Simplify heat recovery steam generator evaluation

Insights, equations and examples illustrate a simpler method for predicting heat recovery steam generator

V Ganapathy, ABCO Industries, Abilene, Texas

HEAT RECOVERY STEAM generators (HRSGs) are widely used in process and power plants, refineries and in several cogeneration/combined cycle systems (Fig. 1). They are usually designed for a set of gas and steam conditions but often operate under different parameters due to plant constraints, steam demand, different ambient conditions (which affect the gas flow and exhaust gas temperature in a gas turbine plant), etc. As a result, the gas and steam temperature profiles in the HRSG, steam production and the steam temperature differ from the design conditions, affecting the entire plant performance and economics. Also, consultants and process engineers who are involved in evaluating the performance of the steam system as a whole, often would like to simulate the performance of an HRSG under different gas flows, inlet gas temperature and analysis, steam pressure and feed water temperature to optimize the entire steam system and select proper auxiliaries such as steam turbines, condensers, deaerators, etc.

HRSG suppliers can provide this information, but if a simpler approach is made available to every engineer involved with HRSGs, it would be a powerful tool for plant engineers, consultants and cogeneration system engineers, who would only like to simulate HRSG performance. Usually they do not want to get involved in the thermal and mechanical design of HRSGs, which is best done by the supplier of the HRSG. This article describes a simplified approach to predicting the performance of HRSGs. Software is available that can save valuable time and drudgery, as performance evaluation of HRSGs is a tedious process.[1,2]

Advantages of this approach. Configuration of the HRSG or its geometrical details such as the width, height, tube size, pitch or fin density, number of rows, etc., need not be known. Based on known or assumed pinch and approach points (see Fig. 2 for definitions), a "design" is simulated, the gas/steam temperature profiles are determined and the steam flow is obtained. Then, using the procedure described below, the "performance" at any other gas or steam parameters can be obtained using quick converging iterative logic. The procedure may be used for both unfired and fired HRSGs.

Heat transfer coefficients or surface area need not be computed per se. The product of U (overall heat transfer coefficient) and S (surface area) is computed for each surface based on the duty and log-mean temperature difference, corrected for gas flow, analysis and temperature, and used in the performance calculations. Since US can be obtained for any HRSG surface for a particular set of design or operating conditions, this information is in everybody's domain.

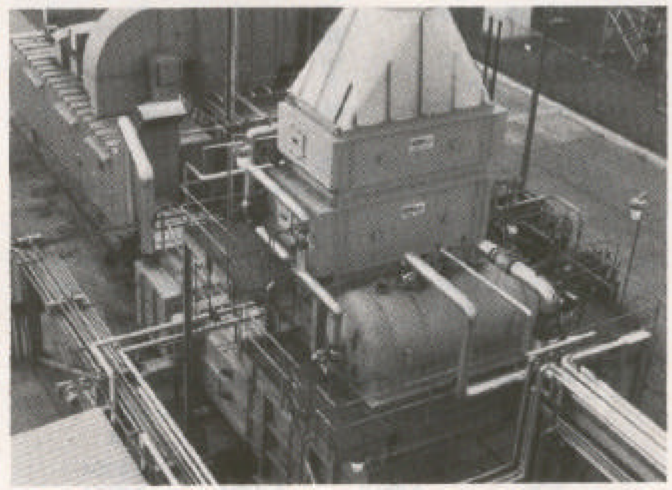
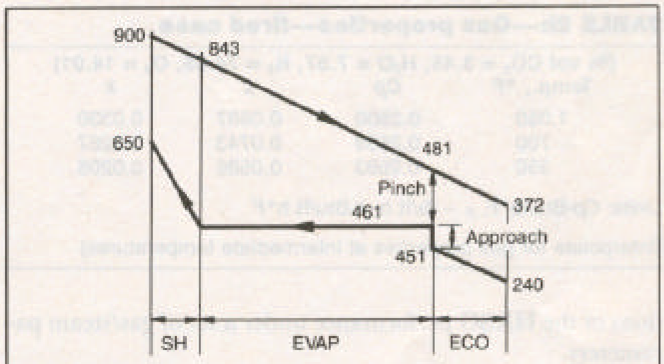


Fig. 1—HRSG for gas turbine exhaust (courtesy ABCO Industries, Abilene).



Results—design case—unfired
Amb. temp. – °F = 60, Rel. hum. – % = 0, Heat loss – % = 1,
Gas temp. to HRSG – °F = 900
Gas flow-pph = 150,000, % vol. CO_2 = 3, H_2O = 7, N_2 = 75, O_2 = 15, SO_2 = 0

	Gas temp. in-out – °F		Wat./stm. in-out – °F		Duty MM b/h	Press. psig	Flow pph	Pstm. %	Pinch °F	Apprch. °F
SH	900	843	461	650	2.35	450	18,610	100.0		
EVAP	843	481	451	461	14.38	467	18,610	100.0	20	10
ECO	481	372	240	451	4.22	467	18,982	0.0		

Fig. 2—Design case 1 results.

