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An Assessment of Advanced Technology for Industrial Cogeneration

N. Moore

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Southern California Edison Company
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ABSTRACT

The potential of advanced fuel utilization and energy conversion technologies to enhance the outlook for the increased use of industrial cogeneration was assessed. The attributes of advanced cogeneration systems that served as the basis for the assessment included their fuel flexibility and potential for low emissions, efficiency of fuel or energy utilization, capital equipment and operating costs, and state of technological development.

Over thirty advanced cogeneration systems were evaluated. These cogeneration system options were based on Rankine cycle, gas turbine engine, reciprocating engine, Stirling engine, and fuel cell energy conversion systems. The alternatives for fuel utilization included atmospheric and pressurized fluidized bed combustors, gasifiers, conventional combustion systems, alternative energy sources, and waste heat recovery. Two advanced cogeneration systems with mid-term (3 to 5 year) potential were found to offer low emissions, multi-fuel capability, and a low cost of producing electricity. Both advanced cogeneration systems are based on conventional gas turbine engine/exhaust heat recovery technology; however, they incorporate advanced fuel utilization systems. One system features a pressurized, fluidized-bed-combustor that is integrated with the gas turbine engine, while the other system incorporates a pressurized, air-blown gasifier that is also integrated with the gas turbine engine. Other advanced cogeneration options with long-term potential include the fuel-cell-based systems, but they require significant improvements in performance and/or costs to reach commercial viability.

PREFACE

This report documents work performed by the Jet Propulsion Laboratory (JPL), California Institute of Technology, under contract Purchase Order No. C2742911 with the Southern California Edison Company (Rosemead, California) through the National Aeronautics and Space Administration (NASA Task RE-152 Amendment 358). The work done under the contract was performed between November 1982 and May 1983.

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SECTION 1

INTRODUCTION AND SUMMARY

1.1 THE CONCEPT OF INDUSTRIAL COGENERATION

Cogeneration is the practice of simultaneously providing thermodynamic work and useful thermal energy from the same fuel or heat source. Typically, the thermodynamic work output of a cogeneration system is shaft power or electricity, and the thermal energy output is steam or hot water at a temperature high enough to be used in industrial processes or other applications. Until the early 1900s, cogeneration was a matter of necessity for many industrial plants. However, with the advent of reliable, relatively inexpensive electricity from utility companies, the practice of cogeneration was largely dropped because industry found utility-purchased electricity to be more economical. Of course, industry continued to produce heat for use in manufacturing processes, largely in the form of low to medium pressure steam. A renewed interest in cogeneration stems from the rising cost of fuel oil and natural gas which encourages their efficient utilization, the substitution of less expensive types of fuels, and the use of alternate energy sources.

The overall efficiency gain resulting from cogeneration occurs because the electricity produced by cogeneration systems displaces an equivalent amount of electricity produced by a conventional utility power plant. The higher overall energy efficiency attributable to cogeneration is not the result of poor fuel utilization or inefficient operation by either conventional industrial boilers or utility power plants. Due to fundamental thermodynamic limitations, utility power plants convert only about 40% of the fuel energy into electricity and must discharge the remaining 60% in the form of heat into cooling towers or other heat sinks. In order to convert as much fuel energy as possible into electricity, the temperature at which the power plants discharge heat must be as low as possible. Cogeneration systems circumvent the problem of achieving a high electrical conversion efficiency by simultaneously producing electricity and heat with a combined efficiency of about 75%, in contrast to power plants that produce only electricity at an efficiency of about 40%. The overall efficiency of energy utilization of typical cogeneration systems is about the same as that of conventional industrial boilers -- about 70% to 80%. An appreciable overall energy savings, considering the fuel consumed by both the utility and the cogenerator, is obtained by cogeneration only if the heat recovery and electrical generating efficiencies of the cogeneration system are sufficient.

In order for cogeneration to be implemented by an industry, the overall cost of cogenerated electricity and heat must be less than the combined costs of purchased electricity and heat produced by conventional systems. From the utility perspective, the cost of production of cogenerated electricity must be competitive with that of conventional power, considering the value of the cogenerated heat. This report presents an assessment of the potential of advanced technology, within the study guidelines, to enhance the economic outlook for increased use of industrial cogeneration. The attributes of advanced cogeneration systems that served as the basis for the assessment include their fuel flexibility and potential for low emissions, their efficiency of energy utilization, their initial capital and operating costs, and their technological state of development and developmental risk. The cogeneration system performance and costs were then used to calculate a representative cost of electricity for each system. These costs of electricity are for relative comparison only and are not intended to illustrate the actual cost of electricity produced by cogeneration.

1.2 STUDY OBJECTIVE, APPROACH, AND GUIDELINES

The major objective of this study is to identify research activities that have the potential to increase substantially the use of industrial cogeneration in the Southern California area. This objective was accomplished through the study approach shown in Figure 1-1. A review of past and ongoing research activities in energy conversion, fuel utilization and cogeneration technology provided the basis for developing a list of advanced cogeneration system options and applicable research areas. These options are presented in Section 1.3. The characteristics of both the advanced and presently available systems were derived from information published in the open literature by means of consistently applied methods to insure that the attributes of the cogeneration system options could be meaningfully compared. These advanced cogeneration system options were compared with presently available cogeneration systems in terms of their attributes and performance characteristics as expressed in terms of the cost of electricity produced by a cogeneration system. The cost of electricity produced by a cogeneration system is subsequently discussed in Section 2.2 of this report. The advanced cogeneration system characteristics that served as the basis for the findings and recommendations of this study included fuel flexibility and the potential for low emissions, technological state of development and developmental risk, the future time frame in which a system might be commercially available, and the cost of electricity produced by the advanced cogeneration system. The recommendations thus derived are given in Section 1.5; the findings with regard to the cogeneration system options are presented in Section 1.4; and a comparative evaluation of the cogeneration system options in the above respects is contained in Section 5.

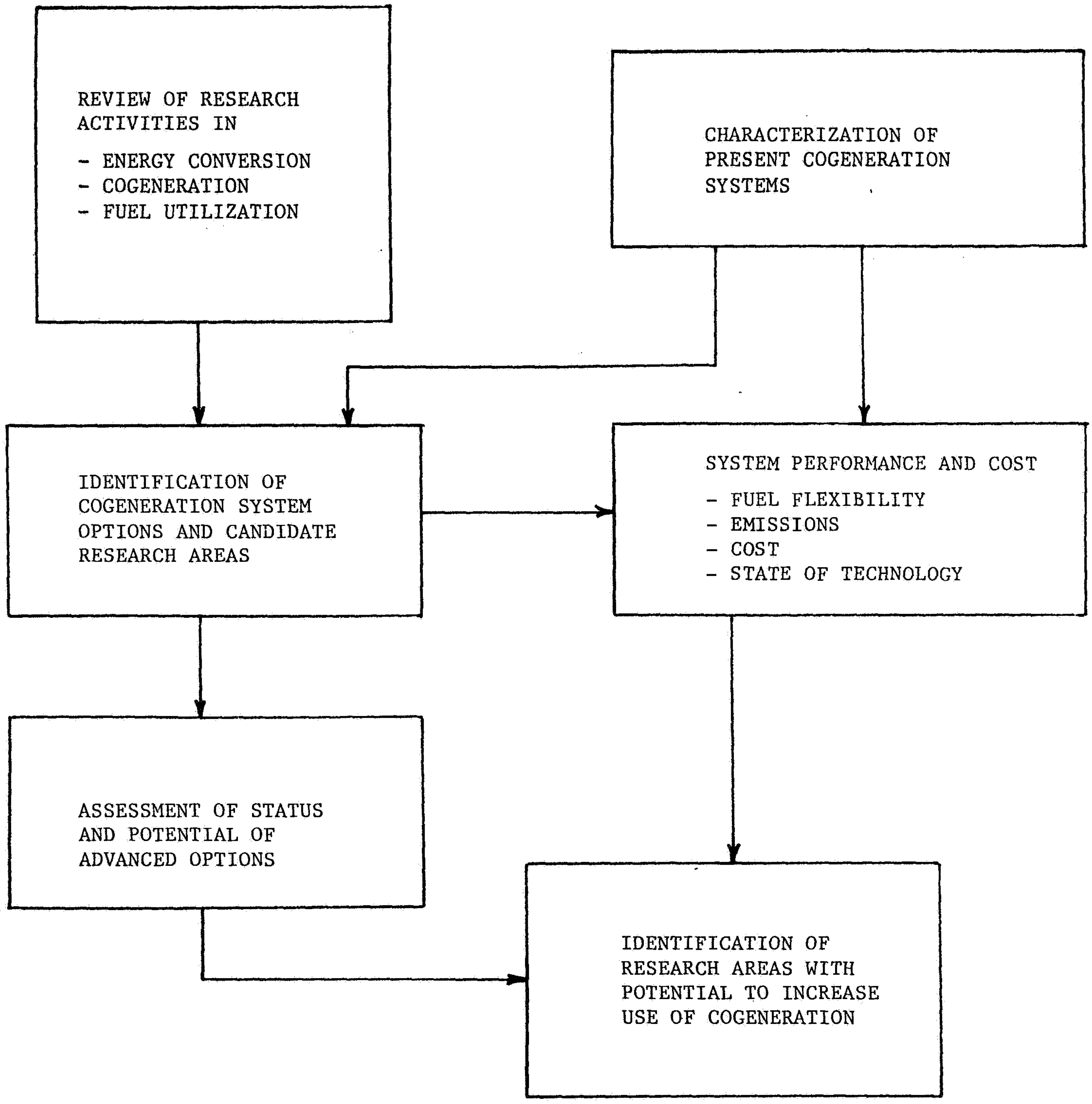


Figure 1-1. Approach Used to Identify Cogeneration Research Areas for Emphasis

A major guideline that directly influenced the outcome of this study is that the effort would be focused on advanced cogeneration system options that make use of advanced energy conversion technology as it develops, but no research activities that would directly pursue advanced energy conversion technology per se would be considered for the study recommendations. For example, research on fuel cells and ceramic heat engines would not be considered in formulating the study recommendations. Such basic energy conversion technology research programs require long-term committment for sponsorship at substantial funding levels and are being pursued by the Electric Power Research Institute and by federal government agencies, including the Department of Defense and the Department of Energy. Additional study guidelines were that realistic emissions standards would be considered, that the study would address both topping and bottoming cycles, and that the capital cost and the performance characteristics of the advanced systems would be expressed in terms of the cost of electricity as defined in Section 2.2. The costs of electricity for the advanced cogeneration systems as given in this report are only for relative comparisons and do not reflect the actual cost of cogenerated electricity. The final guideline specified that in addition to the conventional fossil energy sources (e.g., natural gas, petroleum and coal), the alternative energy resources including biomass, solar, refuse, and geothermal would be addressed.

1.3 COGENERATION OPTIONS CONSIDERED

The cogeneration system options that are addressed in this report are listed in Table 1-1. The presently available systems shown in Table 1-1 serve as a baseline of reference against which to compare the advanced cogeneration systems. The presently available cogeneration systems consist of components and prime movers that are commercially available from equipment manufacturers and vendors. These systems are treated in detail in Section 3 of this report where their system characteristics and cost of producing electricity are given.

Advanced cogeneration system options that are addressed in this study are listed in Table 1-1 beneath the type of energy conversion system on which they are based, or, in the case of the alternative energy systems, according to the type of alternative energy source. The advanced cogeneration system options addressed in this study were identified by means of a comprehensive survey of energy conversion system and cogeneration technology and are intended to include those energy conversion system options that have potential to find use in industrial cogeneration systems in the foreseeable future. The study was directed toward identifying those advanced technologies that would enable both the developing energy conversion systems and those in a relatively mature state of development to find more general use in industrial cogeneration applications. Hence, for the purpose of the study, the energy conversion systems, i.e., the prime movers, were presumed to exist in representative configurations that served as the common elements among the cogeneration options for each type of energy conversion system. The cogeneration options within the energy conversion system categories are based on either the mode

Table 1-1. Cogeneration Systems Addressed in the Study

PRESENTLY AVAILABLE SYSTEMS

Baseline Systems

- Rankine Topping
- Rankine Bottoming (Organic Working Fluids)
- Reciprocating Engine
- Gas Turbine Engine

ADVANCED COGENERATION SYSTEMS

Rankine Cycle Systems

- Topping Cycles
  - Atmospheric Fluidized Bed Combustion
  - Alternative Expanders
  - Alternative Energy Sources
- Bottoming Cycles

Gas Turbine Engine Systems

- Pressurized Fluidized Bed
- Integrated Gasification
- Indirectly Fired
- Direct Coal-Fired

Advanced Reciprocating Engine Systems

- Gasification
- Direct Coal-Fired

Table 1-1. Cogeneration Systems Addressed in the Study (cont'd)

ADVANCED COGENERATION SYSTEMS (cont'd)

Stirling Engine Systems

- o Conventional
- o Fluidized Bed

Fuel Cell Systems

- o Natural Gas/Naphtha Reforming
- o Fuel Oil Reforming
- o Gasification

Alternative Energy Sources

- o Solar
- o Biomass
- o Geothermal
- o Wind

of fuel utilization or the type of energy source. Cogeneration systems based on present technology serve as a baseline of reference against which to compare the advanced systems.

The advanced cogeneration system options that were analyzed in detail in the study are shown in Table 1-2. The matrix of Table 1-2 shows the types of energy conversion systems that were adapted to a given mode of fuel utilization or energy source and, conversely, the applicability of a particular energy source or mode of fuel utilization to different energy conversion systems. The analysis of these systems included the identification of major components, an assessment of their technological status, a thermodynamic analysis to estimate cogeneration system performance, and projections of installed capital costs. The estimates of performance and costs were then used to calculate a representive cost of electricity for each cogeneration system option. The projections for installed capital costs for the advanced cogeneration systems are direct costs and do not include interest on funds during construction, contingencies, and other indirect costs that can significantly increase the cost of acquiring and owning a cogeneration system. Therefore, the costs of electricity for the cogeneration systems are only for relative comparisons and do not reflect the actual costs of cogenerated electricity.

Table 1-2. Matrix Depicting Cogeneration System Options Analyzed in the Study (X indicates combination analyzed)

Energy Source/ Mode of Utilization	Energy Conversion System: Rankine Cycle Topping	Rankine Cycle Bottoming	Gas Turbine Engine	Reciprocating Engine	Stirling Engine	Fuel Cell
Fuel Oil (Distillate)	X[h]		X[h]	X[b]	X[h]	X[e]
Natural Gas				X[c]		X
Direct Coal	X			X[d]		
Gasification (Multi-fuel)			X[f]	X[b,f]		X[e,g]
Atmospheric Fluidized Bed (Multi-fuel)	X		X[a]		X	
Pressurized Fluidized Bed (Multi-fuel)			X			
Process Waste Heat		X				
Solar	X					

[a]The gas turbine engine was analyzed in an additional indirectly-fired configuration not shown in this table; see Section 4.2.2.
[b]Reciprocating engines were analyzed in a conventional version and in an advanced, uncooled version.
[c]Conventional reciprocating engine only.
[d]Advanced, uncooled, reciprocating engine only.
[e]Fuel cells of both the phosphoric acid type and the molten salt type were analyzed.
[f]Air-blown gasifier only.
[g]Both air-blown and oxygen-blown gasifiers were analyzed.
[h]Dual fuel system for both natural gas and distillate.

1.4 STUDY FINDINGS

In general, it was found that the advanced industrial cogeneration systems that rank more favorably in terms of their costs of electricity make use of inexpensive fuel such as biomass, coal or refuse, providing that the additional capital equipment necessary to use such fuel is not excessively costly. The specific findings of this study of advanced industrial cogeneration systems are given below for each of the advanced cogeneration system types as listed in Table 1-1. The recommendations for research are subsequently presented in Section 1.6.

Rankine Cycle Systems. The advanced Rankine topping cycle cogeneration system was found to have a relatively high cost of electricity (COE) and to have little potential for achieving a lower COE through a capital cost reduction or energy efficiency improvement resulting from technology development activities within the guidelines of this study. Likewise, Rankine bottoming cycle systems were found to have a high COE with little potential for reduction. Bottoming cycle systems must be evaluated vis-a-vis recuperative recovery of waste heat for preheating process streams or boiler feed water; and, typically, recuperative waste heat recovery yields a more favorable economic return.

Gas Turbine Engine Systems. The directly-fired gas turbine engine (GTE) cogeneration systems have costs of electricity that are among the lowest of the industrial cogeneration systems due to their moderate capital cost and high overall energy efficiency. The preferred GTE systems are directly-fired by means of a pressurized fluidized bed combustor (PFBC) or pressurized air-blown gasifier (PABG) to enable the use of fuels such as coal, biomass, and refuse. Both the PFBC and the PABG systems have the potential for low emissions, considerably bettering the current federal standards, but perhaps having some difficulty with the most stringent California standards that are under consideration, e.g., oxides of nitrogen emissions of less than 0.1 lb per million Btu's of fuel fired. Post combustion flue gas treatment might be required at the most stringent emissions levels. The directly-fired GTE systems could be available for service in the mid-term time frame (1985-1995) after a nominal development effort for the PABG and/or the PFBC units and their integration with the GTE.

The direct firing of pulverized coal in GTEs has been investigated over the past three decades in the United States and Australia. Generally, turbine blade erosion and corrosion limit turbine hot section life to unacceptably low levels. Directly-firing pulverized coal in a GTE would require the use of flue gas desulfurization systems and some form of post-combustion NO_X reduction. In view of these factors, directly firing a GTE with pulverized coal is not considered feasible for industrial cogeneration applications.

The indirectly-fired gas turbine systems have relatively high costs of electricity due to higher capital costs and lower energy use efficiency than the directly-fired gas turbine systems. The high temperature heat exchanger required in the indirectly-fired systems limits the turbine inlet temperature so that engine efficiency and specific power are reduced. Thus, in addition to the cost of the indirectly-fired burner itself, the gas turbine engine must be larger for a given power output and, therefore, more costly. The energy losses associated with the furnace/high temperature heat exchanger unit reduce the system's energy use efficiency, further increasing the cost of electricity.

Advanced Reciprocating Engine Systems. The advanced reciprocating engine cogeneration systems were all found to have relatively high costs of electricity due to their higher capital cost and limited overall efficiency of energy use. The uncooled, ceramic reciprocating engine does have a significant improvement over the conventional reciprocating engine in that the heat available from the exhaust gas stream is substantially increased. Even with this improvement, however, the reciprocating engine cogeneration system suffers from a comparatively higher heat loss which reduces its overall efficiency of energy usage. The capital cost of the advanced, uncooled, ceramic reciprocating engine cogeneration system is conjectural at this juncture, but was assumed to be that of the conventional engine for the purpose of calculating a cost of electricity. The COE thus calculated for an advanced, uncooled reciprocating engine in which coal is directly burned was found to be in the higher range. Considering that post-combustion flue gas cleanup would be required at an additional capital cost to meet expected emissions requirements, the uncooled, ceramic reciprocating engine is not among the systems of first choice for industrial cogeneration. However, if development of the ceramic reciprocating engine proceeds favorably, this system should be re-examined for industrial cogeneration applications, particularly in the directly coal-fired version.

Stirling Engine Systems. Stirling engine cogeneration systems were found to have a high cost of electricity even with comparatively optimistic assumptions regarding its capital cost. The Stirling engine cogeneration system is indirectly fired, as mandated by the engine's basic configuration, and suffers the efficiency and cost disadvantages associated with the indirect-firing of heat engines. Stirling engines in the size range

range required for industrial cogeneration systems are presently in the conceptual design stage of development. In view of the higher projected capital cost of the Stirling engine system, particularly in the solid fuel burning versions, and its early stage of development in the size range required for industrial cogeneration systems, it is not among the preferred systems at the present time. However, the technological progress of Stirling engines should be monitored; and, as in the case of the uncooled reciprocating engine, the Stirling engine should be re-examined for industrial cogeneration applications as its technology develops.

Fuel Cell Systems. Fuel cell cogeneration systems in their mid-term state of development are projected to have relatively high costs of electricity. However, with the advent of lower capital costs achievable through higher rates of production, improved manufacturing techniques, improvements in fuel cell stack design and the use of lower cost materials, fuel cells could have COEs competitive with the better gas turbine systems examined in this study and at higher overall energy use efficiencies. The lowest cost fuel cell cogeneration system examined herein was a far-term configuration consisting of a molten salt fuel cell stack integrated with an air-blown gasifier. The mid-term, but higher cost system, consisted of a phosphoric acid fuel cell stack again integrated with an air-blown gasifier. Due to their high overall efficiency of energy use and low emissions, fuel cells are an attractive energy conversion system for industrial cogeneration applications. If fuel cell cogeneration system capital costs were competitive, they would be the preferred system in the long term, assuming that favorable projections of performance, durability, and reliability are realized.

Alternative Energy Systems. Energy conversion systems for utilizing the alternative energy resources -- biomass and/or refuse, wind, geothermal, and solar -- have received considerable attention over the past decade. Wind-powered cogeneration systems are generally not feasible due to resource availability and economic considerations. Of the alternative energy resources, only biomass and refuse were found likely to be utilized in industrial cogeneration applications in the near-term to mid-term time frame. The direct combustion or gasification of biomass, refuse or other carbonaceous material was presumed to be an option for the fluidized bed combustors and the solid fuel gasifiers considered in this study. Developmental work for such fuel utilization systems is required, and the delineation of the required effort should be undertaken.

Geothermal energy is usually available as relatively low-temperature brine. The use of geothermal heat to generate electricity by serving as a heat source for a low-temperature Rankine cycle power plant and by the direct expansion of geothermal brines are under active development. In general, the temperature of geothermal brine is too low for use in an industrial cogeneration system. A relatively hot geothermal well, at 270 to 350°F, could directly provide industrial process heat but would not be practicable for use as the high-temperature heat source in a cogeneration system. A limited class of industrial processes could use the heat that is rejected at 150 to 200°F from a geothermal power conversion cycle, but a detailed examination of such limited possibilities is outside the scope of this study.

Solar energy cogeneration systems based on point-focusing or line-focusing concentrators could be attractive in the far-term, but cost reductions in solar collector systems are required to achieve more general economic feasibility.

1.5 RECOMMENDATIONS FOR FURTHER WORK

The mid-term options for advanced industrial cogeneration include two systems with costs of electricity among the lowest found in this study. Both of these advanced cogeneration systems are based on directly-fired gas turbine engines with heat recovery boilers, which represent presently available technology. The technological advancement lies in the mode of fuel utilization by the directly-fired gas turbine engines. The lowest cost system features a pressurized fluidized bed combustor (PFBC) in which coal or other suitable solid fuel is burned. The second system, having a somewhat higher cost of electricity, features a pressurized air-blown gasifier unit also capable of handling coal or other suitable solid fuel. Both of these advanced cogeneration system options are capable of low emissions of nitrogen and sulfur oxides without post-combustion flue gas treatment; however, their capability to meet extremely stringent emissions standards that are much more restrictive than those now in effect without post-combustion treatment remains to be determined.

Air-blown gasifiers that operate at 8 to 10 atmospheres pressure and that handle both coal and other solid fuels would also be adaptable to the advanced cogeneration systems based on fuel cells. The fuel-cell-based systems available in the mid-term have fairly high costs of electricity, but fuel cell cogeneration systems have the potential for substantial cost reduction in longer term.

The findings of this study as synopsized above and as presented in Section 1.5, then, lead to the following recommendations for further work with the objective of generally increasing the utilization of industrial cogeneration systems through technological advancement:

(1) A detailed assessment of the applicability of pressurized fluidized bed combustors to gas turbine engines in the 1 to 15 megawatt range should be carried out. Such an assessment would include an in-depth investigation of alternative PFBC designs, their costs and performance, and their suitability for integration with the smaller gas turbine engines.

(2) A parallel study of air-blown gasifiers operating at 8 to 12 atmospheres pressure should also be performed. In general, gasification technology is quite mature, and several types of gasifiers are commercially available; but integration of a pressurized, air-blown gasifier with gas turbine engines in the size range under consideration here is not a developed technology. The recommended assessment of air-blown gasifier technology for advanced industrial cogeneration would focus on identifying a design for a high-efficiency, pressurized unit suitable for integration with gas turbine engines in the 1 to 15 megawatt range. The study should encompass unique methods of fuel sulfur removal, including in-situ sulfur capture, perhaps in a fluidized bed or moving bed gasifier; and, as with the PFBC unit, fuel feeding and ash removal systems for the smaller units should be given particular attention in a search for innovative improvements.

(3) The study of pressurized air-blown gasifiers should also consider the integration of such units with fuel-cell-based cogeneration systems. Fuel cells have much more stringent requirements for fuel gas composition than do gas turbine engines, and the effects of solid fuel composition on gas cleanup and carbon monoxide/hydrogen shift conversion must be considered.

(4) An in-depth assessment of the nitrogen oxide emissions characteristics of gas turbine engine combustion of low-Btu fuel gas should be performed. Recent reports in the open literature indicate that formation of nitrogen oxides in low-Btu gas combustion may be more prevalent than expected. The status of low-Btu gas combustion technology for gas turbine engines should be ascertained relative to emission standards expected in the Southern California area; and, if required, additional development work should be initiated.

(5) In light of the results of the above recommended work, the findings of this study should be re-examined.

SECTION 2

COGENERATION ENERGY USE AND COSTS

2.1 TYPES OF COGENERATION SYSTEMS

Cogeneration is the practice of simultaneously providing thermodynamic work and useful thermal energy from the same fuel or heat source. The thermodynamic work output of a cogeneration system is usually shaft power or electricity, and thermal energy output is usually steam or hot water at a temperature high enough to be used in an industrial process or other application. Typically, a cogeneration system feeds steam and electricity into an industrial plant's distribution system for use throughout the plant. Industrial process steam is typically generated at pressures ranging from 40 psia to 150 psia. Steam at near atmospheric pressure and hot water generally find little use in typical industrial plants, because low pressure steam is difficult to distribute and the heat available from low pressure steam, hot water, or condensate is usually at too low a temperature to provide process heat. Therefore, in this assessment unless heat from a cogeneration system could be used in the production of saturated steam at 267°F, it was not considered to be a part of the system's useful thermal output.

Cogeneration systems are generally categorized according to the type of energy conversion device, i.e., prime mover, on which they are based. The electricity output of a cogeneration system is provided by the prime mover, while the source of the thermal output depends on the configuration of the cogeneration system. Cogeneration system options considered in this study were based on either heat engines or fuel cells.

In the cogeneration systems based on heat engines, the energy source for the production of electricity and steam is the combustion of a füel. A portion of the energy released as heat by combustion is converted into electricity by the heat engine. Of the remaining portion of the combustion energy, part is utilized to generate steam and part is lost through system inefficiencies and in the flue gas. The heat-engine-based cogeneration systems considered in this study convert thermal energy into mechanical power that is used to drive an electrical generator. The heat engines considered herein all convert heat into mechanical work by increasing the temperature of a working fluid that is confined at elevated pressure within the engine, then allowing the working fluid to produce mechanical work during a restrained expansion process carried out in a positive-displacement expander or a turbine. All heat engines carry the working fluid through a cyclic sequence of thermodynamic states in order to convert thermal energy into thermodynamic work and are, therefore, subject to the Carnot limit for attainable efficiency. Heat engine energy conversion systems are generally classified according to the type of thermodynamic cycle on which they operate and by the type of working fluid used to execute the thermodynamic cycle. In addition, heat engine systems may be either directly fired or indirectly fired and either open cycle or closed cycle.

The distinction between the directly and indirectly fired heat engines lies in the manner in which heat addition to the working fluid is accomplished. In the directly fired engines, the combustion of the fuel is carried out in the working fluid itself, so the products of combustion partially constitute the working fluid as it undergoes expansion. The working fluid in indirectly fired engines receives heat through a high-temperature heat exchanger, so the products of combustion and the working fluid are not mingled. The high-temperature heat exchanger in indirectly fired engines must confine the working fluid at elevated pressure, and the working fluid will necessarily be at a temperature less than that of the combustion products or other heat source. High-temperature heat exchangers operating in a stressed condition represent an additional cost of indirectly fired heat engines.

