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**Characterization of Operating Conditions
for Gas/Water Heat Recovery
Steam Generators**

R. L. Graves

OAK RIDGE NATIONAL LABORATORY
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CHARACTERIZATION OF OPERATING CONDITIONS FOR GAS/WATER
HEAT RECOVERY STEAM GENERATORS

R. L. Graves

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CHARACTERIZATION OF OPERATING CONDITIONS FOR GAS/WATER HEAT RECOVERY STEAM GENERATORS

R. L. Graves

ABSTRACT

The possible operating points of air/water and helium/water steam generators are parametrically presented. Calculation procedures and a sample computer program listing are provided. Multiple saturated steam conditions are shown to be attainable with the same gas inlet and exit temperatures. Design considerations of heat transfer surface area and pumping requirements are also discussed.

1. INTRODUCTION

Gas/water heat exchangers, usually characterized by the names waste heat boiler and heat recovery steam generator (HRSG), are widely used throughout the industrial sector. The recent and somewhat sudden escalation of fuel prices has increased interest in these devices as a means of increasing the fuel economy of industrial processes by recovering the heat from the hot efflux of various furnaces or power systems. In attempting to optimize plant heat recovery systems and cogeneration systems, the HRSG does not lend itself to the routine treatment of gas/gas or water/water heat exchangers due to the so-called pinch point temperature difference (PTD) as defined in Figure 1. The additional perturbation of the PTD makes analysis of the HRSG more than routine and makes optimization rather awkward. Optimization here implies a design which would generate the highest possible temperature steam or a specific flow rate of steam while cooling the gas to the lowest possible temperature. It is the purpose of this report to parametrically characterize the HRSG in order to facilitate systems optimization and to present a method of calculation appropriate for essentially any gas and any fluid which has properties similar to steam, that is, liquid-to-vapor phase change behavior in the temperature region of interest.

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Fig. 1. Definition of pinch point in a counterflow, once-through HRSG.

2. GOVERNING ENERGY RELATIONS

The fundamentals of the HRSG analysis are presented here. The information desired in this analysis can be acquired satisfactorily without involving the various fluid flow and heat transfer relations which appear in many heat exchanger problems. The energy states refer to the numbers identified in Figure 2. The superheat region is not treated in this analysis as it lends itself to conventional heat exchanger and energy balance relations.

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Fig. 2. Reference points for analytical treatment.

In the boiling zone of the HRSG, the governing energy equation is

$$\dot{m}_s h_{fg} = \dot{m}_g (h_{g1} - h_{g2}) . \quad (1)$$

We require, however, that a minimum pinch point result at State 2, that is

$$T_{g2} = T_{sat} + \text{PTD} \quad (2)$$

where T_{g2} is derived by rearranging (1).

$$h_{g2} = h_{g1} - \frac{\dot{m}_s}{\dot{m}_g} (h_{fg}) . \quad (3)$$

In the preheat zone the energy relation is written as

$$\begin{aligned} \dot{m}_s (h_f - h_{w3}) &= \dot{m}_g (h_{g2} - h_{g3}) \\ h_{g3} &= h_{g2} - \frac{\dot{m}_s}{\dot{m}_g} (h_f - h_{w3}) . \end{aligned} \quad (4)$$

We have arrived at three independent equations, (2), (3), and (4) through which the unknowns T_{sat} , T_{g2} , and h_{g3} should be accessible.

In many instances, it may be desirable to specify the gas inlet and outlet temperatures and let the steam temperature and flow rate vary in order to satisfy the energy balance. The following algorithm may be used to accomplish this in an iterative manner:

