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Pipeline Bottoming Cycle Study

Final Report

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June 1980

Prepared for:
U.S. Department of Energy
Assistant Secretary for Conservation
and Solar Energy
Office of Transportation Programs

Under Contract No. AC03-77CS51381

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Washington, D.C. 20585

Prepared by:
General Electric Company
Cincinnati, Ohio 45215
Under Contract No. AC03-77CS51381

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SECTION 1

SUMMARY

The General Electric Company has conducted a program to study the technical and economic feasibility of applying bottoming cycles to the prime movers that drive the compressors of natural gas pipelines. These bottoming cycles convert some of the waste heat from the exhaust gas of the prime movers into shaft power and conserve gas.

Three typical compressor station sites were selected, each on a different pipeline. Although the prime movers were different, they were similar enough in exhaust gas flow rate and temperature that a single bottoming cycle system could be designed, with some modifications, for all three sites. Preliminary design included selection of the bottoming cycle working fluid, optimization of the cycle, and design of the components, such as turbine, vapor generator and condensers. Installation drawings were made and hardware and installation costs were estimated.

Assessments were made in the areas of economics, technological feasibility, environmental and safety, operational reliability and maintainability. Depending on the projected gas prices and the investment criteria used, bottoming cycles can be a very attractive investment. The results of the economic assessment of retrofitting bottoming cycle systems on the three selected sites indicated that profitability was strongly dependent upon the site-specific installation costs, how the energy was used and the yearly utilization of the apparatus. The amount and the worth of the fuel savings were found to be highly variable depending on the mode of station operation. The economics of all applications studied were highly dependent on the projected cost of the natural gas, as well as the rules used to allocate gas costs to pumping use. The study indicated that the bottoming cycles are a competitive investment alternative for certain applications for the pipeline industry, with some applications showing the system to be economically viable right now or in the very near future. The

bottoming cycles will be economic for more applications in the future with some changes in the regulations.

Bottoming cycles are technically feasible; they are being developed in smaller sizes and most of the component equipment is available from commercial manufacturers. The choice of a flammable working fluid was considered carefully and it was concluded that proper design and operating practices would reduce the environmental and safety hazards to acceptable levels. The reliability and maintainability of the bottoming cycle are comparable to conventional power plants, such as the STeam and Gas Turbine (STAG*) plant.

An industrial potential assessment was made to estimate the amount of gas that could be saved through the year 2000 by the adoption of bottoming cycles for two different supply projections. The estimates varied from 51 million barrels of oil equivalent (BOE) (0.296 trillion cubic ft) for a low supply projection to 126 million BOE (0.734 trillion cubic ft) for a high supply projection. The potential market for bottoming cycle equipment varied from 530,000 to 1,334,000 horsepower for the two supply projections or from 170 to 500 units of varying size.

Finally, a program plan was developed that will permit the demonstration of bottoming cycles on gas pipelines in a timely and expeditious manner.

* Trademark of General Electric Co.

SECTION 2

INTRODUCTION

The Department of Energy (DOE) is exploring many means for saving energy and precious fuels in this country. In the category of nonhighway transportation, there are four classes of transportation for which energy savings are sought, namely airlines, marine, pipelines and rail. The DOE also has under study organic Rankine bottoming cycles for waste heat recovery in the power range from about 600 to 800 HP. These organic Rankine cycle systems promise to pay dividends in the use of waste heat through the generation of electricity. Since this concept seemed to be viable economically, it was the desire of the Department of Energy to determine whether the same kind of system would economically save energy on the nation's pipelines.

The Research and Development Program goal of the Department of Energy has two facets with respect to the pipelines. First, to achieve maximum energy efficiency and second, to achieve optimized pipeline applications. With these goals in mind, the Department of Energy set out to demonstrate the viability of a bottoming cycle system, and to measure the energy savings obtainable from such a system. As the first part of this research, development and demonstration (R, D&D) program, the Department of Energy solicited a contract on the Pipeline Bottoming Cycle Study. The objectives of this program included the selection of a potential demonstration site on the nation's pipelines; the development of a preliminary design of the bottoming cycle for the selected site, and the preparation of assessments of the bottoming cycle system in the following areas: economics, environment and safety, technological feasibility, and operational reliability and maintainability. Additional objectives were to provide an assessment of the potential for industry utilization of pipeline bottoming cycles including an estimate of the potential fuel savings to the year 2000, and finally to develop and recommend a demonstration program plan for applying bottoming cycles to pipeline prime movers. The General Electric Company was awarded a contract on September 15, 1977 to study the technical and economic feasibility of bottoming cycles on gas pipeline prime movers. This report describes the study on the above mentioned tasks of the present phase of the program.

Initially, three sites were selected as appropriate applications for the potential demonstration of the bottoming cycle technology. The Department of Energy established three general goals in the selection of these sites: the site should have broad applicability for the new and retrofit applications leading potentially to large domestic fuel savings; the site should have a good potential for near term implementation and the site should provide a substantial percentage reduction in heat rate of the prime mover to which the bottoming cycle is applied. All three sites selected met the above criteria; but each site had a difference in the installation method. Hence, it was decided to include all three sites and to explore the various types of installation at the sites to utilize the power generated by the Rankine bottoming cycle. The three sites were on three different pipelines and the pipeline companies were willing to participate in the study program because of the potential significant amounts of the gas savings resulting from the organic Rankine bottoming cycle apparatus.

Preliminary designs of the bottoming cycle systems were carried out for the selected sites. This included preliminary design of the components such as turbines and the heat exchangers and the preliminary design of the installation of the system at the specific sites. The installation designs and the equipment layouts for the specific sites were carried out in cooperation with the pipeline companies. A consultant was then utilized to compare these installation costs on a common basis. Vendor quotes were obtained to evaluate the bottoming cycle equipment costs. The total cost for a specific site was obtained by combining the equipment and the installation costs.

Based on the three site installations, four assessments have been made in order to determine the viability and commercialization potential of the Rankine bottoming cycle as applied to pipeline compressor stations. These assessments were: economic; environmental and safety; technological feasibility; and operational, reliability and maintainability.

In the environmental and safety assessment task of the bottoming cycles study, the equipment and the system design were reviewed for compatibility with federal, state and local codes governing the selected potential demonstration sites; regulations and codes were examined for possible conflicts and restrictions; and potential design and operational problems were discussed. The three pipeline companies cooperated in obtaining the data on the compliance of the system with the state and local codes by conducting the surveys among their operating engineering departments. The technological feasibility study was conducted by reviewing the existing bottoming cycle systems for other applications and a design review was held to identify the potential problems. The design review team consisted of experts from the appropriate departments in the General Electric Company. In the reliability and maintainability task of the program, potential reliability problems with the proposed bottoming cycle designs were evaluated and the status of research on these problems were presented. The general overall system acceptability from the reliability and maintainability standpoints were examined and the estimated maintenance requirements for the bottoming cycle system were presented. The severity of new maintenance problems and site-sensitive problems was discussed.

The economic assessment included analyses of the costs of the system compared with the economic benefits. System capacity, initial cost, and expected energy savings over the life of the system were calculated. The economic attractiveness of retrofitting the Rankine bottoming cycle power plant equipment to the gas turbines located on the pipeline pumping stations was shown by analyzing the return on capital investment by the incremental cost of service approach. The incremental cost of service approach was suggested by the participating pipeline companies in the bottoming cycle study program as the most representative cost analysis approach for assessing the merits of new investments. The "average" and "new" gas prices have been used to calculate the dollars saved resulting from the fuel savings by using the bottoming cycle equipment. The financial information required to make the cost of service computations was provided by the respective pipeline company for the installation of the bottoming cycle equipment at the specific site. Both the first year cost of service and multiple years cost of service computations were made. The investment criterion used by two of the participating companies was the break-even first year cost of service. The third pipeline company suggested a fifteen year profitability as the financial criterion for the capital investment. Finally, sensitivity analyses were conducted to evaluate the effects of the cost of the fuel used, the cost of the installed system, fuel escalation rates, investment tax credits, rate of return on invested capital and other pertinent system and economic factors, such as the effect of government regulations in performing the economic assessment.

Following the four assessments, an industrial potential assessment was made to estimate the amount of gas that could be saved through the use of the bottoming cycles. These estimates were made for four different projections, two for each of two supply projections for the natural gas. The potential market for the bottoming cycle equipment was also estimated through the year 2000 under the four projections.

Finally, a program plan was developed to make the bottoming cycle system demonstration possible in a period of 48 months. The logical progression of effort leading to the demonstration of the system at a pipeline compressor station includes three phases: directed or related studies; detailed design, fabrication and installation of the bottoming cycle system at a demonstration site; and the system start-up, operation and data acquisition and analysis. The work statement for the entire program was included with the cost estimates and the schedules.

SECTION 3

POTENTIAL DEMONSTRATION SITE SELECTION

3.1 INTRODUCTION

The purpose of this task is to select sites that are typical of the gas pipeline industry for which bottoming cycles will be designed and evaluated. The first step was to prepare and submit to the Department of Energy a site selection methodology plan. Following that and using the resources of the General Electric Company including the Industrial Sales Division, which sells gas turbines to the pipeline industry, a survey was made of the prime movers installed on domestic natural gas pipelines. The purpose of the survey was to determine what was typical relative to prime movers installed on the nation's gas pipelines. Then the sites available to GE from committed pipeline companies were analyzed. The following gas pipeline companies have submitted letters of commitment to the Contractor*:

- Columbia Gulf Transmission Company through Research Department of The Columbia Gas System Service Corporation,
- Pacific Gas and Electric Company, including Pacific Gas Transmission Company, and
- Texas Gas Transmission Corporation.

The analyses of the sites of the committed gas pipeline companies led to the four sites which were recommended to the DOE and which met the following goals:

- The site should have broad applicability for new and retrofit applications, leading potentially to a large fuel saving within the domestic pipeline industry.

- The site should have a good potential for near term implementation and thus represent a favorable cost/benefit ratio, and

*Said Letters are in Appendix A of this Report.

- The site should provide for a substantial percentage reduction in the heat rate of the prime mover to which the bottoming cycle is applied, with a target value of 20%.

In this section the criteria which were used to make the site selections are discussed. This is followed by a review of the survey data on the prime movers installed on domestic gas pipelines. Then available sites from the committed pipeline companies are presented along with the calculations which provide the data from which the recommendations were made. Descriptions of the recommended sites and the final site selections conclude this section.

3.2 SELECTION CRITERIA

Shown in Table 3-1 are the selection criteria which are delineated in the contract for this phase of the Department of Energy Pipeline Bottoming Cycle Program. Comments on some of these criteria are pertinent at this time. Item 1, type of pipeline: as was documented in Reference 1, liquid pipelines which carry either crude oil or petroleum products are in large part driven by electric motors. For this reason, liquid pipelines were excluded from the selection process and only gas pipelines were considered. Item 2, the type of engine: in the selection process reciprocating engines of both the supercharged and naturally aspirated types were considered; both recuperated and simple-cycle gas turbines were also considered. Item 3, system operating parameters and duty cycle: this item was generally characterized by the number of hours per year the site was utilized in 1976 as reported by the pipeline company. Item 4, site location: the site should be a reasonable distance from a commercial airport so that interested parties could tour the demonstration site conveniently. Item 5, owner interest in project (potential for cost participation): a successful demonstration would provide the site owner with apparatus which saved fuel and was economic. With appropriate contractual protection against reduced performance and reduced economic benefits the pipeline company should be willing to contribute to the purchase and installation of the demonstration apparatus. Item 6, severity of problems (disruptions) posed by the demonstration: the control over the points in time when the bottoming cycle apparatus requires the shut down of the

TABLE 3-1

SELECTION CRITERIA

1. Type of Pipeline
2. Type of Engine
3. System Operating Parameters and Duty Cycle
4. Site Location
5. Owner Interest in Project (Potential for Cost Participation)
6. Severity of Problems (Disruptions) Posed by Demonstration
7. Time Frame That Project Could be Initiated
8. Potential for Utilization of Heat Recovered

prime mover for installation or connect up will of necessity have to be retained by the pipeline company and the impact to the program would have to be assessed. Item 7, time frame that project could be initiated: considered must be whether the bottoming cycle is to be used to increase site power coincident with a planned increase in throughput or whether the need for the power depends upon the approval of some new project by the Federal Energy Regulatory Commission (FERC). Finally, Item 8, potential for utilization of heat recovery: two pipelines committed to this program have indicated that the generation of electricity to be put on the grid would be an attractive way to use the bottoming cycle. This would permit utilizing the extra horsepower which the site would produce and it would supplant new equipment purchases for electrical generating equipment on the power generation system. However, this situation is not typical of the vast majority of domestic pipeline companies and for that reason this type of utilization of the bottoming cycle apparatus was not considered in the analysis of available sites. In general, the pipeline companies are only expanding in selected areas where new supplies of gas are becoming available. A large number of compressor stations are operating at only a fraction of the installed power because the gas through-put has been severely limited. For this reason, it has been assumed that no additional horsepower is required at a particular compressor station and that the additional horsepower obtained from the bottoming cycle must displace some other source of power at the compressor site so as to result in a reduction in fuel consumption.

In order to make the four initial site selections, it was necessary to become more definitive with respect to the relative characteristics of the numerous pipeline pumping sites that are available through pipeline companies committed to this program. Shown in Table 3-2 are some additional criteria which were applied in the selection of sites under this contract. It will be noted that the first two selections require achieving the target value of a 20% fuel saving at the site selected. However, since the 20% value was only a target and not a threshold it was decided to permit two of the selections to be made in which this target was not quite met. The overall objective of the Department of Energy Pipeline Bottoming Cycle Program is to save fuel on the domestic pipelines. For that reason the criterion of maximum US fuel saving in barrels per day equivalent (BPDE) was applied to two of the sites selected. However, it is not possible to get

TABLE 3-2

ADDITIONAL SITE SELECTION CRITERIA

Site Selection No.	Fuel Saving %	U.S. Fuel Saving BPDE	Payback
1	$\geq$ 20	Best	
2	$\geq$ 20		Best
3	< 20	Best	
4	< 20		Best

the pipeline companies to accept the pipeline bottoming cycle apparatus unless the benefits are worth the cost. For this reason, a simple payback calculation was used to rank the sites as the economic desirability of a particular site. Later in the Program, a more detailed cost analysis was conducted to determine the economic viability of the system.

3.3 SURVEY OF DOMESTIC GAS PIPELINE COMPRESSOR PRIME MOVERS

The purpose of surveying the domestic prime movers was to determine what is typical about the nation's compressor stations. Shown in Figure 3-1 are some data on the types and sizes of gas pumping equipment on the domestic gas pipelines obtained by analyzing the data of Reference 2. The abscissa shows power ranges of the various types of equipment on the nation's gas pipelines. The ordinate shows the total power of all equipment nationwide of the same power class per site. The data indicate a wide diversity of types and amounts of installed power on the nation's gas pipelines. Contrary to the liquid pipelines, very little electrical equipment is used directly for pumping. It can be seen that there is significantly more reciprocating power installed than gas turbine power.

Shown in Table 3-3 is an estimate of the potential energy saving by the use of bottoming cycle apparatus on gas pipeline pumping prime movers for both gas turbines and reciprocating engines. The data are presented in various classes of unit power for both gas turbines and reciprocating engines. The data for the gas turbines were taken from Reference 3. These data accounted for only about 3 million installed horsepower and included two Canadian pipelines. The data, however, was considered to be typical of the United States and the values were increased by 31% in order to make the total installed gas turbine horsepower agree with the value of approximately 4 million for gas turbines indicated in Reference 4. In a similar way the data from Reference 5 on reciprocating engines were fragmentary but also considered typical of the US and were increased by 11% in order to make the total installed horsepower agree with the value of 9 million found in Reference 4. In both categories of engines the number of units were divided up among the various classes of horsepower per unit. Estimates for the fuel saving per unit were made based upon the engine characteristics in the various classes. For example, in the gas turbine portion of the table the power classes from 5000 to 15,000

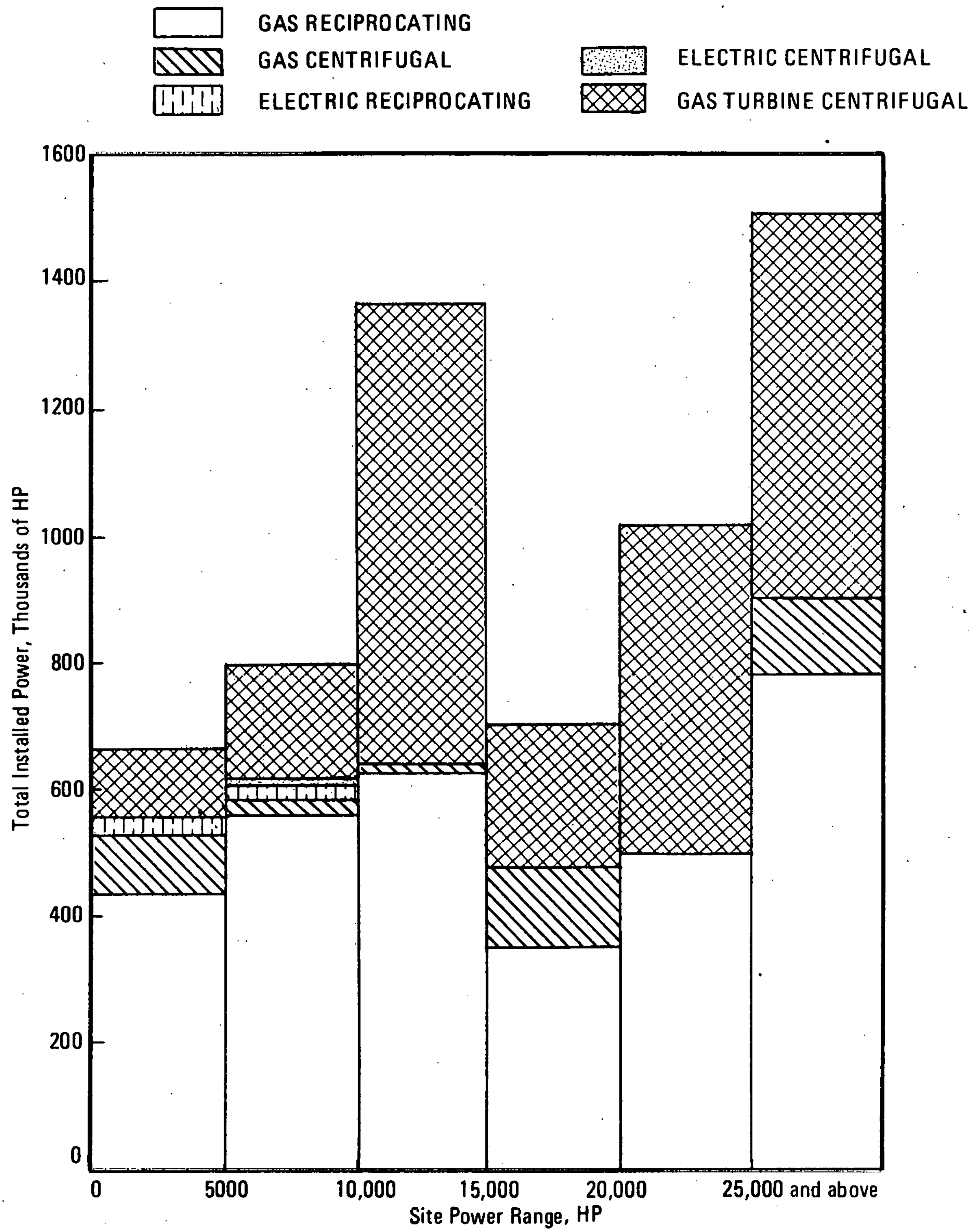

Figure 3-1. Total Installed Power by Site Power Class

TABLE 3-3

POTENTIAL ENERGY SAVINGS BY BOTTOMING GAS PIPELINES PUMPING PRIME MOVERS

GAS TURBINES (Reference 3)

Unit HP	1000 to 5000	To 10000	To 15000	To 20000	To 25000	& Above	Totals
Number	193	183	113	6	33	1	529
Total HP	390,000	1,424,500	1,354,800	104,440	680,400	45,700	3,999,840
Average Unit HP	2,000	8,000	12,000	17,500	20,600	45,700	
Heat Rate, BTU/HP Hr	15,500	8,000	8,000	9,000	9,000	9,000	
Fraction Fuel Saved*	0.24	0.15	0.15	0.21	0.21	0.21	
BPDE	4,234	6,724	6,395	776	5,058	340	23,527

RECIPROCATING (Reference 5)

Unit HP	1000 to 5000	To 10000	To 15000	To 20000	To 25000	& Above	Totals
Number	4,215	129	31				4,375
Total HP	7,869,000	795,300	336,200				9,000,500
Average Unit HP	1,900	6,200	11,000				
Heat Rate, BTU/HP Hr	7,315	7,315	7,315				
Fraction Fuel Saved*	0.13	0.13	0.13				
BPDE	29,435	2,975	1,258				33,668

*Estimated taking into account the variety of kinds of prime movers on the nation's pipelines.

HP were shown having a lower estimated fuel saving because of the large number of recuperated engines in this category. The fuel saving per unit for the reciprocating engines is shown lower than for the gas turbines because of the relatively lower heat rates. The final row in each portion of the table indicates the fuel saving in BPDE obtained from each power category for each type of engine. The total fuel saving for all the gas turbines amount to approximately 24,000 barrels per day equivalent. This amount of fuel can be saved in one category alone of reciprocating engines. In the power class of reciprocating engines from 1000 to 5000 HP better than 29,000 BPDE is estimated to be saved. In total, reciprocating engines are estimated to be able to save approximately 34,000 BPDE compared to 24,000 for the gas turbines. The significance of these values is that although the reciprocating engines save considerably less fuel on a unit basis, the large amount of installed power of this class of engine results in a large saving in fuel nationwide.

As was pointed out earlier, there is a wide diversity in both the gas turbine and reciprocating engine equipment installed in the domestic pipelines. Reference to Table 3-3 indicates there are approximately 1.4 million installed horsepower in the gas turbine category between 5 and 10 thousand horsepower per unit. For the next category from 10 to 15 thousand horsepower there is approximately 1.3 million horsepower. However, because of the large number of recuperated engines in the 5 to 10 thousand HP category the largest block of simple-cycle gas turbines falls in the 10 to 15 thousand horsepower category. For this reason a large block of similar engines in the 10 to 15 thousand horsepower category for simple-cycle gas turbines was sought. Shown in Table 3-4 are five groups of engines which have an aggregated installed power of 544,500 horsepower. The source of this data which is Reference 3, however, accounts for only 2.7 million out of a known 4 million horsepower. Therefore, the aggregate amount of gas turbines in this category could amount to as much as 804,500 horsepower. It will be noted that all the engines delineated in Table 3-4 have a range of power between 12,000 and 12,500 HP. Also the exhaust temperatures and flow rates lie in a very narrow range. This means that bottoming cycle apparatus made for one of these engines could be applied to all other engines with only minimal alterations. Thus there are a total of at least 44 engines which are known to be in this power class

TABLE 3-4

DOMESTIC 12000 TO 12500 HP SIMPLE CYCLE GAS TURBINES

Manufacturer	Model	Power HP	Exh. Temp., °F	Flow Rate PPS	No. of Units	Total Power HP
Dresser Clark	DJ-125	12500	740	154	1	12500
Cooper Bessemer	RT-125	12500	735	139-162	15	187500
Ingersoll-Rand	JP-125	12500	713	135-148	3	37500
Turbo Power & Marine	GG3-C4	12000	700	153	11	132000
Rolls Royce	75G,76G	12500	740	154	14	175000
Totals					44	544500

which have similar conditions of the exhaust temperature and flow rate. Shown in Table 3-5 are three General Electric recuperated gas turbine models. These range in power from 8,000 to 11,100 horsepower. Here again the exhaust temperatures and flow rates are similar. Thus bottoming cycle apparatus applied to one of these engines could be applied to all the rest with minimal changes. For these three models there are 405,000 horsepower known to be installed in domestic pipelines from sales records. In addition to these specific models there are other GE recuperated engines with similar exhaust temperatures and flow rates so as to add up to a total of 676,000 horsepower installed in domestic pipelines. For these three models then there are 49 engines which are installed on domestic pipelines and which have flow rates and exhaust temperatures in a narrow band. The installed power on domestic sites varies through a large range and it is known that a number of sites have both reciprocating and gas turbine units. In attempting to determine what was typical about domestic gas pipeline sites, a survey of two pieces of data was made to determine the general characteristics of these sites.

Initially, the data from Reference 6 was studied to determine the typical power mix between gas turbines and reciprocating engines on compressor sites. These data indicated that 62% of the installed gas turbine power was on unmixed sites; that is, on sites which had gas turbines only as prime movers. Similarly for reciprocating engines, 88% of the installed reciprocating power was on unmixed sites. The data presented by this reference, however, only accounted for about 65% of the installed horsepower of the industry when compared with the data of Reference 4. Also, it was noted that the pipeline companies with the highest installed horsepower were not generally included in this data. For this reason, Reference 7 was surveyed for the ten domestic pipelines having the highest installed power. These 10 companies are listed in Table 3-6 (Reference 8). When this data was studied it was found that 68% of the installed gas turbine power was located on unmixed sites and 94% of the installed reciprocating power was on unmixed sites. The significance of these data is that it is not typical to attempt to use the advantages of gas turbine and reciprocating engines of a given site in a bottoming cycle system. Thus, in evaluating sites for selection bottoming cycle apparatus was assumed to be put either on gas turbines or on reciprocating engines but in no instance was it assumed that it would be necessary for both types of engines to be on the same site.

TABLE 3-5

SELECTED DOMESTIC RECUPERATED GAS TURBINES

MANUFACTURER	MODEL	NO. OF UNITS	RATED POWER HP	EXH. TEMP., °F	FLOW RATE, PPS	TOTAL POWER
GE	M3802R	14	8000	700	108	112000
GE	M3912R	25	9100	700	110	227500
GE	M3112R	10	11100	700	114	111000
TOTAL		49				450500

TABLE 3-6

DOMESTIC PIPELINE COMPANIES WITH THE HIGHEST INSTALLED POWER *

Company	Address	Transmission HP
Tenneco Inc.	Houston, TX	1,244,113
Texas Eastern Transmission Corp.	Houston, TX	1,176,110
Transcontinental Gas Pipe Line Corp.	Houston, TX	951,185
Natural Gas Pipeline Co. of America	Chicago, IL	938,105
Northern Natural Gas Co.	Omaha, NE	919,508
Panhandle Eastern Pipeline Co.	Houston, TX	827,417
El Paso Natural Gas Co.	Houston, TX	803,068
Michigan Wisconsin Pipe Line Co.	Detroit, MI	766,942
Columbia Gulf Transmission Corp.	Houston, TX	470,516
Texas Gas Transmission Corp.	Owensboro, KY	461,710

* Reference 8

Another piece of data which bears upon what is typical in compressor sites was obtained from Reference 5. In this reference the reciprocating engines that are used on pipeline applications were delineated as regards model number, manufacturer, size and other characteristics. From this data it was found that of the 9 million installed reciprocating horsepower in the field only 2 million was of the naturally aspirated type whereas fully 7 million of the reciprocating engines were turbocharged. Therefore, in the evaluation of sites for selection the natural aspirated reciprocating engines were ignored.

