps to complete combustion. Figure 2 shows the typical relationship between excess air and emissions.

Table 1. Boiler efficiency as a function of excess air and exit gas temperature.

Fuel	Gas				Oil			
Excess Air, %	5	5	15	15	5	5	15	15
T_{go}, °F	350	450	350	450	350	450	350	450
Flue gas composition:								
CO_2, %v	9.06	9.06	8.34	8.34	12.88	12.88	11.82	11.82
H_2O, %v	19.11	19.11	17.70	17.70	12.37	12.37	11.47	11.47
N_2, %v	70.93	70.93	71.48	71.48	73.83	73.83	74.19	74.19
O_2, %v	0.90	0.90	2.48	2.48	0.92	0.92	2.53	2.53
MW	27.57	27.57	27.66	27.66	28.86	28.86	28.97	28.97
w_g/w_f	19.17	19.17	20.9	20.9	16.31	16.31	17.77	17.77
Efficiency losses:								
L_1, %	4.74	6.44	5.23	7.09	5.13	6.96	5.62	7.63
L_2, %	0.09	0.12	0.10	0.13	0.09	0.12	0.10	0.14
L_3, %	10.89	11.32	10.89	11.32	6.63	6.89	6.63	6.89
L_4, %	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
Efficiency:								
E_h, %	83.2	81.1	82.9	80.5	87.1	85.0	86.7	84.3
E_l, %	92.3	89.9	91.7	89.2	92.8	90.0	92.3	89.9

Key:

T_{go} = exit gas temperature, °F
w_g/w_f = ratio of flue gas to fuel
E_h = efficiency based on HHV
E_l = efficiency based on LHV

L_1 = dry gas loss
L_2 = air moisture loss
L_3 = loss due to moisture from combustion of fuel
L_4 = radiation losses

Fuel characteristics:

Oil: C = 87.5%, H_2 = 12. 5% (by weight); HHV = 19,727 Btu/lb, LHV = 18, 512 Btu/lb
Gas: CH_4 = 97%, C_2H_6 = 2%, C_3H_8 = 1% (by volume); HHV = 23,789 Btu/lb, LHV = 21,462 Btu/lb

Source: *(3)*.

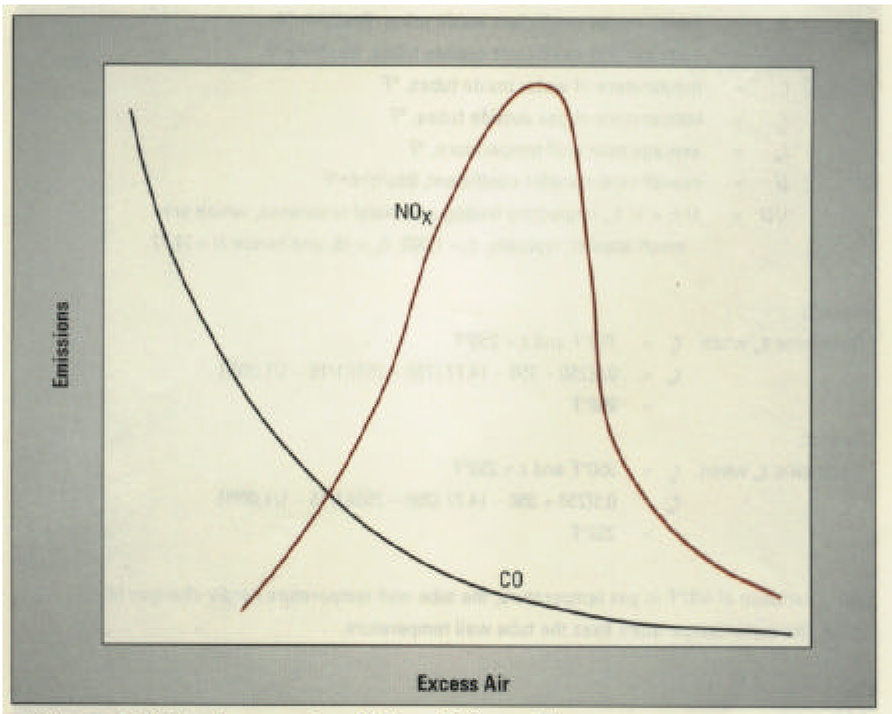


Figure 2. Effect of excess air on CO and NOx emissions.