Heat engines are further distinguished by being either open-cycle or closed-cycle. In the open-cycle machines, the working fluid is air that is inducted from the surroundings, carried through the thermodynamic processes within the engine, then exhausted to the atmosphere. In contrast, the working fluid in the closed-cycle machines is totally confined within the engine to repeatedly execute the thermodynamic cycle. The heat addition and rejection processes for the closed-cycle heat engines considered herein are carried out by heat exchangers. These closed-cycle engines are, therefore, indirectly fired and must also incorporate a low-temperature heat exchanger through which to reject heat.

Cogeneration systems based on heat engine energy conversion systems may be of either the topping cycle or bottoming cycle configuration. In topping cycle cogeneration systems, the heat source provides heat directly to the heat engine, while the thermal output for use in the industrial plant is derived from lower temperature heat that is rejected from the heat engine after the work-producing process. Actually, the heat rejected by certain heat engine energy conversion systems is at a fairly high temperature -- about 800°F to 1000°F for the exhaust gas streams of the gas turbine engine and the reciprocating engine systems. In bottoming cycle cogeneration systems, the industrial process heat is directly provided by the high temperature before heat is supplied to the heat engine. The heat source for electricity production in bottoming cycle systems is usually a flue gas stream or process fluid exit stream at a temperature much lower than that of the combustion heat source of topping cycle systems. Only the Rankine cycle cogeneration systems were considered in both topping cycle and bottoming cycle versions, as shown in Figures 3-1 and 3-2, respectively.

The electrical conversion efficiency, i.e., the ratio of electricity output to system fuel energy input, is higher for topping cycle systems because the heat engine operates with a higher temperature heat source, thus resulting in greater utilization of the thermodynamic availability of the fuel energy. As mentioned in Section 1.1, the overall efficiency of energy use of a cogeneration system, i.e., the ratio of the sum of electrical and thermal energy outputs to the fuel energy, is about the same as that of a conventional industrial boiler supplying only steam. Therefore, the additional cost of a cogeneration system is offset by electricity production and, perhaps, by the utilization of a less expensive fuel. The importance of a cogeneration system's electrical conversion efficiency and fuel type thus becomes evident.

The fuel cells do not make use of the cyclic thermo-mechanical energy conversion process as do the above-discussed heat engines. Instead, fuel cells convert a fuel and an oxidizer to electrical power by means of an electrochemical reaction that takes place within the fuel cell elements. Fuel cells are discussed in Section 4.2.5 of this report. The theoretical electricity output of a typical electrochemical fuel cell is in excess of 80% of the fuel energy; however, thermodynamic irreversibilities associated with practical fuel cell operation reduce the electrical conversion efficiency to about 40% of the fuel energy. Fuel cells do have a high overall energy utilization efficiency because the heat produced in the cell stack is recovered and is available as useful heat in a cogeneration system.

2.2 THE COST OF ELECTRICITY PRODUCED BY COGENERATION

The cost of electricity produced by a cogeneration system, or any other power plant, may be taken as the ratio of the net annual cost of owning and operating the system to its net annual electrical power output. The cost of owning and operating a cogeneration system is made up of the fixed charges on the capital investment, the maintenance cost, and the cost of fuel; all of which are partially offset by the value of the thermal energy produced.

The cost of electricity, as defined above, may be expressed as:

$$COE = (C + M + F - V)/E \qquad (1)$$

where C is the fixed charge, M is the maintenance cost, F is the fuel cost, V is the value of heat produced, all in dollars per year; and E is the electricity output in kilowatt-hours per year. In this study, the annual fixed charge was taken as 25% of the capital cost. The cost and performance parameters of cogeneration systems may be introduced into Equation (1) through the following expressions:

$$\begin{aligned} C &= I/4 \\ M &= mE \\ F &= phE \\ &\text{and} \\ V &= pbLE \end{aligned} \qquad (2)$$

where I is the capital investment in dollars, m is the maintenance cost factor in dollars per kilowatt-hour, p is the price of fuel in dollars per Btu, h is the heat rate of the cogeneration system in Btu of fuel fired per kilowatt-hour of net electricity output, b is the thermal output of the cogeneration system in pounds of steam per kilowatt-hour of net electricity output, and L is the rate at which fuel is consumed in conventional industrial boilers expressed in Btu of fuel fired per pound of steam generated.

Upon substitution of the above expressions, Equation (1) becomes:

$$COE = I/4E + m + ph + pbL \tag{3}$$

The annual electricity output E may be expressed in terms of the electrical power output of the cogeneration system in kilowatts and the number of hours per year that it operates. For purposes of this study, annual averages of all time-varying system performance parameters were used; therefore, E is expressed in terms of the rated electrical power output capacity of the cogeneration system and the equivalent number of hours per year that it operates at rated capacity. The capacity factor is the ratio of the actual annual electricity output E to that which would be obtained if the system continuously operated at rated capacity for an entire year. Hence, the annual electricity output may be expressed as:

$$E = fRA \tag{4}$$

where f is the capacity factor, R is the rated or maximum electrical power output capacity of the cogeneration system in kilowatts, and A is the number of hours in a year (taken as 8760). Upon introducing the expression above and rearranging, Equation (3) becomes:

$$COE = (I/R)/(4fA) + m + p(h - bL) \tag{5}$$

where I/R is the installed capital cost of the cogeneration system in dollars per kilowatt of rated power output, and the other parameters are as defined above.

The costs of electricity for the cogeneration systems analyzed in this study were all calculated using Equation (5). The capacity factor, f, was taken as 70% for all the cogeneration systems considered in this study, except for the solar cogeneration system whose capacity factor is 32%. The heat output of all the cogeneration systems considered in this study is in the form of industrial process steam that was assumed to displace steam generated in conventional boilers at the rate of 750 pounds per million Btu of fuel fired. Therefore, the parameter L is taken as $(10^6)/750$ in Equation (5). The installed capital cost (I/R), the heat rate (h), and the thermal output (b) were determined for each of the cogeneration systems, as discussed in Sections 3 and 4. The projections for the installed capital costs for the advanced cogeneration systems are direct costs and do not include interest on funds during construction, contingencies, and other such items that can significantly increase the cost of acquiring and owning a cogeneration system. The costs of electricity determined for the presently available and the advanced cogeneration system options are only for relative comparisons and do not reflect the actual costs of cogenerated electricity.

SECTION 3

PRESENTLY AVAILABLE COGENERATION SYSTEMS

Presently available industrial cogeneration systems may be based on the Rankine topping cycle, the Rankine bottoming cycle, the gas turbine engine or the reciprocating engine. Each of these four cogeneration system types provides electricity for use in the plant or sale to the utility and heat in the form of steam that is used in the industrial processes in the plant. Generally, a cogeneration system would feed steam and electricity into an industrial plant's existing distribution systems for use throughout the plant.

A Rankine topping cycle cogeneration system is shown in Figure 3-1. The system consists of a natural gas/fuel oil-fired steam boiler and a conventional, multi-stage, axial flow steam turbine whose exhaust directly supplies steam as 40 psia saturated vapor to the industrial processes. The boiler steam exit conditions are 650 psia and 750°F, which is typical of medium pressure industrial boilers. The Rankine bottoming cycle system is shown schematically in Figure 3-2. This system consists of an industrial steam boiler whose flue gas provides heat to drive a commercially available organic Rankine cycle (ORC) prime mover. An ORC prime mover is schematically illustrated in Figure 3-3. The performance and costs for the ORC system were derived from the recent work by Moynihan (Ref. 1). A conventional gas turbine engine (GTE) cogeneration system is illustrated in Figure 3-4. This system consists of a gas turbine engine whose exhaust heat is used to generate steam in a heat recovery boiler that supplies process steam as 40 psia saturated vapor. And finally, a conventional reciprocating engine cogeneration system is shown in Figure 3-5. The reciprocating engine exhaust gas stream is at a temperature high enough to generate steam as 40 psia saturated vapor; however, the heat from the engine's cooling system

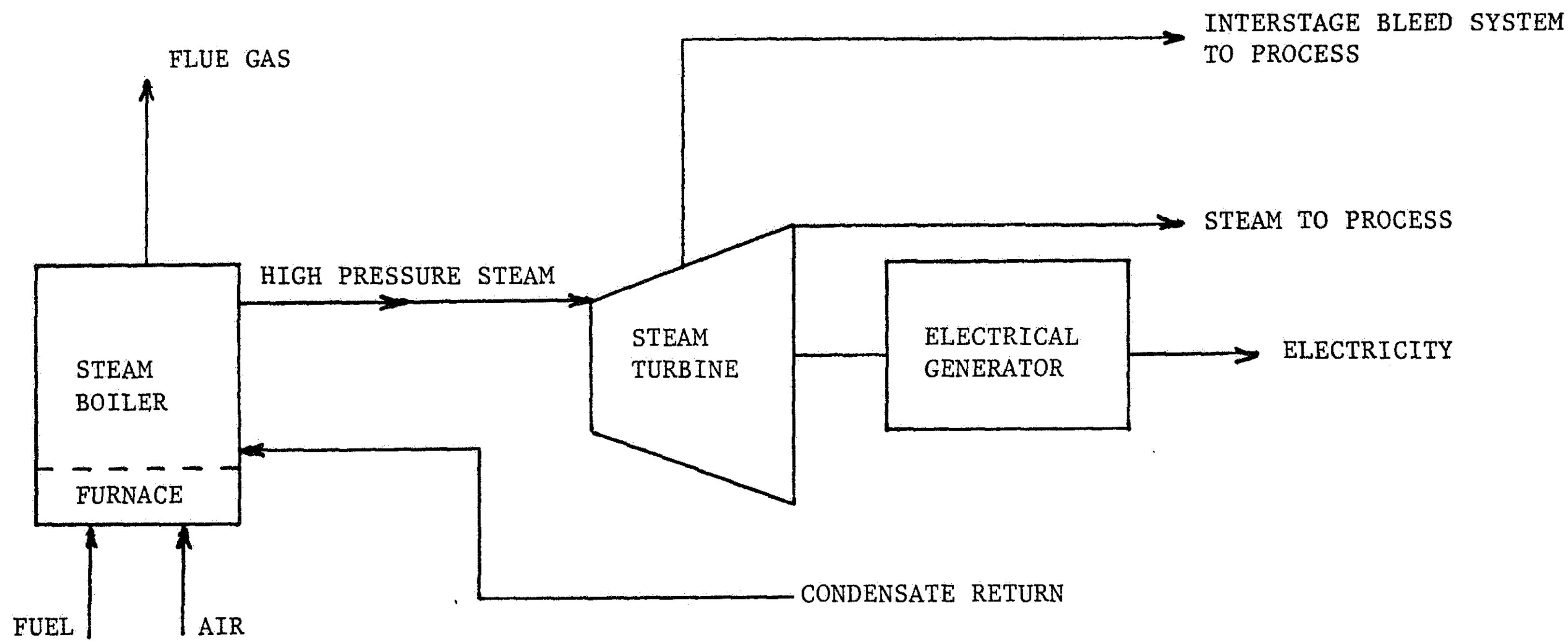


Figure 3-1. Conventional Rankine Topping Cycle Cogeneration System

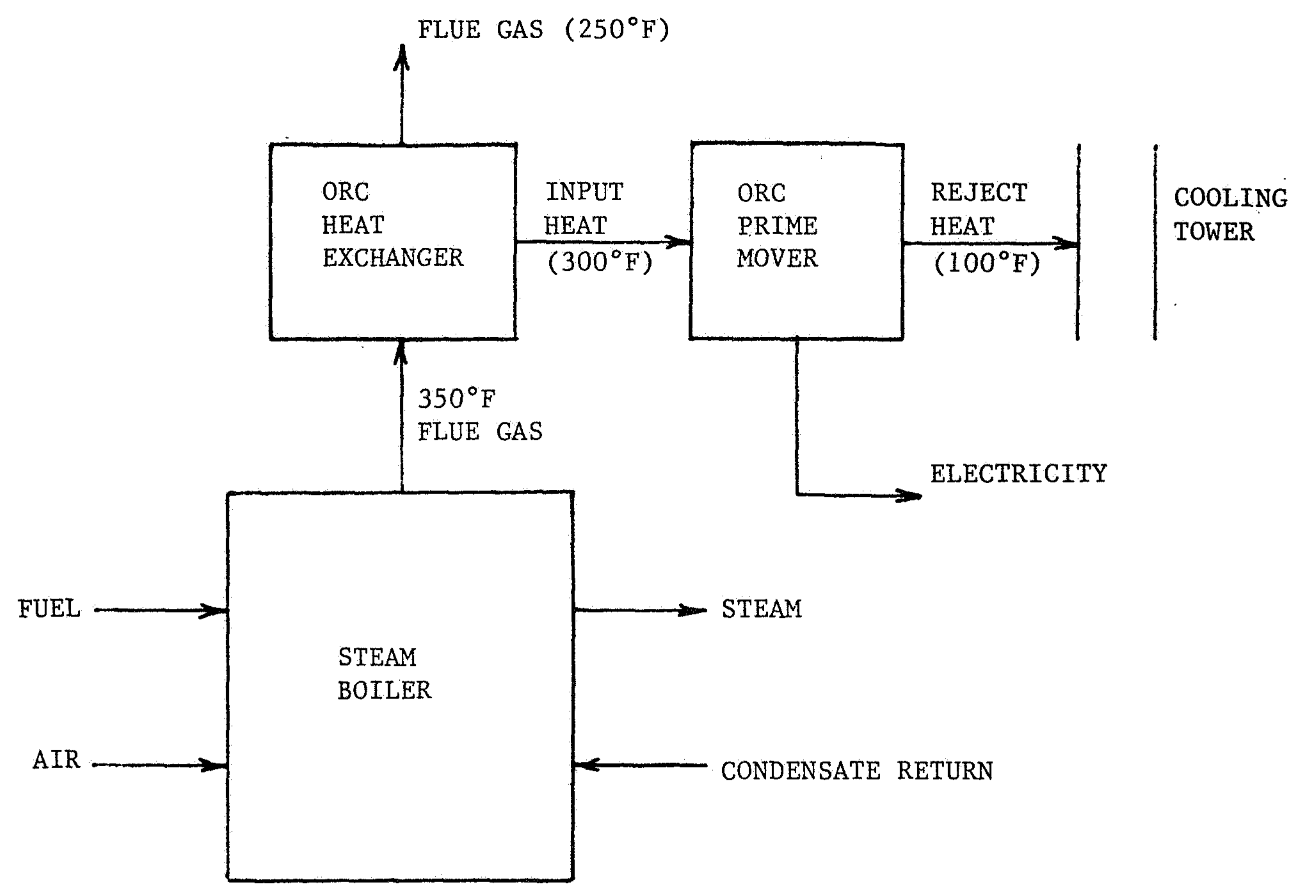


Figure 3-2. Organic Rankine Cycle System for Bottoming Cycle Cogeneration with a Process Steam Boiler

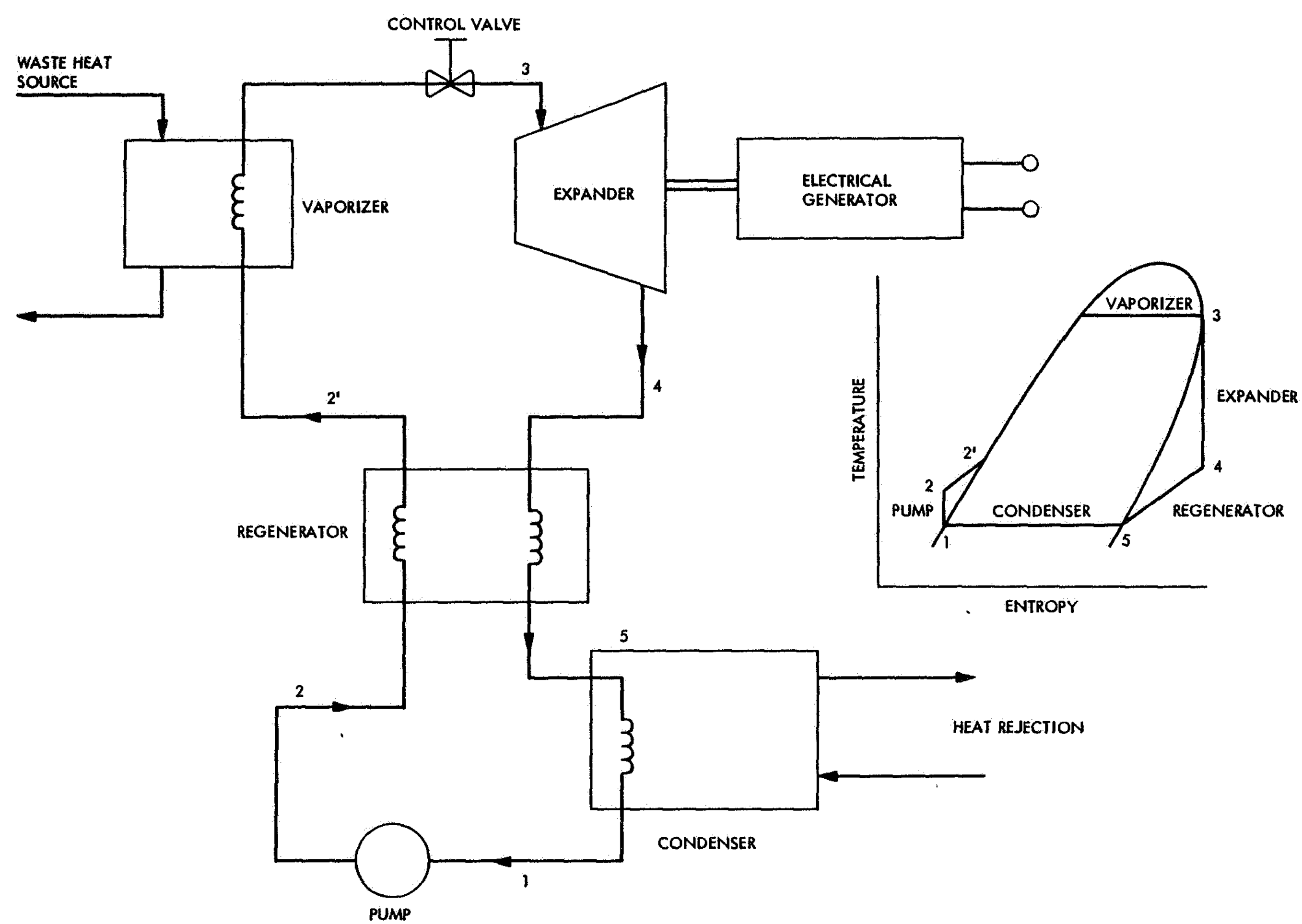


Figure 3-3. A Schematic Illustration of the Organic Rankine Bottoming Cycle with a Temperature-Entropy Diagram

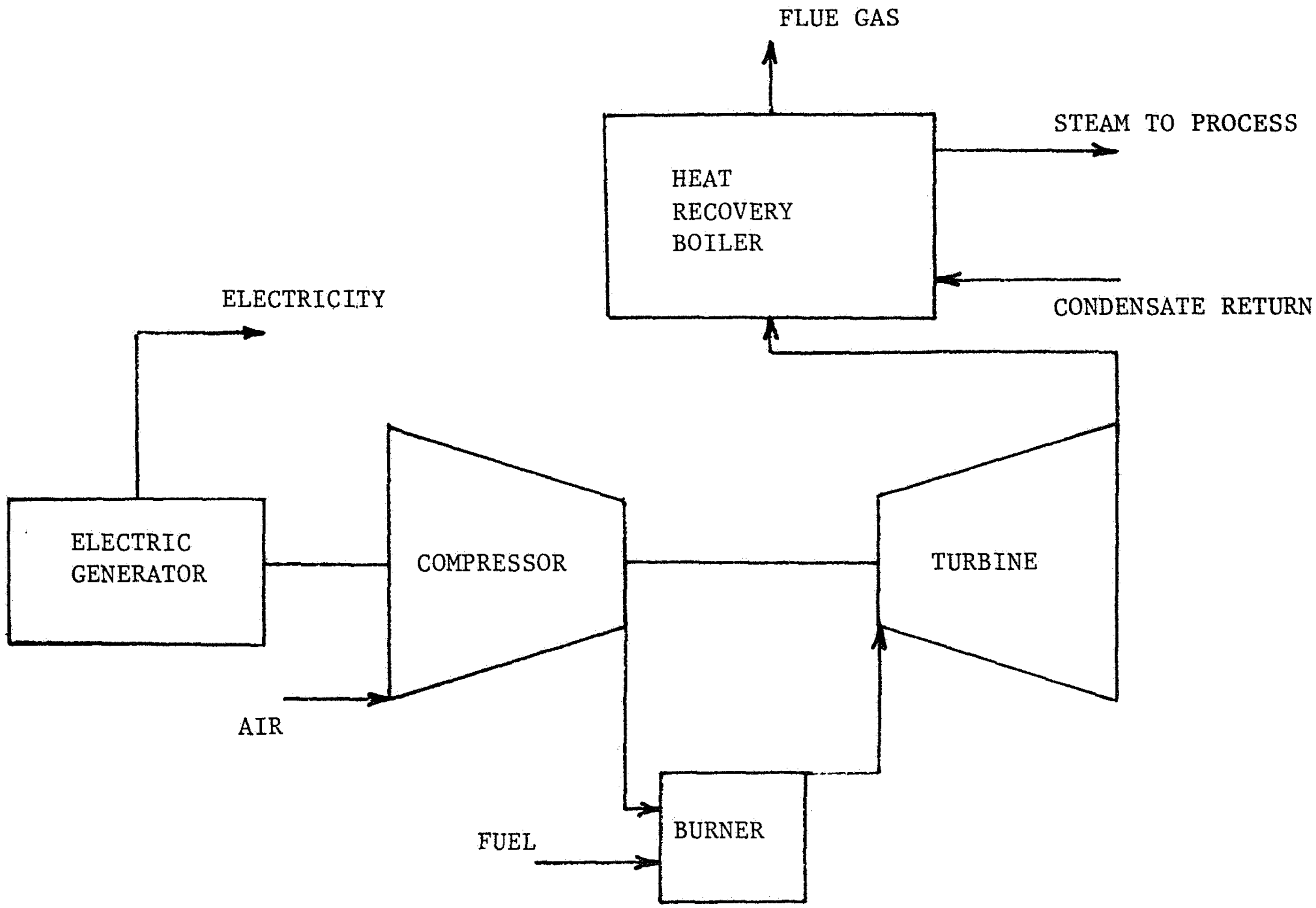


Figure 3-4. Conventional Gas Turbine Engine Topping Cycle Cogeneration System

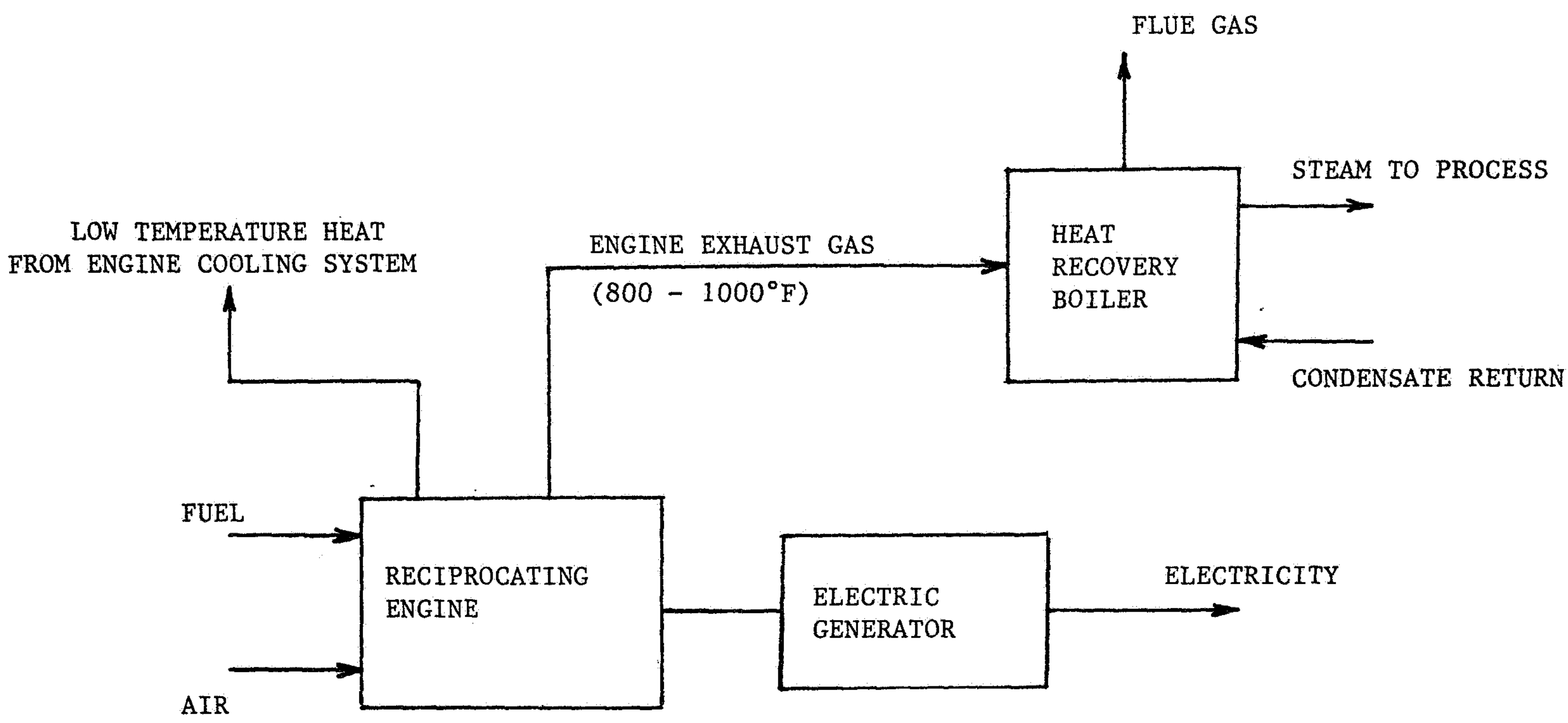


Figure 3-5. Conventional Reciprocating Engine Topping Cycle Cogeneration System

is not at a temperature high enough to generate 40 psia steam, so it is unavailable for use.

The characteristics of the four types of presently available cogeneration systems are shown in Table 3-1. The characteristics are based on system parameters, specifications, and equipment costs that were provided by manufacturers and vendors during the course of the work that is reported in Reference 2. The system performance was established by thermodynamic analyses based on the system parameters and specifications.

The installed capital costs of the systems shown in Table 3-1 were derived from equipment costs provided by suppliers and manufacturers and from estimates of installation costs. The costs were developed for systems assembled from components that are currently available from manufacturers and suppliers. The installed cost shown in Table 3-1 is the ratio of the direct cost of the installed system to the net electrical power output of the system. With the exception of the Rankine bottoming cycle system, the capital cost estimates include the prime mover, the electrical generator, the steam boiler and ancillary equipment. The capital cost estimate for the Rankine bottoming cycle does not include the steam boiler.

The costs of electricity shown in Table 3-1 were calculated at fuel costs of $5.00 and $7.50 per million Btu as calculated by the method given in Section 2.2 of this report. The Rankine topping cycle system and the GTE system have costs of electricity significantly less than that of the reciprocating engine. As previously stated, the reciprocating engine does not lend itself well to cogeneration as considered herein because the heat lost to the engine's cooling system is not available to raise steam. The high capital cost of the reciprocating engine system also contributes to its relatively high cost of electricity. As discussed in Section 4.2.3, an

Table 3-1. Comparison of Presently Available Cogeneration Systems

System Parameter	Rankine Cycle		Gas Turbine Engine[a]	Reciprocating Engine[b]
	Topping[a]	Bottoming		
Electricity Output, kW (net)	3760	1000	3060	1254
Fuel Firing Rate, 10^6 Btu/h	131	806[c]	38.4	14.2
Engine Thermal Input, 10^6 Btu/h	131	21.2	38.4	14.2
System Heat Rate, Btu/kW-h (net)	34840	806000	12800	11320
Thermal Output[d], lb steam/kW-h	21.3	589000	6.27	1.75
System Capital Cost[e], \$/kW	516	1800[g]	636	1290
Maintenance Cost, \$/kW-h	0.005	0.002	0.003	0.004
Cost of Electricity[f], mills/kW-h				
@ Fuel Cost of \$5.00/10^6 Btu	58	180	51	100
@ Fuel Cost of \$7.50/10^6 Btu	74		62	120

[a]Dual fuel (natural gas or distillate).
[b]Natural gas-fired.
[c]Boiler firing rate required to supply heat from boiler flue gas to bottoming cycle at boiler efficiency of 75%; feed water to steam enthalpy rise of 1229 Btu/lb; distillate fuel.
[d]Steam output at 40 psia, saturated vapor, with boiler feed water at 175°F.
[e]1982 \$; cost of prime mover, heat recovery equipment, ancillary equipment, and installation.
[f]Cost of electricity as defined in Chapter 2.2.
[g]Cost of ORC system only; the industrial steam boiler is not included.

an advanced version of the reciprocating engine does not suffer from the loss of the cooling system heat. The highest cost of electricity (COE) shown in Table 3-1 is that of the Rankine bottoming cycle. The bottoming cycle system has a high capital cost even though the cost of the boiler was not included. This system suffers from the thermodynamic disadvantage discussed in Section 2. The electricity costs and characteristics of the presently available systems serve as a baseline to which the advanced cogeneration systems are compared.

