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Thermal Energy Storage for Integrated Gasification Combined-Cycle Power Plants

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July 1990

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SUMMARY

Studies strongly indicate that the United States will face widespread electrical power constraints in the 1990s, and most regions of the country will experience capacity shortages by the year 2000. In many cases, the demand for increased power will occur during intermediate and peak demand periods. While natural gas is currently plentiful and reasonably priced, the availability of an economical, long-term, coal-fired option for peak and intermediate load power generation will give electric power utilities an option in case either the availability of natural gas should deteriorate or its cost increases.

This study describes a technical and economic evaluation of integrating molten nitrate salt thermal energy storage (TES) into a integrated gasification combined-cycle (IGCC) power plant. Currently, a conventional IGCC power plant can only be considered for a base load power generation applications because of difficulties with cycling the gasifier. An IGCC plant with TES would continuously operate the gasifier and gas turbine, storing the thermal energy in the fuel gas stream and gas turbine exhaust. During peak demand periods, the stored thermal energy would be used as a heat source for a steam power cycle, producing electric power.

The following are the results of the economic evaluation of using molten salt TES in an IGCC plant:

- An IGCC plant with molten salt TES could substantially reduce the cost of coal-fired peak and intermediate load power generation. The results of this study show that an IGCC plant with molten salt TES could reduce the cost of peak and intermediate power generation by as much as 20% when compared with other coal-fired alternatives.

- Molten salt TES is technically feasible. The overall judgment, both of this study and of similar evaluations is that molten salt TES is technically feasible, and it is reasonable to assume that the technology could be commercialized.

- Advanced molten salt TES concepts could substantially improve the performance and economics of an IGCC plant with TES. Several advanced concepts, such as direct-contact salt heating, low freezing point salts, dual-storage media, and advanced tank designs, have the potential to substantially improve the performance of an IGCC plant with TES.

Conceptual designs were developed for a number of operating schedules. The conceptual designs had sufficient detail to allow development of capital and levelized energy cost estimates. The resulting costs were then compared with a base case consisting of an IGCC plant producing base load power and a cycling coal plant producing peak or intermediate load power. The technical feasibility of molten salt TES and direct-contact salt heating was investigated by conducting a literature review and contacting researchers working with the technology.

Costs were estimated for six power plant operating schedules. Levelized energy costs were prepared for each operating schedule for both the conventional coal-fired option and the IGCC plant with TES. Results showed that the IGCC plant with TES had a lower levelized energy cost than the corresponding conventional coal-fired option for all plant operating schedules. When high natural gas price escalation rates were assumed, the IGCC plant with TES had lower levelized energy costs than the corresponding natural-gas-fired options for all but one operating schedule. This concept is most attractive at lower plant capacity factors (fewer operating hours) where the coal-firing equipment is downsized and, hence, the capital cost benefits of incorporating thermal energy storage are greatest.

While not currently used in power production, molten salt TES has been extensively investigated as part of the U.S. Department of Energy's (DOE) Solar Thermal Program. The concept has been the subject of bench-scale experimental investigations, several detailed design studies, and small-scale field demonstrations. While problems remain, the balance of opinion is that commercialization of molten salt TES is technically feasible. Recommendations are given for the areas most in need of technology development.

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THERMAL ENERGY STORAGE FOR INTEGRATED GASIFICATION
COMBINED-CYCLE POWER PLANTS

1.0 INTRODUCTION

There are increasingly strong indications that the United States will face widespread electrical power generating capacity constraints in the 1990s; most regions of the country could experience capacity shortages by the year 2000. The demand for new generating capacity occurs at a time when there is increasing emphasis on environmental concerns. These two trends have led to the development of a number of coal-fired power plant technologies that can provide economical base load power while meeting stringent environmental regulations. However, the increased demand for electric power often occurs during periods of peak and intermediate load power demand (U.S. Energy Association 1988). Advanced, clean coal technologies will be limited to base load applications that comprise only 60% of the market for new generating capacity unless these technologies are able to economically provide peak and intermediate loads (Yu 1988).

The integrated gasification combined-cycle (IGCC) power plant is an example of an advanced coal-fired technology that will soon be commercially available. The IGCC concept has proved to be efficient and cost-effective while meeting all current environmental regulations on emissions; however, the operating characteristics of the IGCC system have limited it to base load applications. The integration of thermal energy storage (TES) into an IGCC plant would allow it to meet cyclic loads while avoiding undesirable operating characteristics such as poor turn-down capability, impaired part-load performance, and long startup times. In an IGCC plant with TES, a continuously operated gasifier supplies medium-Btu fuel gas to a continuously operated gas turbine. The thermal energy from the fuel gas coolers and the gas turbine exhaust is stored as sensible heat in molten nitrate salt; heat is extracted during peak demand periods to produce electric power in a Rankine steam power cycle.

The study documented in this report was conducted by Pacific Northwest Laboratory[a] (PNL) and consists of a review of the technical and economic feasibility of using TES in an IGCC power plant to produce intermediate and peak load power. The study was done for the U.S. Department of Energy's (DOE) Office of Energy Storage and Distribution.

1.1 STUDY OBJECTIVES AND SCOPE

This study had the primary goal of assessing the technical and economic feasibility of using molten salt TES in an IGCC plant. The specific objectives were to

1. develop a conceptual design of an IGCC plant using molten salt storage
2. develop a capital cost and levelized energy cost (LEC) estimate for the conceptual design.

1.2 ORGANIZATION OF REPORT

The background of the study is presented in Section 2, and the method of analysis is discussed in Section 3. Section 4 describes the development of the conceptual design (Objective 1). Section 5 presents the capital and levelized cost estimates for the conceptual design (Objective 2). Conclusions are presented in Section 6.

(a) PNL is operated for the U.S. Department of Energy by Battelle Memorial Institute.

2.0 BACKGROUND

This study primarily focuses on evaluating applications of molten salt TES in an IGCC power plant to provide intermediate demand electric power generating capacity. Section 2.1 presents a description of the concept while Section 2.2 discusses the results and relevance of previous investigations of TES for power generation.

2.1 CONCEPT DESCRIPTION

Given the uncertainties in electric power demand, the cost and availability of fuel, and the effects of environmental legislation, utilities are seeking to construct small, low-cost, and fuel-flexible power plants that can economically meet current and future environmental requirements. The IGCC power plant is an attractive option for the expanding current and near-term generating capacity needs of the utility industry. These plants can be built as modular units with phased construction; they use coal (which is abundant and has been historically stable in price); and they can significantly reduce air pollutant emissions when compared with conventional coal-fired power plants.

While the IGCC concept has many attractive features, it has only been considered for base load applications because the concept has several characteristics that make it unattractive for peak and intermediate load applications. These characteristics include 1) poor gasifier turn-down capability, 2) poor part-load performance, and 3) long gasifier startup times. The incorporation of a molten salt TES system into an IGCC plant would allow flexible power production in an intermediate or peak mode.

During the course of this study, several schemes for integrating TES into an IGCC plant were identified. After a preliminary evaluation, the scheme shown in Figure 2.1 was selected for detailed evaluation. With this scheme, a continuously operating Texaco gasifier supplies intermediate-Btu fuel gas to a gas turbine, which is also operated continuously, to generate base load electric power. Thermal energy in the gas turbine exhaust and hot fuel gas is used to charge a TES system.

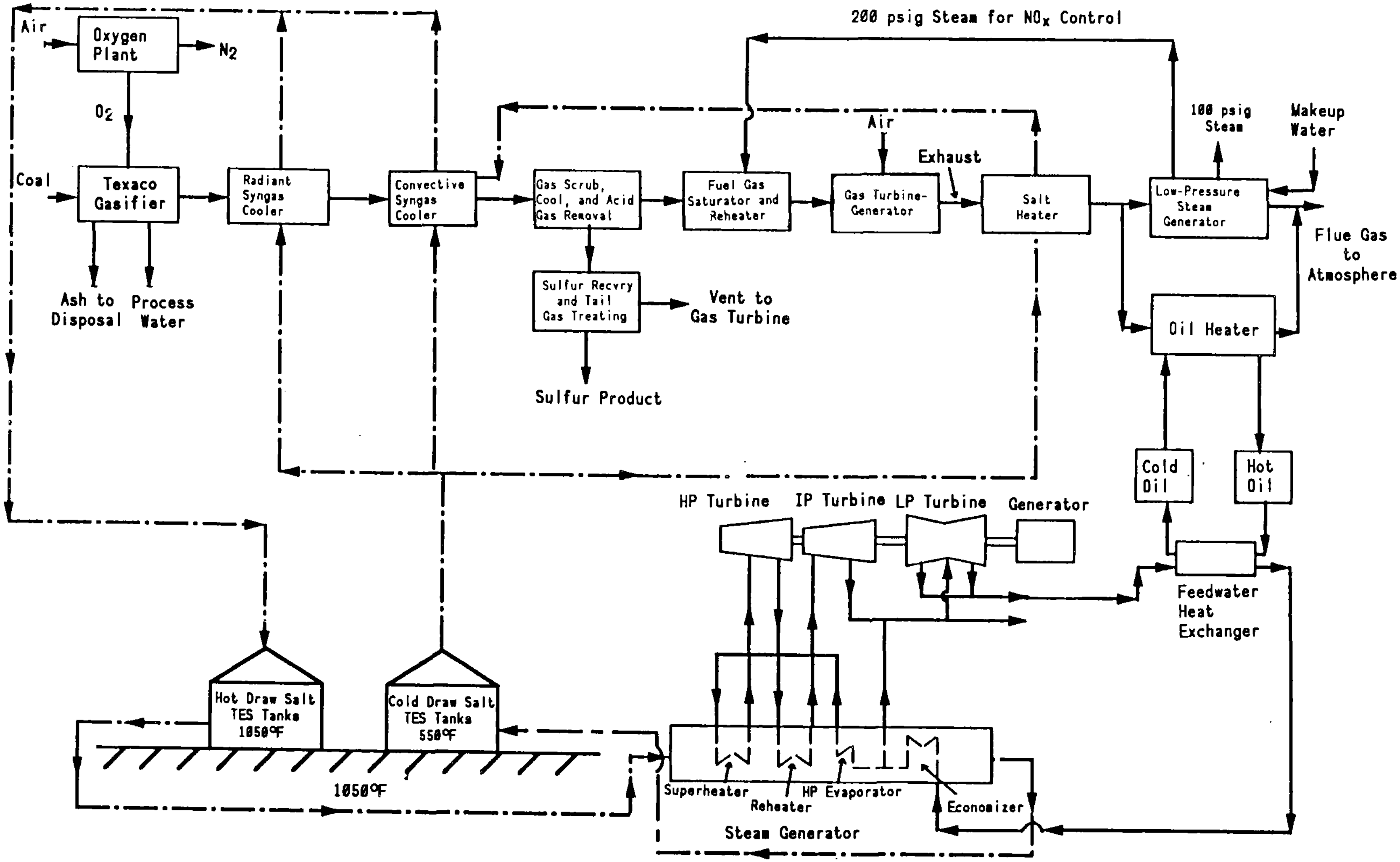

FIGURE 2.1. IGCC Power Plant with Molten Salt TES for Peaking

The TES system consists of a high-temperature and a low-temperature TES system. The high-temperature storage medium is a mixture of sodium nitrate (60 wt%) and potassium nitrate (40 wt%). Thermal energy is stored as sensible heat in this salt mixture. Cool salt stored in the cold salt tank at 280°C (536°F) is continuously pumped through the high-temperature section of the gas turbine waste heat recovery heater and the radiant and convective fuel gas coolers where it is heated to 566°C (1050°F). The salt is then returned to the hot salt storage tank. The salt mixture freezes at a temperature near 240°C (464°F); consequently, precautions must be taken to ensure that the temperature of the molten salt never drops below the freezing point. The maximum temperature of the salt is 566°C (1050°F) and is determined by the chemical stability of the mixture. During peak demand periods, the hot molten salt is used as a heat source to produce 538°C (1000°F) and 16.54-MPa (2400-psi) superheated steam in a steam generator. The steam is then used to produce electricity in a conventional steam Rankine power cycle.

The low-temperature storage medium is a mixture of a heat transfer oil such as Caloria HT-43[a] (25 vol%) and rock (75 vol%) contained in a storage tank. Thermal energy is stored as sensible heat primarily in the rock, and the oil acts as a heat transfer fluid. The storage tank is arranged so that hot oil is always added or removed from the top of the tank while cold oil is added or removed from the bottom of the tank. The lower-density hot oil remains in the top of the tank while the higher-density cool oil occupies the bottom of the tank, forming one hot and one cold region separated by a thermocline. This design eliminates the need for separate hot and cold tanks as are used with the molten salt TES. Low-temperature oil from the bottom of the tank at 121°C (250°F) is pumped to the low-temperature section of the gas turbine waste heat recovery heater where it is heated to 288°C (550°F). The oil is then returned to the top of the oil/rock storage tank. The maximum oil temperature is ~288°C (550°F) and is determined by the chemical stability of the oil. The thermal energy stored in the low-temperature TES is used to preheat the feedwater after it leaves the condenser and to produce process steam for other applications in the IGCC plant.

(a) Trademark of the Exxon Corporation, Houston, Texas.

Although not used to produce power commercially, molten salt TES was extensively investigated as part of DOE's Solar Thermal Program. The concept was the subject of bench-scale experimental investigations, several detailed design studies, and small-scale field demonstrations. While significant problems remain, the general opinion of experts is that commercialization of molten salt TES is technically feasible.

As with the molten salt TES, oil and rock TES was extensively investigated as part of the DOE Solar Thermal Program. One large-scale demonstration has been successfully completed at the Barstow solar thermal power plant (Hallet and Gervais 1977). Oil and rock TES has been proven to be technically feasible and in this study it was assumed to be commercially available.

2.2 RESULTS OF PREVIOUS EVALUATIONS

While TES has never been considered for applications with an IGCC plant, there have been a number of studies on using TES with conventional nuclear and coal-fired power plants. A series of studies on TES for near-term utility applications was conducted by DOE, the Electric Power Research Institute (EPRI), and the National Aeronautics and Space Administration (NASA) in the late 1970s (General Electric 1979a,b,c). The results of these studies did not favor TES, but the ground rules for the studies limited the concepts to schemes that had a minor impact on the overall design of the plant. These limitations resulted in poor applications for TES but they often are more expensive.

In the last 10 years, however, significant changes have altered this situation. First, improved integration schemes based on less-restrictive assumptions have proven to be much more attractive. Second, improved TES designs have resulted in improved costs and performance. Finally, the emergence of advanced coal combustion schemes, such as IGCC, has opened new opportunities for TES applications in power generation.

Based on these new developments, a review of TES applications in conventional, pulverized-coal-fired power plants was conducted under DOE's TES Program (Drost et al. 1989). The results of the study showed that the use of

TES in a coal-fired power plant could reduce the cost of coal-fired intermediate and peak load power generation by 5% to 25% when compared with a conventional cycling coal-fired power plant. Drost et al. also identified attractive applications for using TES with advanced coal technologies such as IGCC. Based on the results of the 1989 study, the current evaluation of integrating TES in an IGCC plant was initiated.

3.0 METHODOLOGY

The use of TES in an IGCC power plant was evaluated by developing conceptual designs for a number of plant operating schedules. The conceptual designs had sufficient detail to allow development of preliminary capital and LEC estimates. The resulting cost estimates were compared with the costs for a base case using more conventional technology.

