Understand the basics of packaged steam generators

A custom design has many advantages over older, standard ones

V Ganapathy, ABCO Industries, Abilene.

Custom-designed steam generators should be considered when evaluating packaged boiler designs because standard designs have several limitations. Merits of convective superheaters over radiant designs should be understood by end-users. Evaluate operating and life-cycle costs for each boiler application. The unit with lowest life-cycle cost and good design should be selected, not the boiler with lowest initial cost alone. Write specifications clearly and avoid comparisons based on surface areas.

Specifying or selecting boilers based on pre-engineered designs/tables is not recommended. Refractory-lined designs have poor design and performance features and serious maintenance concerns. Operating at low loads is difficult for fans and superheaters and should be reviewed carefully.

Oil- and gas-fired packaged steam generators are widely used in chemical plants, refineries and cogeneration systems. They form an important part of the total steam system in any plant and are available in capacities up to 250,000 lb/hr at pressures ranging from 150 to 1,500 psig and temperatures from saturated steam to 1,000°F. They are expected to last about 25 years, and therefore, cost-effectively generate steam.

However, when making purchasing decisions, plant engineers, consultants and engineering firms spend little time on important aspects such as long-term performance and operating costs and often recommend standard, off-the shelf designs that have several limitations. Here are important design and performance aspects of packaged steam generators and recent trends in their design that engineers should be familiar with.

Table 1. Effect of excess air and FGR on flue gas

Cases 1–3: 100,000 lb/h, 150 psig saturated steam, 230°F feed water, 3% blowdown
Cases 4–6: 100,000 lb/h, 600 psig, 750°F steam, 230°F feed water, 3% blowdown
Assumptions: fan efficiency = 70%; 8 in. WC burner/duct losses in all cases

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Duty, MMBtu	100.2	100.2	100.2	118.8	118.8	118.8
Exit gas, °F	320	320	500	320	320	700
Excess air, %	10	15	15	10	15	15
FGR, %	0	15	15	0	15	15
Gas flow, lb/h	100,385	120,730	127,100	119,000	143,150	160,000
Fuel input, MMBtu	119.3	119.6	125.9	141.4	141.8	158.6
Gas pressure drop, in. WC	10	15	16	14	20	27
Fan power, kW	68	104	114	98	150	209
Efficiency, % HHV	84.00	83.80	79.57	84.00	83.80	74.90

Note: With larger flue gas quantity flowing through a given boiler, the exit gas temperature will also be higher, which is not considered in this table. Approximately 40°F increase in exit gas temperature decreases boiler efficiency by 1%, which is equivalent to $30,000 based on a fuel cost of $3/MMBtu.

Standard designs have limitations. Standard boiler designs were developed decades ago by boiler vendors to simplify the process of manufacturing and purchasing packaged boilers. Tables were developed showing major dimensions, surface areas, tube count, etc., and the consultant's job was only to select a model number for a given steam capacity from pre-engineered designs; plant layout was not difficult because the major dimensions and weight details were known. Engineering and manufacturing hours were greatly reduced for the boiler supplier, resulting in lower initial costs.

However, when standard designs were developed, there were no emission regulations. Low excess air, about 5% to 10% air, was used to attain high boiler efficiencies for natural gas and fuel oils. Using flue-gas recirculation (FGR) to reduce NO_x emissions was relatively unknown. Today, 15% to 20% excess air and 15% to 25% FGR rates are common for natural gas-fired boilers to attain NOx less than 30 ppm and CO less than 100 ppm. Also, the flue gas quantity flowing through the boiler is directly proportional to the amount of fuel fired:

Fuel fired = boiler duty/efficiency(1)

Boiler duty, or energy absorbed by steam, depends on whether there is a superheater or not. Efficiency depends on the exit gas temperature, which in turn depends on the presence of economizer. Thus, even if the boiler capacity is a nominal 100,000 lb/h as shown in Table 1, due to the effect of duty and excess air and FGR rates, a significant difference in flue gas mass flow results among the various cases.