Table 2. Acid dew point is a function of partial pressure.

The following equations can be used to calculate the dew points of several acids commonly found in flue gases, where T_{DP} is the dew point in Kelvin and P is the partial pressure in mm Hg:

Hydrobromic Acid:

$$1{,}000/T_{DP} = 3.5639 - 0.1350 \ln(P_{H2O}) - 0.0398 \ln(P_{HBr}) + 0.00235 \ln(P_{H2O}) \ln(P_{HBr})$$

Hydrochloric Acid:

$$1{,}000/T_{DP} = 3.7368 - 0.1591 \ln(P_{H2O}) - 0.0326 \ln(P_{HCl}) + 0.00269 \ln(P_{H2O}) \ln(P_{HCl})$$

Nitric Acid:

$$1{,}000/T_{DP} = 3.6614 - 0.1446 \ln(P_{H2O}) - 0.0827 \ln(P_{HNO3}) + 0.00756 \ln(P_{H2O}) \ln(P_{HNO3})$$

Sulfurous Acid:

$$1{,}000/T_{DP} = 3.9526 - 0.1863 \ln(P_{H2O}) + 0.000867 \ln(P_{SO2}) - 0.000913 \ln(P_{H2O}) \ln(P_{SO2})$$

Sulfuric Acid:

$$1{,}000/T_{DP} = 2.276 - 0.0294 \ln(P_{H2O}) - 0.0858 \ln(P_{H2SO4}) + 0.0062 \ln(P_{H2O}) \ln(P_{H2SO4})$$

Source: *(1, 4, 5)*.

Table 3. Calculating tube wall temperature at different gas temperatures.

The average wall temperature of a bare tube economizer is given by the equation:

$$t_w = 0.5[t_i + t_g - U(t_g - t_i)(1/h_o - 1/h_i)]$$

where:

h_i = heat-transfer coefficient inside tubes, Btu/ft²•h•°F
h_o = heat-transfer coefficient outside tubes, Btu/ft²•h•°F
t_i = temperature of water inside tubes, °F
t_g = temperature of gas outside tubes, °F
t_w = average tube wall temperature, °F
U = overall heat-transfer coefficient, Btu/ft²•h•°F
$1/U$ = $1/h_i + 1/h_o$, neglecting fouling and metal resistance, which are much smaller; typically, $h_i = 1{,}000$, $h_o = 15$, and hence $U = 14.77$

Case 1:

Determine t_w when t_g = 750°F and t_i = 250°F
t_w = 0.5[250 + 750 − 14.77 (750 − 250)(1/15 − 1/1,000)]
= 258°F

Case 2:

Determine t_w when t_g = 350°F and t_i = 250°F
t_w = 0.5[250 + 350 − 14.77 (350 − 250)(1/15 − 1/1,000)]
= 252°F

For a variation of 400°F in gas temperature, the tube wall temperature hardly changes (6°F). Thus, the water temperature fixes the tube wall temperature.

Source: *(1, 2, 6)*.

For each fuel there is range of excess air that achieves the desired CO and NOx levels. Higher molecular weight hydrocarbons have a higher flame temperature, which produces higher NOx, which in turn requires a higher degree of FGR to limit it, which in turn may require higher excess air, depending on the CO levels to be guaranteed. The experience of burner suppliers with units burning similar fuels and having similar emissions often determines this parameter.

Exit gas temperature

The lower the exit gas temperature, the higher the boiler efficiency. A rule of thumb is that every 40°F difference is equivalent to a 1 % change in efficiency. However, if the temperature of the feed water entering the economizer is higher, then the stack temperature will also be higher. Otherwise, a very large economizer may be required to maintain the same exit gas temperature compared to a low feed water temperature case.

One factor influencing the exit gas temperature is the sulfuric acid dew point. When fuels containing sulfur are fired, SO_2 is formed, and some of it (1-5%) is converted to SO_3, particularly if vanadium is present in the oil ash, which acts as a catalyst. When the acid vapor gets cooled by the feed water, condensation may occur if the temperature of the tube surface is at or below the acid dew point. Dew point is a function of the partial pressure of the acid vapor and water vapor in the flue gases. Table 2 shows a few correlations for acid vapors (1, 4, 5).