SECTION 4

ADVANCED COGENERATION SYSTEMS

4.1 FLUIDIZED BED COMBUSTION

Fluidized bed combustion of a solid fuel such as coal is accomplished by injecting fuel particles into a non-combustible bed of granular material that is fluidized by air blown upward from the bottom of the bed. Fluidization occurs when the gas flow through the bed is sufficient to suspend the particles of the bed without entraining them in the air flow. A fluidized bed combustion chamber may operate at pressures from one to several atmospheres. The air that fluidizes the granular bed is fed from beneath through a porous distributor plate. Atmospheric and pressurized fluidized bed combustion systems are illustrated in Figures 4-1 and 4-5, respectively. Typical values of air flow velocity needed to achieve fluidization of bed particles range from 3 to 8 feet per second. The fluidization characteristics change with particle size and density as discussed in Reference 3.

The fluidized bed combustion of coal is usually carried out in a bed of limestone or dolomite which acts as a sorbent for the sulfur in the coal. In the latest FBC designs, it is expected that from 80% to 98% of the sulfur in a high sulfur eastern coal can be retained in the bed as calcium sulfate when the sorbent in the bed is available at a calcium to sulfur mole ratio of 1.7 to 5.0, depending on the design of the fluidized bed combustor. In any fluidized bed combustion system, provisions must be made to inject the sorbent which usually constitutes over 90% of the material contained in the bed, to feed the coal, and to remove the spent bed material and coal ash. The rate of heat removal from the bed via air-cooled or water-cooled tubes within the bed must be matched to the bed heat generation rate in order to achieve stable operation. The technology development of FBC combustion systems has been directed toward achieving economical, reliable operation while maintaining low emissions of sulfur and nitrogen oxides. Fluidized bed steam boilers operating at a pressure of near one atmosphere and of up to 200,000 lb per hour of steam capacity are in the commercial demonstration stage (Ref. 4).

Both atmospheric and pressurized fluidized bed designs are being developed for electric utility applications and could serve as the basis for advanced industrial cogeneration systems. The atmospheric fluidized bed systems consist of a fluidized bed boiler and a steam turbine power conversion cycle that is virtually identical to that of conventional Rankine cycle cogeneration systems. The pressurized fluidized bed system differs from the AFBC system in two major respects: (1) the coal is burned in the fluidized bed at elevated pressure and (2) the pressurized fluidized bed combustor is integrated with a gas turbine engine in which it replaces the conventional gas turbine combustor.

Atmospheric Fluidized Bed Combustion. Atmospheric fluidized bed combustion has been widely investigated in the past five to seven years. The cumulative results of these investigations are reviewed in References 5 through 9. The results and findings of these performance and economic investigations indicate that once-through AFBC units have two major limitations. First, once-through beds require that a high ratio of calcium sorbent in the bed material to sulfur in the coal be maintained in order to achieve 90% sulfur capture in coal of from 3-5% sulfur content. Second, once-through AFBC beds have a comparatively low combustion efficiency of 90 to 95%; hence, secondary carbon burn-up has been used to achieve acceptable overall fuel utilization efficiency. The requirement for calcium to sulfur mole ratios of from three to five coupled with poor combustion efficiency impose significant penalties on the once-through AFBC system (Ref. 8). An alternative AFBC design offers the potential to resolve the problems of high sorbent usage and low combustion efficiency by recycling a large amount of bed material that is purposefully elutriated from the bed (entrained in the combustion product gas stream leaving the fluidized bed). An AFBC unit with provision for recycle is shown in Figure 4-1. The elutriated material consists of coal ash, spent sorbent, unburned coal and unused sorbent. The quantity of elutriated material depends on the gas velocity at the bed surface and on the size distribution of the bed material. AFBC designs that recycle up to five times the mass flow of coal feed are under investigation (Refs. 8, 9). Current investigations of atmospheric fluidized bed combustion for utility-sized power plants include the construction of a twenty-megawatt experimental AFBC facility at a Tennessee Valley Authority site in Kentucky. This facility is designed to experimentally investigate the performance of AFBC units with high recycle rates. The twenty-megawatt test facility is expected to provide necessary information regarding coal/limestone feed system configuration, boiler-tube material, boiler-tube erosion, and recycle rates for best sorbent utilization.

Pressurized Fluidized Bed Combustion. The combustion of coal in a fluidized bed that is maintained at an elevated pressure results in improved fluidized bed performance without recycling or recirculation of elutriated bed material. Relative to AFBC, operation at elevated pressure improves sulfur capture, increases combustion efficiency, reduces nitrogen oxides emissions and reduces the size of the fluidized bed for a given firing rate. The results of extensive development work on PFBC units and studies of systems incorporating PFBC are reported in References 10 through 21. These investigations indicate that better than 95% sulfur capture at calcium to sulfur mole ratios of about 2:1 and combustion efficiencies in excess of 99% can be obtained in a PFBC cell at fluid velocities that are low enough to minimize elutriation of bed material.

The pressurized fluidized bed combustors being considered for utility power plants are cooled by tubes immersed in the fluidized bed. These in-bed tubes carry either water and steam or air as the heat extraction fluid. In either water-cooled or air-cooled PFBC units, the high pressure combustion air is provided by a gas turbine compressor. The combustion product gas from the PFBC unit is cleaned of particulate matter and expanded through a gas turbine. The heat extracted from the in-bed tubes may be used as the heat source for a Rankine/steam turbine cycle if the PFBC unit is water-cooled. However, if the in-bed tubes are air-cooled, the gas turbine compressor supplies sufficient air both to support combustion and to cool the tubes. The hot air from the in-bed tubes is mixed with cleaned combustion gas, and the resulting hot gas stream is expanded through a gas turbine. As in a conventional gas turbine cogeneration system, heat may then be recovered from the gas turbine exhaust in a waste heat boiler to generate steam.

4.2 ADVANCED SYSTEMS CONCEPTS, STATUS, AND CHARACTERIZATION

4.2.1 Rankine Cycle Systems

Topping Cycle Cogeneration Systems

Technology Status. The use of Rankine cycle energy conversion systems in industrial plants dates from the era of the industrial revolution. The principal components in conventional Rankine cycle systems are the steam boiler and steam turbine, as discussed in Section 3.1. Rankine cycle technology is quite mature and no technological advancements in steam generator or turbine technology that would substantially affect the system performance or costs have been identified. However, in systems of industrial size, the atmospheric fluidized bed boiler may offer modest cost advantages relative to stoker fired or pulverized coal fired boilers. At more stringent emmissions standards, any cost advantage of the AFBC (atmospheric fluidized bed combustion) boiler system could be lost, because large quantities of either sorbent or supplementary flue gas desulfurization may be required. With regard to turbines, some manufacturers have offered radial flow steam turbines as an alternative to the commonly used axial flow machines. The radial flow units may offer advantages in performance when compared to the small, simple impulse turbines commonly used for low power mechanical drives in industrial plants, although at some increase in initial cost. However, the multi-stage axial flow turbines currently used for cogeneration applications in the size range of from one to ten megawatts output have cost and performance characteristics that are not likely to be bettered by other types of expanders. At sizes greater than about ten megawatts, conventional axial flow turbines are clearly superior.

Advanced System Characteristics. The advanced technology Rankine topping cycle cogeneration system consists of an atmospheric fluidized bed combustion (AFBC) steam boiler and a conventional, axial-flow steam turbine whose exhaust directly supplies steam for the industrial processes. An AFBC steam boiler is shown schematically in Figure 4-1, and a Rankine topping cycle cogeneration system was previously shown in Figure 3-1. Figure 4-2 shows a typical industrial AFBC boiler installation in greater detail. The system characteristics for the AFBC Rankine cycle cogeneration system are shown in Table 4-1 along with a conventional, stoker-type, coal-fired system and the conventional oil/gasfired system. The performance characteristics of the Rankine topping cycle cogeneration systems shown in Table 4-1 were established by means of a thermodynamic analysis of the Rankine cycle. The system parameters are based on the specifications for currently available commercial equipment as provided by manufacturers and as given in the open literature. The system parameters and the results of this analyses are presented in Table 4-2 for the oil/gas-fired boiler. The boiler efficiency was increased to 80% for both of the coal-fired units, which gives a slight improvement in system heat rate for the stoker-fired system of Table 4-1. The AFBC system also operates at a boiler efficiency of 80%, but the parasitic power of the forced draft fan that supplies air to the fluidized bed causes the system heat rate for the AFBC unit to increase to 35,320 Btu/kW-h.

All three of the Rankine cycle cogeneration systems have a comparatively high thermal output of 21 to 23 lb of steam per kW-h. This characteristic of Rankine cycle systems is a consequence of the thermodynamic operation of the system. The turbine work output is limited by the relatively low temperature of 750°F at which the working fluid enters the turbine and by the turbine

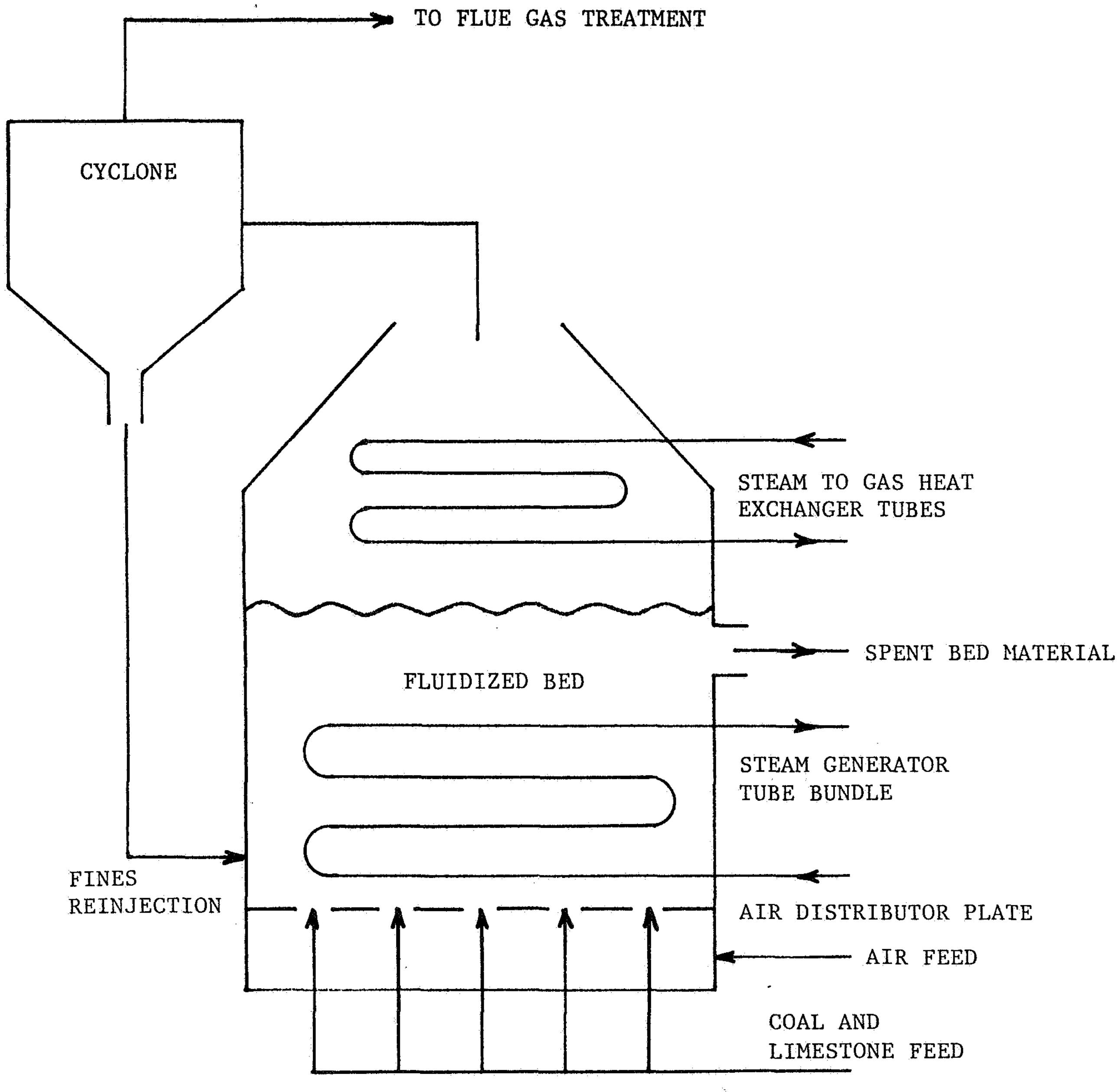
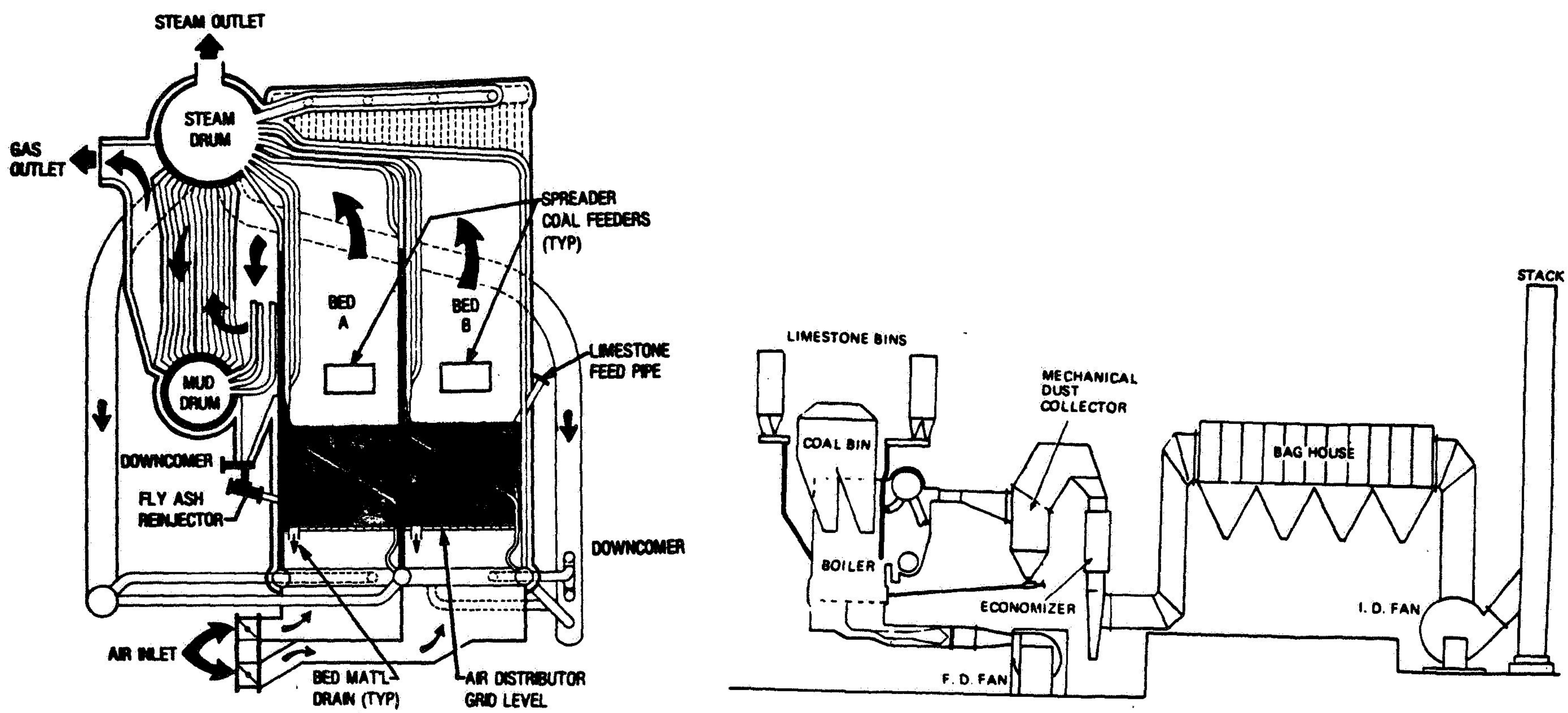


Figure 4-1. A Schematic Illustration of an Atmosphere Fluidized Bed Steam Boiler for Rankine Topping Cycle Cogeneration Systems



Fluidized Bed Steam Generator

General Arrangement

Figure 4-2. Typical Industrial AFBC Installation for 100,000 lb/h Steam Generation (Superheater not shown)

Table 4-1. Rankine Topping Cycle Cogeneration System Characteristics

System Type				
Fuel	Configuration	Capital Cost, $/kW	Heat Rate, Btu/kW-h	Output Ratio of Steam[a] to Electricity, lb/kW-h
Natural Gas/ Distillate	Conventional Boiler/Steam Turbine	516	34,840	21.3
Coal/Other Solid Fuel	Stoker Fired Boiler/FGD[b]/ Steam Turbine	1630	32,660	21.3
Coal/Other Solid Fuel	AFBC[c] Boiler/ Baghouse/ Steam Turbine	1610	35,320	23.0

[a]Steam output @ 40 psia saturated vapor.
[b]Flue Gas Desulfurization via wet scrubbing.
[c]Atmospheric Fluidized Bed Combustion.

Table 4.-2. Rankine Topping Cycle Cogeneration System Parameters

Boiler size lb/h	80,000
Boiler exit steam temperature, °F	750
Boiler steam pressure, psia	650
Condensate return fraction	0.90
Condensate temperature, °F	185
Make-up water temperature, °F	90
Boiler feedwater temperature, °F	175
Enthalpy increase of steam across boiler, Btu/h	1,229
Boiler efficiency, %	75
Fuel energy input to boiler, 10^6 Btu/h	131
Steam turbine efficiency, %	65
Turbine exhaust pressure, psia	40
Isentropic enthalpy drop across turbine, Btu/lb	264
Turbine power output, kW	4,020
Water pump efficiency, %	70
Pumping power, kW	60
Gearbox efficiency, %	98.5
Generator efficiency, %	95
Net electrical power output, kWe	3,760
Steam to plant processes:	
Pressure, psia	40
Temperature, °F	267
Quality, %	100
Enthalpy, Btu/lb	1,170

back-pressure of 40 psia; hence, the turbine extracts only a small fraction of the energy of the boiler exit steam. The turbine exhaust steam is provided directly to the industrial process where the greater portion of the energy from the combustion of the fuel is made available for use. Consequently, the overall efficiency of energy utilization for Rankine cycle systems is reasonably high despite the low power output.

The capital costs of the coal-fired systems given in Table 4-1 are strikingly higher than for the oil/gas-fired system. This cost difference is due solely to the much higher cost of coal-fired boilers and their subsystems. The cost estimates of Table 4-1 are based on the information given in References 2 and 22. Reference 22 is a cost study of oil, gas, and coal-fired industrial size steam boilers and their supporting systems. The study presents a direct comparison of the costs of three types of coal burning boilers -- stoker-fired, pulverized coal-fired, and AFBC -- with oil burning and gas burning boilers. According to Reference 22, the costs of AFBC and stoker-fired boilers with their flue gas treatment systems, but not including ancillary equipment and facilities, are a factor of four to five times the cost of an oil/gas-fired boiler of the same steam generation capacity. The capital costs for AFBC and stoker-fired boilers for Rankine cycle cogeneration systems were obtained by appropriately scaling the boiler costs of Reference 22 to account for differences in firing rate, then combining the boiler cost with that of the turbine/generator and the balance of the plant given in Reference 2. The Rankine cycle cogeneration system costs thus developed show the relative cost differences between the oil/gas-fired, stoker-fired, and AFBC units. The installed boiler costs for oil/gas-fired units in References 2 and 22 agree favorably. Hence, the capital costs of Table 4-1 also reflect the cost of Rankine cycle systems relative to the baseline systems whose costs were derived from Reference 2.

The costs of electricity for the Rankine cycle cogeneration systems are shown in Table 4-3. The coal-fired systems have costs of electricity in the higher range at 76 to 82 mills per kilowatt-hour for fuel costing $1.25 to $2.50 per million Btu. The conventional oil/gas-fired system provides lower cost electricity than the coal-fired systems until the cost of oil/gas exceeds $7.50 per million Btu. The coal-fired systems suffer from the high cost of conventional coal burning equipment.

Table 4-3. Cost of Electricity for Rankine Topping Cycle Cogeneration Systems (mills per kilowatt-hour)

System Type		Fuel Cost, Dollars per Million Btu			
Fuel	Configuration	1.25	2.50	5.00	7.50
Natural Gas/ Distillate	Conventional Boiler/Steam Turbine		42	58	74
Coal/Other Solid Fuel	Stoker Fired Boiler/Steam Turbine	77	82	93	
Coal/Other Solid Fuel	AFBC[a] Boiler/ Steam Turbine	76	82	94	

[a]Atmospheric Fluidized Bed Combustion.

4.2.2 Gas Turbine Systems

Technology Status. The gas turbine engine (GTE) is a steady flow machine in which heat is converted to mechanical power by compression of the working fluid to an elevated pressure, heat addition at constant pressure, and restrained expansion of the hot working fluid to produce mechanical power. Current gas turbine engines rely on aerodynamic compressors and turbines of radial flow or axial flow design operating at pressure ratios from 4:1 to about 20:1, depending on the size of the engine and its application. In the post World War II era, the gas turbine engine has been the focal point of technological development for aircraft propulsion. The huge investments of private and government capital in gas turbine engine development have resulted in power plants that utilize advanced engineering technology and materials, e.g., superalloys and coatings. Modern gas turbine engines can operate with turbine inlet temperatures of from 1800°F to 2100°F, enabled by high-temperature materials in conjunction with air-cooled turbine vanes and blades, and usually feature multi-stage axial flow compressors and turbines. A typical aircraft-derived industrial GTE is shown in Figure 4-3 with component materials designated. Gas turbine engines have the potential for extremely high levels of power output per unit of engine weight, typically about 0.2 to 0.4 lb per shaft horsepower for aircraft turboshaft engines. Several gas turbine manufacturers offer modified aircraft engines for industrial applications. Other manufacturers use heavier, conventional castings for the engine cases and housings if the turbine is for stationary industrial use where increased weight is of no consequence. Gas turbine engines specifically designed for a variety of transportation and stationary applications are offered by manufacturers in the United States, Europe, and Japan.

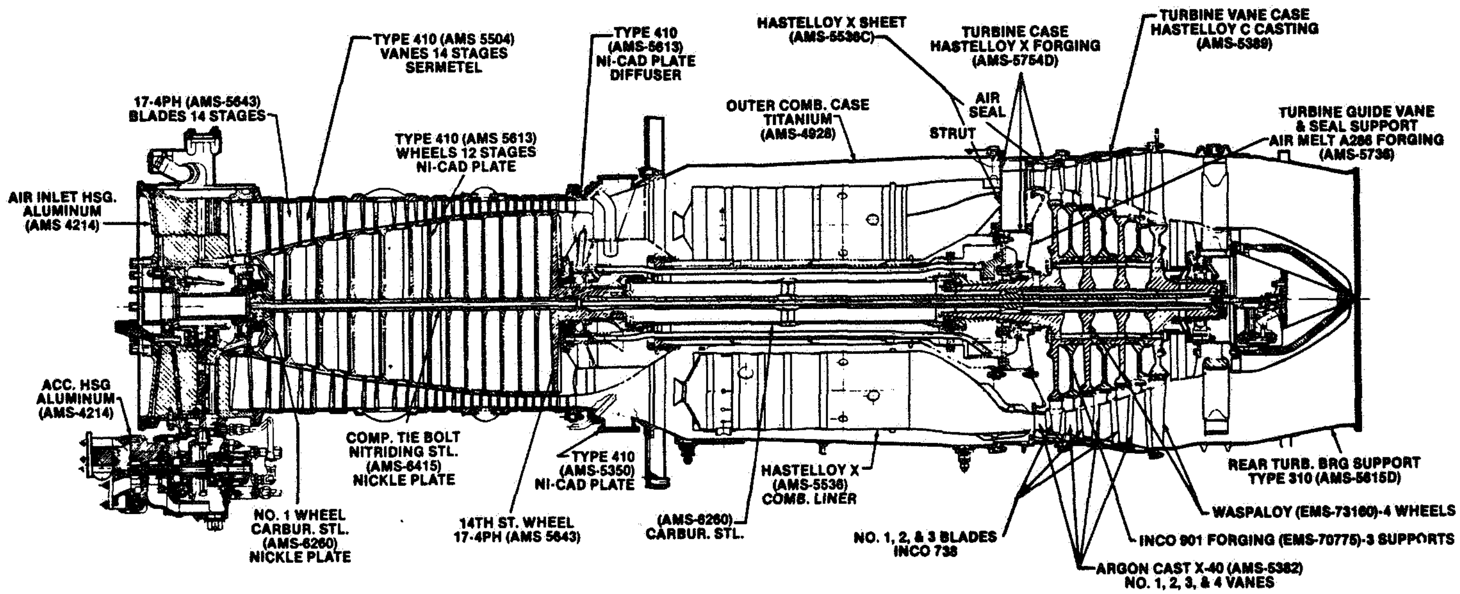


Figure 4-3. Industrial Gas Turbine with Materials Designation

Gas turbine engines (GTEs) are well suited for industrial cogeneration systems because virtually all of the engine's waste heat is available from a single source (i.e., the exhaust gas stream) and the engine is adaptable to different types of combustion systems or high-temperature heat sources. In addition, the heat losses from open-cycle, internal combustion gas turbine engines are small because they are compact and have a high gas throughput. The simple, compact configuration of the GTE results in the working fluid being exposed to a relatively small surface area through which heat losses can occur. The exhaust gas stream from a GTE is typically at 900°F to 1000°F which allows for economical heat recovery for industrial steam generation at normal steam temperatures. The high-temperature heat addition process in the GTE is physically distinct and separated from the compression and expansion processes. This feature, in conjunction with the steady flow of the working fluid through the engine, permits a variety of high-temperature heat sources to be adapted to the gas turbine engine.

Conventional gas turbine engines are directly-fired, internal combustion engines in which liquid or gaseous fuel is injected into compressor discharge air and burned in a relatively compact combustion chamber. About 20 to 25% of the oxygen in the compressor discharge air is consumed in combustion with a typical liquid hydrocarbon fuel. The combustion chamber in aircraft engines is quite compact, usually arranged in an annular configuration around the GTE's "waist." If designed for stationary applications, the combustion chamber can be completely removed from the engine, and compressor discharge air can be supplied to the combustor via ductwork. The hot combustion products must then be ducted to the turbine section of the engine, with the accompanying heat and pressure losses in the ducting. The separate combustion chamber in this arrangement can be replaced by a heat exchanger to heat the compressor discharge air. The heat exchanger may be an integral part of a furnace in which a variety of fuels can be fired. Such a configuration is referred to as an indirectly-fired gas turbine engine.

Gas turbine engines may also be operated as closed cycle machines wherein the working fluid is recirculated through the engine, which is equipped with heat exchangers for heat addition and rejection. However, there is little to recommend the closed-cycle GTE for cogeneration applications. They offer no identifiable performance advantages over open cycle machines and are not commercially available at present. In order to have efficiencies comparable to open cycle machines, the working fluid would have to be cooled to near ambient temperature in the heat rejection process. This would require a large low temperature heat exchanger in addition to the heat recovery boiler of the present cogeneration system. The additional cost and complexity are not offset by an accompanying performance improvement. Therefore, closed cycle GTEs are not considered further in this study.

