Modeling Boiler Economizers With Finned Tubes

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ABSTRACT

The PEPSE® heat balance program can be used to model extended-surface fossil boiler economizers when design or test performance data for at least one load case are available. The geometry of the extended surface does not need to be known. The method includes the use of a multiplier of the overall heat transfer coefficient calculated for smooth tubes. The results of this study indicate that the value of the heat transfer multiplier is essentially constant for the 50% to 100% load range. When design performance data for a load below 50% load are available, load cases between the lowest design load case and 100% load may be accurately represented. The multiplier may vary because corrective heat transfer at low gas flow conditions may become nonlinear due to flow stagnation and nonuniform temperature. The general magnitude of the overall heat transfer multiplier is in agreement with the approximate value of the multiplier obtained from a theoretical approach using a weighted fin efficiency for the finned tube heat exchangers.

INTRODUCTION

This report demonstrates the use of the PEPSE heat balance code to model finned tube fossil boiler economizers. Two finned tube economizers were modeled: one from a drum-type boiler and one from a supercritical boiler. Because PEPSE does not contain extended-surface calculations, the heat transfer calculated for smooth tubes was increased to simulate the added fin heat transfer by applying a multiplier to the PEPSE-calculated overall heat transfer coefficient.

MODEL DEVELOPMENT

The design data available for the study of these two economizers were the following: (1) the fuel composition and firing rate, (2) the flue gas flow rate, (3) the flue gas economizer exit temperature, (4) the economizer heat duty, (5) feedwater flow rate and inlet conditions, (6) pressure drops through the economizers, and (7) economizer tube geometry, such as number of tubes per row, number of rows, tube diameter, and so forth. For Case 1, the economizer from the drum-type boiler, data for the 100%, 75%, 50%, and 35% load cases were available. For Case 2, the economizer from the supercritical boiler, data for the 100%, 75%, 50%, and 25% load cases were available.

The economizer was analyzed in a submodel consisting of a convective heat transfer stage to represent the economizer and sources and sinks to represent the flue gas and feedwater flows. The input specifications in accordance with Reference 1 and the PEPSE-calculated results for the 100% load of Case 1 are shown in Appendix A.

The flue gas constituent mass fractions were determined in a separate model using a furnace/combustor component with the specified fuel composition, firing rate, and percent of excess air. In the economizer submodel, the unknown flue gas temperature entering the economizer was determined from the known economizer heat duty and flue gas exit temperature. Two controls were used for this calculation. The first control adjusted the gas inlet temperature to match the heat duty. The second control was used to obtain the design flue gas exit temperature by adjusting a multiplier of the overall heat transfer coefficient. This overall heat transfer coefficient multiplier was determined for each design load case.

PRESENTATION OF RESULTS

The values of the overall heat transfer coefficients, U, the overall heat transfer coefficient multipliers, HTTIRH, and the outer film convective coefficients, ho, for each load case for Case 1 and Case 2 are shown in Table 1.

Additional analyses were performed in which a single value of the heat transfer multiplier was used throughout the load range. The multiplier seclected was the value determined in the previous analysis at the 100% point. Shown in Figure 1 and Figure 2 are the results of the PEPSE prediction using this 100% load point multiplier. Table 2 summarizes the heat duties predicted by the vendor versus the heat duties predicted by PEPSE when assuming a constant overall heat transfer coefficient multiplier. The results of the PEPSE prediction match those of the design prediction from the 100% load case down to the 50% load case. Below the 50% load case, the heat transfer predicted by PEPSE is greater than the design heat transfer. This decrease in the design heat transfer may be a result of flow stagnation and nonuniform temperature of the flue gas caused by the low flue gas velocities at low loads. The availability of design data for loads below 50% provides the opportunity to tailor the PEPSE analyses to accurately calculate these effects in this region by using a suitable value of the multiplier.

