4. Detailed review of design report

Section 4 includes discussions that are specifically applicable to a single page or a few pages of SAI's design report. Some topics, of a more general nature, were previously discussed in Section 3, which complements Section 4.

Page 3

The report should be clarified to reflect the intent that 200-600 psig is the pressure range of most interest. Operating pressures as low as 14.7 psia and as high as 1000 psig are desired for flexibility; however, instrument accuracy can be sacrificed at the extremes.

Page 5

The SAI report specifies a range of residence times from 5 to 10 sec in SAI Table 3.1, but subsequently calls for residence times of 3 and 12 sec for test IID. The report should make clear the intent that SAI's Table 3.1 should adhere to.

The term "adiabatic" should be clarified to reflect the intent that heat is to be supplied to the reactor vessel walls so the vessel wall temperature will match the reaction mixture temperature in order to prevent heat loss from the reaction mixture. For more flexibility to achieve this condition, a multizone heater and control system is recommended.

The stated method of sampling the solids in batches on completion of the runs seems acceptable; however, it may be found, when the system is operational, that solids samples are desired during the runs. It is recommended that provisions be made for additional remote solid sampling as a potential add-on feature. For example, two valves in series at the bottom of the char pot could be operated remotely to remove samples during runs. The additional cost may not be justified at this time.

The coal throughput rate of 10 lb/hr is too low for the reactor as specified in the report: 15 lb/hr would be more appropriate. Based on the possible new larger reactor size, the throughput would be in the range from 1.8 to 24 lb/hr.

Page 6

The gas supply systems should be revised to reflect the higher gas flow rates if the larger reactor is used (See revised Table 3.2 for maximum flows).

Page 7

The coal feed storage volume (0.8 ft^3) is low based on the original design. It should have been about 1.2 ft^3, but it should be increased to about 2.0 ft^3 to handle the larger reactor size.

Page 8

A more meaningful basis than temperature (2100°F) for determining actual ft^3/hr (ACFH) is the volume of the reactor and the gas residence time.

Nitrogen should replace one of the hydrogens in "Gas Composition."

For the new reactor volume, the flow rates should be 160-1200 SCFH and 70-141 ACFH.

REVISED Table 3.2 - DESIGN CRITERIA GAS SUPPLY SYSTEM

Component	Purity (%)	System Pressure (psig)	Temperature+ (°F)	Maximum Flow (SCFH)	Capacity** (SCF)
He	99.995 (High Purity)†	1100	50	950	7600
N_2	99.998 (Prepurified)†	1100	50	1200 (800)***	9600 (6400)***
CO	99.5 (C.P.)†	700	50	130 (80)***	1040 (640)***
CO_2	99.8 (Bone Dry)†	700	80	145 (75)***	1160 (600)***
H_2	99.95 (Prepurified)†	1100	50	1200 (800)***	9600 (6400)***
Steam	Demineralized	1100	600	1200 (800)***	9600 (6400)***

+Prior to Preheater which raises the nonaqueous gases to 600°F.

†Designation of the Matheson Company.

**Based upon an 8-hr total run time.

***Original in parentheses.

Page 9

The operating temperature of the reactor should be specified over approximate ranges as follows:

Inlet: 2200-3000°F, and

outlet: 1600 to 2300°F.

Page 10

Several ranges of coal analysis were found in the literature: It appears that the solids loading could be 0.3-2 lb/50 SCF of gas. The gas flow rate should be approximately 200-1700 SCFH (25-600 ACFH at 2100°F, 200 to 600 psig) for the new reactor size. The coal feed rate should be 1.8 to 24 lb/hr and the condensate loading should be 55 lb/hr.

Page 11

The pressure criteria should show a design pressure of 1100 psig and an operating pressure of 200-600 psig to be consistent with the rest of the criteria. For the new reactor size, the liquid volume storage size should be around 3.0 ft^3 to hold a 3-hr run at 55 lb/hr of H_2O. The solids volume should be approximately 6.0 ft^3 for 57 lb of char and ash from 3 hr of operation (19 lb/hr and a density allowance of 10 lb/ft^3).

Page 15

It would help clarify "operation at 1800°F and 300 psig" by stating that steam injection will be used only when the reactor pressure is less than 300 psig.

Creep could cause the joints and seals to develop leaks after the reactor is used at high temperature over a period of time. (Reference MRC comments on SAI's page B-15). This would be particularly true with use of 304 or 316 stainless steel in the lower reactor section.

The steam supply subsystem as shown in the P&I diagram indicates measurement of the steam pressure and temperature at two different locations. The same location should be used for both in order to determine the mass flow rate of the steam.

Page 16

The proposed rotary star feeder and other potential feed mechanisms were evaluated for this application. Vibrating feeders, fluid bed feeders, and screw feeders were considered. The design may require several interchangeable star wheels to minimize pulse feeding at low feed rates. A vibrating system on the feed hopper and feeder may be helpful. A fluid-bed coal feeder would not have a wide feed-rate range and the carrier gas could interfere with operation of the reactor. A screw type coal feeder may be acceptable; however, pressure fluctuations between the reactor and the pressurized feed hopper may cause problems since the screw feeder does not provide positive pressure isolation. Calibration provisions are unclear. It was concluded that the proposed rotary star feeder, coupled with modifications based on METC's experience, is the best approach.

The P&I diagrams show three temperature sensors for the gas heater (303, 304, and 305), but only one temperature controller. The intent of this arrangement is not clear. As discussed in response to the heat transfer calculations on SAI's page B-12, more flexibility could be obtained by independently controlling multiple heating zones.

Page 19

The dimensions given for the heater tube do not agree with the dimensions used in the calculations on SAI's page B-12 which were based on a 1/2-in. square channel, 20.2 ft long.

The initial gas temperature entering the heater is stated elsewhere in the report as 600°F, not 50°F.