Advanced Cogeneration Systems. Gas turbine engine cogeneration systems were analyzed in two direct-fired, internal-combustion configurations and in two indirectly-fired configurations. The advanced cogeneration systems make use of presently available gas turbine engines and heat recovery systems. The technology advancements lie only in the method of fuel utilization by the gas turbine engine. The costs and performance characteristics of these four cogeneration systems are shown in Table 4-4, along with those of the conventional gas turbine cogeneration system. The performance and costs for the gas turbine engine/generator and the heat recovery boiler were based on the information given in Reference 2 and on detailed information provided by equipment manufacturers. Energy balances were employed to establish the

Table 4-4. Gas Turbine Cogeneration System Characteristics

System Type		Capital Cost, $/kW	Heat Rate, Btu/kW-h	Output Ratio of Steam to Electricity, lb/kW-h	Relative Power Output
Fuel	Configuration				
Natural gas/ Distillate	Conventional GTE[a]/HRB[b]	636	12800	6.27	1.0
Coal/Multi-Fuel	Integrated, Air-Blown Gasifier/ GTE/HRB	1050	18200	8.23	1.0
Coal/Other Solid Fuel	PFBC[c]/GT/HRB	1030	14300	6.72	0.81
Multi-Fuel (Light Ash)	GTE-Blown Furnace/Boiler	1850	67600	46.9	0.63
Coal/Other Solid Fuel	AFBC[d]/GTE/HRB	1570	18500	7.63	0.63

[a]Gas Turbine Engine.
[b]Heat Recovery Boiler.
[c]Pressurized Fluidized Bed Combustor.
[d]Atmospheric Fluidized Bed Combustor.

performance of each cogeneration system. The performance characteristics of a typical industrial gas turbine engine with a normal pressure ratio of 9:1 and continuous turbine inlet temperature of 1800°F were adopted for the conventional gas turbine cogeneration system and were used to establish the effects of reduced turbine inlet temperature on engine efficiency and power output. These effects were verified by straightforward thermodynamic analyses of the gas turbine cycles. The effects of increased pressure loss in the combustor or high-temperature heat exchanger and of compressor discharge flow diversion for the coal gasifier air supply were determined by thermodynamic analyses of the gas turbine cycle according to the method of Reference 23.

A directly-fired gas turbine cogeneration system with an integrated coal gasifier is shown in Figure 4-4. About one-fifth of the GTE compressor discharge air is diverted to the coal gasifier where it reacts with coal and water/steam to produce low-Btu coal gas. The coal gas is processed to remove particulates and contaminates, primarily sulfur, and is then burned in the GTE combustion chamber. The coal gasifier is postulated to operate at compressor discharge pressure. Considering the pressure losses through the gasifier and gas-cleaning system, a slight reboost of fuel gas pressure, requiring little power, would be necessary prior to the coal gas entering into the combustor (the reboost compressor is not shown in Figure 4-4). The diversion of one-fifth to one-fourth of the compressor discharge air flow into the gasifier and the subsequent cooling of the gasifier products to near ambient temperature cause the gas turbine engine's brake efficiency to be reduced to 92% of the nominal value. This loss, in conjunction with the coal gasifier efficiency of 77%, results in a coal feed to electricity heat rate for

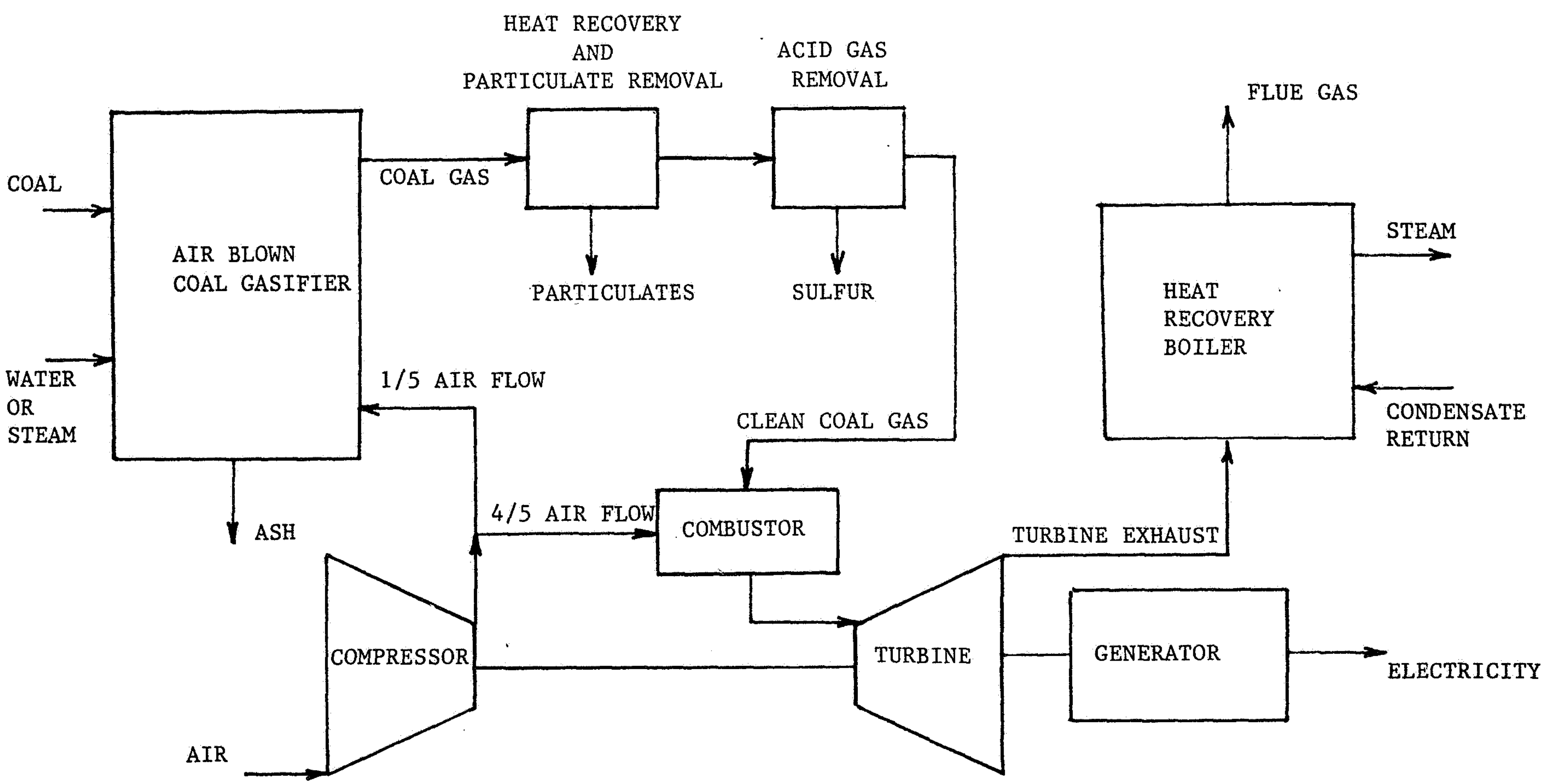


Figure 4-4. Integrated Air-Blown Coal Gasifier/Gas Turbine Cogeneration System

the system of 18,200 Btu/kW-h. The cogeneration system thermal output is increased relative to the conventional GTE cogeneration system because 11% of the coal feed energy is recoverable from the coal gasifier as useful heat, which, in addition to the heat recoverable from the GTE exhaust, gives the thermal output of 8.23 lb/kW-h shown in Table 4-4. Coal gasification gas turbine systems are discussed in References 24 through 30.

A gas turbine cogeneration system that incorporates a pressurized fluidized bed combustor is shown in Figure 4-5. In this system about 30% of the compressor discharge air flow supports coal combustion in the fluidized bed while the remaining compressor discharge air removes heat from the bed as it flows through the heat exchanger tubes immersed in the fluidized bed. The in-bed heat exchanger tubes operate at a very low stress level because the gas pressure inside the tubes and that in the fluidized bed are virtually equal. The combustion product gas stream from the PFBC unit is then cleaned of particulates, mixed with the heated air from the in-bed tubes, and expanded through the turbine. The PFBC bed temperature must be limited to 1550°F to 1750°F to ensure sulfur capture and to control fusion and agglomeration of bed material. A turbine inlet temperature of 1650°F was assumed to be available from the PFBC unit. The reduction in turbine inlet temperature from the normal value of 1800°F causes both engine efficiency and power output to fall. A slight additional penalty results from the somewhat higher pressure loss of the PFBC unit. The overall degradation of the coal feed to electricity conversion efficiency results in a heat rate for the PFBC/gas turbine cogeneration system that is 12% higher than for the conventional system. The thermal output for the PFBC system is 6.72 lb/kW-h, the increase relative to the conventional GTE system being due to the higher heat rate of the PFBC/GTE system. PFBC/GTE systems are discussed in References 31 through 36.

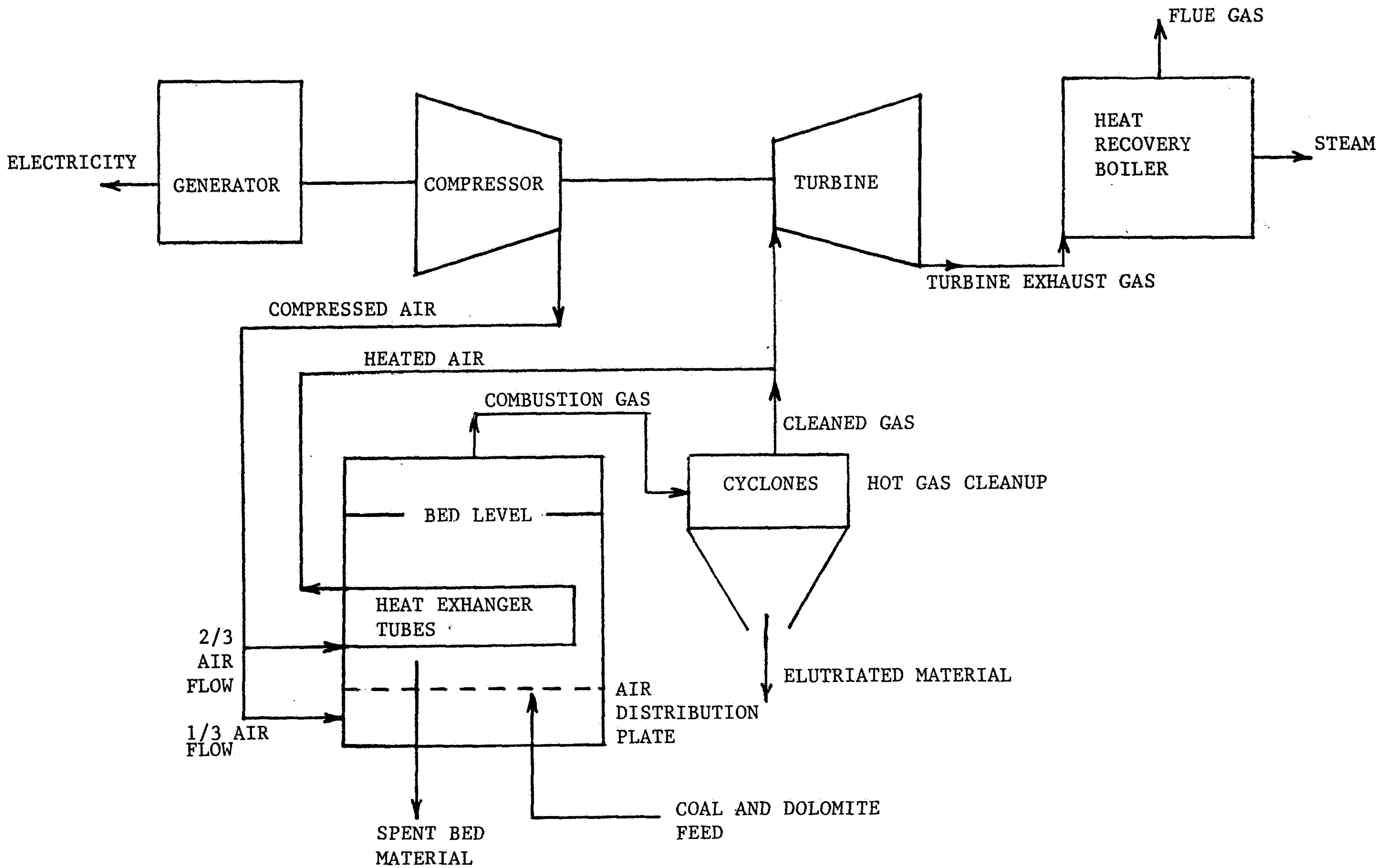


Figure 4-5. A Schematic Diagram of a Pressurized Fluidized Bed Combustion Unit with a Gas Turbine Cogeneration System

Indirectly-fired gas turbine engine cogeneration systems are shown in Figures 4-6 and 4-7. The system of Figure 4-6 consists of a conventional gas turbine engine and an atmospheric fluidized bed combustion (AFBC) furnace/heat exchanger serving as the high-temperature heat source. The compressor discharge air is heated to 1550°F as it flows through the AFBC heat exchanger tubes without contacting or mixing with combustion products. The heat exchanger tubes in this system operate at a stress level determined by the pressure difference across the tube wall since the furnace operates at near atmospheric pressure and the heat exchanger tubes contain compressor discharge air at 100 to 200 psia. The temperature of 1550°F is determined by the long-term, high-temperature strength and durability of heat exchanger materials. The decrease in turbine inlet temperature from the normal value of 1800°F to 1550°F penalizes both engine heat rate and power output; the engine heat rate increases by about 17% and power output falls by about 37%. An additional small loss in engine performance results from the increased pressure loss of the high-temperature heat exchanger tubes. The overall fuel input to electricity output efficiency is further reduced by the heat losses of indirect firing. The furnace/heat exchanger efficiency was assumed to be 85%, which is somewhat higher than current values for units in the size range considered here. The furnace/heat exchanger loss is particularly significant because it increases the system heat rate and simultaneously reduces thermal output. The AFBC/GT/HRB system performance, considering the reduction in turbine inlet temperature and the heat loss of indirect firing, is shown in Table 4.2.2-1. AFBC/GT systems are discussed in References 37 through 42.

The second indirectly-fired gas turbine engine system is shown in Figure 4-7. This gas turbine/furnace arrangement, referred to as the GTE-Blown

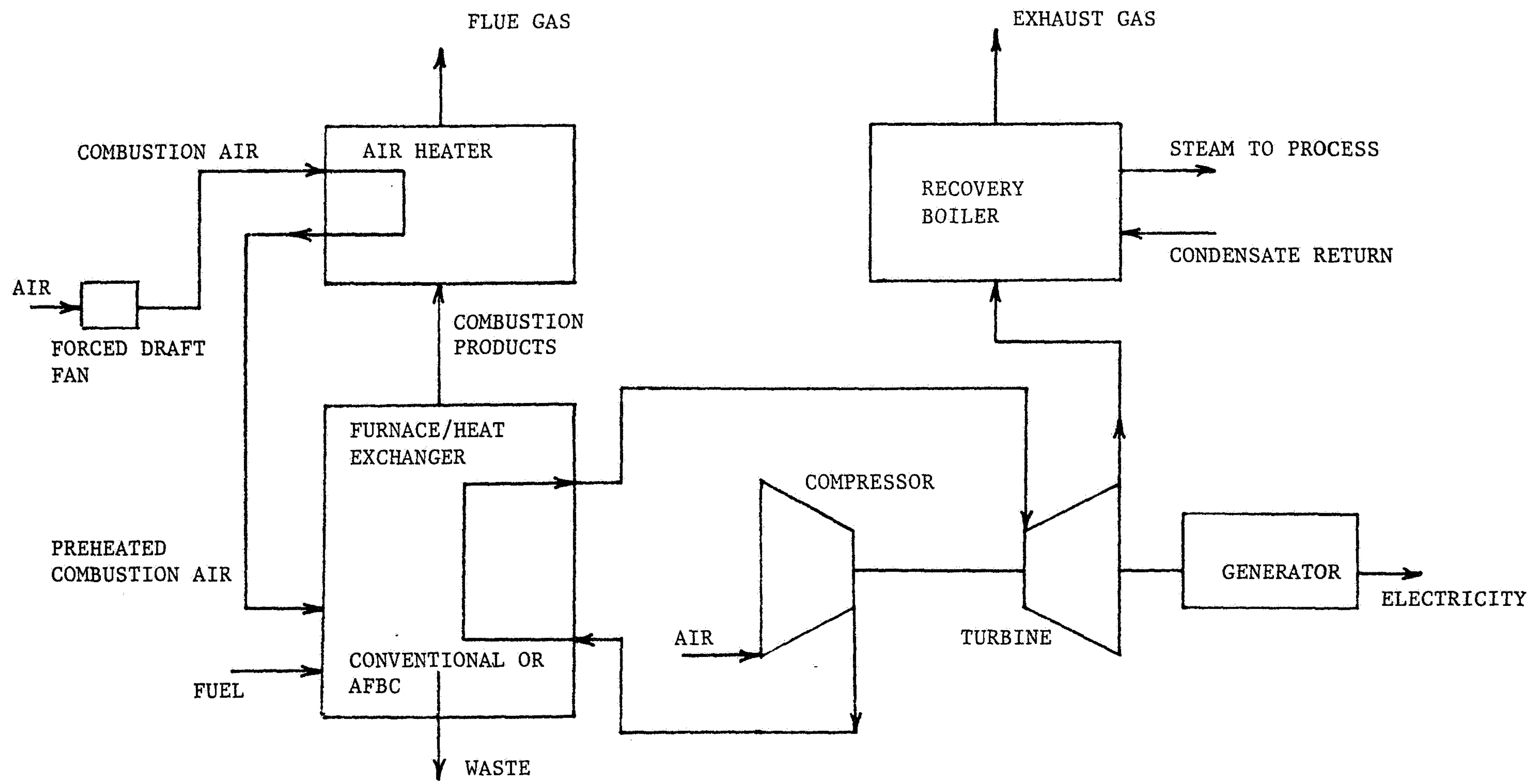


Figure 4-6. Indirectly-Fired Gas Turbine Cogeneration System with a Conventional or Fluidized Bed Furnace/Heat Exchanger

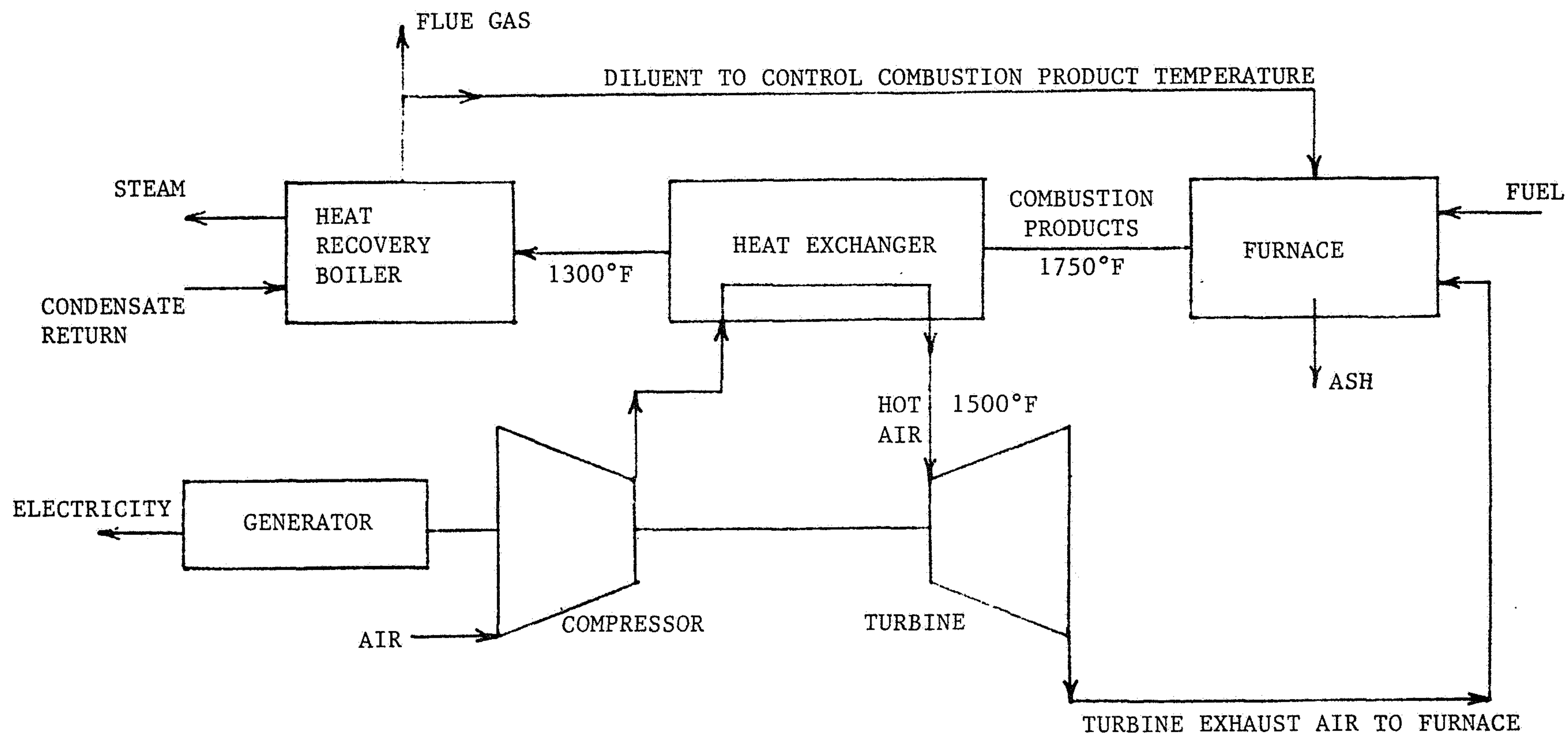


Figure 4-7. Indirectly-Fired Gas Turbine Engine Providing Combustion Air For a Furnace/Steam Boiler (Gas Turbine Engine Blown Furnace/Boiler)

Furnace/Boiler, is discussed in References 43 and 44. The compressor discharge air in this system is indirectly heated to 1550°F as before; however, the similarity of the two indirectly-fired systems ends here. In the GTE-Blown Furnace/Boiler cogeneration system, the turbine exhaust air is ducted to the furnace, where it supports combustion at an excess air level of 15% to 20%. The heat to drive the gas turbine is extracted from the furnace products in the heat exchanger, then the combustion products pass through a heat recovery boiler to generate steam. The air flow through the gas turbine engine will support a combustion firing rate of three to four times that of the indirectly fired system of Figure 4-6. Consequently, the heat available for steam production in the heat recovery boiler is much higher than in systems in which the firing rate is set to match the heat input requirement of the gas turbine engine. Due to the high firing rate in the furnace, the fuel to electricity heat rate for the system is high at 67,600 Btu/kW-h, even though the gas turbine engine itself is operating at the same efficiency as in the AFBC/GTE/HRB system. The additional energy made available by firing beyond that rate required to supply heat for the GTE is recovered for steam generation in the HRB. Consequently, the system's thermal output of 46.9 lb/kW-h is considerably higher than that of the other GTE cogeneration systems. The temperature of the combustion products is controlled by recycling a large fraction of the flue gas. Flue gas recycle is common industrial practice in certain operations, e.g., moderate temperature kiln drying. In this system, the gas mixture entering the GTE air heater must be tempered to 1750°F.

The capital cost estimates shown in Table 4-4 are based on the information of Reference 2 for the basic GTE/HRB module and on the information of References 22, 68, 69, 71 and 72 for the combustor and gasifier, i.e., fuel utilization units. This information was corroborated by the results of an analysis of the costs given in Reference 10. Cost estimates for the fuel utilization units were formulated by scaling the costs given in References 22, 69, 71 and 72 to account for different fuel firing rates and then adjusting the cost of the fuel utilization units to consistently reflect the relative differences among the units in physical size and layout, materials of construction, and the type and number of processes carried out in the units. Based on the electricity output of the particular gas turbine cogeneration system incorporating a given fuel utilization unit, the costs thus obtained were: $410/kW for the air-blown coal gasifier, $250/kW for the PFBC combustion unit, $560/kW for the AFBC combustion unit, and $840/kW for the furnace/heat exchanger/boiler unit of the second indirectly-fired system. These costs are distorted because they only reflect the cogeneration system's electricity output and do not account for the thermal output. Viewed in this manner, the capital cost in $/kW for a cogeneration system would approach infinity as the ratio of electrical to thermal output approaches zero. However, in calculating the cost of electricity, all of the cogeneration system's characteristics are normalized to the basis of electricity output; thus, the thermal output and capital cost are properly considered. The costs of electricity are shown in Table 4-5 for the four advanced gas turbine cogeneration systems and for the conventional system. The coal gasifier and the PFBC systems have the lowest costs of electricity (COE) of the advanced systems. At a fuel cost of $1.25 to $2.50 per million Btu, the coal gasifier and PFBC systems have a COE that is competitive with that of the conventional GTE at a fuel cost of $5.00 to $7.50 per million Btu's. The two indirectly fired GTE systems have high costs of electricity due to the inefficiency and extra cost of indirectly fired systems.

Table 4-5. Cost of Electricity for Gas Turbine Cogeneration Systems (mills per kilowatt-hour)

System Type		Fuel Cost, $ per million Btu			
Fuel	Configuration	1.25	2.50	5.00	7.50
Natural Gas/ Distillate	Conventional GTE[a]/HRB[b]		40	51	62
Coal/ Multi-Fuel	Integrated, Air-Blown Gasifier/ GTE/HRB	55	64	82	100
Coal/Other Solid Fuel	PFBC[c]/GTE/HRB	52	59	72	85
Multi-Fuel (Low Ash)	GTE-Blown Furnace/Boiler	85	91	104	117
Coal/Other Solid Fuel	AFBC[d]/GTE/HRB	78	88	109	130

[a]Gas Turbine Engine.
[b]Heat Recovery Boiler.
[c]Pressurized Fluidized Bed Combustor.
[d]Atmospheric Fluidized Bed Combustor.

4.2.3 Reciprocating Engine Systems

Technology Status. Advanced high-temperature materials that will operate in a more rigorous stress-temperature regime than conventional metals and alloys have been under development for more than two decades. The successful development and utilization of such materials would open new possibilities in the design of reciprocating, internal-combustion engines. The work to date on the use of high-temperature components for reciprocating engines has been carried out only for truck engines with medium-sized cylinders. The approach taken herein is to review the applicable work on such engines and then to estimate by means of similarity principles the impacts of high-temperature components on the performance characteristics of larger engines more suitable for cogeneration applications.

The use of high-temperature components in diesel engines does not, per se, bring about any increase in the thermal efficiency of the basic diesel cycle. As discussed by Taylor (Ref. 45) and Moore (Ref. 46), the thermal efficiency of the ideal diesel or limited pressure cycle depends only on the compression ratio and the schedule of heat addition, i.e., combustion. Therefore, any increase in efficiency as a result of high-temperature components must occur through the reduction of cylinder heat losses. If such reductions are made during the compression and expansion processes, the engine's thermal efficiency could be increased. The actual heat loss for smaller cylinders during these cycle phases has been estimated at 7% to 12% of the fuel energy, according to experimental information given in Reference 43. Thus, with a thermal efficiency of say 35%, the increase in engine efficiency resulting from total elimination of heat losses would be from two to four points of thermal efficiency. In large engines, the heat loss per unit of cylinder volume may be smaller, as indicated by the analyses of References 45 and 46. Therefore, any increase of engine brake thermal efficiency would be even smaller. A reduction of heat losses during the exhaust process would increase the amount and the temperature of the heat available from the engine's exhaust gas. Hence, the prospects for the use of bottoming cycles or waste heat utilization in a cogeneration system would be increased. These possibilities are embodied in the concept of the "uncooled or adiabatic engine." The terms "uncooled" or "adiabatic" indicate only that the use of high-temperature materials permits operation without a means of active cooling and with insulation to reduce heat losses.

