Superheaters: design and performance

Understand these factors to improve operation

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Steam superheaters are widely used in steam generators and heat-recovery steam generators (HRSGs). Their purpose is to raise steam temperature from saturation conditions to the desired final temperature, which can be as high as 1,000°F in some cases. When used in steam turbines, superheated steam decreases steam heat rate of the turbine and thus improves the turbine and overall plant power output and efficiency. Also, steam conditions at the steam turbine exit will have little or no moisture, depending on the pressure ratio; moisture in the last few stages of a steam turbine can damage the turbine blades. This article outlines some of the design considerations and performance aspects of superheaters, which should be of interest to plant engineers.

Superheaters in packaged steam generators and HRSGs-general features. Packaged steam generators generate up to 300,000 lb/h steam, while a few gas turbine HRSGs generate even more depending on the gas turbine size. Steam pressure in cogeneration and combined cycle plants typically ranges from 150 to 1,500 psig and temperature from saturation to 1,000°F Seamless alloy steel tubes are used in superheater construction. Tube sizes vary from 1.25 to 2.5 in. Commonly used materials are shown in Table 1.

Allowable stress values depend on actual tube wall temperatures. Tube thickness is determined based on this using formulae discussed in the ASME Code, Sections 1 and 8. Different designs are available for superheaters depending on gas/steam parameters and space availability. The inverted loop design (Fig. 1) is widely used in packaged boilers, while the vertical finned tube design is common in HRSGs. The horizontal tube design with vertical headers is used in both. Bare tubes are generally used in packaged steam generators, where gas temperatures are high (typically 1,500-2,200°F) and tube wall temperature is a concern.

However, in gas turbine HRSGs, finned superheaters are used. Gas inlet temperature is generally low, on the order of 900-1,400'F, which requires a large surface area. Use of finned tubes makes their design compact. Superheaters can be of convective or radiant design or a combination of these in packaged boilers. Final steam temperature may or may not be controlled. In unfired and supplementary fired HRSGs, the superheaters are or convective design only.

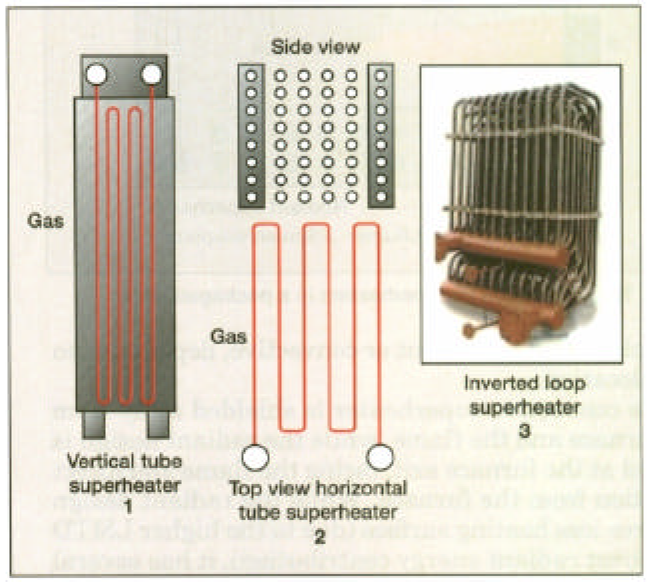

Fig. 1. The inverted loop design is typically used in packaged boilers, while the vertical finned design is common in HSRGs.

Table 1. Superheater materials

Material	Composition	Allowable temperature, °F
SA-192	Carbon steel	950
SA-213 T11	1.25 Cr-0.5 Mo-Si	1,050
SA-213 T22	2.25 Cr-1 Mo	1,125
SA-213 T91	9 Cr-1 Mo-V	1,200
SA-213 TP 304H	18 Cr-8 Ni	1,400
SA-213 TP 347H	18 Cr-10 Ni-Cb	1,400
SB-407-800H	Ni-Cr-Fe	1,500
SA-213 TO 310H	25 Cr 20 Ni	1,500

Steam velocity inside superheater tubes ranges from 50 to 140 fps depending on steam pressure, allowable pressure drop and turndown in load. Typical pressure drop in industrial applications ranges from 10 to 70 psi depending on size, pressure and load turndown conditions. In utility boilers, where multiple stage superheaters are used, pressure drop will be much higher, say 150-200 psi. If the superheater has to operate over **a** wide load range, a higher steam pressure drop at full load ensures reasonable flow at lower loads.

