and explains qualitatively why the configurations employing a supercharged boiler do not perform as well as those incorporating a waste heat boiler.

5.3 Effect of Additional Components

5.3.1 Regenerative Feedwater Heaters

The purpose of this section is to show how the station efficiency is affected by regenerative feedwater heating. As we shall soon see, this change will increase the steam cycle efficiency. It is the station efficiency, however, that is to be improved. Improving the gas cycle, steam cycle, or gasification efficiencies in and of themselves will not necessarily improve the overall performance.

In order to keep the steam cycle simple and so as not to obscure the basic effect of feedwater heating, let us use only one closed feedwater heater and one open feedwater heater, the latter of which can also serve as a deaerator. Following the usual practice of placing the deaerator last in the feedwater train, the steam cycle is modified as shown in Figure 5.3-1. Note that the "condensate" on the shell-side of the closed feedwater heater is flashed into the shell-side of the condenser through a throttle valve. The heating of the feedwater is accomplished by extracting a small portion of the turbine flow. The first extraction occurs at a higher pressure, of course, than the second. The lowest pressure in the steam cycle occurs at the turbine exit, where the exhaust flow enters the condenser. Note that it is necessary to add a condensate pump to the feedwater train for two reasons. The first is that two-phase flow through the tube-side of the closed feedwater heater is undesirable and adding heat to the saturated liquid from the condenser at the condenser shell-side pressure would tend to form vapor. The pump serves to subcool the fluid by increasing the pressure. The second reason for the condensate pump is that the deaerator is an open heater at a pressure above the shell-side condenser pressure. Obviously a pump is needed to take the condensate at the condenser pressure and deliver it to the deaerator at a higher pressure. Figure 5.3-1 is applicable to the steam cycle of all four configurations. Reasonable values for the new specifiable parameters will now be assigned.

The pressure related data are given first followed by the other data. Let us assume that the first extraction flow occurs at a pressure of 30 psia and the second at 6 psia. The shell-side condenser pressure remains the same as before. Let us assume no additional pressure drops, since a lumped loss of 400 psi is already used. The only temperature-related data that needs to be specified is the terminal temperature difference in the closed feedwater heater which is taken to be 3°F. The efficiency of each turbine stage is taken to be 0.90 and that of the condensate pump also 0.90. The feedwater pump efficiency is unchanged from before. In Section 5.4, more realistic values for the low-pressure steam turbine efficiencies will be used.

When the steam cycles of each of the basic cycles of Section 5.2 are modified according to Figure 5.3-1 using the above data, the somewhat surprising results shown in Table 5.3-1 are obtained. The station efficiency for each configuration decreased slightly. As expected, the regenerative feedwater heating, however, did increase the steam cycle efficiency from 35.02 percent to 37.88 percent. The deterioration in in station efficiency may be explained by comparing the heat rejected

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from the steam cycle in the condenser to the heat loss through the stack gas. Feedwater heating tends to increase the stack gas temperature, since hotter

Table 5.3-1
Results with Feedwater Heating

	Configuration 1	2	3	4
Station Efficiency (%)				
Without	34.57	29.87	38.63	32.72
With	34.29	29.47	38.32	32.32
Steam Cycle Efficiency (%)				
Without	35.02	35.02	35.02	35.02
With	37.88	37.88	37.88	37.88
Increase in Stack Gas Heat Loss over Base Case (Btu)	510	732	553	724
Decrease in Steam Cycle Heat Rejected over Base Case (Btu)	470	676	510	669
Final Feedwater Temp. (°F)				
Without	122.8	122.8	122.8	122.8
With	252.9	252.9	252.9	252.9
Stack Gas Temp. (°F)				
Without	324.1	262.8	319.1	422.8
With	405.7	502.9	402.0	622.9
Dew Point Temp. (°F)	93	119	116	141

water enters the economizer. Let us refer to the temperature of the feedwater into the economizer as the final feedwater temperature. From Table 5.3-1, it is seen that for all configurations, this temperature has increased from 122.8°F to 252.9°F. The stack gas temperature is seen to be correspondingly higher. In each case, the increase in heat loss through the stack is greater than the savings in heat rejected from the steam cycle. Although the steam cycle efficiency is improved, this improvement is not enough to improve the station efficiency. However, this could not be known a priori because of the two competing effects.

Although the above trend indicates that cycle performance deteriorates slightly when feedwater heating is employed, the heaters will be kept in the cycle for the reasons which follow. In Table 5.3-1, the dew point temperatures are given for each configuration. Without feedwater heating, the final feedwater temperature is 122.8°F, and Configuration 4 would probably begin to condense some of the water vapor locally within the economizer. Furthermore, this final feedwater temperature is a direct result of the 1.75 psia assumed condenser pressure. If a pressure of 0.75 psia were assumed, the final feedwater temperature without heating would be about 93°F. Clearly, all four configurations would probably have local consensation in the stack gas. This is to be avoided because of the corrosive nature of the acid which would form. Another reason for keeping the feedwater heaters in the cycle is to provide a convenient location to deaerate the water, namely the open feedwater heater. Finally, a higher stack gas temperature increases the so-called stack effect, and a smaller diameter stack may be used. For these reasons, the two feedwater heaters will be kept in each configuration, and the loss of less than 0.5 percentage points in station efficiency will be accepted. Unless otherwise stated, all subsequent results are presented with the regenerative feedwaters heaters in the steam cycle.

5.3.2 Intercooled Compressor Serving the Combustor

Next the air compressor serving the combustor is replaced with a two-stage intercooled compressor on each configuration. The schematic of this modified portion of the gas cycle is shown in Figure 5.3-2 and applies to all four configurations. The intercooler serves to reduce the effective temperature of the gas during compression, tending to make the process more nearly isothermal, which requires much less work than adiabatic compression. It is for this reason that the effect of an intercooled compressor on each of the four configurationsis examined. As always, it is the station efficiency that must be used as a basis for comparison.

The new data applicable to the modified portion of the cycles must first be specified. It is easily shown that the optimum pressure ratio for each stage of the two-stage compression is equal to the square root of the product of the initial and final pressures. This results in the minimum amount of total compressor work being required. Since the combustor is assumed to operate at 10 atmospheres, the first-stage outlet pressure is taken to be 3.162 atmospheres. A 0.1 psi drop through the intercooler is further assumed. The temperature to which the air is cooled in the intercooler must be specified. It is arbitrarily assumed that the air may be cooled to within 50°F of ambient or to 127°F. Each stage of compression is assumed to have an efficiency of 0.9.

With these modifications to each configuration and with the regenerative feedwater heaters now in the system, the results summarized in Table 5.3-2 are obtained. It should first be noted that this modification does not improve station efficiency for any configuration. The most immediate effect of intercooling is seen in the reduction of the temperature of the air to the combustor. In Configurations 1 and 3, this results in less air flow to the combustor since the lower air temperature is a more effective diluent. However, in Configurations 2 and

4, the amount of air to the combustor is fixed since 10 percent excess air is stipulated. Now the lower air inlet temperature serves to reduce the combustor outlet temperature.

Table 5.3-2
Results with Intercooled Compressor
Serving the Combustor

	Configuration 1	2	3	4
Station Efficiency (%)				
Without	34.29	29.47	38.32	32.32
With	32.75	28.74	36.48	31.19
Air Temp. to Combustor (°F)				
Without	622.1	622.1	622.1	622.1
With	378.7	378.7	378.7	378.1
Air Flow To Combustor (lbm)				
Without	19.84	6.73	23.76	10.89
With	17.06	6.73	20.46	10.89
Combustor Outlet Temp. (°F)				
Without	2000.	3250.	2000.	3057.
With	2000.	3154.	2000.	2920.
Gas Cycle Heat Loss (Btu)				
Without	4877.	4228.	4045.	3889.
With	5391.	4681.	4661.	4601.
Heat to Steam Cycle (Btu)				
Without	4852.	6994.	5270.	6891.
With	4375.	6430.	4699.	6001.
Gas Cycle Net Work (Btu)				
Without	4856.	4175.	3432.	1967.
With	4638.	4071.	3387.	2145.

From the tabulation of results, it can further be seen for Configurations 1 and 3 that the gas cycle heat loss increases substantially while the heat transfer to the steam cycle decreases by a smaller amount. The intercooling is the primary reason for the additional heat loss. Since the steam cycle efficiency is unchanged (for all configurations), the station efficiency must decrease. Less heat is transferred to the steam cycle primarily because there is less gas flow. It is further shown for Configurations 1 and 3 that the net amount of work produced in the gas cycle has decreased; the work of compression decreased but the work of expansion decreased by a greater amount, a direct result of the lower gas flow due to the lower air flow. Clearly, for Configurations 1 and 3, intercooled compressors serving the combustor do not improve performance.

Let us now examine Configuration 2. Again the gas cycle heat loss increases but the heat transfer to the steam cycle decreases by an amount greater than this. Recall from Equation (2.1-10) that this heat transfer to the steam cycle is multiplied by η_2. The decrease in $(Q_\ell + Q_2)/Q_1$ is seen to be smaller than the decrease in $\frac{Q_2}{Q_1}\eta_2$, resulting in poorer performance. Again intercooling causes the effective gas cycle heat loss to increase, and the lower heat transfer to the steam cycle is caused by the lower combustor exit temperature. To make matters worse, the stack gas temperature increased about 60°F (not shown). As before, the net gas cycle work has decreased, although some work of compression is saved. Intercooling is not desirable, therefore, in Configuration 2.

A similar line of reasoning applies to Configuration 4 except that now the net gas cycle work has increased. However, the decrease in total heat rejected from the gas cycle compared to the decrease in the weighted heat transfer to the steam cycle, results in overall deteriorated performance.

In summarizing, the addition of an intercooled compressor serving the combustor is not warranted. Unless otherwise stated, all subsequent results are presented without intercooled air compressors serving the combustor.

5.3.3 Regenerator within Gas Clean-up System

Let us examine the temperature of gas leaving the steam generator for the new base cases, which include feedwater heating but not intercooled compression. For Configurations 1 and 2 the steam generator gas exit temperature is 1982°F and for Configurations 3 and 4 only 820°F. Note that the exit temperature of the adiabatic gasification type of configurations is much higher than that of the endothermic type. This is caused by the much higher steam demand to the gasifier for the latter, which is a characteristic of endothermic gasification as seen in Section 5.2. Generating significantly more steam for the gasifier removes more sensible heat from the power gas, thus lowering its temperature substantially. Since the gas must be cooled for the gas cleanup operation, a great amount of sensible heat is lost in the gas cooler when the gas temperature is excessively high. In fact, this is the chief reason why the endothermic configurations perform better than the adiabatic. Clearly, if these heat losses could be reduced, then the cycle performance would be improved.

A gas-to-gas counterflow heat exchanger between the "dirty" power gas entering the gas cleanup system and the "clean" gas leaving this system will reduce these losses. We may also refer to this device as a regenerator, since heat is being transferred from one fluid stream in the cycle to another. Figure 5.3-3 shows how each cycle must now be modified to include this new component. As noted in Section 3.11, the the regenerator effectiveness must be specified, which shall be taken to be 0.80 for now. Later in Section 5.7 a parametric study on this parameter will be done. The pressure drop on each side of the regenerator will be taken to be 0.1 psi.

When this modification is made to the cycles, the results summarized in Table 5.3-3 are obtained. The improvement in cycle performance is dramatic, especially for the first two configurations. Without the regenerator in service, the configurations with an endothermic gasifier performed better than those with an adiabatic gasifier. This is not surprising since the higher steam demand for endothermic gasification results in less heat being lost in the gas cooler, as shown in Table 5.3-3 for the results

Table 5.3-3
Results with 80% Effective Regenerator within Gas Cleanup System

	Configuration 1	2	3	4
Station Efficiency (%)				
Without	34.29	29.47	38.32	32.32
With	41.53	36.04	40.04	34.17
Heat Loss in Gas Cooler (Btu)				
Without	2606.	2606.	623.	623.
With	485.	485.	122.	122.

without regeneration. But with the regenerators in service, the adiabatic configurations are superior to the endothermic. Generating steam from 77°F water with 2000°F gas represents a large irreversibility compared to that associated with 80% effective gas-to-gas regeneration. Since endothermic gasification requires much more steam than adiabatic, the former contribution to the total irreversibility is emphasized, thus de-emphasizing the contribution from the latter. Configuration 1 now

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becomes the best performer. Note however that the two configurations utilizing a waste heat system are still superior to the two incorporating a supercharged boiler system.

Unless otherwise stated, all subsequent results are presented with the regenerator in the gas cleanup system.

5.3.4 Regenerator between Air Stream to Gasifier and Power Gas Stream

Next let us examine the temperatures of the air entering the gasifier and of the power gas leaving the gasifier on Configurations 1 and 2. Since there is no air flow to the gasifier for Configurations 3 and 4, these two cycles are dismissed from further consideration in this section. The air inlet temperatures are shown in Table 5.3-4 to be 651.9°F for Configurations 1 and 2 while the gasifier gas exit temperature is 2000°F. The large temperature clearance between these two fluid streams suggests that a regenerator could be incorporated into these configurations as shown in Figure 5.3-4.

Again the regenerator effectiveness is assumed to be 0.80. The pressure drops on each side of regenerator are assumed to be 0.1 psi. With this modification, the results shown in Table 5.3-4 are obtained.

Table 5.3-4
Results with 80% Effective Regenerator
between Gasifier Air and Gas Streams

	Configuration 1	Configuration 2
Station Efficiency (%)		
Without	41.53	36.04
With	41.80	36.62

Table 5.3-4 (Continued)

	Configuration	
	1	2
Temp. of Air to Gasifier (°F)		
Without	651.9	651.9
With	1730.4	1730.4
Heat Transfer in Regenerator within Gas Cleanup System (Btu)		
Without	2121.	2121.
With	1096.	1096.
Heat Transfer in Present Regenerator (Btu)		
Without	0.	0.
With	902.	902.
Steam Flow to Gasifier (lbm)		
Without	0.022	0.022
With	0.195	0.195
Heat Transfer in Steam Generator (Btu)		
Without	28.	28.
With	252.	252.
Gas Cycle Heat Loss (Btu)		
Without	3240.	1782.
With	3192.	1695.
Air Flow to Gasifier (lbm)		
Without	3.783	3.783
With	3.121	3.121
Enthalpy of Air to Gasifier (Btu)		
Without	1017.	1017.
With	1740.	1740.
Heating Value of Gas (Btu/SCF)		
Without	135.	135.
With	155.	155.