The maximum temperature of the ceramic will exceed 3000°F in order to heat the gas to 3000°F. This is discussed further in response to the heat transfer calculations on SAI's page B-12.

During experiments using the steam injection probe, care must be taken to ensure that the ash will be solidified before passing through the throat of the reactor where it will be cooled by the steam. Otherwise it could accumulate on the throat surface and plug the reactor.

Pages 20 to 24

The specific design has been in a state of development during the time this review is being done. A recent design has a larger reactor section (3 in. i.d. and

4 ft long) than the désigns described in the report and has minimized or eliminated many of the concerns regarding the reactor designs in the report.

The purpose of the taper at the bottom of the reactor is unclear. It appears to be an unnecessary restriction of the flow and potentially a place where plugging could occur. It is recommended that its purpose be reviewed.

Since it is important to be able to replace parts easily, such as the ceramic tube, it is recommended that the steps required to do this be reviewed with the manufacturer during the detailed design phase.

Even though the most recent design has minimized the concern about leakage through the slip joints, it is recommended that consideration be given to specifying a minimum acceptable leak rate and appropriate testing procedures for acceptance from the vendor.

The hole configuration for entry of the gas and the subsequent mixing of the gas and coal can best be dealt with experimentally as suggested in the SAI report. A 30° angle from vertical and a slight radial angle to impart a swirl are recommended for testing. It may be necessary to provide different hole configurations for widely varying flow rates.

The heater and reactor tube wall thicknesses are significantly different. At the heater/reactor joint, excessive stresses may develop in the heater tube as a result of joining irregular cross sections. These stresses may reduce the heater tube life.

If the horizontal heater tube is not properly supported, sagging will result after prolonged use.

A thorough analysis should be made of the deformations resulting from thermal expansion of the reactor. For example, the varying expansion rates of the outer jacket, because of its attachment to the hot char cooler, may cause deformation at the mounts and connections.

It is recommended that dual thermocouples be specified, particularly for the critical temperatures inside the reactor. The cost is small compared to the additional reliability.

Vessel wall thicknesses are discussed in our review of SAI's calculations on SAI pages B-15 through B-18.

Alternative designs for the joints which were reviewed and discussed with METC included flanges, slip joints, threaded joints, and ceramic cemented joints. The new reactor design reflects the use of several of these alternatives.

Page 24

There is some concern regarding the ability of the char cooler to cool the char and gases adequately for worst condition cases. Thus, the following preliminary analysis was considered:

Note Figure 4-1 for the cooler conditions and sizes.

The following assumptions were made in the analysis:

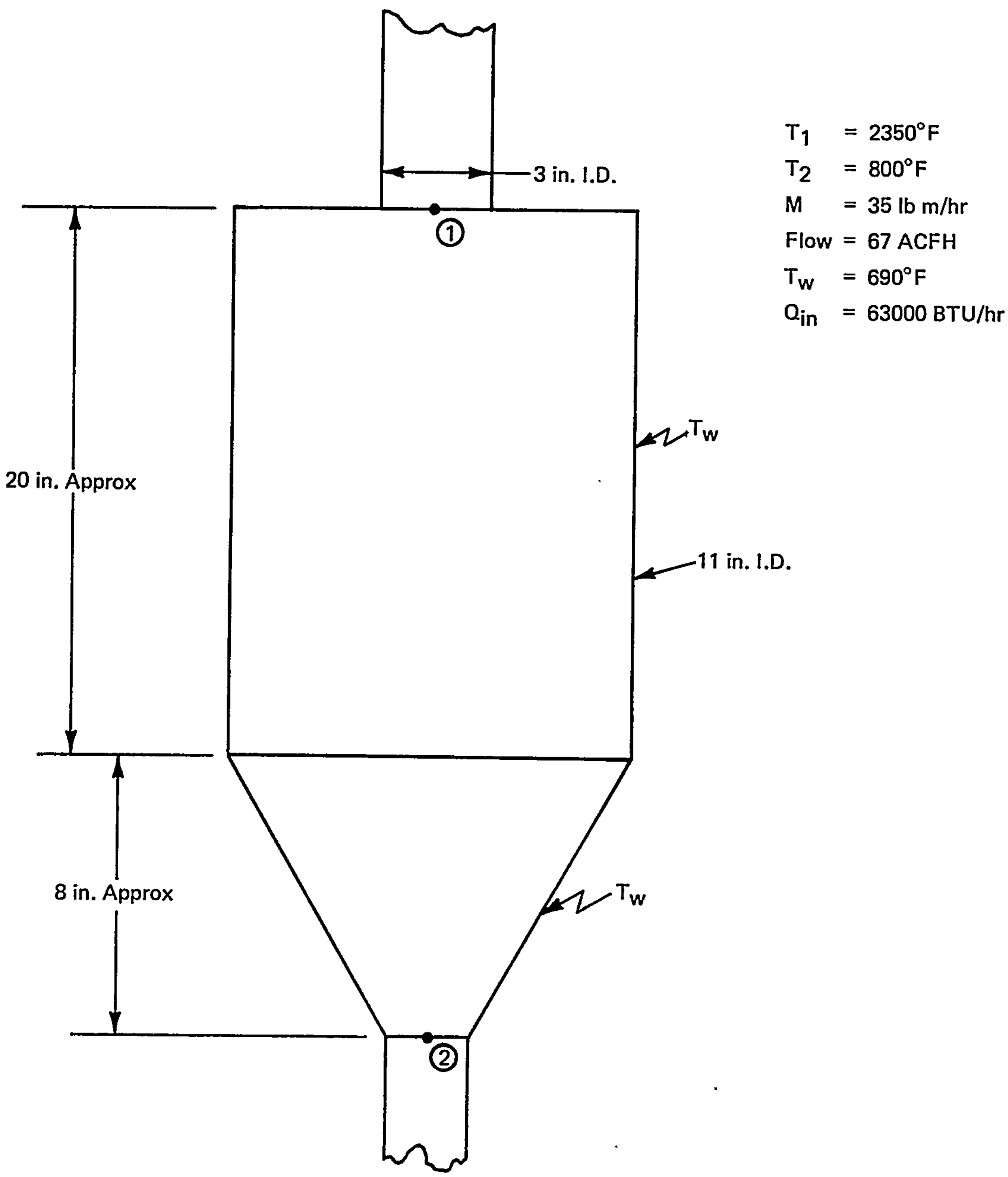

FIGURE 4-1 - Basis for char cooler heat transfer calculations.