Convective and radiant superheaters in packaged boilers. Fig. 2 shows typical location of superheaters in a packaged boiler. Superheaters are basixally

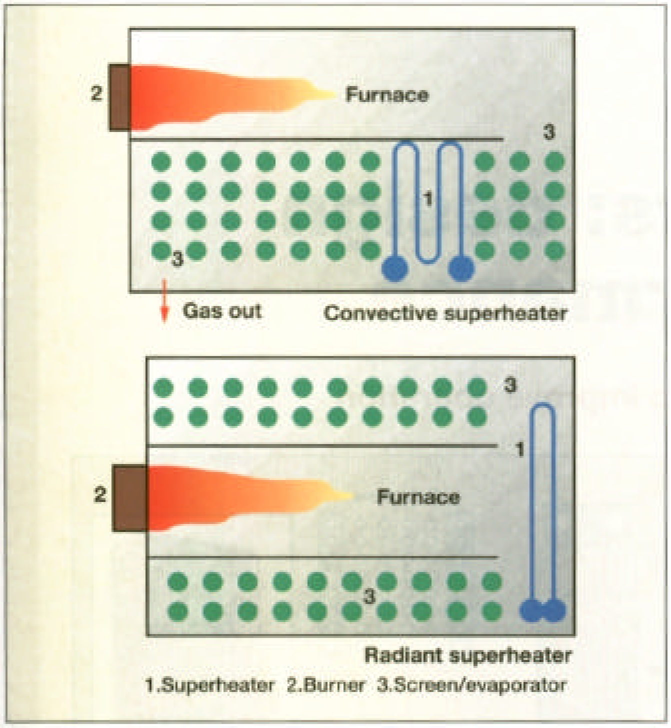

Fig. 2. Typical location of superheaters in a packaged boiler.

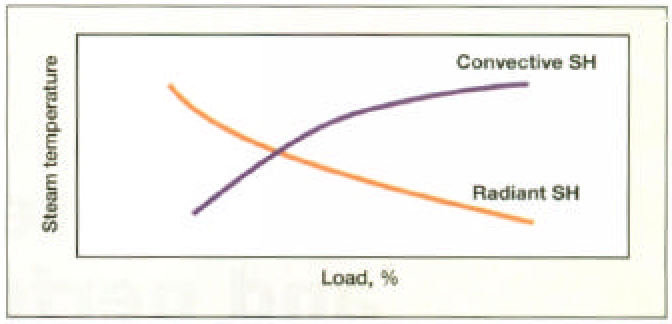

Fig. 3. The radiant design absorbs more energy at lower loads, while the convective design absorbs more at higher loads.

cally classified as radiant or convective, depending on their location.

The convective superheater is shielded away from the furnace and the flame, while the radiant design is located at the furnace exit, facing the flame and direct radiation from the furnace. While the radiant design requires less heating surface (due to the higher LMTD and direct radiant energy contribution), it has several drawbacks compared to the convective design.

• A convective superheater is located behind a screen section, which helps to cool gases from the furnace and also ensures that a uniform gas mixture enters the superheater. This permits the designer to predict superheater performance with much higher reliability and accuracy. The furnace is a difficult section to evaluate due to the complexity of the combustion process. Adding to the difficulty is use of varying excess air and flue gas recirculation rates (used for NO_X control) for different fuels at different loads, which in turn affects flame temperature and its temperature distribution along the flame.

Only models based on experience of similar units could be considered reasonable since no simple mathematical formulae can accurately predict the furnace energy balance. Hence actual furnace exit gas temperature can easily be off from predicted values by 50-150°F, which affects the radiant superheater performance significantly. If actual gas temperature is higher than predicted, we have tube overheating problems and if it is lower, we may not obtain the desired steam temperature.

• The radiant superheater is located in a region where the flue gases make a turn and, hence, the gas flow distribution pattern over the tubes is difficult to predict at various loads.

• The radiant superheater receives direct radiation from the furnace, which in turn depends on exit gas temperature. Again, if we do not predict exit gas temperature accurately, radiant heat flux will be off from estimates, which affects steam and tube wall temperatures. Margin of error on gas temperature estimation is much higher at furnace exit compared to the convective superheater inlet, which is located behind a screen section.