1. Input necessary base data including values for h_{g1} , h_{g3} , PTD, h_{w3} .
2. Make an initial guess for P_{sat} or T_{sat} from which the values of h_{fg} and h_f arise.
3. Calculate h_{g2} , i.e., T_{g2} from Eq. 2 and $\dot{m}_s/\dot{m}_g$ from Eq. 3.
4. Calculate h_{g3} , i.e., T_{g3} from Eq. 4. If the desired value of T_{g3} has been obtained, the calculation is over. If the proper value has not been reached, make a new guess for P_{sat} and repeat steps 3-4. Although once-through boiling has been implied, recirculation of saturated liquid serves only to raise the water inlet temperature without effect on

the gas. Depending on the iteration scheme, some multiple solutions may not be found, hence a parametric survey is recommended by performing steps 1-4 for various gas inlet temperatures. In this parametric mode any value of T_{g3} is acceptable except those, of course, which may be less than T_{w3} . Although this algorithm is quite elementary and requires no unusual programming skills, a listing for a sample program is provided in the Appendix.

Due to the nature of the steam properties (Fig. 3), multivalued solutions to the problem might be anticipated, and such foresight has proven to be correct. As shown in Figure 4, there are some mass flow ratios for which two values of T_{sat} will satisfy a particular pinch point requirement. Other mass flow ratios satisfy either larger or smaller PTD requirements.

3. RESULTS FOR TYPICAL HRSG

A map of operating points for an air-heated HRSG is presented in Figure 5, derived by the procedures outlined in Section 2. Double-valued solutions are evident for cases where the gas inlet temperature is higher than approximately the critical temperature for steam. This characteristic is attributable to the nature of the steam properties.

For waste heat recovery applications it is generally desirable to utilize as much of the sensible heat in the gas as practical. This implies that the gas exit temperature be low. An important revelation of this study is that the gas utilization can be essentially the same for the generation of either high or low pressure steam. For example, with 1000°F (538°C) inlet air exiting at 100°F (38°C), it is possible to generate steam at either 2350 psia (16.2 MPa) or 6.0 psia (0.04 MPa). With a 200°F (93°C) gas exit temperature, the options would be 550 psia (3.8 MPa) or 1550 psia (10.7 MPa). Given the option of two such steam pressures, the high pressure is more advantageous even though the energy content (enthalpy) may be the same or less than the low pressure steam. The advantages are summarized in the following statements.

1. With slight superheating, 200-400°F (93-204°C), the high pressure steam becomes satisfactory for power generation in a back-pressure or condensing turbine.

Fig. 3. Liquid enthalpy, h_f , vapor enthalpy, h_g , and latent heat, h_{fg} , of saturated steam.

Fig. 4. Double value of T_{sat} for satisfying pinch requirement air inlet temperature, $T_{g1} = 1000^\circ\text{F}$ (538°C), water inlet temperature, $T_{w3} = 60^\circ\text{F}$ (16°C).

Fig. 5. Operating points for air-heated HRSG.

2. The high temperature steam is suitable for a broader range of process needs.

A comparable chart for helium/water HRSG is provided in Figure 6. Helium has been examined as a potential working fluid for closed cycle gas turbines^{1,2} and is used as a coolant in some gas cooled reactors.³

The effect of increasing the pinch point temperature difference is depicted in Figure 7. A higher PTD results in a substantially lower steam pressure in the low steam pressure region. The upper limit on attainable steam pressure is seen to be slightly higher, however. A more important parameter influenced by the PTD is the mean temperature difference to which the boiler heat transfer surface area is inversely proportional. This effect will be discussed in a following section.

4. DESIGN CONSIDERATIONS FOR THE HRSG

Having identified the thermodynamic characteristics of the HRSG, two design considerations are briefly discussed in this section, namely the effect of generating high pressure steam on the HRSG size and pumping requirements.

4.1 Heat Transfer Surface Requirements

An example is presented here to demonstrate the effect which the generation of higher pressure steam has on the HRSG surface area. An indicator for surface requirements is the log mean temperature difference (LMTD) which is, identically,

$$\text{LMTD} = \frac{Q}{UA} .$$

The LMTD is derived in terms of the temperatures of the inlet and outlet streams of the two fluids exchanging heat. The primary restrictions on the use of the LMTD is that its derivation presumes that the specific heats of the fluids are constant. Hence, a value for LMTD cannot be calculated through the pinch point. However, an average LMTD can be obtained by respective calculations in the saturation region and subcooled liquid region. Such an average should be weighted in accordance with the amount

Fig. 6. Operating points for helium-heated HRSG.