3.4 AVAILABLE SITES

Shown in Table 3-7 are the sites that have been made available to the contractor from committed pipeline companies. In this table are shown the name of the pipeline company, the location of the site, and the installed gas turbine and reciprocating engine power. Also some comments on these sites are shown in the last column. Concerning the sites of Columbia Gulf Transmission Company it will be noted that a comment is made that several of the sites have low operational hours. Since low operational hours impacts the economic factors in the selection of the site adversely these sites were eliminated from further consideration. Also it should be noted that on the Columbia Gulf Transmission Company pipeline there is only one site where there is an operational gas turbine, namely, Rayne, LA. These sites, then, with the elimination of the ones mentioned above, constitute the sites which were evaluated in order to arrive at the recommended sites.

3.5 SITE COMPATIBILITY CALCULATIONS

As indicated in Table 3-2 where additional site selection criteria were delineated the payback period of each of the sites must be determined. In order to do this it was necessary to carry out a parametric study of bottoming cycles for application to the gas pipelines in which the variation of the cost of the equipment was approximated. Three cycles were considered, two using steam as the working fluid at turbine inlet pressures of 200 and 300 psia and one using toluene at a pressure of 270 psia. The assigned values for each of these cycles is delineated in Table 3-8. The parametric analysis was carried out for a gas turbine having the mass flow rate and exhaust temperature shown in Table 3-8.

TABLE 3-7

AVAILABLE SITES

PIPELINE CO.	LOCATION	GAS TURBINE HP	RECIPROCATING HP	COMMENTS
1. TEXAS GAS TRANSMISSION CORP.	LAKE CORMORANT, MS	20,100	14,000	HOST: GOOD SITE
	COVINGTON, TN	12,000	17,000	BEST SITE
	JEFFERSONTOWN, KY	11,000	13,500	GOOD SITE
	GREENVILLE, MS	20,100	20,600	GOOD SITE
2. PACIFIC GAS & ELECTRIC CO.	TIONESTA, CA	11,100	-	
	BURNEY, CA	19,320	-	HOST: FIRST CHOICE
	GERBER, CA	12,830	-	
	DELEVAN, CA	17,280	-	
	TOPACK, CA	-	35,000	
	HINKLEY, CA	-	39,600	
	KETTLEMAN, CA	-	28,930	
3. PACIFIC GAS TRANS. CO.	EASTPORT, ID	11,400	-	
	SAMUELS, ID	8,090	-	
	ATHOL, ID	19,870	-	
	ROSALIA, WA	19,990	-	
	STARBUCK, WA	20,940	-	
	WALLULA, WA	25,500	-	
	IONE, OR	21,060	-	
	KENT, OR	23,500	-	
	MADRAS, OR	15,580	-	
	BEND, OR	22,300	-	
	CHEMULT, OR	21,900	-	HOST: SECOND CHOICE

TABLE 3-7 (Cont'd.)

AVAILABLE SITES

	PIPELINE CO.	LOCATION	GAS TURBINE HP	RECIPROCATING HP	COMMENTS
3.	PACIFIC GAS TRANS. CO. (Cont'd.)	BONANZA, OR	28,655	-	
4.	COLUMBIA GULF TRANS. CO.	ALEXANDRIA, LA	27,500	14,000	
		BANNER, MS	28,500	14,800	LOW TURBINE USAGE FACTOR
		CLEMENTSVILLE, KY	21,000	14,000	LOW TURBINE USAGE FACTOR
		CORNITH, MS	26,500	14,000	
		DELHI, LA	39,500	14,000	LOW TURBINE USAGE FACTOR
		HAMPSHIRE, TN	23,000	18,000	LOW TURBINE USAGE FACTOR
		HARTSVILLE, TN	32,000	14,000	LOW TURBINE USAGE FACTOR
		INVERNESS, MS	21,000	14,000	
		RAYNE, LA	37,500	14,000	OPERATIONAL GAS TURBINE
		STANTON, KY	28,500	14,000	LOW TURBINE USAGE FACTOR

TABLE 3-8

ASSUMPTIONS AND CYCLE DATA USED IN PARAMETRIC ANALYSIS

Bottoming Cycle Parameters	Working Fluid: Toluene	Working Fluid: Steam	
Turbine Inlet Pressure (psia)	270	200	300
Turbine Inlet Temperature (°F)	500	500	500
Turbine Efficiency	0.8	0.72	0.72
Pump Efficiency	0.7	0.7	0.7
Fan Power, % Heat Rejected	1.0	1.0	1.0

Gas Turbine Parameters

Exhaust Gas Flow Rate - 113.0 lbs/sec

Exhaust Gas Temperature - 660°F

Bottoming Cycle Component Cost is Proportional To $(\text{Heat Exchanger Area})^{0.6}$

This value of exhaust temperature is typical of the value for the 450500 installed horsepower shown on Table 3-5 for the three models of recuperated gas turbines on domestic pipelines and is in the low range for the gas turbines to be considered. At exhaust gas temperatures higher than 660°F the performance and economics is improved. Because several of the committed pipeline companies indicated the undesirability of using water-cooled condensers in compressor stations in remote localities, it was decided to limit the parametric study to the case of an air-cooled condenser. As is known from studies like the one made in Reference 9 the cost of the heat exchangers in a bottoming cycle system are a large portion of the total bottoming cycle system cost. Therefore, in this preliminary calculation it was assumed that the bottoming cycle cost would be made proportional to the heat exchanger area raised to the 0.6 power. Reference 10 indicates that the cost of apparatus varies with capacity, raised to powers somewhat lower than unity. An exponent of 0.6 was selected as being typical of a large amount of apparatus to the bottoming cycle type.

Shown in Figure 3-2 are results of the parametric study for steam. Plotted are the relative cost of the bottoming cycle equipment against the bottoming cycle power for two levels of steam pressure and condenser temperatures of 125, 150 and 200°F. The curves indicate minimum values of relative cost as a function of bottoming cycle horsepower. The indications from the curves are that the bottoming cycle using a pressure of 300 psia into the turbine has a lower cost than one with a pressure of 200 psia. The condenser temperature of 150°F results in the lowest value of relative costs. The square symbol indicates the lowest cost design on the figure. The results of the parametric study for toluene are presented in Figure 3-3. Curves for condenser temperatures of 125, 150 and 175°F are shown. As for the steam cases the condenser temperature of 150°F results in the lowest cost. The square symbol represents the best cycle for toluene. In comparing the optimum cycles for steam and toluene the following results were obtained:

	Steam	Toluene
Turbine inlet pressure, psia	300	270
Condenser temperature, °F	150	150
Bottoming cycle power, HP	2030	2750
Relative cost	1.002	0.98

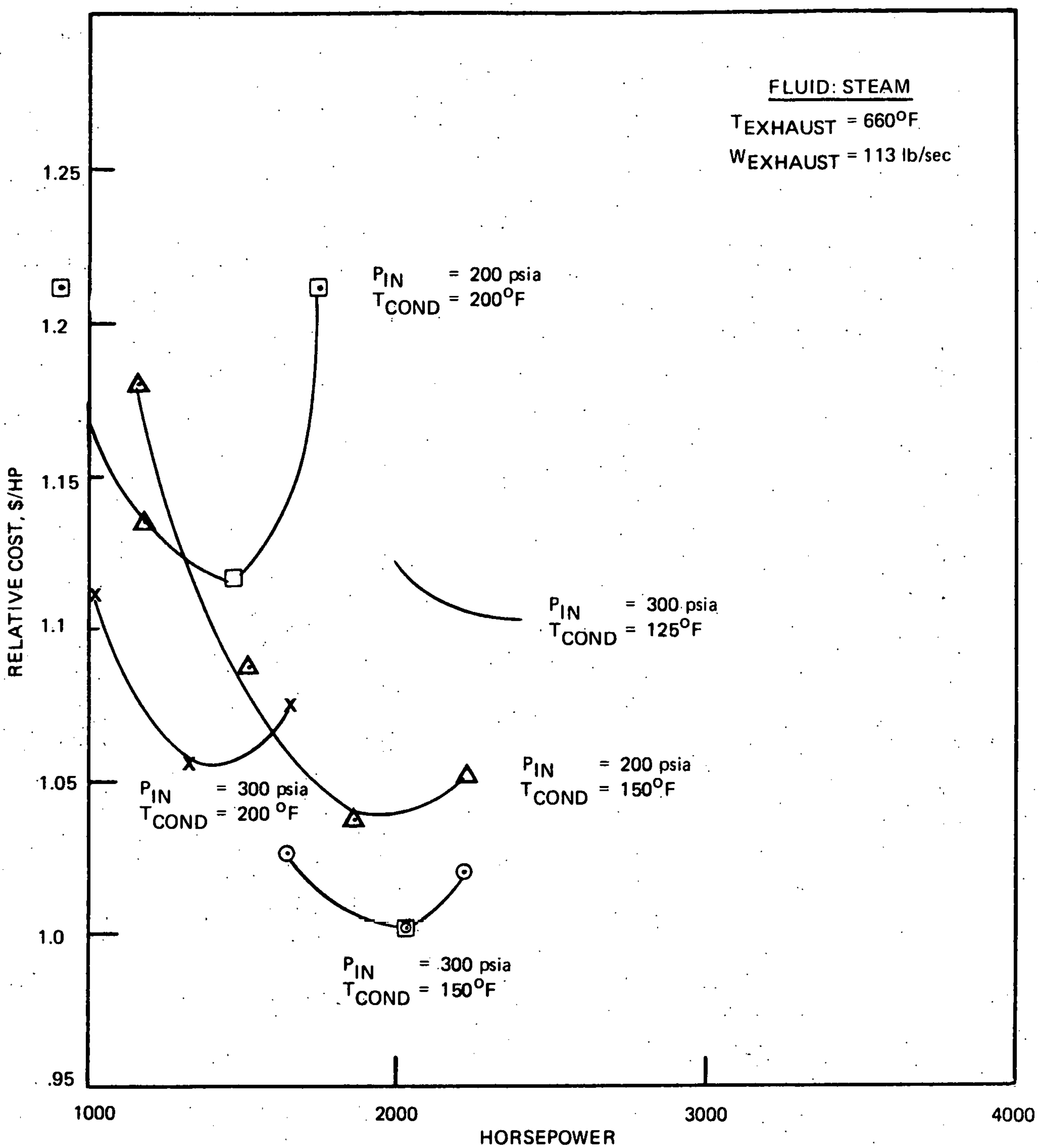

Figure 3-2. Relative Costs of Steam Bottoming Cycles.

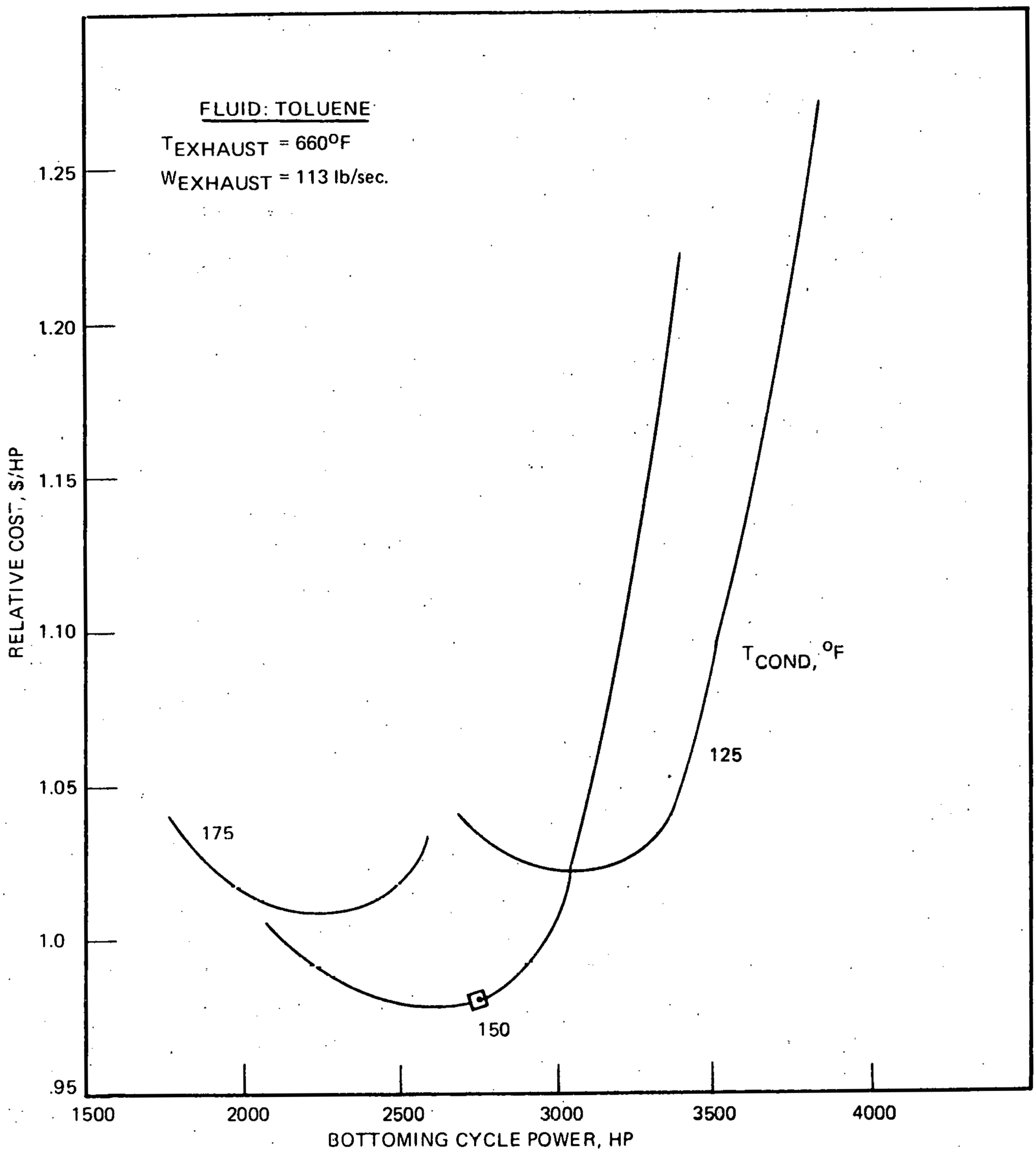

Figure 3-3. Relative Cost of Toluene Bottoming Cycles.

Thus, it can be seen that the toluene cycle both achieves a higher power output from the same exhaust stream than steam and also the apparatus has a slightly lower cost. As a result the cycle represented by the square on Figure 3-3 for toluene was used in computing bottoming cycle power for the prime movers on the site available from the committed pipelines. It should be noted that a toluene turbine is expected to have a higher efficiency than a steam turbine of the same power because the former has a higher volume flow rate than the latter and suffers no loss in efficiency due to the presence of condensate in the last stages. However, if the efficiencies were the same, toluene would still produce more power than steam from the given exhaust gas conditions.

The economic evaluation of the sites was deemed one of the important criteria for site selection. A simple economic indicator - the simple payback period - was utilized for this purpose. Shown in Table 3-9 is the definition of the parameter as used in the evaluation of the sites. The payback period is simply the capital cost for the bottoming cycle equipment in dollars divided by the fuel cost saving in dollars per year. For the gas turbine this payback period can be characterized as shown on Table 3-9. The payback period for the gas turbine is the cost of the bottoming cycle equipment in dollars per horsepower times the horsepower of the bottoming cycle divided by four factors. The first factor is the change in the heat rate between the unbottomed and the bottomed condition, the second factor is the value of the fuel in dollars per million BTU, the third factor is the hours per year that the equipment was used and the fourth factor is the horsepower of the gas turbine. For the reciprocating engine, the calculations payback is somewhat different. The reason for this is that, contrary to the gas turbine case, in the reciprocating engine case sites are generally characterized by a number of reciprocating engines at a given site. Thus, in order to compensate for the extra horsepower generated by the bottoming cycle, one of the reciprocating engines may be shut off or all throttled back slightly. Since the fuel consumption curves of the reciprocating engines are very flat in the power range of 75 to 100% of rated power the heat rate changes very little for the small amount (10 or 15%) that the reciprocating engines need to be throttled back to compensate for the bottoming cycle power. Thus, the payback period for the

TABLE 3-9

ECONOMIC EVALUATION CRITERIA FOR THE SITES*

$$\text{SIMPLE PAYBACK PERIOD} = \frac{\text{CAPITAL COST, \$}}{\text{FUEL COST SAVING, \$/YR}}$$

FOR GAS TURBINE:

$$\text{PAYBACK} = \frac{\left(\$/\text{HP}\right)\left(\text{HP}_{\text{BC}}\right)}{\left(\frac{\Delta\ \text{BTU}}{\text{HP-HR}}\right)\left(\frac{\$}{10^{6}\ \text{BTU}}\right)\left(\frac{\text{HRS}}{\text{YR}}\right)\left(\text{HP}_{\text{GT}}\right)}$$

FOR RECIPROCATING ENGINE:

$$\text{PAYBACK} = \frac{\left(\$/\text{HP}\right)}{\left(\frac{\text{BTU}}{\text{HP-HR}}\right)\left(\frac{\$}{10^{6}\ \text{BTU}}\right)\left(\frac{\text{HRS}}{\text{YR}}\right)}$$

*An elaborate economic assessment of the bottoming cycle system for the three pipeline sites was carried out later in the program based on discounted cash flow methods.

reciprocating engines is simply the cost per horsepower paid for the bottoming cycle divided by the reciprocating engine heat rate, the value of fuel and the hours per year that the equipment was utilized.

In order to have values of bottoming cycle cost which varied with bottoming cycle power the cost was assumed to vary with the 0.6 power of the bottoming cycle power. In Reference 9 the cost per HP of organic Rankine systems was found to be a little less than $300/HP. This latter value when corrected for inflation to 1978 was assumed for the largest bottoming cycle power encountered in the present study which was 4400 HP. The variation of the cost per horsepower used is shown in Figure 3-4. It should be emphasized that all values of payback period shown are relative because of the arbitrary selection of $300/HP selected for 4400 HP. The cost of the apparatus designed in Task II, Systems Preliminary Design will be obtained by estimating the cost of each component and the assembly and installation costs.

3.6 THE SELECTION OF SITES

Selection analyses were performed for both gas turbine and reciprocating sites.

3.6.1 GAS TURBINE SITES

Shown in Table 3-10 are the engine characteristics for the gas turbine sites available from the committed pipeline companies. Shown in the table are the pipeline and the site location and the engine characteristics of the gas turbines on these sites. The first group of sites have a Cooper-Bessemer RT-125 simple-cycle gas turbine which has an exhaust temperature of 735°F and a flow rate in a range from 139 to 162 lbs/sec. This gas turbine must be throttled back to 76% rated power in order to compensate for bottoming cycle power. The bottoming cycle power for these gas turbines ranges from approximately 3000 to 3350 horsepower depending upon the power reported by the pipeline company for the site. The next group of engines shown in the table is the Ingersoll-Rand JP-125 simple-cycle gas turbine. This gas turbine has an exhaust temperature of 713°F and a mass flow ranging from 135 to 148 lbs/sec. Again, this gas turbine is throttled back to 76% power and the value of the bottoming cycle

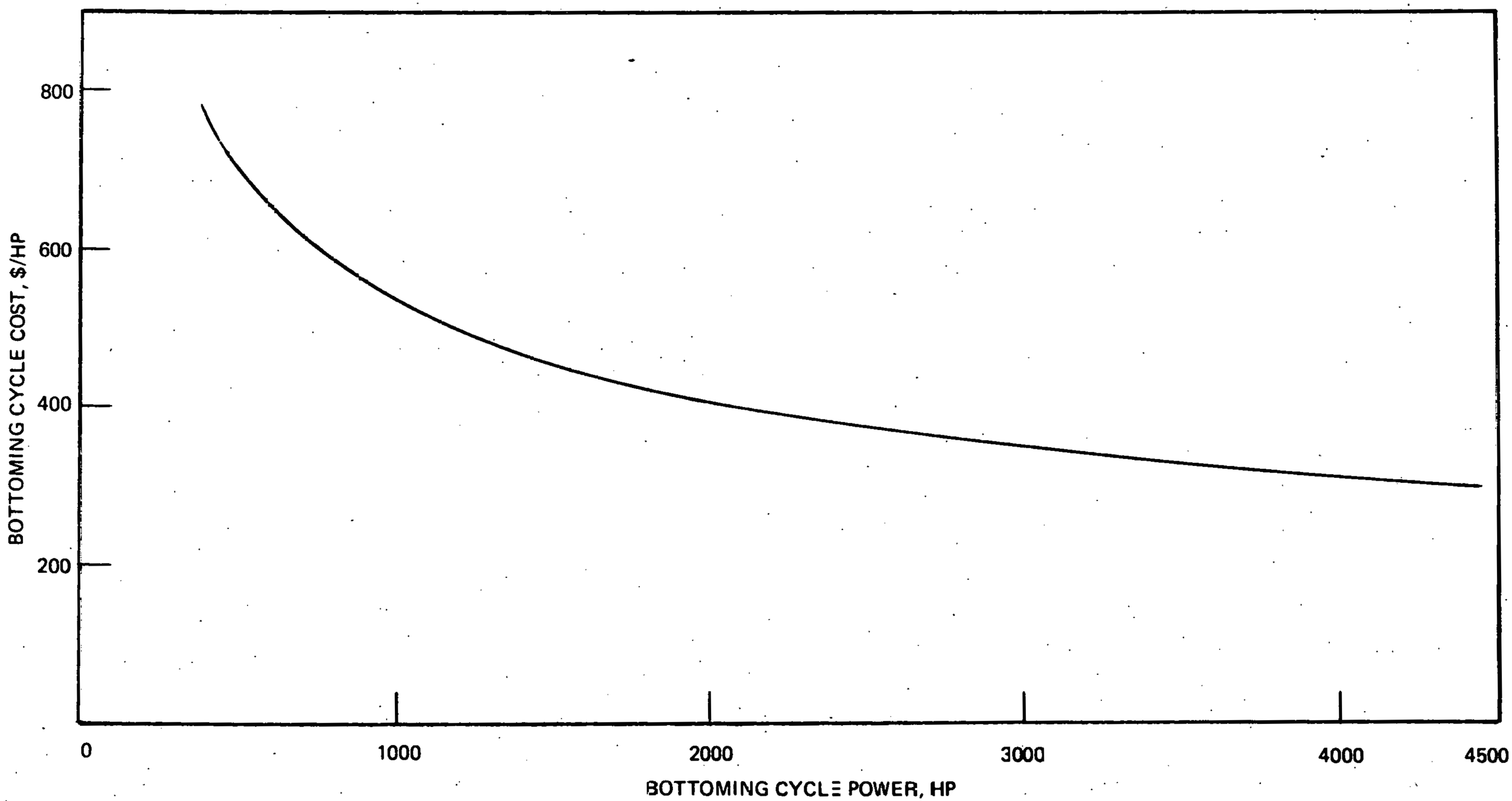

Figure 3-4. Assumed Variation of Bottoming-Cycle Cost With Power.

TABLE 3-10

GAS TURBINE ENGINE CHARACTERISTICS

PIPELINE CO. & LOCATION	NO. OF UNITS	RATED POWER* H.P.	HEAT RATE BTU/HP-HR	MANUFACTURER	MODEL	TYPE	EXH. TEMP., °F	FLOW RATE PPS	% RATED POWER	B/C POWER HP
PACIFIC GAS & ELECTRIC										
TIONESTA, CA	1	11100	10060	COOPER-BESSEMER	RT-125	SIMPLE-CYCLE	735	140	76	2970
GERBER, CA	1	12830	10060	"	"	"	735	162	76	3436
PACIFIC GAS TRANS. CO.										
EASTPORT, ID	1	11400	10060	"	"	"	735	149	76	3161
ATHOL, ID	1	11850	10060	"	"	"	735	150	76	3182
STARBUCK, WA	1	12500	10060	"	"	"	735	158	76	3352
WALLULA, WA	1	12750	10060	"	"	"	735	161	76	3415
WALLULA, WA	1	12750	10060	"	"	"	735	161	76	3415
KENT, OR	1	11750	10060	"	"	"	735	149	76	3161
KENT, OR	1	11750	10060	"	"	"	735	149	76	3161
BEND, OR	1	11150	10060	"	"	"	735	141	76	2992
BEND, OR	1	11150	10060	"	"	"	735	141	76	2992
CHEMULT, OR	1	10950	10060	"	"	"	735	139	76	2949
CHEMULT, OR	1	10950	10060	"	"	"	735	139	76	2949
COLUMBIA GULF TRANS. CO.										
RAYNE, LA	1	12500	10060	"	"	"	735	158	76	3352
RAYNE, LA	1	12500	10060	"	"	"	735	158	76	3352
RAYNE, LA	1	12500	10060	"	"	"	735	158	76	3352

* 80°F, 1000 ft altitude

TABLE 3-10 (Cont'd.)

GAS TURBINE ENGINE CHARACTERISTICS

PIPELINE CO. & LOCATION	NO. OF UNITS	RATED POWER* H.P.	HEAT RATE BTU/HP-HR	MANUFACTURER	MODEL	TYPE	EXH. TEMP., °F	FLOW RATE PPS	% RATED POWER	B/C POWER HP
PACIFIC GAS & ELECTRIC										
BURNEY, CA	1	11550	9300	INGERSOLL-RAND	JP-125	S. C.	713	140	76	2970
PACIFIC GAS TRANS. CO.										
ROSALIA, WA	1	11920	9800	INGERSOLL-RAND	JP-125	S. C.	713	145	76	3076
IONE, OR	1	12220	9800	"	"	"	713	148	76	3146
BONANZA, OR	1	11125	9800	"	"	"	713	135	76	2864
TEXAS GAS TRANS. CO.										
COVINGTON, TN	1	12000	10890	P&WA/TP&M	GG3C-4	S. C.	700	152.8	82	2935

* 80°F, 1000 ft altitude

TABLE 3-10 (Cont'd.)