• Steam temperature characteristics are different between convective and radiant designs. The radiant design absorbs more energy at lower loads, while the convective design absorbs more at higher loads (Fig 3). This is due to the increase in convective heat transfer rates. In large field-erected industrial or utility boilers, the superheater is in multistages. The combination of radiant and convective designs helps ensure a uniform steam temperature over a wide load range.

However, when only a single-stage superheater is used-as in packaged boilers-the radiant design is subject to higher steam temperatures at low loads, when flow distribution on both gas and steam sides is poor. If at 100% load the steam pressure drop is, say, 30 psi, at 25% load it will be hardly 2 psi, which cannot ensure good steam flow distribution in various elements. The same is true of flow distribution on the gas side at low loads due to the low gas velocity. Thus, this fact, along with lack of steam temperature control methods, can result in overheating and possible tube failure.

Creep analysis using Larson-Miller parameter methods may be done to estimate life at various tube wall temperatures and remaining superheater life, based on number of hours of operation at each load. The convective design, on the other hand, is located in a much cooler gas temperature zone-1,600-1,800°F, compared to 2,200-2,400°F for the radiant design. Hence, it runs much cooler and tube wall temperatures are also more accurately predictable. At lower loads, gas temperature to the superheater will be lower as well as the heat transfer rate and tube wall temperature. Thus, convective superheater life is longer than the radiant design.

• Steam turbines usually require a constant steam temperature. Lower steam temperatures affect the heat rate; however, this occurs at a lower load with convective designs and, hence, loss in output is not significant. Oversizing convective superheaters may also be done to ensure that desired steam temperature is achieved over a wider load range if necessary-say from about 30% to 100%.

• With convective designs, it is possible to have twostage designs with interstage attemperation. With radiant designs in packaged boilers, single stages are generally used, which causes concerns with steam temperature fluctuations and tube overheating.

Steam temperature control methods in superheaters. Generally, steam temperature is maintained constant from about 60% to 100% load. Interstage attemperation or spray water injection (Fig. 4) is done to achieve the desired final steam temperature. Water injected should be demineralized since solids contained in feed water can get carried into the superheater and turbine and selective deposition can occur.

Salt deposits in the superheater can result in tube overheating. Turbine blade deposition is a big concern with turbine maintenance engineers since it reduces power output, restricts flow passages, causes corrosion and can damage the blades. Hence, high steam purity on the order of 20-50 ppb is generally desired in high steam temperature applications. Good steam drum internals using a combination of baffles and Chevron separators can achieve the desired steam purity.

In case demineralized water is not available for spray, some of the steam may be condensed using a heat exchanger as shown in Fig. 4, and the condensate is sprayed into the desuperheater. Steam flow through the exchanger and superheater should be balanced in the parallel paths either by using flow restrictions, control valves in each parallel path or by raising the exchanger level to provide additional head for control. Feed water from the economizer cools and condenses steam used for desuperheating (Fig. 4a). In Fig. 4b, the feed water is directly injected into the steam between the stages.

Desuperheating beyond the superheater is not recommended since moisture can be carried to the steam turbine along with the steam if downstream mixing is not good. Also, this method permits steam temperature in the superheater to increase beyond the desired final steam temperature and, hence, the premium on materials used for superheater construction will be high.

There are several other methods used for steam temperature control such as varying excess air, tilting burners, recirculating flue gases, etc., but in packaged boilers and HRSGs, interstage attemperation is generally used.

Superheaters in HRSGs. The basic difference in superheater design used in steam generators and HRSGs is that in HRSGs, as mentioned earlier, finned tubes may be used to make the design compact. The large duty and large gas-to-steam flow ratio coupled with the low LMTD necessitates this. However, while selecting finned tubes, a low fin density should be used considering the low steam side heat transfer coefficient inside the tubes. The heat transfer coefficient due to superheated steam flow is small, on the order of 150-300 Btu/ft^{2}h°F, depending on steam flow, pressure, temperature and tube size.