Note that the station efficiency improves only slightly. The reason for this, of course, is that the other regenerator is already effectively reducing the heat loss from the power gas flow stream. Note also that

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the required steam flow to the gasifier has increased from 0.022 to 0.195 lbm. This results in more heat transfer in the steam generator helping to reduce the total gas cycle heat loss slightly. Note that the hotter air into the gasifier results in less air demand but more total sensible heat is being added to the gasification process from the air. Furthermore, the lower heating value of the power gas is improved from 135 to 155 Btu/SCF. All of these observations are consistent with the conclusions in Chapter 2, concerning the effect of adding heat to the gasification process.

Admittedly, this regenerator improves the performance very little. As implied above, the gasification efficiency is significantly improved (from 86.58 to 94.75 percent). Improving the gasification efficiency significantly does not necessarily result in a significant improvement in station efficiency. Although the incremental improvement in performance is small, this regenerator shall be kept in Configurations 1 and 2 for all subsequent calculations unless otherwise stated. In Section 5.7, the relative importance of the two regenerators in the adiabatic configurations will be determined.

5.3.5 Intercooled Compressor Serving the Gasifier

Finally, an intercooler is added to the compressor serving the gasifier on Configurations 1 and 2 with the hope that the station efficiency may be increased by reducing the work of compression. The cycles must be modified according to Figure 5.3-5 where only the affected portion of the cycle is shown. Note that this modification applies only to the two configurations incorporating an adiabatic gasifier.

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As before, the first-stage pressure ratio is taken to be equal to the square root of the product of the inlet pressure of the first stage and outlet pressure of the second stage; therefore the pressure at the outlet of the first stage is 3.317 atmospheres. Again it is assumed that a 0.1 psi pressure drop occurs through the gas cooler, that the gas is cooled to within 50°F of ambient or to 127°F, and that each compressor stage has an efficiency of 0.90.

With this modification, the results shown in Table 5.3-5 are obtained. Note that the station efficiency of Configuration 1 has decreased, but that of Configuration 2 has just about remained the same. In Configuration 1 the small increase in the net gas cycle work is not enough to offset the even greater increase in the gas cycle heat loss. In Configuration 2 the larger amount of net gas cycle work is just about cancelled by the effect due to the greater gas cycle heat loss.

Table 5.3-5
Results with Intercooled Compressor
Serving the Gasifier

	Configuration 1	Configuration 2
Station Efficiency (%)		
Without	41.80	36.62
With	41.62	36.63
Gas Cycle Net Work (Btu)		
Without	3705.	1609.
With	3709.	1662.
Gas Cycle Heat Loss (Btu)		
Without	3192.	1694.
With	3265.	1774.

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In summarizing, the addition of an intercooled compressor serving the gasifier is not warranted. Unless otherwise stated, all subsequent results are presented without an intercooled air compressor serving the gasifier.

5.4 Optimization

The purpose of this section is to optimize each configuration by determining those operating conditions which maximize the station efficiency. It can be argued that the optimization needs to be done with respect to only three variables: the gasifier exit temperature, the combustor pressure, and the gasifier steam temperature. Increasing the steam cycle peak pressure would improve performance but the steam turbine exit quality is already near the practical lower limit of 88 percent and increasing the steam pressure would make the turbine exhaust even wetter. It has already been shown that feedwater heating is not really desirable in a combined cycle. Clearly the optimum steam extraction pressures would be the limiting low pressure in the steam cycle, namely the condenser pressure. It is not necessary, therefore, to optimize with respect to these pressures. Obviously, increasing the gas turbine inlet temperature would result in improved station efficiency, but this parameter is fixed by present-day gas-turbine technology at 2000°F, similarly for the peak steam-cycle temperature of 960°F. For the supercharged boiler configurations, the excess air fraction is a specifiable parameter. However, when the excess air was increased from the current value of 10 percent, performance did not improve. Consequently this parameter too does not need to be considered in the optimization. All other variables have obvious optimum values (like zero pressure drops) and are dismissed.

The effect on performance of some of these parameters is considered in the parametric studies of Section 5.7.

Before the optimization procedure is described let us adjust the steam turbine second- and third-stage efficiencies to be more realistic, namely 0.825 and 0.750 respectively. The new base case station efficiencies become 41.26, 35.74, 39.52, and 33.47 percent for Configurations 1 to 4 respectively. The steam turbine exit quality is calculated to be 87.9 percent for each configuration and is marginally acceptable.

For each configuration, the following optimization procedure is suggested. Let us use the data for each cycle as already described up to this point. Then, varying only the temperature of the gas leaving the gasifier, let us note the value which results in the highest station efficiency. With this optimum value for the gasifier exit gas temperature, the peak gas cycle or combustor pressure will then be varied and the station efficiency calculated. The optimum value will be noted. Finally, with the above two optimum values being used, the temperature of the superheated steam entering the gasifier will be varied, and the effect on station efficiency noted. Depending on the outcome, this procedure may have to be repeated until no further changes in the optimum conditions occur. In the next four subsections, each configuration is optimized in turn. It should be emphasized that the results of our effort in Section 5.3 are incorporated into all subsequent calculations, unless otherwise stated.

5.4.1 Configuration 1

As already outlined, the optimization begins with the waste heat combined cycle integrated with an adiabatic gasifier. All other parameters have the previously specified values. Table 5.4-1 shows the resulting station efficiencies as this temperature is varied from 1600°F to 2600°F in 200°F increments. Note that the station efficiency is not

Table 5.4-1
Configuration 1 - Optimum Gasifier Temperature

Gas Temperature (°F)	Station Efficiency (%)
1600	40.75
1800	41.17
2000	41.26
2200	41.25
2400	41.22
2600	41.19

a strong function of gasifier exit gas temperature. This shows that if it is desirable to operate the gasifier at higher temperatures, then the station efficiency will not be unduly compromised. For example, the steam flow to the gasifier is 0.457 lbm for gasification at 1600°F but only 0.105 lbm at 2600°F. Furthermore the gasification reactions would proceed faster at the higher temperature. This could result qualitatively in a smaller gasifier design, since the residence time of the species in the gasifier could be reduced. In any event, the optimum value is taken to be 2000°F, which, incidentally, is the value used prior to this phase of the calculations.

With this value for gasification temperature, the gas cycle pressure is now varied. It should be noted, however, that the optimum pressure for the gasification system is not independent of that for the gas turbine cycle. Clearly, these two pressure levels should be as nearly the same as possible, since any difference between them tends to act as an effective gas cycle pressure drop in the throttle valve. In reality, the gasification pressure must be slightly higher than the gas turbine cycle pressure because there will be losses in the real system. Let us use a difference of 1 atmosphere between the two systems. Considering the other pressure drops which have previously been specified, this is equivalent to assuming a 13.5 psi drop through the throttle valve after the steam generator. The resulting station efficiencies are shown in Table 5.4-2, where the combustor pressure is the independent variable. Note that the optimum

Table 5.4-2
Configuration 1 - Optimum Gas Cycle Pressure

Combustor Pressure (atm)	Station Efficiency (%)
5	40.58
10	41.26
15	40.62
20	39.54
25	38.28
30	36.83

pressure occurs at the value that has been used all along, that is, 10 atm. It should be noted that the calculated gasifier steam flow varies only slightly from 0.181 to 0.238 lbm as the pressure is increased from 5 to 30 atm. For pressures above 15 atm, the calculated results indicate that it is not possible to raise the 960°F superheated steam in the waste heat boiler. In fact, at 30 atm, only 792°F steam could be produced. This partly accounts for the poorer performance at increased pressure.

Finally, the temperature of the steam entering the gasifier is varied from 400°F to 1000°F in 100°F increments. The results, shown in Table 5.4-3, clearly show that there is no measurable effect of this parameter on station efficiency. Let us take the "optimum" value to be 600°F, since this will result in a smaller superheat section in the steam generator compared to raising 1000°F steam. One reason for the insensitivity

Table 5.4-3
Configuration 1 - Optimum Temperature of Steam to Gasifier

Steam Temperature (°F)	Station Efficiency (%)
400	41.25
500	41.25
600	41.26
700	41.26
800	41.26
900	41.26
1000	41.27

of the results to this parameter is that very little steam is required by the gasifier; recall that only about 0.2 lbm of steam is needed. The extra sensible heat transferred from the power gas to effect additional superheating is minimal. In fact, only about 50 Btu of additional heat are required as the steam temperature is increased from 600 to 1000°F. Later, for the configurations incorporating an endothermic gasifier, it will be seen that this is no longer the case.

Clearly, the optimization procedure does not need to be repeated. The optimum gasifier gas temperature of 2000°F and the optimum gas cycle pressure of 10 atm were used from the outset. The somewhat arbitrary gasifier steam temperature of 600°F is sufficiently close to the original value of 620°F that a new iteration is not necessary. Furthermore, station efficiency hardly depends on this steam temperature anyway.

We conclude, therefore, that the optimized cycle has a station efficiency of 41.26 percent. It should be emphasized that this includes the 10 percent station load factor. Pressure drops and component inefficiencies are also included. Without this 10 percent factor, the station efficiency would be 45.84 percent. It should be pointed out, however, that we have yet to consider the impact of meeting the federal gaseous emission requirements on nitric oxides. These are considered in Section 5.5.

5.4.2 Configuration 2

Now the indicated optimization procedure is applied to the supercharged boiler combined cycle integrated with an adiabatic gasifier. From Table 5.4-4, the optimum gasification temperature is seen to be 1800°F, giving a station efficiency which is only slightly better than the new base case value of 35.74 percent. The steam flow required by the gasifier is 0.264 lbm at the optimum gasification temperature. As in Configuration 1, the

Table 5.4-4
Configuration 2 - Optimum Gasifier Temperature

Gas Temperature (°F)	Station Efficiency (%)
1600	35.42
1800	35.79
2000	35.74
2200	35.67
2400	35.59
2600	35.51

station efficiency is not drastically affected by the gasification temperature. Again consideration of other factors such as reduced steam flow and smaller gasifier designs at higher temperatures may dictate actual operation off optimal conditions. The above results again indicate that the sacrifice in station efficiency would be minimal.

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The gas cycle pressure is varied next. As before, the value of the gasification temperature which proved to be optimal is now used. The resulting station efficiencies are shown in Table 5.4-5. The optimum pressure is seen to be 10 atm, which is the value that has been used thus

Table 5.4-5
Configuration 2 - Optimum Gas Cycle Pressure

Combustor Pressure (atm)	Station Efficiency (%)
5	32.20
10	35.79
15	35.60
20	35.28
25	34.89
30	34.50

far. The decrease in station efficiency for pressures above 10 atm is primarily due to the reduction in net work (134 Btu for 30 atm) produced in the gas cycle, although there is a small decrease in net work (49 Btu) produced in the steam cycle. As expected, for this configuration it is always possible to raise the 960°F steam in the boiler, since a super-charged boiler is now in the cycle.

Finally, the temperature of the steam to the gasifier is varied, with the results shown in Table 5.4-6. It is emphasized that the gasification

Table 5.4-6
Configuration 2 - Optimum Temperature of Steam to Gasifier

Steam Temperature (°F)	Station Efficiency (%)
400	35.77
500	35.78
600	35.79
700	35.79
800	35.80
900	35.81
1000	35.82

temperature of 1800°F and gas cycle pressure of 10 atm obtained above are used to obtain the results of Table 5.4-6. Again the "optimum" steam temperature will be taken to be 600°F, since a smaller superheat section in the steam generator will be needed compared to raising 1000°F steam. Also, as in the results of Configuration 1, the entire cycle is practically independent of this parameter.

When the above optimization procedure is repeated using the latest optimum values, the same results are obtained. It is concluded that for Configuration 2 the optimum operating conditions are as follows: 1800°F gasification temperature, 10 atm gas cycle pressure, and 600°F gasifier steam temperature. While the last specification is not really optimal, the decrease in station efficiency from that at 1000°F is almost undetectable. It appears that the station efficiency for the optimized cycle is only 35.79 percent, which is significantly below that of Configuration 1. Again it is pointed out, however, that consideration of the pollution criteria may reduce this gap.

5.4.3 Configuration 3

The same optimization procedure is now applied to the waste heat combined cycle integrated with an endothermic gasifier. As before, the gasification temperature is varied first. The results are shown in Table 5.4-7. The optimum gasification temperature is seen to be between 1800°F and 2000°F. The lower temperature is chosen as optimum even though slightly more steam is required in the gasification process (1.038 lbm at 1800°F compared to 0.997 lbm at 2000°F). The reason for this is that the heat source for the endothermic gasifier is the combustor, which, it should be recalled, has a product gas exit temperature of 2000°F in order to be compatible with the turbine inlet temperature. The effective temperature of the heat source for the gasifier must be higher than the gasification temperature because of the second law of thermodynamics, since it is impossible to transfer heat from one temperature to a higher temperature in a cycle without expending work. It is indeed fortuitous that the optimum gasification temperature turned out to be significantly below the 2000°F temperature of the heat source. Unlike Configurations 1 and 2, Configuration 3 cannot be operated at gasification temperatures above this limiting value. For completeness, it is also seen from the program

Table 5.4-7
Configuration 3 - Optimum Gasifier Temperature

Gas Temperature (°F)	Station Efficiency (%)
1600	39.18
1800	39.52
2000	39.52
2200	39.41
2400	39.27
2600	39.12

output that 4627 Btu are required by the gasifier. Of course, the same amount of heat is removed from the combustor, since no losses are assumed.

Next the combustor pressure is varied with the above optimum value being used and the resulting station efficiencies are shown in Table 5.4-8, where the optimum again is the value that has been used all along, namely 10 atm. As in the other waste heat system configuration, it is not possible to raise 960°F steam for combustor pressures above 15 atm.

At 30 atm, the temperature of the steam to the turbine is only 801°F. As before, this accounts in part for the reduced performance at higher combustor pressures.

Table 5.4-8
Configuration 3 - Optimum Gas Cycle Pressure

Combustor Pressure (atm)	Station Efficiency (%)
5	38.53
10	39.52
15	39.12
20	38.26
25	37.20
30	36.00

Finally, the gasifier steam temperature is varied with the above two optimum values now being used. The resulting station efficiencies are shown in Table 5.4-9. It should be noted that there is a larger effect on station efficiency now compared to that of the adiabatic configurations.