3.1 SELECTION OF PLANT OPERATING SCHEDULES

The general approach used in this study was to develop a conceptual design and a cost estimate for the IGCC/TES and to compare these data to the cost for a base case that consists of 1) an IGCC base load power plant and 2) a cycling pulverized-coal-fired power plant providing intermediate load electric power generation.

The comparison between the base case and the IGCC power plant with TES was made for a range of plant operating schedules. Table 3.1 summarizes the assumed operating schedules. The range of operating schedules was selected to include nominal capacity factors ranging from 20% to 40%. Two weekly operating schedules were evaluated. In the first case, the plant was assumed to operate for 5 days per week. In the second case, the plant was assumed to operate for 7 days per week but with a shorter daily operating period. In all cases, the peak plant net output was assumed to be 500 MWe for both the conventional coal-fired plant (the base case) and the IGCC plant with TES.

TABLE 3.1. Plant Operating Schedule

Schedule Numbers	Operating Days/Week	Base Load Hours/Day	Peak Load Hours/Day
1	5	24	8
2	5	24	12
3	5	24	16
4	7	24	6
5	7	24	9
6	7	24	12

3.2 SELECTION OF THE CONCEPTUAL DESIGN

The main element of this study was the development of a conceptual design for an IGCC power plant with TES. The conceptual design process served two purposes. First, the design was detailed enough to allow development of a preliminary capital cost estimate. Second, the design process identified problems that might affect the technical feasibility of the concept.

The integration of TES in an IGCC plant is a challenging design problem because of the large number of options that exist for combining the two technologies. A comprehensive review of the design options was not attempted, but two integration concepts were identified and briefly evaluated. The first concept was described in Section 2.1. In this concept, the gas turbine operates continuously to produce base load power and the Rankine steam power cycle operates intermittently to produce intermediate load power. The second concept was intended to produce only intermediate load power. In this concept, the gasifier operates continuously and the fuel gas is used to heat salt directly during off-peak periods. During peak demand periods, the fuel gas is used to fire a gas turbine; waste heat in the gas turbine exhaust is used to charge the thermal energy storage. The thermal energy in storage is used during peak demand periods to generate steam for the Rankine steam power cycle.

The first concept was selected for further evaluation because it made the maximum use of the combined cycle, increasing the plant's efficiency. This concept was used as the basis for the conceptual design and economic evaluation. Neither design was the subject of rigorous design optimization. Therefore, while the arrangement selected for evaluation is reasonable, it is possible that optimization studies and additional investigations of concept arrangements could result in more attractive plant designs.

The general philosophy of the conceptual design was to minimize modifications to the IGCC because the IGCC plant has already been extensively optimized. Where possible, IGCC component designs were not modified but the size of the components was influenced by the plant operating schedule. New components, such as the molten salt TES and the steam generator that is heated by the molten salt, were characterized and sized.

3.3 ECONOMIC EVALUATION

The IGCC/TES power plant evaluated in this study has base load and intermediate load power production characteristics. Base load power is provided by the gas turbine, which operates continuously 24 hours per day. Intermediate load power is provided by the steam turbine, which operates from 6 to 16 hours per day. The economic evaluation was conducted by calculating and comparing the LEC of IGCC/TES power plants to reference power plants supplying the same mix of base load and intermediate load power output. Levelized cost analysis combines initial cost, annually recurring cost, and system performance characteristics with financial parameters to produce a single figure of merit (the LEC) that is economically correct and can be used to compare the projected energy costs of alternative power plants. The specific economic methodology employed was that defined in Brown et al. (1987). Some of the key factors for the LEC analysis conducted for this study are identified in Table 3.2.

Initial capital costs were first identified for a conventional IGCC power plant at a relatively detailed level. For example, the gas-treating equipment estimate was based on individual estimates for low-temperature coolers, acid gas removal, sulfur recovery, tail gas treating, and the fuel gas saturator. Each individual cost element for the entire plant was assigned to one of the following categories: variable elements (those related to either gas turbine or steam turbine generating capacity) or fixed elements. Some elements were split into both fixed and variable parts. The capacity and cost of the variable elements depend on the plant's planned power generation schedule, and they are generally lower for an IGCC/TES plant than for a conventional IGCC plant. The capacity and cost of the fixed elements are the same for both IGCC/TES and conventional IGCC plants.

TABLE 3.2. Key Levelized Energy Cost Inputs

Initial Costs		
Coal handling	Salt steam generator	Gas turbine
Gasification	Balance of plant	Heat recovery
Steam/condensate	Working capital	Salt storage
Steam turbine	Oxygen plant	Salt piping
Oil/rock storage	Gas treating	Land
		Startup

Recurring Costs
Fixed operation and maintenance
Variable operation and maintenance
Fuel

Design and Performance Factors	
Gas turbine generating capacity	Steam turbine generating capacity
Oil/rock storage capacity	Heat transfer areas of heat recovery heat exchangers
Molten salt storage capacity	Annual plant availability
Annual plant heat rate	Generation schedule

Individual cost elements were also assigned to one of the following cost components:

- coal handling
- gasifier
- radiant cooler
- fuel gas scrubber
- fuel gas saturator
- sulfur recovery
- steam/condensate
- steam power cycle
- oxygen plant
- ash handling
- convective cooler
- low-temperature coolers
- acid gas removal
- tail gas treating
- gas turbine
- balance of plant.

Conventional IGCC power plant component costs were then adjusted, as necessary, to reflect the design conditions of the IGCC/TES power plant. Equations estimating the cost of each of these components as a function of gas turbine or steam turbine generating capacity were derived from IGCC power plant economy-of-scale studies described in the literature.

Salt storage, salt steam generator, and oil/rock storage costs were obtained from research reports on solar thermal power systems. Costs for the heat recovery heat exchangers were estimated based on published cost data from various sources. Finally, molten salt piping designs and cost estimates were prepared by PNL.

Estimates for fixed and variable operation and maintenance (O&M) cost elements were developed in a manner similar to the initial capital costs. First, O&M costs were identified for a conventional IGCC power plant. Individual cost elements were then divided into fixed and variable parts. Cost estimating equations for each part were developed as a function of initial capital cost, steam turbine power output, or gas turbine power output. Current and future coal cost estimates were derived from projections made by several energy forecasting organizations.

Economic assumptions for factors such as the discount rate, income tax rate, and plant life were primarily based on assumptions specified in EPRI's Technical Assessment Guide (1986) for the utility industry. These assumptions are specifically identified in Section 5.

4.0 CONCEPTUAL DESIGN

The conceptual design characterized an IGCC power plant with TES in sufficient detail to develop a meaningful cost estimate. The relevant features of a conventional IGCC power plant are discussed in Section 4.1, and Section 4.2 provides a detailed discussion of the selected arrangement of the IGCC power plant with TES. The remaining sections present the results of the conceptual design by subsystem. Section 4.3 describes the modifications and addition of new components to the IGCC plant. Section 4.4 discusses the TES subsystem, and Section 4.5 presents the steam transport subsystem. The molten salt heated steam generator is described in Section 4.6; the other plant components are discussed in Section 4.7; and Section 4.8 briefly discusses the performance characteristics of an IGCC plant with TES.

4.1 CONVENTIONAL IGCC POWER PLANT

One goal of this study was to develop concept arrangements that minimize the impact of including TES on the design of the IGCC plant. Therefore, there is substantial similarity between the conventional IGCC plant and the IGCC/TES design. Because of the similarity between the two designs, it is important to understand the major characteristics of a conventional IGCC plant, as well as the IGCC/TES plant.

The main features of a conventional IGCC power plant are shown in Figure 4.1. A Texaco coal gasifier is used to produce clean medium-Btu (300 Btu/scf) fuel gas, which is used to fire a gas turbine. The gas turbine exhaust is used in a heat recovery steam generator (HRSG) to produce steam for a Rankine power cycle and to produce process steam required by other components of the IGCC plant. Currently, available fuel gas cleaning equipment requires that the fuel gas be cooled after it leaves the gasifier. This cooling is accomplished through the use of radiant and convective fuel gas coolers. These heat exchangers generate supplemental steam, augmenting the output of the HRSG. The steam power cycle operates with steam turbine inlet conditions of 521°C (970°F) and 10.00 MPa (1450 psi). Some feedwater heating [to 112°C (233°F)] takes place by heat exchange with the fuel gas stream. The plant configuration illustrated in Figure 4.1 is highly efficient: its

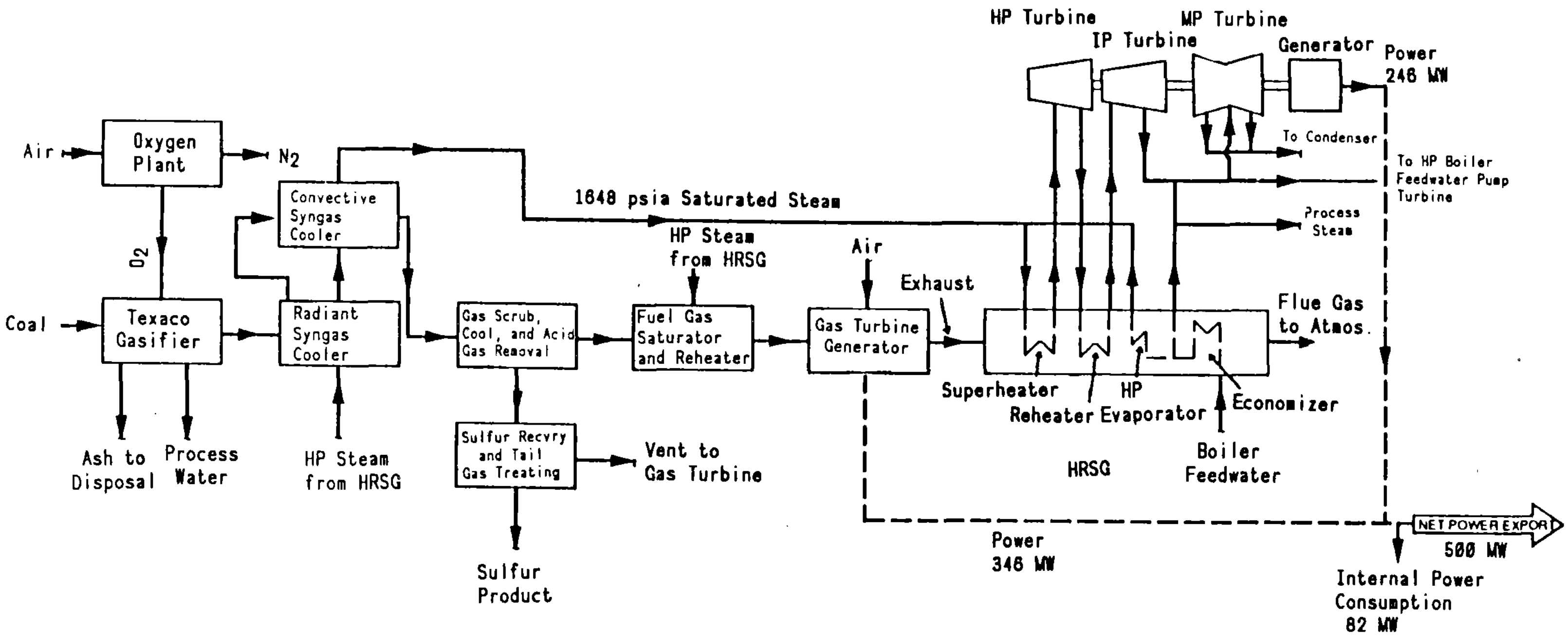

FIGURE 4.1. Conventional IGCC Power Plant with Current Technology Turbine

net efficiency is 36.6% (a heat rate of 9322 Btu/kWh). In general, the power plant uses commercially available equipment; the major exceptions are the high-temperature fuel gas coolers, which require additional development. The gas turbine, with an inlet temperature of 1104°C (2020°F), is representative of current gas turbine technology. As gas turbines with operating temperatures of 1204°C (2200°F) become available, performance is expected to improve.

The Texaco IGCC power plant has been extensively studied and appears to be competitive with a conventional coal-fired power plant on the basis of cost and performance (Matchak et al. 1984). However, the operating characteristics of the IGCC plant are somewhat different from a conventional coal-fired power plant when considering cycling operation. The major differences involve the gasifier, which is not readily cycled or economically operated at part-load. Therefore, the IGCC concept is only being considered for base load service.

The gas turbine also has important operational differences compared with the conventional coal-fired power plant. The gas turbine cycle is a constant-volume heat engine. Consequently, the power output and fuel requirements vary inversely with ambient temperature and will produce the rated output at only one ambient temperature. Depending on the design philosophy and the design ambient temperature selected, either the gasifier or the gas turbine will be oversized during most of the plant operating time. If the plant design point is at a low ambient temperature, the gas turbine's output is significantly reduced [by approximately 20% if the unit was designed using an ambient temperature of -6.7°C (20°F) and was actually operating at 31°C (88°F)] at high ambient temperature. Unfortunately, many utilities experience peak power demand when the ambient temperature is high because of increased air-conditioning loads. This problem can, to some extent, be mitigated by supplemental firing, in which the excess gasifier output is used to heat the gas turbine exhaust, generating additional steam in the HRSG. The increased steam production requires the installation of an oversized steam turbine, which results in increased power production and heat rate. Supplemental firing was not included in the conceptual design of the IGCC/TES concepts.

4.2 IGCC POWER PLANT WITH MOLTEN SALT TES FOR PEAKING

A molten salt TES interposed between the gasifier/gas turbine and the steam generator in an IGCC plant can provide a cycling capability. Instead of generating steam directly, the heat from the fuel gas coolers and turbine exhaust is used to heat molten nitrate salt which is then stored. The gas turbine is operated whenever the gasifier operates. Any excess fuel gas [at ambient temperatures >15°C (59°F) design point] could be used to heat salt in the TES. The increased peaking capacity will be available during periods of high ambient temperature, when peaking capacity is most needed by summer peaking utilities. The TES serves to decouple the steam generator and turbine from the rest of the plant, allowing steam power production as needed for peak power generation.

An earlier study examined molten salt TES application with pulverized-coal-firing technology and determined that conventional materials (various stainless steels) would be adequate for constructing the salt heaters (Drost et al. 1989). In this study, it was assumed that similar materials could be used for the fuel gas coolers. The same design philosophy was applied to the turbine exhaust heat recovery salt heater. In the case of the heat recovery salt heater, it may be possible to use direct heat exchange between the exhaust gas and the salt. If feasible, direct-contact heat exchange would dramatically decrease the cost of the heat recovery salt heater and would improve performance.

The molten nitrate salt freezes at 240°C (464°F); and, therefore, the cold TES tank is designed for 288°C (550°F) operation. Efficient operation of the IGCC plant requires that thermal energy in gas streams at temperatures below 316°C (600°F) be used in the IGCC plant. The concept proposed in this study uses these gas streams to generate low-pressure process steam for use in the gasification process, and the remaining heat is stored in a second low-temperature TES system. The low-temperature TES system stores thermal energy in a medium consisting of oil and rock, which has been developed extensively for solar thermal power generation.