There is a misconception, even among experienced engineers, that condensation in the economizer can be avoided by maintaining the exit gas temperature high enough. When an economizer is used as the heat recovery equipment, the cold end temperature is mainly a function of the feed water temperature entering the boiler and the gas temperature has little effect on it. This is due to the high tube-side heat-transfer coefficient. Table 3 shows a simple calculation where the exit gas temperature varies by as much as

400°F, while the economizer tube wall temperature varies by only a few degrees (/, 2, 6).

However, it is not necessary to have the feed water temperature at or above the acid dew point to minimize corrosion problems. While this is a sure way to prevent acid condensation, research has shown that corrosion is significant at 50-100°F below the acid dew point, as illustrated in Figure 3 (/, 6). Hence, one need not specify a high feed water temperature, which results in a lower efficiency. If the acid dew point is, say, 275°F, a 230-250°F feed water temperature is a good compromise. An exit gas temperature of 300-320°F can be achieved with moderate costs.

Air heaters are often avoided as back-end heat-recovery equipment. This is because they contribute to higher combustion temperature and hence NOx, which calls for higher FGR rates, which in turn results in a higher gas pressure drop through the boiler or a larger boiler or both. Also, the gas pressure drop through an air heater is much higher, by 2-3 in. w.c., which is a continuous loss of energy.

When an air heater is used, it is usually one of two types - the tubular or recuperative, or the regenerative or rotary air heater. Rotary air heaters have the additional problem of leakage from the air to the gas side, which affects the fan size and air heater performance. An advantage of air heaters, though, is that they may be more compact than tubular air heaters. Air heaters are primarily recommended when difficult fuels, such as solid fuels and low-Btu gaseous fuels, are fired.

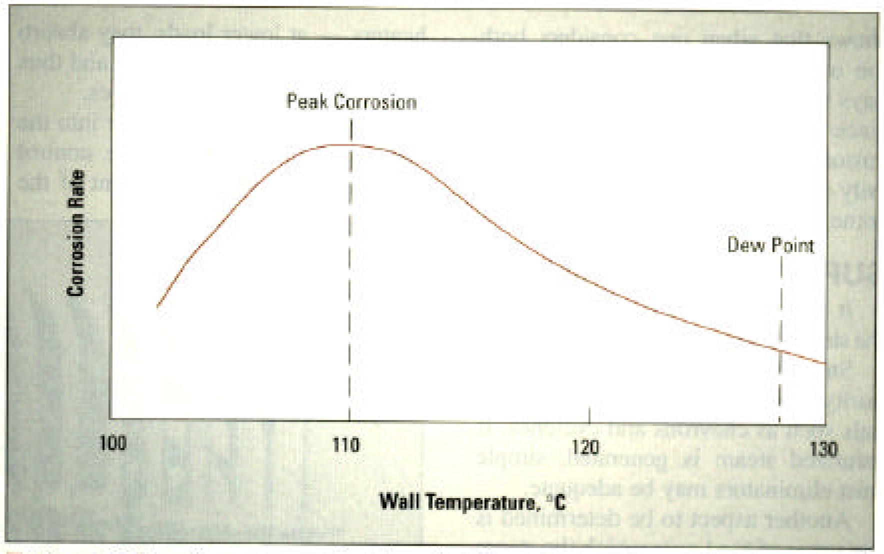


Figure 3. Corrosion rate as a function of wall temperature. Source: (1, 6).

Table 4. Gas flow, in lb flue gas/lb fuel, as a function of excess air and FGR.

	Flue Gas Recirculation, %		
Excess Air, %	0	10	20
5	19.17	21.00	23.00
10	20.00	22.00	24.00
20	21.76	24.00	26.10

Based on natural gas fuel.

Flue gas quantity

Using higher excess air and FGR rates for the same boiler duty increases the flue gas quantity to be handled by the boiler. This naturally results in a larger boiler or, if dimensions are limited because of shipping restrictions (which is often the case), a higher gas pressure drop is incurred in the convection section and economizer. Table 4 shows the flue gas quantity produced with different excess air and FGR factors.

This is one of the reasons why it is often beneficial to go with a custom designed steam generator - tube spacings, tube height, and the number of tube sections can be varied to minimize gas pressure drop.