Table 5.4-9
Configuration 3 - Optimum Temperature
of Steam to Gasifier

Steam Temperature (°F)	Station Efficiency (%)
400	39.44
500	39.48
600	39.52
700	39.55
800	39.59
900	39.62
1000	39.66

Not shown in Table 5.4-9 is the effect of increasing this steam temperature on the steam generator gas-side exit temperature. For a steam temperature of 1000°F, the power gas is cooled to 367°F from the base case value of

568°F. In Section 5.6, it will be shown how the regenerator and possibly the gas cooler may be eliminated by taking advantage of this.

When the above procedure is repeated starting with a steam temperature of 1000°F, a combustor pressure of 10 atm, and a gasifier gas exit temperature of 1800°F, the optimum conditions do not shift. For the time being, these are accepted as optimal. The optimum station efficiency is 39.66 percent without consideration of the pollution criteria.

5.4.4 Configuration 4

Finally, the optimization procedure is applied to the supercharged boiler combined cycle integrated with an endothermic gasifier. As usual, the gasification temperature is varied first with the resulting station efficiencies shown in Table 5.4-10. The optimum gasification temperature is seen to be 1800°F. Unlike Configuration 3, the combustor outlet temperature is well above all feasible gasification temperatures; at the optimum the combustor outlet temperature is 3166°F. Again it would be

Table 5.4-10
Configuration 4 - Optimum Gasifier Temperature

Gas Temperature (°F)	Station Efficiency (%)
1600	33.12
1800	33.55
2000	33.47
2200	33.40
2400	33.21
2500	33.02

possible to operate the gasifier at a higher than optimal temperature if this became desirable for other reasons. For completeness, the amount of heat required by the gasifier is 4627 Btu and, again, must be equal to that supplied by the combustor.

Using the above 1800°F gasification temperature, the combustor pressure is now varied with the resulting effect on station efficiency shown in Table 5.4-11. For the first time the optimum gas cycle pressure is no longer the value that has been assumed all along, but rather 20 atm. Doubling the gas cycle pressure has added more than 2 percentage points to the previous highest station efficiency.

Finally with these two optimum values fixed, the temperature of the steam to the gasifier is varied with the resulting station efficiencies

Table 5.4-11
Configuration 4 - Optimum Gas Cycle Pressure

Combustor Pressure (atm)	Station Efficiency (%)
5	29.00
10	33.55
15	35.29
20	35.81
25	35.54
30	35.23

shown in Table 5.4-12. Note that it is not possible to have a superheated steam temperature of 400°F since this is below the saturation temperature of the water-side of the steam generator. As in the other endothermic configuration, the temperature of the gas leaving the steam generator is reduced substantially as the steam temperature is increased; from 553°F for the new base case to 339°F for the 1000°F steam temperature. More will be said about this in Section 5.6.

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When the above procedure is repeated starting with a steam temperature of 1000°F, a combustor pressure of 20 atm, and a gasifier gas exit temperature of 1800°F, the optimum conditions remain the same. Therefore, for

Table 5.4-12
Configuration 4 - Optimum Temperature
of Steam to Gasifier

Steam Temperature (°F)	Station Efficiency (%)
500	35.77
600	35.81
700	35.84
800	35.87
900	35.90
1000	35.93

now these conditions are accepted as optimal. The optimum station efficiency for Configuration 4 is only 35.93 percent, without consideration of the pollution critera.

5.5 Consideration of the Gaseous Pollution Criteria

The model that has been used so far is capable of predicting the amount of sulfur oxides (SO_x) and nitrogen oxides (NO_x) that enter the atmosphere from the stack. Recall that the combustor model, in particular, provides the composition of the product gas; that is, the mole (and weight) fraction of each constituent in the product gas, and hence stack gas, is known. Recall also that the amount of gas that the enters the atmosphere through the stack is calculated.

Through the EPA, the federal government has set limits on these two types of pollutants from power plants. These limits are summarized in

Table 5.5-1 for the various fuel types. Note that the units used in this table are lbm (of pollutant) per million Btu (of fuel input based on the

Table 5.5-1
Federal Emission Limits[31,25]

Type of Fuel	Limit (lbm per 10^6 Btu) SO_x	NO_x
Coal	1.2	0.7
Oil	0.8	0.3
Gas	0.2	0.2

higher heating value of the fuel). Note further that the limit varies with fuel type, the limits for gaseous fuels being the most stringent. Let us use the limits for a coal-fired plant, since this is in fact our primary fuel. It obviously would be much more difficult to meet the limits for a gaseous fuel.

As alluded to in Chapter 4 during the discussion of subroutine POLUTE, the NO producing reaction is reported[23] to freeze at a temperature of about 2400°F. In other words, the NO producing reaction slows down markedly in the reverse direction for temperatures below 2400°F. So it may be argued that even though combustion takes place at temperatures over 3000°F which produces a larger amount of NO, the concentration of NO will decrease as the stack gas temperature is approached. However, because of the above mentioned freeze phenomenon, the concentration of NO never goes below its equilibrium value at 2400°F.

It should be noted that provisions have been made already for reducing SO_x emissions via the gas cleanup system. Recall that it was assumed that 90 percent of the H_2S is removed from the power gas before it is burned in the combustor. Since there is much free oxygen in the combustor, the sulfur in the remaining H_2S and all the COS ends up mostly in the form of SO_2 with smaller amounts in the form of SO and SO_3.

Although more complete results on the gas composition are presented in the next section, let us present some of these now in order to see the relative magnitudes. These abridged results are shown in Table 5.5-2 and are from the calculations which yield the optimum operating conditions discussed in the previous section. For the values given, it is assumed

Table 5.5-2
Abridged Results on Combustor Product Gas Composition
(PPM by Weight)

Effluent	Configuration 1	2	3	4
SO	0	0	0	0
SO_2	230	620	220	500
SO_3	30	0	20	0
NO	520	3420	480	2810
NO_2	20	0	10	10

that equilibrium exists at the combustor exit temperature, namely 2000, 3610, 2000, and 3354°F respectively for Configurations 1 to 4.

When the amount of SO_x and NO_x are calculated according to the method given in Section 4.6, the results shown in Table 5.5-3 are obtained assuming equilibrium at the just stated combustor exit temperatures. Note that even for equilibrium at 2000°F, Configurations 1 and 3 are unacceptable with respect to NO_x. Configurations 2 and 4 are also unacceptable with respect to NO_x at this point, but it is important to remember that these values correspond to equilibrium at temperatures of 3610 and 3354°F respectively, which are above the 2400°F freeze temperature. All configurations meet the 1.2 lbm/10^6 Btu limit on SO_x; it is concluded that at least

Table 5.5-3
Gaseous Emissions Assuming Equilibrium
at the Combustor Exit Temperature
(lbm/10^6 Btu)

Effluent	Configuration 1	2	3	4
SO_x	0.60	0.57	0.50	0.49
NO_x	1.24	3.21	1.03	2.74

for the coal which is assumed thus far, that 90 percent H_2S removal is effective. In fact, this leaves plenty of margin in the tail gas effluents from the sulfur recovery operation in the Claus plant.

Now let us modify the calculation of the NO_x emissions by the method discussed in Section 4.6, namely by assuming that NO producing reaction freezes at 2400°F. These results are shown in Table 5.5-4. The NO_x for Configurations 1 and 3 must obviously increase while that for Configurations 2 and 4 must decrease. Now the supercharged boiler configurations are

Table 5.5-4
NO_x Emission Assuming NO Producing
Reaction Freezes at 2400°F
(lbm/10^6 Btu)

Effluent	Configuration 1	2	3	4
NO_x	3.54	0.41	2.93	0.47

acceptable with respect to both NO_x and SO_x emissions. The waste heat boiler configurations, however, exceed the limit on NO_x by more than a

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factor of 4. As mentioned during the development of the combustor model, flue gas recirculation is sometimes used as a means to reduce the NO_x. In essence some of the relatively cool flue gas is recirculated back into the combustor thus replacing some of the excess air as the diluent. This serves two purposes: one is to reduce the amount of gas that actually goes to the atmosphere and the other is to reduce the amount of NO produced in the first place.

Let us incorporate flue gas recirculation and modify the waste heat configurations as shown in Figure 5.5-1, where only the affected portion of the cycle is shown. Recall from the development of the combustor model in Section 3.2 that unless the flue gas enters the combustor at a reduced temperature the benefit of flue recirculation is lost. Also gas compressors are needed, since the flue gas is at atmospheric pressure and the combustor operates at elevated pressures. The first gas cooler is added to reduce the work required by the first stage of compression. The intercooler helps to reduce the work required by the second stage of compression. Finally the second gas cooler is utilized to lower the temperature of the compressed flue gas before entering the combustor to maximize the effect of the flue gas recirculation.

Additional data need to be specified before the NO_x emission can be calculated. Let us assume that each of compressor stages has an efficiency of 0.9 and that the first gas cooler reduces the flue gas temperature to 250°F, the second to 300°F, and the third to 350°F. Note that is is not possible to reduce these temperatures to within 50°F of ambient as before because the flue gas has a high volume fraction of water vapor. As the pressure increases, the dew point temperature increases. Condensation in the flue gas recirculation system is to be avoided for the same reasons it is to be avoided elsewhere in the system. A pressure drop of 0.1 psi is assumed in each gas cooler.

With this modification to only Configurations 1 and 3, the results shown in Table 5.5-5 are obtined. Note the decrease in performance as the fraction of flue gas recirculation is increased. In order to ensure some margin, it appears that about 53 percent recirculation is necessary for

Table 5.5-5
Effect of Flue Gas Recirculation
on Configurations 1 and 3

Configuration 1

Fraction of Flue Gas Recirculated	Station Efficiency (%)	NO_x (lbm/10^6 Btu)
0.30	38.36	1.89
0.35	37.95	1.62
0.40	37.55	1.34
0.45	37.19	1.05
0.50	36.85	0.74
0.55	36.56	0.33

Configuration 3

Fraction of Flue Gas Recirculated	Station Efficiency (%)	NO_x (lbm/10^6 Btu)
0.30	37.08	1.37
0.35	36.71	1.10
0.40	36.38	0.82
0.45	36.07	0.49

Configuration 1, while for Configuration 2 only 45 percent is needed. Unless otherwise stated, the fraction of recirculated flue gas is fixed at these values for all subsequent calculations. The NO_x shown in Table 5.5-5 has been calculated assuming the 2400°F freeze temperature.

When Configurations 1 and 3 are checked to determine how flue gas recirculation might effect the previously calculated optimum conditions, no shift in these parameters occurs. That is, the two configurations are still optimum for the previously determined operating conditions..

Finally, let us verify one of the assumptions made in Section 4.6 concerning the numerical equivalence of the mole and weight fraction of NO. For each configuration, these are shown in Table 5.5-6, assuming 53 and 45 percent recirculation for Configurations 1 and 3 and no recirculation,

Table 5.5-6
Numerical Equivalence of Mole and Weight Fraction of NO

	Configuration 1	2	3	4
Mole Fraction	.00018	.00342	.00016	.00263
Weight Fraction	.00019	.00353	.00017	.00277

of course, for Configurations 2 and 4. Clearly, the approximation made in Section 4.6.16 is justified since ample margin exists in the calculated NO_x emissions compared to the limiting value.

5.6 Review of Results

The main purpose of this section is to summarize some of the more important results. After modifying Configurations 3 and 4 still further, new cycle schematics for each optimized configuration will be presented. In addition, the water and heat rejection requirements will be given for each optimized configuration.

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Before summarizing the results, let us review the results for the optimization of Configurations 3 and 4 with respect to the gasifier steam temperature. Recall that as this temperature was increased the temperature of the power gas leaving the steam generator decreased a few hundred degrees. By increasing the steam temperature even further, the gas temperature at the steam generator exit could be decreased to within the temperature range of the gas cleanup system, which is 200 to 260°F. This suggests that the regenerator can be removed from these endothermic configurations at the expense of making the steam generator superheater larger. When this regenerator is removed from these configurations and the temperature of the steam to the gasifier is increased, the results shown in Table 5.6-1 are obtained. Only the results for the steam temperature which gives a steam generator gas exit temperature near 200°F are presented. Clearly, the gas

Table 5.6-1
Results without Regenerator in Service and Elevated Gasifier Steam Temperatures

	Configuration 3	Configuration 4
Gasifier Steam Temp. (°F)	1280	1200
Station Efficiency (%)		
Previous Optimum	36.07	35.93
Without Modifications	36.14	35.95
Steam Generator Gas Exit Temperature (°F)	211.	222.
Heat Removed by Gas Cooler (Btu)	11.	22.

cooler before the gas cleanup system could also be removed, since cleanup at both 211 and 222°F is acceptable. This would result in even higher performance, but the improvement would be small.

It should be noted that the purpose of this modification is to eliminate the use of expensive equipment and not to improve the station efficiency by less than a tenth of a percentage point. By taking advantage of using a higher gasifier steam temperature, it is possible to eliminate an expensive piece of equipment, namely the regenerator, from Configurations 3 and 4. The gas cooler, however, will be left in the cycles, but it should be kept in mind that these too could be eliminated if an actual plant were to be built.

An obvious question arises. Why is it possible to allow steam temperatures in excess of 960°F in the steam generator but not in the boilers? A careful examination of the differences between the two provides the answer. The primary reason for limiting the steam temperature to about 1000°F in the supercharged and waste heat boilers is that stress problems arise at elevated temperatures because of the large pressure differential between the two sides. In the waste heat boiler, the steam-side operates at 1600 psia but the gas-side at atmospheric pressure. In the supercharged boiler, the steam-side again operates at 1600 psia but the gas-side at ten or twenty atmospheres. In any event, at elevated temperatures stress problems arise with these kinds of pressure differentials. This is not the case in the steam generator, however, because both sides are necessarily at approximately the same pressure. Consequently, it is probably no problem to raise the higher temperature steam in the steam generator. Unless otherwise stated the regenerator in Configurations 3 and 4 is removed from the respective cycles for all subsequent calculations. When Configurations 3 and 4 are re-optimized, no changes from the previous optimum conditions result.