4.3 MOLTEN SALT HEATER DESIGNS

The proposed integration of TES in an IGCC power plant requires modifications to the radiant fuel gas cooler, the convective fuel gas cooler, and the heat recovery salt heater to allow the components to heat molten salt (and oil in the case of the heat recovery salt heater). Section 4.3.1 presents a comparison of the heat transfer characteristics of molten salt versus water/steam. Section 4.3.2 summarizes the design of the three salt heaters and one oil heater included in the design of the IGCC plant with TES. Section 4.3.3 discusses the incorporation of other heat exchangers into the IGCC/TES design to provide steam for fuel gas saturation and feedwater heating.

4.3.1 Comparison of Heat Transfer Characteristics of Molten Salt and Steam Generation

Although the relevant molten salt thermal-hydraulic properties are not radically different from water (Martin Marietta 1984), new designs for the radiant fuel gas cooler, convective fuel gas cooler, and HRSG are required because the molten salt does not undergo the phase change that water/steam experiences. The molten salt remains a single phase fluid throughout the heat transfer process, which results in good heat transfer characteristics throughout all zones of the salt heater and avoids the high pressures encountered during steam generation. Because there will not be high pressures in the salt heater, the tube walls can be significantly thinner. Scale buildup is not a problem with molten salt, but tube corrosion and long-term salt degradation are issues that must be addressed in a realistic design.

In this study, it was assumed that the three salt heaters would have many features in common with their counterparts in the conventional IGCC design. The arrangement of heat transfer surfaces and the structural design of the salt heater were also assumed to be similar between the two sets of components.

The major differences between a salt heater and a steam generator are in the heat exchange surfaces. Two problems encountered in the analysis were the uncertainty in salt properties and the lack of a proven design for some components such as the high-temperature radiant and convective fuel gas cooler.

While molten nitrate salt has been extensively investigated, substantial uncertainty still exists in the estimates for some important properties, especially specific heat, where there are significant variations in the reported results (Carling 1983; Martin Marietta 1984; De Laquil, Kelly, and Egan 1988). The second difficulty was a lack of detailed data on fuel gas cooler design. To minimize the impact of these difficulties, the analysis relied on the results of a recent comparison of a coal-fired boiler with a coal-fired salt heater (Drost et al. 1989).

Typical convective heat transfer coefficients for water and molten salt are presented in Table 4.1. The values for molten salt are taken from Martin Marietta (1978), and the values for water and steam are taken from Kreith and Bohn (1986).

A review of Table 4.1 shows that while molten salt is a good heat transfer fluid, its convective heat transfer coefficient is somewhat lower than forced convection with water and substantially lower than boiling heat transfer with water. Conversely, the convective heat transfer coefficient for molten salt is substantially greater than that for steam. While water has a higher convective heat transfer coefficient than molten salt, the impact of this difference on the design of a radiant or convective salt heater is negligible because the resistance to heat transfer from the fuel gas to water or molten salt is dominated by the external convective heat transfer coefficient between the external surface of a tube and the flue gas. Sorensen (1983) and Babcock and Wilcox (1978) report the overall heat transfer coefficient between flue gas and water as being between 50 W/m^2-°C [8.33 Btu/(h-ft^2-°F)] and 75 W/m^2-°C [12.5 Btu/(h-ft^2-°F)]. The resistance to heat transfer attributable to the convective heat transfer coefficient between an internal surface of a tube and water is less than 1% of the overall resistance. Consequently, the impact of water or molten-salt-side convective heat transfer coefficient can be ignored without loss of accuracy. This is not the case for steam because the steam-side convective heat transfer coefficient is much lower than for either water or molten salt. The major conclusion resulting from the

TABLE 4.1. Typical Convective Heat Transfer Coefficients for Water and Molten Salt

Fluid	Convective Heat Transfer Coefficient W/m^2-°C	Btu/(h-ft^2-°F)
Water, single phase	12,000	2,000
Water, boiling	60,000 to 120,000	10,000 to 20,000
Steam	300	50
Molten salt	6,000 to 9,000	1,000 to 1,500

evaluation of heat transfer coefficients is that molten salt will have heat transfer characteristics equivalent to water for convection or boiling heat transfer and will be superior to steam.

One disadvantage of molten salt is its low heat capacity. One kilogram of water will absorb approximately 2.33×10^6 J (1000 Btu/lbm) as it passes through the boiler; the high heat capacity is caused by the water going through a phase change. Molten salt does not experience a phase change; therefore, there is a substantial reduction in its heat capacity and a related increase in the required mass flow rate for a molten salt system when compared to a similarly-sized water/steam system. A study by Martin Marietta (1984) indicates that at 371°C (700°F), the specific heat of molten salt is estimated to be 1616 J/(kg-°C) (0.386 Btu/lbm-°F) with a density of 1849 kg/m^3 (115 lbm/ft^3). There is a substantial variation in the reported values of specific heat for molten salt (Martin Marietta 1978, 1984; Kolb and Nikolai 1988; De Laquil, Kelly, and Egan 1988). This study used a value of 1550 J/(kg-°C) (0.372 Btu/lbm-°F), which was assumed to be independent of temperature.

The major conclusion for this comparison of molten salt and water/steam heat transfer characteristics is that the mass flow rate of molten salt is approximately six times greater than the mass flow rate of water to achieve the same heat removal rate. The volume flow rate for molten salt will be approximately three times greater than that for water.

Molten salt has been evaluated for use in solar central receiver systems (U.S. DOE 1988a,b). Evaluations have included the design and testing of molten salt central receivers. These evaluations identified the preferred tube design for solar applications. The recommended tube diameter was 0.038 m (1.5 in.) with a recommended tube wall thickness of 0.0017 m (0.065 in.) (Martin Marietta 1978). The molten salt would be at an approximate pressure of 1.724 MPa (250 psia), which would only require a wall thickness of 0.0006 m (0.0218 in.); the additional wall thickness facilitates fabrication. Given the low molten salt pressure and the high flow rates required in a coal-fired salt heater, it was concluded that tubing diameters of 0.063 m (2.48 in.) or larger should be used to minimize pressure drop. Pressure drop could probably be achieved using the same 0.0017-m (0.065-in.) wall thickness used in the smaller diameter tubing.

A molten salt solar receiver using 0.0255-m (1.00-in.) diameter Incoloy 800 tubing was initially tested at a peak flux of 400,000 W/m^2 (Martin Marietta 1978). This was followed by a test of a 5-MWt receiver, using 0.019-m (0.75-in.) diameter tubes at a central receiver test facility. The receiver was designed for a peak flux of 0.6 MWt/m^2 (190,000 Btu/ft^2-h) (Delameter 1987; Bergan 1987). A high flux test exposed the receiver to flux levels above 1 MWt/m^2 (317,000 Btu/ft^2-h) for over 1 hour, and the receiver survived the test with no damage. The second test confirmed that a conservative design peak flux for a molten salt receiver with Incoloy 800 tubes is 0.85 MWt/m^2 (269,000 Btu/ft^2-h) (Bergan 1987). Current molten salt solar central receivers are designed for a peak flux of approximately 0.75 MWt/m^2 (238,000 Btu/ft^2-h) using 316 stainless steel tubing (De Laquil, Kelly, and Egan 1988; U.S. DOE 1988b).

The results of the molten salt solar receiver development program suggest that the heat fluxes typically encountered in a coal-fired furnace [0.16 MWt/m^2 (50,700 Btu/ft^2-h)] can be readily accommodated in a coal-fired salt heater. A detailed design study for a radiant fuel gas cooling salt heater could possibly take advantage of the properties of molten salt to further improve the design, but consideration of these issues was beyond the scope of this study.

4.3.2 Salt Heater Design Characteristics

The design of the salt heaters for application with the IGCC concept was based on the results of a recent comparison of a molten salt heater with a coal-fired boiler (Drost et al. 1989). Drost et al. estimated the overall heat transfer coefficients for various regions of a coal-fired molten salt heater, and it was assumed that this information could be used to size the molten salt heaters used with the IGCC concept. The specific assumptions were that 1) the overall heat transfer coefficient in the radiant region of the coal-fired molten salt heater could be used to size the radiant fuel gas cooler and 2) the overall convective heat transfer coefficient in the convective passes of the coal-fired steam generator could be used to size the convective fuel gas cooler and heat recovery molten salt heater and oil heater.

It was assumed that the IGCC radiant salt heater experienced operating conditions similar to the radiant regions of a coal-fired salt heater. Results reported in Drost et al. (1989) estimated the overall heat transfer coefficient as being 70 W/m^2-°C (12.3 Btu/h-ft^2-°F). It was assumed that this value would also apply to the heat exchange between the raw fuel gas and the salt. The log mean temperature difference was calculated to be 639°C (1182°F). An examination of the effect of the fuel gas pressure determined that an adequate factor of safety exists with 1/8-in.-thick walls in 3-in. OD tubes of 304H stainless steel.

The design of the fuel gas convective cooler is also based on the results reported in Drost et al. (1989) for a convective salt heater in a pulverized-coal-fired salt heater. The overall convective heat transfer coefficient in the convective passes of the coal-fired salt heater was calculated to be 150 W/m^2-°C (26.6 Btu/h-ft^2-°F). The same value was assumed for the convective fuel gas cooler. The appropriate log mean temperature difference was calculated to be 112°C (233°F). The tube material is 304H stainless steel, with 1/8-in.-thick walls and 3-in. OD.

The gas turbine heat recovery molten salt heater was also assumed to be similar to the convective passes of a coal-fired salt heater, with an overall convective heat transfer coefficient of 150 W/m^2-°C (26.6 Btu/h-ft^2-°F). However, the clean turbine exhaust permits extensive use of fins to improve heat transfer on the gas side. The log mean temperature difference was estimated to be approximately 14°C (57°F). Tube material was assumed to be 304H stainless steel with approximately 1/16-in.-thick walls and ≤1-in. OD, but the fins were assumed to be carbon steel.

The turbine exhaust oil heater replaces the low-temperature heat exchanger surfaces in the HRSG. It was assumed that the overall convective heat transfer coefficient would approximately equal the value used for the salt heater. This was justified because the dominant resistance to heat transfer is on the gas side. The calculated log mean temperature difference was found to be about 4°C (39°F). With the lower temperatures encountered by the oil heater, the fins and tubing (of a size similar to the salt heater tubing) can be constructed of SA285 Gr. A or other carbon steel.

4.3.3 Other Heat Exchangers

An IGCC plant typically requires two sources of hot feedwater/steam: feedwater (as in the Texaco IGCC plant) or steam (as in the Shell IGCC plant) is used to saturate the fuel gas to a moisture content of about 30 wt% prior to combustion in the gas turbine to reduce NO_x emissions from the turbine; a secondary result of the saturation process is somewhat improved turbine efficiency. Also, 0.74-MPa (100-psia) process steam is required for IGCC plant use. In the conventional Texaco IGCC plant, both needs are met by the HRSG, from which steam and feedwater are extracted.

In this design, it was decided to inject saturated 1.48-MPa (200-psig) steam into the gas turbine, following the design proposed for the Shell process. Thermal energy from the gas turbine exhaust will be used to generate this steam.

A similar rationale holds for the 0.74-MPa (100-psia) process steam. The heat required for this process can be supplied by cooling raw fuel gas from 128°C to 54°C (262°F to 130°F), which in the conventional Texaco plant is used for the feedwater heating.

4.4 SALT TRANSPORT SUBSYSTEM

The salt transport system design was taken from the results of a study of the integration of a TES system with conventional coal-firing technology

(Drost et al. 1989). In that study, a conceptual layout for the nitrate salt storage tanks, hot salt piping, and pumps was developed. The major components of the transport system were sized, and the thermal and pumping requirements were calculated. Thermal losses, which include warmup losses and trace heating, were based on recent experimental and analytical work performed for DOE's Solar Thermal Program (Kolb and Nikolai 1988; Martin Marietta 1985; De Laquil, Kelly, and Egan 1988). The design of the transport subsystem was subsequently used to develop a cost estimate for the subsystem.

4.5 SALT STORAGE SUBSYSTEM

The salt storage subsystem decouples the production of thermal energy in the gasifier and HRSG from the production of electricity. The storage subsystem must store hot salt at 566°C (1050°F) for extended time periods without excessive thermal losses or capital cost. Expensive materials (relative to carbon steel) are required to contain hot molten salt, and the main design problems are related to minimizing the use of these expensive materials in the storage vessels. Molten salt TES has been extensively investigated for solar thermal applications; therefore, where possible, the current evaluation has relied on the results of these studies.

4.5.1 Design Options

Two design options have been proposed for the nitrate salt TES subsystem. The first option uses one tank with the hot and cold salt separated by a thermocline (Martin Marietta 1978). The second option uses separate hot and cold molten salt tanks (Martin Marietta 1985).

The thermocline system is attractive because only one tank is required. The thermocline is formed by adding or removing hot salt from the top of the tank and adding or removing cold salt from the bottom of the tank. The lower density of the hot salt keeps the hot salt in the upper region of the tank while the cold salt occupies the lower region. The narrow zone between the hot and cold regions is the thermocline. However, the most recent studies suggest that the cost savings associated with a thermocline system are small because the cost of the cold tank in the two-tank system is a small fraction of the total cost. In addition, it may be difficult to maintain the thermocline because of radiation heat transfer between the hot and cold regions of the tank.

A system using separate hot and cold tanks avoids the difficulties associated with maintaining the thermocline. All recent studies have selected a two tank system (Martin Marietta 1985; Ross, Roland, and Bouma 1982; Delameter 1987; U.S. DOE 1988a).

4.5.2 Proposed Design

The maximum size of a molten salt storage tank is limited in diameter by the maximum realistic tank wall thickness. Large-diameter tanks with a wall thickness exceeding 0.038 m (1.5 in.) must undergo post-weld heat treatment, which is prohibitively expensive (Martin Marietta 1985). This limits the maximum tank diameter to approximately 25 m (82 ft). The maximum tank height is limited by the soil-bearing strength. Assuming a soil-bearing strength of 0.24 MPa (50 psia), the maximum height of the stored hot salt will be 13 m (42.6 ft). Therefore, the maximum storage tank size would be 25 m (82 ft) in diameter and 13 m (42.6 ft) high, although the actual tank dimensions will be somewhat larger to accommodate an inert cover gas (Martin Marietta 1985). Because the maximum storage volume will be around 6000 m^3 (212,000 ft^3), multiple hot and cold salt tanks will be required. The hot tank design is externally insulated and uses 316 stainless steel or even less expensive 304H stainless steel for the wall material (U.S. DOE 1988a; De Laquil, Kelly, and Egan 1988). With external insulation, the integrity of a tank is unaffected by insulation failure. Nonconventional tank configurations, such as the conical tank design proposed in Kohl, Newcomb, and Castle (1987) have the potential to reduce tank costs but require additional research and demonstration.

The cold tank dimensions are similar to the hot tank dimensions. The cold tank walls will be fabricated from A516 carbon steel with external insulation (U.S. DOE 1988a; De Laquil, Kelly, and Egan 1988). Salt freeze protection may be required; however, because of the long time span before freezing occurs in such large vessels (3 to 6 months), freeze protection may not be a critical issue (Ross, Roland, and Bouma 1982). The tanks will be surrounded by dikes to contain salt spills.

4.6 OIL/ROCK STORAGE SUBSYSTEM

The low-temperature TES consists of a storage medium of a heat transfer oil and river rock. The oil and rock are contained in one or more carbon steel tanks, depending on the size of the low-temperature TES subsystem. The tank or tanks are insulated to reduce heat loss, and appropriate foundations and miscellaneous equipment are included. All piping is assumed to be schedule 40 carbon steel with calcium silicate insulation. The tanks are enclosed in dikes to contain oil spills.