1. Total surfaces and volumes are approximate.

2. The char cooler can be adequately modeled as a "fired heater."

3. The inner wall temperature is a constant 690°F.

4. All properties are evaluated at a mean temperature based on the outlet and inlet conditions (T_1 and T_2).

5. Gases such as He and H_2 are transparent and, thus, have no significant contribution to the overall radiative transfer.

6. The only gases that significantly contribute to the radiative transfer are CO_2 and H_2O (steam). The sum of their expected molal percents equals their additive partial pressures.

The following analysis follows closely a paper written by Norman Wimpress of C. F. Braun and Company which appeared in Chemical Engineering magazine (May 22, 1978).

For the total radiation and convective transfer inside a firebox or, in this case, the char cooler, the governing equation is,

$$q_T = AF\left[\sigma\left(T_g{}^4 - T_t{}^4\right) + 35\left(T_g - T_t\right)\right]$$

where

q_T = Total energy transfer including both convection and radiation, BTU/hr.

A = Plane area of tube heat exchanger. For the char cooler this will be the total internal surface area or

Total Surface Area (Approximate) = Cylinder + Cone + End = 945 in.^{2} or 6.56 ft^2

F = Radiant exchange factor, dimensionless.

σ = Stefan-Boltzmann constant,

$$0.173(10^{-8})\ \frac{\text{BTU}}{\text{hr ft}^2\ {}^\circ\text{R}^4}$$

T_g = Mean gas temperature, $T_g = \frac{T_1 + T_2}{2} + 460°$, °R

$T_t = T_w + 460°$, °R

Evaluating these terms for the char cooler gives

A

A = 6.56 ft^2

F

"F" is determined graphically from Wimpress's paper. Briefly, an effective length, "L", is determined by,

$$L = 3.6\ V/A$$

where A is the char cooler enclosed area, and V is the enclosed volume or Total Volume = Cylinder + Cone = 2192 in.^{3} or 1.27 ft^3. "L" is multiplied by the partial pressure of the CO_2 and H_2O summations ($\sim$0.05 x 69 atm). An emissivity, ε, is determined from a graph using the mean gas temperature.

With ε known, a second graph involving the area of the char cooler is consulted and the exchange factor, "F", is determined.

In this particular case, F equals 0.4.

σ

As noted.

$\underline{T_g}$

According to energy balance calculations, the worst case reactor outlet condition is

$$T_1 = 2350°F.$$

The char cooler outlet temperature was specified as

$$T_2 = 800°F.$$

Thus,

$$T_g = \frac{T_1 + T_2}{2} + 460° = 2035°R.$$

$\underline{T_t}$

$$T_t = T_w + 460 = 690 + 460 = 1150°R$$

All terms in equation (1) have been specified; thus, q_T can be evaluated.

$$q_T = 6.56(.4)\left[.173(10^{-8})\ (2035^4 - 1150^4) + 35(2035 - 1150)\right]$$

$$\boxed{q_T = 150{,}000 \text{ BTU/hr}}$$

Compared to a maximum amount of energy to be removed of 63,000 BTU/hr, the heat exchanger calculations indicate an oversizing by a factor of approximately 2. With the aforementioned assumptions, particularly the constant wall temperature, this sizing would be appropriate. Finally, these calculations are preliminary and further analysis is advisable.

Page 27

There is some concern regarding potential plugging of the sintered metal filter. METC's intent to clean the filter between runs will alleviate this concern.

The symbol $\triangle$P on the char pot outlet is unclear.

Page 28

There is some concern regarding potential plugging of the sample filter.

Pages 30 through 32

The assumptions that the gaseous products would be similar to the devolatilized gases given in SAI Tables 5.1 and 5.2 and that the tars and oils would gasify according to the equation on page 32 are inconsistent with experimental data listed in SAI's reference 2 and the equilibrium constants for the higher temperatures. The amount of CH_4 is much too high. Also, the assumption that 10% of the fixed carbon reacted with water seems low for some of the experiments.

Pages 35 through 37

Tables 6.1 and 6.2 in SAI's report should be revised to reflect METC guidance to limit the calculations to the pressure range of 200 - 600 psig, to gas residence time of 5 to 10 sec, and the actual reactor size.

Page 38

The location of stream #11 is not shown on Figure 6-1 of SAI's report.

Page 39

The system heat and material balance in general appears consistent based on the assumptions on pages 30, 31, and 32 of SAI's report and the smaller reactor size as listed on page 19. The only stream that appears inconsistent is #14, but the basis could not be checked since SAI did not include any calculations for this stream.

Pages 40 through 44

These heat and material balances were not checked in detail since they were based on the smaller reactor size which results in smaller gas and coal flows, and higher pressures than are now anticipated. Also the methane content seems higher than possible.

Page 101

The cooling water total requirement cannot be evaluated until the final design is made of the heater/reactor. See the revised SAI Table 3.2 in this report for the new maximum gas flow rates based on the new larger reactor size.