Table 3 shows the comparison of the vendor-predicted flue gas pressure drop versus the PEPSE-calculated flue gas pressure drop for both Case 1 and Case 2. The flue gas pressure drops are in very close agreement with the design pressure drops in Case 1 and are adequately matched in Case 2.

TABLE 1

VALUES OF THE ho, HTTIRH, AND U

CASE 1

Load, %	ho, Btu/hr-ft^2-°F	HTTIRH (Multiplier)	U, Btu/hr-ft^2-°F
100	16.25	3.71	60.09
75	13.25	3.83	50.55
50	11.33	3.82	43.15
35	8.72	3.51	30.59

CASE 2

Load, %	ho, Btu/hr-ft^2-°F	HTTIRH (Multiplier)	U, Btu/hr-ft^2-°F
100	20.18	2.03	40.14
75	17.03	2.09	34.96
50	13.67	1.92	25.82
25	8.86	1.40	12.38

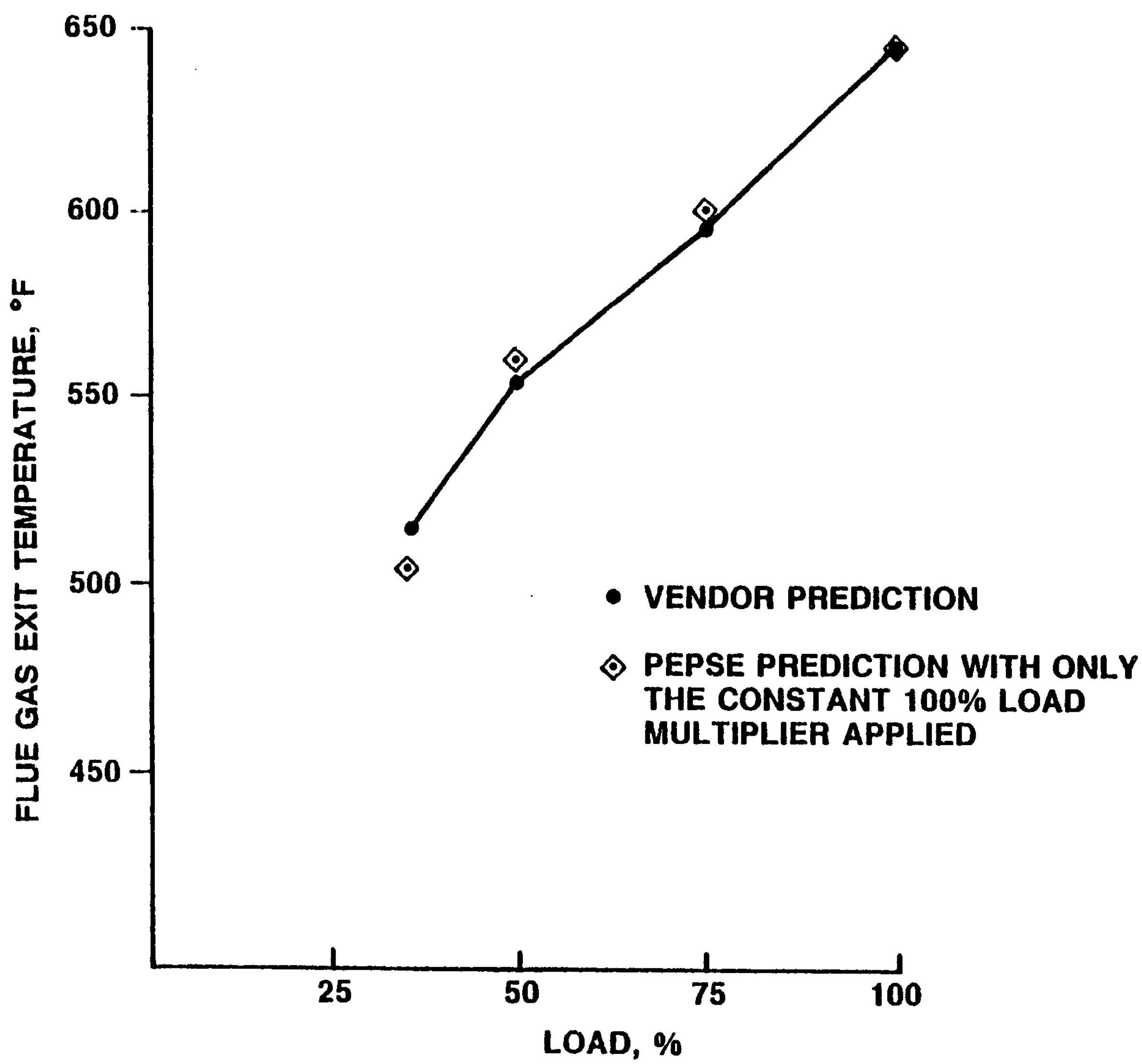

FIGURE 1. ECONOMIZER FLUE GAS EXIT TEMPERATURE VERSUS LOAD FOR CASE 1

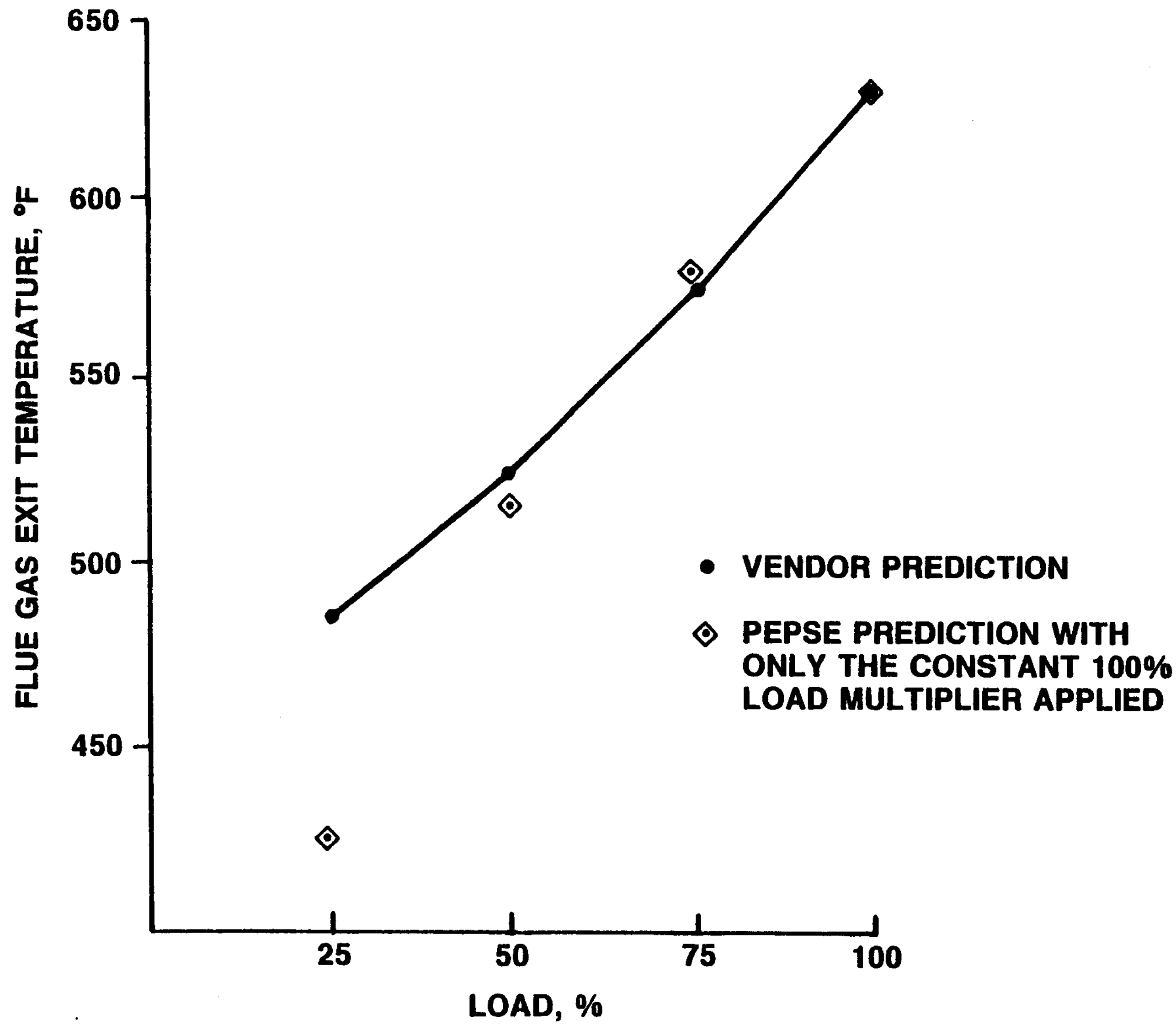

FIGURE 2. ECONOMIZER FLUE GAS EXIT TEMPERATURE VERSUS LOAD FOR CASE 2

TABLE 2

ECONOMIZER VENDOR-PREDICTED VERSUS PEPSE-PREDICTED
HEAT DUTY WHEN ONLY THE 100% LOAD OVERALL HEAT
TRANSFER COEFFICIENT MULTIPLIER IS USED

CASE 1