We are in a position now to modify Figures 2.3-1 to 2.3-4 based on the results up to this point. The modified configurations are shown schematically in Figures 5.6-1 to 5.6-4 for Configurations 1 to 4 respectively. Configuration 1 now has regenerative feedwater heating, single stage air compressors, two regenerators, and flue gas recirculation. Configuration 2 similarly has regenerative feedwater heating, single stage air compressors, and one regenerator, but no flue gas recirculation. Configuration 3 has regenerative feedwater heating, a single stage air compressor, flue gas recirculation, but no regenerators at all. Finally, Configuration 4 is unmodified from original cycle presented in Chapter 2 except for the addition of the regenerative feedwater heating. The new cycle schematics represent the final versions of the original cycles presented in Chapter 2 with gaseous emission criteria now considered.

Let us now summarize some of the key results for each optimized configuration as shown in Table 5.6-2. These results apply to the cycles shown in Figures 5.6-1 to 5.6-4. From this table it is seen that all four configurations have practically the same station efficiency, although Configuration 1 is marginally the best. More importantly, Configuration 1 requires less total steam than the other configurations. As expected, the endothermic configurations require much more steam than the adiabatic ones. Note that the amount of heat rejected through the condenser for the waste heat configurations is significantly below that for the supercharged boiler configurations. For Configuration 1, only about 29 percent of the heat input to the cycle is actually rejected to the heat sink, probably a river, compared to about 63 percent for an equally efficient conventional fossil-fueled plant. This difference could be significant enough to eliminate the need for expensive cooling towers, which seem today to be almost standard equipment on new power plants. This benefit practically disappears for the supercharged boiler configurations. Another interesting trend is the ratio of gas cycle work to steam cycle work. Once again the indication is that de-emphasizing the gas cycle is undesirable. If flue gas recirculation were not needed, then the configurations

Table 5.6-2
Summary of Results for
Final Version of Each Configuration

	Configuration 1	2	3	4
Station Efficiency (%)	36.67	35.79	36.14	35.95
Combined Cycle Efficiency (%)	43.01	41.83	42.08	44.86
Heating Value of Gas (Btu/SCF)	155.	157.	288.	290.
Water Requirement (lbm)				
Gasification	0.195	0.264	1.038	1.068
Gas Cleanup	0.268	0.243	0.123	0.000
Total	0.463	0.507	1.161	1.068
Heat Rejected (Btu)				
Gas Cycle	3861.	1710.	4114.	2243.
Steam Cycle	3692.	5968.	3515.	5412.
Ratio of Gas Cycle Work to Steam Cycle Work	1.44	0.47	1.53	0.63
Fraction of Flue Gas Recirculated	0.53	--	0.45	--
Pollution (lbm/10^6 Btu)				
NOx	0.52	0.41	0.50	0.47
SOx	0.59	0.57	0.49	0.49
Station Efficiency (%) without Pollution Control	41.26	35.79	39.66	35.93

with high gas cycle work to steam cycle work ratios would be superior to those with low such ratios. Flue gas recirculation takes its toll on station efficiency, somewhat masking the correlation between this ratio and cycle performance. For added proof of this, refer to the results presented in Tables 5.2-3 and 5.2-4. For completeness, the amount of flue gas recirculation as well as the amount of NO_x and SO_x emissions are also given in Table 5.6-2.

It is convenient at this point to present typical equilibrium compositions of the clean fuel gas and the combustor product gas. The former are summarized in Table 5.6-3 and the latter in Table 5.6-4. Note that the composition is given by mole fraction (or volume fraction).

Table 5.6-3 shows that the fuel gas for Configurations 1 and 2 is composed mostly of N_2, CO, and H_2. Recall that these configurations incorporate adiabatic gasifiers and that a large amount of air is required. Consequently, a high fraction of the fuel gas is composed of N_2, which

Table 5.6-3
Composition of Clean Fuel Gas
for Final Version of Each Configuration
(Mole Fraction)

Species	Configuration 1	2	3	4
H_2	0.1835	0.1933	0.4968	0.4950
CH_4	0.0021	0.0046	0.0299	0.0530
H_2O	0.0784	0.0784	0.0784	0.0392
CO	0.3017	0.2954	0.3771	0.3822
N_2	0.4256	0.4130	0.0022	0.0024
Ar	0.0053	0.0052	0.0000	0.0000
CO_2	0.0027	0.0094	0.0149	0.0274
H_2S	0.0004	0.0004	0.0006	0.0006
COS	0.0002	0.0002	0.0001	0.0002
NH_3	0.0001	0.0001	0.0000	0.0000

serves to lower the effective heating value of the gas, thus giving a low-Btu fuel gas. For Configurations 3 and 4, the nitrogen in the fuel gas is due only to that in the coal, since endothermic gasification

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requires no air. For these configurations, the fuel gas is composed mostly of H_2 and CO, thus yielding a higher effective heating value for the fuel gas. These results agree with the general trends indicated in Section 2.2. It should also be noted that the amount of CH_4 produced is insignificant for all configurations, although more is produced in the endothermic gasifiers. This is unfortunate since CH_4 has a heating value of about 1000 Btu/SCF and is the primary component in natural gas. Clearly the gasification processes would have to be modified substantially to produce a synthetic natural gas (mostly CH_4). This is, of course, outside the scope of this dissertation.

Table 5.6-4 shows that the combustor product gas is composed mostly of N_2, which is not surprising since air is necessary to the combustion process.

Table 5.6-4
Composition of Combustor Product Gas
for Final Version of Each Configuration
(Mole Fraction)

Species	Configuration 1	2	3	4
H_2	0.0000	0.0006	0.0000	0.0002
H_2O	0.1254	0.1327	0.1995	0.1865
CO	0.0000	0.0037	0.0000	0.0008
N_2	0.7025	0.6922	0.6430	0.6511
O_2	0.0185	0.0121	0.0165	0.0148
Ar	0.0088	0.0087	0.0081	0.0082
CO_2	0.1443	0.1441	0.1325	0.1343
NO	0.0002	0.0034	0.0002	0.0027
OH	0.0000	0.0020	0.0000	0.0012
H	0.0000	0.0001	0.0000	0.0000
O	0.0000	0.0001	0.0000	0.0000
NO_2	0.0000	0.0000	0.0000	0.0000
SO	0.0000	0.0000	0.0000	0.0000
SO_2	0.0003	0.0003	0.0002	0.0002
SO_3	0.0000	0.0000	0.0000	0.0000

The main products of combustion are H_2O and CO_2 as expected. Practically no CO is produced by Configurations 1 and 3 with a significant amount produced by Configuration 2 and a moderate amount by Configuration 4. Note that in Configurations 1 and 3, the amount of NO is quite low because of the flue gas recirculation. Note that practically no NO_2, no SO, and no SO_3 are produced.

5.7 Parametric Studies

Let us determine the sensitivity of station efficiency to variations in some of the parameters, the values of which have been assigned somewhat arbitrarily in Section 5.2. Among the parameters that will be varied are coal composition, regenerator effectiveness, pressure drops and component efficiencies, boiler pinch points, and gas turbine inlet temperature.

5.7.1 Coal Composition

As mentioned before, the coal which has been assumed in all the previous calculations is a Pennsylvania high volatile bituminous coal. Let us now use three other coals, the compositions of which are given in Table 5.7-1. For convenience, the composition of the Pennsylvania coal is listed in this table also. Note that a typical eastern coal[25], a Wyoming coal[27], and an Illinois coal[13] are used. It should be noted that the Illinois coal is the same one used in ECAS[14] and is actually referred to as Illinois No. 6. The effect on the station efficiency for only Configuration 1 will be shown since similar results are obtained for the other configurations. For each of these three other coals, the station efficiency is improved slightly

Table 5.7-1
Coal Compositions for Parametric Study
(Weight Fractions)

	Reference Coal	Typical Eastern Coal	Wyoming Coal	Illinois No. 6
C	0.7304	0.786	0.730	0.596
H	0.0528	0.049	0.056	0.059
O	0.0616	0.020	0.151	0.200
N	0.0088	0.005	0.012	0.010
S	0.0264	0.010	0.005	0.039
$H_2O(\ell)$	0.0300	0.020	0.000	0.000
Ash	0.0900	0.110	0.046	0.096

with the change in station efficiency being less than one percentage point as shown in Table 5.7-2. As expected, the amount of SO_x produced varies directly with the weight fraction of sulfur in the coal. Note that even for the higher sulfur Illinois No. 6 the amount of SO_x produced is still below the 1.2 lbm/10^6 Btu limit. For completeness, the lower heating values of each coal, calculated from the Dulong approximation, are also tabulated.

Table 5.7-2
Results of Parametric Study on Coal Composition
(Configuration 1)

	Reference Coal	Typical Eastern Coal	Wyoming Coal	Illinois No. 6
Station Efficiency (%)	36.67	36.84	37.33	36.76
SO_x (lbm/10^6 Btu)	0.59	0.21	0.12	1.10
Lower Heating Values (Btu/lbm)	12,747	13,526	12,222	10,334

From this brief parametric study, it can be seen that the cycle performance is fortunately not a strong function of coal composition.

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5.7.2 Relative Importance of Regenerators in Configurations 1 and 2

Next let us try to determine the relative importance of the two regenerators incorporated in the adiabatic configurations. This is most easily done by varying the effectiveness of one regenerator while keeping the other one fixed at its nominal value of 0.80. Let us refer to the regenerator in the gas cleanup system as RG2 and that adjacent to the gasifier as RG1.

Let us hold the effectiveness of RG2 at 0.80 and vary the effectiveness of RG1 as shown in Table 5.7-3, where the resulting station efficiencies are

Table 5.7-3
Results of Parametric Study on the Effectiveness of RG1 with that of RG2 Held at 0.80

Effectiveness of RG1	Station Efficiency (%) Configuration 1	Station Efficiency (%) Configuration 2
0.80	36.67	35.79
0.60	36.60	35.68
0.40	36.52	35.58
0.20	36.44	35.46
0.00	36.35	35.34

also shown. It is noted that the decrease in station efficiency is not significant as the effectiveness of RG1 is decreased. In fact, if RG1 were removed completely, the station efficiency would drop less than 0.5 percentage points from its nominal value.

Next let us hold the effectiveness of RG1 at 0.80 and vary the effectiveness of RG2 as shown in Table 5.7-4, where the resulting station efficiencies are also shown. Now the deterioration in performance is dramatic. In fact, for both Configurations 1 and 2, the station efficiency would drop about 3 percentage points if RG2 were removed from the cycle. Clearly the

the regenerator in the gas cleanup system is the more important one. The effect of this trend on economic decisions concerning these regenerators is obvious.

Table 5.7-4
Results of Parametric Study on the Effectiveness of RG2 with that of RG1 Held at 0.80

Effectiveness of RG1	Station Efficiency (%) Configuration 1	2
0.80	36.67	35.79
0.60	35.90	35.14
0.40	35.11	34.40
0.20	34.31	33.63
0.00	33.51	32.85

5.7.3 Pressure Drops and Component Efficiencies

In order to see the effect of the assumed pressure drops and component efficiencies on station efficiency let us first make all the pressure drops zero and note the results. Then with zero pressure drops, let us calculate the station efficiencies assuming all the compressors, pumps, and turbines are 100 percent efficient. The effect of the assumed pressure drops is shown in Table 5.7-5. Note that the station efficiency increases

Table 5.7-5
Results of Parametric Study on Pressure Drops

	Station Efficiency (%) Configuration 1	2	3	4
With Pressure Drops	36.67	35.79	36.14	35.95
Without Pressure Drops	37.53	36.78	36.95	36.85

by about 1 percentage point for all configurations. The improvement in performance by assuming ideal components is much more dramatic as shown in Table 5.7-6. In fact, more than 6 percentage points are lost because of

Table 5.7-6
Results of Parametric Study on Efficiencies of Compressors, Pumps, and Turbines (Without Pressure Drops)

	Station Efficiency (%)			
	Configuration			
	1	2	3	4
Non-ideal Components	37.53	36.78	36.95	36.85
Ideal Components	44.73	43.04	43.49	43.74

the inefficiencies associated with the pumps, compressors, and turbines compared to the zero pressure drop cases. Marked improvement in overall performance can be expected by decreasing the irreversibilities associated with these components, although significant improvements in these component efficiencies are unlikely.

5.7.4 Boiler Pinch Point Temperature Differences

Let us now decrease the pinch point temperature differences in the boiler of each configuration from 40 to 20°F. This would require a large boiler, since more heat transfer area would be needed. As shown in Table 5.7-7, the improvement in performance is not great. In fact, for Configuration 2 the performance is unchanged since a 20°F pinch point temperature difference is not possible. For this case, the computer output indicates that the pinch point temperature difference has to be 40°F to ensure 7°F subcooled water in the economizer. Based on these results, it is unlikely that decreasing the pinch point temperature difference in the boilers from 40°F to 20°F could be economically justified.

Table 5.7-7
Results of Parametric Study on
Boiler Pinch Point Temperature Differences

Station Efficiency (%)

	Configuration 1	2	3	4
With 40°F	36.67	35.79	36.14	35.95
With 20°F	37.21	35.79	36.64	36.13

5.7.5 Gas Turbine Inlet Temperature of 2400°F

Finally, let us determine how much the station efficiency could be improved by increasing the turbine inlet temperature to 2400°F. Because Configuration 1 has resulted in the best performance, let us now restrict our attention only to this configuration. It is reasonable to expect that similar improvements in station efficiency for each of the other configurations would result for a similar increase in gas turbine inlet temperature. For the remainder of this subsection, therefore, we restrict our attention to Configuration 1 only.

The higher gas turbine inlet temperature will require a different amount of flue gas recirculation. Let us vary the fraction of flue gas recirculated for the new turbine inlet temperature of 2400°F as shown in Table 5.7-8. Also shown in this table are the resulting station efficiencies and the amounts of NO_x that enter the atmosphere. It should be emphasized that since the "freeze" and "equilibrium" temperatures are both 2400°F, the calculated amounts of NO_x are essentially identical for the two methods of computation. Note that about 40 percent flue gas recirculation is sufficient to reduce the NO_x to below the 0.7 lbm/10^6 Btu limit with ample

Table 5.7-8
Parametric Study on Flue Gas Recirculation at Gas Turbine Inlet Temperature of 2400°F for Configuration 1

Fraction of Flue Gas Recirculated	Station Efficiency (%)	NO_x (lbm/10^6 Btu)
0.00	44.93	2.36
0.10	44.07	1.94
0.20	43.23	1.51
0.30	42.43	1.06
0.40	41.66	0.54

margin. It may be recalled from Section 5.5 that with a turbine inlet temperature of 2000°F, Configuration 1 required 53 percent flue gas recirculation reducing the amount of NO_x to 0.52 lbm/10^6 Btu.