A substantial fraction of the low-temperature TES tank volume is filled with the inexpensive rock; the remaining volume is filled with the more costly oil. The thermal energy is stored as sensible heat in both the oil and rock: the oil, which represents 25% of the storage volume, stores 19.2% of the thermal energy as sensible heat; and the rock, which represents the balance of the volume, stores 80.8% of the thermal energy as sensible heat. Hot oil is added or removed from the top of the tank; cool oil is added or removed from the bottom of the tank. This arrangement maintains a density-driven segregation (thermocline) between the hot oil in the top of the tank and the cool, denser oil in the bottom of the tank.

Heat transfer oils tend to degrade at elevated temperatures. Above 304°C (580°F), the degradation rate is very high and heat transfer oils cannot be successfully used. Below 304°C (580°F), degradation still occurs but at a substantially reduced rate. The oil and rock TES subsystem operates at a maximum temperature of 304°C (580°F), which is low enough to preclude substantial oil degradation. However, during operation of the low-temperature TES system, the products of degradation must be removed and replacement oil must be added (Hallet and Gervais 1977). A fluid maintenance system filters the oil to remove suspended solids, distills a side stream to remove high boiling polymeric compounds, and adds fresh makeup fluid to replace decomposed fluid. In addition to maintaining an oxygen-free gas above the heat transfer fluid, the maintenance system also removes the volatile fractions of the degradation products that evaporate into the ullage space above the heat transfer oil.

4.7 STEAM GENERATOR SUBSYSTEM

During peak demand periods, thermal energy is extracted from storage and used to produce steam for a conventional Rankine power cycle. Steam production occurs in the steam generator subsystem, which consists of a number of heat exchangers where thermal energy is transferred from the molten salt to water or steam. DOE's Solar Thermal Program conducted five design studies of molten salt steam generator systems for molten salt solar central receiver systems (Martin Marietta 1978; Ross, Roland, and Bouma 1982; Weber 1980; U.S. DOE 1988b; De Laquil, Kelly, and Egan 1988). These studies form the basis for the design described below.

4.7.1 Description

The steam generator subsystem consists of four separate heat exchangers: 1) a preheater where the temperature of the feedwater is raised to the saturation temperature; 2) an evaporator where steam is generated; 3) a superheater where the saturated steam is superheated; and 4) a reheater where the high-pressure turbine exhaust is heated to 538°C (1000°F). The design will be presented as if each heat exchanger were in one shell; but given their large size, multiple parallel heat exchanger trains will be used.

A recirculating steam generation arrangement is proposed. In this arrangement, the evaporator includes a steam drum and steam separator. A mixture of water and steam enters the steam separator where the vapor phase is separated from the liquid phase and sent to the superheater. The liquid is recirculated to the evaporator.

4.7.2 Proposed Design

The heat exchangers are single-pass tubular heat exchangers; water/steam is contained within the tubes. A single-pass design was selected because of the desire to have counterflow heat exchange in the preheater, superheater, and reheater. The internal baffling will be arranged so that molten salt flow approximates a counterflow arrangement (Martin Marietta 1978). This internal baffling arrangement significantly reduces the salt pressure drop but adversely affects the heat exchangers (Martin Marietta 1978; Kays and London 1964). The evaporator uses a parallel-flow arrangement (Ross, Roland, and Bouma 1982).

The heat exchanger designs use long tubes with the smallest shell diameter consistent with reasonable salt pressure drop. This results in a heat exchanger design with a 15- to 20-m (49- to 66-ft) length and a 1- to 2-m (39- to 79-in.) shell diameter (Martin Marietta 1978).

The characteristics of the four heat exchangers are presented in Table 4.2. The design specifications are based on the designs proposed by Martin Marietta (1978), but they have been modified to reflect current design practices in regard to design margins and materials selection. Alternative designs using a U-shell design (U.S. DOE 1988a; De Laquil, Kelly, and Egan 1988) and hemispherical head arrangements (Ross, Roland, and Bouma 1982) have been proposed but would most likely be more expensive than the conventional designs selected here.

4.8 BALANCE-OF-PLANT

The balance-of-plant includes conventional electric power-generating equipment and cooling towers. The power generating equipment consists of a

TABLE 4.2. Heat Exchanger Specifications

Parameter	Preheater	Evaporator	Superheater	Reheater
Heat transfer surface, m^2	21,700	21,800	3,150	4,100
Approximate shell diameter, m	1 to 1.5	1 to 1.5	1.5 to 2	1.5 to 2
Approximate tube length, m	20	20	15	15
Tube OD, m	0.016	0.016	0.016	0.016
Tube wall thickness, m	0.00163	0.00163	0.00163	0.00163
Baffle spacing, m	0.75	0.75	0.00	0.75
Material	Carbon-steel	2¼ Cr-1 Mo[a]	SS 304[b]	SS 304
Number of heat exchangers required	2 to 4	2 to 4	2	2

(a) 2¼% chrome, 1% molybdenum.
(b) SS = stainless steel.

reheat condensing steam turbine with separate high- and low-pressure turbines. The efficiency of the combined IGCC plant with TES is enhanced through the use of a 16.54 MPa/538°C/538°C (2400 psi/1000°F/1000°F) reheat steam turbine with feedwater heating; all of the feedwater heating to about 178°C (353°F) is accomplished by means of the oil/rock TES. Two stages of conventional extraction steam feedwater heaters (not shown or discussed further) provide the final heating to 253°C (488°F). Condensate pumps, a deaerator, feed pumps, feedwater heat exchangers, and a shell-and-tube type condenser operating at 2.5 in. Hg (absolute) plus wet cooling towers complete the equipment necessary for a working electric power-generating facility.

4.9 PLANT PERFORMANCE

The heat rate of the IGCC plant with molten salt TES will be different than the heat rate for a conventional IGCC plant. Additional parasitic losses associated with the TES system will tend to increase the heat rate, but this effect is more than offset by the improved efficiency of the steam cycle.

TES parasitic losses include increased parasitic electric power consumption related to salt pumping and heat tracing and thermal losses from the storage tanks. Thermal losses are approximately 25 MWht per day while the increased parasitic electric load is between 600 and 1100 kWe. The calculation of parasitic losses was based on performance estimates developed by Kolb and Nikolai (1988) and Martin Marietta (1985).

The steam turbine used in a conventional IGCC plant operates with turbine inlet conditions of 10 MPa (1450 psia) and 521°C (970°F) and has a gross heat rate of 9322 Btu/kWh. The IGCC plant with TES is projected to operate with turbine inlet conditions of 16.54 MPa (2400 psia) and 538°C (1000°F) with a gross heat rate of 8690 Btu/kWh. The improved steam conditions result from the cycle arrangement that allows a maximum molten salt temperature of 565°C (1050°F) and from the larger size of the steam turbine. The result is that the net heat rate of the IGCC unit with TES is around 9180 Btu/kWh as compared with a net heat rate of 9322 Btu/kWh for the conventional IGCC plant.

5.0 ECONOMIC EVALUATION

This section presents detailed information regarding the cost and economic analysis for IGCC power plants using molten salt TES and for reference power plants supplying the same mix of base load and intermediate load power output. Section 5.1 defines the cost estimating and economic assumptions used in the analysis. Sections 5.2 and 5.3 discuss the estimating approach and cost results for the initial capital, operation, and maintenance of an IGCC/TES power plant. Section 5.4 presents and discusses the main results of the LEC analysis. Additional LEC results from two sensitivity studies are presented in Section 5.5.

5.1 GROUND RULES AND ASSUMPTIONS

The economic assumptions used to calculate the LEC are listed in Table 5.1. Each of these assumptions was either taken directly or calculated from data in EPRI's Technical Assessment Guide (1986) except for the combined federal and state income tax, price year, the first year of plant operation, and the fuel inflation rate. Brown et al. (1987) was the reference for the combined state and federal income tax rate. The first year of operation was set at the year 2000 because this was felt to be a reasonable time frame for bringing TES on-line with new power plants. A price year of 1987 was selected

TABLE 5.1. Financial Assumptions

Description	Assumption
Discount rate	10.5% (nominal)
General inflation rate	6.0%
Capital inflation rate	6.0%
Operation and maintenance inflation rate	6.0%
Fuel inflation rate	7.0%
Investment tax credit	0.0%
Property tax and insurance rate	2.0%
Combined state and federal income tax rate	39.1%
Plant economic life	30 years
Plant depreciable life	20 years
Plant construction period	3 years
Price year	1987
First year of plant operation	2000

to be consistent with the approach used in Drost et al. (1989), which evaluated the integration of molten salt TES with pulverized-coal-fired power plants. The selection criteria for the fuel price inflation rate are discussed in Section 5.3.

5.2 CAPITAL COST ESTIMATES

Capital cost estimating equations were developed for 16 conventional IGCC power plant components, 8 heat recovery and energy storage components, startup, working capital, and land. The following are the capital cost components:

- coal handling
- gasifier
- radiant cooler
- fuel gas saturator
- sulfur recovery
- steam/condensate
- steam power cycle
- salt heater
- oil heater
- oil/feedwater heat exchanger
- salt piping
- oxygen plant
- ash handling
- convective cooler
- acid gas removal
- tail gas treating
- gas turbine
- LP steam generator
- oil/rock storage
- salt storage
- salt steam generator
- balance of plant
- working capital
- startup
- land.

5.2.1 Conventional IGCC Power Plant Component Costs

The general approach to characterizing the costs associated with conventional IGCC power plant components was to determine the costs for a specific plant generating capacity, to adjust the costs as necessary to reflect IGCC/TES plant design conditions, and to develop individual cost estimating equations for the major power plant components. The reference IGCC power plant is based on Texaco's coal gasification technology, which uses radiant and convective coolers and current generation [1093°C (2000°F)] combustion turbines. This plant partially oxidizes a concentrated water slurry of high-sulfur midwestern bituminous coal to produce an intermediate-Btu gas at approximately 1316°C (2400°F). After passing through radiant and convective coolers (where steam is produced for a Rankine cycle turbine), the intermediate-Btu gas is further cooled and cleaned and finally is combusted in a gas turbine. More detailed design information can be found in Section 4 and in several reports prepared for EPRI (e.g., Matchak et al. 1984; Snyder et al. 1986; Jacob and Chu 1988).

Relatively detailed cost estimates for the conventional IGCC power plant were necessary to segregate cost elements into those that would remain the same, change, or be deleted in an IGCC/TES power plant. Detailed estimates and design descriptions also allowed the grouping of costs into elements that are fixed or that vary with gas turbine and/or steam turbine generating capacity. The capacity and costs for variable elements depend on the planned power generation schedule for an IGCC/TES plant. In general, the capacity and costs of the variable elements are lower for the IGCC/TES plant than for the conventional IGCC plant. The capacity and cost of the fixed elements are the same for both IGCC/TES and conventional IGCC plants. Reference IGCC power plant costs were taken from a design and cost study prepared by Jacob and Chu of Fluor Engineers, Inc. (1988) for EPRI. The Jacob and Chu study was selected as the reference cost source because it was the latest publicly available study on the Texaco IGCC power plant with radiant and convective cooling.

The direct installed capital cost estimates published in Jacob and Chu (1988) are listed in Table 5.2. Supporting cost detail acquired directly from Fluor Engineering, Inc. was used to develop cost estimating equations for each of the list of PNL cost components shown in Table 5.2, but these details are not shown in order to protect the proprietary interests of Texaco. Table 5.2 also indicates whether the component was presumed to vary with the plant's gas turbine generating capacity or steam turbine generating capacity, or was fixed for the power plants evaluated in this study. Assignment to a particular PNL capital cost component and fixed or variable status was established by reading the specific equipment descriptions provided in the Fluor study for their capital cost components. The Fluor study estimates are in January 1987 dollars and were adjusted upward by 1.5% to reflect price increases to mid-1987, the reference date for this study.

The costs for several items included in the Fluor study were either deleted, replaced, or modified to adjust to the specific design conditions of

TABLE 5.2. Reference IGCC Power Plant Costs

Source: Jacob and Chu (1988)
Gas Turbine Generating Capacity = 340 MWe
Steam Turbine Generating Capacity = 260 MWe
Thousands of January 1987 dollars

Fluor Cost Component(s)	PNL Cost Component(s)	Direct Field Costs	PNL Cost Variable[a]
Site work and buildings	Balance of plant	19,581	1,3
Fuel systems	Coal handling	33,200	1
Air separation	Oxygen plant Balance of plant	77,297	1 3
Gasification	Coal handling, Gasifier Ash handling, Radiant cooler, Convective cooler, Fuel gas scrubber	110,428	1 1 1 1 1 1
Gas treating	Low-temp. coolers, Fuel gas saturator, Acid gas removal, Sulfur recovery, Tail gas treating	43,485	1 1 1 1 1
Power generation	Steam/condensate, Gas turbine, Steam power cycle	195,994	1 1 2
Water systems	Balance of plant	53,859	1,2,3
Electrical systems	Balance of plant	29,956	1,3
Balance of plant	Balance of plant	21,164	1,3

(a) "1" indicates costs assumed to vary with gas turbine generating capacity.
"2" indicates costs assumed to vary with steam turbine generating capacity.
"3" indicates costs assumed to be fixed for plants evaluated in this study.

an IGCC/TES plant. Deleted items include the costs for fuel oil and natural gas supply systems. These items were included in the Fluor estimate because they assumed a phased construction scenario in which the plant would operate initially with oil-fueled or natural-gas-fueled turbines and the coal gasification system would be constructed later.

The radiant and convective coolers for the IGCC/TES plant were estimated to cost less than those for a conventional IGCC plant because of the difference in operating pressure. In a conventional IGCC power plant, steam is produced in the radiant and convective coolers at approximately 10.3 MPa (1500 psi), but the salt-filled tubes in the radiant and convective coolers of the IGCC/TES plant would operate at less than 1.48 MPa (200 psi). Based on an estimating rule of thumb from Guthrie (1974) which defines the relative costs of process furnaces as a function of tube pressure, the radiant and convective coolers for the IGCC/TES plant are estimated to be about 30% less expensive than for a conventional IGCC plant.

All costs in the Fluor study related to the steam turbine system, including the HRSGs, were replaced because of changes in the the hardware for the gas turbine exhaust heat recovery and in the steam turbine inlet conditions. The steam power cycle cost estimating equation in Drost et al. (1989) was adjusted for feedwater heating requirements supplied by the oil/feedwater heat exchanger in the IGCC/TES plant design and was substituted for the Fluor costs. Costs for the IGCC/TES gas turbine exhaust heat recovery heat exchangers are discussed in Section 5.2.2.

Two other direct construction cost elements (the switchyard and the generator step-up transformer) were not included in the Fluor estimate, so they were estimated separately. The costs for these two elements were derived from data in EPRI (1986) and Dunlop and Slambrook (1986) and were estimated to be \$11,080,000 for the switchyard and \$4,430,000 for the generator step-up transformer in mid-1987 dollars.

Equations estimating the direct installed construction costs of conventional IGCC power plant components as a function of gas turbine and/or steam turbine generating capacity were developed by a two-step process. The first step was to determine the economy-of-scale factor controlling relative costs at different plant power ratings for each major plant cost component (the PNL cost components are listed in Table 5.2). The economy-of-scale factor is defined as the exponent "B" in the following generalized equation: Cost = A*(Plant Power Rating)^B. The second step was to solve for the values of "A" in the above generalized equation that would yield the estimates from Jacob and Chu (1988) for the same PNL cost components, given the economy-of-scale factors ("B" values) identified in the first step.