Importance of pinch and approach points. To obtain the performance of an HRSG with a different set of gas/steam conditions, one should have either the design condi

TABLE 1-Data for design and performance calculations

		Desig	Perf	Perf	Perf
1	Case no.	*1*	2	3	4
2	Gas flow, pph	150,00	165,000	165,00	165,000
3.	Exhaust temp., °F	900	840	840	840
4.	% vol $C0_2$	3	3	3	3
	H_2O	7	7	7	7
	N_2	75	75	75	75
	0_2	15	15	15	15
5.	Steam press., psig	450	450	450	300
6.	Steam temp., °F	650	?	?	650
7.	Feed water temp., °F	240	240	240	240
8.	Blow down, %	2	2	2	2
9.	Process steam, pph	-	-	-	2,500
10.	Heat loss + margin,	1	1	1	1
11.	SH press. drop, psi	7	?	?	?
12.	Pinch point, °F	20	?	?	?
13.	Approach point, °F	10	?	?	?
14.	Steam flow, pph	?	?	26,000	26,000
15.	Ambient temp., °F	80	50	50	50

Natural gas used: % vol C1= 96, C2 =2,C3=2

Note that steam is required at a controlled temperature of 650°F in case 4. In cases 2 and 3 it is uncontrolled. Also, in case 4, 2,500 pph of saturated steam is taken off the drum and the balance of 26,000 pph is to be superheated to 650°F. The steam exit pressure is 300 psig in case 4. It will be seen later that cases 3 and 4 are fired and cases 1 and 2 are unfired.

TABLE 2a-Gas properties-unfired case

(% vol CO_2 = 3, H_2O = 7,N_2=75,O_2=15

Temp., °F	Cp	µ	*k*
900	0.2736	0.083	0.0304
650	0.2658	0.0724	0.0261
400	0.2584	0.0612	0.0218

TABLE 2b-Gas properties-fired case

(% vol CO2=3.45,H2O=7.87,N2=74.65,O2=14.01

Temp.,	Cp	µ	*k*
1,050	0.2800	0.0887	0.0330
700	0.2689	0.0743	0.0267
350	0.2583	0.0586	0.0208

Units: Cp-Btu/Ib°F, µ - Ib/ft h, k-Btu/fth°F

(Interpolate for gas properties at intermediate temperatures)

tions or the HRSG performance under a set of gas/steam parameters.

HRSGs for gas turbine exhaust are usually designed in unfired conditions and the performance evaluated at other unfired or fired conditions. The reason for this is that two of the important variables which affect the gas and steam temperature profiles, namely the pinch and approach points, cannot be arbitrarily selected in the fired mode. The following problems can result if this is done:

1. A realistic temperature profile may not result. That is, a temperature cross can occur in the economizer, with the gas exit temperature being lower than the incoming feed water temperature.

2. An unrealistic boiler size can result due to very small pinch or approach points especially if that is the basis for selecting the temperature profile in the fired mode. Pinch and approach points up to 10°F can be achieved with practical boiler configurations in unfired modes. Unless one has a great deal of experience in designing HRSGs, pinch and approach points should be selected in unfired modes and the HRSG performance evaluated in fired modes.

3. Steaming in the economizer is likely if one selects an HRSG temperature profile in the fired mode and operates the boiler in unfired mode. The performance of the HRSG has to be checked in the unfired coldest ambient conditions, when the gas flow is the highest and the gas inlet temperature to the HRSG is the lowest to make sure the economizer does not steam. If it does, the original temperature profile selected has to be revised, resulting in a waste of time and effort.

4. The amount of spray water used for steam temperature control cannot be simulated a priori in the fired mode. This is particularly important in HRSGs where the steam temperature has to be controlled over a wide load. The steam temperature will fall off as the gas inlet temperature reduces and as a result, if the steam temperature is chosen (say 700°F) in the fired mode, it will be lower in the unfired mode. Several performance checks would have to be made to ensure that the steam temperature is being achieved over the desired load range. On the other hand, if the steam temperature is selected in the unfired mode, it will increase with an increase in gas inlet temperature and hence, can be controlled with spray water or other means.

Hence, it is prudent to arrive at a design temperature profile in gas turbine HRSGs based on cold ambient, unfired conditions and then check the HRSG performance in other unfired or fired conditions, even if the HRSG operates in the fired mode most of the time.

Pinch and approach points lie in the range of 10 to 40°F in unfired conditions for clean applications such as gas turbine exhaust, where extended surfaces can be used. Higher numbers may be selected if the steam generation can be lower. In the case of HRSGs for applications such as incineration exhaust or chemical plants where the gas inlet temperature could be in the range of 1,400 to 1,800°F, and in HRSGs where extended surfaces cannot be used due to dirty gas, a higher pinch point in the range of 100 to 250°F should be used.

Design temperature profile and calculations. A superheater and economizer are assumed to be in counterflow arrangement, which is the widely used configuration.

Example 1. A gas turbine HRSG is to be designed for the parameters shown in Table 1. Determine the gas/steam profiles and the steam flow. Let the gas pressure drop = 6.0 in. WC.