The reduction of heat losses through the use of advanced materials and improved design has been the subject of research by the Cummins Engine Company under sponsorship of the U. S. Army Tank-Automotive Research and Development Command (References 47, 48, and 49). This work has been conducted for truck-sized engines with cylinders of 5 1/2 inches bore and 6 inches stroke whose typical characteristics are shown in Table 4-6. Cummins has concluded that the most important components to insulate are the piston, cylinder liner, cylinder head, valves, exhaust ports, and exhaust manifold. These components could be insulated by using a material that has low thermal conductivity and high-temperature durability. Unfortunately, many insulating materials have poor strength at high temperatures, e.g., lithium alumina silicates. Cummins has found that a good combination of insulation and durability can be achieved with silicon nitride engine components when surface-roughened metal shims are stacked and inserted at various locations along the head and block to reduce thermal conductance. The use of zirconium ceramics for engine components is being actively investigated with promising results thus far

Table 4-6. Performance of Conventional and Uncooled Diesel Engines (Reference 47)

(Engine Parameters: 2100 rev/min, 28:1 air-fuel ratio; Turbo-charged; After cooler coolant temp. = 180°F; Peak cylinder pressure = 1800 psi; $BSNO_2$ emissions = 5 gm/Hp-h)

Engine Configuration	Fuel Consumption BSFC, lb/Hp-h	Brake Thermal Efficiency, %	Relative Efficiency Improvement Ratio
C1. Conventional Engine	0.373	37.9	1.00
U1. Uncooled Engine with Ceramic Hot Parts	0.362	39.1	U1:C1 = 1.03
C2. Turbo-Compounded, Conventional Engine	0.309	45.8	C2:C1 = 1.21
U2. Turbo-Compounded, Uncooled Engine	0.285	49.6	U2:C1 = 1.31 U2:U1 = 1.27 U2:C2 = 1.08
U3. Uncooled Engine with Rankine Bottoming Cycle	0.272	52.0	U3:U1 = 1.32
U4. Turbo-Compounded, Uncooled Engine with Rankine Bottoming Cycle	0.255	55.5	U4:U2 = 1.12

(Ref. 50). Measurements on insulated engines with ceramic components have shown an increase in exhaust gas temperature from 1150°F to 1450°F.

Kamo and Bryzik (Ref. 47) have analytically investigated the performance of alternative configurations of diesel engines, including naturally aspirated, turbo-charged, turbo-compounded, and Rankine bottomed versions of both conventional and uncooled engines. These analytical investigations have been supported by experimental work with uncooled engines; thus, the comparative and absolute values for nominal engine efficiencies are considered to be valid for the design parameters employed, e.g., cylinders of about 6 x 6 in., bore x stroke. A portion of the results given in Reference 47 are presented in Table 4-6. These results show that only minor gains in brake efficiency are the direct result of uncooled/insulated engine operation, but the additional exhaust gas energy can be recovered by turbo-compounding, by incorporating a Rankine bottoming cycle or by cogeneration. The overall brake efficiency for different engine configurations is given in Table 4-6.

The parameters and energy balances for four sizes of engine cylinders ranging from 143 to 82,740 cubic inches per cylinder are shown in Table 4-7. The remarkable similarity among the energy balances for these cylinders shows how little the basic cycle thermodynamic parameters change with cylinder size. However, the exhaust gas temperature of the large two-stroke cylinders is notably lower than that of the four-stroke cylinders. The two-stroke cylinders are uniflow-scavenged as discussed in References 51, 52 and 54. The two-stroke scavenging process requires that a large amount of air be blown longitudinally through the cylinder from bottom to top in order to expell the expanded combustion products. Considerable mixing of the combustion products with inlet air occurs with the concomitant reduction of the average temperature of the cylinder exhaust gas as measured in the exhaust manifold.

Table 4-7. Parameters and Energy Balance for Cylinders of Differing Size

Engine Parameter	Conventional Four-Stroke	Conventional Four-Stroke	Conventional Two-Stroke	Advanced Un-cooled Engine Four-Stroke
Bore X stroke, inches	5 1/2 x 6	17 x 22	26 x 55	5 1/2 x 6
Single cylinder displacement, in.^{3}	143	4993	29200	143
Crankshaft speed, rev/min	2100	360	150	
Single cylinder power, BHP	58	256	2145	60 to 75
Inlet manifold pressure, atm	2.0	1.6	2.9	2 to 3
Mean piston speed, ft/min	2100	1320	1375	2000
Brake mean effective pressure, psi	154	135	188	100 to 200
Brake thermal efficiency, %	36	40	43	37 to 41
Exhaust heat, % of fuel energy	35	35	32	50
Coolant heat, % of fuel energy	25	4	22	None
Misc. losses, % of fuel energy	4	5	3	5
Exhaust gas temperature, °F	1070	850-950	590	1450
Reference	(45,46,47)	(53)	(51,52,54)	(47,48,49,50)

Due to the cylinder scavenging method that is used for the large two-stroke engines, the use of uncooled cylinder parts would have a substantially smaller effect on their exhaust gas temperature. Thus, the potential for recovering exhaust gas energy for cogeneration would be correspondingly less. However, significant economic benefits may be available if ceramic materials are applied to increase the durability of piston crowns, exhaust valves, and fuel injectors, among other parts, especially when burning fuels containing corrosive agents.

The larger four-stroke engine cylinders could show about the same efficiency characteristics as are shown in Table 4-7 for smaller cylinders if certain changes in engine design are made to fully exploit the anticipated high-temperature durability of ceramic parts. In particular, exhaust valve and piston crown temperatures are now controlled through an extended valve overlap period that occurs near the piston's top center position between the exhaust and intake strokes. During this valve overlap period, inlet air is blown through the combustion chamber to cool the piston crown and exhaust valves. This cooling air dilutes the cylinder exhaust gas and lowers the average gas temperature in the exhaust manifold in a manner similar to, but not to the extent of, the scavenging air of the two-stroke engine. The cooling of hot upper cylinder parts would not be necessary with ceramic high-temperature materials; hence, the valve overlap period could be changed to reduce the amount of exhaust gas dilution. Such a change could be implemented in the four-stroke cycle because cylinder scavenging to expell exhaust gas is accomplished by a complete upward stroke of the piston. In contrast to the larger four-stroke clyinders, smaller four-stroke cylinders have an advantage of scale in that the exhaust valves and piston crowns have shorter conductive heat flow paths. Thus, the extended air flow period is not required for cooling. The characteristics of the advanced uncooled engine of Table 4-7 were developed for cylinders of about 6 inches x 6 inches, bore x stroke. These characteristics could apply to the larger cylinders if the increased durability of ceramic parts were exploited in order to provide the increase in exhaust gas temperature that is available from the uncooled engine. The uncooled operation of medium to large fourstroke cycle diesel engines could give an increase in exhaust gas temperature from typical values of 850-1100°F to 1200-1500°F.

The Direct Combustion of Coal in Diesel Engines. The concept of burning coal directly in reciprocating, compression-ignition, internal-combustion engines dates back to the early experiments of Rudolf Diesel (circa 1892). Diesel's early work revealed problems that remain unresolved in developing an acceptable fuel metering and delivery system, in achieving acceptable combustion characteristics, and in avoiding excessive cylinder wear when coal is burned directly in internal-combustion engine cylinders. Subsequent work in Germany from the early 1900s through World War II failed to resolve these difficulties. Consequently, little attention has been given to coal burning engines in the post World War II era until recent work sponsored by the U. S. Department of Energy. In this work, mixtures of coal and fuel oil are being burned in very large, low-speed, marine-type diesel engines. Even if these tests show that fuel oil mixed with nominal amounts of coal can be burned in such engines, the ramifications in terms of expanded use of coal are extremely limited due to the very narrow and specialized applications for which these large, low-speed engines are suited. In order for compression-ignition, internal-combustion engines to realize the benefits of burning inexpensive fuel for cogeneration, the fundamental problems that now preclude burning coal directly must be resolved.

In view of the lack of success of past work in addressing the above mentioned problems of fuel delivery, combustion rate, and cylinder wear, a new approach to the direct use of coal in compression ignition engines is in order. Past work carried out at the Jet Propulsion Laboratory has shown that coal can be extruded as a plastic medium when subject to the proper conditions of shear deformation, pressure, and elevated temperature. The coal extrusion process might be utilized to directly inject plasticized coal into the cylinder of a compression ignition engine. The combustion rate of the extruded coal might be increased through the blending of appropriate additives during the fuel delivery and injection processes. The injection of coal in the plastic into the engine cylinder might also provide a means to control the dispersion of the ash-plume within the cylinder, thereby reducing the amount of ash reaching the cylinder wall to prevent excessive wear. Additives that modify the physical characteristics of the coal ash might also be used as a means of reducing ash-induced cylinder wear due to abrasion. And with the advent of dry lubricants for ceramic pistons and cylinder walls, ash-induced wear might be reduced to acceptable levels.

Coal Gas-Fired Engines. Reciprocating engines may be fired with gaseous fuel that is inducted into the cylinder as a fuel/air mixture and ignited at the proper instant with an electrical spark as in a conventional Otto cycle engine. Combustion may, in some engines, be initiated with a pilot spray of fuel oil which ignites as in a compression ignition engine. Usually, the auto-ignition temperature of gaseous fuels is above that of a fuel oil spray, so pilot spray ignition is feasible in engines with the proper compression ratio. Gasfired engines are presently in widespread commercial use.

Coal gas from an air-blown gasifier has a heating value of about 125 Btu per standard cubic foot (SCF), which is low in comparison to natural gas with a typical heating value of about 1000 Btu per SCF. However, the heat release per unit volume of a stoichiometric, i.e., chemically correct, fuel air mixture of low Btu coal gas is about 60% of that for a stoichiometric fuel-air mixture of methane (natural gas). If a normally aspirated reciprocating engine were fueled with coal gas, the power output would be reduced by about 40%; however, turbocharging to increase the density of the coal gas fuel/air mixture inducted into the cylinder can be used to restore the engine's output. Gas-fired reciprocating engines with natural gas fuel are normally turbocharged to increase the induction pressure to about 2 atmospheres. The turbocharged outlet pressure could be increased to about 4 atmospheres to allow an engine fueled with air-blown coal gas to achieve the normal power output. Some engine system modifications would be necessary but are well within present technological capability.

Cogeneration System Characteristics. In this study, reciprocating engine cogeneration systems were based on both conventional and uncooled ceramic engines. Both types of engines were analyzed with air-blown gasifiers to enable the cogeneration system to be coal fueled. The uncooled ceramic engine was also considered in a direct coal-fired configuration and as a distillate fueled diesel engine. The advanced uncooled reciprocating engine cogeneration system is shown in Figure 4-8.

The system characteristics shown in Table 4-8 for distillate and gas-fired conventional engines were based on present commercial technology.

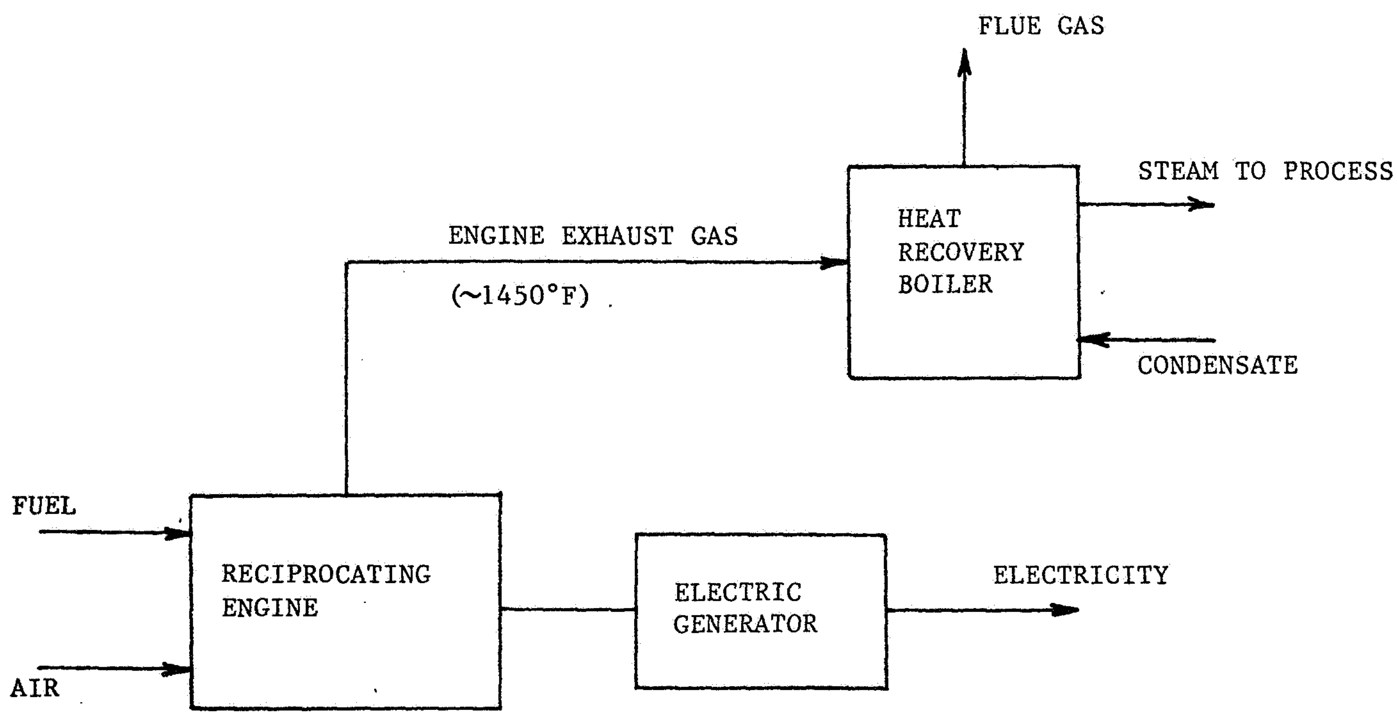


Figure 4-8. Advanced Uncooled (Ceramic) Reciprocating Engine Cogeneration System

Table 4-8. Reciprocating Engine Cogeneration System Characteristics

System Type		Capital Cost, $/kW	Heat Rate, Btu/kW-h	Output Ratio of Heat to Electricity, lb of Steam/kW-h
Fuel	Configuration			
Natural gas	Conventional Gas-Fired Engine	1290	11320	1.7
Distillate	Conventional Diesel Engine	1290	9980	1.6
Coal	Conventional Engine/ Coal Gas Induction/ Coal Gasifier	1706	14700	3.3
Distillate	Uncooled Ceramic Diesel Engine	1290[a]	9690	3.1
Coal	Uncooled Ceramic Engine/ Direct Coal Injection	1290[a]	9690	3.1
Coal	Uncooled Ceramic Engine/ Coal Gas Induction/ Coal Gasifier	1706[a]	14300	4.9

[a]Capital costs for uncooled ceramic engines are unknown but are arbitrarily assumed to be those of the conventional engine to calculate a cost of electricity.

The cost and performance parameters for the conventional engines were derived from the information given in Reference 2 and from detailed performance information provided by engine manufacturers. The heat rates of the ceramic uncooled diesel and gas-fired engines were taken as 97% of those of their conventional counterparts, according to prior discussion. The exhaust gas temperature of the uncooled diesel engine was assumed to be 1450°F, as shown in Table 4-7. Assuming that the engine exhaust gas mass flow rate, the flue gas stack temperature of 280°F, and other pertinent engine and heat recovery system parameters are not changed in the uncooled engine system, the heat recoverable from the uncooled diesel engine's exhaust gas stream will be increased by a factor of about 1.9. The thermal output of the uncooled gas-fired engine was assumed to increase by the same factor of 1.9. The heat rate and thermal output of the direct coal-fired, uncooled ceramic engine were assumed to be the same as those of the distillate-fueled, uncooled engine.

The air-blown coal gasifier characteristics used for the gas turbine engine and the fuel cell cogeneration systems were also used in estimating the overall system heat rates and thermal output ratios shown in Table 4-8 for the reciprocating engine system. The air-blown coal gasifier has a coal feed to coal gas energy efficiency of 77%; hence, the coal feed to electricity efficiency of the gasifier engine system is reduced to 77% of the engine heat input to electricity efficiency. However, the system thermal output is increased because 11% of the coal feed energy is recoverable from the gasifier as useful heat. The total thermal output is that of the gasifier plus that of the engine exhaust gas heat recovery system.

The capital costs shown in Table 4-8 for the conventional reciprocating engine systems were taken from Reference 2. The costs of Reference 2 are based on supplier's quotations for equipment and standard estimates for installation/construction costs. The cost of the air-blown coal gasifier was $420/kW. Gasifier costs were previously discussed in Section 4.2.2 and 4.2.5. Capital cost estimates for uncooled, ceramic engines are speculative at the present time because manufacturing processes and production techniques are not established. Also, no generically similar types of equipment with ceramic components are known to exist, thus no reference is available for relative cost estimates. Consequently, costs of electricity for the uncooled, ceramic engine systems were calculated using the capital costs of the conventional engine systems. The costs of electricity obtained in this manner indicate the potential of uncooled ceramic engines if their assumed capital costs and performance gain some measure of credibility as uncooled engine development and ceramic materials research continue.

The costs of electricity for the reciprocating engine systems are shown in Table 4-9. The costs of electricity for the conventional distillate and gas-fired systems are in the higher range at 77 to 124 mills/kW-h, reflecting the relatively low energy usage efficiency of the conventional reciprocating engine cogeneration systems. The coal gas-fired systems all have high electricity costs due to the additional capital cost of the coal gasifier. The uncooled ceramic engine systems benefit from the additional heat available in the engine's exhaust, and, if direct coal-firing were possible, from the low fuel cost of coal. These factors are responsible for the low electricity cost of from 65 to 99 mills/kW-h for the direct coal-fired and distillate fueled, uncooled, ceramic engine systems.

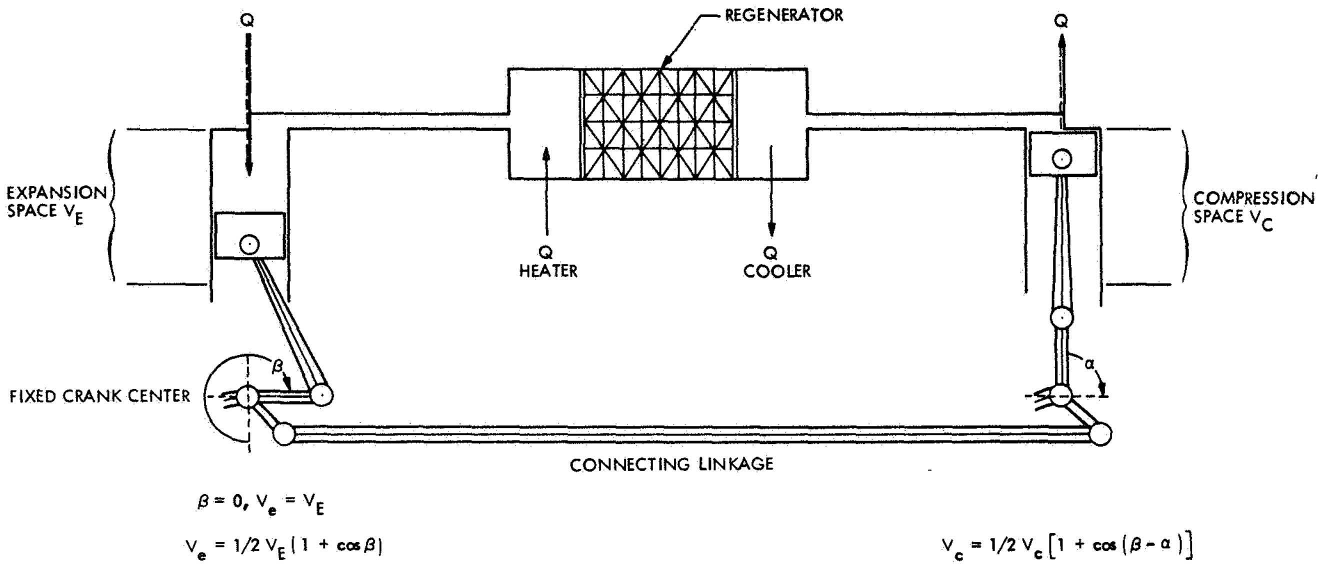
Table 4-9. Cost of Electricity for Reciprocating Engine Cogeneration Systems (mills per kilowatt-hour)

System Type		Fuel Cost, Dollars per Million Btu			
Fuel	Configuration	1.25	2.50	5.00	7.50
Natural Gas	Conventional Gas-Fired Engine		79	102	124
Distillate	Conventional Diesel Engine		77	97	117
Coal/Multi-Fuel	Conventional Engine/ Coal Gas Induction/ Coal Gasifier	87	100	126	
Distillate	Uncooled Ceramic Diesel Engine		72	85	99
Coal/Multi-Fuel	Uncooled Ceramic Engine/ Direct Coal Injection	65	72	85	
Coal/Multi-Fuel	Uncooled Ceramic Engine/ Coal Gas Induction/ Coal Gasifier	84	94	113	

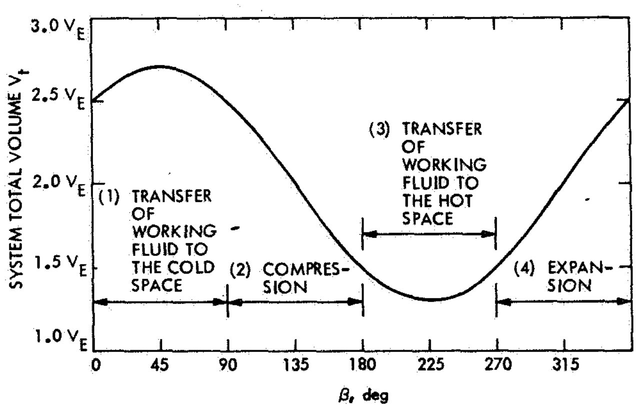
4.2.4 Stirling Engine Systems

Technology Status. Stirling engines are closed-cycle heat engines that convert heat to mechanical work by the cyclic compression and expansion of a confined working fluid, usually hydrogen or helium, which is maintained at a higher temperature during expansion and a lower temperature during compression. All Stirling engines operate on the same basic thermodynamic cycle regardless of the means of mechanizing the engine. The ideal Stirling thermodynamic cycle consists of isothermal compression and expansion processes with regenerative recovery of heat from the working fluid after expansion. Thermal regeneraton in Stirling engines is accomplished via an unsteady flow heat exchanger that stores heat recovered from the working fluid after expansion and returns heat to the working fluid after compression.

The Stirling engine is distinguished from other heat engines in that the cyclic flow of the working fluid within the engine is achieved only through geometric volume changes without the use of intermittently opened ports or valves. The Stirling cycle may be implemented with a mechanical, kinematic linkage of pistons to accomplish the required volume changes in the hot and cold sections of the engine in the proper phase relationship. The phased volume changes are illustrated in Figures 4-9 (a) and (b). The working fluid is cyclicly displaced between the hot section of the engine where expansion occurs and the cold section where compression occurs. The hot and cold engine sections are connected by a gas flow path consisting of a heater section, a regenerator, and a cooler section. Historical reviews of the Stirling engine and discussions of operating principles along with mechanical implementations of the Stirling cycle are given in References 1 through 9.



(a) Two-piston mechanism illustrating the operation of a Stirling engine



(b) Variation of total working space volume with crank rotation angle

Figure 4-9. Operating Principles of Stirling Engines

The Stirling engine offers high energy conversion efficiencies circa 40% (mechanical power output/heat input) at the present state of technological development, as shown in Table 4-10. The Stirling engine is currently limited to a hot side temperature of about 1400°F by the creep-strength/temperature characteristics of the nickel-based superalloys used to fabricate the engine's hot components. The stresses in the Stirling engine are high due to the high pressure (1500 to 3000 psia) at which the working fluid must be confined in order to obtain a satisfactory specific power output from the engine, i.e., power output per unit weight and volume. Even in stationary applications, a reasonable specific power output is required to prevent high manufacturing costs due to the engine's excessive size. The development of ceramic materials for use in hot stressed components could offer efficiency improvements and, more importantly, extend the cyclic fatigue life of such components. Stirling engines have only been operated for relatively short periods of time (hundreds of hours) in prototype configurations. Major difficulties have been encountered in controlling leakage of the high pressure working fluid past seals between moving parts, e.g., piston rods. The reliability, durability, and acceptable manufacturing cost of the Stirling engine must be demonstrated before the Stirling engine will be a viable contender for industrial cogeneration applications.

Cogeneration System Characteristics. The advanced Stirling engine cogeneration system considered in this study is shown schematically in Figure 4-10. The heat rejection temperature of the engine is increased from a typical value of 160°F as shown in Table 4-10 to 280°F in order to generate 40 psia steam in a heat recovery boiler. The increase in heat rejection temperature reduces the engine brake efficiency by a factor of 0.1 (i.e.,

Table 4-10. Typical Stirling Engine Parameters

Manufacturer	Philips 4-215	Philips	United Stirling	GMRL GPU-3	Philips 4-235	Philips 40 hp	United Stirling	MAN-MWM 4-400
Status	Proto (Ford)	Analy. (optimized)	Proto	Proto	Proto	Proto	Analy. phase I	Proto
Type	Two piston	Piston-disp	Two piston	Piston-disp	Piston-disp	Piston-disp	Two piston	Piston-disp
Working fluid	H_2	He	H_2	H_2	He	H_2	H_2	H_e
Max press. $\overline{P}$, psi	2850	3200	2100	1000	3200	2058	2100	1570
No. of cylinders	4	4	4	1	4	1	8	4
Max bhp	170	275	49	11	200	40	200	120
RPM at max power,	4000-4200	1600	3400	3600	3000	1500	2400	1500
Max torque, ft-lb	300	1287	120	19	253	108	520	475
RPM at max torque	1400	400	955	1200-2400	1000	900	600	700
Gas temp (hot), °F	1300	~1400	1275	1400[a]	1260	1200	1325	1170
Gas temp (cold), °F	175	160	160	180	108	60	160	105
Efficiency at max BHP (%)	24	30	24	25	30	30	30	29
Max efficiency, %	32[b]	43[b]	30	26.5[b]	31	38	35	32
Power at max efficiency, BHP	75	100	35	~7	175 (approx)	23	76	88
RPM at max efficiency	1100-2000	600	2000	1900	1800	725	1200	1000
Weight,[c] lb	750	N/D	N/D	165[d]	1272	N/D	1435	N/D
Dimensions,[c] ft	N/D	4.9 x 4.3 x 2.2	N/D	1.3 x 1.3 x 2.4[e]	4.1 x 1.7 x 3.6	N/D	3.7 x 2.7 x 3.1	5.0 x 2.3 x 4.3
Applications	Auto	Bus	Auto	EPS	Bus	LRE	Bus, truck	LRE
References	6-6, 13, 22, 27	6-5	6-25, 28	6-3, 26, 38	6-39	6-8, 10	6-25	6-24

[a]Heater tube wall temperature.

[b]Net brake efficiency accounting for all auxiliaries including cooling fan, combustion blower, and water pump, among others.

[c]Includes all auxiliaries except cooling system with fan and transmission.

[d]Engine and auxiliaries less electrical power generator.

[e]Engine only.

Abbreviations:

Proto: operating prototype engine; LRE: Laboratory Research Engine; Analy: computer design projection; N/D: no date; EPS: electric power supply.