Fig. 7. Effect of increased PTD on operating points of HRSG.

of heat transferred in the two regions. The respective relations for calculating the LMTD are, in the boiling region,

$$\text{LMTD} = \frac{(T_{g2} - T_{\text{sat}}) - (T_{g1} - T_{\text{sat}})}{\ln \left(\frac{T_{g2} - T_{\text{sat}}}{T_{g1} - T_{\text{sat}}} \right)}$$

and, in the subcooled liquid region,

$$\text{LMTD} = \frac{(T_{g3} - T_{w3}) - (T_{g2} - T_{\text{sat}})}{\ln \left(\frac{T_{g3} - T_{w3}}{T_{g2} - T_{\text{sat}}} \right)}$$

The steam conditions and results of the comparison are presented in Table 1. This comparison provides a good indication that the surface area for the high pressure HRSG will be considerably greater. Although the convective and boiling heat transfer coefficients will generally increase with increasing pressure, the gas side coefficient is dominant such that the overall coefficient of heat transfer, U , will not be greatly influenced. For the same reason the effective overall coefficient will not be greatly dependent on the relative quantities of heat transferred in the respective saturation and subcooled liquid zones of the HRSG.

The generation of higher pressure steam would obviously require a larger capital investment. This situation can be ameliorated by increasing the PTD. For example, doubling the PTD to 50°F (28°C) raises the LMTD by about 25%. This change in design conditions appears to offer some advantage at the high steam pressures since neither the steam pressure nor the fuel economy would be reduced (see Fig. 7).

Table 1. Comparison of HRSG LMTD for two steam pressures

	Steam at 330°F (166°C) 103 psia (0.71 MPa)	Steam at 645°F (341°C) 2130 psia (14.7 MPa)
Air inlet temperature	1000°F (538°C)	1000°F (538°C)
Air exit temperature	150°F (66°C)	150°F (66°C)
Pinch point ΔT	25°F (13.9°C)	25°F (13.9°C)
Water inlet temperature	60°F (16°C)	60°F (16°C)
Percent of heat load transferred in saturation region	76.7%	40.0%
Percent of heat load transferred in sub-cooled region	23.3%	60.0%
Weighted average LMTD	162.3°F (90.2°C)	80°F (44.4°C)

4.2 Pumping Power

The steam side pumping power is about 20 times higher for the high pressure case for the comparison presented in Table 1. This power, however, is still small, especially relative to the power producing capacity of the high pressure steam.

Perhaps of more importance is the gas side pressure drop which would be proportional to the size of the HRSG. Since the gas flow rate is often 5-6 times the steam flow rate, the parasitic compressor or fan power could be quite noticeable for the larger, high pressure HRSG. For instance, with reference to the high pressure steam boiler in Table 1, the air side pumping power is about twice that of the steam side, assuming a 3% pressure drop in the air side.

5. CONCLUDING REMARKS

The information in this report was developed primarily for use in parametric studies of the Coal Combustor for Cogeneration (CCC) program at the Oak Ridge National Laboratory.⁴ This cogeneration system consists of a gas (air) turbine cycle coupled to a fluidized bed coal combustor via a closed heat exchanger in the fluidized bed with the optional use of the HRSG. The data and calculation methods are, however, of a general nature and may be useful in evaluating other gas turbine systems or nonpower systems.

REFERENCES

1. D. H. Pomeroy et al., Advanced Energy Conversion Systems-Conceptual Designs, Part 2, Closed Turbine Cycles, ECAS General Electric Phase II Final Report, NASA-CR 134949, December 1976.
2. A. P. Fraas, Summary of the R&D Efforts on Closed Cycle Gas Turbines, ORNL/TM-6237 (to be published), Oak Ridge National Laboratory.
3. Small HTGR Steamer for Industrial Applications, GA-A13888, General Atomic Company, April 1976.
4. R. L. Graves, R. S. Holcomb, and J. R. Tallackson, Assessment of an Atmospheric Fluidized Bed Coal Combustion Gas Turbine Cogeneration System for Industrial Application - Summary Report, ORNL/TM- (to be published), Oak Ridge National Laboratory.