The drum pressure = 450 + 7 = 457 psig. The saturation temperature is 460°E Gas temperature leaving the evaporator = 460 + 20 = 480°E Compute the gas properties for the given analysis.' The data are shown in Table 2. Using an instantaneous specific heat of 0.267 for the range 900 to 480°F, and a heat loss factor of 0.99, the duty in the superheater and evaporator is: $Q_1 + Q_2$ = 150,000 (0.267) (0.99) (900 - 480) = 16.65 x 10^6 Btu/h = Wsd[(1,330.8 - 431.2) + 0.02(442.3 - 431.2)] = 899.8 Ws_d. Where 1,330.8 = enthalpy of superheated steam at 450 psig, 650°F, 442.3 = enthalpy of saturated water at drum pressure, 431.2 = enthalpy of water entering the evaporator at 450°E 0.02 is the blow down factor.

From the above, Wsd = 18,510 pph. Superheater duty, Q_1 = 18,510(1,330.8 - 1,204.4) = 2.34 x 10^6 Btu/h, where 1,204.4 is the enthalpy of 'saturated steam. Gas temperature drop in the superheater = 2.34 x 10^6/(150,000) (0.273) (0.99) = 58°E Hence, gas temperature to evaporator = 900 - 58 = 842°F. Q_2 = Evaporator duty = 16.65 - 2.34 = 14.31 x 10^6 Btu/h. Economizer duty = 18,510 (1.02) (431.2 - 209.6) = 4.19 x 10^6 Btu/h, where 209.6 is the enthalpy of feed water at 240°F.

Gas temperature drop in the economizer = 4.19 x 10^6/ (150,000) (0.99) (0.26) = 109°F. The gas specific heat at the average gas temperature in the economizer, obtained from Table 2 by interpolation is 0.26. Hence, the exit gas temperature = 480 - 109 = 371°F. The temperature profile is shown in Fig. 2. Using a similar approach, the temperature profiles for any other pinch or approach points can be obtained.

To proceed with the performance calculations for case 2 shown in Table 1, a few parameters should first be computed, as discussed in Appendix 2. These parameters help relate the heat transfer coefficients in the "design" mode to those in "performance."

For the superheater: $K_1 = Q_1/(\Delta T_1)\ (Wg^{0.65})\ (Fg)$ where ΔT_1 = log-mean temperature difference = [(842 - 460) - (900 - 650)]/ln[(842 - 460)/(900 - 650)] = 311°F, Fg = $Cp^{033} k^{0.67}/\mu^{0.32}$ = 0.135, using a Cp = 0.273, k = 0.029, and μ = 0.0826. Hence K_1 = 2,340,000/150,$000^{0.65}$/311/ 0.135 = 24.10. Similarly for the evaporator K_2 = 387.6 and K_3 = 218.4 for the economizer.

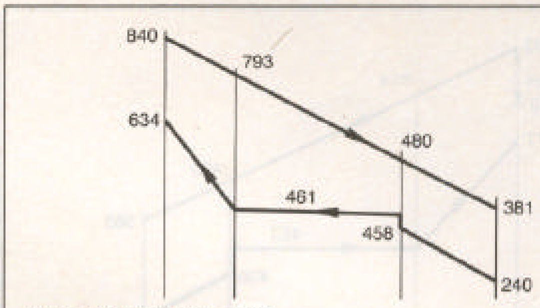
K_1, K_2 and K_3 will be used to compute $(US)_p$ the product of U and S in the performance modes as discussed in Appendix 2.

Performance calcbulations. Let us see how the unit performs when the conditions are as shown in case 2, Table 1. The gas flow is 165,000 pph at 840°E The gas analysis, feed water temperature and steam pressure remain the same as earlier. The performance of the HRSG is arrived at through an iterative process described in Appendix 1, using the equations discussed in Appendix 2.

Trial 1. As a first approximation, assume that the steam flow is proportional to the gas flow and temperature drop. Ws = 18,510(165,000/150,000) (840 - 371)/(900 - 371) = 18,050 pph.

Superheater performance. Let ts_2, the steam exit temperature = 640°F. Then, from steam tables, the enthalpy = 1,325 Btu/lb. The assumed duty = 18,050 (1,325 - 1,204.4) = 2.177 x 10^6 Btu/h. Gas temperature drop = 2,177,000/(165,000) (0.99) (0.271) = 49°F. Hence, gas temperature leaving the superheater = 840 - 49 = 791°F. Compute the transferred duty, Q1t, using Eq. 2 in Appendix 2. Fg = 0.135, Wg = 165,000, *K,* = 24.1, Ws_d = 18,510, Ws = 18,050. Hence $(US)_p$ = 165,$000^{0.65}$(0.135) (24.1) (18,050/18,510)$^{o.5}$ = 7,974. ΔT = log-mean temperaturedifference = [(840 - 640) - (791 - 460)]/ln[(840 - 640)/(791 - 460)] = 260°F. Hence, Q_{1t} = 7,974(260) = 2,074,000 Btu/h. This is close to the assumed value. If it were not, we would have to assume another steam temperature and repeat the steps. Let us continue.**Evaporator performance**. Compute Fg at the average gas temperature in the evaporator. Fg = 0.129, K_2 = 387.6. Then, (US)p = 165,$000^{0.65}$(0.129) (387.6) = 123,123. Using Eq.8,[(791-460)/(Tg_3-460)]= $_e$(123,123/165,000/0.99/0.266) = 17,00. Hence Tg_3 = 480°F; Q_2 = 165,000(0.99) (0.266) (791 - 480) = 13.522 x 10^6 Btu/h. Note that the gas properties have to be interpolated for the values at the average gas temperature in the section.