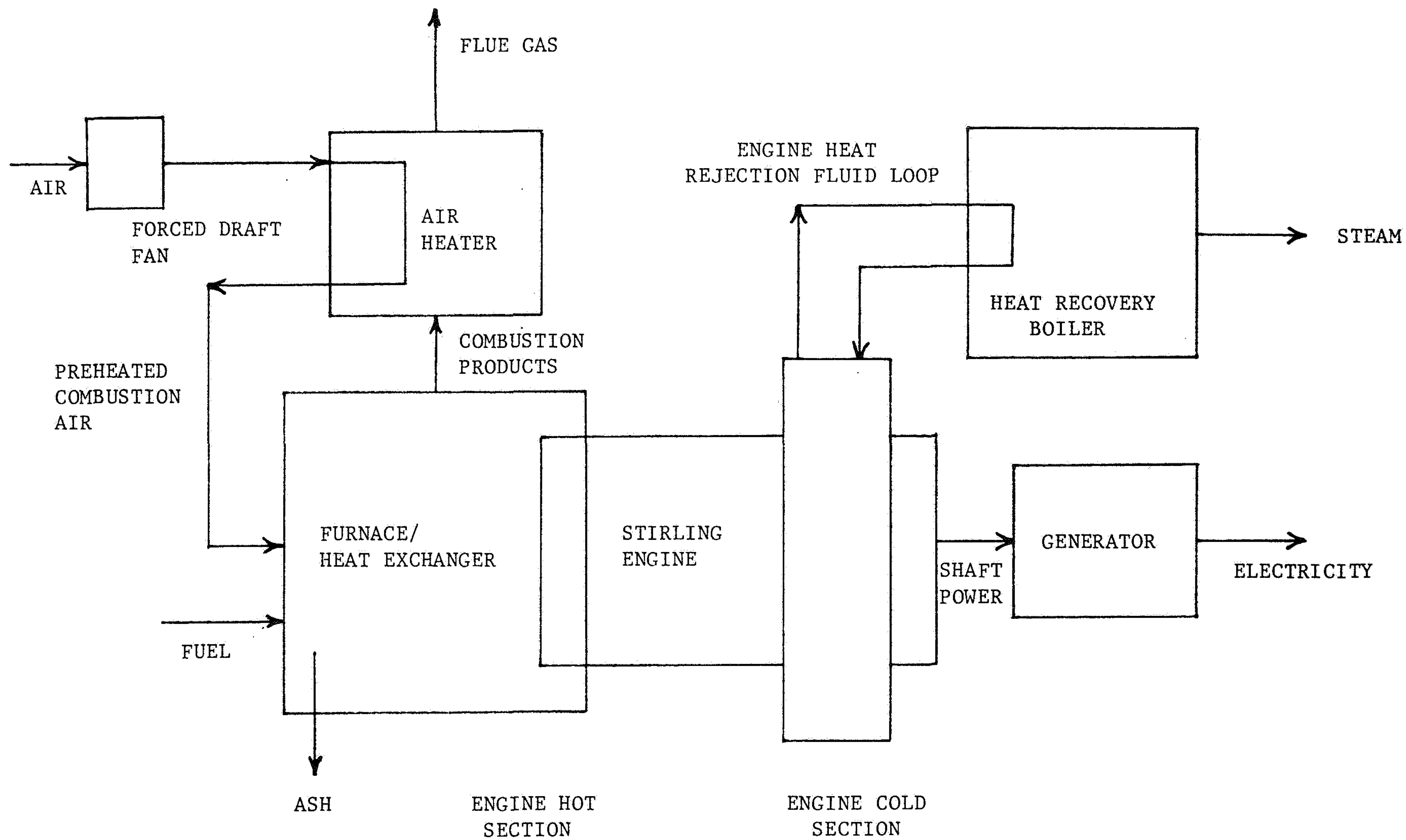


Figure 4-10. Schematic of a Stirling Engine Cogeneration System

efficiency at 280°F = 0.9 x efficiency at 160°F). The engine's brake efficiency with a heat rejection temperature of 280°F was taken as 39%, before accounting for electrical power generation at an efficiency of 95% and combustor losses of 15%. The high-temperature heat for the closed cycle Stirling engine is provided by an atmospheric fluidized bed combustion furnace capable of burning biomass, coal, or refuse-derived fuel. This furnace was assumed to deliver 85% of the heat of the burning fuel to the Stirling engine, which would result in an overall system efficiency of 31% after furnace and electrical generator losses. The furnace efficiency of 85% is typical of units of this type and perhaps optimistic for furnaces of the required size. The Stirling engine itself is conjectured to be of the double-acting, Rinia type with the cylinders in a V arrangement. In such an engine, the angle between the V cylinder banks can provide the proper volume change phasing if opposing cylinders are connected to the same crankshaft throw. The engine would likely have 16 to 24 cylinders to provide 3000 to 5000 kW of power output. The engine is assumed to reject 51% of the heat input to the working fluid to the heat rejection cooling loop. After a 3% heat loss in the cooling loop and the 15% loss of the furnace, the rejected heat generates 40 psia steam at the rate of 4.5 lb per kilowatt-hour of electrical output. These system characteristics are summarized in Table 4-11.

The capital cost estimate for the Stirling engine cogeneration system was based on that of the reciprocating engine system with adjustments for major differences between the systems. The manufacturing cost of Stirling engines is about 26% higher than that of turbocharged diesel engines according to the analyses presented in Reference 1 (Chapter II). After deleting the cost of the combustion system from the Stirling engine (the estimate of

Table 4-11. System Characteristics and Cost of Electricity for Stirling Engine Cogeneration Systems

System Characteristics	
Engine Type	Multi-Cylinder, Double-Acting, Rinia
System Heat Rate, Btu/kW-h	10,900
Output Ratio of Heat to Electricity, lb steam/kW-h	4.5
Capital Cost, $/kW	
AFBC Furnace	1,670
Conventional Fired	1,460
Engine Hot Side Temperature, °F	1,400
Engine Cold Side Temperature, °F	280
Furnace Efficiency, %	85
Engine Efficiency, %	39
Generator Efficiency, %	95
Engine Heat Output, % of Fired Fuel	50

Cost of Electricity

	mills/kW-h	
Fuel Cost, $\$/10^6$ Btu	$1670/kW	$1460/kW
1.25	79	
2.50	85	77
5.00	98	89
7.5	110	100

Reference 1) the Stirling engine is found to cost 5% more than a diesel of the same power output. Assuming that the \$1290/kW for the reciprocating engine systems of this study is evenly divided between equipment and labor, the Stirling engine system is found to cost \$1320/kW without a combustion system. The cost of an AFBC furnace for a Stirling engine with a firing rate of 37 x 10^6 Btu/h was estimated to be \$350/kW by means of an analysis based on the information given for AFBC industrial boilers in Reference 22. The total capital cost of the Stirling cogeneration system with an AFBC furnace is then found to be \$1670/kW. This estimate is in all likelihood quite optimistic, but serves to represent the Stirling cogeneration system relative to the other systems of this study. A conventionally fired Stirling engine burning natural gas or fuel oil would, as discussed above, cost 26% more than a conventional diesel engine, which would place the installed cost of the conventionally fired Stirling engine at \$1460/kW.

Costs of electricity for the Stirling cogeneration system are also shown in Table 4-11. The cost of electricity ranges from 79 to 110 mills per kW-h as fuel cost increases from \$1.25 to \$7.50 per million Btu. The high cost of electricity is due to the high capital cost for the AFBC/Stirling engine cogeneration system. The electricity cost for the conventionally fired Stirling engine ranges from 77 to 100 mills per kilowatt-hour as fuel cost increases from \$2.50 to \$7.50 per million Btus; these costs are somewhat greater than those shown in Table 4-9 for the uncooled, ceramic reciprocating engine that burns distillate fuel.

4.2.5 Fuel Cell Systems

Technology Status. Fuel cells are energy conversion devices that continuously produce electricity by means of the electrochemical reaction of a fuel and an oxidizer, normally hydrogen and oxygen, in the presence of an electrolyte. The operating principles and technological status of fuel cells are discussed by Voecks (Ref. 64), by Fickett (Ref.65), and in Reference 66, and the present discussion is chiefly drawn therefrom. Fuel cells are generally considered to be of four types: (a) acid, (b) molten salt, (c) alkaline, and (d) solid oxide. For seemingly valid considerations given in the above mentioned references, the first two types may be reasonably expected to reach commercial status for industrial and utility applications. Hence, only acid and molten salt fuels will be considered herein for advanced cogeneration systems.

The four types of fuel cells are shown in Figure 4-11. All the fuel cells require hydrogen as the fuel; therefore, a fuel processor is used to convert a hydrocarbon or carbonaceous fuel into a hydrogen rich fuel gas to be fed into the cell. A notable difference between acid and molten salt fuel cells shown in Figure 4-11 is that the acid fuel cell requires a shift converter to reduce the molar concentration of carbon monoxide in the fuel gas to less than about 2% to avoid poisoning the platinum catalyst of the fuel cell anode. In contrast to the acid fuel cell, the molten salt fuel cell is not poisoned by carbon monoxide. The carbon monoxide (CO) is indirectly utilized in the molten salt cell in that CO is shifted within the fuel cell to form hydrogen, which is consumed at the anode, and carbon dioxide, which is recycled to the cathode where it participates in the electrochemical reaction. As shown in Figure 4-11, acid fuel cells operate at temperatures of about 400°F while molten salt cells operate near 1000°F.

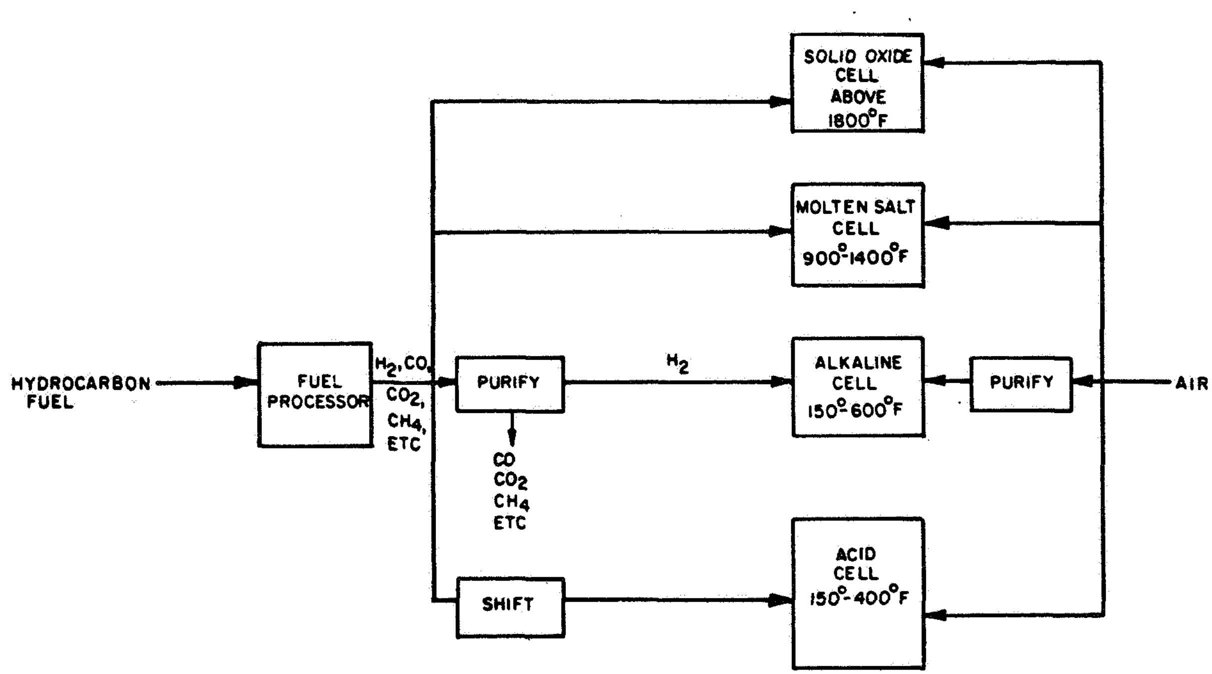
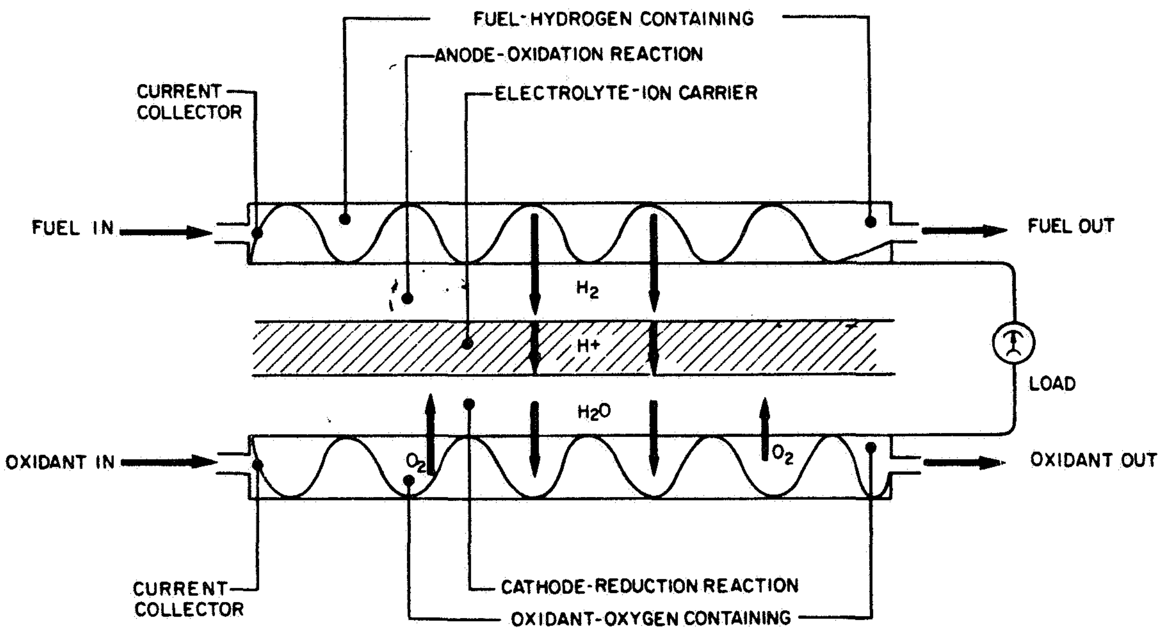


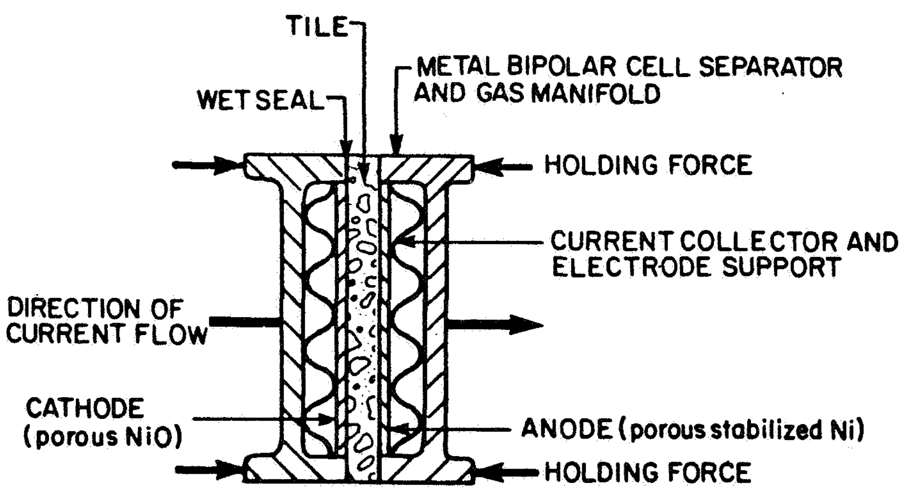
Figure 4-11. Illustration of Fuel Cell Types (Ref. 66)

Figure 4-12 illustrates the configuration of both phosphoric acid and molten carbonate fuel cells. Each fuel cell element consists of a porous anode to which fuel is fed, a porous cathode to which oxidizer is fed, and the electrolyte saturated wick structure that separates the anode and cathode. In the acid fuel cell, hydrogen in the fuel stream surrenders electrons at the anode, then the hydrogen ions diffuse through the acid electrolyte to the cathode where the ions react with oxygen and accept electrons to form water vapor. The product water diffuses out of the cathode and is carried away with excess air flowing through the cathode channels. In the molten carbonate fuel cell, hydrogen also surrenders electrons at the anode, subsequently reacting with carbonate ions to form water vapor and carbon dioxide gas that diffuse out of the anode and are carried away by the anode gas stream. The carbon dioxide is externally recycled to the cathode where it reacts with oxygen and accepts electrons to form carbonate ions that diffuse through the electrolyte to the anode.

Phosphoric acid fuel cell electrodes typically consist of agglomerated carbon particles loaded with the platinum catalyst that are held together by a fluorocarbon binder. The phosphoric acid electrolyte is held in an inert wick matrix of inorganic or polymeric material. Molten carbonate fuel cells use porous, sintered nickel electrodes, and their electrolyte is held as a porous, ceramic tile made of lithium-aluminum-oxide containing a mixture of alkali carbonates, usually lithium carbonate and potassium carbonate. Both types of fuel cells must operate at elevated pressure to achieve satisfactory power density, about 120 psia for the next generation of acid fuel cells and for the postulated generation of commercial molten salt fuel cells.



Phosphoric Acid Fuel Cell Element



Molten Carbonate Fuel Cell Element

Figure 4-12. Schematic Illustrations of Phosphoric Acid and Molten Carbonate Fuel Cell Elements

A fuel cell energy conversion module consists of many individual fuel cell elements connected in series to give the desired output voltage; such modules are referred to as cell stacks. These modules are then connected in parallel to give higher electrical current output levels. Because a fuel cell produces direct current (dc), electrical power conditioning equipment is required to convert dc power to 60 hertz alternating current for industrial applications. A fuel cell power plant for industrial cogeneration applications would then consist of three basic elements, as shown in Figure 4-13. The fuel processor converts a feedstock of liquid, gaseous or solid fuel into a hydrogen-rich fuel gas stream. This stream electrochemically reacts in the fuel cell stack to produce electricity. At practicable rates of energy conversion, the thermodynamic irreversibilities of the fuel cell energy conversion process result in 50% to 60% of the energy available from the fuel being dissipated as heat within the fuel cell elements. Thus, fuel cells require that heat be removed from the cell stack in order to maintain a constant fuel cell operating temperature. This heat serves to produce steam in a fuel cell cogeneration system.

Phosphoric acid fuel cells have been developed to the commercial demonstration stage over the past fifteen years. In the early 1970s, a group of 60 units of 12 kW output were tested in 35 locations in a variety of applications by United Technologies Corporation under the TARGET Program -- the Team to Advance Research for Gas Energy Transformation, which included a consortium of 32 gas utilities. These 12-kW units operated on natural gas fed to a self-contained fuel processing unit. A larger, 40-kW unit was developed with the benefit of the experience of the 12-kW program. The 40-kW unit has operated for approximately 1000 hours at two industrial sites in a program that is focused on field testing pre-prototype units. These units achieve

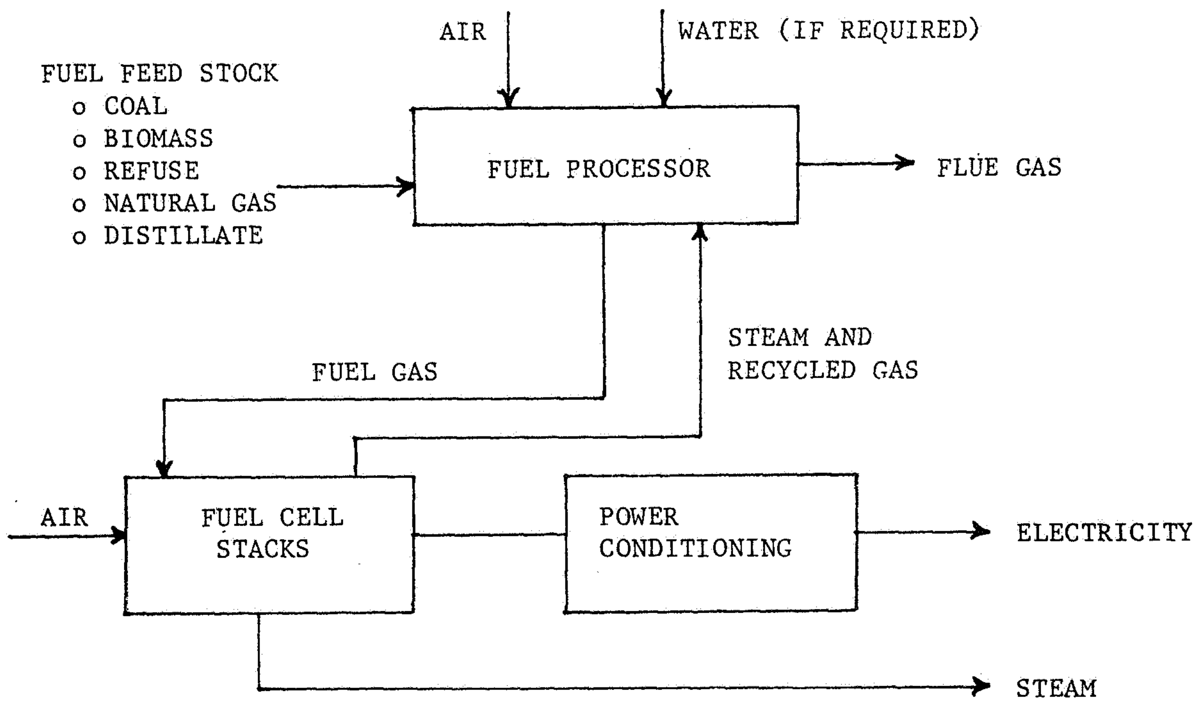


Figure 4-13. Schematic of a Fuel Cell Power Plant

an electrical conversion efficiency of approximately 38% of the fuel energy and demonstrate the potential useful heat recovery of up to 40% of the fuel energy (Ref. 67). A one-megawatt unit was tested for 1100 hours at 50 psia in preparation for constructing 4.5-MW units. A 4.5-megawatt unit is now being readied for testing in parallel, grid-connected operation in New York while another is in operation in Tokyo. These units are designed to operate on natural gas or naphtha that is processed through a self-contained reformer to obtain a fuel gas containing about 70% hydrogen. Developmental work for phosphoric acid fuel cells is aimed at increasing cell operating pressure to about 120 psia in order to provide a substantial increase in power density relative to the 4.5-MW units. Performance and durability improvements are needed in several areas in order for phosphoric acid fuel cells to be commercially viable. In present units, for example, electrolyte evaporation creates corrosion problems, and limited cathode performance presently suppresses fuel cell efficiency. The economic viability of phosphoric acid fuel cell systems in utility and industrial applications remains to be proven when viewed relative to the Rankine cycle, gas turbine engine, and reciprocating engine systems.

Molten carbonate fuel cells have the potential to reduce system costs and improve overall electrical conversion efficiency when combined with a bottoming cycle. Presently, molten carbonate fuel cells are being developed in the laboratory with attention directed to identifying cell configurations and materials with the required performance and lifetime.

Cogeneration System Characteristics. Advanced fuel cell cogeneration systems based on both phosphoric acid and molten carbonate fuel cell stacks were analyzed. The phosphoric acid fuel cell (PAFC) system was considered in three individual configurations that differ according to fuel type.

Phosphoric acid fuel cell systems were configured to run on natural gas/naphtha, liquid hydrocarbon distillate, or solid carbonaceous fuels such as biomass, coal, or refuse. The molten carbonate fuel cell (MCFC) was analyzed only in coal or other solid carbonaceous fueled configurations. The solid carbonaceous fueled systems for the PAFC and the MCFC were considered both with air-blown and oxygen-blown gasifiers. The two types of gasifiers have different fuel gas compositions and capital costs.

The natural gas/naphtha fueled PAFC system is shown schematically in Figure 4-14. In the system shown, the fuel is desulfurized and fed to the steam reformer where it reacts with steam generated from the cell stack to yield a gaseous mixture of hydrogen, carbon monoxide, carbon dioxide, and water vapor. Since only limited amounts of carbon monoxide can be fed to the PAFC, a shift converter is used to increase the hydrogen concentration in the gas stream by reacting carbon monoxide and water to form hydrogen and carbon dioxide. The hydrogen-rich fuel gas then flows to the fuel cell anode where about 80% of the hydrogen in the gas stream is electrochemically consumed. The anode gas exhaust stream with the residual hydrogen and unconverted methane is then burned with air to provide heat for the endothermic reforming process. The fuel cell cathode is supplied with air from the turbocompressor. The water of reaction and excess air are contained in the cathode exhaust gas stream. In the particular system shown in Figure 4-14, virtually all of the water vapor in the cathode exhaust is condensed, then returned as liquid water to the fuel cell to remove the heat being generated therein by means of heat exchange with a coolant circulating through the cell stack. A portion of the steam generated in cooling the cell stack is used in the fuel processor and the remaining steam constitutes the cogeneration

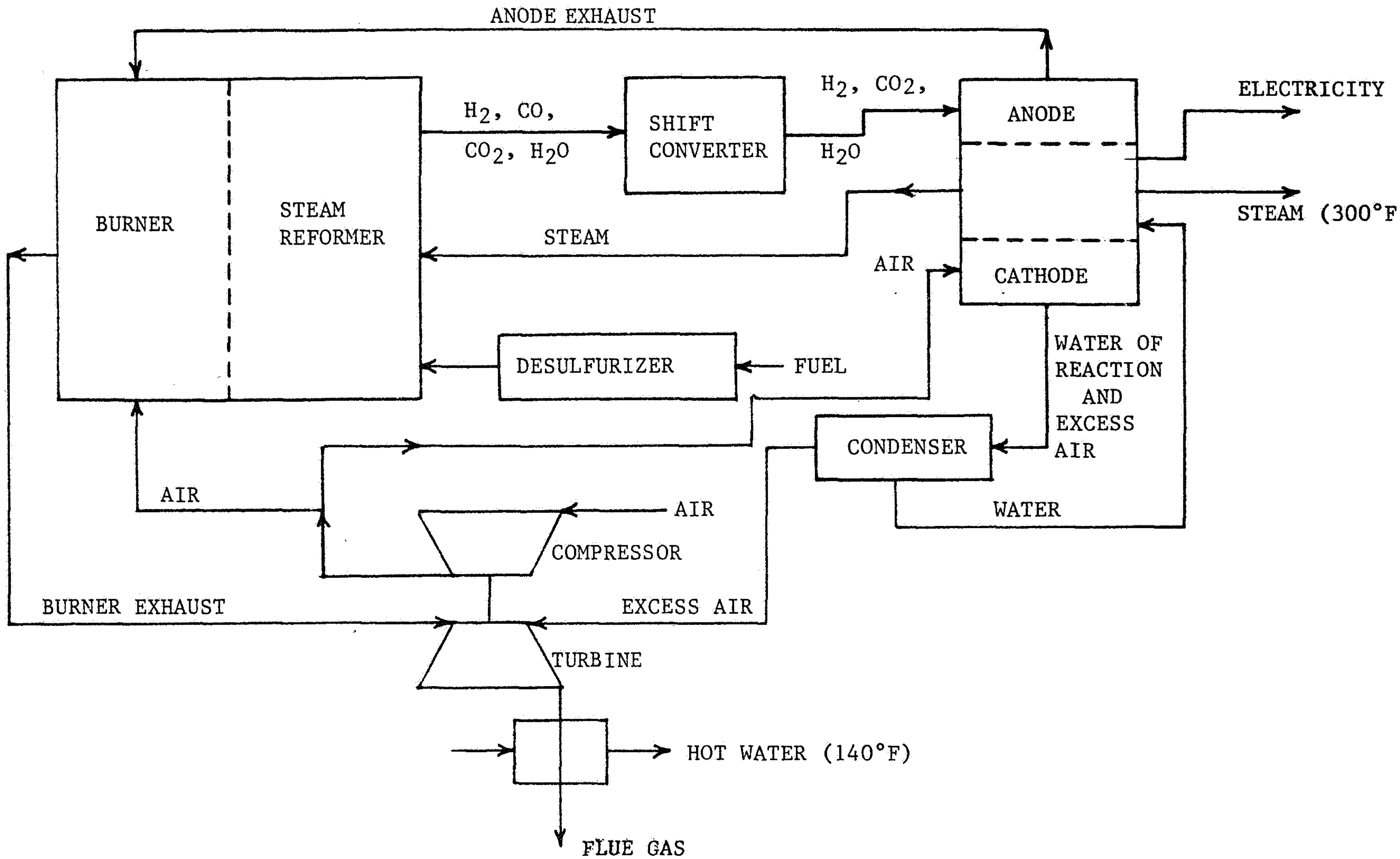


Figure 4-14. Phosphoric Acid Fuel Cell Cogeneration System Using Light Distillate Fuel (Naptha) in a Steam Reforming Fuel Processor

system's heat output. The performance of this system is presented in Reference 67 by the system's developers -- United Technologies Corporation. According to that information, the overall fuel to electricity efficiency of the system is 38%, which gives a system heat rate of 8980 Btu/kW-h shown in Table 4-12. The thermal output of this system is quite low at 0.79 lb of steam per kW-h because the steam fed to the reformer is obtained by condensing the water of reaction and then revaporizing the condensate to cool the cell stack.