Page 201

The amount of heat to be exchanged in the high pressure gas preheater, H-101, has increased from 9400 to 12,600 BTU/hr because of the larger gas flow in the larger reactor. To match the heat transfer, the Dowtherm 'G' flow rate should have a corresponding increase from 655 to 1600 lb/hr. The heat exchanger still appears to be adequate to handle the additional load.

Page 202

The heat load for this exchanger condenser H-501, would increase from 49,800 to about 57,400 BTU/hr because of the larger reactor. The large load occurs with H_2O but the overall transfer should be divided into three sections, super heated steam to water, condensing steam to water, and water to water. The heat exchanger, as sized, is marginal without any allowance for fouling by organic films. A larger transfer area should be specified or two exchangers should be used.

Page 203

The heat load for the condenser H-602 should be around 3100 rather than 625 BTU/hr. The specified heat exchanger should be able to handle the load.

Page 206

A check cannot be made of this cooling water heat exchanger, H-805, until the heater/reactor design is completed.

Page 300

At this time it is difficult to predict what the exact pressure drops, pipe lengths, and tubing configurations will be in order to predict total dynamic head for pump sizing. SAI's parameters predicted for the various pumps in Appendix B, page B-14, look reasonable for the system shown. The required horsepower equations appear mathematically correct and should be adequate for the system. The capacity for the P-101 steam generator water pump should be increased from 0.1 to 0.11 gpm. The capacity for the Dowtherm pump should be increased to minimize the high film temperature problem in the char cooler.

Pages 500 through 538

The flexibility of the instrumentation for this project could be increased by the use of dual thermocouples eliminating unnecessary shutdowns and by the use of data loggers. A digital data logger which can be programmed for increased flexibility has the capacity to easily monitor one hundred temperature points and could replace many three-pen recorders for a cost savings. Since the system is heavily instrumented with top-of-the-line equipment, cost saving potentials also exist by reducing the quantity of instruments and possibly the quality of the instruments after the accuracy requirements on the process have been defined.

Conceptually, the P&I diagrams provide sufficient information as to what kinds of instrumentation are being proposed, but METC is making major changes which have not been evaluated. The two major areas of concern on instrumentation are the need to further define the safety interlocking system and the need to define what valves require remote operation to eliminate the need to enter the pressure cell during operation.

In some cases the instrumentation specifications are too specific: They appear to be rigid specifications from a particular manufacturer, e.g., multipoint recorders. In other cases the specifications are too general. The instrumentation specifications should be reviewed in greater detail after the process accuracy requirements are defined and completion of the P&I diagram.

<u>Page 602</u>

The coal feeder should be sized to handle the flow rate required by the actual reactor size (range of from 1.8 to 24 lb/hr for the 3 in. i.d. x 4 ft long reactor). METC may want to consider a non-electrical drive rather than E.P. electrical.

<u>Pages 604 and 605</u>

The coal hopper should be increased in size to about 2.0 ft^3 to handle the maximum flow rate for 3 hr if the larger reactor is used. A cone shaped bottom would probably feed better.

<u>Page 607</u>

The use of Hastelloy X for the transition tube from the reactor to the char cooler will cause a problem. The design temperatures are in the range of 2200 to 2300°F and Hastelloy X will not take that high a temperature (melting point 2300-2470°F).

<u>Pages 608 and 609</u>

The char pot should be increased in size to handle the additional volume if the larger reactor is used. Estimates indicate a worst condition of about 5.3 ft^3 based on a density of 10 lb/ft^3 for the char and ash.

<u>Page 611</u>

The process line between condenser H-501 and vessel V-505 is 1/2 in. o.d. tube but V-505 has a 1 in. pipe nozzle. Such a large reduction in size requires more fittings than necessary.

<u>Page 701</u>

The steam generator B-101 should be sized to generate at least 55 lb/hr of steam. It would be much cheaper and safer to locate the generator outside the cell rather than use E.P. electrical per code.

Pages 702 through 704

The specifications for the low pressure superheater for the steam could not be checked since steam injection requirements were not established.

Pages 906 through 911

These specifications for the gas heater/reactor assembly, M301/401, should be revised to agree with capacity and sizes associated with the actual reactor size.

Pages 1100 through 1104

There are no specific comments on the P&I diagrams since MRC plans to provide feedback to METC as a follow-up to this report after the revised P&I diagrams are received from METC.

Page B-1

SAI used a different gas composition for the feed gas than that listed on page 11 of the project proposal (SAI Reference 2). The comparison follows:

	SAI	Reference 2
N_2	57.7	56
H_2	4.8	7
H_2O	14.7	13
CO	12.2	16
CO_2	10.6	8

The major impact is the 16% requirement of CO to feed the system, the other changes have no significant impact. The gas flow rate in SCFH is essentially correct for the 2 3/4 in. i.d. x 3 ft reactor size in SAI's report. This needs to be corrected for the actual reactor size as mentioned earlier in the comments.

Page B-2

The friction factor, f, used in the friction pressure drop equation is too high (0.031 vs. 0.008). The pressure drop for the tube listed should be 0.05 psi instead of 0.071 psi. The 3/8 in. i.d. tube size does not agree with the heat exchange surface calculation for the size on SAI's B-13.

Page B-3

There appear to be several errors in the calculations of the heat and energy balances. The amount of coal (carbon and ash) in stream #12 should be 3.935 lb and the amount of gas should be 35 lb (see SAI's page B-5). The specific heat equation for coal is wrong. Handbooks indicate a range of 0.26 to 0.37 BTU/lb °F for coal. If this method is used, the approximate temperature for stream #12 would be around 2250°F.

Pages B-4, 5, and 6

These calculations of the heat and material balance appear to be correct based on the assumptions listed on SAI's pages 30, 31, and 32. As stated earlier in Section 4 the basis for these assumptions is questionable.

Page B-7

It is not clear what pressure drop (ΔP) is being calculated.