Load, %	Vendor Heat Duty, 1×10^8 Btu/hr	PEPSE-Calculated Heat Duty, 1×10^8 Btu/hr	Percent Deviation
100	3.6298	3.6298	+0.0
75	2.8842	2.8471	-1.3
50	2.2578	2.2330	-1.1
35	1.5566	1.5884	+2.0

CASE 2

Load, %	Vendor Heat Duty, 1×10^8 Btu/hr	PEPSE-Calculated Heat Duty, 1×10^8 Btu/hr	Percent Deviation
100	15.0650	15.0650	+0.0
75	11.5632	11.7715	-0.8
50	7.3080	7.41266	+1.4
25	4.1700	4.5193	+8.4

TABLE 3

FLUE-SIDE VENDOR-PREDICTED VERSUS
PEPSE-PREDICTED PRESSURE DROPS

CASE 1

Load, %	Vendor Pressure Drop, psi	PEPSE-Calculated Pressure Drop, psi
100	.105	.102
75	.058	.058
50	.039	.037
35	.016	.018

CASE 2

Load, %	Vendor Pressure Drop, psi	PEPSE-Calculated Pressure Drop, psi
100	.229	.375
75	.146	.228
50	.087	.118
25	.029	.032

DISCUSSION OF RESULTS

Overall Heat Transfer Coefficient

The calculation of the overall heat transfer coefficient as described in Reference 2 includes terms for the effects of the inner and outer film convective heat transfer coefficients, intertube radiation, tube metal conductivity, and fouling. Table 4 presents an independent hand calculation of the outer convective heat transfer coefficient for one load case. Table 5 shows the hand calculation of the overall heat transfer coefficient. By far the dominant term in the equation for the overall heat transfer coefficient is the outer film convective coefficient. In fact, if all the terms except the outer film convective coefficient are neglected, the resulting overall heat transfer coefficient is within 2% of the value that includes all of the terms.

Magnitude of the Overall Heat Transfer Coefficient Multiplier

The favorable comparison between the design specifications and the PEPSE results using a single-valued multiplier is encouraging evidence that the approach is sound. A further check of the approach was made by estimating the values of the heat transfer multiplier based on the approach taken by Kern in Reference 3. This, then, allows a comparison between the overall heat transfer multiplier resulting from the PEPSE analyses and those based on the theory from the literature.