These systems may be modified, at some sacrifice of electricity conversion efficiency, to substantially increase the system's thermal output. As described in Reference 67, the fuel processor may be directly fed with the cathode exhaust stream to provide the water vapor required by the reforming process. Such a system is shown in Figure 4-15. Only the water to be used for cell stack cooling is condensed and returned to the fuel cell. This modification increases the system thermal output to 4.4 lb of steam per kW-h, but decreases the electrical conversion efficiency to 34%, giving a heat rate of 10,000 Btu/kW-h, as shown in Table 4-12.

Both of the phosphoric acid fuel cell systems operate at a pressure of 120 psia to increase cell stack power density. These systems incorporate an integrated turbo-compressor to provide compressed air to the fuel processor and to the fuel cell cathode. The compressor is driven by a turbine through which the burner exhaust gas is expanded.

The fuel processor of the phosphoric and fuel cell cogeneration system shown in Figure 4-15 could be a gasification system that provides low or medium Btu fuel gas to the sulfur removal and shift converter units. Coal gasification systems for phosphoric acid fuel cells are discussed by Krasicki and Pierce (Ref. 68). The gasifier may be oxygen-blown or air-blown to provide medium or low Btu fuel gas, respectively. In either case, the fuel

Table 4-12. Fuel Cell Cogeneration System Characteristics

System Type				
Fuel	FC/FP	Capital Cost, $/kW	Heat Rate, Btu/kW-h	Output Ratio of Heat to Electricity, lb of steam/kW-h
Natural gas/ Naphtha	PA/Reformer	1460	8980	0.79
Petroleum Distillate	PA/Reformer	1390	10000	4.4
Solid[a]	PA/Gasifier (Air)	1530	14220	7.76
Solid[a]	PA/Gasifier (O_2)	1680	11380	6.03
Solid[a]	MC/Gasifier (Air)	1090	14200	7.48
Solid[a]	MC/Gasifier (O_2)	1230	11770	6.08

[a]Solid fuel may be biomass, coal, or refuse-derived.

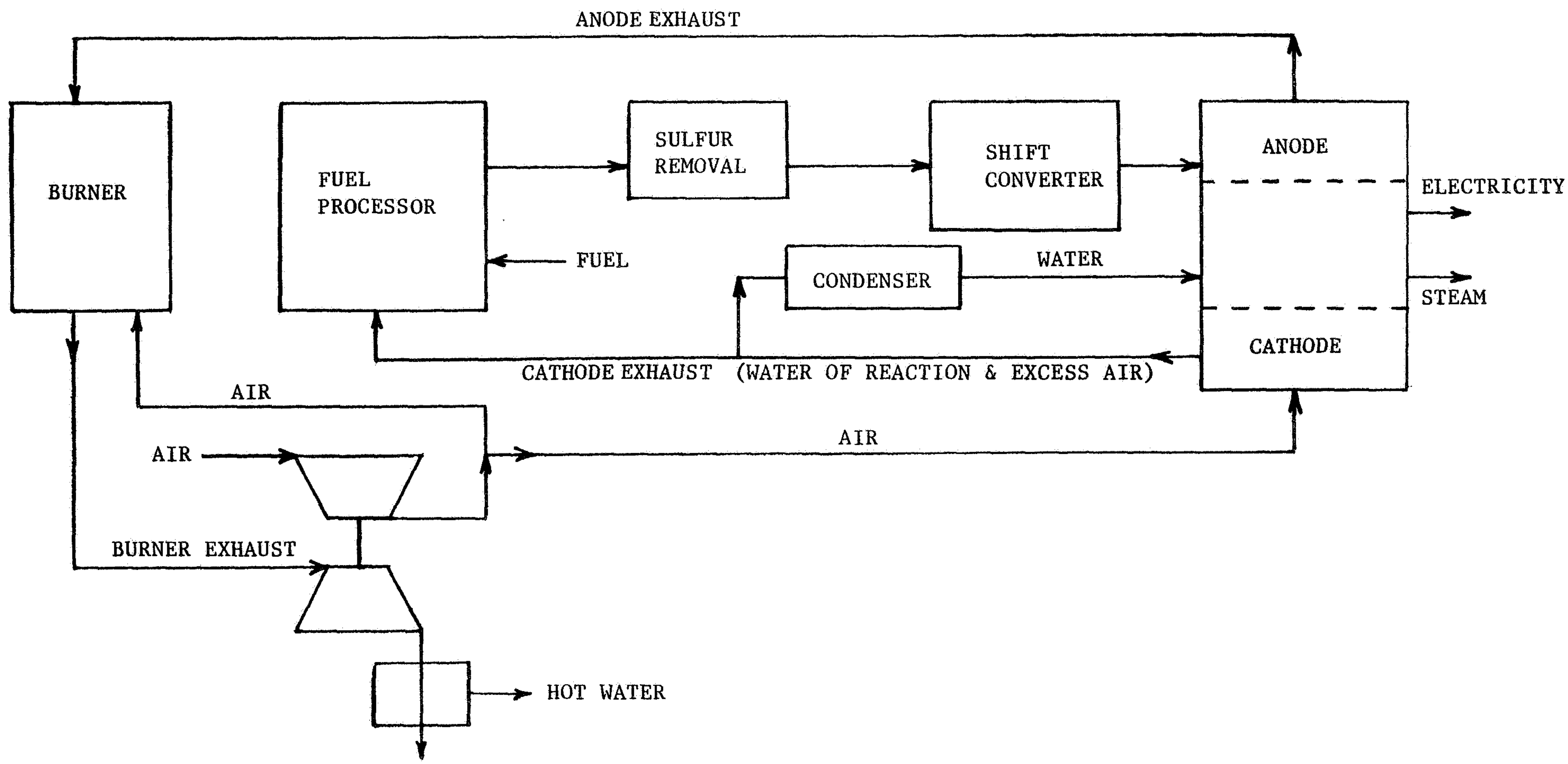


Figure 4-15. Phosphoric Acid Fuel Cell Cogeneration System Using Hydrocarbon Fuel in an Adiabatic Reforming Fuel Processor (Steam for Fuel Processor Supplied Directly by Cathode Exhaust)

gas must be processed through the shift converter to reduce the carbon monoxide concentration to about 2%. Typically, raw, low Btu fuel gas from an air-blown gasifier will consist of about (in molar concentration) 15% hydrogen, 30% carbon monoxide, 50% nitrogen, small amounts of methane and hydrogen sulfide, and traces of other contaminants. After shifting and cleaning, the fuel cell stack feed would consist of about 30% hydrogen with nitrogen and carbon dioxide constituting the rest of the stream. An oxygen-blown gasifier would have a product gas stream undiluted by nitrogen from air; after cleaning and shifting, the hydrogen molar concentration of the fuel cell feed gas would be about 50%, with carbon dioxide constituting most of the remaining gas stream. Either gas stream is suitable for feeding the anode of a phosphoric and fuel cell; however, slightly lower hydrogen utilization at the anode would result from using the more dilute fuel gas stream from the air-blown gasifier.

The overall energy conversion efficiency of the gasifier/PAFC system is less than that of the previously discussed fuel cell systems because of coal gasifier energy losses. Coal gasifiers typically produce output gas streams that have about 75% of the heating value of the coal feed. In addition, the heating value of the product gas must be available from hydrogen and carbon monoxide in order to be consumed by the fuel cell. Some gasifiers produce appreciable amounts of methane -- up to about 18% of the heating value of the coal feed according to Reference 69 -- which cannot be electrochemically utilized in the fuel cell stack. However, methane passes through the fuel cell and is combusted in the burner section of the plant; hence, a portion of the energy of the fuel fed to such a gasifier is recoverable as sensible heat from the burner.

An air-blown gasifier and an oxygen-blown gasifier were considered with the PAFC in this study, as illustrated in Figures 4-16 and 4-17. The air-blown gasifier has a high methane output, and the oxygen-blown gasifier is a high efficiency unit. The characteristics of the oxygen-blown gasifier shown in Figure 4-17 could also represent air-blown gasifiers with a low methane output, although the amount of sensible heat recovered from the gasifier unit would be somewhat reduced for air-blown units. In each of the systems shown, the phosphoric acid fuel cell system converts 41.5% of the hydrogen (H_2) and carbon monoxide (CO) fuel energy into net (ac) electricity output. In the fuel cell stack, 50% of the H_2 + CO fuel energy is dissipated as heat. The total heat output of each system consists of the heat recovered from the gasifier unit, the heat produced by the methane in the burner, and the heat produced in the fuel cell stack. The heat rate and thermal output for each of the gasifier/PAFC systems are shown in Table 4-12. The heat rates for both gasifier systems are higher than those of the PAFC/reformer systems due to the fuel gas energy being less than that of the solid fuel feed to the gasifier. However, the gasifier systems have a high thermal output which maintains the overall energy utilization at 80% to 84% of the gasifier feed HHV.

Molten carbonate fuel cell systems are shown in Figures 4-18, 4-19, and 4-20. Molten carbonate fuel cell systems differ from PAFC systems in that a shift converter is not required by the MCFC (although it is used in certain system concepts), and the carbon dioxide formed at the MCFC anode is externally recycled to the cathode. A basic MCFC system concept with fuel gas provided by an adiabatic reformer is shown in Figure 4-18. The fuel gas from the reformer is desulfurized and fed to the fuel cell anode. The anode discharge stream supplies water to the adiabatic reformer and a mixture of

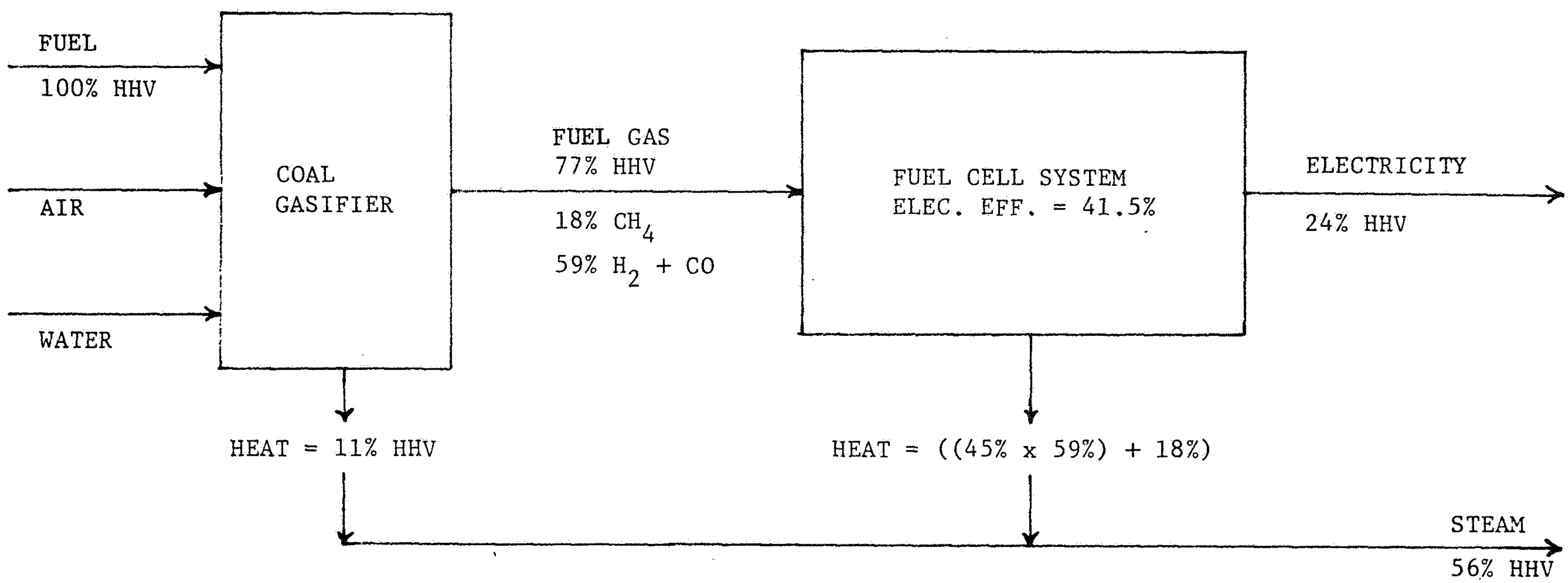


Figure 4-16. Gasifier/Phosphoric Acid Fuel Cell System with High Methane, Air-Blown Gasifier

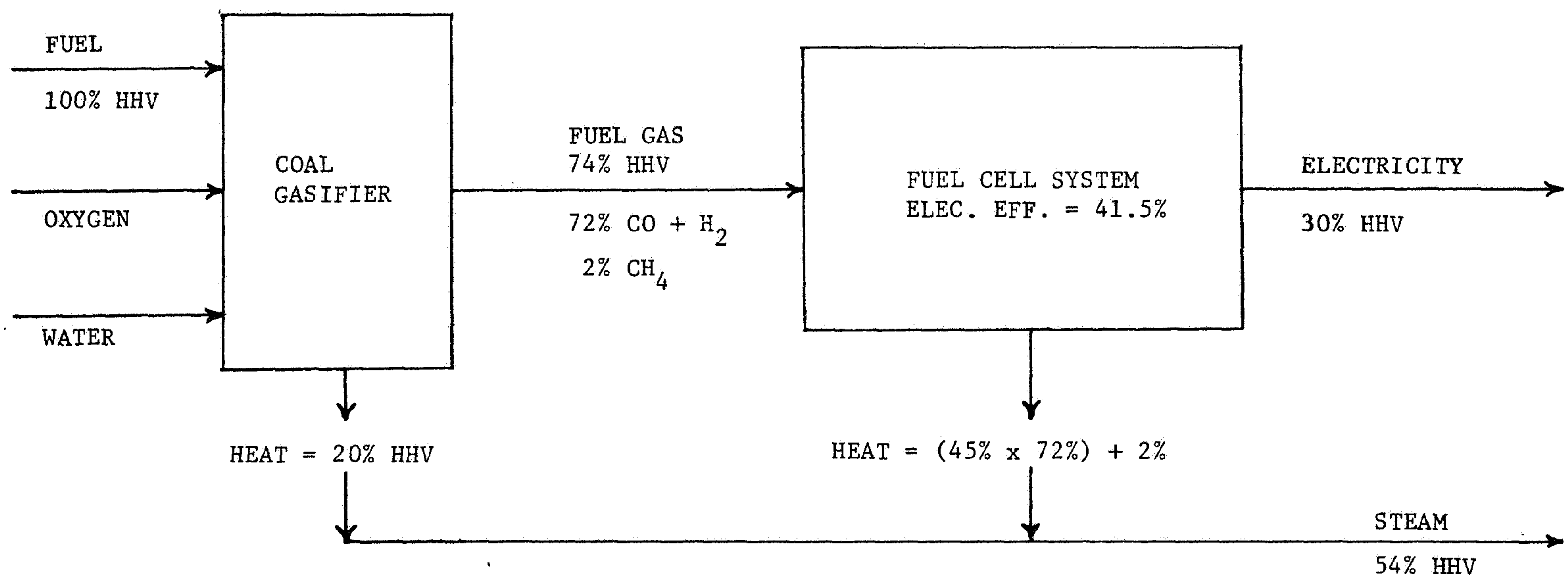


Figure 4-17. **Gasifier/Phosphoric Acid Fuel Cell System with High Efficiency, Oxygen-Blown Gasifier**

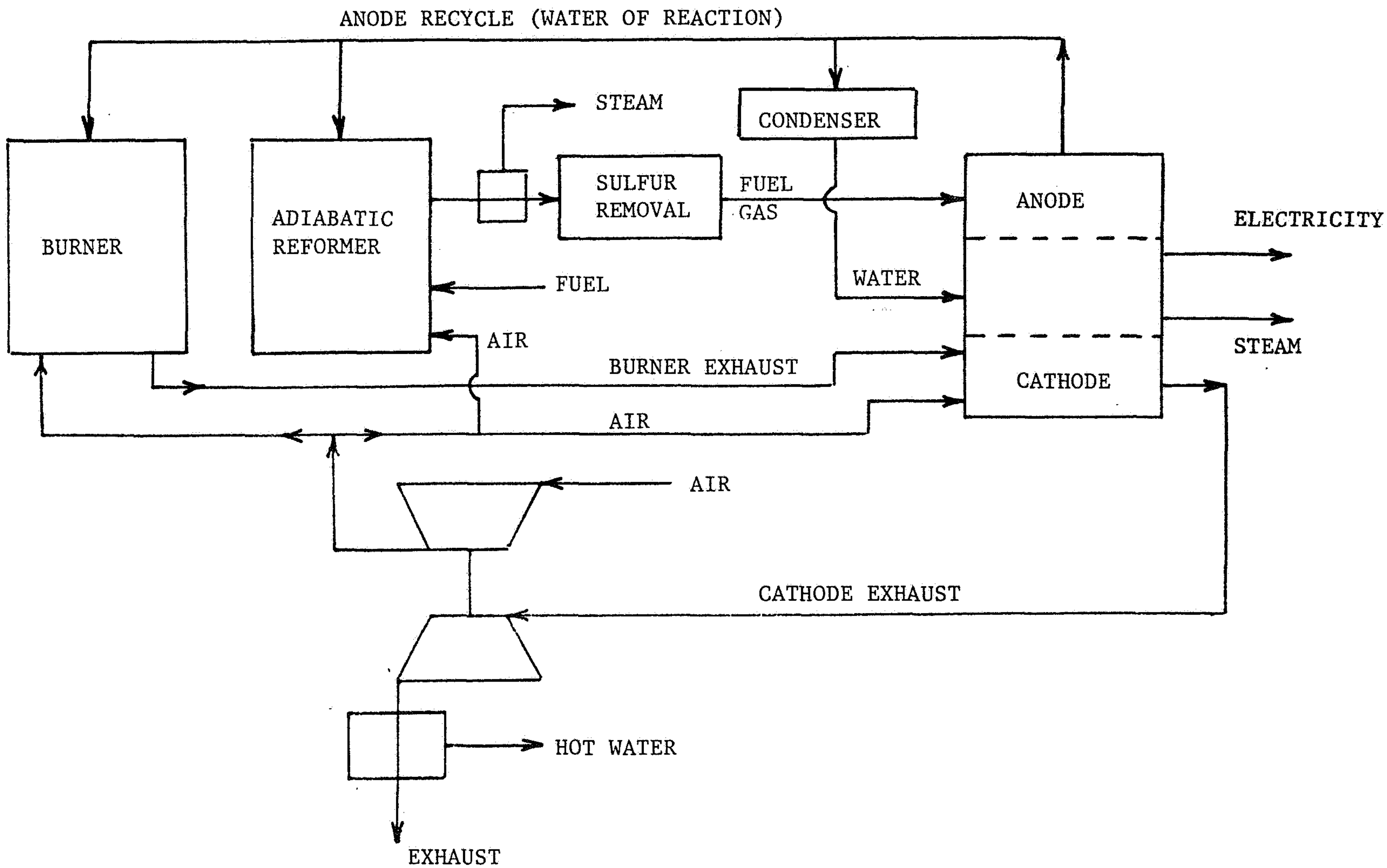


Figure 4-18. Molten Carbonate Fuel Cell Cogeneration System Using Liquid Hydrocarbon Fuel in an Adiabatic Reforming Fuel Processor (Steam for Adiabatic Reformer Supplied Directly by Anode Exhaust)

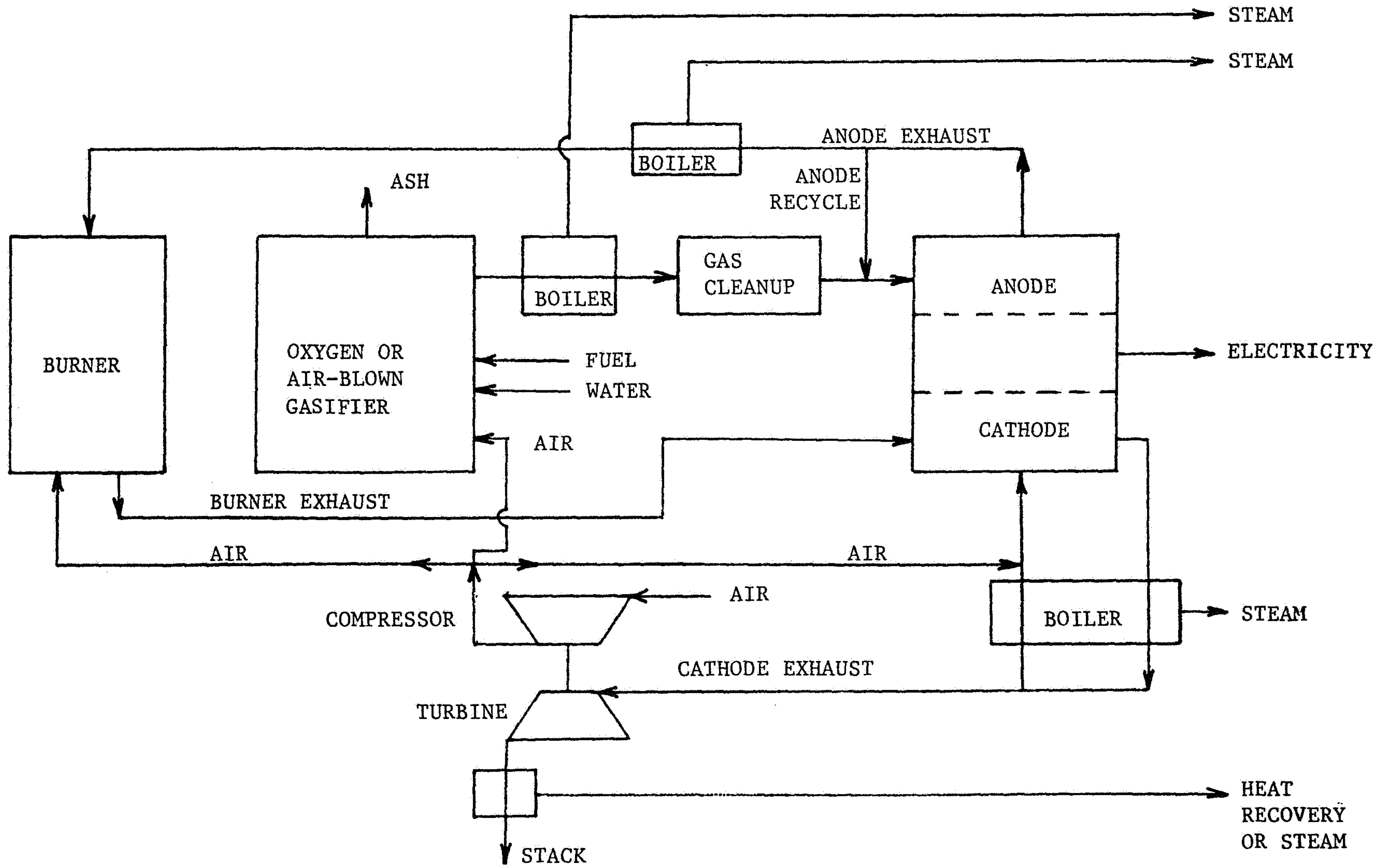


Figure 4-19. Advanced Gasifier /Molten Carbonate Fuel Cell Cogeneration System with Anode Gas Recycle

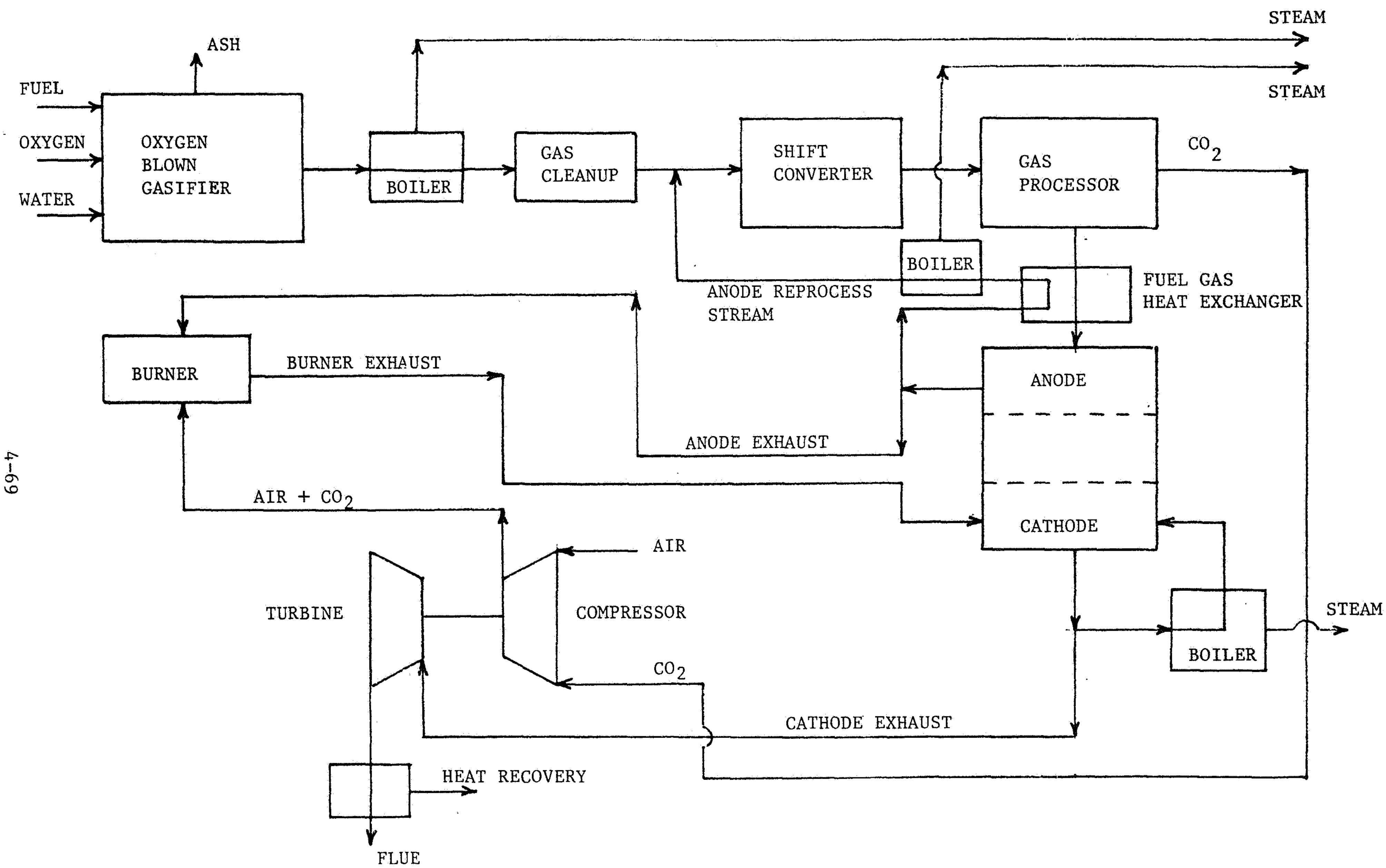


Figure 4-20. Advanced Gasifier Molten Carbonate Fuel Cell System with Anode Gas Reprocessing

residual fuel gas and carbon dioxide to the burner. The recycling of carbon dioxide is accomplished by routing the burner exhaust to the cathode along with air from the compressor. The cathode exhaust is expanded through the turbine to provide power to drive the air compressor. Advanced concepts for MCFC systems with coal gasifiers have been considered by Reinstrom (Ref. 70). MCFC systems with anode gas recycle and anode gas reprocessing are shown in Figures 4-19 and 4-20, respectively. The fuel cell stacks in these systems are gas cooled; hence, heat recovery boilers are used to generate steam with heat from the anode and cathode discharge streams. The MCFC/coal gasifier systems shown in Table 4-12 are based on the performance characteristics of the anode gas recycle concept as described by Reinstrom (Ref. 70). The gasifier characteristics, as previously discussed for the PAFC systems, were used in conjunction with the anode recycle MCFC performance to estimate the heat rate and thermal output for the MCFC/coal gasifier systems shown in Table 4-12. The MCFC system performance is quite similar to that of the PAFC. The major advantage of the MCFC is that the heat dissipated in the cell stack is available at about 1000°F, instead of 400°F as in the PAFC. The higher temperature is of little advantage in this study because steam is supplied to the industrial process as steam vapor at 267°F. In utility power plant applications, the higher temperature of the heat from the MCFC would allow increased bottoming cycle energy conversion efficiency.