GAS TURBINE ENGINE CHARACTERISTICS

PIPELINE CO. & LOCATION	NO. OF UNITS	RATED POWER* H.P.	HEAT RATE BTU/HP-HR	MANUFACTURER	MODEL	TYPE	EXH. TEMP., °F	FLOW RATE PPS	% RATED POWER	B/C POWER HP
TEXAS GAS TRANS. CO.										
GREENVILLE, MS	1	11000	8020	GENERAL ELECTRIC	M3112R	RECU-PERATED	750	113.9	82.0	2009
LAKE CORMORANT, MS	1	11000	8020	"	M3112R	"	750	113.9	82.0	2009
JEFFERSONTOWN, KY	1	11000	8020	"	M3112R	"	750	113.9	82.0	2009
GREENVILLE, MS	1	9100	8690	"	M3912R	"	700	109.7	82.0	1935
LAKE CORMORANT, MS	1	9100	8690	"	M3912R	"	700	109.7	82.0	1935
PACIFIC GAS & ELECTRIC										
BURNEY, CA	1	7770	8750	"	M3912R	"	625	98.	82.0	1728
DELVAN, CA	1	8640	8750	"	M3912R	"	625	106.	82.0	1870
DELVAN, CA	1	8640	8750	"	M3912R	"	625	106.	82.0	1870
PACIFIC GAS TRANS. CO.										
SANDPOINT, ID	1	8090	8750	"	M3912R	"	625	102.	82.0	1799
ATHOL, ID	1	8070	8750	"	M3912R	"	625	102.	82.0	1781
ROSAHIA, WA	1	8070	8750	"	M3912R	"	625	102.	82.0	1799
STARBUCK, WA	1	8440	8750	"	M3912R	"	625	106.	82.0	1870
IONE, OR	1	8840	8540	"	M3912R	"	625	104.	82.0	1834

* 80°F, 1000 ft altitude

power ranges from 3000 to 3146 horsepower. One site on the Texas Gas Transmission Company pipeline has a Pratt-Whitney GG3C-4 simple-cycle gas turbine which has an exhaust temperature of approximately 700°F and a mass flow of 152.8 lbs/sec. This gas turbine must be throttled back to 82% power and the bottoming cycle horsepower is approximately 2900 horsepower. The next group of gas turbines are recuperated engines built by the General Electric Company; two models are shown. Both models have exhaust temperatures in the range from 625 to 750°F and flow rates in the range from 102 to 109.7 lbs/sec. These gas turbines must be throttled back to 82% power in order to compensate for the bottoming cycle power that is generated and the bottoming cycle power is in the range from 1800 HP up to 2009 HP.

Shown in Table 3-11 are the data computed for the various sites to determine the compatibility of the site with the additional selection criteria. In this Table the gas turbines were assumed to be throttled back the amount of the bottoming cycle power and the gas turbine heat at throttle condition was estimated. The first group of sites are those which have the Cooper-Bessemer RT-125 simple cycle gas turbines. It can be seen that bottoming cycle apparatus for these gas turbines have payback periods that vary from approximately 2 years to over 6 years. These engines with their bottoming cycles have a fuel saving capability in a throttled back condition of 22.3%. These engines are in a class of engines that have power ratings at 80°F and 1000 ft in the range of 12,000 to 12,500 horsepower. The nationwide potential horsepower for bottoming as indicated previously is in the range of 544,500 to 804,500. If all engines in this class are bottomed, the domestic fuel saving would be somewhere between 3806 to 5630 BPDE. The next group of engines are Ingersoll-Rand JP-125 simple-cycle gas turbines which have a rated power at 80°F and 1000 ft of approximately 12,500 HP. The payback period for these engines ranges from 2 years to 2.7 years. Because they are in the same power class and have approximately the same exhaust conditions as the Cooper-Bessemer engines, there is the same potential U.S. horsepower for bottoming and the same potential BPDE in fuel saving. Like the Cooper-Bessemer engines these engines will have approximately 22.3% fuel saving per unit. The last engine in this group is the Pratt-Whitney GG3C-4 simple-cycle gas turbine which has a rated power of approximately 12,000 HP

TABLE 3-11

SITE COMPATIBILITY, GAS TURBINE SITES

PIPELINE CO. & LOCATION	MANU-FACTURER	MODEL	B/C POWER* HP	COST, $/HP	DIFFERENCE IN HEAT RATE, BTU/HP-HR	DUTY, HRS/YR	FUEL VALUE, $/10⁶ BTU	PAYOUT YRS.	% FUEL SAVING	U.S. POTEN. HP	U.S. FUEL SAVING, BPDE
PACIFIC GAS & ELECTRIC											
TIONESTA, CA	COOPER-BESSEMER	RT-125	2970	356	2240	8085	2.20	2.39	22.3	544,500 TO 804,500	3806 TO 5630
GERBER, CA	"	"	3436	336	2240	8261	2.20	2.21	22.3		
PACIFIC GAS TRANS. CO.											
EASTPORT, ID	"	"	3161	347	2240	8348	2.20	2.34	22.3		
ATHOL, ID	"	"	3182	346	2240	8532	2.20	2.21	22.3		
STARBUCK, WA	"	"	3352	339	2240	8357	2.20	2.21	22.3		
WALLULA, WA	"	"	3415	337	2240	4511	2.20	4.06	22.3		
WALLULA, WA	"	"	3415	337	2240	6859	2.20	2.67	22.3		
KENT, OR	"	"	3161	347	2240	8226	2.20	2.30	22.3		
KENT, OR	"	"	3161	347	2240	8217	2.20	2.30	22.3		
BEND, OR	"	"	2992	355	2240	3013	2.20	6.41	22.3		
BEND, OR	"	"	2992	355	2240	7840	2.20	2.46	22.3		
CHEMULT, OR	"	"	2949	357	2240	6535	2.20	2.99	22.3		
CHEMULT, OR	"	"	2949	357	2240	5046	2.20	3.87	22.3		
COLUMBIA GULF TRANS. CO.											
RAYNE, LA	"	"	3352	339	2240	8280	2.21	2.23	22.3		
RAYNE, LA	"	"	3352	339	2240	240	2.21	76.5	22.3		
RAYNE, LA	"	"	3352	339	2240	240	2.21	76.5	22.3		

*Based on gas turbine operation at 80°F, 1000 ft altitude

TABLE 3-11 (Cont'd.)

SITE COMPATIBILITY, GAS TURBINE SITES

PIPELINE CO. & LOCATION	MANU-FACTURER	MODEL	B/C POWER,* HP	COST, $/HP	DIFFERENCE IN HEAT RATE, BTU/HP-HR	DUTY HRS/YR	FUEL VALUE, $/$10^6$ BTU	PAYOUT YRS.	% FUEL SAVING	U.S. POTEN. HP	U.S. FUEL SAVING, BPDE
PACIFIC GAS TRANS. CO.											
ROSALIA, WA	INGERSOLL-RAND	JP-125	3076	351	2182	8375	2.20	2.25	22.3	544,500 TO 804,500	3806 TO 5630
IONE, WA	"	"	3146	348	2182	8559	2.20	2.18	22.3		
BONANZA, OR	"	"	2864	361	2182	8121	2.20	2.38	22.3		
TEXAS GAS TRANS. CO.											
COVINGTON, TN	P&WA/ TP&M	GG3C-4	2935	358	2122	5420	1.01	7.54	19.5		
PACIFIC GAS & ELECTRIC											
BURNEY, CA	INGERSOLL-RAND	JP-125	2970	356	2182	7064	2.20	2.70	22.3		

*Based on gas turbine operation at 80°F, 1000 ft altitude

TABLE 3-11 (Cont'd.)

SITE COMPATIBILITY, GAS TURBINE SITES

PIPELINE CO. & LOCATION	MANU-FACTURER	MODEL	B/C POWER,* HP	COST, $/HP	DIFFERENCE IN HEAT RATE, BTU/HP-HR	DUTY, HRS/YR	FUEL VALUE,, $/$10^6$ BTU	PAYOUT YRS.	% FUEL SAVING	U.S. POTEN. HP	U.S. FUEL SAVING, BPDE
TEXAS GAS TRANS. CO.											
GREENVILLE, MS	GENERAL ELECTRIC	M3112R	2009	402	1198	6009	1.01	10.09	14.9	405,000 TO 676,000	1770 TO 2950
LAKE CORMORANT, MS	"	"	2009	402	1198	6406	1.01	9.47	14.9		
JEFFERSONTOWN, KY	"	"	2009	402	1198	2367	1.01	25.63	14.9		
GREENVILLE, MS	"	"	1935	408	1308	5506	1.01	11.94	15.0		
LAKE CORMORANT, MS	"	"	1935	408	1308	7519	1.01	8.74	15.0		
PACIFIC GAS & ELECTRIC											
BURNEY, CA	"	"	1728	427	1317	8085	2.20	4.055	15.0		
DELVAN, CA	"	"	1870	414	1317	8392	2.20	3.684	15.0		
DELVAN, CA	"	"	1870	414	1317	8313	2.20	3.719	15.0		
PACIFIC GAS TRANS. CO.											
SANDPOINT, ID	"	M3912R	1799	420	1317	8488	2.20	3.8	15.0		
ATHOL, ID	"	"	1781	422	1317	7463	2.20	4.58	15.0		
ROSALIA, WA	"	"	1799	420	1317	7656	2.20	4.22	15.0		
STARBUCK, WA	"	"	1870	414	1317	7463	2.20	4.15	15.0		
IONE, OR	"	"	1834	417	1285	7788	2.20	3.93	15.0		

*Based on gas turbine operation at 80°F, 1000 ft altitude

at 80°F and 1000 ft. The payback period on this engine is very high, about 7 years, but it is primarily due to the low number of hours it was used per year and the fact that the value of the fuel on the Texas Gas Transmission Company line is less than half of that on the other two lines shown. Bottoming this engine results in a fuel saving per unit of 19.5%. The final group of engines shown in the table comprise the two models of General Electric recuperated gas turbines. These engines save only approximately 15% of the fuel per unit because of their already low heat rate and have payback periods that range from about 3.7 years to as high as 10 years. The payback periods are primarily governed by the number of hours per year that the pipeline uses the engines and by the value that is placed on the fuel that is burned in these engines. Here again, note that the Texas Gas Transmission Company sites have a fuel value which is less than half of the value of that for the sites of the Pacific Gas and Electric Company, the Pacific Gas Transmission Company and the Columbia Gulf Transmission Company sites and, therefore, they have a much longer payback period. Because the recuperated engines miss the 20% fuel saving per unit by a large margin they were dropped from further consideration.

Based upon the desire to find sites on which the bottoming cycle could expect low payback periods the sites presented in Table 3-12 were tentatively selected. The sites span two different gas turbine engines and three pipeline companies. After the selections were made the pipeline companies were contacted for approval of the selected sites. Columbia Gulf Transmission Company was in agreement to utilize their best candidate site at Rayne, LA for further study. However, Pacific Gas and Electric Company, which is the parent company for Pacific Gas Transmission Company, proposed alternate sites for study which are shown as recommended selections on Table 3-13. As can be seen the first two engines on Table 3-13 have a lower number of hours of utilization than the same sites shown in Table 3-12. However, on each of the two recommended sites there is another gas turbine which now is sharing the load. By putting a bottoming cycle on the gas turbine shown, the other gas turbine on each site will not be needed and the utilization will go up to approximately the values shown in Table 3-12.

TABLE 3-12

TENTATIVE SELECTIONS, GAS TURBINE SITES

PIPELINE CO. & LOCATION	MANU-FACTURER	MODEL	B/C POWER, HP	COST, $/HP	DIFFERENCE IN HEAT RATE, BTU/HP-HR	DUTY, HRS/YR	FUEL VALUE, $/$10^6$ BTU	PAYOUT YRS.	% FUEL SAVING	U.S. POTEN. HP	U.S. FUEL SAVING, BPDE
PACIFIC GAS TRANSMISSION CO.											
1. IONE, OR	INGERSOLL-RAND	JP-125	3146	348	2182	8559	2.20	2.18	22.3	544,500 to 804,500	3806 to 5630
PACIFIC GAS & ELECTRIC CO.											
2. GERBER, CA	COOPER-BESSEMER	RT-125	3436	336	2240	8261	2.20	2.21	22.3		
COLUMBIA GULF TRANSMISSION CO.											
3. RAYNE, LA	COOPER-BESSEMER	RT-125	3352	339	2240	8280	2.21	2.23	22.3		

TABLE 3-13

RECOMMENDED SELECTIONS, GAS TURBINE SITES

PIPELINE CO. & LOCATION	MANU-FACTURER	MODEL	B/C POWER, HP	COST, $/HP	DIFFERENCE IN HEAT RATE, BTU/HP-HR	DUTY, HRS/YR	FUEL VALUE, $/10^6 BTU	PAYOUT YRS.	% FUEL SAVING	U.S. POTEN. HP	U.S. FUEL SAVING, BPDE
PG&E											
1. BURNEY, CA	INGERSOLL-RAND	JP-125	2970	356	2182	7064	2.20	2.70	22.3	544,500 to 804,500	3806 to 5630
PACIFIC GAS											
2. CHEMULT, OR	COOPER-BESSEMER	RT-125	2949	357	2240	6535	2.20	2.74	22.3		
COLUMBIA											
3. RAYNE, LA	COOPER-BESSEMER	RT-125	3352	339	2240	8280	2.21	2.04	22.3		

3.6.2 RECIPROCATING ENGINE SITES

Shown in Table 3-14 are the engine characteristics of reciprocating engines on the sites available from the pipeline companies committed to this program. Shown in the table are the pipelines, site locations, number of like units, manufacturer of the reciprocating engines, model numbers, type (turbocharged or naturally aspirated), horsepower rating, and the heat rate. Also shown are the exhaust temperature and flow rates of these engines. Except for the C-B LSV-16 engines, these engines have exhaust temperatures and flow rates in a narrow band. The power of the bottoming cycles for these sites varies from 436 HP to 4362 HP.

Shown in Table 3-15 are site compatibility calculation results for the reciprocating engine sites. (Because the bottoming cycle power is a small fraction of the engine rated power, the rated heat rate was used as a first approximation to the throttled back heat rate.) The cost in dollars per horsepower varies widely among these engine sites because of the variation in the size of the bottoming cycle equipment. In each instance the group of engines on a site that were bottomed were all alike. The reciprocating engines have a percentage fuel saving less than 20% (7.2 to 14.3%). Also the potential U.S. horsepower to be bottomed varies quite widely among these sites. This was arrived at in the following manner. From the survey data the total amount of reciprocating power on sites with various levels of reciprocating power per site was known. From these data the number of sites could be determined which have a certain amount of reciprocating power or more. Thus, for each level of power bottomed on the available sites the total amount of power to be bottomed in blocks of that level was determined. From these results and the percentage fuel savings per unit the nationwide fuel saving in BPDE was determined. The nationwide fuel saving, therefore, is large for bottoming cycles with low power because a greater percentage of the total power can be bottomed than for the high power bottoming cycles. It should be noted that the domestic fuel saving in BPDE for the classes of engines corresponding to each of the reciprocating sites except one (Topock, CA) are larger than the values obtainable with any gas turbine power block shown. In one instance the reciprocating site can produce 5 times the BPDE reported for the best gas turbine block of power.

TABLE 3-14

ENGINE CHARACTERISTICS, RECIPROCATING SITES

PIPELINE CO. & LOCATION	NO. OF UNITS	RATED POWER HP	HEAT RATE, BTU/HP-HR	MANUFACTURER	MODEL	TYPE	EXH. TEMP., °F	FLOW RATE PPS	TOTAL POWER, HP	B/C POWER HP
COLUMBIA GULF TR. COMPANY										
Alexandria, LA	4	3430	6420	C-B	LSV-16	SC	898	8.7	13720	1447
Corinth, MS	4	3430	6420	C-B	LSV-16	SC	898	8.7	13720	1447
Inverness, MS	4	3430	6420	C-B	LSV-16	SC	898	8.7	13720	1447
Rayne, LA	7	2000	7500	C-B	GMWA-8	SC	725	10.56	14000	1781
PACIFIC GAS & ELECTRIC CO.										
Topock, CA	10	3500	6900	C-B	GMW10C	SC	725	18.1	35000	4362
Hinkley, CA	1	3500	6900	C-B	GMW10	SC	725	18.1	3500	436
	2	7250	6850	C-B	W330	SC	750	40.0	14500	2064
Kettleman, CA	8	2050	7500	Clark	HBA8T	SC	645	11.4	16400	1696
	1	7250	6850	C-B	W330	SC	750	40.0	7250	1032

TABLE 3-14 (Cont'd.)

ENGINE CHARACTERISTICS, RECIPROCATING SITES

PIPELINE CO. & LOCATION	NO. OF UNITS	RATED POWER HP	HEAT RATE, BTU/HP-HR	MANUFACTURER	MODEL	TYPE	EXH. TEMP. °F	FLOW RATE PPS	TOTAL POWER, HP	B/C POWER HP
TEXAS GAS & TRANS. COMPANY										
Greenville, MS	4	2600	8310	Clark	HBAT10	SC	775	13.05	10400	1435
	4	1550	8020	Clark	HBAT6	SC	775	7.92	6200	871
	2	2000	7830	Clark	TLA6	SC	750	8.33	4000	430
Lake Cormorant MS	4	2000	7830	C-B	GMB8TF	SC	575	10.5	8000	580
	3	2000	7500	C-B	GMWA-8	SC	725	10.5	6000	756
Covington, TN	5	1500	7740	C-B	GMW6TF	SC	575	8.9	7500	614
	3	1500	7450	C-B	GMWA6	SC	725	8.9	4500	644
	2	2500	7310	C-B	GMWA10	SC	725	12.5	5000	603
Jeffersontown, KY	5	1500	7740	C-B	GMW6TF	SC	575	8.89	7500	614
	4	1500	7450	C-B	GMWA6	SC	725	8.89	6000	857

TABLE 3-15

SITE COMPATIBILITY, RECIPROCATING SITES

PIPELINE CO. & LOCATION	MANU-FACTURER	MODEL	B/C POWER, HP	COST $/HP	HEAT RATE BTU/HP-HR	DUTY HRS/YR	FUEL VALUE $/$10^5$ BTU	PAY-OUT YRS	% FUEL SAVING	U.S. POTEN. HP 10^6	U.S. FUEL SAVING BPDE
COLUMBIA GULF TR. COMPANY											
Alexandria, LA	C-E	LSV-16(1)	1447	459	6420	4380	2.21	7.38	10.5	3.523	10233
Corinth, MS	C-E	LSV-16(1)	1447	459	6420	4380	2.21	7.38	10.5	3.523	10233
Inverness, MS	C-E	LSV-16(1)	1447	459	6420	4380	2.21	7.38	10.5	3.523	10233
Rayne, LA	C-B	GMWA8	1781	422	7500	7509	2.21	3.39	12.7	3.595	14754
PACIFIC GAS & ELECTRIC CO.											
Topock, CA	C-B	GMW10C	4362	295	6900	8283	1.12	4.61	12.5	1.255	4664
Hinkley, CA	C-B	GMW10	436	741	6900	8283	1.12	11.6	12.5	5.974	22200
	C-B	W330	2064	398	6850	8283	1.12	6.26	14.2	3.723	15603
Kettleman, CA	Clark	HBA8T	1696	430	7500	8283	1.12	6.19	10.3	2.91	9686
	C-B	W330	1032	525	6850	8283	1.12	8.26	14.2	5.037	21110

TABLE 3-15 (Cont'd.)

SITE COMPATIBILITY, RECIPROCATING SITES

PIPELINE CO. & LOCATION	MANU-FACTURER	MODEL	B/C POWER, HP	COST $/HP	HEAT RATE BTU/HP-HR	DUTY HRS/YR	FUEL VALUE $/$10^6$ BTU	PAY-OUT YRS	% FUEL SAVING	U.S. POTEN. HP 10^6	U.S. FUEL SAVING BPDE
TEXAS GAS TRANS. COMPANY											
Greenville, MS	Clark	HBAT10	1435	460	8310	6661	1.01	8.23	13.8	3.377	16686
	Clark	HBAT6	871	562	8020	6508	1.01	10.65	14.0	5.185	25085
	Clark	TLA6	430	745	7830	5485	1.01	17.15	10.7	6.094	22000
Lake Cormorant, MS	C-B	GMB8TF	580	661	7830	6291	1.01	13.3	7.2	4.924	11960
	C-B	GMWA8	756	595	7500	5631	1.01	13.95	12.6	5.018	20432
Covington, TN	C-B	GMW6TF	614	646	7740	5460	1.01	15.15	8.2	5.211	14250
	C-B	GMWA6	644	634	7450	6291	1.01	13.38	14.3	6.219	28547
	C-B	GMWA10	603	651	7310	8283	1.01	10.66	12.1	6.91	26335
Jeffersontown, KY	C-B	GMW6TF	614	646	7740	5990	1.01	13.81	8.2	5.211	14250
	C-B	GMWA6	857	566	7450	6967	1.01	10.8	14.3	5.018	23035

Because there is so much reciprocating power to be bottomed the reciprocating engines were left in the final selections in spite of the fact that the maximum fuel saving per unit is only 14.3%.

Shown in Table 3-16 is the methodology for selecting the reciprocating engine site. Because the reciprocating bottoming cycle systems are low powered in many instances, have low utilization in others or low fuel values on still others, the payback periods reported in Table 3-15 range from 3.4 to 17.2 years whereas the better gas turbine sites have payback periods of as low as 2 years. In the selection of a demonstration site one could look ahead 2 to 3 years when the value of natural gas may increase. In a still later time period as the higher value natural gas becomes more available, it results in an increase in hours of utilization. In order to make a selection among the reciprocating sites a figure of merit was used which was the sum of three rankings, with the lowest value going to the best site. The first was a ranking corresponding to values in the first payback period column of Table 3-16, computed with a standard fuel value of $2.20/10^6 BTU. The second was a ranking corresponding to values in the second payback period column of Table 3-16, computed with the standard fuel value ($2.20/10^6 BTU) and with a utilization factor of 95% (8322 hrs/yr). The third was the ranking corresponding to the fuel savings in Table 3-16. Thus the payback period made two contributions to the figure of merit and the fuel savings also made one.

The result of this ranking procedure was for two types of Clark engines on the Texas Gas Transmission Corp. site at Greenville, MS to receive the number one ranking. Arbitrarily the four HBAT-10 engines were selected because they had the lower payback periods of the two, and three times the BPDE of the best gas turbines. The final selection is shown in Table 3-17 with the actual reported hrs/yr, the standard fuel value of $2.20/10^6 BTU and the throttled-back heat rate.

3.7 RECOMMENDED SITE SELECTIONS

Shown in Table 3-18 is the way in which the four selected sites relate to the contractual site selection criteria. For selections 1 and 2 the bottoming cycles will permit the shutting down of companion gas turbine units at each site and thus increase utilization of the bottomed unit. Selection 1 will have to be throttled back to 75% of the gas tur-

TABLE 3-16

FINAL SELECTION, RECIPROCATING SITES

PIPELINE CO. & LOCATION	MANU-FACTURER	MODEL	B/C POWER, HP	PAYBACK AT $2.20/ 10^6 BTU YRS.	RANK	PAYBACK AT $2.20/ 10^6 BTU & 8322 HRS/ YRS.	RANK	U.S. FUEL SAVING BPDE	RANK	FIGURE OF MERIT	RANK
TEXAS GAS TRANS. CO.											
GREENVILLE, MS	CLARK	HEAT10	1435	3.78	5	3.02	2	16686	9	16	1
	CLARK	HEAT6	871	4.89	7	3.82	6	25085	3	16	1
	CLARK	TLA6	430	7.87	19	5.19	18	22000	6	43	
LAKE CORMORANT, MS	C-B	GMB8TF	580	6.11	11	4.61	15	11960	14	40	
	C-B	GMWA8	756	6.40	14	4.33	12	20432	8	34	
COVINGTON, TN	C-B	GMW6TF	614	6.96	15	4.56	14	14250	12	41	
	C-B	GMWA6	644	6.14	12	4.64	16	28547	1	29	
	C-B	GMWA10	603	4.89	8	4.87	17	26335	2	27	
JEFFERSONTOWN, KY	C-B	GMW6TF	614	6.34	13	4.56	13	14250	13	39	
	C-B	GMWA6	857	4.96	9	4.15	10	23035	4	23	

TABLE 3-16 (Cont'd.)

FINAL SELECTION, RECIPROCATING SITES

PIPELINE CO. & LOCATION	MANU-FACTURER	MODEL	B/C POWER, HP	PAYBACK AT $2.20/ 10^6 BTU YRS.	RANK	PAYBACK AT $2.20/ 10^6 BTU & 8322 HRS/ YRS YRS.	RANK	U.S. FUEL SAVING BPDE	RANK	FIGURE OF MERIT	RANK
COLUMBIA GULF TR. COMPANY											
ALEXANDRIA, LA	C-B	LSV-16	1447	7.35	16	3.9	9	10233	17	42	
CORINTH, MS	C-B	LSV-16	1447	7.35	17	3.9	8	10233	16	41	
INVERNESS, MS	C-B	LSV-16	1447	7.35	18	3.9	7	10233	15	40	
RAYNE, LA	C-B	GMWA8	1781	3.38	4	3.06	3	14754	11	18	2
PACIFIC GAS & ELECTRIC OO.											
TOPOCK, CA	C-B	GMW10C	4362	2.35	1	2.33	1	4664	19	21	3
HINKLEY, CA	C-B	GMW10	436	5.91	10	5.88	19	22200	5	34	
	C-B	W330	2064	3.19	3	3.17	5	15603	10	18	
KETTLEMAN, CA	CLARK	HBA8T	1696	3.15	2	3.15	4	9686	18	24	
	C-B	W330	1032	4.21	6	4.18	11	21110	7	24	

TABLE 3-17

FINAL SELECTIONS, RECIPROCATING SITES

Pipeline Co. & Location	Texas Gas, Greenville, MS
Manufacturer	Clark
Model	HBAT10
B/C Power, HP	1435
Cost, $/HP	460
Throttled-Back Heat Rate, BTU/HP-Hr	8671
Duty, Hrs/Yr	6661
Fuel Value, $/10^6 BTU	2.20
Payout, Yrs.	5.2
% Fuel Saving	10.0
U.S. Potential, HP 10^6	3.4
U.S. Fuel Saving, BPDE	12100

TABLE 3-18

COMPARISON WITH CRITERIA

SITE SELECTION NO.		1	2	3	4
CRITERIA					
1.	TYPE OF PIPELINE	GAS	GAS	GAS	GAS
2.	TYPE OF ENGINE	SIMPLE CYC.	GAS TURB.	GAS TURB.	SUPERCHG
	MANUFACTURER & MODEL	INGERSOLL-RAND JP-125	COOPER-BESSEMER RT-125	COOPER-BESSEMER RT-125	CLARK HBAT10
3.	SYSTEM OPERATING PARAMETER AND DUTY CYCLE (HOURS/YEAR in 1976)	7064	6535	8280	6661
4.	SITE LOCATION NEARBY CITIES AND DISTANCE (MILES)	BURNEY, CA SACREMENTO 170	CHEMULT, OR PORTLAND 180 EUGENE 90	RAYNE, LA NEW ORLEANS 130 BATON ROUGE 70	GREENVILLE, MS MEMPHIS 135 JACKSON 90
5.	POTENTIAL FOR COST SHARING	YES	YES	YES	YES
6.	SEVERITY OF PROBLEMS (DISRUPTIONS)	NO PROBLEM	NO PROBLEM	NO PROBLEM	NO PROBLEM
7.	TIME FRAME SITE AVAILABLE	NO RESTRICTIONS------------	------------	------------	------------
8.	POTENTIAL FOR USE OF RECOVERED HEAT	NEED 75% OF POWER WITH BOT. CYCLE	NEED ADDITIONAL POWER	BOT. CYCLE POWER REPLACES RECIP.ENGINES	MUST THROTTLE PRIME MOV
9.	PIPELINE COMPANY	PACIFIC GAS & ELECTRIC CO.	PACIFIC GAS TRANS. CO.	COLUMBIA GULF TRANS. CO.	TEXAS GAS TRANS. CORP.

bine plus bottoming cycle power. More power is needed at site selection 2, therefore, full gas turbine plus bottoming cycle power will be used at this site. There are reciprocating units at site Selection 3 that would be shut down and thus make possible using the gas turbine unthrottled. Bottoming site Selection 4 will increase the utilization because these engines when bottomed would have the lowest heat rate at the site.