NOMENCLATURE

A	heat transfer surface area
h_f	saturated water enthalpy
h_{fg}	enthalpy of vaporization
h_g	gas enthalpy
h_w	subcooled water enthalpy
HRSG	heat recovery steam generator
LMTD	log-mean temperature difference
$\dot{m}_g$	mass flow rate of gas
$\dot{m}_s$	mass flow rate of steam
PTD	pinch point temperature difference

Q	heat transfer rate
T_g	gas temperature
T_{sat}	steam saturation temperature
T_w	water temperature

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APPENDIX

A sample listing of the computer program for HRSG calculations is included here. Major parameters are defined in the program list. Steam properties were available through stored FUNCTION subroutines. Polynomials were derived for air properties, and a constant specific heat was found satisfactory for helium. Sources of the respective thermodynamic properties are listed below.

Steam:

C. A. Meyer et al., Thermodynamic and Transport Properties of Steam, American Society of Mechanical Engineers, 1967.

Air:

J. H. Keenan and J. Kaye, Gas Tables — Thermodynamic Properties of Air Products of Combustion and Component Gases, John Wiley & Sons, Inc., London, 1950.

Helium:

S. W. Akin, General Electric Company, NP-124, May 1947.

F. G. Keyes, Project Squid, NP-4621, April 1952.

(Information from these two reports collated in an unpublished ORNL report.)


```

COMPILER OPTIONS - NAME= MAIN,OPT=02,LINECNT=60,SIZE=0000K,
                   SOLRCE,EBCCIC,NOLIST,NODECK,LOAD,MAP,ACEOIT,IC,ACXREF
ISN 0002      IMPLICIT REAL*8(A-H,C-Z)
ISN 0003      REAL*8 MRAT
C              HW1-INLET ENTHALPY OF SUBCOOLED WATER BTU/LBM
C              TW1-INLET TEMPERATURE OF SUBCOOLED WATER -F
C              PSAT-STEAM SATURATION PRESSURE PSIA
C              TG1- INLET GAS TEMPERATURE -F
C              MRAT-STEAM FLOW/GAS FLOW
C              HG1-INLET ENTHALPY OF GAS BTU/LBM
C              DTPIN-PINCH POINT TEMPERATURE DIFFERENCE -F
C              TGEX-DESIRED EXIT TEMPERATURE OF GAS -F
C              TSAT- SATURATION TEMPERATURE OF STEAM -F
C              HF-SATURATED LIQUID ENTHALPY BTU/LBM
C              HG- SATURATED VAPOR ENTHALPY BTU/LBM
C              TG2-TEMPERATURE OF GAS AT THE PINCH POINT -F
C              TG3- TEMPERATURE OF GAS AT EXIT -F
C      *****INITIALIZE*****
ISN 0004      C=0.0
ISN 0005      C=C.0
C      ***** THIS VALUE OF MRAT IS ONLY AN INITIAL GUESS FOR ITERATION *****
ISN 0006      MRAT=0.25
ISN 0007      HW1=28.08
ISN 0008      TW1=60.0
C      ***** THIS VALUE OF PSAT IS ONLY AN INITIAL GUESS FOR ITERATION*****
ISN 0009      PSAT=7.0
ISN 0010      TG1=1000.0
ISN 0011      TGEX=200.
ISN 0012      T=TG1+460.0
ISN 0013      HA=-2.92799+C.256036*T-3.71459E-05*T**2+3.44164E-08*T**3
              #=-1.0595E-11*T**4+1.19813E-15*T**5
ISN 0014      HG1=HA
ISN 0015      DTPIN=25.0
C      ***** DTPL, CTPI, TEXPL, TEXPI SET LIMITS OF CONVERGENCE *****
ISN 0016      DTPL=1.005*DTPIN
ISN 0017      CTPI=0.995*CTPIN
ISN 0018      TEXPL=1.005*TGEX
ISN 0019      TEXPI=0.995*TGEX
C      *****STARTING ITERATION LOOP*****
ISN 0020      SS CONTINUE
ISN 0021      1CC CONTINUE
ISN 0022      TSAT=TSL(PSAT)
ISN 0023      HF=HFP(PSAT)
ISN 0024      HG=HGP(PSAT)
ISN 0025      HFG=HG-HF
ISN 0026      TG2=TSAT+CTPIN+460.0
ISN 0027      T=TG2