Kern (Reference 3) presents a method of defining a composite heat transfer coefficient for the combination of fin and unextended surfaces. Kern develops an equation for this composite heat transfer coefficient, hc, based on the fact that the total heat transfer is the sum of the fin heat transfer and the smooth tube heat transfer. The final form of the equation is shown as follows:

TABLE 4

CALCULATION OF THE OUTER FILM HEAT TRANSFER CONVECTIVE COEFFICIENT

$$h_o = \frac{Co\, G^{0.61} C_p^{0.33} k^{0.67}}{D_o^{0.39} \mu^{0.28}}$$

where

h_o = outside tube heat transfer coefficient, Btu/hr-ft^2-°F
k = flue gas average conductivity, Btu/ft-hr-°F
D_o = tube outside diameter, ft
Co = correlation fit coefficients
μ = flue gas average viscosity, lbm/ft-hr
C_p = flue gas average specific heat, Btu-lbm-°F
G = flue gas mass flux through the stage, lbm/hr-ft^2

For the 100% load case of Case 2, the following approximate film temperature properties were found:

k = 0.029 Btu/ft-hr-°F
D_o = 0.1458 ft
Co = .26528
μ = 0.0823 lbm/ft-hr
C_p = 0.2837 Btu/lbm-°F
G = 10748.63 lbm/hr-ft^2
h_o = 20.12 Btu/hr-ft^2-°F

(PEPSE-calculated value of h_o = 20.18 Btu/hr-ft^2-°F.)

TABLE 5

CALCULATION OF THE OVERALL HEAT TRANSFER COEFFICIENT, U

$$U = \left[\frac{1}{h_i(\frac{D_i}{D_o})} + \frac{1}{h_f} + \frac{1}{(h_{RAD} + h_o)} + \frac{D_o \ln (D_o/D_i)}{2k} \right]^{-1}$$

where

U = overall flue-to-tube fluid heat transfer coefficient, Btu/hr-ft^2-°F
D_i = tube inside diameter, ft
D_o = tube outside diameter, ft
h_f = 1/(Fouling Factor), Btu/hr-ft^2-°F
k = tube metal conductivity, Btu/hr-ft-°F
h_i = tube inside heat transfer coefficient, Btu/hr-ft^2-°F
h_o = tube outside heat transfer coefficient, Btu/hr-ft^2-°F
h_{RAD} = gas-tube effective heat transfer coefficient for radiation, Btu/hr-ft^2-°F

For the 100% load case of Case 2, the following data were determined:

h_i = 2042.6 Btu/hr-ft^2-°F
D_i/D_o = 1.23/1.75 = .702857
h_f = 0.0
h_{RAD} = 0.322585 Btu/hr-ft^2-°F
h_o = 20.1811 Btu/hr-ft^2-°F
D_o = 0.1458 ft
k = 23. Btu/hr-ft-°F
U = 19.768 Btu/hr-ft^2-°F

(If $U = h_o$, then U = 20.18 Btu/hr-ft^2-°F.)

$$hc = (N_f\ A_f + A_o)\ \frac{ho}{Ai} = \text{(a multiplier) } ho$$

where

N_f = fin efficiency,

A_f = fin surface area (both sides),

A_o = tube outside surface area (not including fin base area),

Ai = tube inside surface area,

ho = average value of fin and tube outside surface heat transfer coefficient, and

hc = composite heat transfer coefficient corrected to tube inside surface area.

This approach weights the fin efficiency. The fin efficiency, which is based on the fin geometry and finned surface convective heat transfer coefficient, may be obtained from equations or graphs developed by Gardner in Reference 5. Although the idealized flow pattern assumed in Gardner's development of fin efficiency differs somewhat from the actual flow pattern, this approach is used with the understanding that the heat transfer coefficient, ho, is an average value.

The method used by Kern to calculate the overall heat transfer coefficient, U, is similar to that shown in Table 5. The composite value of hc is used instead of ho. The method used by PEPSE differs in that the multiplier is applied to U instead of ho. Because of the dominant influence of ho on the value of U, any resultant difference caused by the differing approaches is slight.

The approach taken by Kern also differs from the approach used in the PEPSE calculation and the hand calculation in that ho is based on a gas mass flux term that does not include the fin area. This difference also affects the value of the fin efficiency, but the effects of this variation on the value of the multiplier were calculated to be slight for this case.