Page B-9

The equation used to calculate terminal velocity from Stokes's Law is not valid for Re >0.1 per Reference 3 page 60. This is not a major problem since the friction factor equation gives only a 30% higher number (0.17 vs. 0.13 ft/sec).

Page B-10

SAI's calculations were checked based on the physical property data and the equations presented. The Nusselt number, coal heat capacity, and dimensionless temperature calculations are in need of clarification; however, the overall conclusion resulting from the calculations may not change.

It is not clear why the Nusselt number calculation is based on the entrained particle being at the terminal velocity of a falling particle: It is doubtful that the terminal velocity assumption applies during the brief time involved. SAI calculated Nu = 2.24 on this basis, whereas a value of Nu = 2.0 would be obtained if the gas and the particle were assumed to be at the same velocity.

The heat capacity of coal, C, listed by SAI to be 1.05 BTU/lb°F could not be calculated from the equation given on Page B-10. Perry's Chemical Engineer's Handbook (Page 3-136) gives the range of C for coal as 0.26 to 0.37 BTU/lb °F.

It is not clear how SAI obtained a value of $(T-T_\infty)/T_O-T_\infty) = 0.05$ or why the symbol T_1 was used rather than T_O as given in SAI's Reference 7. If values of T = 2030°F, T_∞ = 3000°F, and T_O = 50°F are substituted into the equation, the result is (2030-3000)/(50-3000) = 0.329.

The above concerns do not seem to alter the apparent conclusion drawn by SAI that the coal particles can be heated quickly enough. If the calculations are modified using the above value of $(T-T_\infty)/(T_O-T_\infty) = 0.329$, the resulting value of θ = 0.013 sec, which is less than the θ = 0.035 sec obtained by SAI. It is recommended that these calculations be again reviewed for the larger particle sizes based on the final reactor size and residence time.

Page B-12

The equations and numerical results presented on pages B-12 and B-13 for heat transfer from the ceramic spiral channel to the steam were reviewed, and it was found that some of the numerical results could not be reproduced. An independent calculation was also performed.

Apparently two of the equations were miscopied: The sixth equation should be

$$h_i^* = \frac{Nu^* k_f}{4d^*}$$

and the seventh equation should be

$$\frac{1}{U^*} = \frac{1}{h_i^*} + \frac{D_i \ln\left(\frac{D_o}{D_i}\right)}{2k_s}$$

It was assumed that the steam properties given were at 3000°F and 1000 psig and a value for the density was determined at these conditions since none was given. The results compare as follows:

	SAI	Review
$q, \frac{BTU}{hr}$	57,500	57,500
$U^*, \frac{BTU}{hr\ ft^2\ °F}$	590	60
ΔT_{ln}, °F	45	45
Area, ft^2	2.16	21
L_{Helix}, ft	20.2	127
ℓ , ft	1.1	17

Also, the calculations were repeated using property values from THERMAL (a commercially available heat transfer computer code) at 1500°F: A value of U* = 53 resulted.

Thus, SAI calculations are numerically incorrect, although it is difficult to determine exactly where since the detailed sample calculations were not provided.

An independent approach was taken to evaluate the maximum temperature of the ceramic wall. SAI's approach involved assuming the maximum temperature to be 3001°F. This provides conservatism in evaluating the length, but does not provide an assessment of the maximum temperature.

THERMAL was used to make the calculations for pure helium, nitrogen, hydrogen, steam, carbon monoxide, and carbon dioxide at the following conditions:

Volumetric Flow	= 800 SCFH
Pressure	= 1000 psig
Inlet gas temperature	= 600°F
Outlet gas temperature	= 3000°F
Channel dimensions	= 0.0147 ft x 0.0147 ft x 20.2 ft long

The results obtained are as follows:

	Q (BTU/hr)	Inside Surface Temperature of Ceramic Channel: At Inlet (°F)	Inside Surface Temperature of Ceramic Channel: At Outlet (°F)
Helium	25,000	830	3190
Nitrogen	40,000	1000	3280
Hydrogen	36,000	700	3080
Steam	53,000	900	3330
Carbon Monoxide	40,000	1000	3410
Carbon Dioxide	66,000	1070	3350

The statement of the problem, analysis, and results for the steam case are as shown in the computer printout on the following pages.

The gas stream compositions summarized in Table 6.2, page 37 of the SAI report were then used to estimate the ceramic surface temperature at the outlet for the mixtures. It was assumed that the ceramic temperature increase above the bulk fluid temperature necessitated by each gas was proportional to the concentration of the gas and that the total volumetric flow in all cases was 800 SCFH. The results are as follows:

Case	Ceramic Inside Surface Temperature at Outlet (°F)
II A	3300
II D	3290
III B	3080
III D	3330 (pure steam)
III F	3210
IV B	3080

The hottest temperature calculated was for Case III D which is pure steam. The temperature increase across a 1/4-in. ceramic wall was estimated to be 30°F so that the maximum calculated ceramic temperature is 3360°F.

Mark's Handbook (7th Edition, page 6-171) states that silicone carbide decomposes above 4060°F. Manufacturers' recommended maximum temperatures are substantially less than this.