Economy-of-scale factors for each account were determined by analyzing cost estimates published in Matchak and Lawrence (1983). This study specifically looked at the impacts on plant performance, capital cost, and the cost of electricity caused by varying the generating capacity of Texaco-based IGCC power plants from 50 to 1000 MWe.

Cost data from Matchak and Lawrence (1983) are shown in Table 5.3 for the three gas turbine generating capacities of interest to this study. Regression analysis techniques were applied to the data in Table 5.3 to determine the economy-of-scale factor for each of the cost components. In turn, the economy-of-scale factors and costs for the IGCC components in Jacob and Chu (1988) were used to develop direct construction cost estimating equations for each PNL cost component. Again, the individual cost component estimating equations are not presented to protect the proprietary interests of Texaco.

5.2.2 Heat Recovery and Energy Storage Component Costs

The heat recovery and energy storage portion of an IGCC/TES power plant includes the salt heater, low-pressure steam generator, and oil heater fired by the gas turbine exhaust; oil/rock storage and oil/feedwater heat exchanger; and salt storage, salt piping, and salt steam generator. This section of the report discusses the estimating approach and cost results for these components.

The costs of the three gas turbine exhaust heat recovery heat exchangers and the oil/feedwater heat exchanger were estimated from PNL calculations of the required heat transfer area and published estimating data and procedures for the heat exchanger types and design conditions encountered (Foster-Pegg 1986; Corripio, Chrien, and Evans 1982; Guthrie 1974; Peters and Timmerhaus 1980). The resulting cost estimating equations are shown in Table 5.4.

TABLE 5.3. Economy-of-Scale Factor Cost Data (thousands of 1981 dollars)

Cost Account	Gas Turbine Generating Capacity, MWe			
	58.75	151.40	303.58	Scale Factor
Coal handling	7,036	14,981	23,888	0.747
Oxidant feed	14,551	30,569	59,269	0.851
Gasification and ash handling	7,016	13,915	23,066	0.725
Gas cooling	17,961	40,566	67,428	0.809
Acid gas removal	2,841	5,672	9,535	0.737
Sulfur recovery	2,000	3,480	5,700	0.635
Tail gas treating	4,050	6,900	10,800	0.595
Steam, condensate and boiler feedwater	860	1,940	2,940	0.755
Combined cycle	37,386	70,120	135,110	0.737
General facilities	19,462	27,870	41,061	0.450

TABLE 5.4. Heat Recovery Heat Exchanger Costs

Component	Direct Installed Cost
Salt heater	$133 * (HTA)^{0.95}$
Low-pressure steam generator	$66 * (HTA)^{0.95}$
Oil heater	$66 * (HTA)^{0.95}$
Oil-fired feedwater heat exchanger	42 * HTA

HTA = heat exchanger heat transfer area in square feet.

Molten salt transport costs were estimated from designs prepared by PNL for the piping system that interconnects the salt heaters (radiant and convective coolers, gas turbine exhaust heat recovery heat exchanger), salt storage, and salt steam generator. Design information was prepared for three different thermal capacities covering the range of plant configurations being investigated. Regression analysis was applied to the costs estimated at the three design points to develop cost estimating equations as a function of storage charging and steam generator thermal capacity. The resulting estimating equations are presented in Table 5.5.

Molten salt systems have been a key element in the development of solar thermal central receiver power plant technology for many years. Although the technology has not been deployed on a full scale, considerable analysis and pilot-plant testing has been completed. In short, the solar thermal program sponsored by the DOE has spearheaded the development and analysis of molten salt storage, steam generation, and related components. The latest developmental effort has been focused on a design study conducted by Arizona Public Service (U.S. DOE 1988a,b), Pacific Gas and Electric (PG&E) (De Laquil, Kelly, and Egan 1988), and a number of other organizations. The information developed in these studies represents the current state of the art for molten nitrate salt system designs and costs. Therefore, the studies completed by APS and PG&E were used as the reference for developing estimating equations for molten salt storage and molten salt steam generators.

Designs and cost estimates were prepared for storage system capacities ranging from about 100 MWht to 5000 MWht in the studies mentioned above. Multiple hot and cold tanks are required at the upper end of this range, but the maximum allowable size of a single hot or warm tank is subject to debate. Two APS contractors prepared different tank designs and reached different conclusions regarding the maximum permissible size of the hot tank. CBI Industries recommended limiting the capacity of a single hot tank to approximately 1500 MWht, while the designs prepared by Pitt-Des Moines, Inc., included

TABLE 5.5. Molten Salt Transport Costs (mid-1987 dollars)

Component	Direct Installed Cost
Salt heater loop piping	$\$4590 * (MWt_1)^{0.974}$
Steam generator loop piping	$\$4840 * (MWt_2)^{0.974}$

MWt_1 = storage charging thermal capacity.
MWt_2 = steam generator thermal capacity.

a single hot tank with a capacity of 3120 MWht. A single 3120-MWht capacity warm tank was specified by both organizations (U.S. DOE 1988b).

Storage system costs for IGCC power plants with molten salt TES were based on maximum hot and warm tank capacities of 1500 MWht and 3000 MWht, respectively. The IGCC plants with molten salt TES have storage capacity requirements ranging from 3000 MWht to 4000 MWht. This results in multiple hot and warm tanks in approximately a 2:1 ratio. With little difference in individual tank sizes, the same unit cost was presumed to apply for all of the IGCC plants with molten salt TES systems. An average direct installed cost of $11/kWht was established based on estimates prepared by CBI Industries for storage systems with two hot tanks and one warm tank (U.S. DOE 1988b).

A cost estimating equation for molten salt steam generators was also developed from cost data presented in the APS/PG&E studies. Steam generator costs were estimated in those studies for power plants with capacities of 100 MWe, 200 MWe, and 400 MWe. Steam turbine generating capacities for the IGCC/TES power plants investigated in this PNL study range from approximately 230 MWe to 390 MWe. Regression analysis of the APS/PG&E data yielded the following cost estimating equation for molten salt fired steam generators: direct installed cost = 1,516,000 * $(\mathrm{MWe})^{0.426}$.

Oil/rock storage technology has also been extensively developed by various solar thermal programs sponsored by the DOE. Unit capital cost data ($/kWht) presented in Williams et al. (1987) for oil/rock storage systems were revised to reflect an increased temperature range and increased heat capacity of the oil/rock mixture for the increased temperature range for the IGCC/TES applications as compared with the solar thermal applications evaluated in Williams et al. (1987). Revised estimates were developed for four designs covering the range of oil/rock storage capacities encountered in the IGCC/TES plants investigated. Regression analysis of the revised data yielded the following equation for estimating the cost of oil/rock storage: direct installed cost = $318 $(\mathrm{kWht})^{0.682}$.

5.2.3 Indirect Costs, Sales Tax, and Contingency

Indirect construction costs, sales tax, and a contingency were added to the direct construction costs to arrive at the complete "overnight" construction cost (i.e., not including the interest or escalation during construction which were included in the economic methodology). Indirect construction costs were estimated at 25% of direct costs based on the ratio of indirect to direct costs in Jacob and Chu (1988). State and local sales tax was estimated at 3% of the sum of direct and indirect costs based on Drost et al. (1989). Contingency was estimated to be 14% of the sum of direct and indirect costs based on data presented in Jacob and Chu (1988).

5.2.4 Startup, Land, and Working Capital Costs

The total initial capital costs were calculated by adding costs for startup, working capital, and land to the overnight construction cost. The costs for these three elements were derived from cost data and estimating procedures presented in Jacob and Chu (1988). The resulting cost estimating equations, which are a function of overnight construction costs, annual O&M costs, or annual plant power output, are shown in Table 5.6.

5.3 OPERATION AND MAINTENANCE COST ESTIMATES

O&M costs calculated in this study include fuel, operating labor, maintenance labor and materials, overhead, and consumables. Nonfuel O&M costs were split into fixed (constant regardless of plant power output) and variable (proportional to plant power output) elements for the conventional IGCC power plant components. Aggregated nonfuel O&M costs were estimated for heat recovery and energy storage components. The following three sections define the specific estimating approach and results for fuel costs, IGCC component O&M, and heat recovery and energy storage component O&M.

5.3.1 Fuel Costs

Current and future fuel costs were established from projections by several organizations for high-sulfur coal delivered to a utility in the Midwest from midwestern mines. The sources consulted were published by the EPRI (1986), the Energy Information Administration (1988), Data Resources Inc. (1988), the

TABLE 5.6. Startup, Land, and Working Capital Costs (mid-1987 dollars)

Startup
Initial catalysts and chemicals: 0.000762 * annual gas turbine kWh + 0.000053 * annual steam turbine kWh
Prepaid royalties: 0.005 * nonbalance of plant initial capital cost + 0.0036 * balance of plant initial capital cost
Other startup: 0.02 * plant initial capital cost + 1/12 * annual operation and maintenance cost + 126 * plant heat rate, Btu/kWh (cost of 1 week of fuel)
Land
Fixed at $1,950,000
Working Capital
2190 * plant heat rate, Btu/kWh (cost of 4 months of fuel) + 1/6 * annual operation and maintenance cost + 1/12 * annual operation and maintenance labor cost + 0.005 * nonbalance of plant initial capital cost + 0.0036 * balance of plant initial capital cost

Gas Research Institute (Holtberg, Woods, and Ashby 1987), and Wharton Econometric Forecasting Associates (1987). Based on a coal price of $1.50/million Btu in mid-1987, the rate of real (relative to general inflation) price increases predicted by the several sources ranged from 0 to 2%/year. A real price annual escalation of 1% was chosen as the baseline assumption for the economic analysis.

5.3.2 IGCC Component Nonfuel Operation and Maintenance Costs

As noted above, IGCC component nonfuel O&M costs were divided into fixed and variable categories. Fixed costs include operating labor, maintenance labor and materials, and overhead. Variable costs include charges for raw water, catalysts and chemicals, and slag disposal, with a by-product credit for the sale of sulfur.

Data in Jacob and Chu (1988) were used as the basis for establishing IGCC component nonfuel O&M costs. Operating labor cost was set at $3,344,000 for all IGCC/TES plant configurations. Maintenance labor and material costs were estimated by multiplying the initial capital cost of each IGCC component by the fraction indicated in Table 5.7. Forty percent of maintenance labor and materials was presumed to be labor. Overhead costs were estimated to be 30% of O&M labor. Equations for estimating variable costs as a function of annual gas turbine and/or steam turbine power output are shown in Table 5.8.

5.3.3 Heat Recovery and Energy Storage Component Nonfuel Operation and Maintenance Costs

O&M costs for heat recovery and energy storage components were estimated separately from the IGCC power plant components. Aggregated nonfuel O&M cost estimates were made by multiplying the fractions shown in Table 5.9 by the initial capital costs of the individual heat recovery and energy storage components. Heat recovery heat exchanger O&M fractions were based on data in

TABLE 5.7. IGCC Component Maintenance Labor and Material Fractions

Component	Fraction
Coal handling	0.03
Oxygen plant	0.02
Gasifier	0.045
Ash handling	0.045
Radiant cooler	0.03
Convective cooler	0.03
Fuel gas scrubber	0.03
Low-temperature coolers	0.03
Fuel gas saturator	0.02
Acid gas removal	0.02
Sulfur recovery	0.02
Tail gas treating	0.02
Steam condensate	0.015
Gas turbine	0.015
Steam power cycle	0.015
Balance of plant	0.015

TABLE 5.8. IGCC Component Variable O&M Cost Equations

Component	Cost Equation
Raw water	\$0.000533/g.t.[a] kWh + \$0.000698/s.t.[b] kWh
Catalysts and chemicals	\$0.003143/g.t. kWh + \$0.000217/s.t. kWh
Slag disposal	\$0.001577/g.t. kWh
Sulfur credit	\$0.001437/g.t. kWh

(a) g.t. = gas turbine.
(b) s.t. = steam turbine.

TABLE 5.9. Heat Recovery and Energy Storage Component Nonfuel O&M Fractions

Component	Fraction
Heat recovery heat exchangers	0.03
Oil/rock storage	0.024
Molten salt storage	0.0007
Molten salt steam generator	(a)
Molten salt piping	0.016

(a) Fraction = 0.197 * (steam turbine generating capacity, MWe)$^{-0.35}$

Jacob and Chu (1988). Other component O&M fractions were derived from data in the capital cost references cited above for each component.

5.4 LEVELIZED ENERGY COST ESTIMATES

Initial capital cost, annual O&M costs, and annual performance characteristics were combined with the economic methodology and assumptions to produce LEC estimates. LEC estimates were prepared for the six planned generating schedules identified in Table 5.10 for IGCC/TES and two reference power plant systems. The IGCC/TES power plant evaluated in this study has base load and intermediate load power production characteristics. Base load power is

TABLE 5.10. Planned Generating Schedules

Schedule Number	Operating Days/Week	Base Load Hours/Day	Peaking Hours/Day
1	5	24	8
2	5	24	12
3	5	24	16
4	7	24	6
5	7	24	9
6	7	24	12

provided by the gas turbine, which operates continuously, 24 hours per day. Intermediate load power is provided by the steam turbine, which operates from 6 to 16 hours per day, depending on the planned generating schedule. The economic evaluation was conducted by calculating and comparing the LEC of IGCC/TES power plants to reference power plant systems supplying the same mix of base load and intermediate load power output.

Two reference plant systems were evaluated for comparison to the IGCC/TES power plants. Both systems use an IGCC plant to supply the base load portion of the power. A cycling pulverized-coal-fired (PC) power plant is presumed to supply the intermediate load power in the first reference system; a gas-fired combined-cycle (CC) plant is presumed to supply the intermediate load power for the second reference system. The reference system LECs were set equal to the weighted average LEC of the individual LECs calculated for the (IGCC) PC or CC power plants. The weighting was based on the relative amount of base load and intermediate load power produced; approximately 47% of the total annual power output for each production schedule is classified as intermediate. Cost and performance assumptions for PC and CC power plant LEC calculations are shown in Table 5.11. Financial assumptions are the same as for IGCC/TES plants. Gas for the CC plant was presumed to cost $2.25/million Btu in mid-1987 dollars and to escalate at 4%/year in excess of general inflation (see references for initial coal price and future coal price escalation rate).