Economizer performance. Let the water temperature leaving the economizer be 450°F. hw_2 = 431.2 from steam tables. Assumed duty Q_{3a} = 1.02(18,050) (431.2 - 209.6) = 4.08 x 10^6 Btu/h. The gas temperature drop = 4,080,000/ 165,000/0.99/0.26 = 96°F, exit gas temperature = 480 - 96 = 384°F. Fg = 0.130, K_3 = 218.4. Hence $(US)_p$ = 218.4



Results—performance case—unfired
Amb. temp. – °F = 50, Rel. hum. – % = 60, Heat loss – % = 1,
Gas temp. to HRSG – °F = 840
Gas flow-pph = 165,000, % vol. CO_2 = 3, H_2O = 7, N_2 = 75, O_2 = 15, SO_2 = 0

	Gas temp. in-out – °F		Wat./stm. in-out – °F		Duty MM b/h	Press. psig	Flow pph	Pstm. %	Pinch °F	Apprch. °F
SH	840	793	461	634	2.08	450	17,822	100.0		
EVAP	793	480	458	461	13.61	456	17,822	100.0	19	3
ECO	480	381	240	458	4.19	466	18,179	0.0		

Fig. 3—Performance case 2 results.

($165,000^{065}$) (0.120) = 64,535. Transferred duty = Q_{3t} = 64,535(72.7) = 4.69 x 10^6 Btu/h, where 72.7 is the log mean temperature difference.

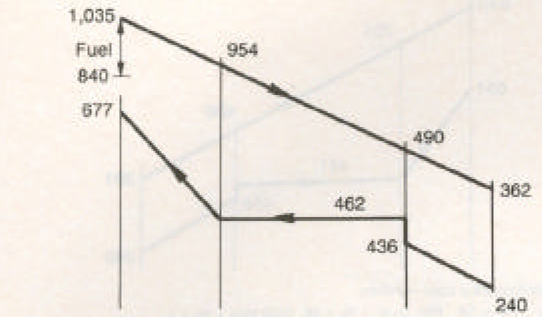
Since the transferred duty is more than the assumed, let us repeat the calculations with say t_{w2} = 457° F.Q_{3a} = 18,050((1.02) (439 - 209.6) = 423,000 Btu/h. The exit gas temperature = 381. ΔT= 65. Then, Q_{3t}= 64,535(65) = 4,190,00(Btu/h. Since this is closer to Q_{3t} let us continue. The total transferred duty = Q_{1t}+ Q_{2t}+ Q_{3t} = 2.07 + 13.52 + 4.19= 19.78 MMBtu/h. The corrected steam flow, Ws = 19.78x 10^6/[1,325 - 209.6 + 0.02(442 - 209.6)] = 17,660 pph. per Eq. 14, Appendix 2. Since this is not close to the assumec value of 18,050 pph, another trial is warranted. Try Ws = 17,770 pph.

Trial 2. Let the revised steam flow = 17,700 pph. Follow a similar procedure as before.

Superheater performance. Let ts_2 = 640° F. Q_{1a} = 17,700((1,325 - 1,204.4) = 2.134 MMBtu/h. Gas temperature drop = 2,134,000/(165,000) (0.99) (0.271) = 48°F. T_{g2} = 840 - 48 = 792^0 F. ΔT = 260°F. Fg = 0.135. *K,* = 24.1. Then, $(US)_p$ = 165,000^{o65}(0.135) (24.1)(17,700/18,510)$^{0.15}$ =7,957. Q_{1t} = 7,957(260) = 2.07 MMBtu/h. Since Q_1, is less than Q_{1a}, try a lower steam temperature, say 635° F. Then Q_{1a} = 17,700(1,322 - 1,204.4) = 2.081 MMBtu/h. Gas temperature drop = 47°F. T_{g2} = 840 - 47 = 793°F. ΔT = 264°F. Hence, Q_{1t} = 7,957(264) = 2.1 MMBtu/h. This is close enough. Continue.

Evaporator performance. Solve for Tg_3 as before. [(793 - 460)/(Tg_3 - 460)] = 17.00; hence T_{g3} = 480°F. Q_2 = 165,000 (0.99) (0.266) (793 - 480) = 13.6 MMBtu/h. (The factor 17 computed from Trial 1 is unchanged.)

Economizer performance. Let t_{w2} = 455; hw_2 = 436.8; Q_{3a} = 17,700(1.02) (436.8 - 209.6) = 4.1 MMBtu/h. Gas temperature drop = 96°F. Tg_4 = 480 - 96 = 384°F. ΔT = 68°F. Using the same $(US)_p$ as before, Q_{3t} = 64,535(68) = 4.36 MMBtu/h. Since the variation between Q_{3a} and Q_{3t} is large, try t_{w2}=458°F. Then, Q_{3a} = 4.14 MMBtu/h. Tg_4 = 383°F. ΔT = 64.6°F. Hence, Q3a = 64.6(64,535) = 4.16 MMBtu/h. This is quite close. The total transferred duty = 2.1 + 13.6 + 4.16 = 19.86 MMBtu/h. The corrected steam flow, Ws = 19.86x10^6/[(1,322 - 209.6) + 0.02(442 - 209.6)] = 17,770 pph.