Cases 3 and 6 have no economizer, which affects the fuel input. In case 5, which has a superheater, the flue gas quantity is nearly 40%

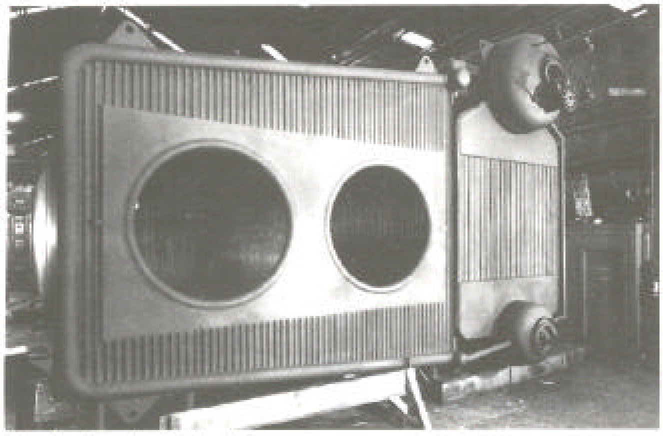

Fig. 1. A packaged boiler.

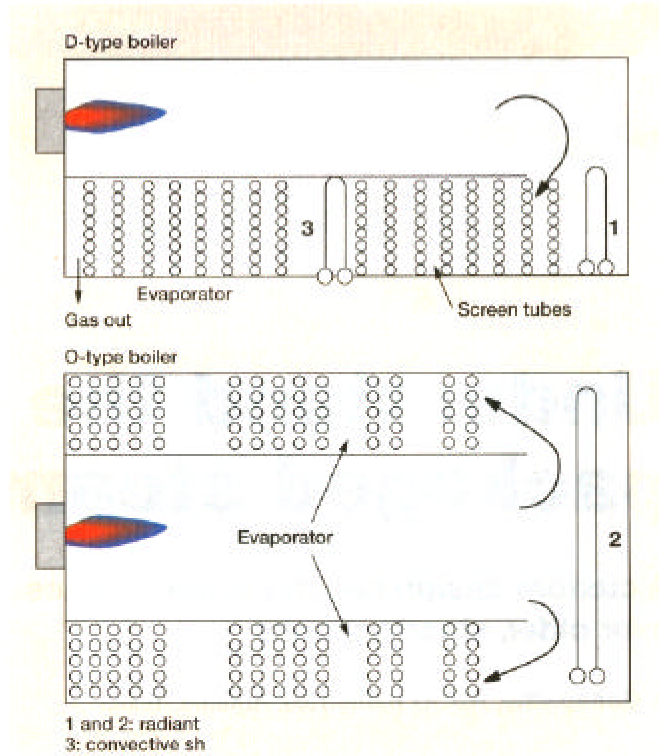

Fig. 2. Radiant and convective superheaters in a D-type and O-type boiler.

more than that of case 1 and, therefore, gas pressure drop across the convection section is doubled. Calculations are based on the assumption that no efforts were made by the boiler supplier to offer a larger unit or change the tube lengths, pitches, tube rows, etc., to lower the gas pressure drop as in custom-designed units. The increase in gas pressure drop causes additional fan power consumption, which is an operating cost.

Fan power consumption (kW)

$$= \frac{W \times \Delta h_w \times 62.4 \times 0.746}{(3{,}600 \times 12 \times 550 \times 0.7 \times 0.075)} \quad (2)$$

$= 0.0000373\ W \times \Delta h_w$ where W = flue gas flow, lb/h

Δh_w = static head or pressure drop in boiler system, in.WC

62.4 = density of water

0.746 = conversion from hp to kW 550 = conversion from ft-lb/s to hp .075 = density of air, lb/ft^3

70% efficiency was used for the fan and an additional 8 in.WC loss was assumed for the burner and duct work. Case 1 power consumption =100,385 x 18 x 0.0000373 = 68 kW and case 5 power consumption = 143,150 x 28 x 0.000373 = 150 kW. The difference is about 82 kW In reality, the difference is more if actual burner drop and duct losses are considered.

Over a year, additional operating costs = 82 x 8,000 x 0.05 = $32,800 (at $0.05/kWh), which is not a small amount. The fan size is also larger. Hence, if actual operating cost is evaluated, the standard design is not cost-effective, although its initial price may be attractive. This could be a costly error for the end-user, who will operate the boiler for 20 to 30 years.