From Table 5.7-8 it is easily seen that increasing the gas turbine inlet temperature to 2400°F from 2000°F would result in a substantial improvement in overall performance. A 400°F increase in this parameter causes the station efficiency to increase almost exactly 5 percentage points. Furthermore, Table 5.7-8 shows the price that must be paid to meet the current pollution criteria with respect to NO_x emissions. Meeting the criterion on NO_x by using flue gas recirculation results in lowering the station efficiency by more than 3 percentage points.

5.8 Discussion of Assumptions

In this section, some of the assumptions that have been explicitly made or implied are now discussed in light of the results which have been

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obtained. There are two basic kinds of results which have been presented: one type is relative and the other absolute. When each configuration was compared with the others, the results which were used in this comparison were all relative. Because consistent assumptions were always made among the four configurations, this comparison was not only valid but also quite instructive. Obviously, these kinds of relative results cannot be that sensitive to the assumptions that have been made. Each configuration was modeled in parallel to ensure this consistency throughout. The second kind of result is necessarily more sensitive to the assumptions. If we were to build the type of plant which has been referred to as Configuration 1, how close could we expect to come to the calculated station efficiency of 36.67 percent or a heat rate of 9307 Btu/kwhr? It is believed that the calculated results are a best estimate of the results which would be obtained from an actual plant.

5.8.1 Dulong Approximation

Several sources[18,26] give the accuracy of the Dulong approximation to be within 2 to 3 percent. That is to say, the heating value of the coal based on measurements from a bomb calorimeter agree to within 2 or 3 percent of that obtained from application of the Dulong approximation. While this error may seem to be substantial, it is well within usual engineering accuracy. Furthermore, to determine the actual enthalpy for for a wide variety of coals and conditions would be an impossible task. The Dulong approximation provides a practical means of obtaining the heating value and enthalpy of the coal for analytical purposes.

5.8.2 Chemical Equilibrium

It is generally agreed that thermodynamic equilibrium would exist in the gasifier for temperatures above 1700°F[28]. Since the optimum value of gasification temperature is well above this for each configuration, the assumption of thermodynamic equilibrium appears to be justified. This is indeed fortuitous since the kinetics for reactions involving coal are extremely unpredictable and are highly dependent on coal type[29]. Also, as pointed out in Section 2.4, a high gasification temperature is advantageous from the standpoint of not producing problemsome tars, phenols, mercaptans, and so forth.

As mentioned in Chapter 3, the equilibrium composition of the product gas leaving the combustor is also calculated. Again equilibrium is assumed at the existing flame temperature. With the exception of the computation of frozen NO for pollution purposes, the composition of the fuel gas is assumed to be frozen at the gasifier exit conditions and composition of the products of combustion is assumed to be frozen at the combustor exit conditions. Although an equilibrium composition could have been calculated at each cycle point, this was not deemed practical for two reasons. First, many reactions slow down considerably as temperature is reduced and it is unlikely that equilibrium is achieved for reasonable time periods. Second, much more computational time would be required. Since it is believed that this would have a very small effect on results, this refinement is not justified.

5.8.3 Feasibility of an Endothermic Gasifier

There are many possible ways to deliver heat to the gasifier. Among these are gases, pebbles, molten salts, and slag. For each of these, heat could be removed from a high temperature heat source, presumably the combustor, and transferred to the gasifier by one of these heat transfer media. The high-temperature gas-cooled reactor provides an immediate example of how a gas may be used. The Mayland Pebble-Bed Gasifier[29] uses pebbles to effect the necessary heat transfer. Molten salts are used in the Kellogg gasification processes[29]. Finally, the Rummel Double-Shaft Gasifier[29] utilizes the coal slag to provide the necessary heat transfer. As noted in Section 2.4, Texaco is reporting progress on material problems associated with coal slag. Although the reason for heat transfer to the gasification process in each of these cases may be different, the basic ideas should be applicable to Configurations 3 and 4.

5.8.4 Limits on NO_x and SO_2 Effluents

It appears that the appropriate limits on NO_x and SO_2 emissions will be changed in the very near future: probably from 0.7 lbm/10^6 Btu to 0.6 lbm/10^6 Btu for NO_x and from 1.2 lbm/10^6 Btu to 99 percent removal for SO_2. From the results presented in this chapter, the projected limit on NO_x is already met. The more stringent limit on SO_2 will cause only a slight decrease in station efficiency, but the capital cost of the sulfur removal system will more than double[1]. Also, new combustor designs are now emerging which are effective in reducing the NO_x emissions; if flue gas recirculation could be eliminated on the waste heat configurations, the station efficiency could be significantly improved.

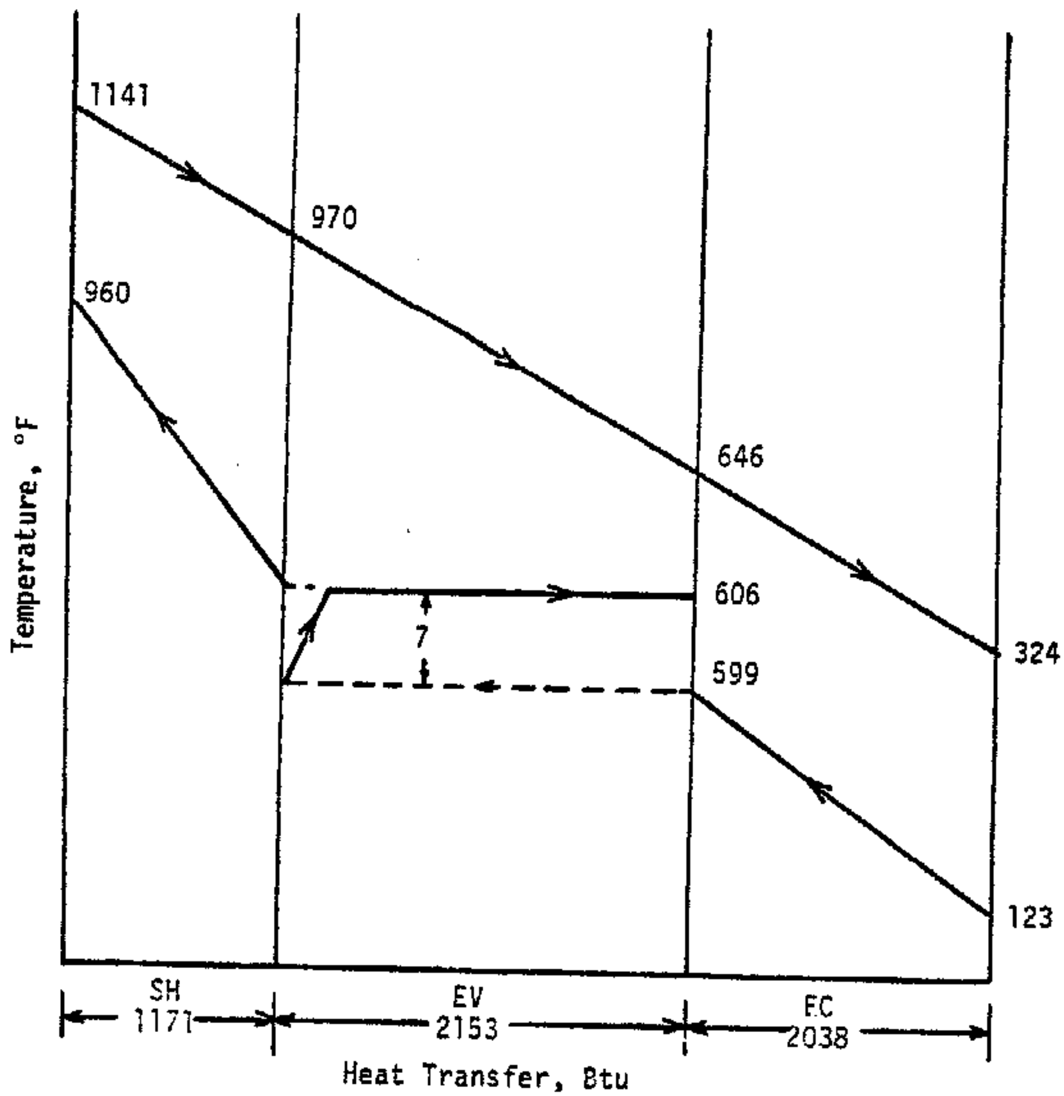

Figure 5.2-1 Temperature-Heat Transfer Diagram for Waste Heat Boiler (Typical Results from Configuration 1)

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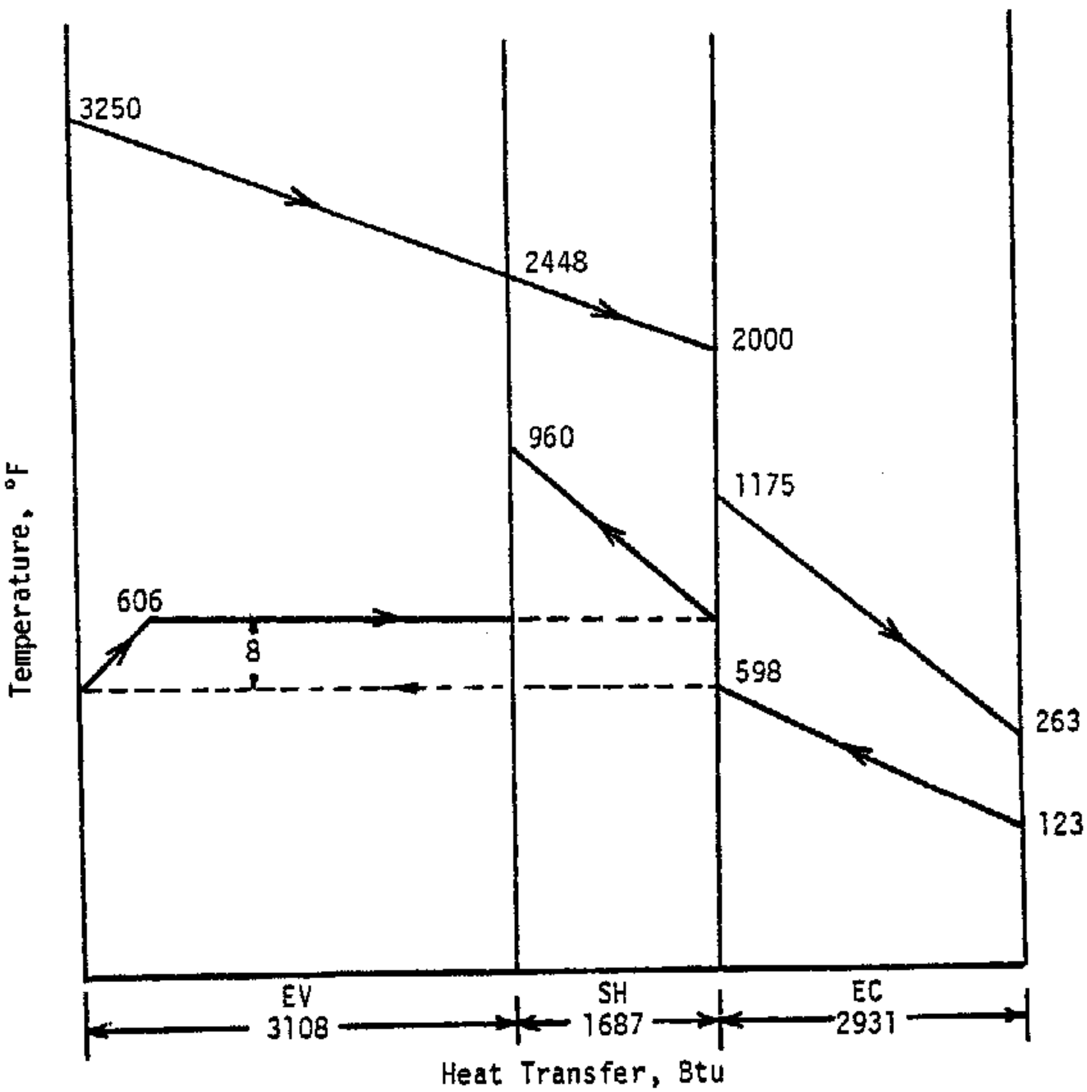

Figure 5.2-2 Temperature-Heat Transfer Diagram for Supercharged Boiler (Typical Results from Configuration 2)