TABLE 5.11. Reference Power Plant Cost and Economic Assumptions (all costs in mid-1987 dollars)

LEC Input	IGCC Plant	PC Plant	CC Plant
Initial capital, $/kWe	1520.0	1525.0	447.0
Land, $/kWe	3.9	3.9	3.9
Startup, $/kWe	53.5	36.5	36.4
Working capital, $/kWe	82.4	16.7	16.7
Fixed O&M, $/kWe-year	43.1	25.8	6.6
Variable O&M, mills/kWh	2.6	5.9	1.7
Heat rate, Btu/kWh	9322	10192	8394
Availability, %	83.2	71.2	90.3

Sources: Jacob and Chu 1988; Drost et al. 1989; EPRI 1986

LEC results for IGCC/TES, IGCC/PC, and IGCC/CC power plants are shown in Table 5.12 for each of the six planned operating schedules. The results indicate that the LEC for the IGCC/TES plant is less than the LEC for the IGCC/PC power plant, but greater than the LEC for the IGCC/CC power plant for the baseline fuel escalation rates assumed (1%/year real escalation for coal; 4%/year real escalation for gas). If a higher fuel escalation scenario (2%/year real escalation for coal; 6%/year real escalation for gas) is assumed, the IGCC/TES plant LEC is lower than both IGCC/PC and IGCC/CC plants for all generation schedules except the first, as shown in Table 5.13. Also note that the IGCC/TES plant looks best for the planned production schedules with fewer peaking hours per day (i.e., at lower annual capacity factors). The fundamental advantages of the IGCC/TES plant are its reduced capital cost at lower annual capacity factors (where the gasification-related components are downsized the most and, hence, the capital cost benefit of incorporating TES is the greatest), higher availability compared to a PC plant, and a lower heat rate compared to either IGCC or PC plants. A summary comparison of capital cost and performance characteristics is provided in Table 5.14.

TABLE 5.12. Levelized Energy Cost Results; Median Fuel Escalation Rates (mid-1987 levelized $/kWh)

Generation Schedule	Power Plant IGCC/TES	IGCC/PC	IGCC/CC
1	0.0892	0.0996	0.0717
2	0.0758	0.0809	0.0670
3	0.0692	0.0715	0.0647
4	0.0760	0.0911	0.0652
5	0.0656	0.0732	0.0608
6	0.0600	0.0643	0.0585

TABLE 5.13. Levelized Energy Cost Results; High Fuel Escalation Rates (mid-1987 levelized $/kWh)

Generation Schedule	Power Plant IGCC/TES	IGCC/PC	IGCC/CC
1	0.0946	0.1054	0.0934
2	0.0811	0.0867	0.0887
3	0.0745	0.0773	0.0863
4	0.0814	0.0969	0.0869
5	0.0710	0.0791	0.0824
6	0.0653	0.0701	0.0801

5.5 SENSITIVITY STUDIES

Two sensitivity studies were conducted to evaluate the impact of decreases in the cost of two key molten salt components on IGCC/TES plant LEC. The first study examined the effect of reducing the molten salt TES cost by presuming that a less-expensive TES medium could be identified. The second study investigated the potential cost reductions associated with substituting a direct-contact gas turbine exhaust/molten salt heat exchanger for the conventional finned-tube heat exchanger design assumed for the baseline conditions.

The reference direct installed capital cost for the complete molten salt TES system, including salt medium, containment, and ancillary equipment is $11/kWht. A little more than half of this total, or about $6/kWht, represents

TABLE 5.14. Comparative Cost and Performance Data

Generation Schedule	Power Plant IGCC/TES	IGCC	PC	CC
Capital Cost, 1987$/kWe				
1	1556	1660	1582	534
2	1650	1660	1582	534
3	1721	1660	1582	534
4	1419	1660	1582	534
5	1559	1660	1582	534
6	1650	1660	1582	534
Availability, %				
All	83.2	83.2	71.2	90.3
Heat Rate, Btu/kWh				
1	9181	9322	10192	8394
2	9182	9322	10192	8394
3	9175	9322	10192	8394
4	9192	9322	10192	8394
5	9180	9322	10192	8394
6	9182	9322	10192	8394

the cost of the molten salt medium. The impact of reducing TES media costs was examined by presuming alternative media costs of $0/kWht and $3/kWht. For the 5 days per week, 16 hours per day operating schedule, the total salt storage cost, including indirect costs, sales tax, and contingency, is about $49,000,000 for media costs of $6/kWht. If media costs could be reduced to $3/kWht or $0/kWht, the corresponding total TES costs would be about $35,000,000 and $22,000,000, respectively. The plant LEC would be lowered from $0.0692/kWh for the baseline case to $0.0686/kWh and $0.0680/kWh for the alternative cases.

In the baseline IGCC/TES power plant design, the cost for the gas turbine exhaust salt heater is based on a conventional finned-tube design. The second sensitivity study investigated the cost impact of using a direct-contact heat exchanger design instead. This direct-contact design exchanges heat between the molten salt and turbine exhaust gases by direct counter-current contact of the two fluids through a packed tower. A detailed discussion of current experience and theory with direct-contact heat exchanger design and performance is presented in the Appendix.

The direct-contact heat exchanger offers better overall heat transfer between the two fluid streams and a significant reduction in equipment cost compared to the finned-tube heat exchanger design. For the 5 days per week, 16 hours per day production schedule, the direct installed cost of the finned-tube heat exchanger was estimated to be approximately $33,000,000 based on the estimating relationships presented in Foster-Pegg (1986). The estimated direct installed capital cost of the direct-contact heat exchanger, based on three separate sources (Mullet, Corripio, and Evans 1981; Peters and Timmerhaus 1980; Guthrie 1974), was approximately $6,000,000. This reduction in capital cost lowered the LEC from $0.0692/kWh for the baseline case to $0.0668/kWh for the direct-contact heat exchanger design.

The potential cost advantages of using a direct-contact heat exchanger look tremendous. However, some caution is advised. As discussed in the Appendix, there is a great deal of uncertainty in predicting the performance of direct-contact salt/gas heat exchangers, and uncertainty in performance is directly translated to uncertainty in design and cost.

6.0 CONCLUSIONS AND RESEARCH NEEDS

The results of this study have led to a number of conclusions and suggestions for further research. Section 6.1 presents a summary of the conclusions; Section 6.2 describes research needs associated with using TES with an IGCC plant.

6.1 CONCLUSIONS

The significant conclusions from this evaluation of TES for application with IGCC are summarized below:

- Molten salt TES appears technically feasible. While acknowledging that problems exist with certain aspects of salt handling, these problems appear to be resolvable. The overall judgment, both of this study and similar evaluations in the solar thermal area, is that molten nitrate salt TES is technically feasible, and it is reasonable to assume that the technology can be commercialized.

- An IGCC plant with molten salt TES can substantially reduce the cost of coal-fired peak and intermediate load power. The results of this study show that an IGCC plant with molten salt TES produces lower cost peak and intermediate load power than the conventional coal-fired alternative over a range of operating schedules. The LEC of an IGCC/TES plant can be reduced by as much as 20% over the LEC of a conventional plant. This concept produces lower cost power than the natural-gas-fired alternative if significant escalation rates in the price of fuel are assumed.

- Advanced molten salt TES concepts can substantially improve performance and economics. Several advanced concepts such as direct-contact salt heating, low-freezing-point salts, dual storage media, and advanced tank designs have the potential to substantially improve the performance and economics of combining IGCC with TES.

6.2 RESEARCH NEEDS

The results of this study show that TES has substantial promise when used in an IGCC power plant, but additional research, described below, is needed to advance the technology.

- Resolve remaining technical issues associated with molten salt TES. The remaining technical issues associated with molten salt handling need to be resolved and demonstrated in field tests.

- Conduct a more detailed evaluation of using molten salt TES with IGCC technology. The evaluation documented in this report was a scoping study and could not address second-order issues. Before proceeding with a technology development program, a more detailed evaluation should be conducted. This evaluation should include a vendor-developed design and cost estimate.

- Develop advanced molten salt TES technology. Several advanced concepts such as direct-contact salt heating, low-freezing-point salts, dual storage media, and advanced tank designs require research and development. If successful, these concepts have the potential to substantially improve the performance and economics of an already attractive concept.

- Conduct a large-scale field test of molten salt TES. The acceptance of TES technology by the utility industry will depend on a successful large-scale field test of the concept including direct-contact salt heating. A meaningful technology development program must result in such a field test.

7.0 REFERENCES

Babcock and Wilcox. 1978. Steam/Its Generation and Use. The Babcock and Wilcox Company, New York.

Bergan, N. E. 1987. "An External Molten Salt Solar Central Receiver Test. In Proceedings of the ASME-JSMS-JSES Solar Energy Conference, eds. D. Y. Goswami, K. Watanabe, and H. M. Healey, pp. 474-476. The American Society of Mechanical Engineers, New York.

Brown, D. R., J. A. Dirks, M. K. Drost, G. E. Spanner, and T. A. Williams. 1987. An Assessment Methodology for Thermal Energy Storage Evaluation. PNL-6372, Pacific Northwest Laboratory, Richland, Washington.

Carling, R. W. 1983. "Heat Capacities of $NaNO_3$ and KNO_3 from 350 to 800 K." Thermochimica Acta 60(1983):265-275.

Corripio, A. B., K. S. Chrien, and L. B. Evans. 1982. "Estimate Costs of Heat Exchangers and Storage Tanks Via Correlations." Published in Chemical Engineering, January 25, 1982. McGraw Hill Publishing Company, New York.

Data Resources, Inc. 1988. 1988 Energy Review. Lexington, Massachusetts.

De Laquil, III, P., B. D. Kelly, and J. C. Egan. 1988. Solar Central Receiver Technology Advancement for Electric Utility Applications. Report No. 007.2-88.2, Vols. I and II, prepared by Bechtel National, Inc., San Francisco, California, for Pacific Gas and Electric Company, San Ramon, California.

Delameter, W. R. 1987. "The Molten Salt Electric Experiment: A Project Overview." In Proceedings of the ASME-JSME-JSES Solar Energy Conference, eds. D. Y. Goswami, K. Watanabe, and H. M. Healey, pp. 468-478. The American Society of Mechanical Engineers, New York.

Drost, M. K., Z. I. Antoniak, D. R. Brown, and K. Sathyanarayana. 1989. Thermal Energy Storage for Power Generation. PNL-7107, Pacific Northwest Laboratory, Richland, Washington.

Dunlop, W., and R. T. Slambrook. 1986. Capital Cost Estimates and Schedules for Coal-Fired Power Plants. EPRI P-4542, prepared for the Electric Power Research Institute, Palo Alto, California, by Bechtel Group, Inc., San Francisco, California.

Electric Power Research Institute (EPRI). 1986. Technical Assessment Guide. EPRI P-4463-SR, Palo Alto, California.

Energy Information Administration. 1988. Annual Energy Outlook 1987 With Projections to 2000. U.S. Department of Energy, Washington, D.C.

Foster-Pegg, R. W. 1986. "Capital Cost of Gas-Turbine Heat-Recovery Boilers." In Chemical Engineering, July 21, 1986. McGraw-Hill Publishing Company, New York.

General Electric Company. 1979a. Conceptual Design of Thermal Energy Storage Systems for Near-Term Electric Utility Applications. Volume 1: Screening of Concepts. EPRI EM-1037, Vol. 1, Electric Power Research Institute, Palo Alto, California.

General Electric Company. 1979b. Conceptual Design of Thermal Energy Storage Systems for Near-Term Electric Utility Applications. Volume 2: Appendixes - Screening of Concepts. EPRI EM-1037, Vol. 2, Electric Power Research Institute, Palo Alto, California.

General Electric Company. 1979c. Combined Thermal Storage and Transport for Utility Applications. EPRI EM-1175, Electric Power Research Institute, Palo Alto, California.

Guthrie, K. M. 1974. Process Plant Estimating, Evaluation, and Control. Craftsman Book Company of America, Solana Beach, California.

Hallet, R. W., and R. L. Gervais. 1977. Central Receiver Solar Thermal Power System, Phase I, CDRL Item 2, Vol 5. SAN/1108-8/5, McDonnell Douglas Astronautics Company, Huntington Beach, California.

Holtberg, P. D., T. J. Woods, and A. B. Ashby. 1987. 1987 GRI Baseline Projection of U.S. Energy Supply and Demand to 2010. Gas Research Institute, Chicago, Illinois.

Jacob, J. T., and L. A. Chu. 1988. Evaluation of a Texaco Gasification-Combined-Cycle Plant With Kraftwerk Union Gas Turbines. EPRI GS-6160, prepared for the Electric Power Research Institute, Palo Alto, California, by Fluor Engineers, Inc., Irvine, California.

Kays, W. M., and A. L. London. 1964. Compact Heat Exchangers. McGraw-Hill Publishing Company, New York.

Kohl, A. L., J. C. Newcomb, and W. R. Castle. 1987. Evaluation of a Conical Tank for High Temperature Molten Salt Containment. SERI/STR-231-3193, Solar Energy Research Institute, Golden, Colorado.

Kolb, G. J., and U. Nikolai. 1988. Performance Evaluation of Molten Salt Thermal Storage Systems. SAND87-3002, Sandia National Laboratories, Albuquerque, New Mexico.

Kreith, F., and M. S. Bohn. 1986. Principles of Heat Transfer. 4th ed., Harper and Row, New York.

Martin Marietta Corporation. 1978. Conceptual Design of Advanced Central Receiver Power System. DOE/ET/20314-1/2, National Technical Information Service, Springfield, Virginia.

Martin Marietta Corporation. 1984. Alternate Central Receiver Power System, Phase II Volume II - Molten Salt Receiver. SAND81-8192/2, Sandia National Laboratories, Albuquerque, New Mexico.

Martin Marietta Corporation. 1985. Volume 1 - Molten Salt Thermal Energy Storage Subsystem Research Experiment. SAND80-8192, Sandia National Laboratories, Albuquerque, New Mexico.

Matchak, T. A., A. D. Rao, V. Ramanathau, and M. T. Sander. 1984. Cost and Performance for Commercial Applications of Texaco-Based Gasification-Combined-Cycle Plants. EPRI AP-3486, prepared for the Electric Power Research Institute, Palo Alto, California, by Fluor Engineers, Inc., Irvine, California.

Matchak, T. A., and C. H. Lawrence. 1983. Economic Assessment of the Impact of Plant Size on Coal Gasification-Combined-Cycle Plants. EPRI AP-3084, prepared for the Electric Power Research Institute, Palo Alto, California, by Fluor Engineers, Inc., Irvine, California.

Mullet, A., A. B. Corripio, and L. B. Evans. 1981. "Estimate Costs of Distillation and Absorption Towers via Correlations." In Chemical Engineering, December 28, 1981. McGraw Hill Publishing Company, New York.

Peters, M. S., and K. D. Timmerhaus. 1980. Plant Design and Economics for Chemical Engineers. McGraw-Hill Publishing Company, New York.

Ross, K., J. Roland, and J. Bouma. 1982. Solar 100 Conceptual Study. Southern California Edison, Los Angeles, California.

Snyder, W. G., et al. 1986. Planning Data Book for Gasification-Combined-Cycle Plants: Phased Capacity Additions. EPRI AP-4395, prepared for the Electric Power Research Institute, Palo Alto, California by Fluor Engineers, Inc., Irvine, California.

Sorensen, H. A. 1983. Energy Conversion Systems. John Wiley and Sons, New York.

U.S. Department of Energy (DOE). 1988a. Alternate Utility Team Utility Solar Central Receiver Study. Report No. DOE/AL/37841-2, U.S. Department of Energy, Washington, D.C.

U.S. Department of Energy (DOE). 1988b. Arizona Public Service Utility Solar Central Receiver Study. Report No. DOE/AL/37841-1, U.S. Department of Energy, Washington, D.C.

U.S. Energy Association. 1988. U.S. Energy'88: Countdown to the Next Crisis, Second Annual Assessment of U.S. Energy Policy and Prospects. U.S. Energy Association, U.S. Member - Committee of the World Energy Conference, April 1988.

Weber, E. R. 1980. Saquaro Power Plant Solar Repowering Project. DOE/SF/10739-1, Arizona Public Service Company, Phoenix, Arizona.