In Table 1, the effect of additional mass flow on boiler exit gas temperature is not considered, which makes the situation even worse. Exit gas temperature increases by 10°F to 40°F if gas flow increases in a given boiler. Note that a 40°F rise in exit gas temperature is equivalent to 1% change in boiler efficiency, or about $30,000/yr, based on a fuel cost of $3/MMBtu. A better selection would have been a model generating 20% to 40% more steam or a 130,000lb/hr to 140,000-lb/h boiler vs. the nominal 100,000-lb/h unit. This does not happen often because vendors want to be competitive and push the lowest cost option. Custom designing is the only way to arrive at optimum designs.

Custom-designed boilers. Custom designing starts with an understanding of the boiler parameters, desired emission levels and fuels fired. The starting point is a furnace design based on discussions with a burner supplier, who reviews information on furnace dimensions, excess air and FGR rates and gives approval. The furnace, convection section, superheater and economizer are then designed.

Based on actual gas flow generated, efforts are made to minimize the gas pressure drop by adjusting the boiler height, tube spacings, tube counts and even possibly using finned tubes in the convection section if the gas is clean. Thus, every boiler is designed new and not pulled up from a pre-engineered table. The result is a unit with high efficiency and low operating cost, meeting the desired emission levels without flame impingement concerns.

Furnace performance. This is the most important component of any steam generator. Its performance affects not only the combustion process but also the heat-transfer surfaces located beyond the furnace such as superheaters, convection section and economizer. Using techniques such as low-NOx burners, staged combustion and FGR, the flame shape in the furnace will be different. Result: a possibility that the furnace dimensions of standard designs are inadequate and flame impingement may occur on the furnace walls.

With custom-designed boilers, furnace dimensions are reviewed with the burner supplier along with a fuel analysis and required emission levels. The furnace dimensions are based on the burner supplier's recommendations and not vice-versa (checking if a given furnace is adequate for the project in question).

Another poor practice carried over from decades ago is

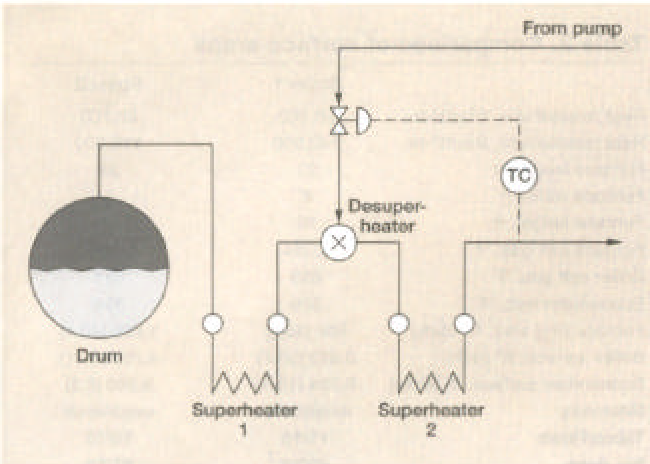

Fig. 3a. Final steam temperature is controlled by water in the desuperheater.

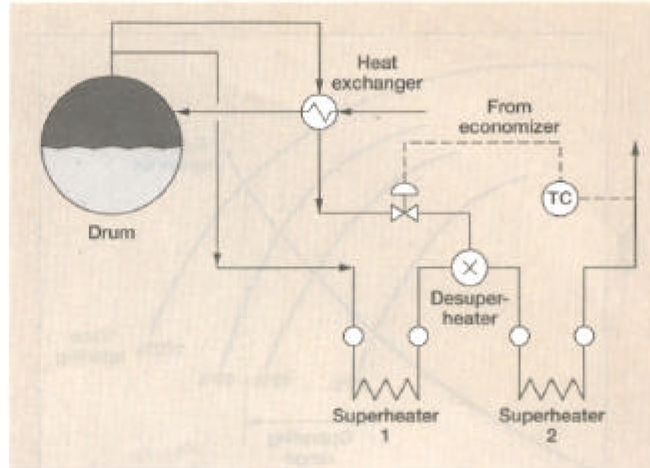

Fig. 3b. Condensate is used in this configuration to control the steam temperature.

using refractory in the furnace floor, front and side walls. Engineers and boiler designers who had little experience with boiling heat transfer and circulation in those days found this practice convenient to prevent overheating of floor tubes. However, with the abundance of information on boiling heat transfer and boiler circulation, backed up by the operation of several hundred units, this practice is not necessary. Using refractory also reduces the furnace effective radiant surface area and increases the area heat release rate and heat flux.