Unfortunately, packaged boilers are often treated as predesigned or off-the-shelf items. Some consultants even suggest model numbers while developing specifications. This practice should be avoided. Otherwise you could be purchasing a design that was developed several decades ago, when emissions were not a concern and FGR was not considered while sizing the convection section and economizer. These steam generators were designed with skimpy furnaces and low excess air factors and without any FGR. Hence, even if the steam parameters may be the same, the flue gas flow through the unit can be nearly 30-40% more, resulting in higher gas pressure drops, higher exit gas temperatures, and therefore lower efficiency and much higher operating costs. Engineers should understand these aspects and opt for custom designed units, which can incorporate several design features to minimize operating costs and improve efficiency.

As an example, if 130,000 lb/h of flue gases have to be forced through an additional 3 in. w.c. in the boiler because a standard rather than custom design is used, the additional fan power consumption is 14.5 kW, based on a 70°F air temperature and a fan efficiency of 70%. Assuming that the boiler operates for 8,000 h/yr and that electricity costs 1 ¢/kWh, the additional cost is $11,600/yr. Capitalizing this cost over the lifetime of the boiler shows that when one considers both the operating and the initial costs, it pays to select a custom designed unit, since the additional capital costs for a custom-designed system is generally only about $20,000 to $30,000, and in some cases is virtually nil.

SUPERHEATER DESIGN

It is very important to know where the steam generated by the boiler is used. Steam turbines require a high steam purity, which calls for good drum internals such as chevrons and cyclones. If saturated steam is generated, simple mist eliminators may be adequate.

Another aspect to be determined is the range of load over which the steam temperature has to be maintained. A wide load range calls for a large and, therefore, expensive superheater. Consultants must discuss with clients and turbine suppliers before specifying this requirement. While 70-100% load range of superheat temperature control is common, some designers unknowingly specify steam temperature control from 40% to 100% load, which complicates the superheater design.

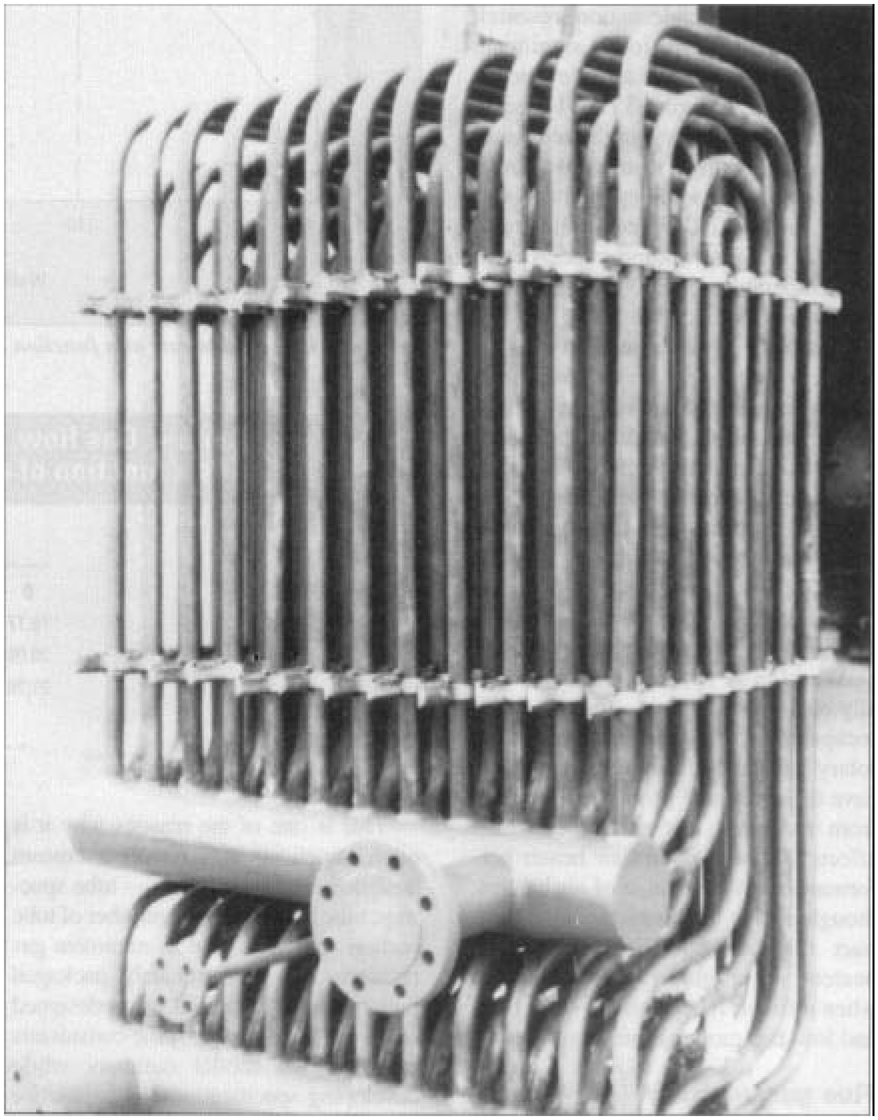
The superheater is the equipment most significantly affected by parameters such as excess air, flue gas recirculation, and furnace sizing. A larger furnace results in a lower exit gas temperature, which increases the superheater size due to the lower log mean temperature difference available. A high FGR rate increases the size of the superheater and the gas pressure drop.