Uncertainties regarding the heat transfer calculations exist as follows:

STATEMENT OF PROBLEM
****

CALCULATE THE FORCED CONVECTION HEAT FLOW Q FROM THE LENGTH OF RECTANGULAR DUCT BETWEEN SECTIONS 1 AND 2. THE FLOW AND THERMAL CONDITIONS ARE:

. TURBULENT FLOW
. UNIFORM HEAT FLUX
. FULLY DEVELOPED VELOCITY PROFILE AT SECTION 1
. FULLY DEVELOPED TEMPERATURE PROFILE AT SECTION 1

THE GIVEN CONDITIONS ARE:

DUCT LENGTH L, FT	=	.2020E+02		
DUCT WIDTH W, FT	=	.4170E-01		
DUCT HEIGHT B, FT	=	.4170E-01		
FLUID TEMPERATURE TB1, F	=	.6000E+03		
MASS FLOW RATE M, LBM/HR	=	.3720E+02		
HEAT FLUX Q/A, BTU/HR-SQ FT	=	.1563E+05	TO	.1564E+05

ANALYSIS
****

HEAT FLOW Q FROM A DUCT OF LENGTH L TO THE FLUID FLOWING IN THE DUCT MAY BE EXPRESSED BY

$$Q = H\,A(TS - TB) \quad , A = 2(W + B)L \qquad (1)$$

HOWEVER, EQN. (1) IS OF LIMITED VALUE FOR THE FOLLOWING REASONS:

. THE SURFACE TEMPERATURE TS VARIES ALONG THE DUCT

. THE FLUID BULK TEMPERATURE TB VARIES ALONG THE DUCT DUE TO HEATING OR COOLING

. THE HEAT TRANSFER COEFFICIENT H WILL VARY DUE TO ITS DEPENDENCE ON FLUID PROPERTY VALUES WHICH DEPEND ON TB

. H WILL VARY IN REGIONS OF DEVELOPING VELOCITY AND/OR TEMPERATURE PROFILES

IT IS NECESSARY, THEREFORE, TO USE THE DIFFERENTIAL FORM OF EQN. (1)

$$DQ = H(TS - TB)DA \qquad (2)$$

IN THIS ANALYSIS THE HEAT FLUX DQ/DA IS SPECIFIED. SINCE H MAY DEPEND ON TB THROUGH TEMPERATURE DEPENDENT PROPERTY VALUES, THE VARIATION OF TB ALONG THE DUCT IS NEEDED. THIS IS OBTAINED BY NOTING THAT THE HEAT TRANSFERRED TO THE FLUID RESULTS IN A RISE IN THE BULK TEMPERATURE,

$$DQ = M\ SPH\ DTB \quad , M = \text{MASS FLOW RATE} \qquad (3)$$

EQNS. (2) AND (3) ARE INTEGRATED NUMERICALLY TO OBTAIN THE SURFACE AND BULK TEMPERATURES.

THE FORCED CONVECTION HEAT TRANSFER COEFFICIENT IS OBTAINED FROM THE FOLLOWING CORRELATION.

REF: W. M. ROHSENOW AND J. P. HARTNETT, HANDBOOK OF HEAT TRANSFER, SECT. 7, P. 33, MC GRAW-HILL(1973)

10,000<RE
0.7<PR
REFT = TB AT LOCATION X

$$NU = 0.022\ RE^{0.8}\ PR^{0.6}$$

THE DIMENSIONLESS PARAMETERS ARE DEFINED AS FOLLOWS,

$$PR = VIS\ SPH/CON$$

$$RE = (M/AC)(4\ W\ B/2(W + B))/VIS$$

$$NU = H(4\ W\ B/2(W + B))/CON$$

RESULTS

IN THE CALCULATIONS FOR THE FOLLOWING BLOCK OF RESULTS THE REFERENCE TEMPERATURE FOR THE INDICATED MATERIAL WAS ABOVE OR BELOW THE TEMPERATURE RANGE FOR THE INDICATED PROPERTY. THE PROPERTY VALUE AT THE UPPER OR LOWER LIMIT WAS USED IN THE CALCULATIONS.

OUTPUT COLUMN	MATERIAL NUMBER	PROPERTY	TEMPERATURE LIMITS, F LOWER	UPPER
1	1	DENSITY	.5463E+03	.1500E+04
1	1	SP HEAT	.5463E+03	.1500E+04
1	1	TH COND	.5463E+03	.1500E+04
1	1	DYN VISC	.5463E+03	.1500E+04
2	1	DENSITY	.5463E+03	.1500E+04
2	1	SP HEAT	.5463E+03	.1500E+04
2	1	TH COND	.5463E+03	.1500E+04
2	1	DYN VISC	.5463E+03	.1500E+04
3	1	DENSITY	.5463E+03	.1500E+04
3	1	SP HEAT	.5463E+03	.1500E+04
3	1	TH COND	.5463E+03	.1500E+04
3	1	DYN VISC	.5463E+03	.1500E+04
4	1	DENSITY	.5463E+03	.1500E+04
4	1	SP HEAT	.5463E+03	.1500E+04
4	1	TH COND	.5463E+03	.1500E+04
4	1	DYN VISC	.5463E+03	.1500E+04

MATERIAL PROPERTIES	Case 1	Case 2	Case 3	Case 4*
PRESSURE, ATM	.6900E+02	.6900E+02	.6900E+02	.6900E+02
DENSITY, LBM/CU FT	.8797E+00	.8797E+00	.8797E+00	.8797E+00
SP HEAT, BTU/LBM-F	.5805E+00	.5805E+00	.5805E+00	.5805E+00
TH COND, BTU/HR-FT-F	.6720E-01	.6720E-01	.6720E-01	.6720E-01
DYN VISC, LBM/FT-HR	.1010E+00	.1010E+00	.1010E+00	.1010E+00
RANGED INPUT PARAMETERS				
Q/A, BTU/HR-SQ FT	.1563E+05	.1563E+05	.1564E+05	.1564E+05
OUTPUT PARAMETERS				
D, FT	.4170E-01	.4170E-01	.4170E-01	.4170E-01
PR1	.1324E+01	.1324E+01	.1324E+01	.1324E+01
RE1	.1765E+05	.1765E+05	.1765E+05	.1765E+05
NU1	.6501E+02	.6501E+02	.6501E+02	.6501E+02
H1, BTU/HR-SQ FT-F	.5352E+02	.5352E+02	.5352E+02	.5352E+02
TS1, F	.8920E+03	.8920E+03	.8921E+03	.8922E+03
PR2	.8724E+00	.8724E+00	.8724E+00	.8724E+00
RE2	.8834E+04	.8834E+04	.8834E+04	.8834E+04
NU2	.2909E+02	.2909E+02	.2909E+02	.2909E+02
H2, BTU/HR-SQ FT-F	.4688E+02	.4688E+02	.4688E+02	.4688E+02
TS2, F	.3331E+04	.3332E+04	.3333E+04	.3334E+04
TB2, F	.2998E+04	.2998E+04	.2999E+04	.3000E+04
H, BTU/HR-SQ FT-F	.4603E+02	.4603E+02	.4603E+02	.4603E+02
Q, BTU/HR	.5265E+05	.5266E+05	.5268E+05	.5270E+05

*Case 4 results in a final gas temperature of 3000°F.