An important recent development is a completely water-cooled furnace (Fig. 1) with several advantages:

1. The furnace front, rear, side walls and floor are completely water-cooled and are of membrane wall construction, resulting in a leak-proof enclosure for the flame. The entire furnace expands and contracts uniformly, thus avoiding casing expansion problems. When refractory is used on front and rear walls, hot gases leaking from the furnace are always possible. Casing corrosion is also likely since a gas-tight joint is difficult to ensure between refractory-lined casings and water-cooled walls; when corrosive gases condense there is corrosion.

2. Problems associated with refractory maintenance are eliminated. There is no need for a shutdown to check the refractory or replace it.

3. Fast startup rates are difficult with refractory-lined boilers. With a completely water-cooled furnace, quick startups are possible. This is important in cogeneration projects where the packaged boiler must supply steam to the end-user as soon as the heat recovery system fails.

4. Heat-release rates, on an area basis, are lower with a water-cooled furnace (for the same furnace volume) compared to a refractory-lined unit which has less cooling surface. This also results in reduced heat flux. Note that area heat-release rate is a more significant parameter than volumetric heat-release rate, which gives an indication of residence time for combustion products and is pertinent only for difficult-to-burn fuels like solid fuels. Area heatrelease rate affects furnace heat flux and departure-fromnucleate boiling (DNB) conditions and is significant. Typical area heat-release rates vary from 100,000 to 175,000 Btu/ft^2-hr for packaged boilers. It makes more sense to specify area heat-release rates rather than volumetric rates.

5. Reradiation from the refractory increases the flame's local combustion temperature, which in turn increases NOx formation. Water-cooled front walls, especially the front wall where NO, formation potential is the highest, have a beneficial effect on the flame -they cool it effectively so NOx formation is reduced.

6. Another problem with using refractory in the furnace is an increase in furnace exit gas temperature, which raises the radiant heat flux and causes tube failures in the radiant superheater (if present).

Radiant vs. convective superheaters. Radiant superheaters are widely used in packaged steam generators (Fig 2). They are prone to frequent tube failures because of their location. Convective superheaters, located behind several screen tubes, have fewer maintenance concerns and a much longer life due to their lower tube wall temperatures; but their size and cost are higher due to a lower log-mean temperature difference. The following points on radiant vs. convective designs should be understood by potential end-users, who can influence design specifications and evaluation.

1. Radiant superheaters are located at the furnace exit or turning section as shown in Fig 2. The furnace exit gas temperature is a difficult parameter to estimate. Variations in excess air, FGR rates and flame shape also add to the difficulty. The furnace exit gas temperature could be off by 100°F to 200°F from predicted values. The turning section is also subject to turbulence and nonuniformity in gas temperature profiles, which also hinders an accurate superheater performance evaluation. Thus, radiant superheater tube wall temperatures could be underestimated significantly, leading to tube overheating and failures.

2. Several boilers operate at partial loads of less than 50% for significant time periods. The radiant superheater, by its nature, absorbs more enthalpy at partial loads compared to convective designs. Also, at partial loads, steam flow distribution inside the superheater tubes is less uniform and often questionable. If at 100% load, the superheater pressure drop is 30 psi, then at 25% load the pressure drop is barely 2 psi-this may not ensure good steam flow distribution through all the tubes. Gas-side mixing will also be poor due to low gas velocities. So there is a double negative of higher radiant energy and poor steam and gas flow distribution, which is likely to cause overheating of a few tubes in the radiant superheater.