Results—performance case—fired

Amb. temp. – °F = 50, Rel. hum. – % = 60, Heat loss – % = 1,

Gas temp. to HRSG – °F = 840

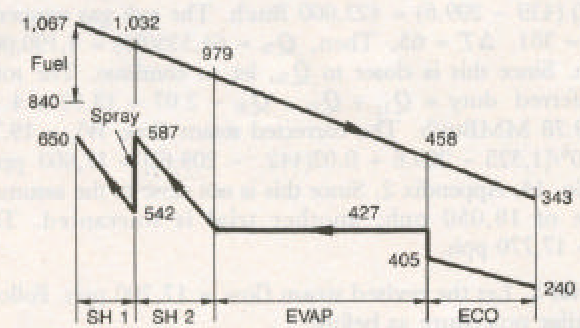
Gas flow-pph = 165,000, % vol. CO_2 = 3, H_2O = 7, N_2 = 75, O_2 = 15, SO_2 = 0

	Gas temp. in-out – °F		Wat./stm. in-out – °F		Duty MM b/h	Press. psig	Flow pph	Pstm. %	Pinch °F	Apprch. °F
BURN	840	1,035	0	0	9.29	0	434	0		
SH	1,035	954	462	677	3.69	450	25,963	100.0		
EVAP	954	490	436	462	20.52	464	25,963	100.0	27	26
ECO	490	362	240	436	5.45	474	26,482	0.0		

Gas flow after HRSG = 165,434, % vol. CO_2 = 3.45, H_2O = 7.87, N_2 = 74.65, O_2 = 14.01, SO_2 = 0.00

Fuel – gas: analysis – % volume, 1 methane = 96, 2 ethane = 2, 3 propane = 2, lhv-Btu/cu ft = 957, lhv-Btu/lb = 21,398, aug. air-pph = 0

Fig. 4—Performance case 3 results.



Results—performance case—fired

Amb. temp. – °F = 50, Rel. hum. – % = 60, Heat loss – % = 1,

Gas temp. to HRSG – °F = 840

Gas flow-pph = 165,000, % vol. CO_2 = 3, H_2O = 7, N_2 = 75, O_2 = 15, SO_2 = 0

	Gas temp. in-out – °F		Wat./stm. in-out – °F		Duty MM b/h	Press. psig	Flow pph	Pstm. %	Pinch °F	Apprch. °F
BURN	840	1,067	0	0	10.82	0	506	0		
SH	1,067	979	427	650	4.05	300	25,968	100.0		
DESH	1,032	1,032	587	542	0.00	309	608	0		
EVAP	979	458	405	427	23.03	318	28,468	100.0	31	22
ECO	458	343	240	405	4.90	328	28,610	0.0		

Gas flow after HRSG = 165,506, % vol. CO_2 = 3.52, H_2O = 8.01, N_2 = 74.60, O_2 = 13.84, SO_2 = 0.00

Fuel – gas: analysis – % volume, 1 methane = 96, 2 ethane = 2, 3 propane = 2, lhv-Btu/cu ft = 957, lhv-Btu/lb = 21,398, aug. air-pph = 0

Fig. 5—Performance case 4 results.

Since this is close to the assumed value of 17,700, let us stop here. The final temperature profile is shown in Fig. 3. The gas pressure drop, using Eq. 15, Appendix 2 = $6(165{,}000/\ 150{,}000)^2[0.5(840 + 383) + 460]/[0.5(900 + 371) + 460)] = 7.1$ in. WC.

Performance check-fired case. Let us check the performance for case 3 shown in Table 1, where it is desired to make 26,000 pph of steam. The steam temperature is uncontrolled.

It is obvious that with the same inlet gas conditions as in the earlier case, we need additional fuel input to the HRSG to generate 26,000 pph. The procedure is similar to the earlier one. However, additional steps are necessary to iterate for the firing temperature, as discussed in Appendix 2. The method of computing the fuel input, firing temperature and gas analysis is discussed elsewhere.[6] Let us only check the final results which are shown in Fig. 4.

Superheater performance. Table 2b shows the gas properties for the gas analysis after combustion. From the printout, Fig. 4, it is seen that the HRSG gas inlet temperature is 1,034°F and the burner fuel input is 9.29 MMBtu/h (LHV basis).

Wg = 165,430; Ws = 26,000; ts_2 = 677°F. Fg at the average gas temperature is 0.142. The saturation temperature is 462°F, at the corrected drum pressure of 463 psig. Q3a = 26,000(1,346.0 - 1,204.3) = 3.69 MMBtu/h. Gas temperature drop= 3,690,000/(165,430) (0.99) (0.278) = 81°F. Exit gas temperature, Tg_2 = 1,034 - 81 = 953°F; ΔT = 420°F. K_1 = 24.1. $(US)_p = 165{,}430^{o65}(0.142)\ (24.1)\ (26{,}000/18{,}510)^{^\circ 15}$ = 8,840. Then, Q_{st} = 420(8,840)=3.71 MMBtu/h.