1. The amount of gas which will leak past the spiral channel was not estimated.

2. The additional heat transfer from the ceramic channel to the gas by radiation was not estimated.

3. The calculation model assumed a 1/2-in. square channel with a 1/4-in. thick wall. The proposed design, a spiral groove in a solid cylinder, is physically different.

4. The extent of hot spots is unknown.

5. The computer code did not contain fluid property data up to 3000°F in several cases. The steam fluid properties did not exceed 1500°F.

In conclusion, it appears that the SAI proposed design will provide sufficient heat transfer area to heat the gases to 3000°F, but concern exists regarding the temperature limitations for silicone carbide. A more conservative approach is available. It is recommended that additional preheater capacity be provided and that the heater section of the reactor be used only for high temperatures that would exceed the capabilities of the preheater. This would enable the reactor heater to operate with lower ceramic temperatures and/or provide some extra capacity. Detailed specifications for manufacture of the square channel should specify an ample radius to avoid stress concentration problems. The heat transfer calculations should again be reviewed when the heater design is firmer and the maximum gas flow rates have been firmly established for the actual reactor design.

Page B-13

SAI's approach and numerical values were used to check the calculation of T_4, the heating element temperature. The calculated value of T_4 was found to be 2940°F compared to SAI's value of 3518°F: Apparently, there is a numerical error in SAI's calculation. Also, SAI's assumption that

$$T_{ceramic} = \left[(T_2^{\,4} + T_3^{\,4})/2\right]^{1/4} = 2452°F,$$

leads to the erroneous conclusion that T_4 = 2940°F is adequate, but the gas cannot be heated to 3000°F if the heater temperature is only 2940°F.

In addition, THERMAL, a commercially available heat transfer computer code, was used to provide a second approach. For this approach, a value of 2.73 ft (rather than 1.1 ft stated in the report) was used for the heater length since this is the length corresponding to a 20.2 ft long spiral according to the last equation on page B-12. Also to be conservative, it was assumed that the entire outside surface temperature of the ceramic tube was at 3360°F, the maximum calculated temperature for 800 SCFH of pure steam (Case III D). The resulting value of T_4 was calculated to be 3440°F.

Since both values of T_4 calculated in this report were less than the value of 3518°F reported by SAI, any conclusions drawn by SAI regarding the acceptability of the heating element materials would still appear to be valid. It is recommended that the heater be designed with multiple independently monitored and controlled heating zones. This would allow the flexibility in providing a greater heat flux near the cooler inlet and a lesser heat flux near the hotter outlet to minimize the ceramic temperature at the outlet.

The statement of the problem, analysis, and results are as shown in the computer printout on the following pages.

Page B-14

At this time it is difficult to predict what the exact pressure drops, pipe lengths, and tubing configurations will be in order to predict total dynamic head for pump sizing. SAI's parameters predicted for the various pumps look reasonable for the system shown. The required horsepower equations appear mathematically correct and should be adequate for the system. The capacity for the P-101 steam generator water pump should be increased from 0.1 to 0.11 gpm. The capacity for the Dowtherm pump should be increased to minimize the high film temperature problem in the char cooler.

Page B-15

Maximum allowable stress values used to calculate wall thicknesses are too high resulting in the calculation of thinner walls than are acceptable according to ASME code. The equations should all be reviewed.

Though this is a very minor error, the head stress equation should be,

$$t = \frac{PD}{2SE - 0.2P} \quad (1)$$

rather than

$$t = \frac{PD}{2(SE - 0.2P)}$$

Example:

For V202 coal hopper, the original values are

design temperature	400°F
design pressure	1100 psig
material	304L
allowable stress	15800 psi
E	0.9

Thus, the resulting vessel thicknesses calculated by SAI are

wall t = 0.405 in.
head t = 0.390 in.

Using Equation 1 and an allowable stress of 14700 psi for 304L seamless pipe gives

wall t = 0.426 in.
head t = 0.409 in.

Thus, in either case, 10 in. Sch 80 (0.500 in. thick) 304L stainless steel is acceptable. However, erroneous stress values and equations give a feeling of incompleteness and uneasiness.

The sample receiver and Dowtherm surge tank should be made out of 304L rather than 304 stainless steel. Since welding is required during fabrication, 304L would be a much better choice.

Tubing materials appear adequate at all locations. Tubing wall thicknesses are adequate, though for safety, appropriate relief valves made of compatible material should be employed. Note that 304L tubing and pipe should be used rather than 304 because its corrosion resistance is better after welding.