The convective superheater, conversely, is located

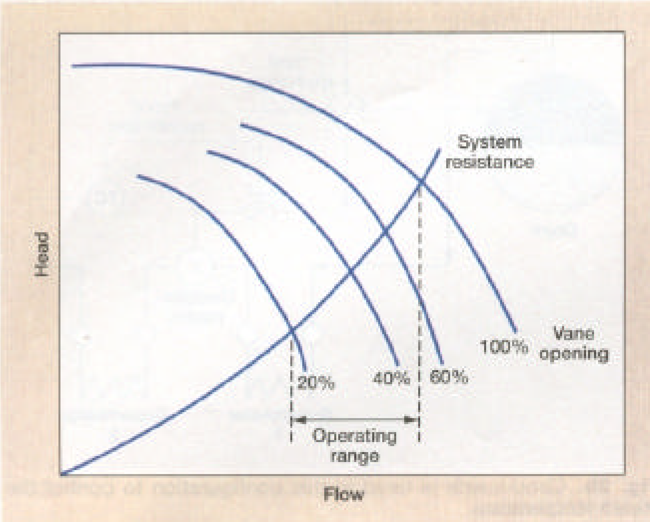

Fig. 4. Fans can operate at 20% to 100% vane openings.

beyond several screen tubes. Thus, better gas-side mixing is likely. The heat flux and gas temperature entering it are lower, resulting in a less hostile environment at all loads. So, their performance can be predicted more accurately than radiant designs. The screen section can be larger when the required steam temperature is less, thus ensuring low tube-wall temperatures. Radiant superheaters, however, are always located at the highest gas temperature zone irrespective of whether the degree of superheat is 20°F or 400°F.

3. Multistage superheaters with interstage desuperheating can be used with convective designs to ensure that steam temperatures are not exceeded and the tube-wall temperatures are predictable and under control.

End-users are better off with convective superheater designs-their size and cost may be more but their life is longer, with fewer maintenance concerns.

Developing boiler specifications. Consultants and AE firms responsible for purchasing steam generators should develop good and clear specifications. Highlight the following aspects.

Steam parameters such as flow, pressure, degree of superheat and feed water temperature should be stated along with feed water quality and steam purity desired. This enables the designer to select proper drum internals. Amount of blowdown to be used while determining the boiler duty can also be estimated if feed water quality is specified. This is important since it affects boiler duty and the amount of fuel fired.

If feed water is used for interstage attemperation, the water should be demineralized with preferably zero solids. Otherwise, solids from the spray water will carry over into the steam and deposit in the superheaters or steam turbine. If demineralized water is not available, then the boiler designer can engineer a condensate spray system, which essentially condenses the desired quantity of steam using feed water and uses it for desuperheating (Fig 3).

Often, specifications do not state if steam for deaeration is from the boiler or another source. This is important if the deaerator is supplied by the boiler vendor. The total steam must be increased by 10% to 15%, depending on the temperature and amount of condensate

Table 2. Comparison of surface areas

	Boiler 1	Boiler 2
Heat release rate, Btu/ft^3-hr	90,500	68,700
Heat release rate, Btu/ft^2-hr	148,900	116,500
Furnace length, ft	22	29
Furnace width, ft	6	6
Furnace height, ft	10	10
Furnace exit gas, °F	2,364	2,255
Boiler exit gas, °F	683	611
Economizer exit, °F	315	315
Furnace proj area, ft^2 (duty)	802 (36.6)	1,026 (40.4)
Boiler surface, ft^2 (duty)	3,972 (53.7)	4,760 (52.1)
Economizer surface, ft^2 (duty)	8,384 (10.5)	8,550 (8.3)
Geometry	evap/econ	evap/econ
Tubes/Rows	11/15	10/15
No. deep	66/14	87/10
Length, ft	9.5/11	9.5/10
Eco fins/in, ht, thick, serr	3x0.75x0.05x0.157	5x0.75x0.05x0.157
Transverse pitch, in.	4.0/4.0	4.375/4.0
Overall heat transfer coefficient	18.0/7.35	17.0/6.25

Parameters: 100,000 lb/h, 300 psig steam, 230°F feed water, 2% blowdown, natural gas fuel, 10% excess air: Boiler duty = 100.8 efficiency (HHV) = furnace back pressure = 7.0 in.

steam for deaeration is supplied from the boiler. Ignoring this could result in a smaller boiler.