TABLE 6

CALCULATION OF THE OVERALL HEAT TRANSFER COEFFICIENT MULTIPLIER FROM THEORETICAL METHODS FOR CASE 2

$$hc = (Nf\ Af + Ao)\ \frac{ho}{Ai} = (C)\ ho$$

where

$Nf = .54$
$Af = 23.44\ in^2/in$
$Ai = 3.8646\ in^2/in$
$Ao = 5.3778\ in^2/in$
$ho = 20.18\ Btu/hr\text{-}ft^2\text{-}°F$
$hc = 94.18\ Btu/hr\text{-}ft^2\text{-}°F$
$C = 4.667$

The overall heat transfer coefficient, U, can be defined as ho multiplied by the overall heat transfer coefficient multiplier when neglecting the slight effects of the other terms defining U as shown in Table 5. Because hc is based on tube inside area, hc must be converted to be based on smooth tube outside area.

Hand Calculated $Ui = (C)\ ho$ - based on tube inside area
$U = C\ (.718)\ ho$ - based on smooth tube outside area
$= (4.667)\ (.718)\ ho$
$= (3.35)\ ho$

PEPSE Calculated U = [multiplier] ho - based on smooth tube outside area
= [2.03] ho

Shown in Table 6 is the hand calculation of the overall heat transfer coefficient multiplier for the 100% load of Case 2 from the theoretical approach outlined by Kern. The magnitude of the multiplier from the calculation for Case 2 is 3.35. This value is higher than the PEPSE-calculated value of 2.0, but this calculation provides a general guideline for the value of the multiplier that can be expected.

The theoretical calculation of extended-surface heat exchangers is complicated by the impact of the presence of the fins on the values of convective coefficients, by the presence of multiple finned tubes, and by the variation of the temperature of the fluid surrounding the fins. These complications make the theoretical calculation of actual fin performance very difficult. Reference 5 points out the dependence of existing methods on test results of the heat transfer characteristics for different extended-surface heat exchanger configurations, especially for flows in the laminar region.

Economizer Pressure Drops

The pressure drops in an economizer are not as important as in other boiler components because the flue gas properties are much more temperature dependent than pressure dependent. Also, the feedwater flow is usually subcooled by about 50°F for economizers (other than for steaming economizers). For typical changes in the pressure of the flue side and of the feedwater side, the effects on heat transfer are negligible.

The feedwater pressure drops for subcooled liquid with the tube velocities commonly found in economizers are largely due to elevation head. The friction pressure drop may be less than 1. psi, but the pressure drop for 30 feet of elevation may be greater than 12. psi.

CONCLUSION

The results of this study indicate that an overall heat transfer coefficient multiplier (essentially constant for the 50% to 100% load range) can be used to predict economizer performance. Finned tube economizers may be modeled by using the PEPSE heat balance program when design performance for at least one load case above 50% is known. When design performance data for a load below 50% load are available, load cases between the lowest design load case and 100% load may be accurately represented. The general magnitude of the overall heat transfer multiplier is in agreement with the approximate value of the multiplier obtained from a theoretical approach using a weighted fin efficiency for the finned tube heat exchangers. The value of the outer convective film heat transfer coefficient along with the overall heat transfer coefficient multiplier dictate the value of the overall heat transfer coefficient.

REFERENCES

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2. Kettenacker, W. C., Minner, G. L., Hansen, E. J., and Klink, P. H., PEPSE Volume II: Engineering Model Description, Revision 4, 1985.

3. Kern, D. Q., Process Heat Transfer, McGraw-Hill Book Company, New York, 1950.

4. Gardner, K. A., Efficiency of Extended Surfaces, Trans. ASME, Vol. 67, pp. 621-631, 1945.

5. Johnson, B. M., Kreid, D. K., and Hauser, S. G., A Method of Comparing Performance of Extended-Surface Heat Exchangers, Heat Transfer Engineering, Vol. 4, No. 1, pp. 32-42, January-March 1983.