The remaining stress calculations in Appendix B were checked and are acceptable,

STATEMENT OF PROBLEM

CALCULATE THE THERMAL RADIATION EXCHANGE BETWEEN TWO FINITE LENGTH CONCENTRIC CYLINDERS. THE THERMAL CONDITIONS ARE:

. UNIFORM SURFACE TEMPERATURES
. ABSORPTANCE EQUAL TO EMITTANCE
. DIFFUSE EMITTANCE AND REFLECTANCE

THE GIVEN CONDITIONS ARE:

SURFACE LENGTH L, FT	=	.2730E+01	
DIAMETER OF SURFACE ONE D1, FT	=	.2384E+00	
SURFACE SPACING R, FT	=	.4750E-01	
SURFACE TEMPERATURE TS1, F	=	.3360E+04	
SURFACE TEMPERATURE TS2, F	=	.3430E+04	TO .3445E+04
EMITTANCE EM1	=	.9500E+00	
EMITTANCE EM2	=	.9500E+00	

ANALYSIS

THE NET RADIATION EXCHANGE BETWEEN TWO SURFACES IN THE ABSENCE OF ANY OTHER PARTICIPATING(REFLECTING)SURFACES IS DEFINED TO BE:

> THE ENERGY ABSORBED BY SURFACE 2 WHICH WAS ORIGINALLY EMITTED BY SURFACE 1 MINUS THE ENERGY ABSORBED BY SURFACE 1 WHICH WAS ORIGINALLY EMITTED BY SURFACE 2, TAKING INTO CONSIDERATION ALL INTERREFLECTIONS BETWEEN THE TWO SURFACES.

UNDER THE ASSUMPTIONS THAT

. THE SURFACES ARE ISOTHERMAL
. THE ABSORPTANCE OF EACH SURFACE IS EQUAL TO ITS EMITTANCE, I.E., THE SURFACES ARE GRAY
. THE SURFACES EMIT AND REFLECT DIFFUSELY

IT CAN BE SHOWN THAT THE NET RADIATION EXCHANGE IS GIVEN BY

$$Q = (EM1\ EM2\ F12/((1 - (1 - EM1)F11)(1 - (1 - EM2)F22) - (1 - EM1)(1 - EM2)F12\ F21))\ A1\ SIG(TS1^4 - TS2^4)$$

FOR THE CASE OF TWO CONCENTRIC CYLINDERS OF EQUAL LENGTH, THE INNER CYLINDER BEING SURFACE 1, THE ANGLE FACTOR F11 IS ZERO AND F21 = (D1/D2)F12 GIVING

$$Q = (EM1\ EM2\ F12/(1 - (1 - EM2)F22 - (1 - EM1)(1 - EM2)(D1/D2)F12^2))A1\ SIG(TS1^4 - TS2^4)$$

THE ANGLE FACTORS F12 AND F22 ARE GIVEN BY

$$F12 = 1 - (1/PI)(ARCCOS\ C/B$$

$$- (1/2\ Y)(((B + 2)^2 - 4\ X^2)^{1/2}\ ARCCOS\ (C/X\ B)$$

$$+ C\ ARCSIN\ 1/X - PI\ B/2))$$

$$F22 = 1 - 1/X + (2/PI\ X)ARCTAN(2(X^2 - 1)^{1/2}/Y) - (Y/2\ PI\ X)$$

$$(((4\ X^2 + Y^2)^{1/2}/Y)ARCSIN((4(X^2 - 1) + (Y^2/X^2)(X^2 - 2))/$$

$$(Y^2 + 4(X^2 - 1)))$$

$$- ARCSIN((X^2 - 2)/X^2) + (PI/2)((4\ X^2 + Y^2)^{1/2}/Y - 1))$$

WHERE $X = (D1 + 2\ R)/D1$, $Y = 2\ L/D1$, $B = Y^2 + X^2 - 1$, $C = Y^2 - X^2 + 1$

RESULTS
****

RANGED INPUT PARAMETERS

TS2, F	.3430E+04	.3435E+04	.3440E+04	.3445E+04

OUTPUT PARAMETERS	Case 1	Case 2	Case 3*	Case 4
F12	.9866E+00	.9866E+00	.9866E+00	.9866E+00
F22	.2743E+00	.2743E+00	.2743E+00	.2743E+00
F21	.7055E+00	.7055E+00	.7055E+00	.7055E+00
Q/A1, BTU/HR-SQ FT	-.2485E+05	-.2668E+05	-.2851E+05	-.3035E+05
Q/A2, BTU/HR-SQ FT	-.1777E+05	-.1908E+05	-.2039E+05	-.2170E+05
Q, BTU/HR	-.5081E+05	-.5454E+05	-.5830E+05	-.6206E+05

*Case 3 results in a heating element temperature of 3440°F.

with the exception of allowable stress values and the head equation noted above. Neither of these errors will change the calculated vessel thicknesses enough to require drawing changes.

Tubing thicknesses were checked via ANSI B31.1 and found to be adequate; however, one exception to this is the tube from the steam superheater to the injector. This tube is called out as Hastelloy X and would not meet code at 1800°F. This could be enclosed and shortened to minimize potential dangers.

Note that the steam superheater is to be enclosed in a purged enclosure because of electrical concerns; however it should also be mandatory from pressure considerations. The stress analysis for tubing and pipe was as follows:

P = 1100 psig
Allowable stress, SE = 9000 psi at 800°F (ANSI B31.1, 304L SST seamless tubing and pipe)

From B31.1, the governing equation for calculation of the wall thickness, t, based on the diameter, D_o, is

$$t = \frac{PD_o}{2(SE + Py)} \quad \text{where } y = 0.5$$

for

<u>1/4 in. tubing</u>

$$t = \frac{1100\ (0.250)}{2[9000 + 1100\ (0.5)]} = 0.014 \text{ in.}$$

<u>3/8 in. tubing</u>

$$t = \frac{1100\ (0.375)}{2[9000 + 1100\ (0.5)]} = 0.022 \text{ in.}$$

<u>1/2 in. tubing</u>

$$t = \frac{1100\ (0.5)}{2[9000 + 1100\ (0.5)]} = 0.029 \text{ in.}$$

<u>1/2 in. pipe</u>

$$t = \frac{1100\ (0.840)}{2[9000 + 1100\ (0.5)]} = 0.048 \text{ in.}$$