Typically, the steam temperature is maintained at 70% to 100% load. With convective type superheaters (Figure 4), this means that the steam temperature will be higher at higher loads. Interstage attemperation should be incorporated to control the steam temperature at higher loads.

Radiant superheaters, which are located in the furnace zone or exposed to direct flame radiation, generally operate at higher tube wall temperatures. Thus, unless these units are very carefully designed, failures are more likely.

Radiant superheaters behave differently than convective superheaters - at lower loads, they absorb more energy due to radiation and thus have higher steam temperatures.

The spraying of feed water into the steam for steam temperature control can increase the solids content of the steam. Thus, the feed water should have the same amount of solids as the final steam - in the ppb range. Demineralized water is preferred for such applications. If demineralized water is not available, saturated steam may be condensed in a heat exchanger and sprayed into steam, as shown in Figure 5a and 5b.



• Figure 4. Superheater for steam generator.

Low-load operation

There are also a few issues of concern at low loads, particularly with superheater and fan operation.

The number of streams (the area through which steam flows in a superheater) has to be chosen so that a good distribution occurs even when the boiler operates at the lowest load. Some consultants specify turndown of 1:8 or even 1:10. From a practical point of view, a turndown this high is not recommended.

The real problem is in the ability of the superheater and fan to handle such low-load conditions. If the steam pressure drop is, say, 50 psi at 100% load, it will be 3 psi at 25% load. At lower loads, it is difficult to ensure that the flow distribution will be uniform through all the tubes. One has to be also concerned about reverse flows, which can result in overheating of some tubes and possibly failure.

The author recommends a load range of 50% to 100%, not 10% to 100%, since performance is difficult to predict at low gas and steam velocities.

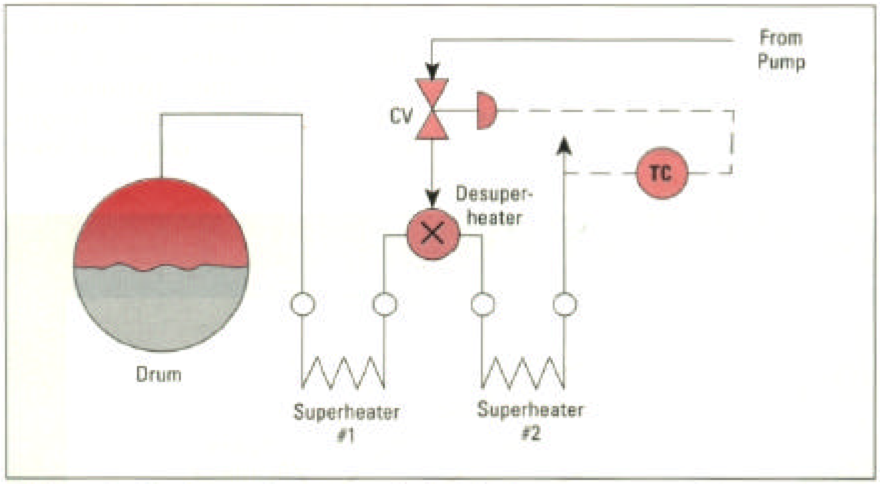
Another problem with low-load operation is the fan performance. If the fan is selected with high margins on flow and head, then at low loads, the operating point may fall below the capacity of the fan even at the lowest vane opening position (Figure 6). This may cause problems with fan operation, such as vibrations, instability, and noise. This is likely in packaged boilers, which typically have one forced draft fan. If two fans each having half the capacity are used instead, then a higher turndown is feasible. Also, at low flows, the fan operating point can drift into the unstable operating regimes of the fan curve.