3.8 FINAL SITE SELECTIONS

The work statement for the subject contract originally called for the Contractor to recommend four natural gas compressor stations as potential demonstration sites for the pipeline bottoming cycle. The Department of Energy was then to select one site from the recommended sites. The remainder of the program was to be carried out using the selected site.

The recommended sites included three from different pipelines at which there were similar simple-cycle gas turbines having exhaust temperatures and flow rates in narrow ranges. Bottoming these gas turbines resulted in slightly exceeding the target value of 20% reduction in heat rate when the prime movers were throttled so that there was no increase in power at the site. A fourth pipeline also had such a gas turbine which was not originally selected because of low annual utilization and thus showed a high cost/benefit ratio. The gas turbines of these four sites fell into a group of gas turbines with a total installed power of between 554,000 and 804,000 HP, a large block of power providing a good potential for near term implementation through retrofitting.

It was the DOE view that there would be a definite added value in the program if several pipeline companies could be retained in the program and if the design problems for several gas turbines and several sites could be factored into the study. As a result the contract was changed to incorporate three pipeline companies, three sites, three different gas turbine prime movers, and three different ways to use the bottoming cycle power. Information of interest for the three sites is summarized in Table 3-19.

TABLE 3-19

DESCRIPTION OF COMPRESSOR SITES

Company[1]	Columbia Gulf Transmission Co.	Pacific Gas & Electric Co.	Texas Gas Transmission Corp.
Site Location	Rayne, LA	Burney, CA	Covington, TN
Nearby Airport	Lafayette, LA	Redding, CA	Memphis, TN
Distance (miles)	16	60	35
Gas Turbine To Be Bottomed			
Type	Simple	Simple	Simple
Rated Power,[2] HP	12500	12500	12500
Heat Rate,[2] Btu/HP-Hr	10060	9800	10890
Gas Turbine Manufacturer	Rolls Royce	General Electric	Pratt & Whitney
Power Turbine Manufacturer	Cooper Bessemer	Ingersoll-Rand	Cooper Bessemer
Exhaust Temp., °F	735	713	700
Flow Rate, lb/sec	158	140	153
Operating Time, 1976 hrs.	8280	8085	5420
Gas Compressor			
Type	Centrifugal	Centrifugal	Centrifugal
Manufacturer	Cooper Bessemer	Ingersoll-Rand	Cooper Bessemer
Model	RF2BB-30	CVP-30	RF2B-24
Other Gas Turbines	2 Each	1 Each	None
Type	Simple	Recuperated	--
Rated Power,[2] HP	12500	9100	--
Heat Rate,[2] Btu/HP-Hr	10060	8750	--
Reciprocating Engines	7 Each	None	5 Each, 3 Each, 2 Each
Rated Power, HP	2000	--	1500, 1500, 2500
Heat Rate, Btu/HP-Hr	8780		7740, 7450, 7310
Bottoming Cycle Function	Pump Natural Gas	Pump Natural Gas	Generate Electricity for Sale to TVA
Load	Gas Compressor in Parallel With Reciprocating Engine Driven Compressors	Same Gas Compressor as Driven by Bottomed Gas Turbine	Generator
Power[3]	5976 HP	5397 HP	3694 KW

(1) Alphabetical Order.
(2) At 80°F, 1000 Ft Altitude
(3) Based on gas turbine operation at site conditions

3.8.1 COLUMBIA GULF TRANSMISSION CO., RAYNE, LA SITE

At the Rayne site there are three Cooper-Bessemer RT-125 gas turbines, with Rolls Royce Avon 1533-76 gas generators. In addition there are seven 2000 HP GMWA-8 Cooper-Bessemer reciprocating units, having a heat rate of 8780 BTU/HP-hr and located on a separate pipeline. It is planned that the bottoming cycle will drive a compressor in parallel with the reciprocating engines and some of the reciprocating engines will be shut down. Because of the high heat rate of the reciprocating engines the bottoming cycle will save substantial fuel.

3.8.2 PACIFIC GAS & ELECTRIC CO., BURNEY, CA

At the Burney site there is an Ingersoll-Rand JP-125 gas turbine consisting of a General Electric Model 7-LM1500 GB 101, 7300 RPM gas generator driving an Ingersoll-Rand 5000 RPM model HP-125 power turbine. At the same site there is a General Electric Model 3912R recuperated gas turbine. There are no reciprocating engines at this site. The bottoming cycle will increase the power of the IR JP-125 sufficiently that the recuperated gas turbine can be shut down, increasing the utilization of the bottomed gas turbine to nearly full time. The bottomed gas turbine will have a heat rate that is substantially less than that of the recuperated gas turbine.

3.8.3 TEXAS GAS TRANSMISSION CORP., COVINGTON, TN

At the Covington site there is a Pratt & Whitney GG3C-4 gas generator with a Cooper Bessemer power turbine. In addition, there are three models of Cooper-Bessemer reciprocating engines (GMW-6TF, GMWA-6 and GMWA-10) for a total of ten engines. These reciprocating engines have heat rates between 7310 and 7740 BTU/HP-hr which are very competitive with the anticipated heat rate of the gas turbine with bottoming cycle. Thus, replacing reciprocating engines with the bottomed gas turbine would not save much, if any, fuel. In addition, the gas turbine is underutilized because of its high heat rate. At this site, therefore, it was decided that the gas turbine bottoming cycle should be used to drive an electric generator and the power should be sold to the TVA.

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SECTION 4

PRELIMINARY SYSTEM DESIGN

4.1 INTRODUCTION

The purpose of this task is to develop preliminary designs of bottoming cycle systems for several sites that will provide the basis for assessments of technical and economic feasibility. Data developed during this task permitted estimates to be made of the cost of fabricating and installing bottoming cycle systems on typical gas compressor sites. All of the subsequent tasks will use information developed from the preliminary designs.

The preliminary designs identified aspects of the bottoming cycle equipment relative to the technical feasibility of such systems. These designs incorporated environmental and safety regulations including those for the specific sites. The preliminary designs also incorporated operational reliability and maintenance requirements which were compatible with the requirements for the existing prime movers. Bottoming cycle performance was factored into an assessment of the potential gas saving in the industry. Any undeveloped technologies required to produce bottoming cycle equipment became apparent during the preliminary design.

In this section the working fluid of the Rankine bottoming cycle is discussed and an optimized cycle is defined. The performance of the bottoming cycle system at each of three sites is presented. The preliminary designs of system components such as turbines, vapor generators and condensers are discussed along with the designs required to install the systems on three sites.

4.2 FLUID SELECTION

In selecting the working fluid for the Rankine bottoming cycle, five criteria were established to carry out the selection. The purpose of using the bottoming cycle is to improve the performance of the installation and so the amount that a given fluid contributes to the reduced heat rate was taken as one important criterion. Since the working fluid must be contained and must flow through apparatus made of various materials, it was necessary to establish the compatibility of the fluid with the ordinary materials of construction for Rankine cycles. Most of the organic compounds that are usable in the Rankine bottoming cycles have upper temperature levels at which the fluids breakdown into other components and therefore are no longer stable. These breakdown temperatures must be higher than the maximum fluid temperature expected in the Rankine bottoming cycle in order to assure that the working fluid will operate in the same way throughout the life of the Rankine bottoming cycle. Two other aspects of the fluid selection are the safety requirements and the toxicity of the fluids to personnel that are operating the pipeline systems.

Shown in Table 4-1 are a number of fluids which were studied by Miller in Reference 11 as possible Rankine cycle working fluids. All these fluids are of the organic type with the exception of water. Shown in the last column of the table are Rankine bottoming cycle efficiency levels attainable with the various fluids using a turbine inlet temperature of 450°F, a condensing temperature of 150°F, and turbine efficiency of 75% and regenerator effectiveness of 80%. As would be expected water has a much lower efficiency in this type of a system than the other organic fluids used; that is the primary reason for using organic compounds as the working fluids in bottoming cycles which have very limited waste heat temperature levels. It was necessary to limit the fluids that were investigated in this study by using the criteria that were stated above. Benzene was immediately rejected because it has been shown to be carcinogenic. Toluene, which shows a relatively high cycle efficiency, was not ruled out even though it has a certain level of flammability. Hexafluorobenzene, which is nonflammable, was also considered to be a potential working fluid for systems of this type. Flutex PP3 is nonflammable and has a relatively good cycle efficiency, but thermodynamic data over the range of temperatures and pressures required for a Rankine bottoming cycle were not available and so it was ruled out. Pyridine, although flammable, produces a rather high cycle efficiency and was not immediately

TABLE 4-1

MAXIMUM USE TEMPERATURE RANKINGS
OF CANDIDATE WORKING FLUIDS

Use Temp., °F		Fluids			Cycle*
Max.	Min.	No.	Name	Safety	Eff. %
1050	32	1	Water	Nonflammable	8.7
800	42	5	Benzene	Flammable	20.0
750	-139	3	Toluene	Flammable	20.8
750	41	6	Hexafluorobenzene	Nonflammable	18.9
700	- 67	10	Flutex PP3	Nonflammable	17.9
670	- 43	9	Pyridine	Flammable	20.9
630	- 50	4	Chlorobenzene	Flammable	21.3
625		11	2,2,2-Trifluoroethanol	Ignitable	17.4
500	- 37	7	Thiophene	Flammable	20.3
450	-255	2	Isobutane	Flammable	13.8
300	- 31	8	Freon 113	Nonflammable	16.0

*Turbine inlet temp.: 450°F, condensing temp.: 150°F; turbine efficiency: 75%, regenerator efficiency: 80%.

ruled out as a working fluid. Chlorobenzene was ruled out because it not only has a low maximum use temperature of 630°F, but it is also flammable. Trifluoroethanol (Fluorinol), has about the same maximum use temperature but it is only ignitable with difficulty. When the source of ignition is taken away the flame readily goes out. For that reason, and because this fluid is used in two other Department of Energy developed Rankine systems, it was not ruled out. The remaining fluids, thiophene, isobutane, and Freon 113 were all ruled out of the study because the maximum use temperatures of these fluids are too close to the desired turbine inlet temperature in the system (500°F). The actual organic fluids which were studied in this program are four; namely, toluene, fluorinol-50, RC-2 and RC-1. Toluene is a very prevalent hydrocarbon, made more prevalent by the fact that benzene has been taken off the market for certain uses because of its carcinogenic qualities. Fluorinol-50 is 50 molar percent of 2,2,2-trifluoroethanol and 50 molar percent of water. RC-1 is a 60 percent mole fraction of pentafluorobenzene and a 40 percent mole fraction of hexafluorobenzene. RC-2 is a 65 percent mole fraction of H_2O and a 35 percent mole fraction of methyl pyridine. These last two working fluids were recommended by Miller in Reference 12 as the optimum working fluids for automotive Rankine cycles.

Shown in Table 4-2 are some comparative thermodynamic performance data on the four fluids studied in this program as regards cycle efficiency, ultimate horsepower and fluid conditions. The net bottoming cycle horsepower excludes the power required to drive the air cooled condenser fans, the boiler feed pump and other accessories required to operate the Rankine Cycle. As can be seen the largest bottoming cycle net horsepower was obtained with toluene and is closely followed by RC-2 and RC-1 and finally Fluorinol-50. The values of the exhaust temperatures from the vapor generator are shown in the last column and the indications are that the RC-1 obtains the largest amount of heat from the exhaust stack but is penalized by the fact that a large amount of power is required to drive the liquid feed pump compared to some of the other fluids. It should be noted that these cycles, although not all of them are optimized, have a performance difference of about 5%. This small a difference does not constitute a basis for selection among working fluids. Indeed the preliminary selection of these four fluids has reduced the significance of performance as a criterio[n]

<u>TABLE 4-2</u>

<u>PERFORMANCE COMPARISON OF CYCLES WITH VARIOUS WORKING FLUIDS</u>

	Turbine Inlet Conditions	Cycle Efficiency	Net Bottoming Cycle Horsepower	Feed Pump Horsepower	Exhaust Temperature °F
Toluene	270 psia/500°F	.1986	5067	147	276
F-50	400 psia/500°F	.1875	4796	111	275
RC-2	270 psia/500°F	.1919	5002	76	267
RC-1	600 psia/500°F	.1704	4923	437	220

These cycles were not optimized and performance difference is about 5%.

and the selection must be made on the basis of other important factors.

A further evaluation of these fluids is shown in Table 4-3. Performance from the previous table is repeated here but in addition materials compatibility, thermal stability tempeatures, safety, toxicity, and costs per unit weight are also recorded. With regard to materials compatibility, toluene has no known compatibility problems. Fluorinol-50 is quite compatible with conventional containment materials except that corrosion has been experienced when air and an antiwear additive were present. RC-2 was found to be incompatible with SAE 4130 steel boiler tubing material, and with aluminum in Reference 12 by Miller et al. RC 1 was tested for 1000 hours in 4130 steel, which is a low alloy steel, with no difficulties and this laboratory fluid is therefore considered to be compatible with containment materials but certainly further materials testing should be carried out before it is committed in the design of a Rankine bottoming cycle. In regard to thermal stability temperature, the one with the lowest value is Fluorinol-50 closely followed by RC-2; RC-1 and toluene have stability temperatures of 750°F or above. Only one of the fluids is nonflammable, RC-1. Fluorinol-50 is classed as ignitible rather than designated as flammable. When the source of ignition is removed the flame goes out. Toluene and RC-2 are flammable which must be taken into account in the use of the fluid in systems of this type. Most of the flammability data can be found in Reference 12. From the standpoint of toxicity a standard has been recommended for toluene which would permit a 100 ppm concentration of toluene in air based on a time weighted average with a maximum of 200. RC-2 is about as toxic as Toluene. Fluorinol-50 has a recommended time weighted average concentration of only 5 ppm. The least toxic of the four fluids studied is RC-1. The costs of these fluids are important, especially for the laboratory type fluids such as RC-1. The cost of the charge of fluid for a bottoming cycle system is a significant portion of the system costs for RC-1.

More information on the toxicity of these four fluids is presented in Table 4-4. In this table an attempt was made to rank the fluids according to their toxicity, from any source that was available which provided toxicity measurements. The same tests were not run on all four fluids and the data came from several sources as can be seen by the notes at

TABLE 4-3

FLUID EVALUATION

	Toluene	F-50	RC-2	RC-1
Performance	5067 HP	4796 HP	5002 HP	4923 HP
Materials Compatibility	good	good in absence of air	incompatible with Al or steel	good tested in 4130 steel
Thermal Stability	700°F	> 625°F	> 670°F	> 700°F
Safety	flammable	ignitable	ignitable	nonflammable
Toxicity	100 ppm TWA 200 ppm max.	toxic	comparable to toluene	least toxic
Cost $/lb.	0.10	7 now < 3 future	0.50	30 now < 5 future

TABLE 4-4

TOXICITY RANKING (descending order)

	orl mus* LD50	ihl* rat LC50	ihl* rat LC Lo
F 50**	432 mg/kg [13]***	897 ppm/6H [14]	-
Toluene	-	-	4000 ppm/4H [13]
RC-2**	916 mg/kg [13]	8000 ppm/4H [12]	5435 ppm/4H [13]
RC-1	-	6000 ppm/4H [14]	-
	-	16000 ppm/4H [12]	-
Freon 22	-	-	250000 ppm/4H [13]

* orl = oral

mus = mouse

ihl = inhalation

** Data for 2,2,2 trifluoroethanol and 2 methylpyridine were converted to F-50 and RC-2 concentrations.

*** Numbers in parentheses indicate references from Section 11.

the bottom of the table. However, a comparison can be made between F-50 and RC-2 in oral tests with mice where the lethal dose for 50% of the population was 432 milligrams per kilogram for F-50 and 916 milligrams per kilogram for RC-2. Further comparisons between fluids are given by considering the second column in which inhalation in rats was studied and the results are shown of concentrations for which 50% of the population survived. If we take the two values that came from Reference 14 we see the F-50 has 897 ppm in 6 hours whereas the RC-1 has 6000 ppm in 4 hours so certainly RC-1 is less toxic from this standpoint than the F-50. Now, if one looks at the data from Reference 12 we see that RC-2 has 8000 ppm for 4 hours whereas RC-1 has 16,000 ppm for 4 hours. For inhalation by rats, column 3, the lowest published lethal concentration for toluene is 4000 ppm in 4 hours compared with 5435 ppm for 4 hours for RC-2. Freon 22 is very much less toxic than any of these fluids.

To summarize, the following reasons are set forth for the selection of toluene for the design studies in this program. First of all, toluene gives more bottoming cycle power from the same waste heat stack than any of the other fluids. Toluene is a very common solvent used in paints and inks, is handled every day by many people, sometimes in large quantities, is very well documented and the precautions that need to be taken are well understood. Although other fluids could provide about the same system performance as Toluene with suitable system changes and would have only a small effect on the program, Toluene is the working fluid of a DOE-supported organic Rankine system development and will thus get considerable developmental attention. Toluene has to be handled like other flammable liquids but is shipped in many types of containers without any particular precautions. However, it has to be used in ventilated areas which are away from excessive heat and ignition sources. There is some toxicity associated with the toluene if the concentration is too high. As indicated above, the standard has been suggested at the level of 100 ppm for an 8 hour day and 40 hour week with a ceiling of 200 ppm. Toluene is nonreactive with ordinary metallic containment materials and has a thermal stability temperature of 700°F[15].

4.3 CYCLE OPTIMIZATION

Having selected the working fluid as toluene, the next step in the program was to carry out a cost optimization of the bottoming cycle in order to select the working conditions which would be most economical. Shown in Figure 4-1 is a schematic diagram of the pipeline bottoming cycle. At the lower left can be seen a gas turbine with a free power turbine which drives the natural gas compressor. The exhaust gas from the power turbine is ducted to a vapor generator in which toluene is heated and vaporized and sent to a vapor turbine. The turbine drives a load through a gear box. In the case of the three sites selected the load can be a generator, a small centrifugal compressor, or it could be the same large compressor driven by the gas turbine. After the toluene leaves the vapor turbine it can be passed through a regenerator so that some of the superheat can be removed and thus added to the organic working fluid as a liquid on its way to the vapor generator. After passing through the regenerator the toluene vapor is sent to the condenser where it is liquefied and dumped into a surge tank. The boiler feed pump is supplied with toluene liquid from the surge tank which is pumped through the regenerator into the vapor generator completing the circuit of the toluene through the Rankine bottoming cycle system. The controls are described in Section 4.5.4.

A number of major design decisions and assumptions were arrived at to make it possible to carry out the design and optimization of the Rankine bottoming cycles in this study. First of all, all the prime movers are simple cycle aircraft derivative gas turbines with free power turbines and as a consequence of this selection the exhaust gas flow rate is in a range from 147 to 168 lbs/sec and the exhaust temperature at rated conditions is between 700-740°F. Since the fuel for the gas turbines to be bottomed is natural gas, which burns cleanly and has negligible sulfur in it, it was decided that fouling of the heat exchanger with unburned hydrocarbons or corrosion of the heat exchanger surfaces from the condensing of sulphuric acid should not be considered in the design. Inasmuch as toluene was selected as the working fluid, the design must take into account the fact that the fluid is flammable and that certain minimal health precautions are necessary. The material compatibility was considered to be no problem and the fact that the stability temperature was around 750°F, which is just slightly higher than the exhaust gas temperature from the gas turbines, it was felt that with proper control the stability limit would not be exceeded for toluene as the working fluid.

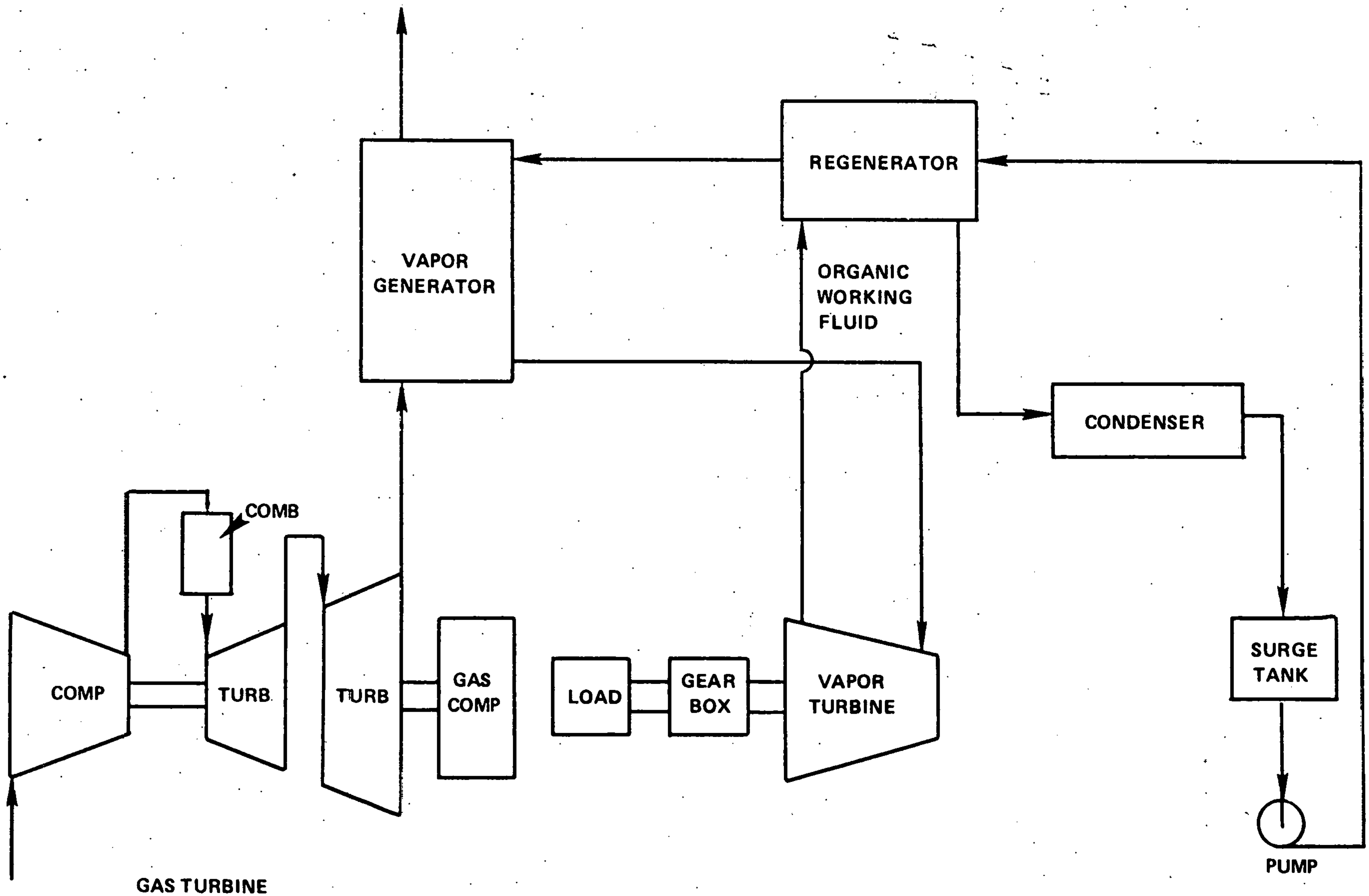


Figure 4-1. Pipeline Bottoming Cycle.

Cooling towers were not considered for the pipeline bottoming cycle design for a number of reasons. Maintenance requirements for cooling towers are difficult. Rust and corrosion seem to take place in hard-to-inspect regions. By the time a rust spot is found so much damage has occurred that a major expense is required for rebuilding the cooling tower. Cooling towers generate a brine from the salt residue formed in the evaporation of the water. The brine must be disposed of in an environmentally acceptable manner periodically. Cooling towers require a water supply; however, many compressor sites are without adequate water. The makeup water for a cooling tower must be treated; this requires a knowledgable person at each site where one is operated, adding to the operating cost. The aggregate of these costs associated with a wet cooling tower are not justified in the size needed for the bottoming cycle of the size under study (5000-6000 HP).

In carrying out the selection study for the sites, it was necessary to make certain calculations with regard to the anticipated saving of fuel and a simplified optimization was carried out to determine the value of certain design variables giving the most cost effective system. The result of that optimization was the establishing of a reference cycle which was used during part of the preliminary design. Shown in Table 4-5 are the pressure drops that were used in the various parts of the reference cycle to obtain this performance and also the other assigned values that were used in the analysis. The turbine efficiency was based on a conceptual seven stage axial turbine design described in Section 4.5.1. A turbine design computer program, with loss calculations based on the Ainley and Mathieson method, was used to estimate turbine efficiency. Shown in Figure 4-2 is the reference pipeline bottoming cycle with the energy transferred values shown on the figure. In excess of 52 million BTU/hr is transferred in the vapor generator and in excess of 39 million BTU/hr is rejected by the condenser. The load receives 4890 HP including the losses in the gearbox; a total of 97 kilowatts are required to drive the feed pump, and 259 kilowatts are required to drive the fans for the condenser. The regenerator transfers nearly 6 million BTU/hr to the toluene liquid on its way to the vapor generator. Shown in Figure 4-3 is the same schematic diagram showing the pressures and the temperature of 700°F which is reduced to 333°F in a vapor generator. It shows the turbine inlet temperature and pressure

TABLE 4-5

ASSIGNED VALUES FOR THE REFERENCE CYCLE

Working Fluid: Toluene

Ambient Air Temperature = 68°F

Fan Power Required = 2% of the total rejection

Regenerator Effectiveness	0.50	
Turbine Efficiency	0.86	
Pump Efficiency	0.70	
Gearbox Efficiency	0.98	
Electric Motor Efficiency	0.90	
Pressure Drops	$\Delta P/P^*$	ΔP, psi
Vapor Generator	0.0815	24.0
Regenerator (Hot Side)	0.0275	0.095
Regenerator (Cold Side)	0.0121	3.6
Condenser	0.0202	0.068
Vapor Gen. to Turbine	0.0025	0.675
Turbine to Regenerator	0.015	0.052
Condenser to Pump	0.0015	0.005
Pump to Regenerator	0.0005	0.15

*The pressure drop, ΔP, across a specific component is normalized with respect to the pressure upstream of that component.