The capital cost estimates for the PAFC modules and subsystems were based on the information given by Krasicki and Pierce (Ref. 68), with appropriate adjustments to account for differences in the configuration of the cogeneration systems considered herein. The fuel processor costs were derived from information given in References 71, 72, and 67. Based on a fuel cell system fuel gas input to ac electricity conversion efficiency of 40%, the costs of the different types of fuel processor units used in this study are: \$550/kW for the oxygen-blown coal gasifier; \$410/kW for the air-blown coal gasifier; \$260/kW for the adiabatic reformer; and \$325/kW for the naphtha/methane steam reformer. The capital costs thus developed for the PAFC cogeneration systems are shown in Table 4-12. The MCFC capital costs were similarly developed using the fuel cell module and subsystem costs given by Lance and VanBibber in Reference 73.

The costs of electricity for the fuel cell cogeneration systems are shown in Table 4-13. The PAFC/coal gasifier systems have costs of electricity in the higher range at 70 to 88 mills per kilowatt-hour with fuel at \$1.25 to \$2.50 per million Btu, respectively. The MCFC systems have lower costs of electricity due to the conjecture of lower capital costs for MCFC fuel cell modules. It may be possible to achieve cost reduction in the PAFC/ air-blown gasifier system through integration of the air-blown gasifier, the turbo-compressor, and the fuel cell module. A 20% capital cost reduction in the PAFC/coal gasifier systems would reduce their costs of electricity to levels competitive with the better gas turbine engine systems of Section 4.2.2.

Table 4-13. Cost of Electricity for Fuel Cell Cogeneration Systems (mills per kilowatt-hour)

System Type		Fuel Cost, $\$/10^6$ Btu			
Fuel	Fuel Cell/ Fuel Processor	1.25	2.50	5.00	7.50
Methane/Naphtha	PAFC/Reformer		82	100	120
Petroleum Distillate	PAFC/Reformer		70	81	91
Solid[a]	PAFC/Gasifier (A)[b]	70	75	85	
Solid[a]	PAFC/Gasifier (O)[c]	76	80	88	
Solid[a]	MCFC/Gasifier (A)	53	58	69	
Solid[a]	MCFC/Gasifier (O)	58	62	71	

[a]Solid fuel may be biomass, coal or refuse derived.
[b]Air-blown coal gasifier.
[c]Oxygen-blown coal gasifier.

4.2.6 Systems Using Alternative Energy Sources

Energy sources and systems that do not utilize such conventional fuels as natural gas, coal, and petroleum have received much attention over the past decade. The class of alternative energy sources is generally considered to include solar, biomass, geothermal, and wind. The generation of electric power has been a focal point of interest in alternative energy. Biomass is the easiest alternative energy source to utilize in an electrical power plant or cogeneration system because cellulosic and other plant material, i.e., wood crop, waste, etc., may be directly burned, gasified or converted into methanol in essentially the same manner as coal. In the present study, the direct combustion or gasification of biomass is considered to be an option in any of the systems considered that use solid fuel. Modification of direct combustion or gasification systems may be necessary to burn cellulosic material, but a detailed treatment of such modification is beyond the scope of this study. Wind power systems directly generate electricity and may produce economically competitive electricity in certain circumstances. Although wind cogeneration systems have been alluded to in the literature, this thermodynamically contorted notion could be taken seriously only if combustible fuel were scarce, extremely expensive or its use otherwise precluded, because thermodynamic work, i.e., electricity, would be directly dissipated as heat in such a system.

Geothermal energy occurs in the form of heat that is discharged as hot brine or steam from geothermal wells, and, in some instances, from natural springs or vents. If the geothermal heat is of adequate temperature, it can be used to generate clean steam for use in a power conversion cycle. Where such geothermal sources exist, their utilization usually is complicated by the potentially high maintenance of fluid handling equipment exposed to the corrosive geothermal brine. Several different systems for conversion of geothermal heat to electricity are currently under investigation. Lacy (Ref. 77) describes a binary conversion process in which a geothermal source provides heat for a closed Rankine cycle system using a hydrocarbon working fluid. Cerini and Record (Ref. 78) describe a rotary separation turbine for directly handling geothermal brines that are at a temperature high enough to become steam-liquid mixtures during expansion. McKay (Ref. 79) describes a novel geothermal expander for directly expanding geothermal brine as a two-phase mixture. Crane (Ref. 80) presents a discussion of three geothermal energy conversion projects in the Imperial Valley in California. In general, geothermal sources are not available at a high enough temperature to make cogeneration practicable. A relatively high temperature for a geothermal brine would be about 300 to 400°F, and the brine would be returned to a reinjection well at about 150°F. Conceivably, a cogeneration system could function over such a temperature difference, but an unacceptable sacrifice in electricity production would be incurred if industrial process steam were generated at 267°F. In unique circumstances, geothermal heat sources could be used in industrial cogeneration systems, but geothermal heat is not a generally viable energy source for industrial cogeneration as considered herein.

Of the alternative energy options, solar energy systems are the most amenable to industrial cogeneration. A comprehensive assessment of alternative solar thermal electric power plants is presented by Rosenberg and Revere in Reference 81, wherein solar electric power plants that employ solar energy collectors of the low concentrating, line-focusing and point-focusing variety are evaluated. Heat engines operating on the Rankine, Brayton, and Stirling thermodynamic cycles were considered as appropriate to the temperature at which heat is available from the different solar collector systems. The assessment of Reference 81 included an extensive cost study of the components of the solar electric systems at production levels of from 1000 to 100,000 units per year. A total of nine solar collector/heat engine systems were evaluated for applications in the 1 to 10 megawatt range of electricity output. This study concluded that the lowest cost electric power is produced by the point-focusing systems that do not include energy storage to increase their capacity factor. These lower cost systems produce electricity at 89 to 111 mills per kilowatt-hour in 1978 dollars, expressed as levelized bus bar energy cost (BBEC) as defined in Reference 82.

One of the lower cost point-focusing systems -- the Point-Focusing Distributed Receiver/Central Rankine (PFDR/R) -- may be readily adapted to industrial cogeneration at no increase in capital cost. In fact, according to the method used here, the system's capital cost was reduced when expressed in equivalent dollars. The PFDR/R system, as considered in Reference 81, is illustrated in Figures 4-21 and 4-22. The sensible thermal storage and condenser/cooling tower portions of the PFDR/R system shown in Figure 4-21 were deleted from the industrial cogeneration system considered herein. As shown in this figure, the parabolic dish solar energy collector field produces heat to generate steam that is expanded in a single turbine/generation unit. In the industrial cogeneration system, the turbine exhaust steam would then be directly provided to the industrial plant.

A cost estimate for the solar thermal industrial cogeneration plant was formulated by using cost information for the solar collector/receiver and steam generation portions of the PFDR/R power plant from Reference 81, and the costs of the steam turbine generator and associated equipment from Reference 2. The PFDR/R collector area was increased to supply 80,000 lb/h

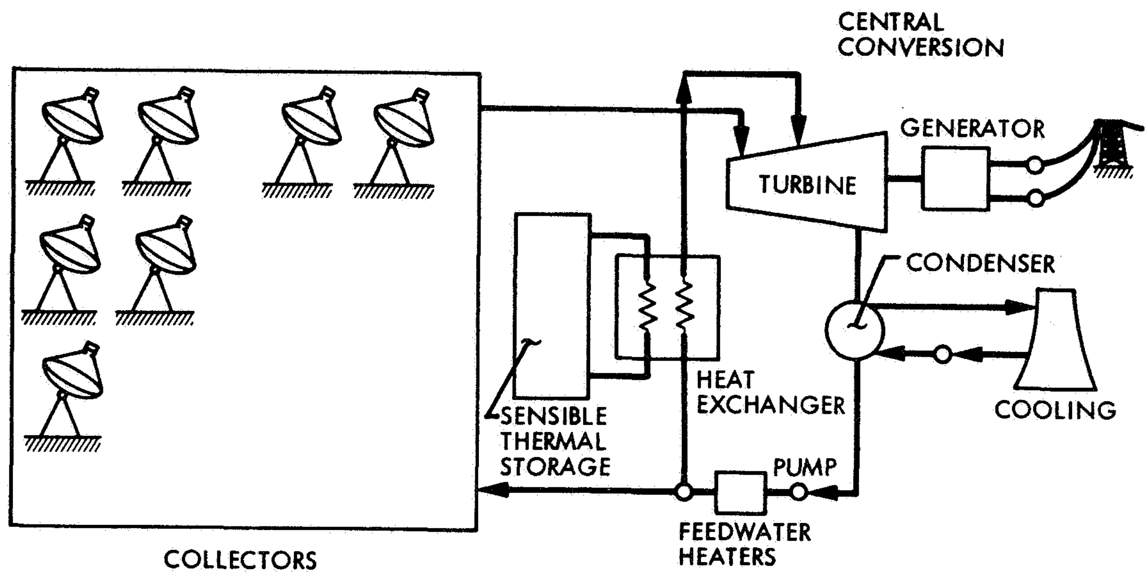


Figure 4-21. Schematic of a PFDR/R Power Plant

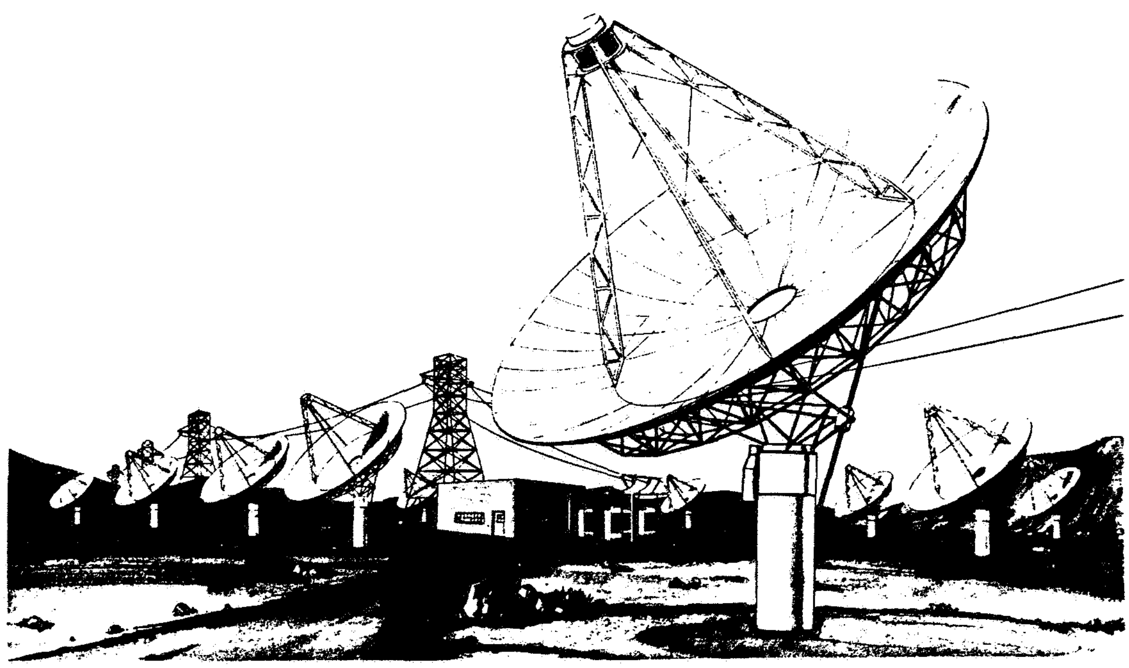


Figure 4-22. Illustration of a Point-Focusing Distributed Receiver Solar Energy System

of steam at peak output, which is the steam capacity of the other Rankine topping cycle systems discussed in Sections 3.1 and 4.2.1 of this report. The system parameters and costs of the solar PFDR industrial cogeneration system are shown in Table 4-14. The peak electrical output and the thermal to electrical output ratio of the solar cogeneration system are the same as for the Rankine topping cycle cogeneration systems previously discussed. The capital cost of the solar cogeneration system is shown as $\$9.8 \times 10^6$ in Table 4-14. The cost of the solar energy system, as taken from Reference 81, is based on a production rate of 25,000 units per year of technologically mature systems. Component cost information given in Reference 81 shows that solar energy system costs change significantly with production rate.

The cost of electricity shown in Table 4-14 for the solar cogeneration system depends on the cost of displaced fuel that would normally be fired in an industrial boiler to supply steam. The cost of electricity was calculated as discussed in Section 2.2 with a system heat rate of zero (because the solar cogeneration system requires no fuel) and a capacity factor of 32%. The maintenance cost was taken as $0.005/kW-h, as for the fuel-fired Rankine systems.

The cost of electricity for the solar cogeneration system decreases as more costly fuel is displaced. At fuel costs of $5.00 to $7.50 per million Btu, the cost of electricity for the solar cogeneration system at a capital cost of $2600/kW becomes competitive with that of the conventional Rankine and gas turbine cogeneration systems previously shown in Section 3.1 of this report. However, the solar cogeneration system capital costs are projections based on favorable assumptions for production rates and for the state of the technology after an extensive development effort. The costs of electricity for a two-fold increase in system capital cost are also shown in Table 4-14. These electricity costs illustrate the effect of a higher capital cost.

Table 4-14. System Characteristics and Costs for Solar PFDR Industrial Cogeneration System

System Parameters	
Collector Area, m^2	40,000
Required Collector Field Land Area, acres	30
Peak Electrical Output, kW	3760
Peak Thermal Output, lb/h	80,000
Capacity Factor (no storage), %	32
Output Ratio of Steam to Electricity, lb/kW-h	21.3

System Capital Costs, 1982 \$ x 10^6	
Solar Collector/Receiver/Transport	6.85
Balance of Solar Plant (Land, Site Prep, Controls)	1.75
Turbogenerator/Electrical System	1.20
Total Capital Cost	9.8

Cost of Electricity

Cost of Displaced Fuel, \$ per 10^6Btu	mills per kW-h	
	\$2600/kW	\$5200/kW
1.25	200	430
2.50	170	400
5.00	95	330
7.50	24	260

SECTION 5

EVALUATION OF ADVANCED COGENERATION SYSTEMS

5.1 STATE OF TECHNOLOGY

The systems shown in Table 5-1 represent vastly differing states of technological development; for instance, the Atmospheric Fluidized Bed/Rankine (AFB/R) cogeneration system is nearing commercial status while the molten carbonate fuel cell system is an advanced concept based on laboratory work with molten salt fuel cell stacks. Hence, the uncertainty in the estimates of cost and performance varies greatly among the systems, as does the risk of failure in achieving expected system cost and performance characteristics. The technological state of development of the advanced cogeneration systems may be categorized as near-term, mid-term or far-term, as designated in Table 5-1. The near-term category includes those systems in advanced stages of commercial demonstration that are nearing commercial readiness; the AFB/R system shown in the Table is a near-term system. The mid-term category encompasses those systems whose major components have been extensively tested at the pilot plant stage in configurations similar to those required by operational cogeneration systems. The mid-term category also includes systems whose components would represent an improvement in existing systems which extends beyond the normal practice of modifying present designs. The far-term designation applies to systems in an early stage of development where basic performance and operating parameters, component configurations, and materials of fabrication are being defined. The far-term systems presuppose that ongoing research and development work will result in energy conversion systems with the projected performance and cost characteristics.

The mid-term systems shown in Table 5-1 include the pressurized fluidized bed/gas turbine (PFB/GT) system, the air-blown gasifier/gas turbine (ABG/GT)

Table 5-1. Cogeneration System Characteristics

Cogeneration System Type	State of Development	Fuel Flexibility	Cost of Electricity, mills/kW-h: Fuel Cost, $/10^6 Btu, $1.25	$5.00	Long-Term Potential
Rankine Topping					
Oil/Gas Fired Boiler	Present	Dist.[a], NG[b]	NA[c]	58	Low
Atm. Fluidized Bed	Near-term	Solid	76	94	Low
Rankine Bottoming	Present	Stack Gas Waste Heat	100	180	Low
Gas Turbine Engine Systems					
Conventional	Present	Dist., NG	NA	51	Low
Press. Fluidized Bed	Mid-term	Multi-fuel	52	72	Limited
Press. Air-Blown Gasifier	Mid-term	Multi-fuel	55	82	High
Atm. Fluidized Bed	Mid-term	Multi-fuel	78	109	Low
Reciprocating Engine Systems					
Conventional	Present	Dist., NG	NA	102	Low
Conventional Engine, Air-Blown Gasifier	Mid-term	Multi-fuel	87	126	Low
Ceramic Engine, Direct Coal Injection	Far-term	Coal	65	85	Limited
Stirling Engine Topping					
Conventional	Mid-term	Dist., NG	NA	89	Low
Atm. Fluidized Bed	Mid-term	Solid	79	98	Limited
Solar Point-Focusing Rankine	Mid-term	None	430	330	
	Far-term		200	95	Limited
Fuel Cell Systems					
Phosphoric Acid Cell, Press. Air-Blown Gasifier	Mid-term	Multi-fuel	70	85	Limited
Molten Carbonate Cell, Press. Air-Blown Gasifier	Far-term	Multi-fuel	53	69	High

[a]Petroleum Distillate. [b]NG = Natural Gas. [c]NA = Not Applicable.

system and the ABG/phosphoric acid fuel cell system. PFB/GT systems have received extensive development in the past ten years, but additional work remains, particularly in satisfactorily integrating PFB units and gas turbine engines to achieve acceptable turbine service life. Atmospheric pressure coal gasifiers have been commercially available for years, as have larger high pressure units. However, an air-blown gasifier operating at 8 to 10 atmospheres pressure that is suitable for integration with a gas turbine engine or fuel cell for the industrial cogeneration application in the 10-MW power range is not available. Several candidate system designs are available for consideration, but further development work is required. The differing requirements for coal gas cleanup between gas turbine engines and fuel cells may have a significant impact on gasifier system configuration and cost.

The far-term systems shown in Table 5-1 are the direct coal injection/ ceramic reciprocating engine and the ABG/molten carbonate fuel cell. Both the ceramic reciprocating engine and the molten carbonate fuel cell have been operated in experimental configurations. These two energy conversion systems represent the introduction of new, high-temperature materials technology into energy conversion systems. The performance and cost characteristics of these systems as projected in this study, although conjectural, are taken to represent the impact of advances in materials technology.

The expenditure of research and development funds with the expectation that a system of presupposed performance and cost characteristics will result, certainly carries an element of risk. The risk of failure or underachievement may be reasonably presumed to increase as the expected characteristics of the advanced system become more dependent on unproven technology. Of the cogeneration systems shown in Table 5-1, the near-term AFB/R system is considerably less attractive than the mid-term, higher-risk PFB/GT and ABG/GT systems. And of particular interest is the lack of benefit for the cogeneration application considered herein to be gained from the far-term, highest-risk systems. Table 5-1 shows the cost of electricity for the best far-term system (the ABG/MCFC) to be matched by two of the mid-term systems (the PFB/GT and the ABG/GT). The cost projections and performance estimates for the far-term systems are considerably more uncertain than for the mid-term systems. If significant cost reductions and/or performance improvements for the far-term systems were attained, they would be considerably more attractive.

5.2 EMISSIONS AND FUEL FLEXIBILITY

The advanced cogeneration systems of Table 5-1 make use of four different generic techniques of fuel utilization: fluidized bed combustion, air-blown gasification, coal extrusion, and conventional diffusion flame combustion. A Rankine cycle system, two of the gas turbine engine systems, and a Stirling engine system incorporate fluidized bed combustion to simultaneously achieve emissions control and solid fuel burning capability. The fuel cell systems, a gas turbine engine system, and a reciprocating engine system feature air-blown gasifiers that produce fuel gas from a raw feed stock of carbonaceous material such as biomass, coal or refuse. The fuel gas would be cleaned of contaminants prior to combustion in the heat engines or electrochemical reaction in the fuel cell. The uncooled, ceramic reciprocating engine is fueled by coal that is directly injected into the engine's cylinders by means of an extrusion process that plasticizes the coal. The conventional systems of each type burn natural gas and liquid fuels. Emissions control for all the systems could be augmented through exhaust gas treatment, including sulfur removal and catalytic reduction of nitrogen oxides if necessary.

The advanced systems do differ significantly in techniques of emissions control. The AFB systems would probably require flue gas cleanup systems for both sulfur and nitrogen oxides in order to meet the more stringent air pollution rules. In contrast, the PFB system has demonstrated that virtually all of the fuel sulfur can be captured in the bed sorbent material, and PFB units have also shown extremely low levels of nitrogen oxides emissions (appreciably less than 0.1 lb/10^6Btus in some tests). Hence, the PFB system would likely require supplementary control only to meet the most stringent standards. The coal gasifier systems are dependent on cleanup of contaminants in the fuel gas stream produced by the gasifier in order to achieve emissions control. Commercially available desulfurization units are employed to remove sulfur compounds in the fuel gas stream in both the gas turbine and fuel cell systems. Additional gas cleaning measures for the fuel cell system would depend on the allowable amounts of feed gas contaminants. Nitrogen oxides are not formed in any appreciable amounts in the fuel cell system's electrochemical energy conversion process, and only a small amount of fuel is consumed in the system's burner at a low combustion temperature. Consequently, nitrogen oxides emissions are not a problem with the fuel cell systems. The combustion process in the gasifier/gas turbine system would require close control and sophisticated combustor design in order to avoid appreciable formation of nitrogen oxides while maintaining satisfactory operating characteristics. Such performance remains to be demonstrated at very stringent nitrogen oxides standards in systems suitable for industrial cogeneration applications. The PFB/GT and ABG/GT systems are, at this juncture, undifferentiated with respect to the potential to meet stringent nitrogen oxides emissions limits. Post combustion treatment of exhaust gas might be required for the more stringent nitrogen oxides standards if adequate control during the combustion process is not achieved.

The advanced cogeneration systems using fluidized bed or gasifier units have the potential to accommodate a variety of fuels. Additional burner units for liquid and gaseous fuel could be fitted to the fluidized bed systems, and the gasifier systems may, with development, accommodate liquid or gaseous fuel. Advanced reciprocating engines could be fitted with liquid fuel injection systems and gaseous fuel induction systems similar to those of presently available dual-fuel engines.

5.3 COST OF ELECTRICITY

The cost of electricity produced by cogeneration systems is primarily determined by the system's capital cost, overall efficiency of energy usage, and cost of fuel consumed by the cogeneration system. As the costs of natural gas and distillate fuel escalate, the cost of electricity produced by the presently available cogeneration systems increases substantially. At these higher costs of electricity, an additional capital investment to enable the use of less expensive fuel such as biomass, coal or refuse would be justified. The presently available technology for utilization of solid fuel is restricted to the Rankine topping cycle cogeneration systems. These solid fuel or coal burning Rankine cycle cogeneration systems have a comparatively high capital cost; as a result, the cost of electricity produced by such systems is competitive with electricity purchased from the utility company or produced by natural gas and oil-fueled cogeneration systems only in limited circumstances. For example, a very large industrial plant may achieve economies of scale that make coal-fired cogeneration attractive, or the availability of refuse or a waste product to be used as fuel may justify an investment in a Rankine cycle cogeneration system.

Certain advanced systems for industrial cogeneration considered in this study offer the potential for use of less expensive solid fuels such as biomass, coal or refuse-derived fuel in systems whose overall efficiency of energy utilization and capital cost are such that attractive costs of electricity are obtained in moderately sized systems suitable for use in typical industrial plants. An additional capital investment in such systems could likely be justified, particularly in an era of expensive natural gas and distillate fuel. The costs of electricity for advanced cogeneration systems are shown in Table 5-1 for fuel costs of $1.25 and $5.00 per million Btu. The pressurized fluidized bed/gas turbine engine system has the lowest cost of electricity at 52 mills/kW-h at a fuel cost of $\$1.25/10^6$Btu. The systems of Table 5.1 that have electricity costs of from 50 to 60 mills/kW-h when burning solid fuel or coal at $\$1.25/10^6$Btu include the ABG/GT, the PFB/GT and the ABG/MCFC. These systems would be economically competitive with presently available cogeneration systems fueled by natural gas or distillate at current prices. Since biomass, coal, refuse, and other solid fuels are not expected to undergo long-term price escalation to the extent of natural gas and petroleum derived fuels, the real cost of electricity produced by such advanced cogeneration systems would be expected to remain relatively stable.

5.4 LONG-TERM POTENTIAL FOR PERFORMANCE IMPROVEMENT AND COST REDUCTION

The long-term potential for improvement in performance and/or cost of the cogeneration systems is categorized in Table 5-1 as low, limited or high. This categorization is based on an assessment of the present state of the technologies on which the cogeneration systems are based and of intrinsic characteristics that may limit a particular technology.

The Rankine cycle energy conversion systems are fundamentally quite mature and are likely to undergo only marginal evolutionary improvements in performance and cost. The AFB boilers will certainly enjoy the benefit of further improvements in durability and will likely decrease in capital cost. However, their position relative to the other advanced cogeneration systems is not likely to change.

The gas turbine engine will benefit from the continued development of gas turbine technology which is being intensively pursued in both privately and governmentally sponsored activities throughout the world. Gas turbine engines will are likely to continue to experience improvements in performance and specific power brought about by higher temperature materials and improved designs. The air-blown gasifier/gas turbine engine cogeneration system (ABG/GT) can experience the full advantage of such improvements, providing that nitrogen oxides control technology keeps pace with the increased temperature capability. In contrast to the ABG/GT, the fluidized bed/gas turbine systems are limited in operating temperature by the techniques of sulfur capture and retention in the fluidized bed. Therefore, unless some basic change in these techniques permits an increase in operating temperature, the fluidized bed/gas turbine systems are limited to the 1500°F - 1700°F temperature range. The pressurized fluidized bed/gas turbine system considered in this study was limited to a turbine inlet temperature of 1650°F with a concomitant reduction in engine efficiency and power output. The atmospheric fluidized bed/gas turbine system is limited to a lower temperature of 1500°F because the heat exchanger tubes operate at higher stress levels. The AFB/GT suffers the further disadvantage of having a substantially larger and more costly fluidized bed due to the larger volume required for combustion at atmospheric pressure. The reduction of gas turbine engine efficiency does not adversely affect the overall efficiency of energy utilization because the otherwise lost energy is recovered as industrial process heat. The power output penalty due to the reduction of turbine inlet temperature does increase system cost because the gas turbine engine must be larger for a given power output. The cost of the AFB/GT system is further increased due to the cost of the atmospheric fluidized bed and the high-temperature heat exchanger.

The uncooled ceramic reciprocating engine is a major advancement in heat engine technology. The cogeneration system based on the uncooled engine overcomes a major disadvantage of the conventional reciprocating engine in that the quantity of heat available to generate industrial process steam is increased by about two-fold. However, the overall efficiency of energy utilization does not match that of the gas turbine and fuel cell systems. Unless the capital cost of the reciprocating engine is reduced by the advent of ceramic materials, the gas turbine and fuel cell based systems will continue to be preferred for industrial cogeneration.

The Stirling engine cogeneration system has limited long-term potential because, relative to the other systems, its capital cost is high and no credible approach to adequate cost reduction is extant. The system cost problem is more severe in the atmospheric fluidized bed Stirling system due to the additional cost of the AFB unit.

The solar point-focusing Rankine cogeneration system requires a major capital cost reduction before it is competitive with the better systems. Table 5-1 shows the solar cogeneration system in a mid-term and a far-term version, the capital cost of the far-term version being one-half that of the mid-term version. With such a capital cost reduction, the far-term system has a cost of electricity among the higher cost options with fuel at $5.00/

10^6Btu. When displacing more expensive fuel, the far-term solar cogeneration system becomes economically attractive.

Fuel cells are in an early stage of technology development and could show substantial improvements in system durability, reliability, lifetime, and capital cost. The performance of the phosphoric acid fuel cell is established, and major breakthroughs in electrochemical energy conversion technology seem unlikely. But improvements in fuel cell systems, primarily reductions in capital cost coupled with extended service life, could improve their relative position in the ranking of advanced cogeneration systems. The far-term fuel cell option -- the ABG/MCFC -- has the potential to be the principal component in an extremely attractive industrial cogeneration system.

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