A large fin area would only increase heat flux inside the tubes, tube wall temperature and possibly gas pressure drop as discussed in an earlier article.' Note that the gas side heat transfer coefficient is lower with higher fin density or surface area. Hence, it is misleading to evaluate finned superheater designs based on surface areas.' In large gas turbines, steam after expanding from the steam turbine is again reheated in the HRSG to generate additional power. Design considerations

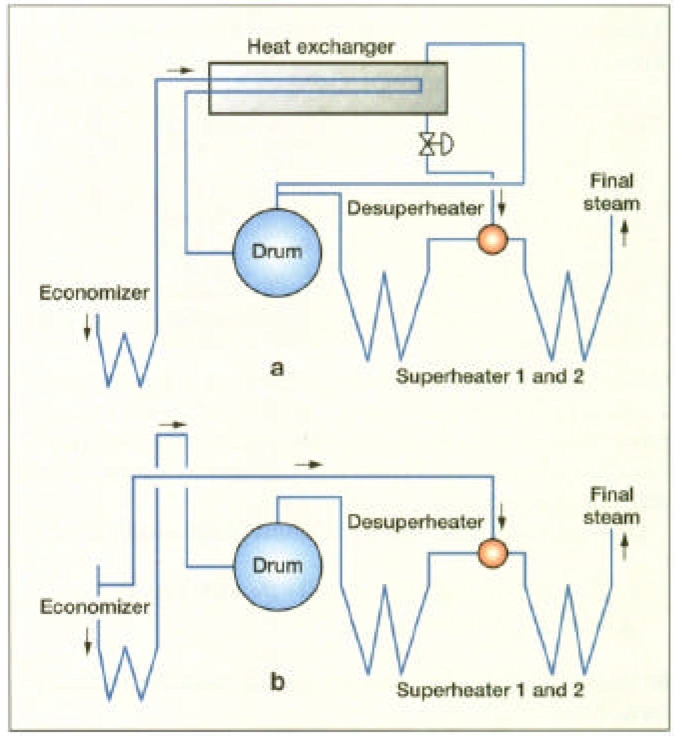

Fig. 4. Interstage attemperation or spray water injection is done to achieve the desired final steam temperature.

of the low-pressure superheater, also known as a reheater, are similar to those discussed previously for superheaters.

Sizing procedure. There are two basic types of calculations with any heat transfer surface. One is the design calculation, in which the objective is to arrive at the surface area, tube layout, steam and gas-side pressure drops, and preliminary material selection. The off-design performance calculation tells us how the same surface will perform at other gas or steam conditions. There is only one design calculation, but performance could be checked at different loads or off-design conditions. A computer program is generally used for these purposes since the calculations are quite involved and iterative. Only after these two types of calculations are completed can we say that the engineering process is over.

Here's the energy balance equation for a superheater. Total energy absorbed by the superheater is:

$$Q_s = W_s(h_{s2} - h_{s1}) = Q_c + Q_n + Q_r \quad (1)$$

Relating this to the gas temperature drop:

$$(Q_s - Q_r) = Q_n + Q_c = US\Delta T = W_g(h_{g1} - h_{g2}) \quad (2)$$

External radiation, Qr, is generally absent in convective type designs. Also, if a screen section is used, Qr gets absorbed in about four to six rows of screen tubes depending on tube spacing.

The fraction of energy, F, absorbed in each row is given by:

$$F = 3.14(d / 2St) - (d / St)[\sin^{-1}(d / St) + \sqrt{(St / d)^2 - 1} - (St / d)] \quad (3)$$

If S/d = 4, then F = 0.361. The first row absorbs

Table 2. C factors for saturated and superheated steam

Temp–°F/pres–psia	100	500	1,000	2,000
Sat	0.244	0.417	0.490	0.90
400	0.271	–		
500	0.273	0.360		
600	0.281	0.322	0.413	
700	0.291	0.316	0.358	0.520
800	0.305	0.30	0.345	0.420
900	0.317	0.327	0.347	0.394
1,000	0.325	0.340	0.353	0.386

0.361 of the external radiation. The second row absorbs: (1- 0.361)0.361= 0.23 and so on. If (S / d) were smaller, fewer rows would be required to absorb the direct radiation. Qr is estimated from gas emissivity and furnace exit gas temperature.

Overall heat transfer coefficient is

obtained from: $1/U = (At/Ai)/hi + ffi (At/Ai) + ffo + (At/Aw) d/24Km \ln(d/di) + 1/\eta ho$ (4)

for finned tubes. For bare tubes, the same equation is used, however,

$At/Ai = d/di$ and fin effectiveness η=1 =1

Fouling factor, *ffo*, is typically 0.001 ft^2hF/Btu in clean gas applications, while *ffi* ranges from 0.0005 to 0.001.

The gas side heat transfer coefficient, $ho = hn + hc$.