Wharton Econometric Forecasting Associates. 1987. 1987 Energy Analysis Quarterly. Bala Cynwyd, Pennsylvania.

Williams, T. A., J. A. Dirks, D. R. Brown, M. K. Drost, Z. I. Antoniak, and B. A. Ross. 1987. Characterization of Solar Thermal Concepts for Electricity Generation. PNL-6128, Pacific Northwest Laboratory, Richland, Washington.

Yu, O. S. 1988. "The Potential Impact of Information on Automation Technologies and Load Factor." Presented at Information and Automation Technology: Serving Electric Utility Customers in the 1990s (a conference sponsored by the Electric Power Research Institute, Palo Alto, California) December 5-7, 1988.

APPENDIX

<u>REVIEW AND FEASIBILITY OF DIRECT-CONTACT HEAT EXCHANGE FOR MOLTEN NITRATE SALT THERMAL ENERGY STORAGE</u>

APPENDIX

REVIEW AND FEASIBILITY OF DIRECT-CONTACT HEAT EXCHANGE FOR MOLTEN NITRATE SALT THERMAL ENERGY STORAGE

A.1 INTRODUCTION

The technical and economic feasibility of using a molten nitrate salt thermal energy storage (TES) system in a coal-fired power plant has been examined in a previous report (Drost et al. 1989). The present review examines the technical feasibility of a direct-contact heat exchanger (DCHX) system with nitrate salt as a heat exchange medium for an integrated gasification combined-cycle (IGCC) power plant. The current status of research on DCHXs, future research needs, and a rudimentary conceptual design of a salt heater for the present application are discussed below.

A.2 DIRECT-CONTACT HEAT EXCHANGE

Molten nitrate salts have received considerable attention in the solar energy community both as a heat exchange medium and for TES. Nitrate salt is an excellent sensible heat storage medium because it has a high heat capacity per unit volume, low vapor pressure, and good heat transfer properties. Its abundant availability at low cost and the minimal hazards associated with its use make it attractive for heat exchange and storage applications. A molten salt TES interposed between the gasifier/gas turbine and the steam generator portions of an IGCC plant can provide a means of cycling and producing peak power. Instead of generating steam directly, the heat from the gasifier and the turbine exhaust streams is used to heat the nitrate salt, which is then stored in suitably sized tanks for future use.

Currently, a molten salt heater would use a salt-in-tube design where the gas turbine exhaust passes over finned-tubes containing molten salt. An alternative approach would allow the gas turbine exhaust and molten salt to be in direct contact. In such a system, the fluid streams are brought into direct contact with each other, resulting in improved heat transfer and reduced capital cost. This approach has been widely applied in situations requiring enhanced mass transfer, such as gas separation and flue gas scrubbing; but its application to heat transfer duty has been somewhat limited. There are several reasons for the limited application of DCHX including 1) the fluids must be chemically compatible, 2) separation of the two fluid streams must be easily accomplished after they have been in direct contact, and 3) unlike the standard tubular heat exchangers, there is a lack of design information sufficiently detailed to allow an engineer to confidently design a heat exchange system based on direct contact.

There are several different ways of accomplishing direct-contact heat exchange including spray columns, falling film columns, plate columns, and packed beds. The last method is especially effective where low pressure drop (on the gas side) or low liquid holdup is important and a high volumetric efficiency is needed (Bohn 1989). A packed bed consists of a vertical column randomly filled with rings, saddles, or other packing material. Figure A.1 shows examples of typical packing material. The liquid is distributed over the top of the bed and trickles down the large surface area of the packing material. The gas flows counter to the liquid, and heat exchange takes place primarily at the gas-liquid interface. Except for very high liquid flow rates, the packed bed remains relatively open, and gas-side pressure drop is low. As the liquid rate is increased, a larger fraction of the bed flow area is occupied by the liquid, and the gas-side pressure drop increases rapidly. The amount of the liquid in the column as a fraction of the total volume of the bed is quantified by the liquid holdup, h. It consists of the nonflowing part (the static holdup) and the flowing part (the dynamic holdup). This quantity is critical to the overall design of the system. The factors influencing liquid holdup will be discussed later. The major benefits of using this direct-contact heat exchange include improved heat transfer, low first costs, low operating costs, and its applicability to corrosive or fouling fluids and to very high temperature fluids.

A direct-contact molten salt air heater (Bohn 1989) has been tested for use on a laboratory scale in solar heat recovery. This concept is the basis

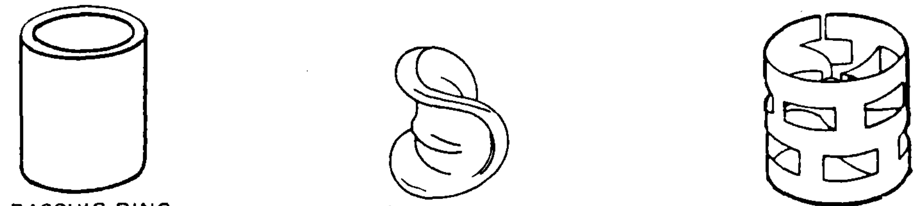

FIGURE A.1. Typical Packings for a Direct-Contact Molten Salt Heat Exchanger

for the direct-contact salt heater discussed in this Appendix. The major difference in the design of the system is that instead of having the molten salt heat the incoming air (as in the solar heat recovery system), the exhaust gases (air) would be used to heat the molten salt from the cold TES tank. The heated salt would then be stored in a separate tank for power cycling purposes. Additional heat recovery from the exhaust gas streams below about 316°C (600°F) would have to be accomplished by generating low-pressure process steam and/or a second TES system using an inexpensive oil/rock technology.

A.3 LITERATURE REVIEW

A review of the existing literature on DCHXs identified a number of investigations by several authors (Bemer and Kalis 1978; Bohn 1985; Bravo et al. 1986; Bohn 1987; Buchanan 1988; Bohn 1989; Huang and Fair 1989). The literature review was limited to packed bed designs for the reasons given in Section A.2, and it showed that both pressure drop and heat transfer model calculations for a packed bed have been developed and verified against the limited experimental data that are available. A discussion of the various considerations and the final correlations for the liquid holdup, the gas pressure drop, and the direct-contact heat transfer are given in the following paragraphs.

A.3.1 Liquid Holdup

The pressure drop and the flooding point in irrigated packed beds are estimated with empirical correlations by relating them to the liquid holdup in the bed (Buchanan 1967, 1969; Bemer and Kalis 1978). One of the more general forms of the empirical equations (Buchanan 1967) combined the liquid holdup considerations in the gravity-viscosity regime (where the film number is the controlling parameter) and the gravity-inertia regime (Froude number being the controlling parameter) in the form:

$$h = A\,Fi^{1/3} + B\,Fr^{1/2} \qquad (1)$$

where Fi = film number for the liquid (= Fr/Re_L)

Fr = Froude number for the liquid (= $U_L^2/d_{eq}\,g$)

Re_L = Reynolds number for the liquid (= $d_{eq}\,U_L\,\rho_L/\mu_L$)

d_{eq} = packing equivalent diameter

U_L = liquid velocity

ρ_L = liquid density

μ_L = liquid viscosity

g = acceleration due to gravity.

The coefficients A and B were established to be 2.2 and 1.8, respectively, for Raschig rings.

Bemer and Kalis (1978) proposed a simpler and a more convincing model by assuming that the film flow was affected only by friction and that the friction factor approaches a constant value for large Reynolds numbers. Their equation was of the form:

$$h = A\,Fr^{1/3} \qquad (2)$$

However, both models assumed a constant wetted area of the bed in considering the effect of the Froude number and assume that neither the liquid viscosity nor the liquid flow rate had any effect on it. It appears that, to take account for the influence of the liquid flow rate on wetted area, the exponent on the Froude number should be greater than 1/3. With that in mind, Buchanan (1988) proposed a more flexible holdup equation:

$$h = A\,Fi^{n} + B\,Fr^{m} \qquad (3)$$

where the exponents n and m are each expected to be somewhat larger than 1/3, and the values of A, B, n, and m vary with the size and shape of the particular packing material used (see Figure A.1). For example, the final equation for ceramic Raschig rings (Buchanan 1988) is

$$h = 9.25\ Fi^{0.48} + 0.805\ Fr^{0.36} \tag{4}$$

which gave the best correlation with the experimental data.

A.3.2 Pressure Drop

Following Bemer and Kalis (1978) and Bravo, Rocha, and Fair (1986), the pressure drop through beds of random packing materials is given by Equation (5):

$$\Delta P = \Delta P_o\ (1 - Kh)^{-5} \tag{5}$$

per unit length of packed bed,

where ΔP_o = dry bed pressure drop per unit length
h = total liquid holdup from Equation (4)
K = constant (characteristic of the packing size and shape, if different from Raschig rings).

The dry bed pressure drop, ΔP_o, is caused by frictional resistance of gas flow with no liquid present in the bed. This may, therefore, be given in terms of a conventional Fanning or Darcy-type relationship (Bravo, Rocha, and Fair 1986) as

$$\Delta P_o = (f\ \rho_g\ U_{ge}^2)/(d_{eq}\ g_c) \tag{6}$$

where f = friction factor (= $C_1 + C_2/Re_g$)
Re_g = gas Reynolds number (= $d_{eq}\ U_{ge}\ \rho_g/\mu_g$)
ρ_g = gas density
U_{ge} = effective gas velocity inside the flow channel (= $U_{gs}/\epsilon \sin \theta$)
U_{gs} = superficial gas velocity

ϵ = packing void fraction

θ = angle of inclination of flow channel from the horizontal

g_c = conversion factor (unity in SI system)

μ_g = gas viscosity.

For the packing materials studied by Bravo, Rocha and Fair (1986), the friction factor expression was given by the following equation:

$$f = 0.171 + 92.7/Re_g \quad (7)$$

Therefore, for the metallic Pall rings packing, for example, the final expression for the pressure drop will be

$$\Delta P = (0.171 + 92.7/Re_g)\ (\rho_g\ U_{ge}^2/d_{eq}\ g_c)\ (1 - Kh)^{-5} \quad (8)$$

with the constant, K, assumed to be **0.75**, thus resulting in a lower pressure drop than for the Raschig rings.

A.3.3 Heat Transfer

The heat transfer modeling of irrigated packed beds has traditionally relied on mass transfer correlations and the analogy between heat and mass transfer. However, the mass transfer analogy has consistently underpredicted the experimentally determined heat transfer rates. The reason for this is attributed to heat transfer by conduction in the packing material (from dry to wet areas), for which there is no equivalence in mass transfer correlations (Huang and Fair 1989). Also, empirical correlations derived from the same mass transfer analogy by different investigators have had large discrepancies in determining the overall heat transfer coefficient, sometimes even by a factor of four or more.

In an attempt to overcome these limitations, each heat transfer mechanism in a packed bed has been individually modeled, rather than lumping them all into an overall heat transfer coefficient (Bohn 1987). The convection heat transfer was determined from a mass transfer correlation and a liquid holdup correlation, while conduction in the packing elements and convection from the dry portions of the bed were based on a model of conduction through and convection from a fin. The radiation terms were also included in the model but in the final analysis were found to be unimportant relative to the convective heat transfer terms, which is consistent with previously obtained data for high-temperature liquid metal systems. The conduction correction term is also quite small because the wetted areas are a large fraction (80% to 90%) of the total packing area, but it nevertheless improves the agreement between the predictions and the data of Huang and Fair (1989). Furthermore, these correlations have been thoroughly verified and are based on a wide range of packing sizes, so it is expected that this model of Bohn (1987) would apply to commercial-size heat exchangers as long as the liquid used wets the packing material. It should be noted, however, that even though the aforementioned mass transfer analogy models exhibit a qualitative agreement with the available data, the slopes of the experimental data are quite different, suggesting a more fundamental limitation with using mass transfer data to predict heat transfer for packed beds.

Bohn (1989) used a packed column filled with stainless steel Pall rings, 15.9 mm in diameter and height, for a total bed height of 610 mm and a void fraction of 0.947. The main test series involving the simultaneous measurement of heat transfer and column differential pressure were conducted at a single salt temperature of 400°C (752°F), salt (liquid) flow rates (L) of 6 to 18 kg/m^2-s and air (gas) rates (G) of 0.3 to 1.2 kg/m^2-s. The resulting net salt heat transfer ranged from 1000 to 4500 W, and the column differential pressure ranged from 163 to 1225 Pa/m. The general operating condition for a packed column is at 400 Pa/m, with 1000 Pa/m being a safe upper limit to avoid column flooding. Based on the correlation of data (Bohn 1989), the net salt heat transfer, Q_s, is given by

$$Q_s \sim G^{0.92} \qquad (9)$$

with no discernible dependence on the liquid flow rate (L). At a liquid flow rate of 17 kg/m^2-s, the column seemed to operate in one of two modes (with poor data repeatability) with significantly different heat transfer rates in the gas rate range of 0.35 to 0.55 kg/m^2-s. But for $G > 0.6$ kg/m^2-s, the column seemed to settle back in a single but lower heat transfer mode. The condition of flooding was indicated when the heat transfer ceased to increase beyond a gas rate of approximately 1.0 kg/m^2-s at a liquid rate of 6.6 and 12.3 kg/m^2-s.

The column differential pressure, on the other hand, is related to gas flow rate through

$$\Delta P \sim G^{1.67} \tag{10}$$

with no significant change in the exponent even as L was increased. For a given value of G, as the value of L was increased from 0 to 6 kg/m^2-s, there was a large increase in ΔP, while any further increases in L produced only relatively small increases in ΔP.

A.4 FUTURE RESEARCH NEEDS

The packed column has been used as an efficient device for transferring mass in gas-liquid contacting operations in the chemical industry. There has also been a long history of development of prediction methods for the pressure drop, flooding, and mass transfer efficiency of packed columns; and yet they cannot be considered fundamental and reliable. There is a fair amount of empiricism associated with the performance evaluation of the columns, and predicting the heat transfer performance based on heat/mass transfer analogy introduces even more. Nevertheless, extending the large amount of available mass transfer data to applications that pertain primarily to heat transfer and developing correlations on the basis of that are the only legitimate methods presently being used for a DCHX system analysis. Hence, there is an urgent need for a fundamental and generalized modeling of the heat exchange performance in packed beds.

The modeling would have to be a combination of more detailed experimentation (involving different fluids, different packing materials, a range of flow rates and temperatures, and both small- and large-scale testing) and an analytical/numerical solution scheme of the basic conservation equations.

Particular emphasis should be placed on testing the molten salt-air system, especially in a large-scale field test unit, to expand the existing database and to offer a stronger case for commercialization of such a system. Alternative molten salts may be less expensive than the current nitrate salt and might have a lower freezing temperature. The lower freezing temperature reduces the need for heat-tracing molten salt components and can decrease the cost of the TES subsystem by decreasing the amount of salt needed for storage. Some alternative salts have been investigated for solar applications, and the results are promising, though a number of practical issues must be resolved (Bradshaw and Tyner 1988). Also relevant is the continual introduction of new packing materials with claims of improved performance but not supported by objective performance data. This would be especially important if pressure drops could be reduced and at the same time the DCHX performance improved in the new beds.

The second major research issue is the impact of the flue gas on the chemical composition of the molten salt. While the exhaust of a natural-gas-fired turbine is very clean, particularly for particulate, it is possible that the trace chemicals in the exhaust may react with the molten salt or be absorbed in the salt and buildup over time. Salt/flue gas compatibility tests will have to be conducted to ensure the success of the concept.