Evaporator performance. Fg = 0.135; K_2 = 387.6; hence $(US)_p = 165{,}430^{o65}(0.135)\ (387.6)$ = 129,437. Using Eq. 8, $[(953 - 462)/(Tg_3 - 463)] = {}_e(129{,}437/165{,}430/0.99/0.27)$ = 18.67. Hence, T_{g3} = 489°F. Q_2 = 165,430(0.99) (0.27) (955 - 489) = 20.52 MMBtu/h.

Economizer performance. T_{w2} = 435; hw_2 = 414.45; Q_{3a} = 26,000(1.02) (414.45 - 209.6) = 5.43 MMBtu/h. Gas temperature drop = 128°F; Tg_4 = 489 - 128 = 361°F. ΔT = 83°F. K_3 = 218.4; Fg = 0.120; $(US)_p = 165{,}430^{0.65}\ (218.4)\ (0.120)$=65,000. Hence Q_{st} = 83(65,000) = 5.4 MMBtu/h. Total energy transferred = 3.71 + 20.52 + 5.4 = 29.63 MMBtu/h. Ws=29.63 x 10^6/[(1,346.7 - 209.6) + 0.02(442.6 - 209.6)] = 25,970 pph. The gas pressure drop could be corrected as before.

This gives an idea of the complexity of performance calculations if fuel firing is involved. Several iterations of performance calculations would be required before the correct firing temperature is arrived at. Also, if the steam temperature has to be controlled, the superheater has to be split up into two stages with a spray desuperheater in between. The method of computing the spray water for steam temperature control is discussed elsewhere.' In such an HRSG, more iterations are involved before the spray water flow and the final temperature profiles are arrived at. Without a computer it would be extremely tedious and time consuming. Fig. 5 shows the results of case 4 where steam temperature control and fuel firing are involved.

Note that gas inlet temperature is 1,067°F. The spray quantity has been arrived at based n a split in the ratio of 6:4 in design U times S values between the first and second stages of the superheater. This ratio is built into the program. Slight changes in the temperature profile and spray quantity can result due to a different split in the surfaces between the two stages of the superheater while actually building the HRSG. Also, note the higher steam pressure drop in the superheater due to the lower steam pressure. The economizer flow includes the 2,500 pph saturated steam taken off the drum.

A note of caution on U, S and U times S values. Note that US values could be computed for each surface from its Q and ΔT data. For instance in the "design" case, for the superheater, US = 234,000/311 = 7,524. These would naturally change depending upon the gas flow, analysis and temperature profile. Hence, these values should be interpreted with caution.After

Appendix 1: Performance calculation procedure

The procedure is discussed for a single pressure HRSG. Fig. 6 shows the various configurations of HRSGs considered. The first case is quite involved. The methodology for this case will be discussed. The gas flow, gas inlet temperature and analysis, steam pressure and feed water temperature are assumed to be known. The design calculations, which are the basis of establishing an initial design, are assumed to be done and the results available, along with K_1, Kz, K_3 factors.

1. Assume the steam flow. A good estimate is obtained by using a ratio of the "performance" to "design" gas flows and temperature drop.
2. Solve the superheater performance. This is an iterative process. See Appendix 2, Eqs. 1 to 5. If the transferred and assumed duty are not equal, repeat with another steam temperature or else continue.
3. Solve the evaporator performance. Obtain the duty and exit gas temperature using Eqs. 6 to 9.
4. Solve economizer performance using Eqs. 10 to 13. This is again an iterative procedure. Calculate the total transferred duty.
5. The steam flow is then corrected based on the total transferred duty and enthalpy rise, Eq. 14. If this is close to the assumed steam flow in step 1, continue or else repeat steps 1 to 5.
6. If the final steam temperature is greater than that desired, the steam flow is corrected for the desired steam temperature.
7. If the desired steam flow is zero (unfired mode) or less than the corrected flow, proceed to step 11.
8. If the desired steam flow is larger than the corrected flow, calculate the fuel input required to raise the gas temperature to the required level to achieve the desired steam flow. This again involves several iterations, and for each firing temperature, all the steps from 1 to 8 have to be repeated until they match.
9. If the final steam temperature is higher than desired, calculate the interstage spray quantity based on a split superheater.
10. Another round of fine tuning is done to check the temperature profiles and steam flow.
11. It can be easily seen that a lot of iterative calculations are involved. For each round, the gas and steam properties have to be computed based on the gas analysis and temperature. If there is steaming in the economizer, the economizer is split up into two stages, a small evaporator and an economizer and calculations are done to evaluate the extent of steaming. It is obvious that without a computer, the calculations can be overwhelming, particularly if there are several alternate performance conditions, and steam is generated at several pressure levels.