The procedure for determining these are well documented.[2,3]

Gas side heat transfer coefficient, *hc*, for finned and bare tubes may be obtained from published charts or equations[1,2,3] as also the procedure for determining *hn*. A simplified approach to estimating *hi* is:

$$hi = 2.33 w^{0.8} c / di^{1.8} \quad (5)$$

where factor C is given in Table 2.

Once the duty Qs are known and resultant gas and steam temperatures at the superheater inlet and exit, then the LMTD may be estimated. Knowing the various gas and steam side heat transfer coefficients, fouling factors and tubes sizes, overall heat transfer coefficient, U, is obtained. Surface area, S, is determined as shown previously from Eq. 2. Then the tubes are laid out and the gas/steam side pressure drops are evaluated. Several factors such as tube size, number of streams carrying the steam flow, tube spacing, gas mass velocity, etc., are selected based on experience.

Gas side pressure drop may be found from equations discussed in citations 2 and 3.

Tube side pressure drop is give by:

$$DP = 3.36 \times 10^{-6} f w^2 Le\, v / di^5 \quad (6)$$

If superheaters are of such a design (say the inverted loop design in Fig. 2) that tube length in various elements is different, a flow balance calculation has to be performed to determine steam flow in each element. The tube with the lowest flow is likely to have the highest tube wall temperature.

If there are, say, four elements, the following equations help evaluate flow in each.

Pressure drop across the headers is the same across each element and from the previous equation for pressure drop:

$w1^2R1 = w2^2R2 = w3^2R3 = w4^2R4$ = M(constant) R = resistance of each element = $fLe/di5$

Thus, we first determine resistance of each element with different tube lengths, R_1, R_2, R_3, R_4. Then we can solve for flow through each element:

$w_1 + w_2 + w_3 + w_4$ = known as the total steam flow is known

or

$\sqrt{M/R_1} + \sqrt{M/R_2} + \sqrt{M/R_3} + \sqrt{M/R_4}$ = total flow

or

M can be obtained from the above since all resistances are known.

Then w_1, w_2, w_3 and w_4 can be obtained.

The tube wall temperature calculations are done. In simple terms, heat flux is first estimated:

$Q = U(tg - ts)$] Temperature drop across the steam film is: *heat flux / hi.*

Temperature drop across the fouling layer inside is: *heat flux x* ffi.

Then drop across the tube wall is determined by multiplying tube wall resistance, (d/24K)ln(d/di), by heat flux. All these are added to the steam temperature to arrive at the tube wall temperature. This calculation may be done at different tube locations using local *tg*, *ts*, *hi* and U values.

Off-design performance. To arrive at the off-design performance, one can resort to the NTU method.[2,3] In this calculation, surface area is known and gas flow, steam flow and their inlet temperatures are known. It is desired to predict the duty and exit gas and steam temperatures. This method is discussed in various textbooks [2,3] and will not be explained here. Based on actual gas/steam flow conditions, an estimation of U is done. Using the NTU method, can be found. Tube wall temperatures at various locations are again evaluated. Based on both design and off-design conditions, a final material selection is made. One may revise the design if off-design performance is not up to expectations.

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NOMENCLATURE

Af, Ai, At = area of fins, inside tube area and total tube surface area per unit length, ft^2/ft
C = constant for determining tube side coefficient
d, di = tube outer and inner diameters, in.
F = fraction of direct radiation absorbed
f = friction factor inside tubes
ffo, ffi = fouling factors outside and inside tubes, fth°F/Btu
hg_1, hg_2 = gas enthalpy at inlet and exit of superheater, Btu/lb
hc, hn, ho = convective, nonluminous and outside heat transfer coefficients, Btu/ft^2hoF
hsl, hs_2 = steam enthalpy at superheater inlet and exit, Btu/lb
K= tube thermal conductivity, Btu/fth°F
Le = tube effective length, ft
M = a constant
Qc, Qn, Qr, Qs = energy due to convection, nonluminous heat transfer, direct radiation and that absorbed by steam, Btu/h
S = surface area, ft^2

Tg, is = local gas and steam temperatures, °F

Wg, WS = gas and steam flow, lb/h
w = steam flow per tube, lb/h

ΔT = log-mean temperature difference, °F

ΔP = pressure drop inside tubes, p si
v = steam specific volume, ft^3/lb
η = fin effectiveness

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