Some consultants think that if a boiler is designed for 700 psig, then it can operate at any lower pressure, even 100 psig. This is not so. Due to the significant difference in specific steam volume, the velocity of steam at lower pressure will be high, about 300 to 400 ft/s in the superheater or in the steam lines and could be a serious operating problem. The steam drum internals will also not operate well at lower pressures and carryover of water into steam is likely. A possible option is to reduce the steam capacity at lower pressures or design the boiler for the varying pressures. But state this point up front in the specifications.

If superheaters are present, the steam temperature control range should be specified. Typically, 50% to 100% load range is feasible. However, some consultants not familiar with flow distribution problems at low loads, suggest a load range of 10% to 100%. This is not meaningful since it is very difficult to predict superheater performance when gas/steam flow maldistribution problems are likely.

Fuels and emissions. Fuel analysis and emissions to be met should be stated clearly. Gaseous fuels containing hydrogen have a higher combustion temperature, which increases NO, formation. Low-Btu fuels result in large amounts of flue gas to be handled by the boiler. This affects fan power consumption and efficiency due to a higher exit gas temperature.

The burner supplier must also ensure that emission levels can be met with the fuels in question and suggest appropriate excess air and FGR rates. If both natural gas and fuel oils are used, the specifications should state this. Furnace exit gas temperature is higher for natural gas compared to oil. If convective superheaters are used, the desired steam temperatures have to be attained on oil firing. Desuperheating could be done on gas firing to control it. Presence of nitrogen in fuels also affects NO_x formation. The burner supplier must be aware of this if NO_x guarantees are made.

One reason for using an economizer and not an air heater as the heat recovery system is the impact on NO, by the higher combustion temperature with hot air. Also the gas- and air-side pressure drops are higher in an air heater, which is an operating penalty.

Fan operation. A small margin should be used for flow and head while sizing fans. This is because, unlike utility boilers where multiple fans are used, a single fan is used in packaged boilers. If a large margin is used, the fan is not likely to operate well below 30% to 40% (Fig. 4) unless a variable speed drive is used. At low loads, even with fully closed damper positions, the leakage air flow could be enough to blow out the flame. These aspects must be discussed with the burner and fan supplier and with the end-user to check if low-load operation for long duration is really likely and necessary.

Surface areas can be misleading. A common problem even among experienced engineers is the comparison of surface areas of different designs. Some consultants even specify required surface areas. This is a poor practice and should be avoided. Surface area is defined as:

$$S = Q/(U \cdot T)$$

(3) where Q= duty or energy absorbed by the surface, Btu/hr U= overall heat transfer coefficient, Btu/ft^2-hr-°F AT= log-mean temperature difference, °F.

U depends on gas velocity, temperature, tube pitch and arrangement. Also, when extended surfaces are used, the variations in S could be 100% to 300%.1, 1,2 Using a large fin density decreases U while a lower fin density increases U. Therefore, unless one knows how to compute U for bare and finned tubes, comparing surface areas can be misleading and should be avoided.

Rules of thumb for surface areas should also be avoided as they can lead to improper conclusions. Table 2 shows the design of two boilers for the same duty, efficiency and gas pressure drop with different surface areas. The reason for variations in S is that the amount of energy absorbed in the furnace, convection and economizer sections are different. Also, using different fin configurations in the economizer distorts the picture.

LITERATURE CITED

Ganapathy, V, "Design and evaluate finned tube bundles," Hydrocarbon Processing, September 1996.
Ganapathy, V., "Evaluate extended surfaces carefully," Hydrocarbon Processing, October 1990.

The author

V. Ganapathy is a heat transfer specialist with ABCO Industries Inc., Abilene, Texas. He is engaged in the engineering *of heat recovery boilers for process,* incineration *and cogeneration applications and packaged water tube steam generators. He also develops software for engineering of heat recovery systems and components. He holds a B Tech degree in mechanical* engineering *from Indian Institute of Technology, Madras, India, and an MSc(eng) in boiler technology from Madras. University. Mr. Ganapathy is the author of over 175 articles on boilers, heat transfer and steam plant systems and has written five books: Applied Heat Transfer, Steam Plant Calculations Manual, Nomograms for Steam Generation and Utilization, Basic Programs for Steam Plant Engineers (book and diskette), and Waste Heat Boiler Deskbook, copies of which are available from him. He also has contributed several chapters to the Encyclopedia of Chemical Processing and Design,*