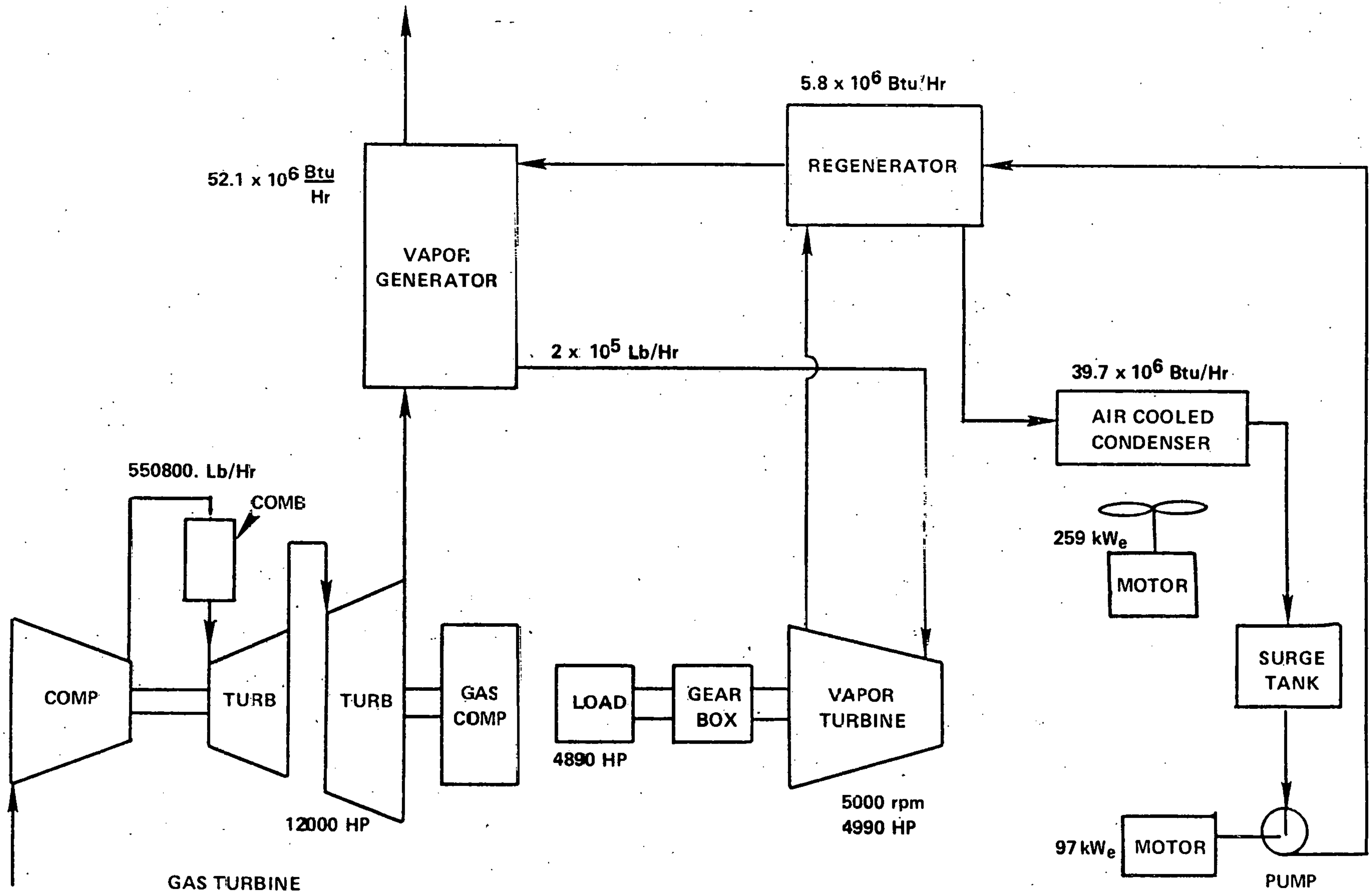


Figure 4-2. Pipeline Bottoming Toluene Reference Cycle.

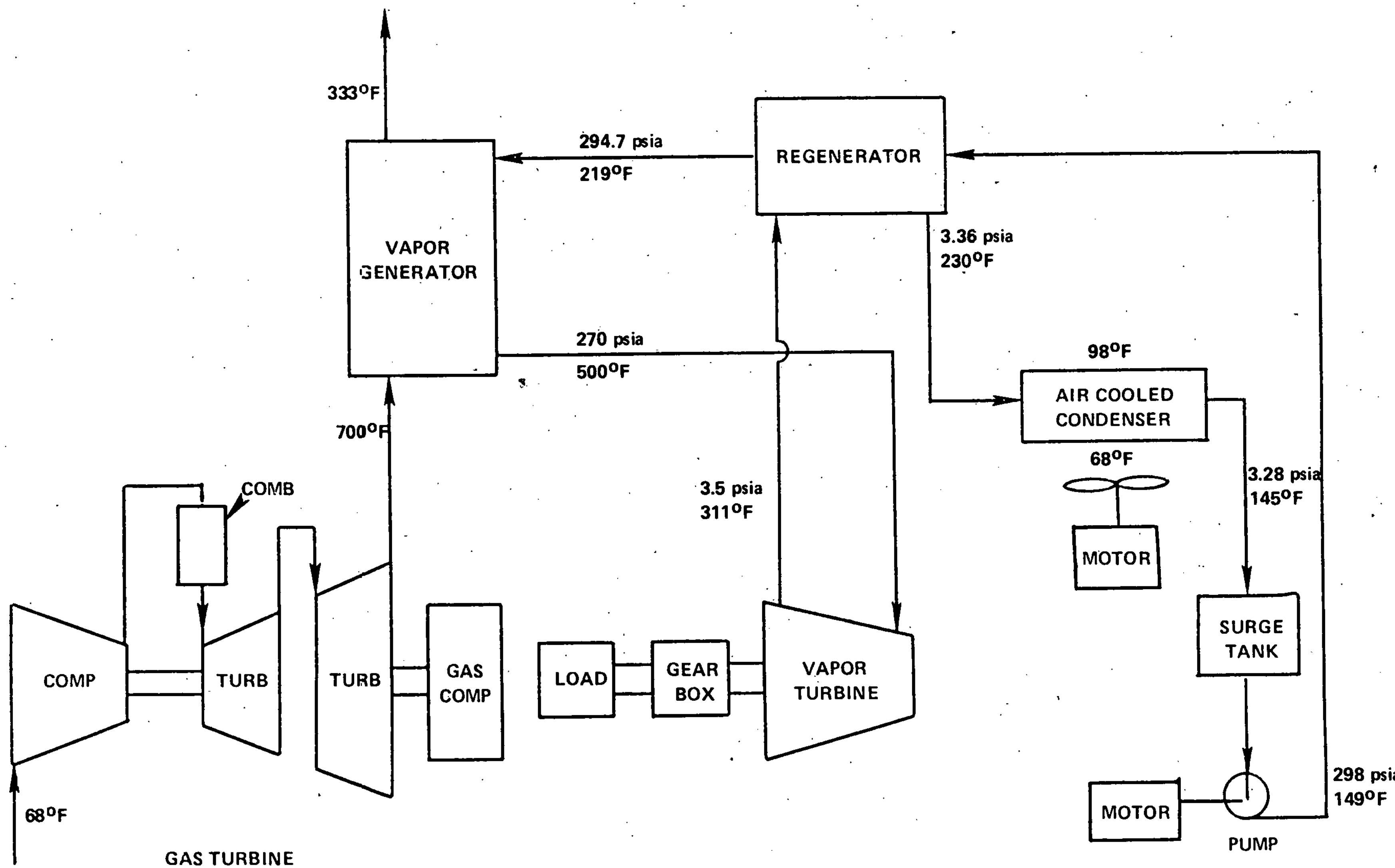


Figure 4-3. Pipeline Bottoming Toluene Reference Cycle.

to be 500°F and 270 psia, respectively. For this cycle an ambient air temperature of 68°F was used and this temperature rises to 98°F as the cooling air passes through the air cooled condenser. The air cooled condenser operates down to a pressure of 3.28 psi and with 5°F of subcooling the temperature of the liquid toluene going to the surge tank is 145°F. The flow leaves the vapor turbine at 311°F and is desuperheated in the regenerator and the air cooled condenser down to a value of 150°F. Shown in Table 4-6 are the particulars of the toluene reference cycle discussed above.

The reference cycle was utilized in the preliminary design to make the initial sizing of the Rankine bottoming cycle components for the purpose of cost estimation. The components for the near optimum design were supplied to the pipeline companies in order that they might make installation studies of the bottoming cycle in their respective sites. In parallel with this effort by the pipeline companies, the Rankine bottoming cycle cost optimization was carried out. In the optimization process there were five variables which had to be determined. These are the turbine inlet temperature, the turbine inlet pressure, the condensing temperature, the recuperator effectiveness, and the pinch point temperature difference. The pinch point temperature difference is the minimum difference in temperature between the exhaust gas flowing through the vapor generator and the vaporizing fluid which is in the tubes of the vapor generator.

In the determination of the optimum cycle the costs were estimated for the reference cycle described above. Then these costs were used as a starting point to generate cost algorithms for each of the major components in the system. These algorithms gave systematic and realistic variations of the component costs as the five prime variables mentioned above were varied over their ranges. Shown in Table 4-7 are the cost algorithms for the vapor generator, the vapor turbine, the vapor condenser, the feed pump, controls and instrumentation, auxiliary equipment and the equipment skid. The equipment skid contains the surge tank, the feed pump, the regenerator, if there is one, and auxiliary equipment.

Shown in the following series of figures are the variations in the cost per unit horsepower of the various Rankine bottoming cycle systems

TABLE 4-6

TOLUENE REFERENCE CYCLE DATA

Turbine Inlet Pressure		270 psia
Saturation Temperature		495°F
Turbine Inlet Temperature		500°F
Condensing Temperature		150°F
Toluene Flow Rate		200,000 lb/hr
Heat Added to Working Fluid		52.1×10^6 BTU/hr
Heat Transferred in Regenerator		5.8×10^6 BTU/hr
Heat Rejected by Working Fluid		39.7×10^6 BTU/hr
Parasitic Power Requirements (Motor Efficiency, 0.9)		
Pump	97 KWe	
Fan	259 KWe	
Total	356 KWe	
Turbine Power	4990 HP	
Net Power to Load (Gearbox Efficiency, 0.98)	4890 HP	

TABLE 4-7

COST ALGORITHMS FOR THE BOTTOMING CYCLE COMPONENTS

Component	Component Cost	Explanation
1. Vapor Generator	$C_{VG} = a\ A_{HT} + b\ V^c + d$	C = Cost, \$ A_{HT} = Vapor Generator Heat Transfer Area, ft^2 V = Heat Exchanger Volume, ft^3 a,b = Constants
2. Vapor Turbine	$C_{Turb} = a + b\ (HP_T)$	HP_T = Turbine Horsepower, HP a,b = Constants
3. Condenser	$C_{Cond} = a + b\ A_{HT}$	A_{HT} = Condenser Heat Transfer Area, ft^2 a,b = Constants
4. Feed Pump	$C_{Pump} = a\ (HP_P)^b$	HP_P = Pump Horsepower, HP a,b = Constants
5. Controls and Instrumentation	$C_{Cont} = a$	a = Constant
6. Auxiliary Equipment	$C_{Ax.Eq} = a\ (HP_T)^b$	HP_T = Turbine Horsepower, HP a,b = Constants
7. Equipment Skid	$C_{Skid} = \left[C_{Turb} + C_{Pump} + C_{Cont} + C_{Ax.Eq}\right]$	

as the prime variables are changed through the usable range. These costs are normalized by dividing by the cost per unit horsepower of the reference cycle. Also shown in some of these figures is the variation of the horsepower output of the bottoming cycles as a function of the prime variables. Shown in Figure 4-4 is the variation of the Rankine bottoming cycle cost per unit horsepower as a function of the turbine inlet pressure at inlet temperatures of 600°F. The other conditions at which the Rankine cycles were operated are indicated in the face of the figure. This figure indicates that as the turbine inlet pressure increases the cost per unit power decreases and the lowest cost per unit power occurs at the saturation pressure. Shown in Figure 4-5 is a similar plot in which the turbine inlet temperature is varied at constant inlet pressure. Here again the indications are that the lowest cost per unit horsepower is obtained at the saturation conditions. Shown in Figure 4-6 is the variation in the gross power of the bottoming cycle as a function of the turbine inlet pressure at an inlet temperature of 600°F indicating that the gross power is highest at the saturation pressure. Shown in Figure 4-7 is the variation of the bottoming cycle horsepower with the variation in turbine inlet temperature at a constant pressure and again the highest horsepower is obtained at the saturation temperature. It is frequently advantageous to use waste heat from the expander to preheat the vapor generator feed in a recuperator (also referred to herein as a regenerator). In this application, however, this process intrinsically reduces the amount of heat that can be extracted from the gas turbine exhaust and hence limits the contribution of the primary heat source to the bottoming cycle power. For that reason, the recuperator was explored over a range of values of effectiveness (zero refers to no recuperator). Shown in Figure 4-8 is the variation in the cost per unit power with recuperator effectiveness. It can be seen that the cost per unit power gradually decreases as the effectiveness approaches zero. The horsepower generated by the bottoming cycle is plotted against recuperator effectiveness in Figure 4-9. Here it can be seen that the minimum horsepower is reached at around 0.5 effectiveness and that the power increases slightly as the recuperator effectiveness drops to a value of zero. The reason that the use of no recuperator is shown to be advantageous in these figures is that with natural gas as the fuel no limitation on the minimum stack temperature

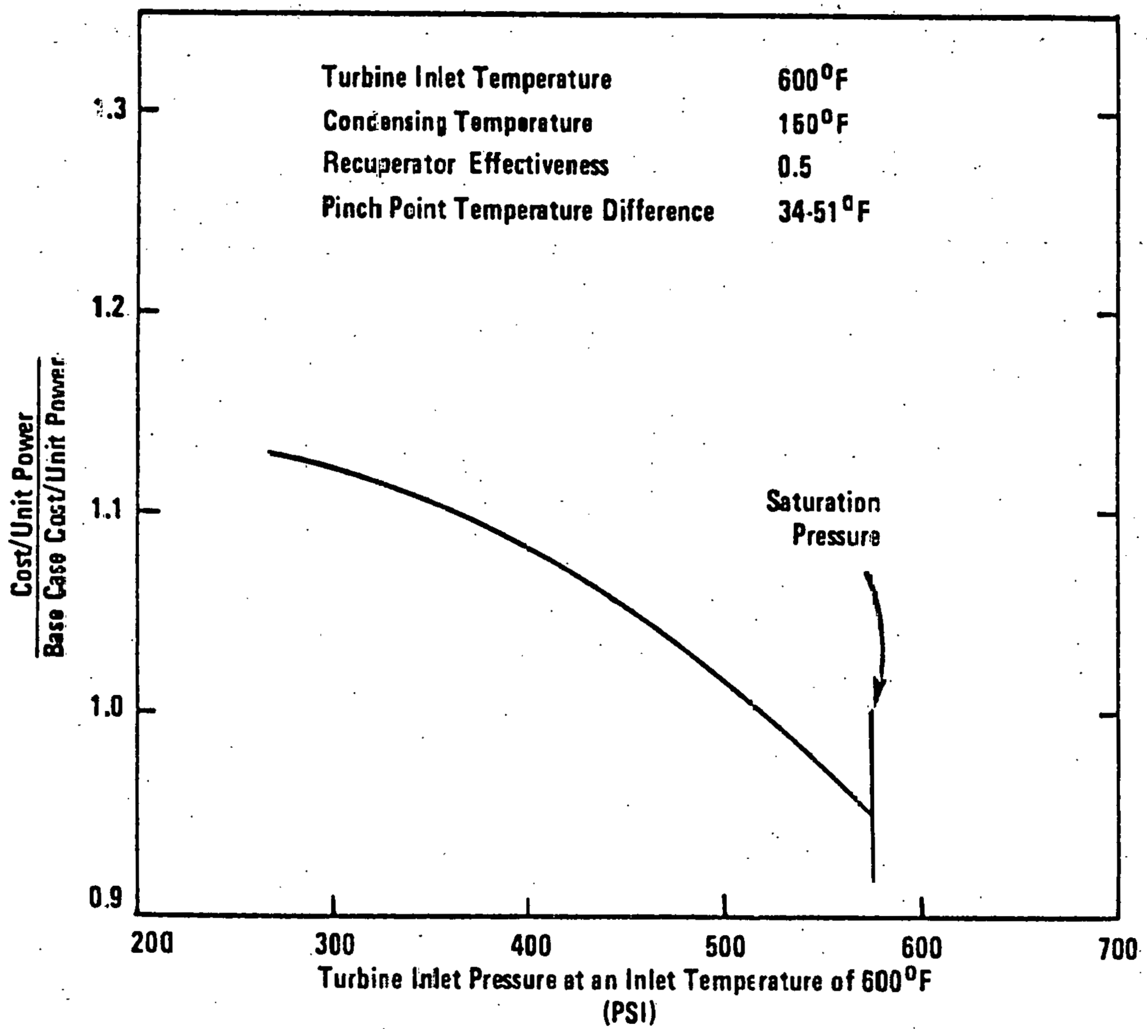


Figure 4-4. Effect of the Turbine Inlet Pressure on the Cost Per Unit Power.

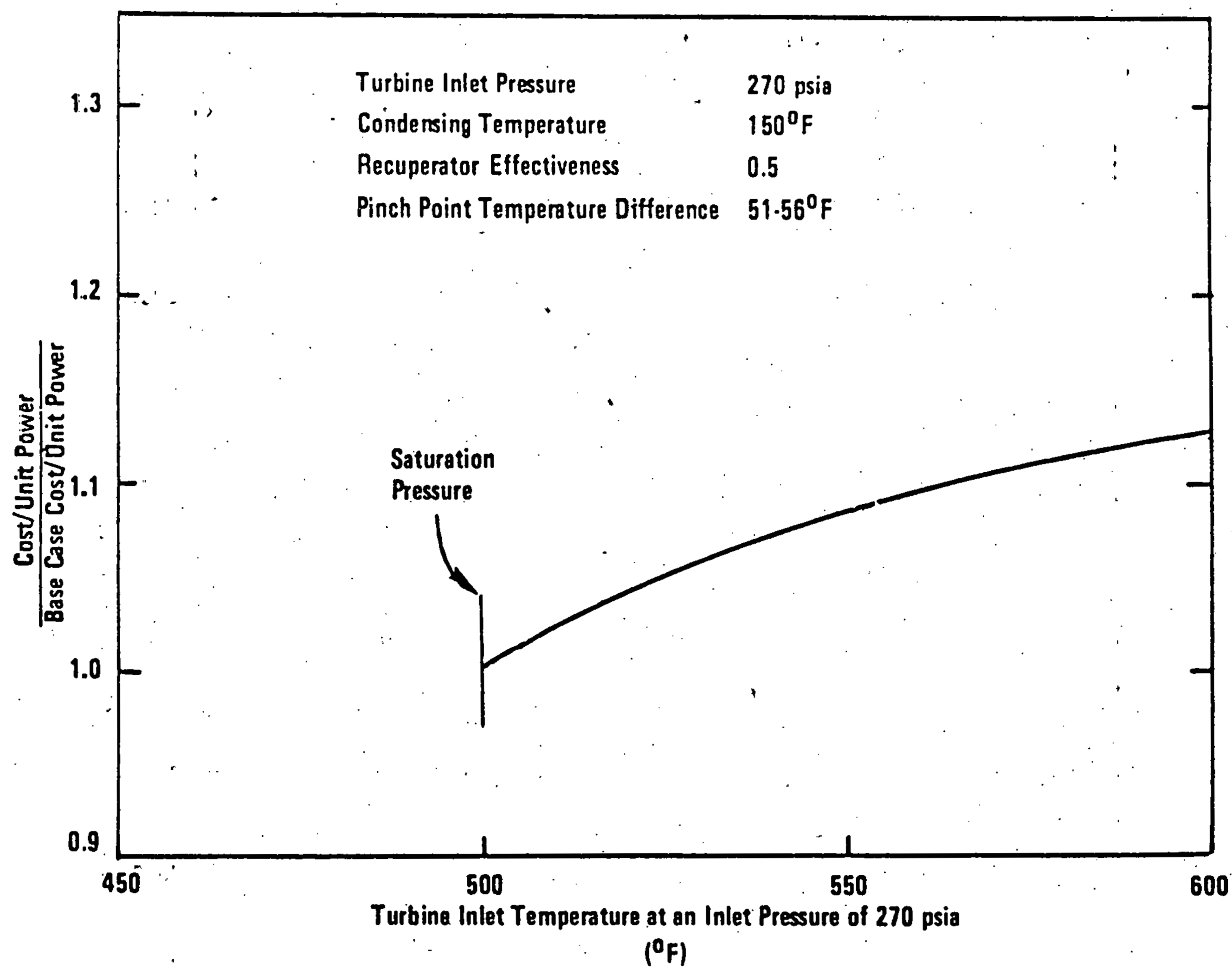


Figure 4-5. Effect of the Turbine Inlet Temperature on the Cost Per Unit Power.

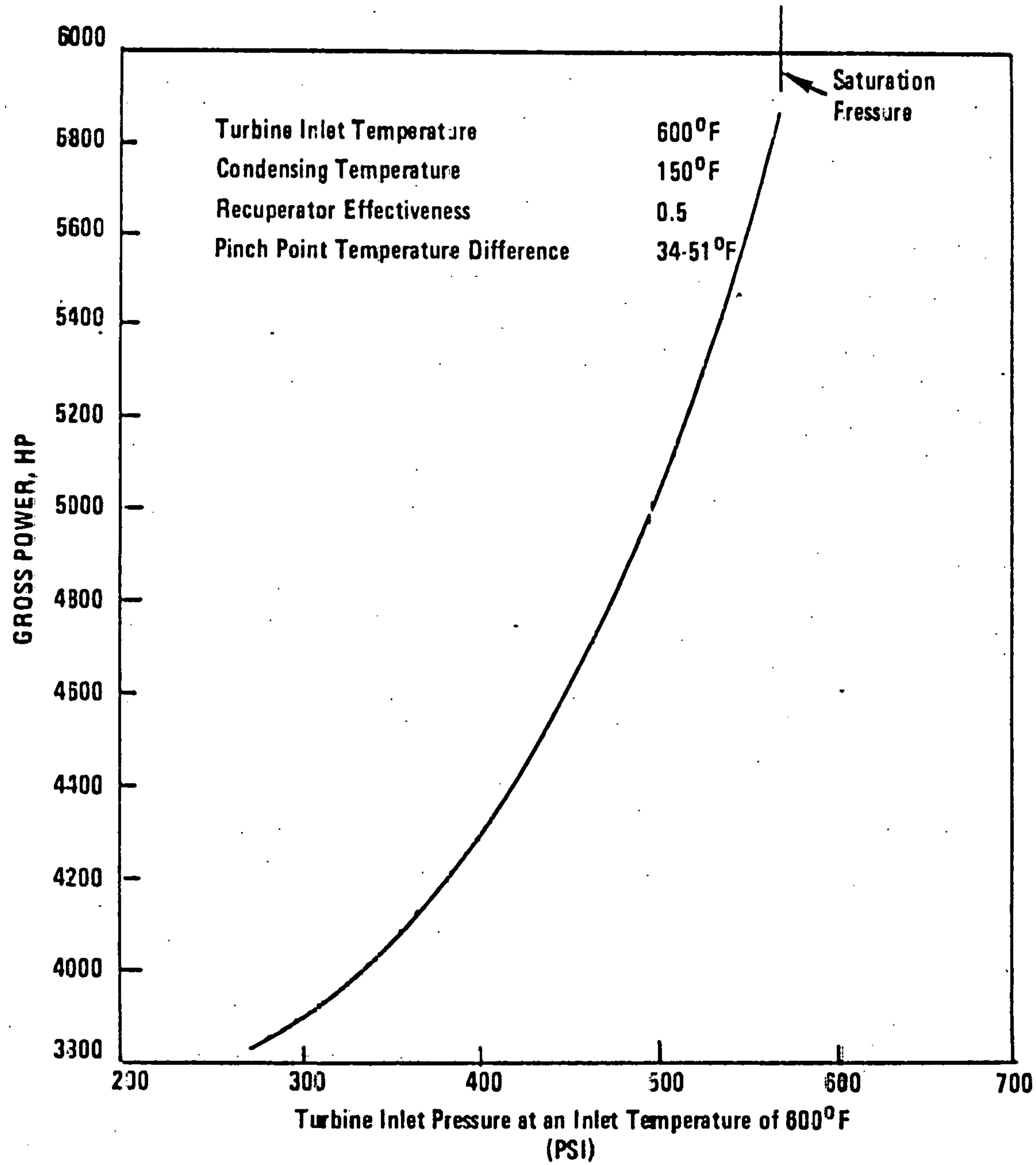


Figure 4-6. Effect of the Turbine Inlet Pressure on the Gross Horsepower.

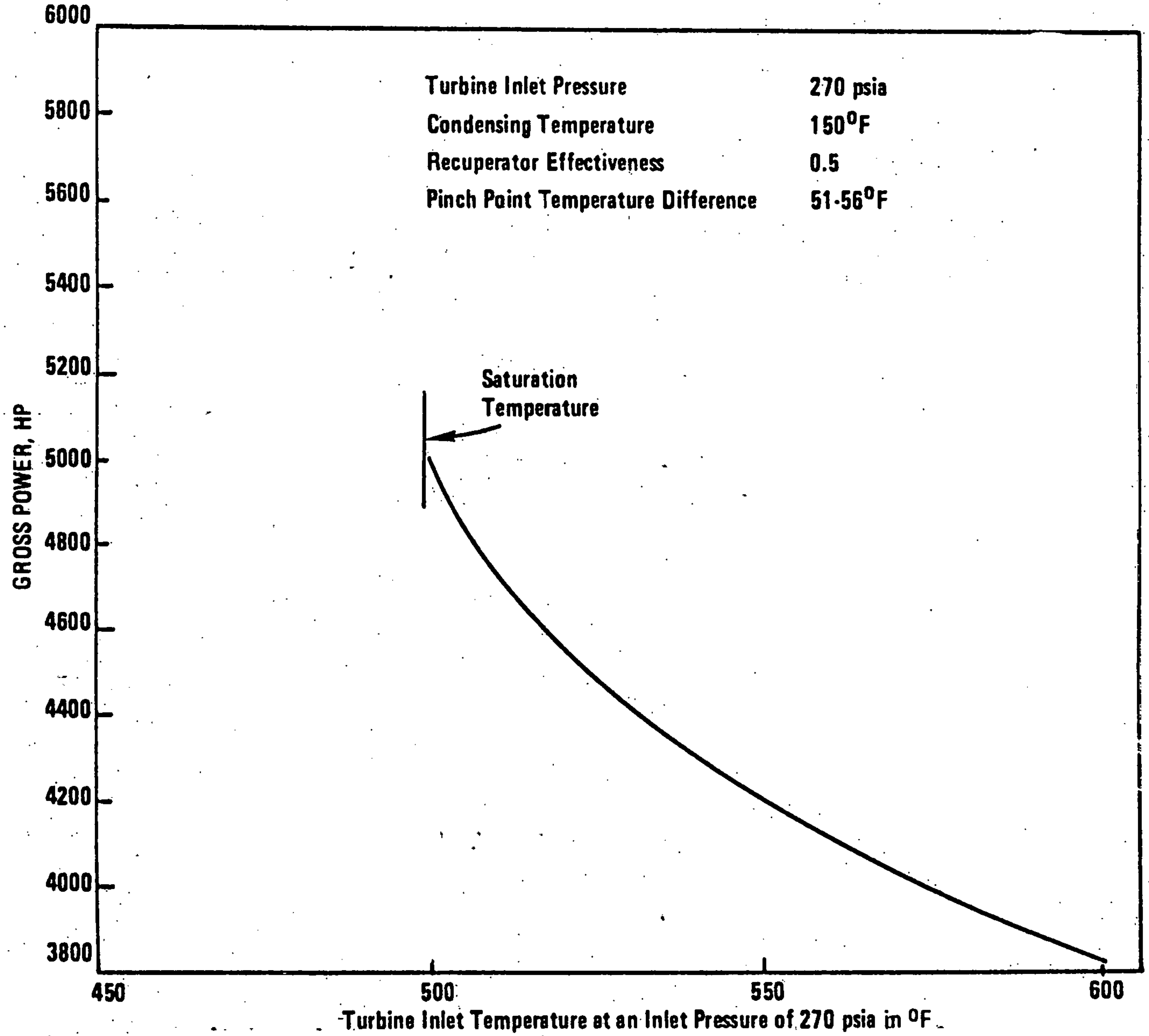


Figure 4-7. Effect of Turbine Inlet Temperature on the Horsepower.