Using a high margin for the fan flow and head should be avoided. The author suggests 10% on flow and 20% on head.

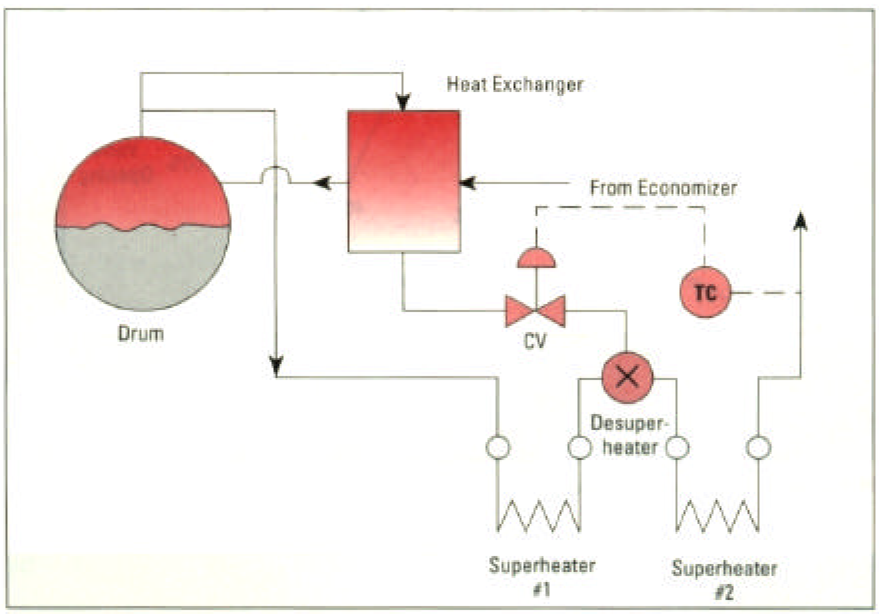
LOAD VS. PERFORMANCE

Figure 7 shows the performance of a boiler at different loads. The efficiency peaks at a certain load and then drops off. This is expected, as the nature of the two important losses, namely radiation and flue gas heat losses, differ. At higher loads, the radiation loss will be lower and the heat losses due to the flue gases will be higher; the opposite is true at lower loads. The combination of these losses results in a peak efficiency, at some load between 0% and 100%.

The exit gas temperature drops off with load. An economizer acts as a heat sink, which limits the gas temperature so that gas is not cooled to dew point levels.



■ *Figure 5a. The desuperheater controls the final steam temperature by spraying feed water into the steam.*



■ *Figure 5b. Saturated steam is condensed in a heat exchanger and condensate is sprayed into the desuperheater.*

The approach point, or the feed water temperature leaving the economizer, decreases when load decreases (unlike in a gas turbine heat-recovery steam generator, or HRSG). Hence, steaming is not a problem at low loads.

CUSTOMIZED DESIGNS As mentioned earlier, adopting a standard design developed decades ago to present-day operating conditions with high excess air and FGR rates results in a compromise on performance and operating costs. Customized designs can overcome these concerns. Following are some of the aspects reviewed in customized designs.