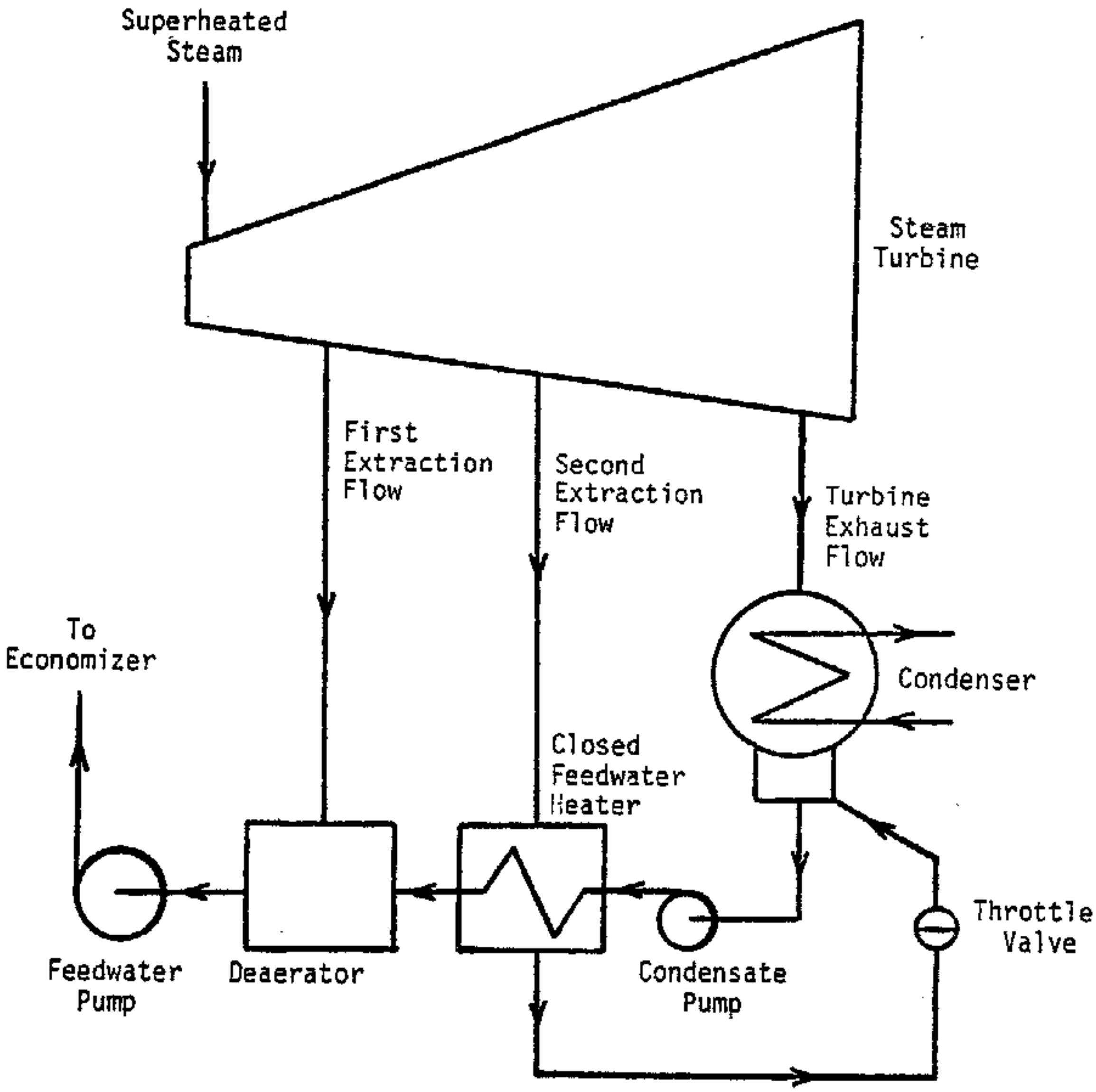

Figure 5.3-1 Steam Cycle Schematic with Regenerative Feedwater Heaters

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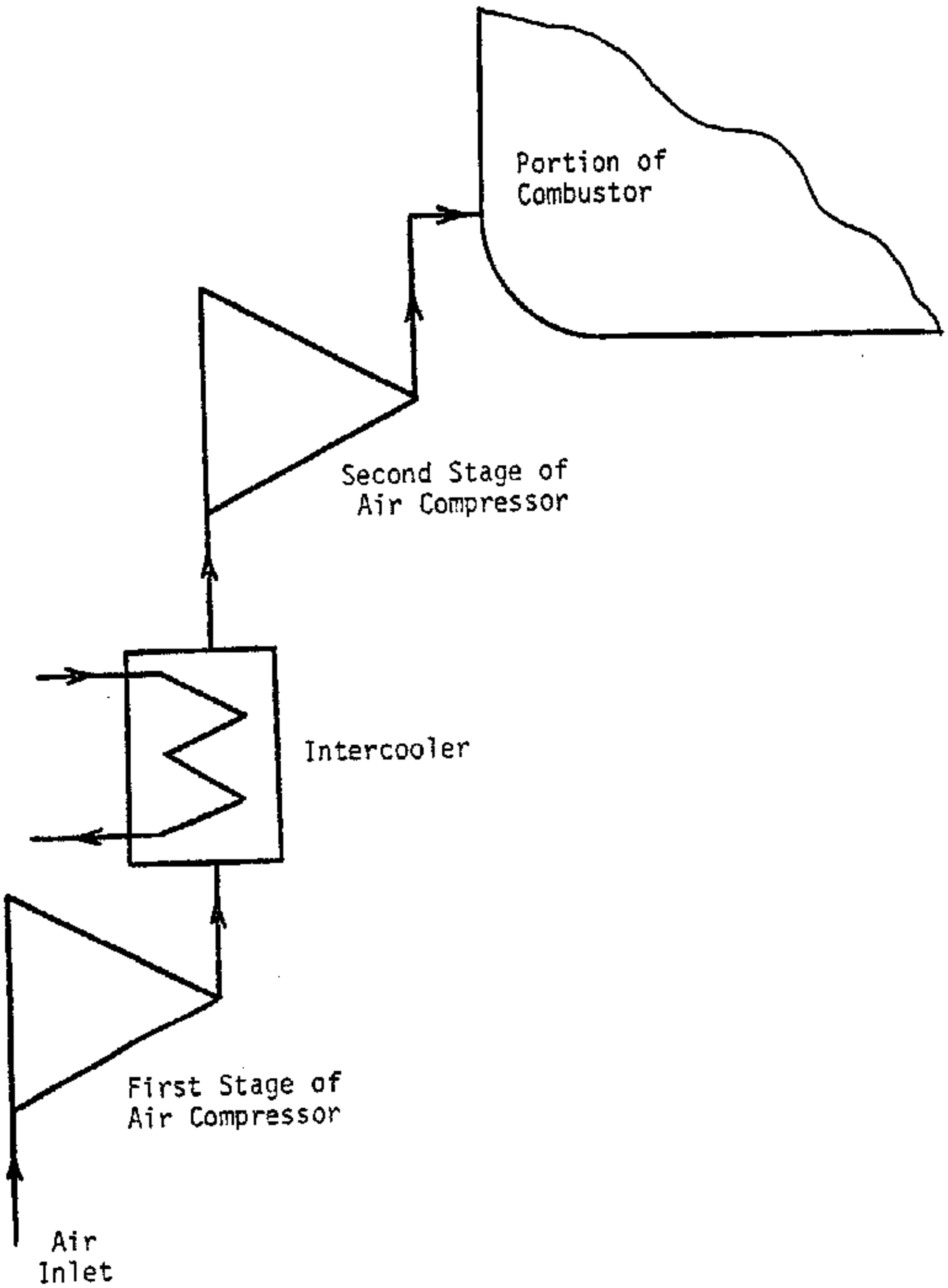

Figure 5.3-2 Schematic of Two-Stage Intercooled Air Compressor Serving Combustor

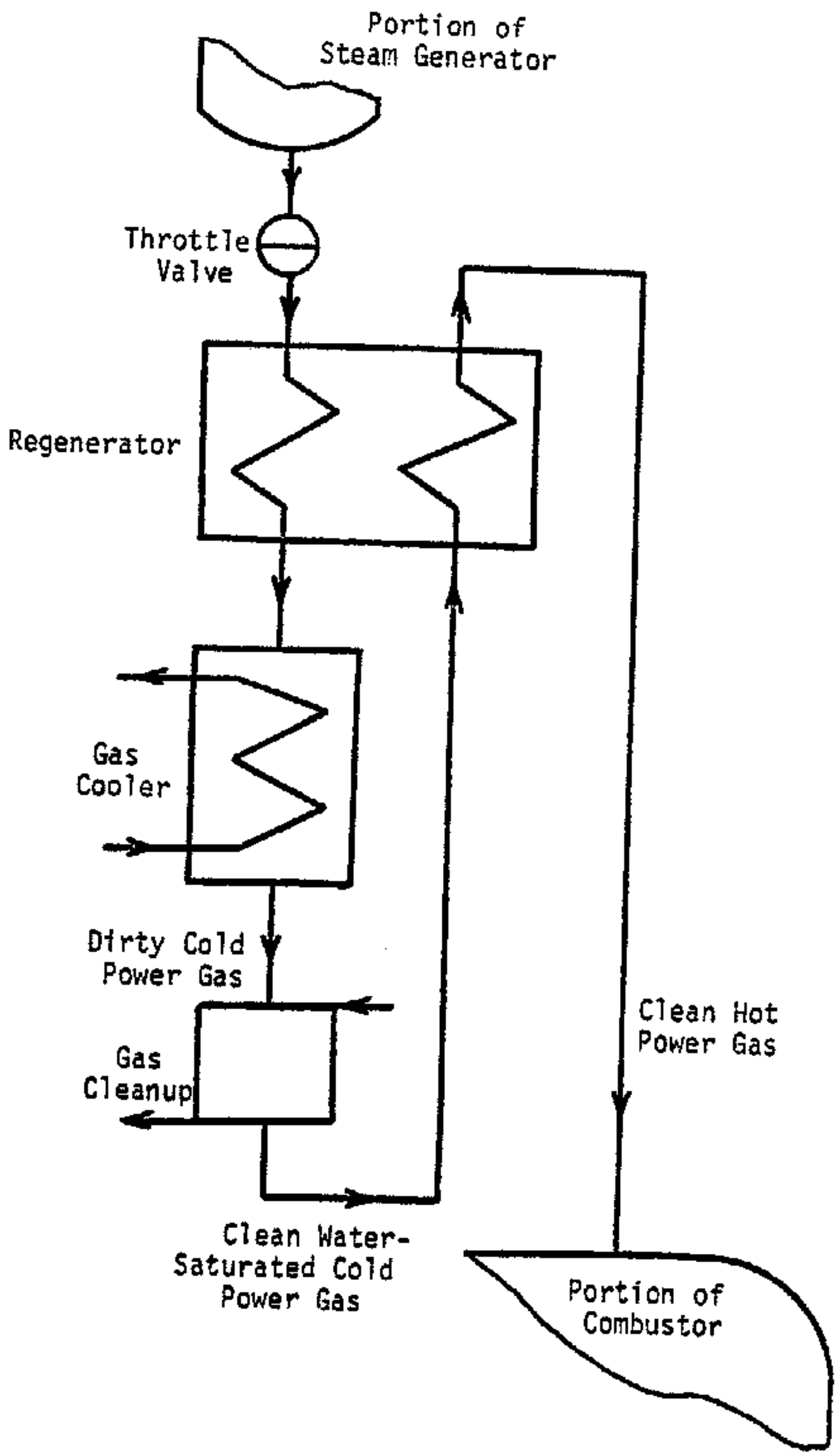

Figure 5.3-3 Schematic of Portion of System from Steam Generator to Combustor with Regenerator within Gas Cleanup System

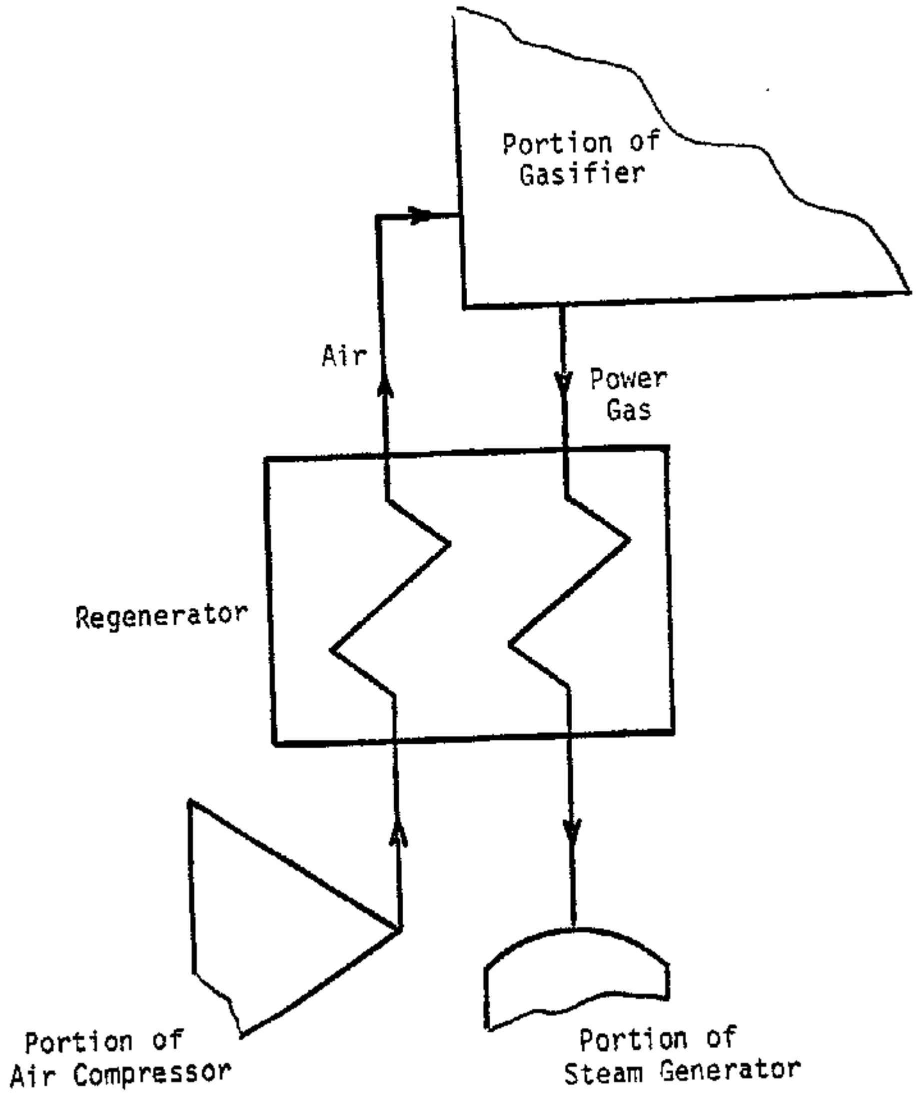

Figure 5.3-4 Schematic of Portion of System from Air Compressor Serving Gasifier to Steam Generator with Regenerator Included in System (Applicable to Configurations 1 and 2 Only)