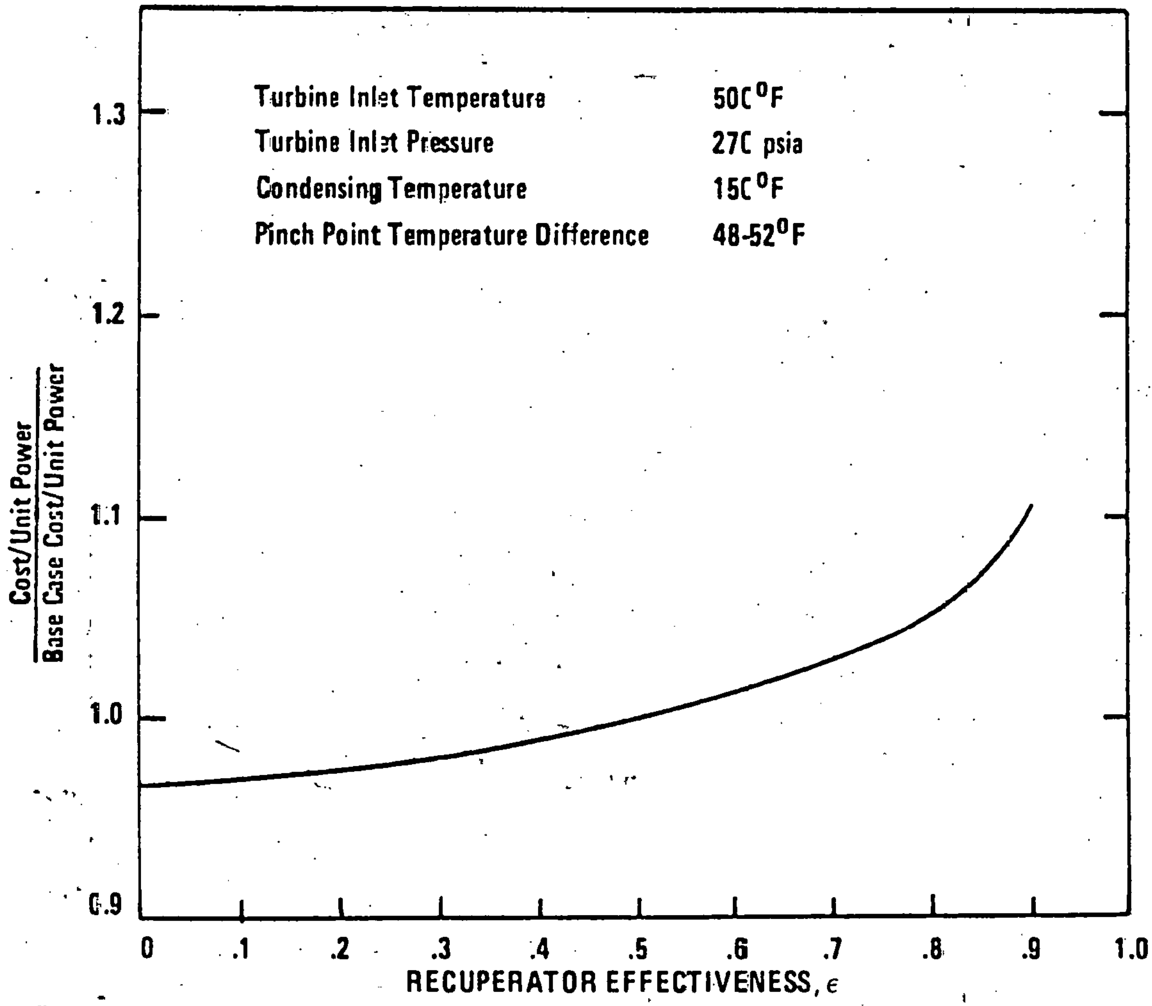


Figure 4-8. Effect of Recuperator Effectiveness on the Cost per Unit Power

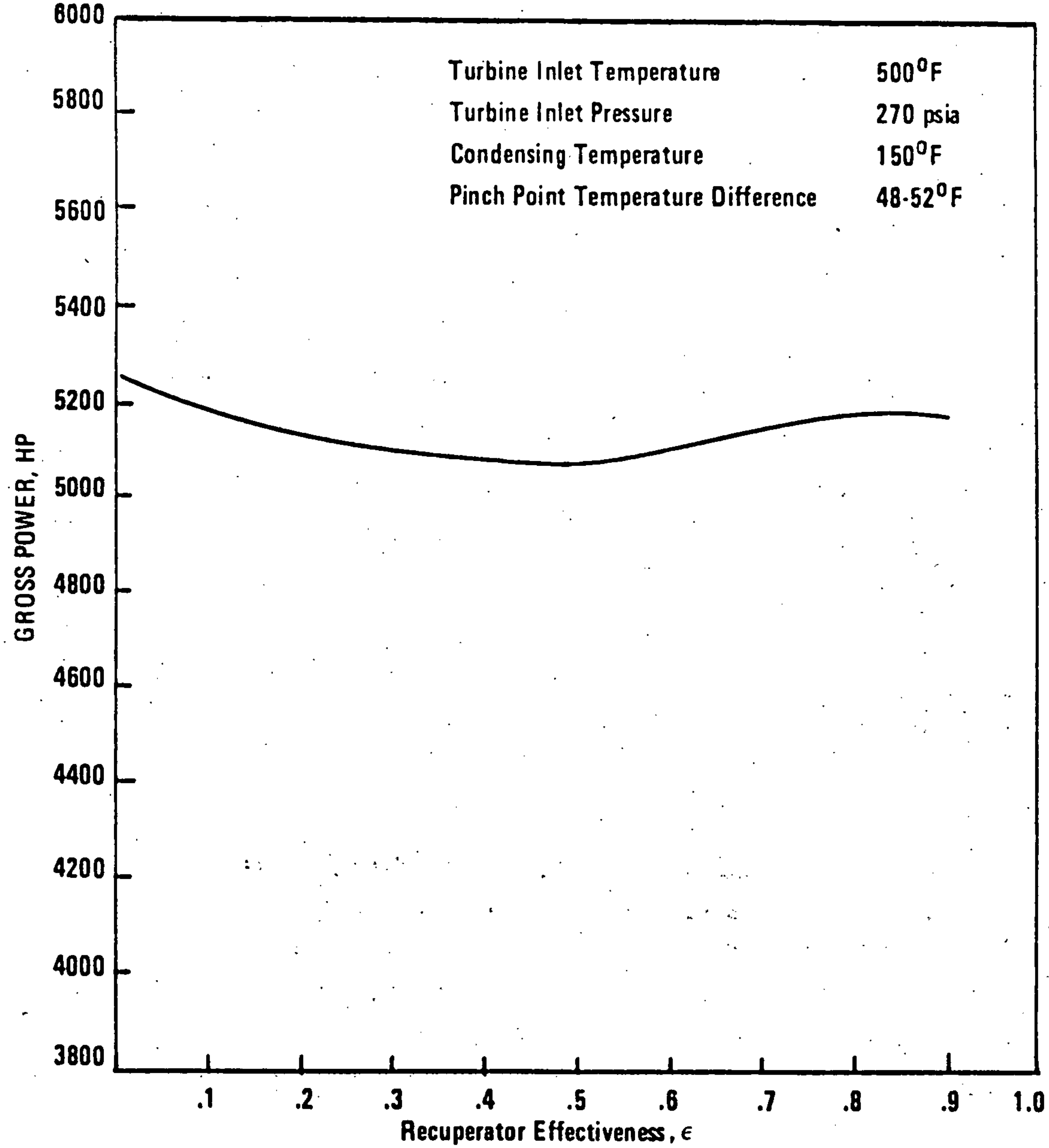


Figure 4-9. Effect of Recuperator Effectiveness on the Horsepower

from the vapor generator is necessary to prevent the condensing of sulfuric acid. Thus, a larger amount of heat can be taken out of the stack if no recuperator is used. The reason for this is that the recuperator increases the temperature of the boiler feed liquid thereby limiting the amount of heat which can be extracted from the stack gas as discussed above.

Shown in Figure 4-10 is the variation in cost per unit power of the bottoming cycle against saturation conditions typified by various temperatures for two values of recuperator effectiveness. First of all, the cost per unit power curves are very flat with temperature minimizing in the neighborhood of 500°F. Also it can be seen that the cost per unit horsepower is significantly lower at a recuperator effectiveness of zero compared to a recuperator effectiveness of 0.5. Shown in the next figure, Figure 4-11, is the variation of bottoming cycle horsepower with turbine inlet temperature. Here the bottoming cycle without a recuperator produces a higher bottoming cycle power for most of the values of turbine inlet temperature shown. Shown in Figure 4-12 is the variation in the cost per unit as a function of the condensing temperature. Values are shown for two conditions of recuperator effectiveness. For the 0.5 recuperator effectiveness the minimum point on the curve is around 125°F. However, for zero effectiveness the value is very close to 150°F. It can be seen that the two optimum values are very close together. Shown in Figure 4-13 is the variation of horsepower with the same set of conditions indicating that as condensing temperature decreases the horsepower increases. However, it is apparent from the previous curve as the condensing temperature becomes lower with the same outside air temperature obviously more condenser surface is required to obtain the condenser temperature desired. Three points, case 20, 24 and 9 are shown on this curve. On the previous curve point 24 had the lowest cost per horsepower, point 20 was the next lowest. Shown in Figure 4-14 is the variation in cost per unit horsepower against pinch point temperature difference. The pinch point temperature difference is the minimum difference in temperature between the exhaust gas flowing through the vapor generator and the vaporizing fluid which is in the tubes of the vapor generator. It can be seen on the figure that for the effectiveness of 0.5 and zero, the optimum pinch point temperature difference is approximately 50°F. The recuperator ef-

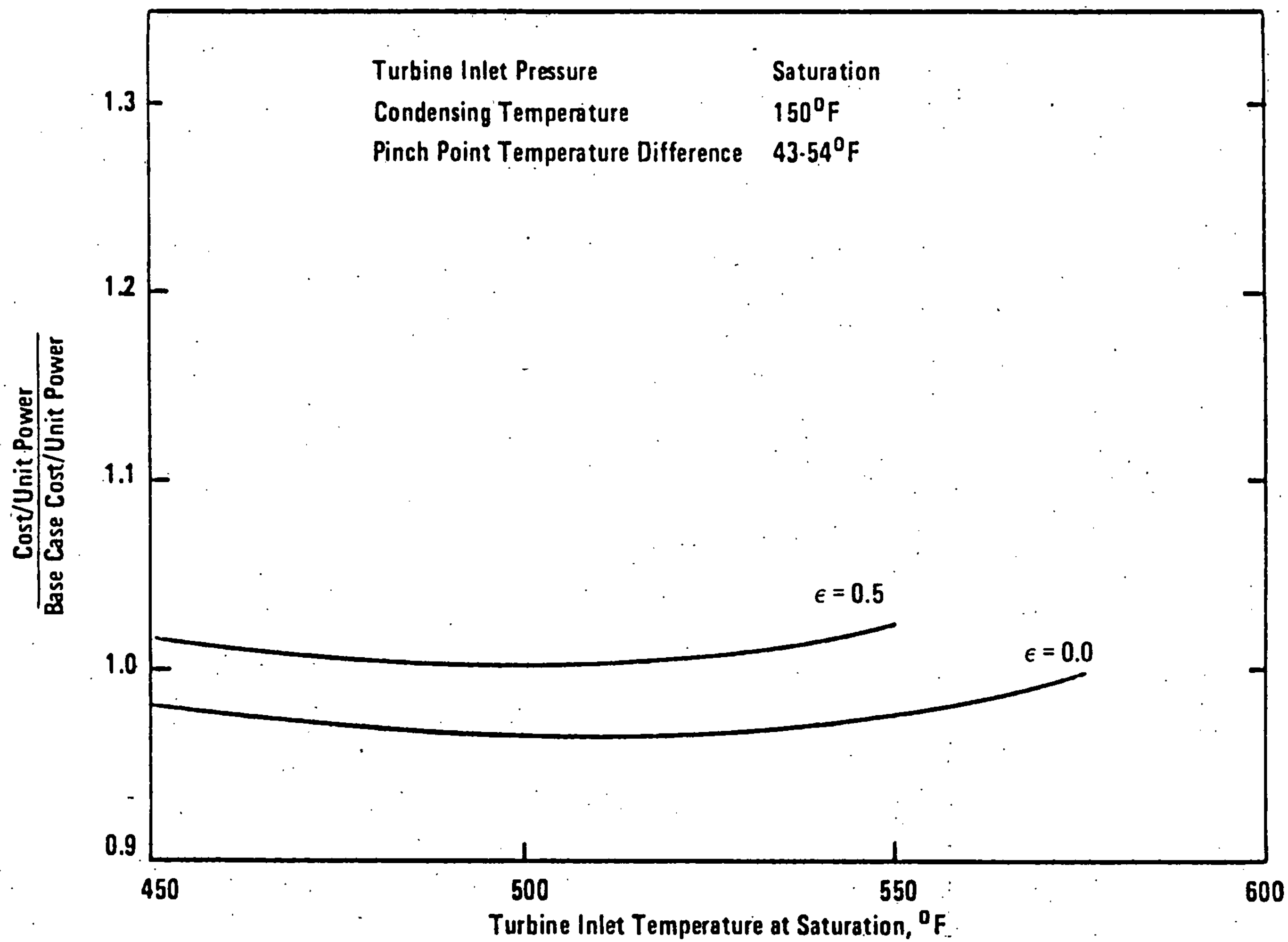


Figure 4-10. Effect of Saturation Condition and Recuperator Effectiveness (ε) on the Cost Per Unit Power.

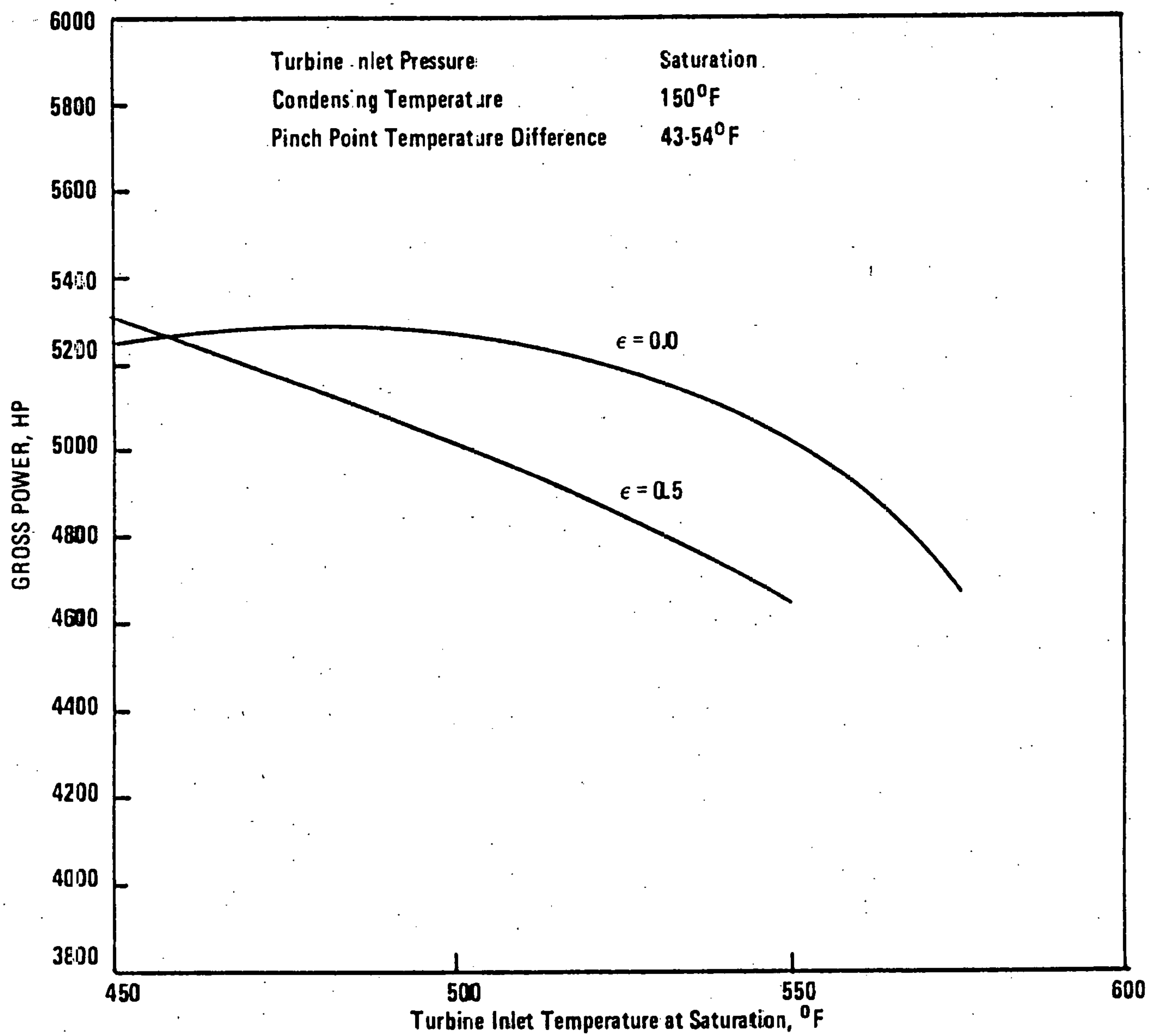


Figure 4-11. Effect of Turbine Inlet Temperature and Recuperator Effectiveness (ε) on the Gross Power.

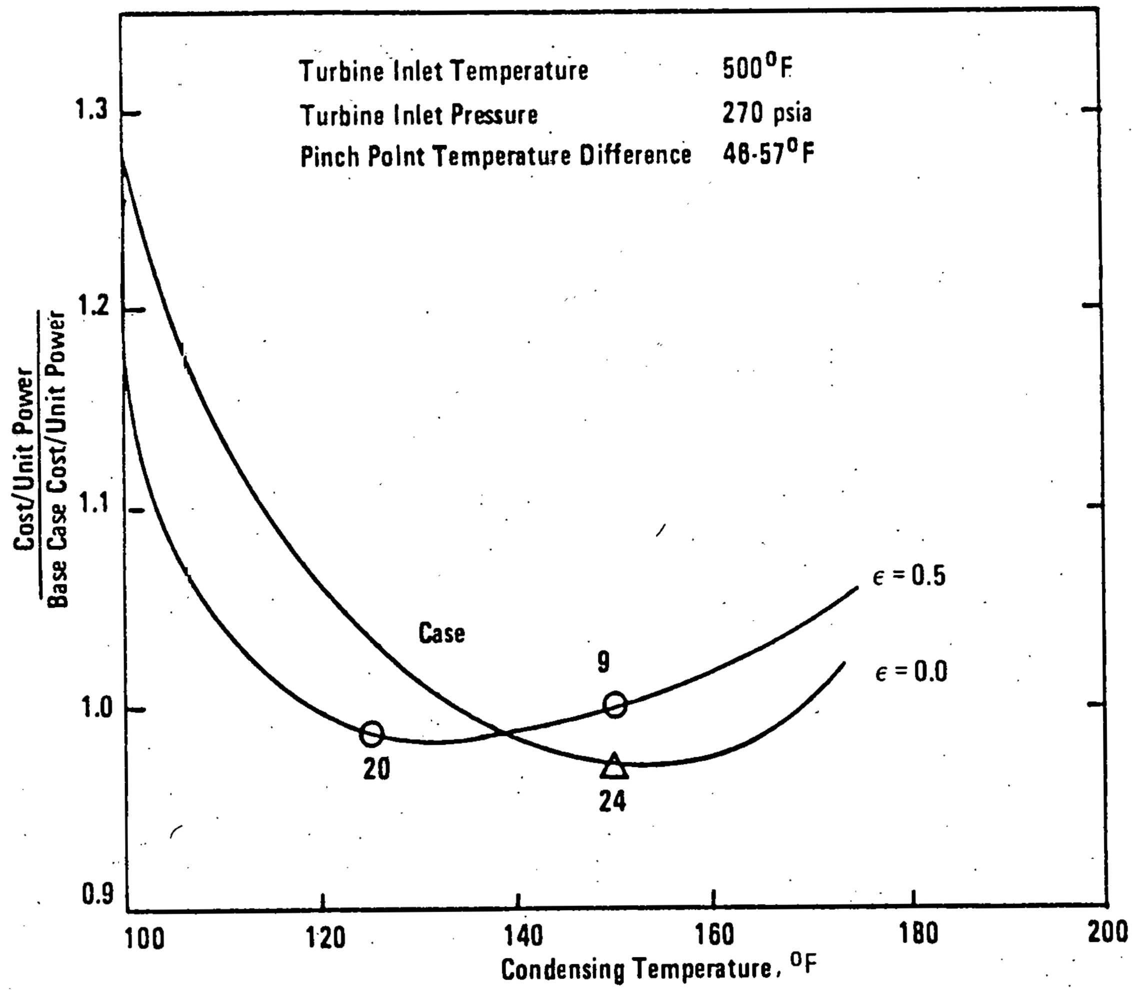


Figure 4-12. Effect of Condensing Temperature and Recuperator Effectiveness (ε) on the Cost Per Unit Power.

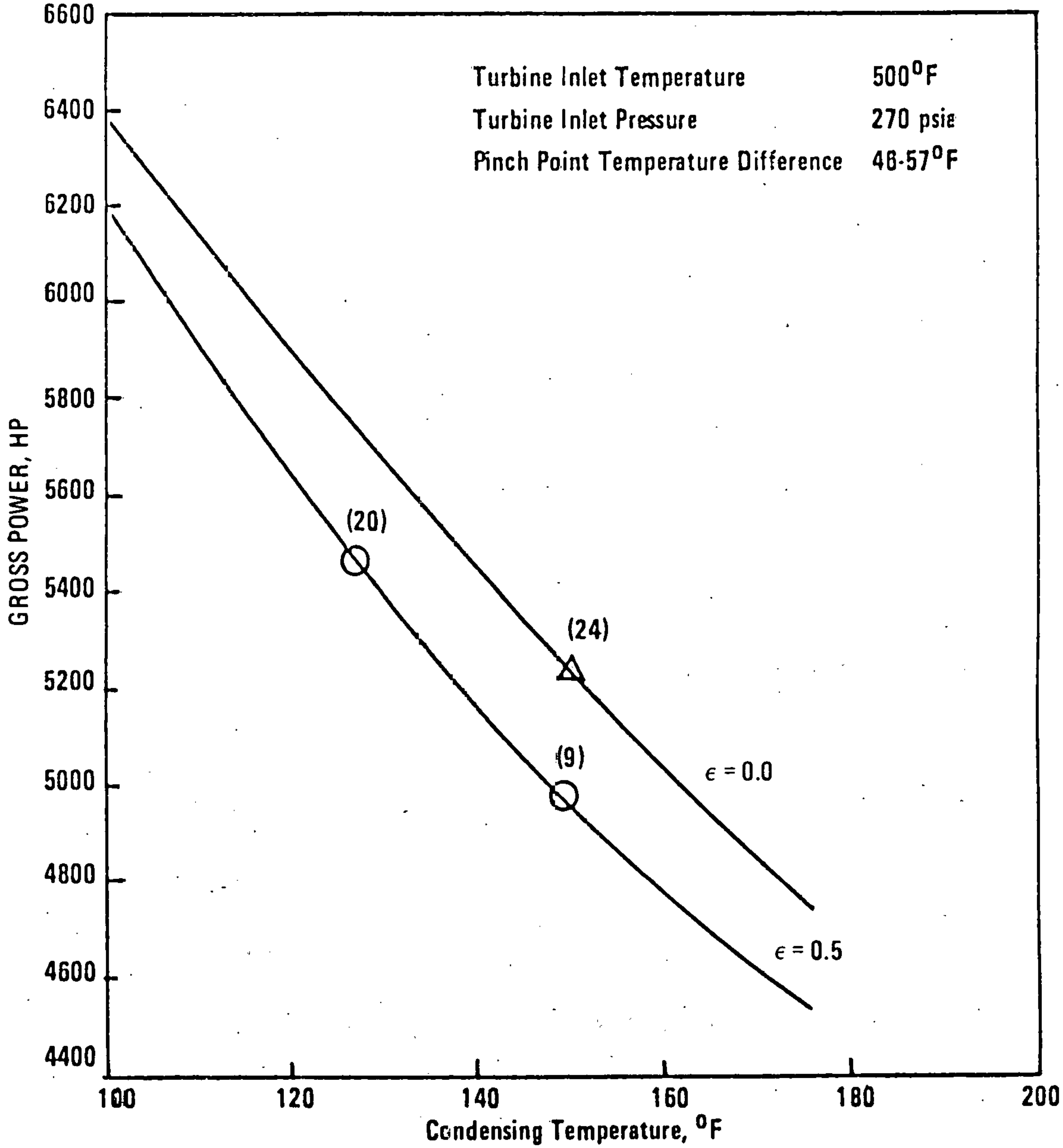


Figure 4-13. Effect of Condensing Temperature and Recuperator Effectiveness (ε) on the Gross Power.