Appendix 2: Equations used in performance calculations

Superheater performance. Assuming that the steam flow = Ws, from energy balance we have:

$$Q1a = Ws(hs_2 - hs_1) = Wg(Cp)(hf)(Tg1 - Tg2) \quad (1)$$

where ts2 = exit steam temperature and hs_2, the enthalpy. Compute the exit gas temperature, Tg2, from the above. The transferred duty is then:

$$Q_{1t} = (US)_p \Delta T \quad (2a)$$

ΔT = log-mean temperature difference

$$\Delta T = [(Tg1 - ts_2) - (Tg_2 - ts_1)]/\ln[(Tg_1 - ts2)/(Tg2-ts1)]$$

assuming counter flow configuration, which is widely used. (US)p is the product of S and U in performance mode and is obtained from the (US) value in the design case by adjusting as follows for the gas properties and flow.

$$(US)_p = Wg^{0.65} FgK_1(WS/Wsd)^{0.15} \quad (3)$$

K_1 is obtained from Q1, ΔT, Wg and Fg values in design case:

$$K1 = Q1/(\Delta T(Wg^{0.65})(Fg)) \quad (4)$$

$$Fg = (Cp^{0.33}k^{0.67}/(\mu^{0.32}) \quad (5)$$

If for the assumed steam temperature Q1a and Q1, do not come close (say within 0.5%), another iteration is warranted. All of the above steps are repeated until Q1a and Q1, match.

Evaporator performance. From energy balance, Q2 = $Wg(Cp)$ (hf) (Tg2 - Tg3) = $(US)_p \Delta T$ (6) where ΔT = $[(Tg_2 - ts) - (Tg_3 - ts)]/\ln[(Tg_2 - ts)/(Tg_3 - ts)]$

$$= (Tg2 - Tg3)/\ln[(Tg2 - ts)/(Tg3 - ts)] \quad (7)$$

From Eqs. 6 and 7 after simplification, we have:

$$[(Tg2 - ts)/(Tg3 - ts)] = e^{[(US)p/(Wg)(cp)(hlf)]} \quad (8)$$

where:

$$(US)p = Wg^{0.65}FgK_2 \quad (9)$$

K_2 is computed as in Eq. 4 from the design conditions. Fg is computed for the performance conditions. Tg_3 is solved from Eq. 8 without iteration. Q_2, the duty, can be obtained from **Eq. 6.**

Economizer performance. Assume tw2, the water exit temperature. Then,

$$Q_{3a} = Ws(hw2 - hw1)(1+bd) = W_gCp(Tg_3 - Tg_4)hf \quad (10)$$

Obtain Tg4 and then the ΔT, assuming counter flow conditions

$$\Delta T = [(Tg4 - tw_1) - (Tg3 - tw2)]/\ln[(Tg4 - tw_1)/(Tg3 - tw2)] \quad (11)$$

$$\text{transferred duty } Q3t = (US)p\,\Delta T \quad (12)$$

where

$$(US)_p = Wg^{0.65}.Ks \quad (13)$$

K_3 is obtained as in Eq. 4 from design conditions. If Q3a and Q_3, are close, continue or else the iteration continues from Eqs. 10 to 13 with a different tw2. The steam flow is then corrected as follows:

$$Ws_c = (Q_{1t} + Q_{2t} + Q_{3t})/[(hs_2 - hw_1) + bd(hf - hw1)] \quad (14)$$

If Ws, is not close to the assumed flow, Ws, the calculations are repeated starting with the superheater. The gas pressure drop is corrected for performance conditions:

$$\Delta P = (\Delta P)d\,(Wgl\,Wgd)^2\,[(Tavg + 460)1(Tavgd + 460)] \quad (15)$$

try to split up the U and S values and compare alternate designs based on S values alone. This can lead to very misleading conclusions and the author strongly recommends against it, particularly if extended surfaces are used.

With finned tubes, the gas side heat transfer coefficient and fin efficiency are affected by variables such as fin density, height, thickness and fin or tube material .6, ',s By using tubes with high fin density, say six, one could show more surface in the HRSG, but due to the lower U associated with it, it does not mean that the energy transferred is more compared to a design which has a lower fin density, say two to four, and hence, lower S. Lower fin density should be used whenever possible to increase U and minimize gas pressure drop and fin and tube wall temperatures. This is more im

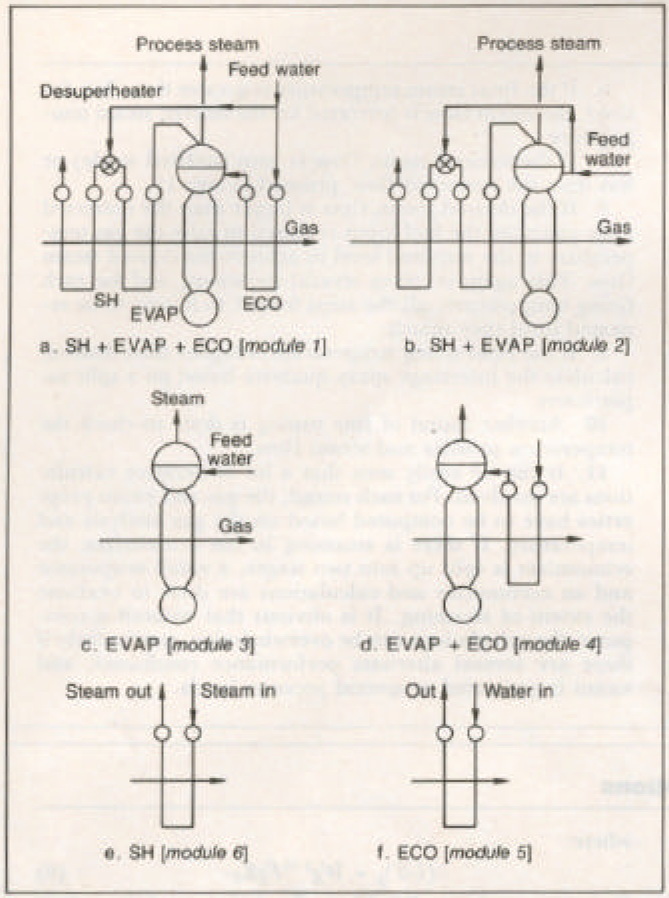


Fig. 6—Various modules can be combined to represent multiple pressure and complex HRSG configurations.