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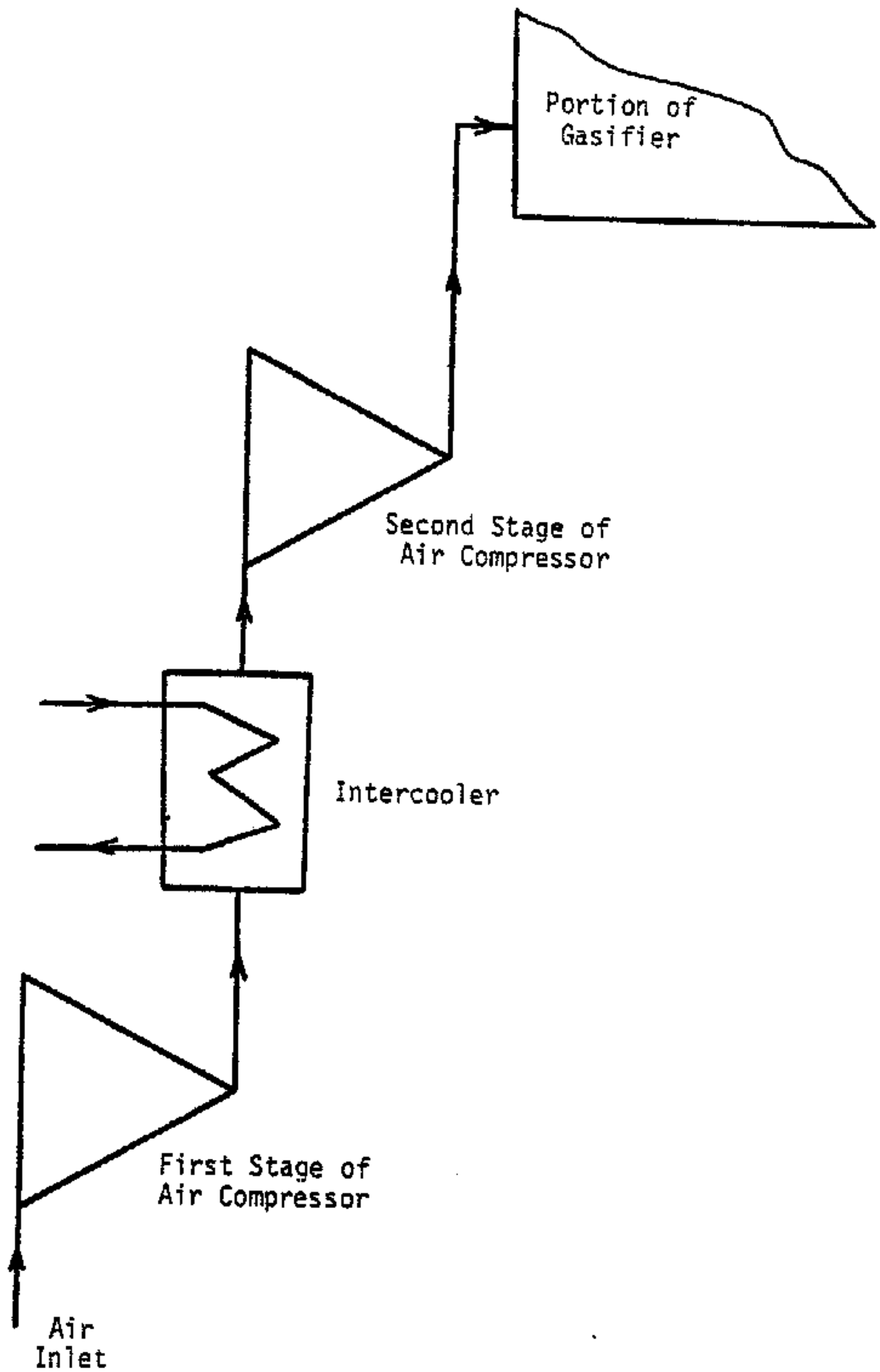

Figure 5.3-5 Schematic of Two-Stage Intercooled Air Compressor Serving Gasifier (Applicable to Configurations 1 and 2 Only)

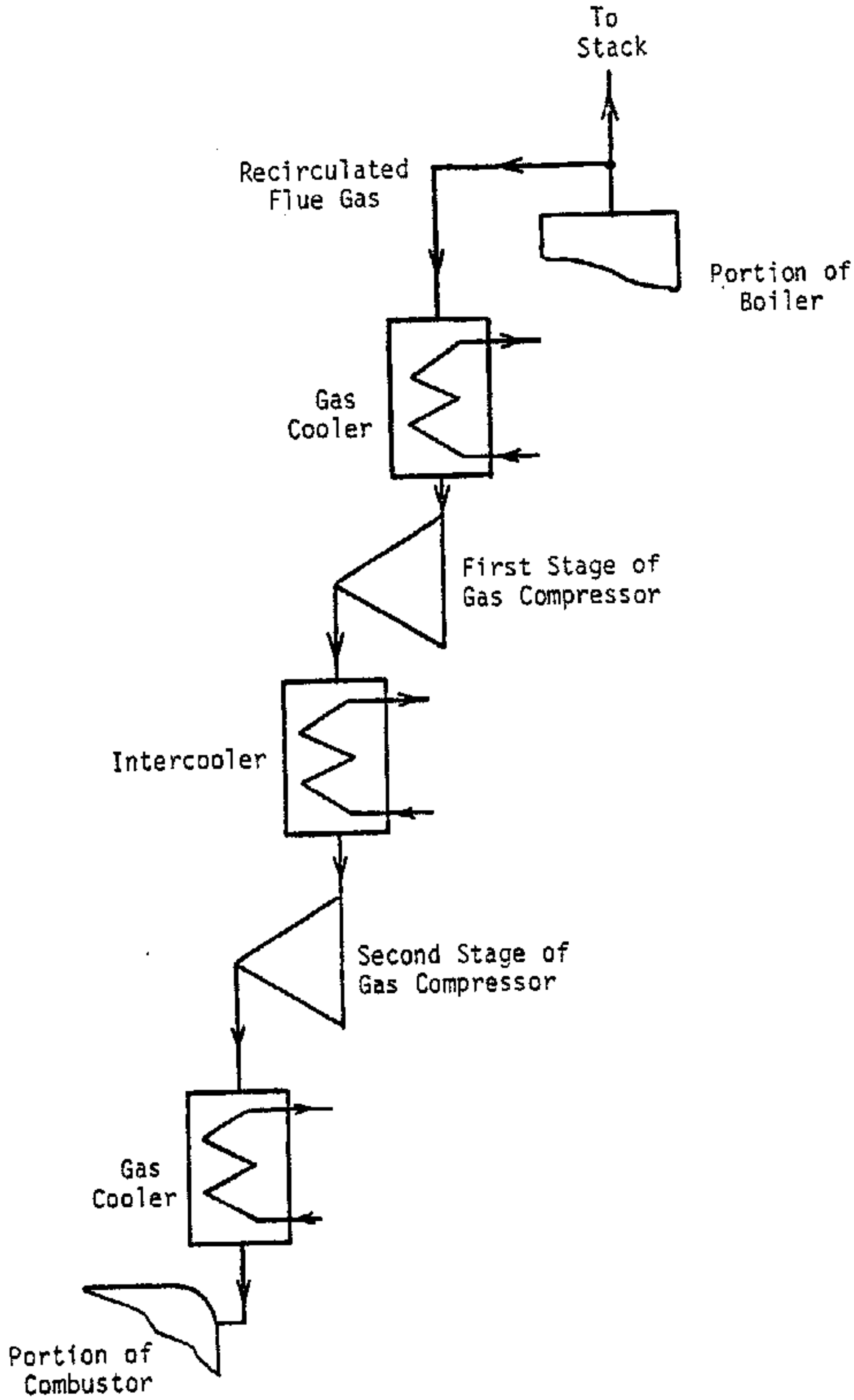

Figure 5.5-1 Schematic of Portion of System from Stack to Combustor Showing Flue Gas Recirculation Train

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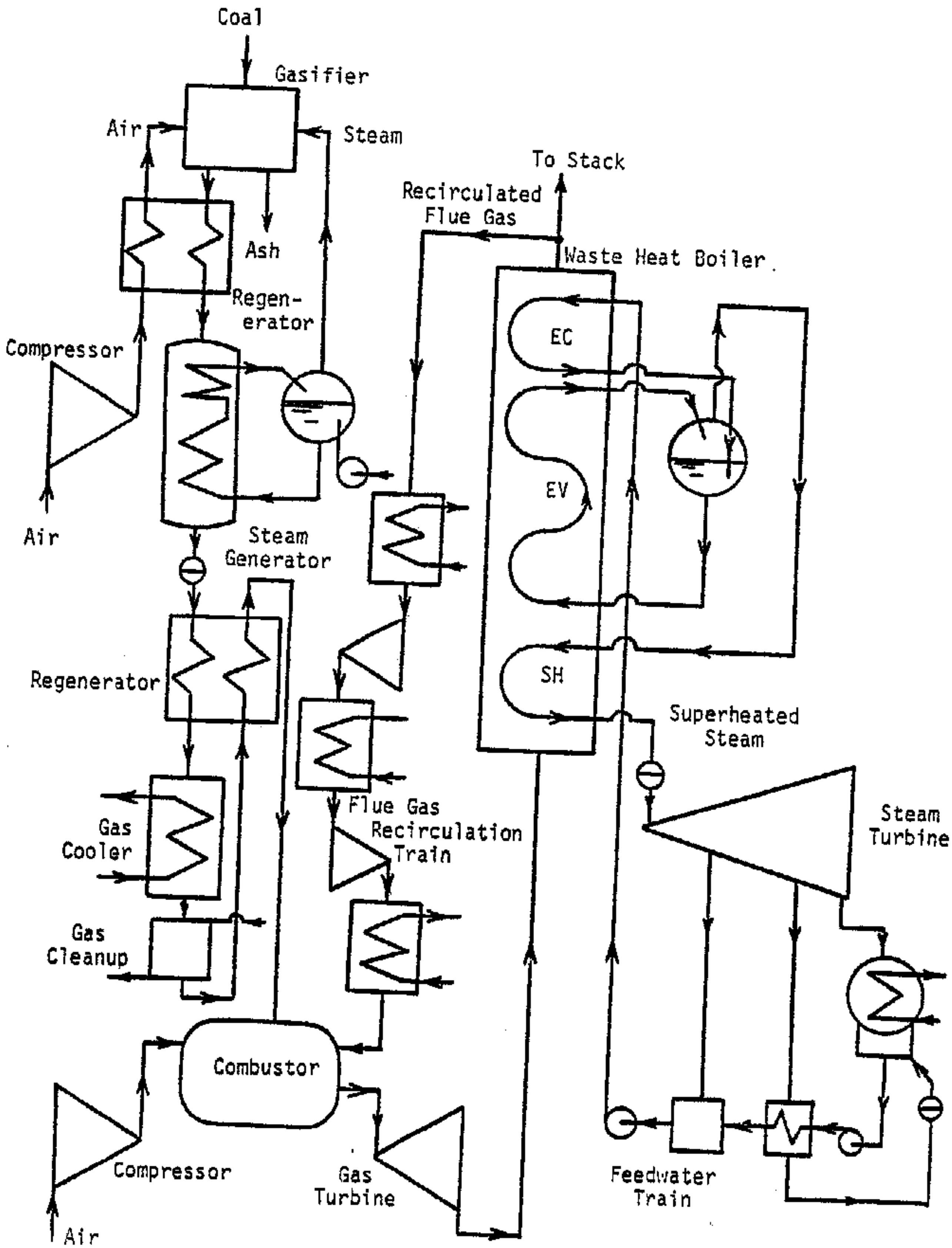

Figure 5.6-1 Schematic of Final Version of Configuration 1

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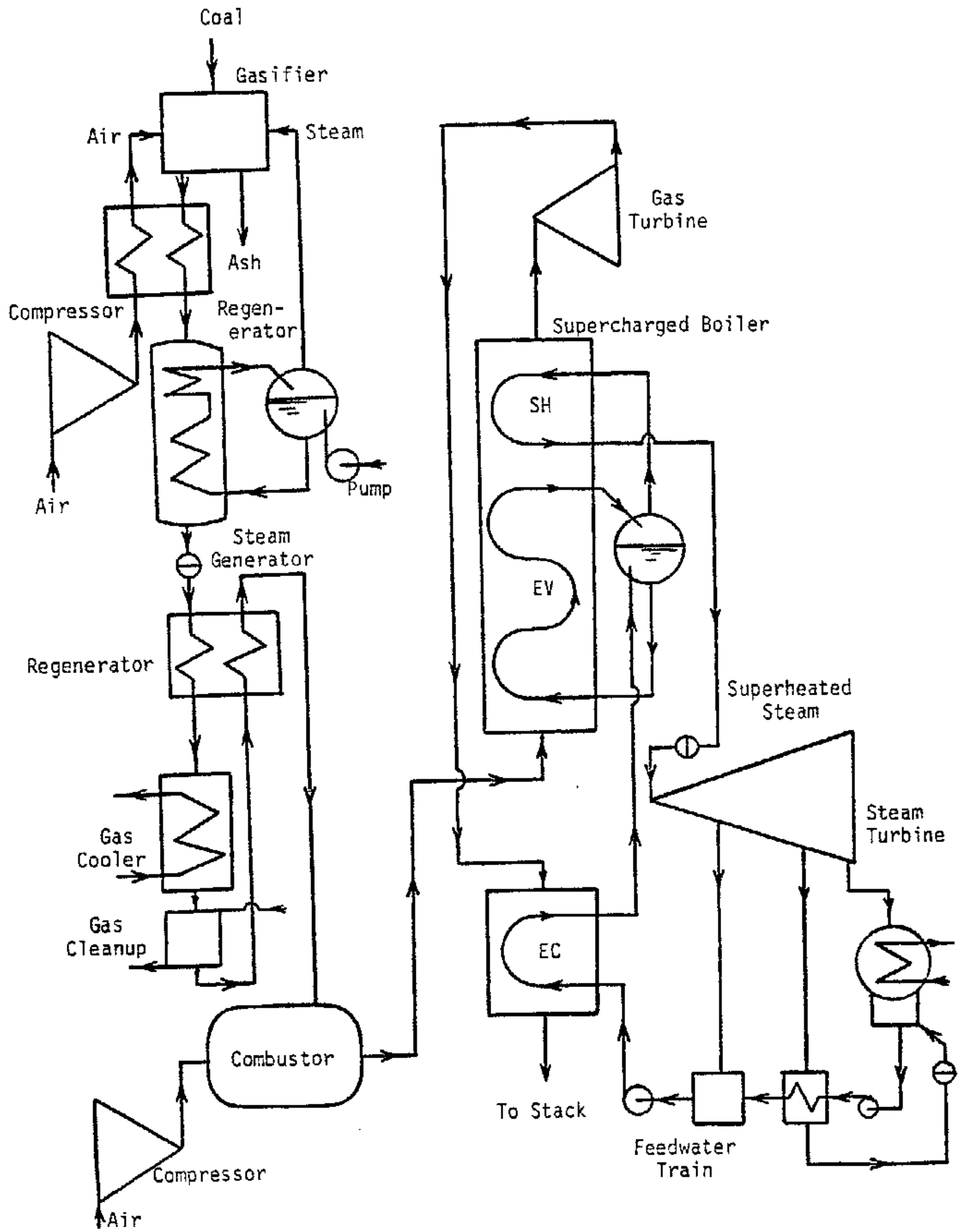