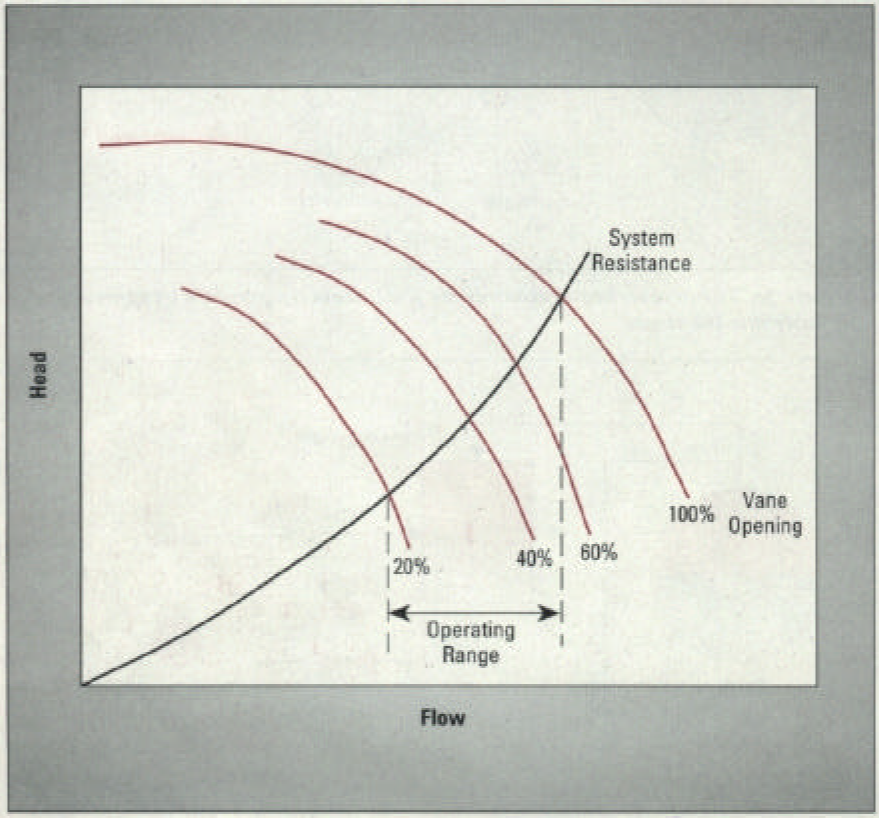
1. Furnace design, which considers the excess air and FGR rates and flame dimensions for each fuel, after discussions with the burner supplier so that the flame is contained entirely within the furnace.

2. Convection section design, which can have longer tubes, wider tube spacing, and more tube sections to reduce gas velocity and hence pressure drop to acceptable levels.

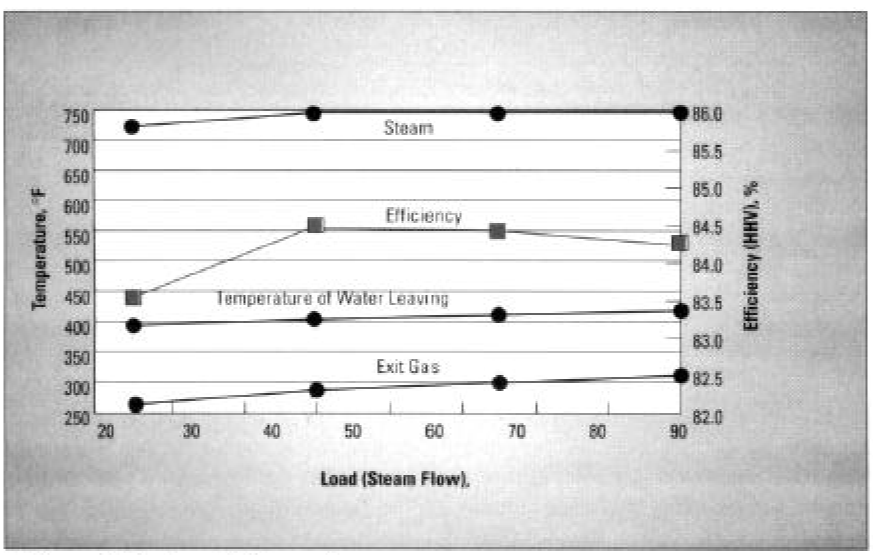
3. The possible use of extended surfaces in the convection section if clean fuels are fired. Extended surfaces can result in compact designs, lower gas pressure drop, and lower exit gas temperatures from the convection section (1).

4. Superheater location and whether it is in the appropriate gas temperature zone in the convection section to operate safely over a wide load range. If the steam temperature is low enough, the superheater could even be located downstream of the convection section and ahead of the economizer.

5. Horizontal or vertical economizers to match the layout requirements.



■ *Figure 6. Fan operating range.*



■ *Figure 7. Packaged boiler performance.*

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Further Reading

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