portant in surfaces with low tube side heat transfer coefficients such as superheaters. One could show that S can be 100 to 200% more by using six fins/in. compared to two, but due to the higher U, the duty can be the same or even more. The author has performed studies on optimization of finned tubes[6, 7] and advises engineers against comparing and selecting HRSGs simply because the surface area, S, is more compared to another design which uses lower fin density. **Unless the engineer is familiar with all aspects of heat transfer with extended surfaces and the impact of each variable on U, comparisons of S alone can be misleading and should be avoided.**

Limitations and software. The approach discussed has a limitation. It cannot be used in HRSGs which have a radiant section. However, the author is of the view that 80 to *90%* of HRSGs fall under the category discussed in Fig. *6*, and hence the methodology discussed can be applied to a wide variety of HRSGs used in the industry.

While the method of predicting performance using U values based on actual tube geometry, fin configuration, etc., gives accurate results, this methodology has been checked against several designs and operating results. For the purposes of engineering analysis, trend projections, evaluation of alternate designs and for studying the effect of different gas/steam parameters on performance, this approach is very effective and hence a powerful tool.

Considering the complexity of the calculations and iterative nature of the procedure, particularly if multipressure HRSGs are involved, a program has been developed by the author for HRSG design and performance evaluation. For more information on the software and its availability, contact the author at *PO. Box 673*, Abilene, Texas *79604, USA*.

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. 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 150 articles on boilers, heat transfer and steam plant systems and has written four books: Applied Heat Transfer, Steam Plant Calculations Manual, Nomograms for Steam Generation and Utilization and Basic Programs for Steam Plant Engineers (book and diskette), copies of which are available from him. He also has contributed several chapters to the Encyclopedia of Chemical Processing and Design, Vols. 25 & 26, Marcel Dekker, New York.

NOMENCLATURE

bd-Blow down fraction; if blow down = 2%, then $bd = 0.02$
Cp-Gas specific heat, Btu/1b °F

Fg-A factor accounting for gas properties, defined in Eq. 5
hlf--Heat loss factor; if heat loss = 2%, then $hlf = 0.98$
hs2, hs1-Enthalpy of superheated steam and inlet steam, Btu/lb
hw_1, hw_2-Enthalpy of water at eco inlet and exit, Btu/ lb
k-Gas thermal conductivity, Btu/ft h°F

K_1, K_2, K_3-Factors obtained from design conditions, Eq. 4
Q_1, Q_2, Q_3 -Energy absorbed in superheater, evaporator and economizer, Btu/h; subscript a = assumed and t = transferred
Tg_1, Tg_2, Tg_3, Tg_4-Gas temperature distribution,
S-Surface area, sq ft
ΔT-Log-mean temperature difference, °F
Tavg, Tavgd-Average gas temperature in HRSG in performance and design modes
Tw_1, tw2-Water temperature at inlet and exit of economizer, F
ts_1, ts_2-Saturated and superheated steam temperature, °F
U-Overall heat transfer coefficient, Btu/sq ft hF
$(US)_p$-Product of U and S in performance mode Wg,
Wgd-Gas flow in performance and design modes, pph
Ws, Wsd-Steam flow in performance and design mode
$(\Delta P)_{d,p}$-Gas pressure drop in design and performance, in. WC
μ-Gas viscosity, lb/ft h

Note: "Design" case is the basis used to arrive at initial temperature profiles, steam flow, and the design. "Performance" case predicts the performance of the HRSG so designed at different gas or steam parameters.

LITERATURE CITED

1.Ganapathy, V., ***Applied heat transfer,*** Pennwell Books, Tulsa, 1982.

2 Ganapathy, V., "HRSG features and applications," ***Heating, piping and air-conditioning,*** Jan. '89.
s Ganapathy, V., et al., "Heat recovery boilers for process and cogeneration applications," Seventh Industrial Energy Technology Conference, Houston, May '85.
4 Ganapathy, V., "HRSG temperature profiles guide energy recovery," ***Power,*** Sept. '88.
s Ganapathy, V., "HRSGs for gas turbine applications," ***Hydrocarbon Processing,*** Ang. '87.
b Ganapathy, V., "Charts simplify spiral finned tube calculations," ***Chemical Engineering,*** April 25, 1977, p. 117.
7 Ganapathy, V., "Charts help evaluate finned tube alternatives," ***Oil and Gas ,Journal,*** Dec. 3, 1979, p. 74.
e Ganapathy, V., ***Nomograms for steam generation and utilization,*** Von Nostrand Reinhold, 1988, p. 77.
9 Ganapathy, V., "Program computes fuel input, combustion temperature," ***Power Engineering,*** July '86.
10 Ganapathy, V., "Determine spray water to desuperheat steam," ***Healing, piping and air-conditioning,*** Dec. '87.