Figure 5.6-2 Schematic of Final Version of Configuration 2

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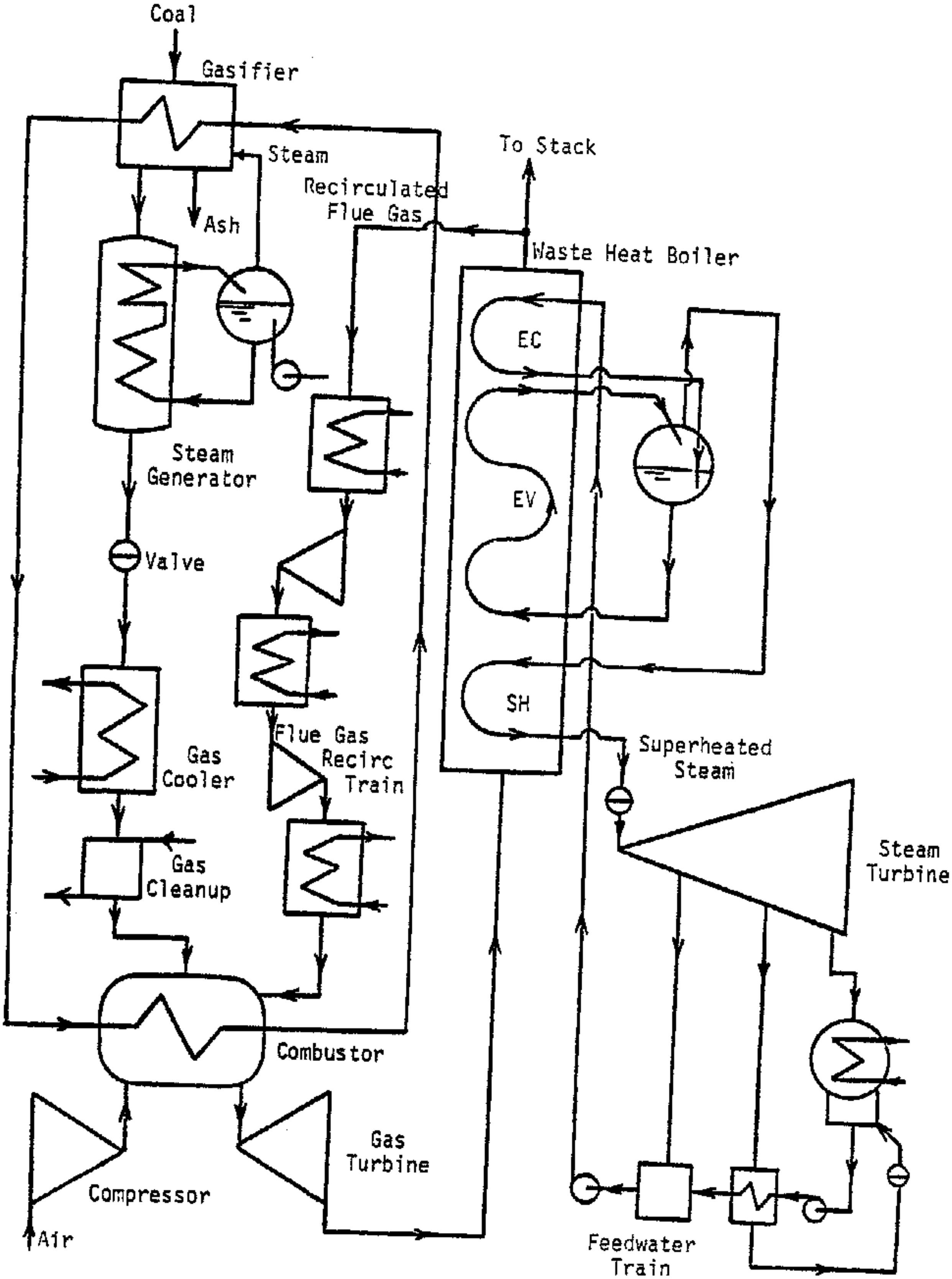

Figure 5.6-3 Schematic of Final Version of Configuration 3

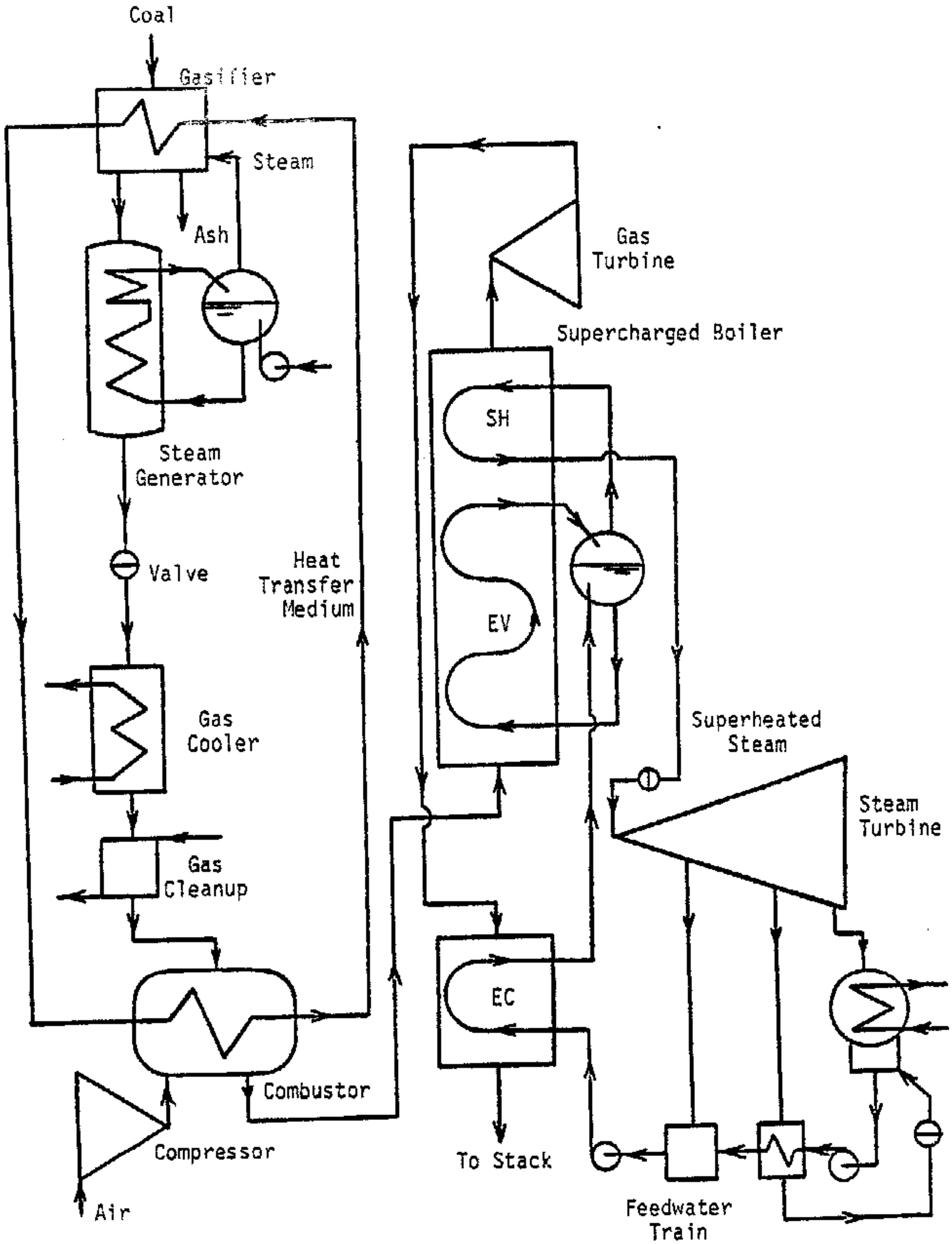

Figure 5.6-4 Schematic of Final Version of Configuration 4

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS FOR FURTHER STUDY

Several conclusions may be drawn from the results presented in the previous chapter. First and foremost is the result of the optimization with respect to both components and operating conditions with consideration of the emission criterion on nitric oxides. As summarized in Table 5.6-2, Configuration 1 results in the best station efficiency but only by a marginal amount. In fact, the difference between the highest and lowest station efficiencies is less than 1 percentage point. If the nitric oxide problem that exists on the configurations which utilize waste heat boilers could be solved without the use of flue gas recirculation, then these configurations could have higher station efficiencies by as much as five percentage points over those incorporating supercharged boilers. All of these comments depend, of course, on the validity of the 2400°F freeze temperature for the NO producing reaction.

While Configuration 1 is marginally the best performer, it does require the most equipment. Further complicating the trade-offs which must be made concerning the search for the best configuration is the relatively small consumable water requirement for Configuration 1. In general, the configurations employing endothermic gasifiers require about twice as much total water as those incorporating adiabatic gasifiers. Finally, the waste heat configurations reject a much smaller amount of heat from the steam cycle compared to that of the supercharged boiler configurations. This could result in substantial capital cost savings if cooling towers could be eliminated on the waste heat configurations.

It is instructive to compare these results with the station efficiency of a conventional coal-fired plant with stack gas scrubbers. Osterle, Impink, et al.[17] calculate the station efficiency of a coal-fired plant under assumptions very similar to those made in this work to be 37.5 percent, without consideration of the penalty from the stack gas scrubbers. Rubin[30] presents data which shows that about 2 1/2 percentage points should be subtracted from the above station efficiency to include the energy requirements of the scrubbers. Therefore, the station efficiency of 36.67 percent for Configuration 1 is slightly better than that of 35.0 percent for a conventional coal-fired plant. Configuration 1 appears to be significantly better than a nuclear plant, the station efficiency of which is usually given as 33 percent. In terms of heat rates, these station efficiencies correspond to 9307, 9750, and 10,300 But/kwhr for Configuration 1, the coal-fired plant, and the nuclear plant respectively.

As already mentioned, it appears that better performance can be expected when the amount of work produced by the gas cycle is a high fraction of the total work. This is a characteristic of the configurations employing a waste heat boiler. Unfortunately, these same configurations are unacceptable with respect to nitric oxide emissions. When flue gas recirculation is used as a means to reduce the amount of this effluent, the station efficiency decreases substantially to very nearly the values of station efficiency for the supercharged boiler configurations.

Regenerative feedwater heating in the steam cycle portion of a combined cycle results in a deterioration of plant performance. While the steam cycle efficiency improves, the station efficiency does not. As seen in Section 5.3, the decrease in heat rejection from the steam cycle through the condenser is smaller than the increase in the gas cycle heat loss through the stack. Feedwater heating raises the final feedwater temperature. A higher stack gas temperature, of course, results in a higher sensible heat loss through the stack. It should be noted that this conclusion is a result of the concept of a combined cycle and does not apply to a conventional fossil-fueled power plant. In the latter, feedwater heating does improve the plant performance significantly. Several reasons have been identified, however, which make some feedwater heating desirable. In a combined cycle only a minimum number of feedwater heaters should be used.

For the configurations employing an adiabatic gasifier it appears that some kind of heat recovery system is necessary beyond that of the steam generator. A gas-to-gas counterflow heat exchanger between the gas streams to and from the cleanup process is seen to improve station efficiency more than 3 percentage points if the device is 80 percent effective. Increasing the temperature of the steam to the gasifier in the endothermic configurations allows the elimination of this regenerator. In these configurations the steam generator is capable of reducing the gas temperature to the proper level required by the cleanup process. It should also be noted for the adiabatic configurations that the regenerator near the gas cleanup system is much more important than the one between the air and gas streams to and from the gasifier respectively. In fact, with the former in service at an effectiveness of 0.80, the latter may be removed completely with the station efficiency decreasing less than 1/2 percentage point.

With the exception of Configuration 3, the effective gasification temperature may be chosen to be higher than that which results in the optimum station efficiency. The advantages of this are faster reaction rates, a lower steam requirement, and conditions which are more conducive to attaining chemical equilibrium. Since the heat source for the endothermic gasification required by Configuraion 3 is at 2000°F, the effective gasification temperature must necessarily be below this. The effect of gas cycle pressure on station efficiency is much larger than that of the effective gasification temperature. Although the temperature of the steam to the gasifier hardly affects the plant performance, proper specification of this parameter for the endothermic configurations does result in the saving of expensive equipment, namely a regenerator.

In all cases, the use of intercooled air compressors does not appear to be justified. The station efficiencies either dropped slightly or remained the same when this modification was made to each configuration.

Fortunately, each configuration is fairly insensitive to different types of coal. When three other types of coal were used in the analysis, the station efficiency hardly changed. Although this study was shown for Configuration 1 only, similar results are obtained for the other configurations. Boiler pinch point temperature differences, too, are unimportant. Halving the 40°F minimum pinch point temperature difference results in a relatively small improvement in station efficiency with the largest increase of slightly more than 1/2 percentage point occurring on Configuration 1.

The inefficiencies associated with the cycle components, particularly the turbines and compressors, play a major role in reducing plant performance. For the hypothetical case of ideal components, the station efficiency would increase more than 6 percentage points. The pressure drops apparently play a much smaller role in determining station efficiency.

Finally, it was seen that the real success of the combined cycle concept integrated with a coal gasifier depends on the attainment of the 2400°F gas turbine inlet temperature. It was shown for Configuration 1 that an increase of almost exactly 5 percentage points would result, after consideration of the pollution criteria. The use of flue gas recirculation to control the production of NO on the waste heat configurations results in decreasing the station efficiency about 3 percentage points.

One obvious extension to the above work is the task of sizing the equipment necessary to obtain a specified electrical power output, say 500 MWe. The results of this could be used for two further studies: an economic study where the trade-offs could be assessed quantitatively between reduced station efficiency on one hand and reduced capital, operating, and maintenance costs on the other; and a transient study where the controls for the best configuration with respect to both performance and economics could be developed. The question of load follow capability could be addressed more appropriately during the course of the design of the control systems. The economic and technical feasibility of gas storage for later use either as a fuel for a combined cycle or as a chemical feedstock for some other process could also be determined.

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