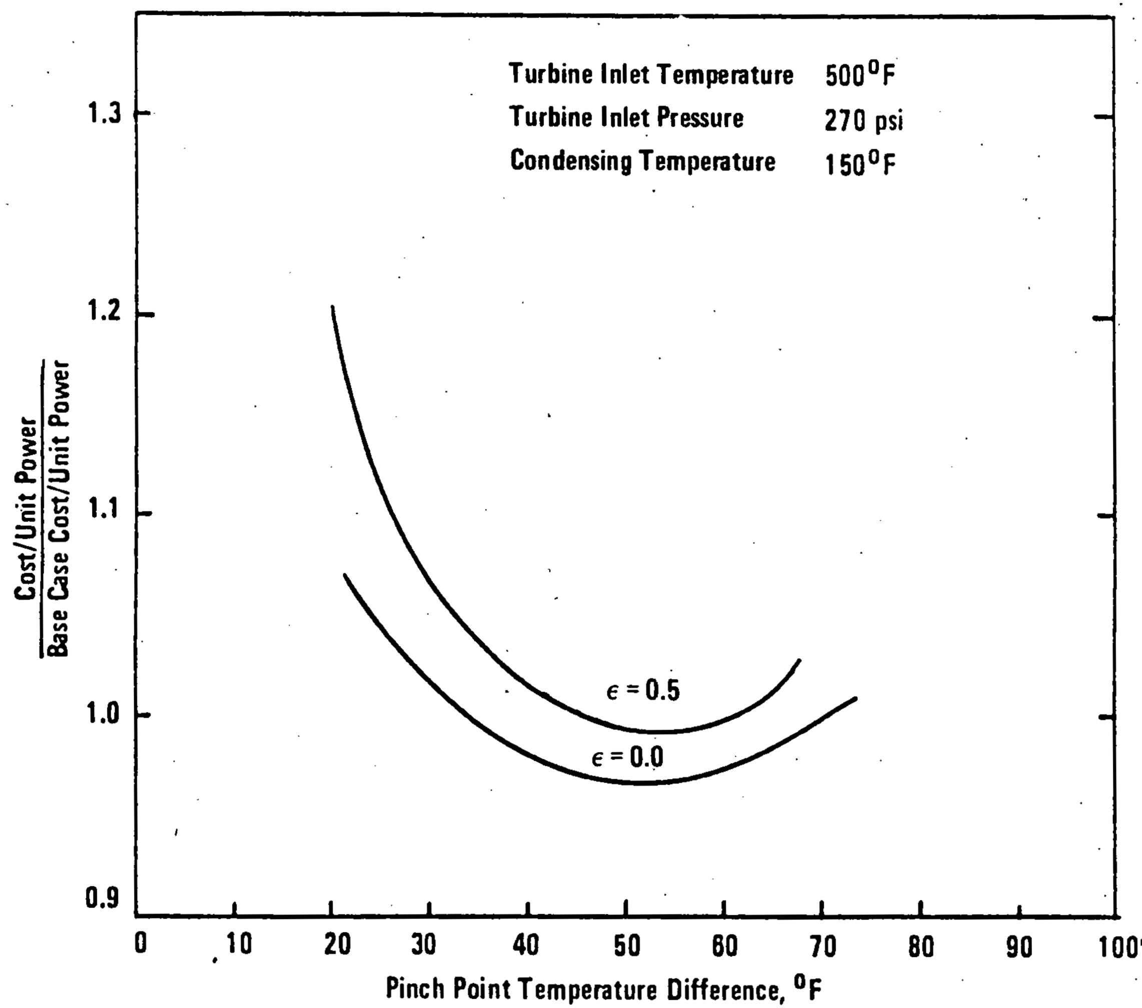


Figure 4-14. Cost per Unit Power as a Function of the Pinch Point Temperature Difference and Recuperator Effectiveness, ε

fectiveness of zero produces the lowest cost per horsepower. Figure 4-15 shows the variation of bottoming cycle horsepower with pinch point temperature difference, the curves for zero and 0.5 effectiveness being approximately linear and increasing as would be expected as the pinch point temperature approaches zero.

Shown in Table 4-8 are the data for the three points designated in some of the previous figures, Cases 9, 20 and 24. Case 9 is the reference case which was used to make the first sizing of components and to determine an estimate of the cost of the bottoming cycle. Case 20 is one in which a recuperator effectiveness of 0.5 was utilized and Case 24 is a case where zero was utilized. It can be seen by the last row of figures that the lowest cost per unit horsepower relative to the reference case occurs for Case 24 which is one that has zero effectiveness of the recuperator. No recuperator was included in the final optimization because of the results shown in Table 4-8. The selected configuration not only minimizes the $/HP but is simpler and has fewer components. Shown in Table 4-9 are the data obtained for the optimized bottoming cycle, that is for the bottoming cycle associated with Case 24. The duties of the vapor generator and the condenser are shown as well as the outputs and the specific parasitic power requirements. The net power to load is 5142 HP which is 5% larger than the HP assumed in the reference cycle. Shown in Figure 4-16 is the schematic diagram of the bottoming cycle with the pertinent data for the optimized cycle appearing on the figure.

4.4 SITE PERFORMANCE RESULTS

After having selected the optimum bottoming cycle as was just described, the design variables which came out of that optimization were applied to bottoming cycles for all three sites that are being studied. Shown in Table 4-10 are the site conditions for each of the three sites. Shown in the table are the gas turbines utilized, the flow rate, exhaust temperature, output and heat rate of the gas turbines shown. These data were determined for the average annual temperature at the weather stations nearest to the sites. It should be noted that because of the altitude, the Burney, CA site has the lowest mass flow rate and this has an effect on the amount of energy that can be conserved with a bottoming cycle.

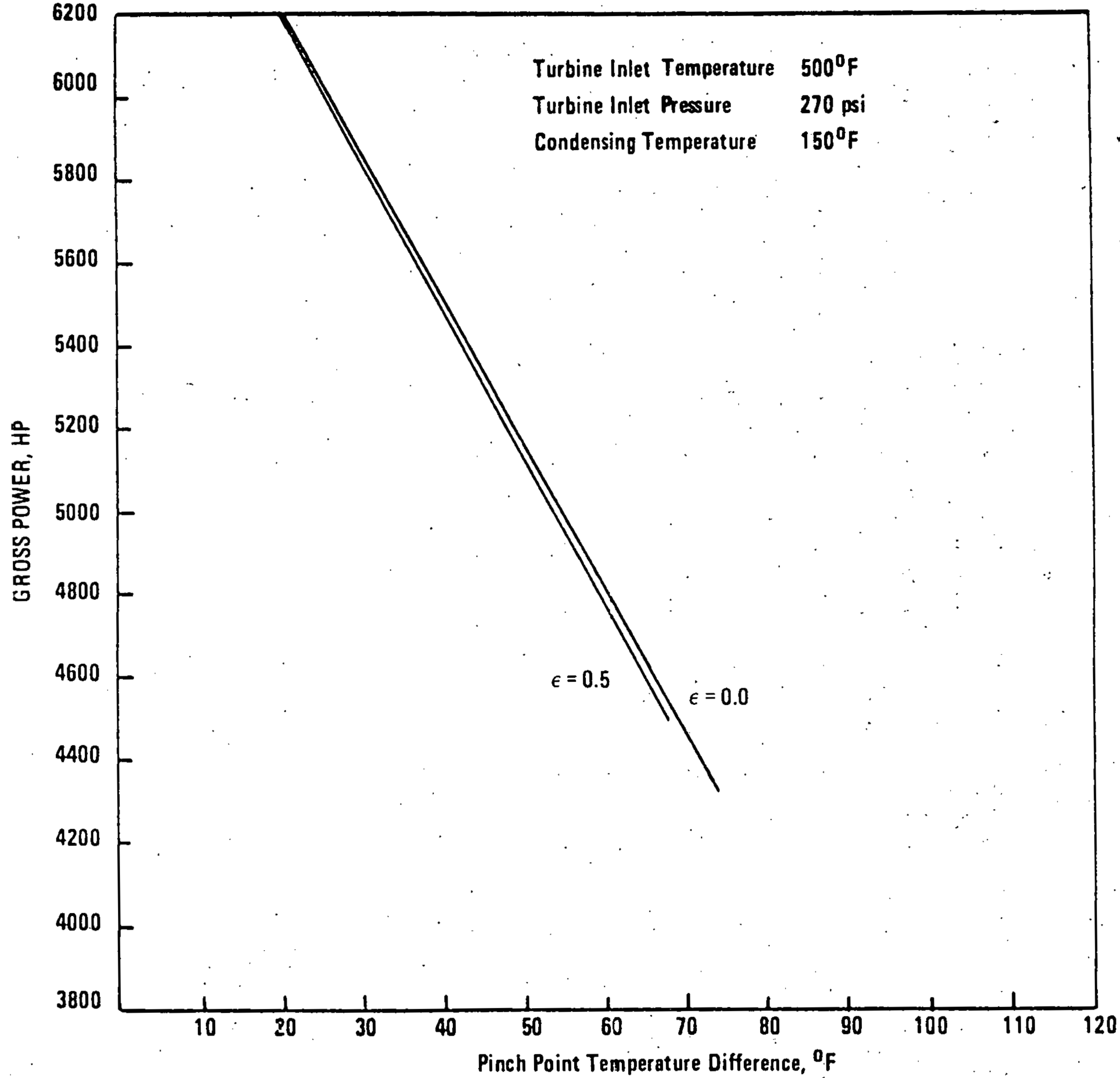


Figure 4-15. Effect of the Pinch Point Temperature Difference on the Horsepower

TABLE 4-8

COMPARISON OF THE REFERENCE DESIGN CASE WITH THE TWO NEAR OPTIMUM CASES FOR RECUPERATOR EFFECTIVENESS (ε)

	Case 9 (Reference Case)	Case 20	Case 24
Turbine Inlet Temperature, °F	500	500	500
Turbine Inlet Pressure, psia	270	270	270
Recuperator Effectiveness	0.5	0.5	0.0
Condenser Temperature, °F	150	125	150
Pinch Point Temperature Difference, °F	54	57	48
Vapor Generator Stack Temperature, °F	333	327	274
Turbine Exit Temperature, °F	311	290	310
Condenser Inlet Temperature, °F	229	207	310
Gross Power, Horsepower	4991	5440	5247
Cost Per Unit Power Relative to the Reference Case 9	1.0	.980	.966

TABLE 4-9

TOLUENE BOTTOMING CYCLE CHOSEN AFTER COST OPTIMIZATION

Turbine Inlet Pressure	270 psia
Saturation Temperature	495°F
Turbine Inlet Temperature	500°F
Condensing Temperature	150°F
Toluene Flow Rate	209,000 lb/hr
Heat Added to Working Fluid	60.6×10^6 BTU/hr
Heat Rejected by Working Fluid	47.55×10^6 BTU/hr

Parasitic Power Requirements
(Motor Efficiency, 0.9)

Pump	100 KWe
Fan	310 KWe
Total	410 KWe

Turbine Power	5247 HP
Net Power to Load (Gearbox Efficiency, 0.98)	5142 HP

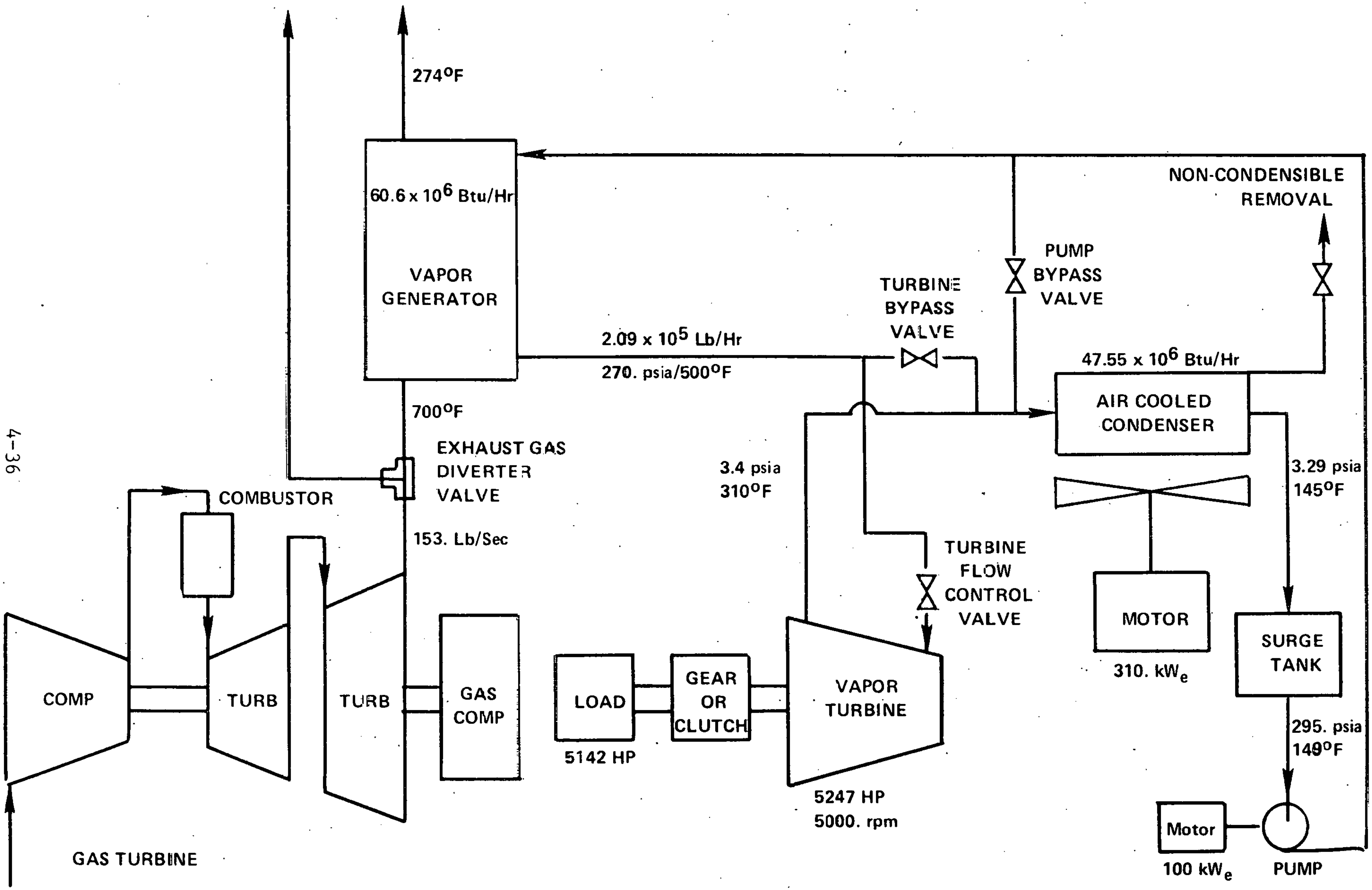


Figure 4-16. Optimized Bottoming Cycle Schematic.

TABLE 4-10

DATA ON THE PIPELINE SITE CONDITIONS

Pipeline Site	Gas Turbine	Avg. Temp./ Altitude	Flow Rate lbs/sec	Exhaust Temp. °F	Output HP	Heat Rate BTU/HP hr
Rayne, LA Columbia Gulf	RR76G/CB125 (Rolls Royce/ Cooper Bessemer)	68°F/Sea Level	164	727	13622	9935
Covington, TN Texas Gas	P&W GG3C-4 (Pratt & Whitney)	62°F/Sea Level	165	700	13835	10567
Burney, CA PG&E	LM1500/IR125 (General Electric)	56°F/3200 Feet	147	713	13170	9394

By the time the optimization was complete and performance for the various sites was to be calculated, the pipeline companies were able to provide enough information about the site installations that pressure drop characteristics could be determined for each site. These values are shown in Table 4-11 and indicate the variation in pressure drops in the systems for the three sites being studied. These pressure drop values were incorporated into the calculation of site performance. Shown in Table 4-12 are the performance values for each of the three sites in terms of bottoming cycle power to the load which includes the gearbox except for the case for Burney where there was no gearbox. The gas turbine power in Table 4-12 includes the effect of the gas turbine exhaust pressure drop shown in Table 4-11. Shown also are the combined power of the system with the gas turbine, the combined heat rate of the system, and the required electrical power needed to drive the fans and the pump. It can be seen that the bottoming cycle power varied between a little over 6000 HP down to almost 5400 HP at the higher altitude site of Burney.

4.5 SYSTEMS DESIGN

The bottoming cycle system was designed to have three major subsystems. The first is the vapor generator, the second is the air cooled condenser, and the third is the equipment skid which has on it the turbine, the feed pump, the surge tank, and the auxiliary equipment and, in the case of the reference cycle, regenerators. In the design of bottoming cycle apparatus the minimum health and safety protection requirements for the system based upon toluene as the working fluid must be observed in order to provide the proper fire protection. The first of these requirements is that the vapor generator be located 50 feet from all other structures. In addition emergency relief venting for the vapor generator and condenser should be provided. A water spray system should be provided for the vapor generator and supports from the ground up to 10 feet above the toluene line. This system should both be automatic and manual in operation. A CO_2 extinguishing system to protect the core of the vapor generator should be provided. All electrical wiring installed on the apparatus should be of the type for use in a Class I, Division 2 location, and this should apply inside all of the bottoming cycle apparatus, within 5 feet of the vapor generator, and within 25 feet from all bottoming cycle apparatus from grade level up to 3 feet in height. In addition an interlock to shut off the heat source

TABLE 4-11

CYCLE PRESSURE DROPS FOR THE SPECIFIC SITES

	Rayne, LA	Covington, TN	Burney, CA
Gas Turbine Exhaust, inches of water	5.3	7.0	5.3
Vapor Generator to Turbine, psi	2.53	9.38	5.1
Turbine to Condenser, psi	0.062	0.055	0.039
Condenser, psi	0.11	0.11	0.11
Condenser to Pump, psi	0.415	0.199	0.501
Pump to Vapor Generator, psi	0.772	2.94	1.1
Vapor Generator, psi	50.6	50.6	50.6

TABLE 4-12

CYCLE PERFORMANCE DATA FOR THE SPECIFIC SITES

Site	Gas Turbine[a] Power HP	B/C Power HP (gross)	B/C Power (to load) $\eta_{gear} = .98$	Combined Power HP	Combined Heat Rate BTU/HP hr	Required Electrical Power, kWe	Heat Rate[b] Improvement %
Rayne, LA	13434	6098	5976	19410	6973	538	27.8
Covington, TN	13587	5614	5502	19089	7659	500	25.9
Burney, CA	12972	5397	5397	18369	6735	476	26.5

[a]Values reduced for vapor generator back pressure

[b]Includes an allowance for electrical power

should be provided in the event that the power to the condenser fans is interrupted. As regards piping, valves, fittings, and relief valves these should be in accordance with the American Society of Mechanical Engineers Code for pressure vessels and the American National Standards Institute power piping code B-31.1. An emergency trench-type drain to a safe location should be provided for flammable leakage and for the fire protection water. A controlled area should be established within a 50 foot radius of bottoming cycle apparatus with limited access, "Restricted Area" signs and "No Smoking" signs. In addition provisions should be made for stack sensors which would automatically activate the water spray systems, the CO_2 system, shut off the heat source and dump the toluene to a safe location following procedures recommended by the National Fire Protection Agency (NFPA). Shown in Table 4-13 are the safety and environmental codes which are applicable to the bottoming cycle apparatus.

4.5.1 TURBINES

Initially radial inflow turbines were considered for the toluene bottoming cycle; however, because of the low acoustic velocity in organic working fluids, a single stage radial inflow turbine for the engine work the turbine must perform will result in supersonic velocities approaching the rotating blade. This can be avoided by a multistage axial flow design. Therefore a seven stage axial flow machine was designed for this application. Shown in Table 4-14 are the design data for the seven stage axial flow turbine. Shown are the pitch line wheel speeds, the enthalpy drops in each stage, the loading parameters at the hub, the length of the buckets, the tip diameter of the blade, the root centrifugal stresses, the metal temperatures, and the ratio of the annulus area of any stage divided by the annulus area of the previous stage. The turbine is designed with many stages mainly because of the rapid variation of the annulus area required to pass the toluene flow especially at the low pressures. Usually one would keep the ratio of the annulus areas within 1-1/2 for good design. However, in this instance it was necessary to exceed that value in the rear stages while it was not possible to obtain that value in the front stages. Blade heights for the turbine vary from 0.69 inch in the first stage to over 6 inches in the last stage. The root centrifugal stresses are relatively low and the temperatures of the blades are lower than the high pressure stages of a stream turbine, for example. This

TABLE 4-13

SAFETY AND ENVIRONMENTAL CODES
APPLICABLE TO THE PIPELINE BOTTOMING CYCLE

Vapor Generator	ASME[a] Boiler Code
Pressure Vessels	ASME Unfired Pressure Vessel Code Sec. VIII, Div. 1
Piping	ANSI[b] B31.1 Power Piping Code
Heat Exchangers	TEMA[c]
Electrical	National Electrical Code Class I, Div. 2, GpD
Handling Working Fluid	NFPA[d] 30, Flammable and Combustible Liquid Code

(a) American Society of Mechanical Engineers

(b) American National Standards Institute

(c) Tube Exchanger Manufacturers Association

(d) National Fire Protection Agency

TABLE 4-14

TURBINE DESIGN DATA

Stg.	U_p, ft/sec	Δh, Btu/lb	$gJ\Delta h/U_p^2$	L, in.	D_T, in.	σ_{rc}, psi	T_m, °F	An/An-1
1	350	6.1	1.25	.69	16.73	870	490	
2	360	7.1	1.37	.85	17.35	1105	475	1.274
3	375	8.1	1.44	1.15	18.34	1555	455	1.405
4	395	9.1	1.46	1.70	19.8	2410	435	1.552
5	420	10.2	1.45	2.47	21.72	3740	413	1.552
6	450	11.2	1.38	4.04	24.66	6540	386	1.75
7	500	12.2	1.22	6.45	29.37	11620	361	1.78

Definitions

U_p Pitch line wheel speed, ft/sec

Δh Enthalpy drop, Btu/lb

$\frac{gJ\Delta h}{U_p^2}$ Kinetic energy ratio

L Blade height, in.

D_T Stage tip diameter, in.

σ_{rc} Root centrifugal stress, psi

T_m Metal temperature, °F

$\frac{An}{An-1}$ Annulus area ratio

means that the conventional steam turbine materials can be utilized. Shown in Figure 4-17 is a cross section of the turbine for the bottoming cycle. The vapor enters through a scroll from the left and discharges through a double exit on the right. The turbine is shown supported on pivoted pad bearings and uses toluene working fluid as the lubricant since toluene is a good solvent for lubricating oil. Also shown are floating ring seals isolating the flow path from bearing areas in the turbine. These seals contain two close running clearance rings between which toluene liquid is injected in order to effect the seals. Additional features of the turbine include a solid rotor which has blades attached to the wheels by dovetails. Bucket covers are used in order to improve the efficiency of all stages and to prevent destructive vibration of the blading in the latter stages. The thrust bearing is located at the high pressure end so the small blade/height stages will be closely controlled within their running positions. Labyrinth seals that isolate the various stages are designed to have a low radius from the center of the machine thus reducing the flow area in a clearance space and, therefore the flow that leaks through. In order to increase the efficiency of the turbine, an exit diffuser has been utilized which recovers some of the static pressure drop that was necessary to get the flow through the latter stages of the machine. The wheels and blades of the turbine are specified to be made of B5F5 wrought steel alloy and the casing and stators of 2-1/4 Cr-Mo alloy steel.

4.5.2 VAPOR GENERATORS

Several bottoming cycle machines that are being developed by the Department of Energy utilize various types of vapor generators. The three most common types are the natural circulation boiler, the forced circulation boiler and the once-through boiler. For the toluene bottoming cycle it was decided to utilize the benefits of a once-through design. The once-through design eliminates the drum and separators and the natural circulation circuits. It can be used below and above the critical pressure of the fluid that is being used. It costs less because no circulating pumps or vapor drums are required, and it tends to reduce the number of places that leakage of the working fluid could occur. Shown in Figure 4-18 is a temperature heat flow plot for the vapor generator. The upper line shows the variation of gas turbine exhaust gas tempera-