```

```

ISN 0028      HA=-2.92795+C.256036*T-3.71459E-05*T**2+3.44164E-08*T**3
              #=-1.0595E-11*T**4+1.19812E-15*T**5
ISN 0029      HG2=HA
ISN 0030      MRAT=(HG1-HG2)/HFG
ISN 0031      DTREAL=IG2-TSAT-460.C
              C***** CHECK FOR SATISFACTION OF PINCH POINT REQUIREMENT *****
ISN 0032      IF (DTREAL.LE.DTFL.AND.DTREAL.GE.DTMI) GO TO 200
ISN 0034      IF (DTREAL.GT.DTFL) MRAT=MRAT*1.CC5
ISN 0036      IF (DTREAL.LT.DTMI) MRAT=MRAT*C.99
ISN 0038      C=C+1.
ISN 0039      GO TO 100
ISN 0040      200 CONTINUE
ISN 0041      HG3=HG2-MRAT*(HF-HM1)
ISN 0042      H=HG3
ISN 0043      T=16.5159+3.77376*T+3.89543E-03*T**2-1.67681E-05*T**3
              #+3.06137E-08*T**4-2.80029E-11*T**5+1.04638E-14*T**6
ISN 0044      TG3=T-460.C
ISN 0045      TG2=IG2-460.C
ISN 0046      PRINT 44,PSAT,TSAT,TG2,TG3,DTREAL,MRAT
ISN 0047      44 FORMAT (1H,6F9.3)
              C***** CHECK FOR SATISFACTION OF EXIT GAS TEMP REQUIREMENT *****
ISN 0048      IF (TG3.LE.TEXPL.AND.TG3.GE.TEXMI) GO TO 300
ISN 0050      IF (TG3.LT.TEXMI) PSAT=PSAT+5.C
ISN 0052      IF (TG3.GT.TEXPL) PSAT=PSAT*C.95
ISN 0054      GO TO 99
ISN 0055      300 CONTINUE
ISN 0056      PRINT 22,C
ISN 0057      PRINT 66
ISN 0058      66 FORMAT (1H1,30X,*** WASTE HEAT BOILER OPERATION ***,//)
ISN 0059      PRINT 77,TG1,TM1,TG3,TSAT
ISN 0060      77 FORMAT (1H, 'GAS INLET TEMP=',F8.2,3X,'WATER INLET TEMP=',F8.2,3X,
              # 'GAS EXIT TEMP=',F8.2,3X,'STEAM EXIT TEMP (SAT)=',F8.2,/)
ISN 0061      PRINT 88,MRAT,DTREAL,PSAT
ISN 0062      88 FORMAT (1H, 'MASS FLOW RATIO-STEAM/GAS=',F5.3,3X,'PINCH POINT TEMP
              # DIFFERENCE=',F8.2,3X,'SAT PRESSURE-PSIA=',F8.2,/)
ISN 0063      22 FORMAT (1H, F8.3)
ISN 0064      STOP
ISN 0065      END

```

*** WASTE HEAT BOILER OPERATION ***

GAS INLET TEMP= 1000.00 WATER INLET TEMP= 60.00 GAS EXIT TEMP= 199.14 STEAM EXIT TEMP (SAT)= 479.22

MASS FLOW RATIO-STEAM/GAS=0.171 PINCH POINT TEMP DIFFERENCE= 25.00 SAT PRESSURE-PSIA= 562.00

IF=00021 STOP C

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