APPENDIX A

INPUT SPECIFICATIONS AND PEPSE-CALCULATED RESULTS FOR CASE 1

LISTING OF INPUT DATA FOR CASE 1

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 ECONOMIZER SUBMODEL CASE 1 - 100% LOAD
*DRUM-TYPE FOSSIL BOILER
*
*CALCULATE FLUE GAS ECONOMIZER INLET TEMPERATURE
*AND HEAT TRANSFER MULTIPLIER
*
010200 0
*
*GEOMETRY
500150 10 U 20 T
500250 20 T 30 I
500450 40 U 20 S
500550 20 D 60 I
*
*STEAM IN
700100 33 490. 2815. 2594014.
*
*FLUE IN
700400 31 1040.   14.7  3240000.
700403 CO2, .20373, H2O, .05112, SO2, .00361
700404 O2, .03464, N2 .69112, C, .00022, ASH .01556
*
700600 30
700300 32
*
*CONVECTIVE STAGE ECONOMIZER
*
*            3-WGUESS 4-PGUESS 5-TGUESS
700200 28 1 2594014.    2815.    490.
*
*        1-NNFUEL  2-NCONV  3-NFLG  4-XHT
700204    3          0        2     30.
*
*                                    *AVE*TUBES
*       5-XWID      6-RHLL    7-NRW    8-NTFR
700205   40.         30.        9.       156.
*
*      9-NPASS   10-DDTIRH  11-DDTORH   12-PTCHLN
700206  9.            1.88        2.0           3.5
*
*
*           14-ZTIN     16-EMISS     18-FOLL     20-FRMLS   22-AATIRH
*      13-PTCHTR    15-ZTOUT    17-CKTIRH  19-EDDTUB  21-HTTIRH   23-TOP
700207 3.5  0.      0.  .2      25.  0.     0.   0.     -3.8
*
*FIND PROPERTIES FOR HAND CALC
880070 PP,55,GPTC,TT,55,DPVB,11
880080 PP,45,GPTC,TT,45,DPVB,12
880100 PP,15,GPTM,TT,15,DPVB,20
880110 PP,25,GPTM,TT,25,DPVB,21
880120 PP,45,GPTM,TT,45,DPVB,22
880130 PP,55,GPTM,TT,55,DPVB,23
880140 PP,15,GPTK,TT,15,DPVB,24
880150 PP,25,GPTK,TT,25,DPVB,25
880160 PP,45,GPTK,TT,45,DPVB,26
880170 PP,55,GPTK,TT,55,DPVB,27
*
*FIND HEAT TRANSFER MULTIPLIER
840100 HTTIRH, 20.645.,0.,1.,TT,55
*
*FIND INLET TEMP OF FLUE GAS BASED ON HEAT DUTY
840200 TTVSC,40,362985280.,1.0E-6,1.0,BDMXFL,20
.
```

OPERATION SET VALUES CALCULATED
AT THE START OF ITERATION 20

SET	VARIABLE (ID) VALUE	OPERATION	VARIABLE (ID) VALUE	=	VARIABLE (ID) VALUE
7	PP (55) 1.45977E+01	GPTC	TT (55) 6.44693E+02	=	OFVB (11) 2.70108E-01
8	PP (45) 1.47000E+01	GPTC	TT (45) 1.04721E+03	=	OFVB (12) 2.86415E-01
10	PP (15) 2.81500E+03	GPTM	TT (15) 4.90000E+02	=	OFVB (20) 2.52818E-01
11	PP (25) 2.81042E+03	GPTM	TT (25) 6.03658E+02	=	OFVB (21) 2.02839E-01
12	PP (45) 1.47000E+01	GPTM	TT (45) 1.04721E+03	=	OFVB (22) 8.67685E-02
13	PP (55) 1.45977E+01	GPTM	TT (55) 6.44693E+02	=	OFVB (23) 6.93315E-02