There are a number of secondary issues that are also important in evaluating the performance of a DCHX system. The measurement of salt carryover rates out of the bed has not been attempted. Also, a more detailed study of the degradation of the various heat transfer salts from contact with air is important. The degradation issue is especially critical in the case of the power industry because the quality of the primary heat transfer medium (the salt) is to be maintained throughout the life of the system. The effect of radiation resulting in thermal backmixing thereby reducing the heat exchange effectiveness may still be important at higher operating temperatures, for which more data and analysis are necessary. In the context of higher temperature applications, the common nitrate salts decompose at temperatures above

600°C (1112°F). Therefore, research is ongoing to identify salts and compatible containment materials for temperatures above 600°C (1112°F). For temperatures up to 800°C (1472°F), the DCHX consists of an internally insulated, carbon-steel column using a high-purity (99% alumina) packing. If the packing and the associated insulation costs get too high, as can be expected, a spray column DCHX may have to be considered for high-temperature applications.

A number of other applications involving TES may be considered in the future, in addition to the conventional and IGCC power plants. For independent power production methods, including cogeneration alternatives and other energy conservation methods, the efficient storage of useful thermal energy for later use in producing supplementary power or process steam is an extremely appropriate concept for integration. Therefore, an inexpensive and efficient way of transferring heat from the exhaust gas streams using a DCHX system is very important for several different TES applications.

A.5 CONCEPTUAL DESIGN OF A DCHX SYSTEM

Based on the heat transfer achieved by the small test beds that have been investigated so far, the bed volume may be scaled up to provide the necessary thermal output. Assuming that a total heat transfer equivalent of 128 MWt is required, the minimum packed bed volume required would be 450 m^3 (15,450 ft^3), which may be provided by two equal sized beds of 5.64 m (18.4 ft) diameter and 9.0-m (29.4-ft) height. The beds would be filled with metallic 0.0254-m (1-in.) Pall rings in a random packing mode. The bed containers would be made of 304H stainless steel (wall thickness of 0.0254 m) as shown in Figure A.2. Because the scaled-up tanks have not been field-tested for their heat transfer duties and because very limited data are available on small experimental beds, no extensive designing of the packed bed has been possible. This lack of information is an additional motivation for conducting scale-up testing and feasibility analyses as quickly as possible.

With the estimated maximum pressure drop of 1000 Pa/m for the two beds of 9.0-m (29.4-ft) height each, the total pressure drop would be 18 kPa

(2.61 psia). The salt (liquid) flow rate would range from 10 to 20 kg/m^2-s, depending on the duty cycling, while the corresponding air (gas) flow rates would range from 10.8 to 21 kg/m^2-s. Because the two beds would be connected in series (Figure A.2), the salt flow stream at 288°C (550°F) would enter sequentially from the colder (C) to the hotter (H) bed, while the exhaust gas at approximately 566°C (1050°F) would flow from the hotter to the colder bed. The total temperature rise in the salt would be about 204°C (400°F), while the air temperature would decrease by 212°C (414°F). The series design of the packed beds would also provide adequate "residence time" for the two streams to contact each other and achieve an effective heat exchange. But, with a pressure drop of 20 kPa (2.9 psi) across the two packed beds, the exhaust stream from the turbine would have to be at 120 kPa (17.4 psi) if the final exit pressure of the gas is assumed to be 100 kPa (14.5 psi). The curtailed expansion in the turbine would reduce its net power output by almost 10% and would also force the exhaust to be at a higher temperature than if it were completely expanded. Most of this increased enthalpy of the exhaust gas stream could be recovered in the packed beds. Nevertheless, the penalty on the turbine power appears to be excessive and necessitates consideration of alternate configurations for the packed beds.

If, instead of having the two packed beds in series, they were connected to each other in a parallel configuration (Figure A.3), the total pressure drop would be decreased and the flow rate across each bed would now be halved. The total pressure drop across each bed is estimated to be approximately 9 kPa (1.31 psi), while the two flow streams would have to be divided into two equal streams for the two beds. The total heat exchange (and the temperature rise of the salt) would now have to occur within each of the two beds. The reduced flow rates of the gas and salt streams would also reduce the velocities through the bed; and, therefore, the pressure drop would remain well below the projected value. This latter effect would decrease the pumping power required for the liquid stream and would reduce the gas turbine output by less than a 5%. The reduced velocities of the two streams will also increase the "residence time" even further and improve the heat exchange characteristics, compared with the series configuration.

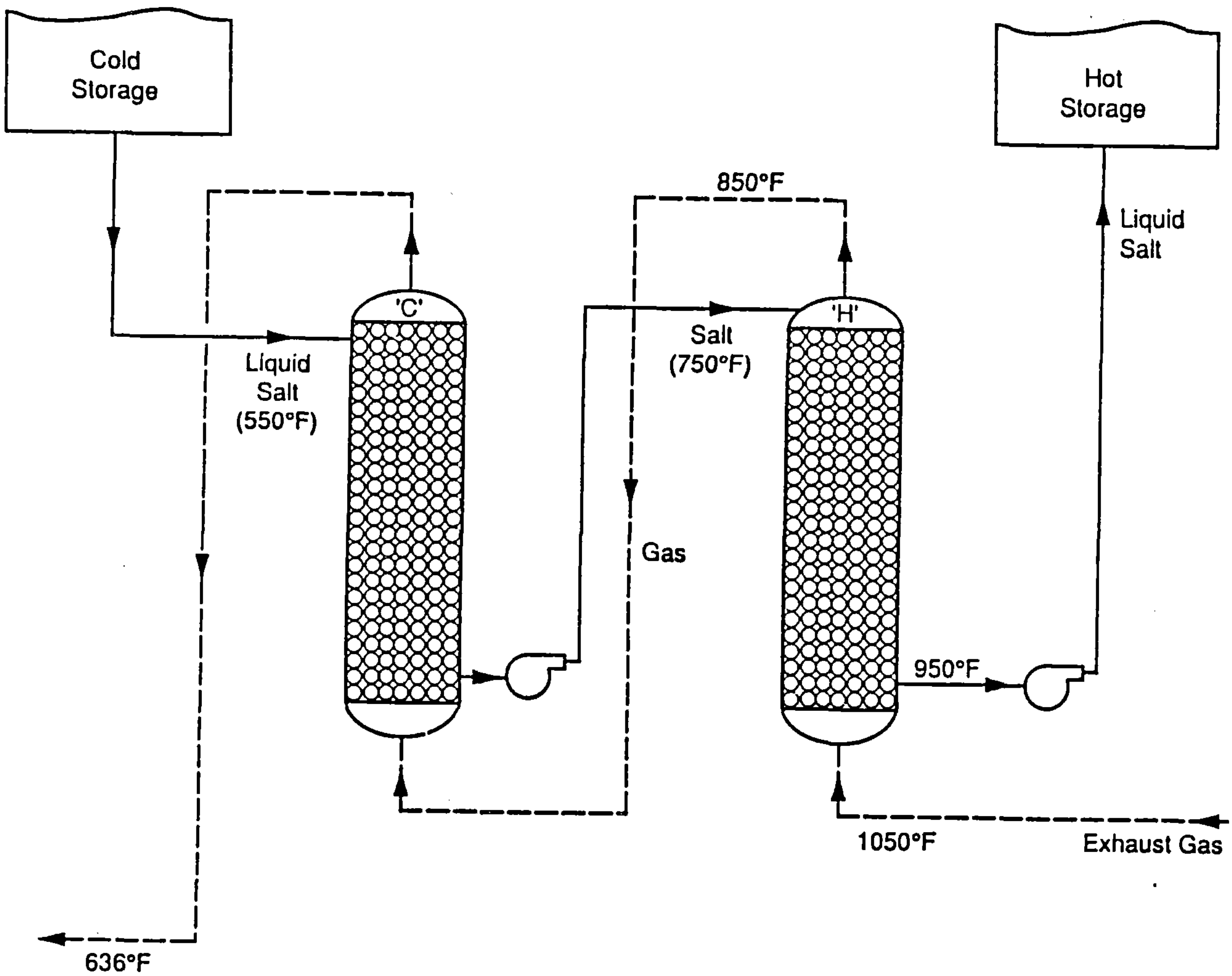

FIGURE A.2. Conceptual Design of Molten Salt Direct Contact Heat Exchanger

If an even smaller pressure drop were desired [say, 5 kPa (0.66 psi)] across each bed, the bed capacity can be maintained by decreasing the height of the bed and dividing the bed into four smaller beds arranged for parallel flow. The gas turbine power penalty would then be reduced to about 2.5%, and the heat exchange characteristics would be enhanced even further. These preliminary calculations show that the molten salt DCHX can impact the gas turbine's performance but that there is substantial flexibility in the design of the DCHX that allows the designer to minimize the impact.

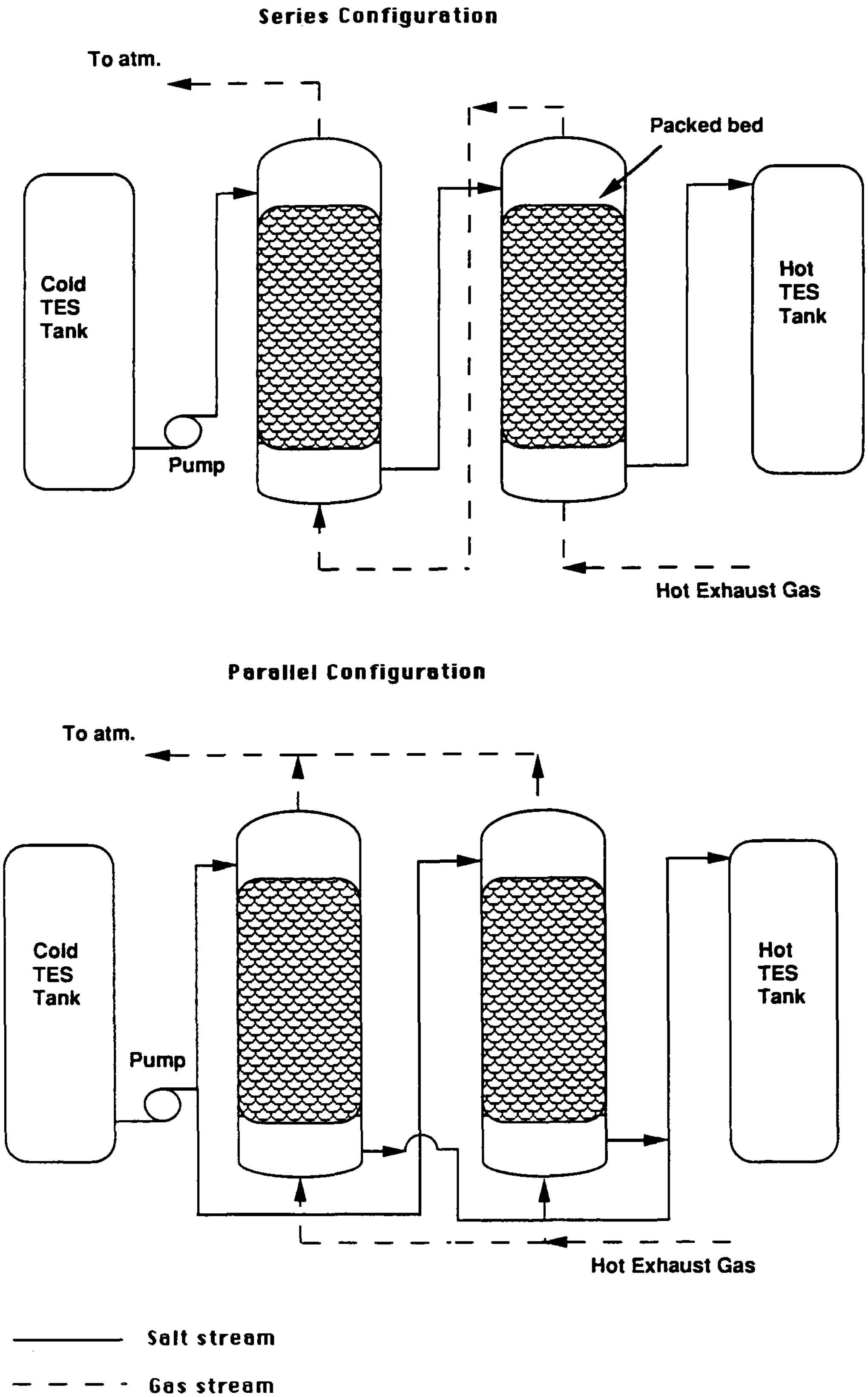

FIGURE A.3. Alternative Configurations for Direct-Contact Heat Exchanger

A.6 CONCLUSIONS

The current status of research on DCHX systems shows that only preliminary modeling and experiments have been performed to quantify the heat transfer aspects of packed beds. A more fundamental analysis and extensive experimentation with larger beds and over wider ranges of temperatures and flow rates have to be performed before any commercialization of a DCHX system for thermal storage applications can be achieved. However, there is sufficient evidence and experience at the present time to indicate the feasibility of the concept; therefore, further investigation of a DCHX system using molten salts for thermal energy storage application in a power plant is warranted.

A.7 REFERENCES

Bemer, G. G., and G. A. J. Kalis. 1978. "A New Method to Predict Hold-up and Pressure Drop in Packed Columns." Trans. Instn. Chem. Engrs. 56:200-204.

Bohn, M. S. 1985. "Air/Molten Salt Direct Contact Heat Transfer." J. Solar Energy Engineering 107:208-214.

Bohn, M. S. 1987. "Analytical Model of an Irrigated Packed-Bed Direct Contact Heat Exchanger at High Temperature." Proceedings of the 2nd ASME-JSME Thermal Engineering Conference, American Society of Mechanical Engineers, New York.

Bohn, M. S. 1989. "Heat Transfer and Pressure Drop Measurements in an Air/Molten Salt Direct-Contact Heat Exchanger." Proceedings ASME 1989 Annual Solar Energy Division Conference, American Society of Mechanical Engineers, New York.

Bradshaw, R. W., and C. E. Tyner. 1988. Chemical and Engineering Factors Affecting Solar Central Receiver Applications of Ternary Molten Salts. SAND88-8686, Sandia National Laboratories, Albuquerque, New Mexico.

Bravo, J. L., J. A. Rocha, and J. R. Fair. 1986. "Pressure Drop in Structured Packings." Hydrocarbon Processing, March 1986, pp. 45-49.

Buchanan, J. E. 1967. "Holdup in Irrigated Ring-Packed Towers Below the Loading Point." Ind. Eng. Chem. Fundam. 6:400.

Buchanan, J. E. 1969. "Pressure Gradient and Liquid Holdup in Irrigated Packed Towers." Ind. Eng. Chem. Fundam. 8:502-511.

Buchanan, J. E. 1988. "Operating Holdup on Film-Type Packings." AIChE Journal 34(5):870-872.

Drost, M. K., Z. I. Antoniak, D. R. Brown, and K. Sathyanarayana. 1989. Thermal Energy Storage for Power Generation. PNL-7107, Pacific Northwest Laboratory, Richland, Washington.

Huang, C., and J. R. Fair. 1989. "Direct-Contact Gas-Liquid Heat Transfer in a Packed Column." Heat Transfer Engineering 10(2):19-28.

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