14	PP (15) 2.81500E+03	GPTK	TT (15) 4.90000E+02	=	OFVB (24) 3.66062E-01
15	PP (25) 2.81042E+03	GPTK	TT (25) 6.03658E+02	=	OFVB (25) 3.09443E-01
16	PP (45) 1.47000E+01	GPTK	TT (45) 1.04721E+03	=	OFVB (26) 3.07075E-02
17	PP (55) 1.45977E+01	GPTK	TT (55) 6.44693E+02	=	OFVB (27) 2.41099E-02

PEPSE CODE BY ENERGY INCORPORATED, IDAHO FALLS, ID. VERSION 51L CREATED 23 DEC 85 DATE 05/05/86.
ECONOMIZER SUBMODEL CASE 1 - 100% LOAD

CONTROLLED VARIABLE VALUES CALCULATED

CONTROL SET	Y VALUE FROM ITERATE 20	X VALUE USED FOR ITERATE 20	CONVERGENCE?
1	6.44693E+02	-3.71718E+00	YES
2	3.62986E+08	1.04721E+03	YES

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ECONOMIZER SUBMODEL CASE 1 - 100% LOAD

COMPONENT PROPERTIES

COMPONENT NUMBER	COMPONENT DESCRIPTION	CONNECTING STREAM NUMBER	PORT ID	FLUID ID	MASS FLOW (LBM/HR)	TEMPERATURE (F)	PRESSURE (PSI)	THERMO. QUALITY (-)	ENTHALPY (BTU/LBM)	ENTROPY (BTU/LBM-F)	SPECIFIC VOLUME (FT3/LBM)
10	INPUT COMPONENT	15	U (OUTPUT)	0	2594014.	490.000	2815.000	-1.05608	476.1801	0.66840	1.9785E-02
20	CONVECTIVE STAGE	15	T (INPUT)	0	2594014.	490.000	2815.000	-1.05608	476.1801	0.66840	1.9785E-02
		45	S (INPUT)	2	3240000.	1047.209	14.700	N.A.	429.3120	1.82570	7.1293E+01
		25	T (OUTPUT)	0	2594014.	603.658	2810.418	-0.55293	616.1121	0.80737	2.2992E-02
		55	D (OUTPUT)	2	3240000.	644.693	14.598	N.A.	317.2795	1.73975	5.2612E+01
30	OUTPUT COMPONENT	25	I (INPUT)	0	2594014.	603.658	2810.418	-0.55293	616.1121	0.80737	2.2992E-02
40	INFINITE SOURCE	45	U (OUTPUT)	2	3240000.	1047.209	14.700	N.A.	429.3120	1.82570	7.1293E+01
60	INFINITE SINK	55	I (INPUT)	2	3240000.	644.693	14.598	N.A.	317.2795	1.73975	5.2612E+01

PEPSE CODE BY ENERGY INCORPORATED, IDAHC FALLS, ID. VERSION 51L CREATED 23 DEC 85 DATE 05/05/86.
ECONOMIZER SUBMODEL CASE 1 - 100% LOAD

CETAILED HEAT EXCHANGER PERFORMANCE OUTPUT

COMPONENT NUMBER	COMPONENT DESCRIPTION	HEAT EXCH SHELL LOSSES (BTU/HR)	HEAT TO TUBE SIDE FLUID (BTU/HR)	HEAT EXCH EFFECTIVENESS (-)	PRESSURE CROP TUBE (PSI)	SHELL (PSI)
20	CONVECTIVE STAGE	0.0000	3.62986E+08	0.7317	4.582	C.102

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ECONOMIZER SUBMODEL CASE 1 - 100% LOAD

DETAILED HEAT EXCHANGER DESIGN OUTPUT

COMPONENT NUMBER	COMPONENT DESCRIPTION	HEAT TRANS. COEFFICIENT (BTU/HR-FT2-F)	EFFECTIVE HEAT TRANS. AREA (FT2)
20	CONVECTIVE STAGE	60.231	22053.98