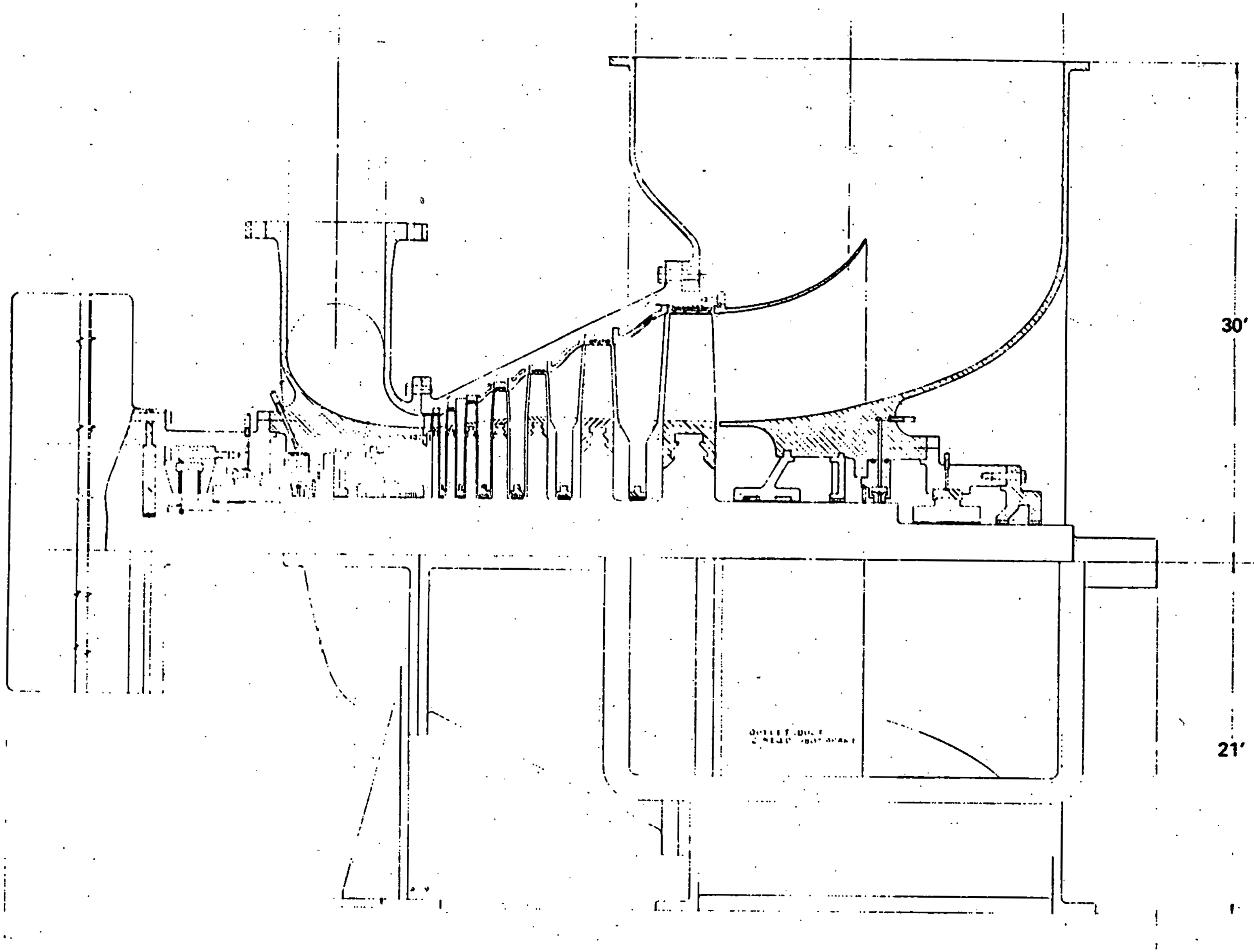


Figure 4-17. **Toluene Vapor Turbine for Pipeline Bottoming Cycle (GE-AEP)**

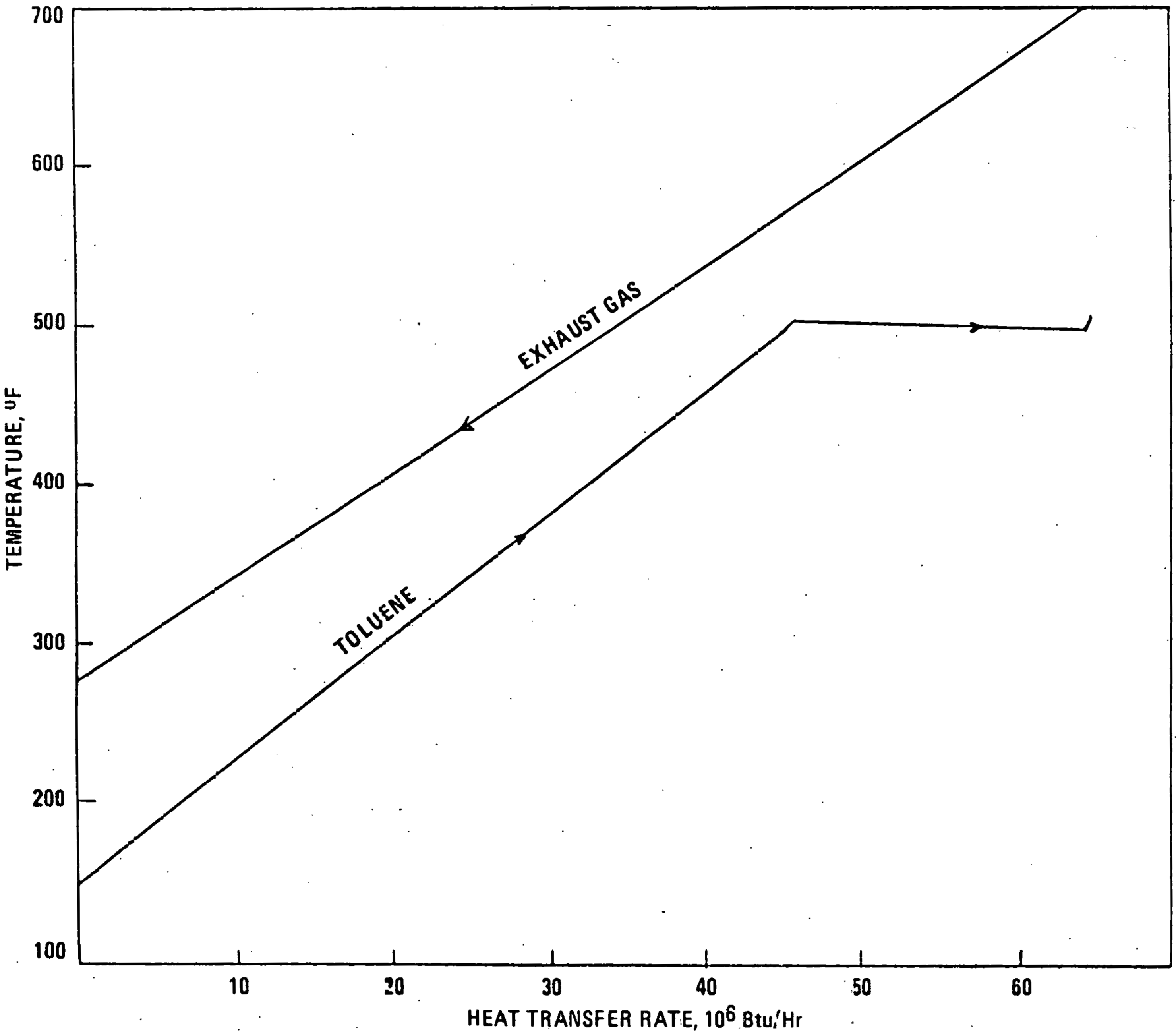


Figure 4-18. Vapor-Generator Temperature Distribution

ture as it leaves the turbine and proceeds through the vapor generator. The lower toluene line starts from the condensing condition of 145°F, goes through the preheating section and then changes over to the evaporative section and finally a small amount of superheat is shown at the extreme right. It is necessary that a minimum temperature difference be maintained between these two lines in order that the log mean temperature difference between the combustion gas and the toluene be large enough. Shown in Table 4-15 are the parameters which define the heat exchanger geometry. This vapor generator is a multipass cross-counterflow design with tubes that have disc fins and a shell and tube heat exchanger. The dimensions for the tubing are shown in the table. Shown in Figure 4-19 is the vapor generator tube geometry. As can be seen there are two headers, one at the top and one at the bottom. Toluene comes in at the top and leaves as vapor at the bottom. The exhaust gas flows vertically upward through the tube bank. With this type of design, in which both headers are on the same side, differential thermal expansion between the tubes and their attachments to the tube bundle frame can be minimized. Shown in the upper right hand corner of the diagram is the geometry of the tubing and the spacing pattern that was chosen for this design. Shown in Table 4-16 is a comparison of the designs for vapor generators that will be required at the three different sites which are in consideration on this program. Near the top of the table is shown the duty in BTU per hour required of each of the vapor generators for the three sites. Also shown are the values of the exhaust gas inlet temperature to the vapor generator because of the existence of the three different gas turbines at the three sites and different ambient temperatures which were assumed for the three sites. The gas flow rates also vary for the gas turbines because one of the units, namely the one at Burney, CA, is designed for 3200 feet whereas the other two are at sea level. The toluene inlet and exit temperatures are approximately equal for each of these three designs. The heat transfer area varies from 57,800 to 61,900 sq. ft. The pressure drops on the shell side are within the range from 3.6 to 4.14 inches of water. A screen is provided upstream of the tube bundle in order to help distribute the combustion gas over the entire face of the vapor generator. Because of the use of a once-through design substantial orifice pressure drops in the liquid end of the tubing are provided to maintain parallel channel stability in the operation of the vapor generator. A sketch is

TABLE 4-15

HEAT EXCHANGER GEOMETRY

TYPE: Multipass Cross-counterflow; Tubes With Disc Fins. Shell and Tube

Tube Diameter (inch)	1.5
Tube Wall Thickness (inch)	0.083
Transverse Pitch (inch)	4.125
Longitudinal Pitch (inch)	4.125
Fin Outer Diameter (inch)	2.75
Fin Thickness (inch)	0.08
No. of Fins/Inch	6
Length of the Tube (feet)	15
Width of the HX (feet)	10

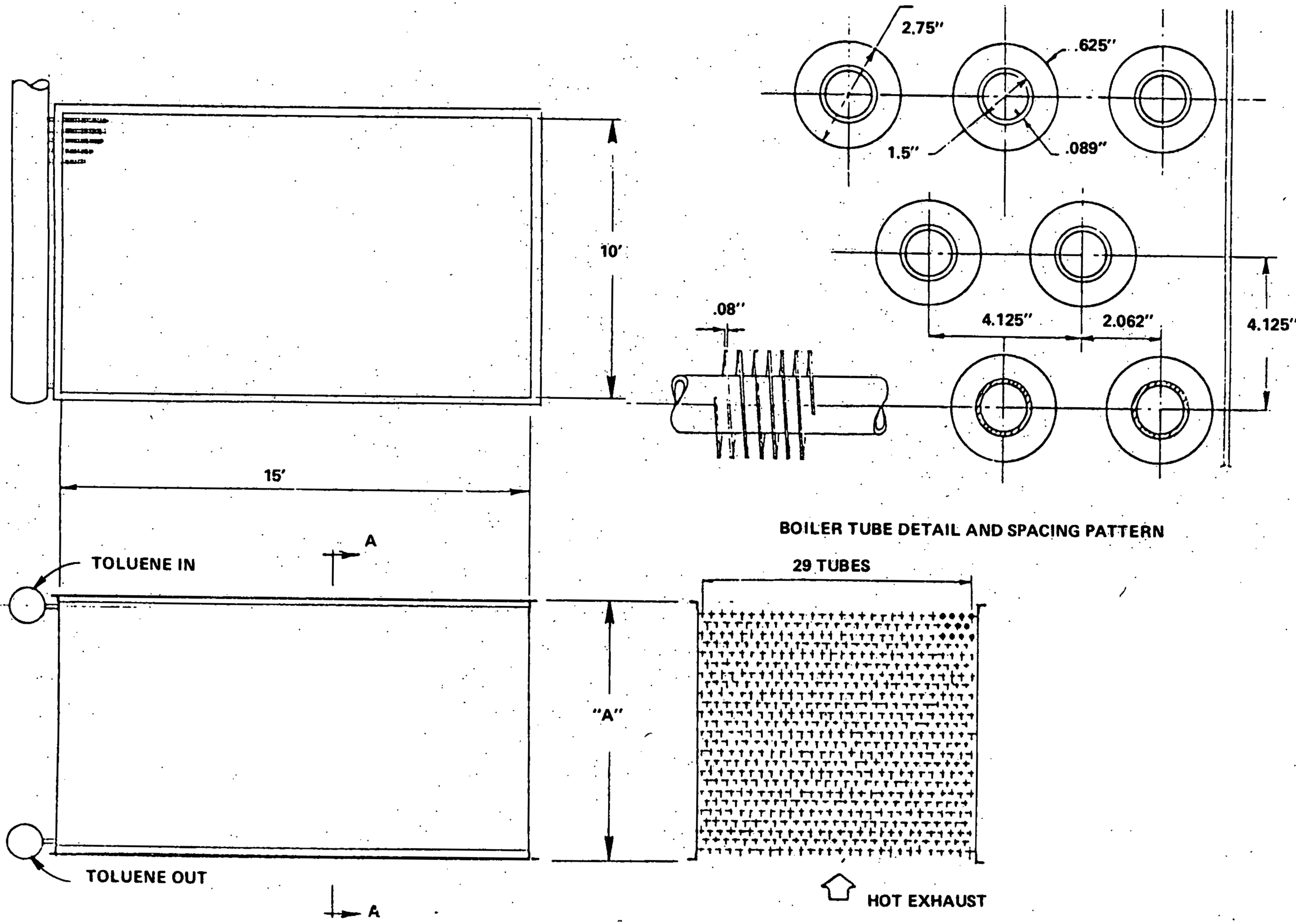


Figure 4-19. Vapor Generator Tube Geometry.

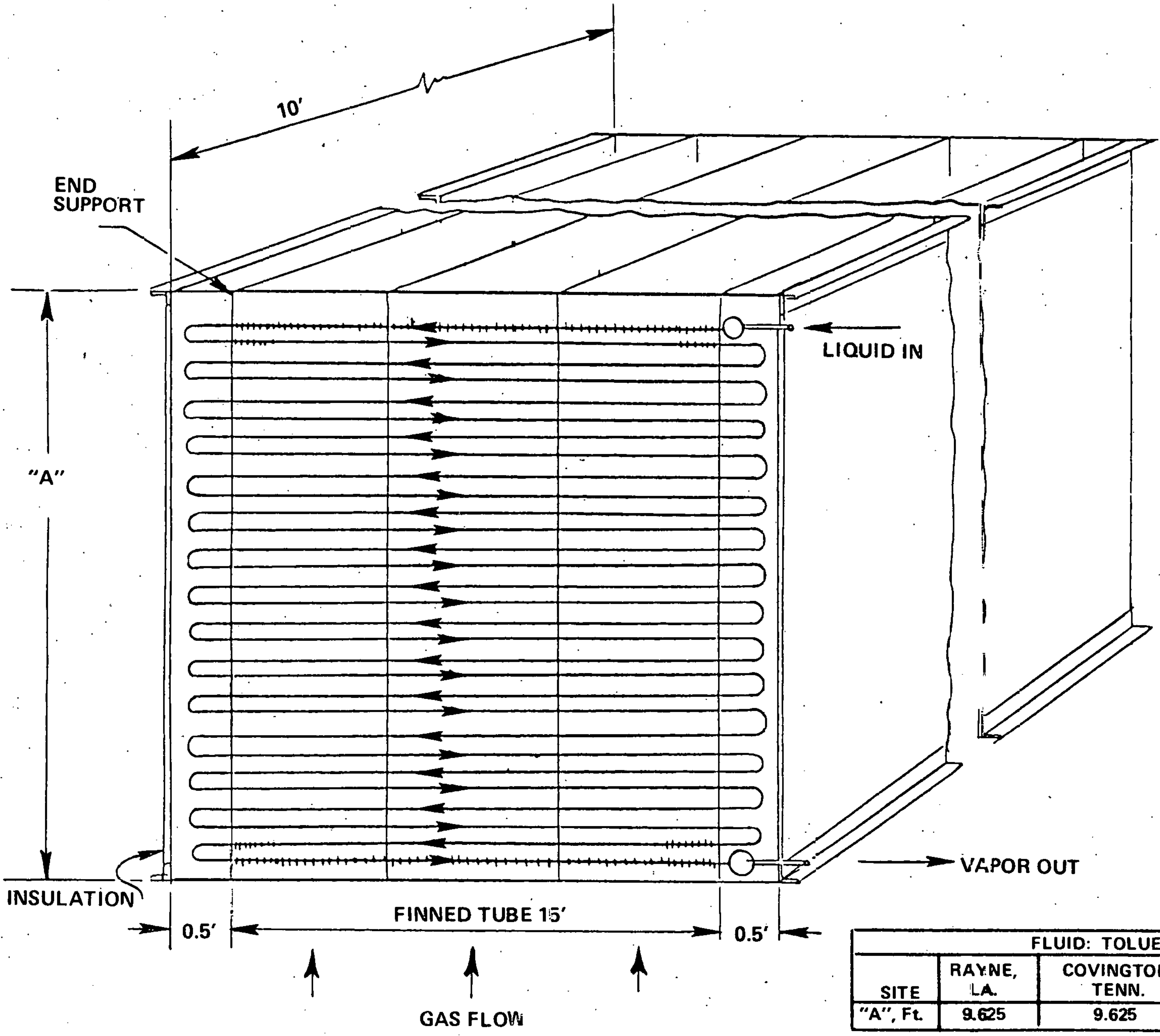
TABLE 4-16

COMPARISON OF THE TOLUENE BOTTOMING CYCLE VAPOR GENERATOR DESIGNS FOR THE THREE SITES

	Rayne, LA	Covington, TN	Burney, CA
Duty (Btu/Hr)	7.061×10^7	6.4974×10^7	6.240×10^7
Exhaust Gas			
Inlet Temperature (°F)	727	700	713
Outlet Temperature (°F)	263.4	276	256
Mass Flow Rate (lbs/sec)	164	165	147
Toluene			
Inlet Temperature (°F)	148.9	149	149
Outlet Temperature (°F)	500	501.3	500.7
Mass Flow Rate (lbs/hr)	243,610	224,190	215,290
Height of the HX (feet)	9.625	9.625	10.3
Number of Passes	28	28	30
Number of Tubes Across	29	29	29
Heat Transfer Area (ft^2)	57,774	57,774	61,900
Shell Side Pressure Drop (inch water)	4.06	4.14	3.6
Screen Pressure Drop (inch water)	0.5	0.5	0.5
Tube-Side Pressure Drop (psi)	21.0	19.0	17.5
Orifice Pressure Drop (psi)	34.2	34.2	34.2

shown in Figure 4-20 of the vapor generator and indicates the way in which it will be constructed. In addition the height of the heat exchanger which is different for the different designs is tabulated in the lower right corner. The other two dimensions, the finned tube length and the length of the heat exchanger, are the same for all three designs. Shown in Table 4-17 are variations in the vapor generator design that come about through the use of alternate fluids, Fluorinol-50, RC-1 and RC-2. These designs all apply to the Covington gas turbine conditions for comparison. It can be seen that heat transfer surface areas vary from 61,900 down to 53,790 ft^2. Shown in Figure 4-21 is the sketch of the vapor generator in which the height of the tube bundle is shown for each of the four working fluids at the Covington, TN site of the Texas Gas Transmission Corp. Shown in Figure 4-22 is an installation drawing of the vapor generator showing the exhaust gas entering the vapor generator at the bottom of the stack at the top. Outside dimensions of the vapor generator are shown in this drawing. Also shown are the liquid manifold in at the top and the vapor manifold out at the bottom of the vapor generator.

In addition to the apparatus depicted in the foregoing drawings, the following apparatus is required for the vapor generator. First of all there must be a diverter valve which will permit diverting the exhaust gas around the vapor generator when the bottoming cycle apparatus is not operational. In addition the vapor generator must have access doors for maintenance. These should be on the ends of the vapor generator so that access can be had to the tube bends. CO_2 fire extinguishing nozzles are to be located inside the vapor generator so that in the event of a toluene leak the CO_2 extinguishing system located near the vapor generator can provide CO_2 to extinguish the fire. In addition a water spray system must be provided as indicated above to spray the vapor generator and its structure from the ground to ten feet above the tube bundle. A trench and drainage system to a safe location arranged so that the toluene could not soak into the ground will be used to drain off the working fluid and the fire protection water following the procedures recommended by the National Fire Protection Association. And finally, since the toluene vapor pressure is higher than the ambient air pressure, there will be safety vents and interlocks on the vapor generator control in the event of an overpressure situation. An interlock will also be used to actuate the diverter valve in the event the cooling air failed to flow through the condenser.



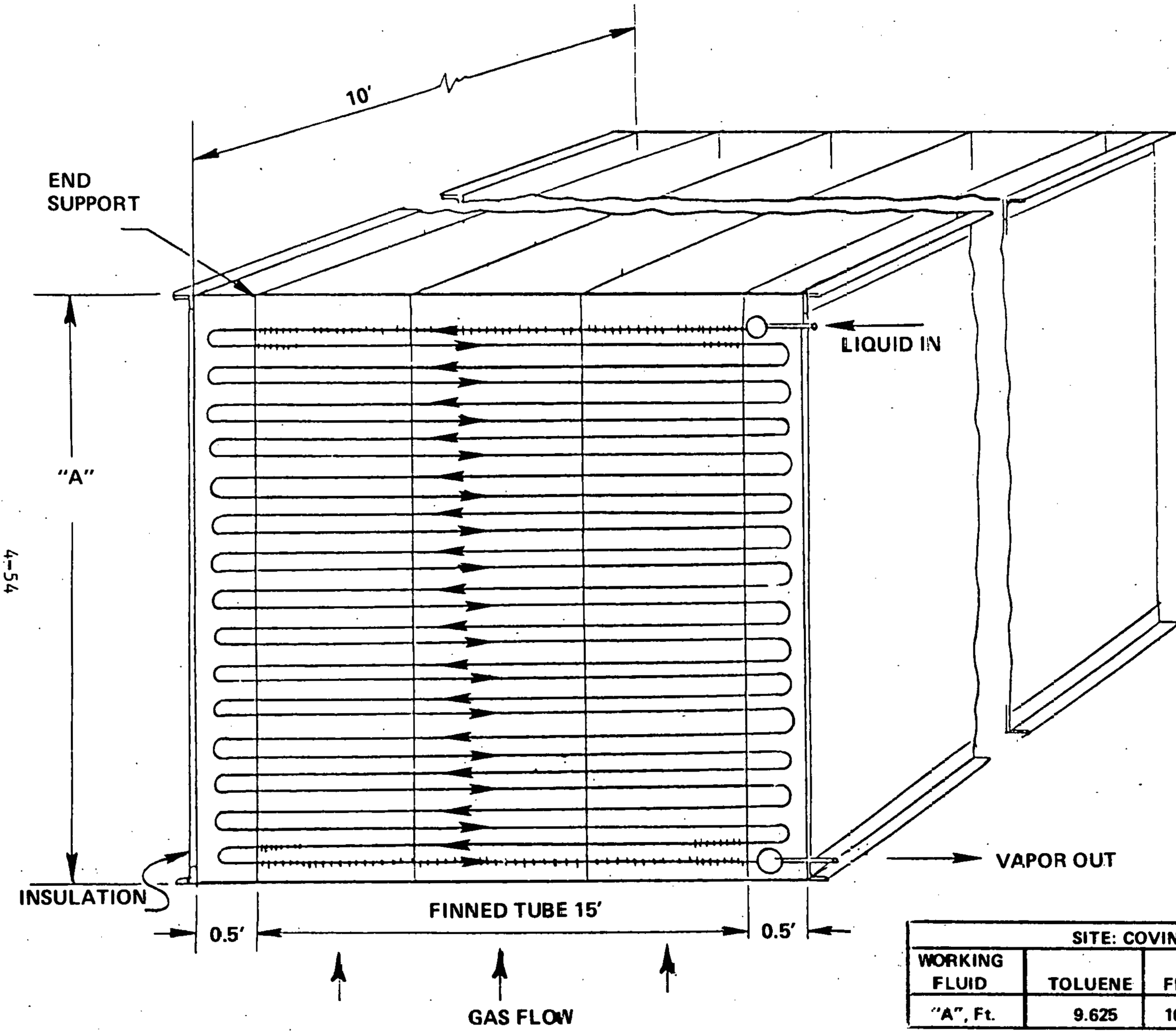
FLUID: TOLUENE			
SITE	RAYNE, LA.	COVINGTON, TENN.	BURNEY, CA.
"A", Ft.	9.625	9.625	10.3

Figure 4-20. Vapor Generator Schematic.

TABLE 4-17

VAPOR GENERATOR DESIGNS WITH OTHER ORGANIC FLUIDS (FOR COVINGTON, TN SITE)

Fluid (Operating Pressure)	Fluorinol-50 (400 psi)	RC-1 (600 psi)	RC-2 (270 psi)
Duty (Btu/Hr)	6.5127×10^7	7.3578×10^7	6.6377×10^7
Exhaust Gas			
Inlet Temperature (°F)	700	700	700
Outlet Temperature (°F)	275	219.9	266.9
Mass Flow Rate (lbs/sec)	165	165	165
Organic Fluid			
Inlet Temperature (°F)	148	151.9	146.9
Outlet Temperature (°F)	500.9	504.9	500
Mass Flow Rate (lbs/hr)	173,050	530,670	126,440
Height of the HX (feet)	10.3	9.28	9.97
Number of Passes	30	27	29
Number of Tubes Across	29	29	29
Heat Transfer Area (ft^2)	61,900	53,790	59,837
Shell-Side Pressure Drop (inch water)	4.94	3.61	4.6
Screen Pressure Drop (inch water)	0.5	0.5	0.5
Tube-Side Pressure Drop (psi)	15.8	12.3	44.5
Orifice Pressure Drop (psi)	32.4	32.4	32.4



SITE: COVINGTON, TENN.				
WORKING FLUID	TOLUENE	FL-50	RC-1	RC-2
"A", Ft.	9.625	10.3	9.28	9.97

Figure 4-21. Vapor Generator Schematic.

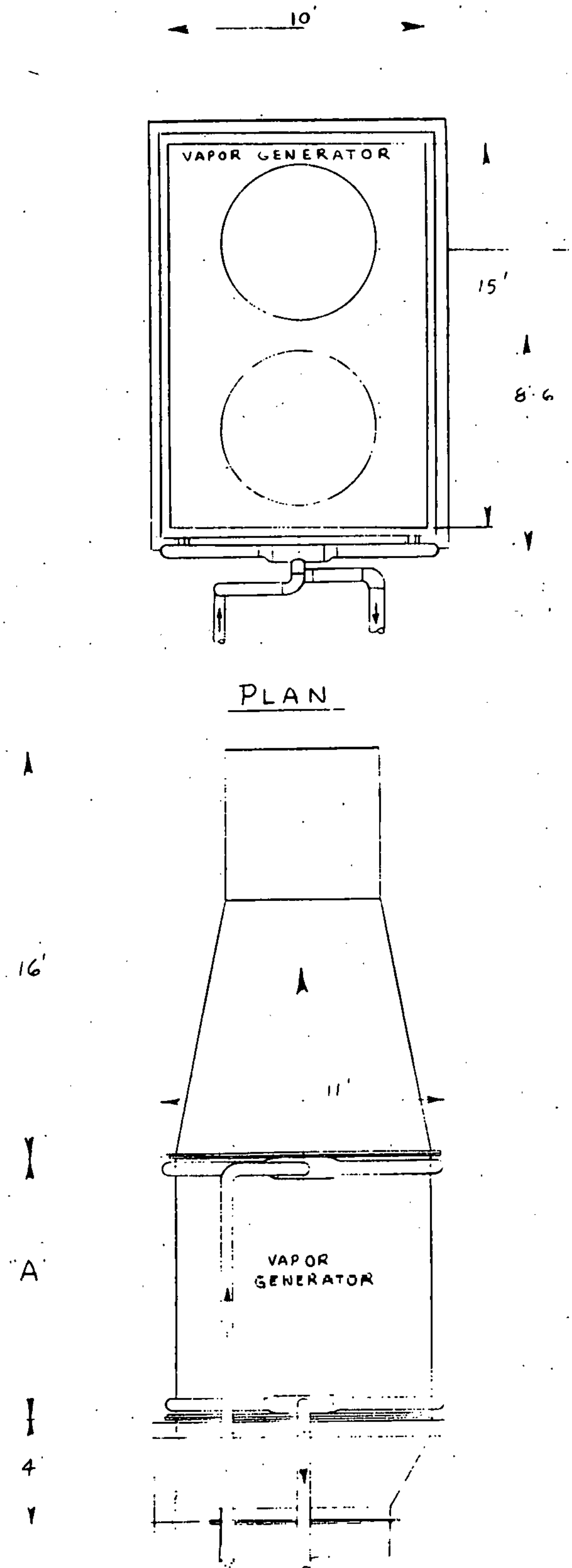


Figure 4-22. Vapor Generator Plan and Elevation Views.

The vapor generator will be made of the following materials. The boiler tubes and fins would be SA 178A, the tube supports would be A36, the headers would be SA 108B, and the casing would be low carbon steel.

4.5.3 CONDENSER

The condenser would be of the air cooled type made of a number of modules of similar design. Shown in Figure 4-23 is a plot of the temperature of working fluid and the cooling air versus the amount of heat rejected by the condenser. From the figure it can be seen that a large amount of superheat has to be removed by the condenser because of the fact that no regenerator was used in the system. Shown in Table 4-18 are the geometrical details of the tubing for the air cooled condenser. The condenser has a single pass on the working fluid side, the tubes are finned and aluminum fins are placed on steel tubes. Each unit has a forced draft fan to circulate the cooling air and one can either use a variable pitch fan and fixed speed motor or a two speed motor and fixed pitch of fan to supply different amounts of air flow to the condenser. Shown in Figure 4-24 is a drawing of the tube bundle geometry showing the proportions of the tube, fin and the spacing.

An air cooled heat exchanger was sized for each of the compressor sites and in Table 4-19 is shown the results of that sizing. The duty for each of the three condensers is shown at the top of the table. All the pertinent conditions of the working fluid and the cooling air are also shown. It can be seen that the heat transfer area is largest for the Rayne site and smallest for the Burney site. The estimated fan horsepower is also shown in the table. All three sites require ten units in parallel, each with a slightly different unit length. Shown in Table 4-20 is a comparison of the condensers for the Covington site calculated for three other fluids, Fluorinol-50, RC-1 and RC-2. The RC-1 requires the most heat transfer area, over 195,000 ft^2, and Fluorinol-50 requires the least heat transfer area approximately 130,000 ft^2. Because of this variation in the heat transfer area, the Fluorinol condenser can be made up of 8 units instead of 10 like the other fluids. Shown in Figure 4-25 is a plan view of the condenser. In this drawing are tables which show the dimension A on the drawing for each fluid and for each site. Also

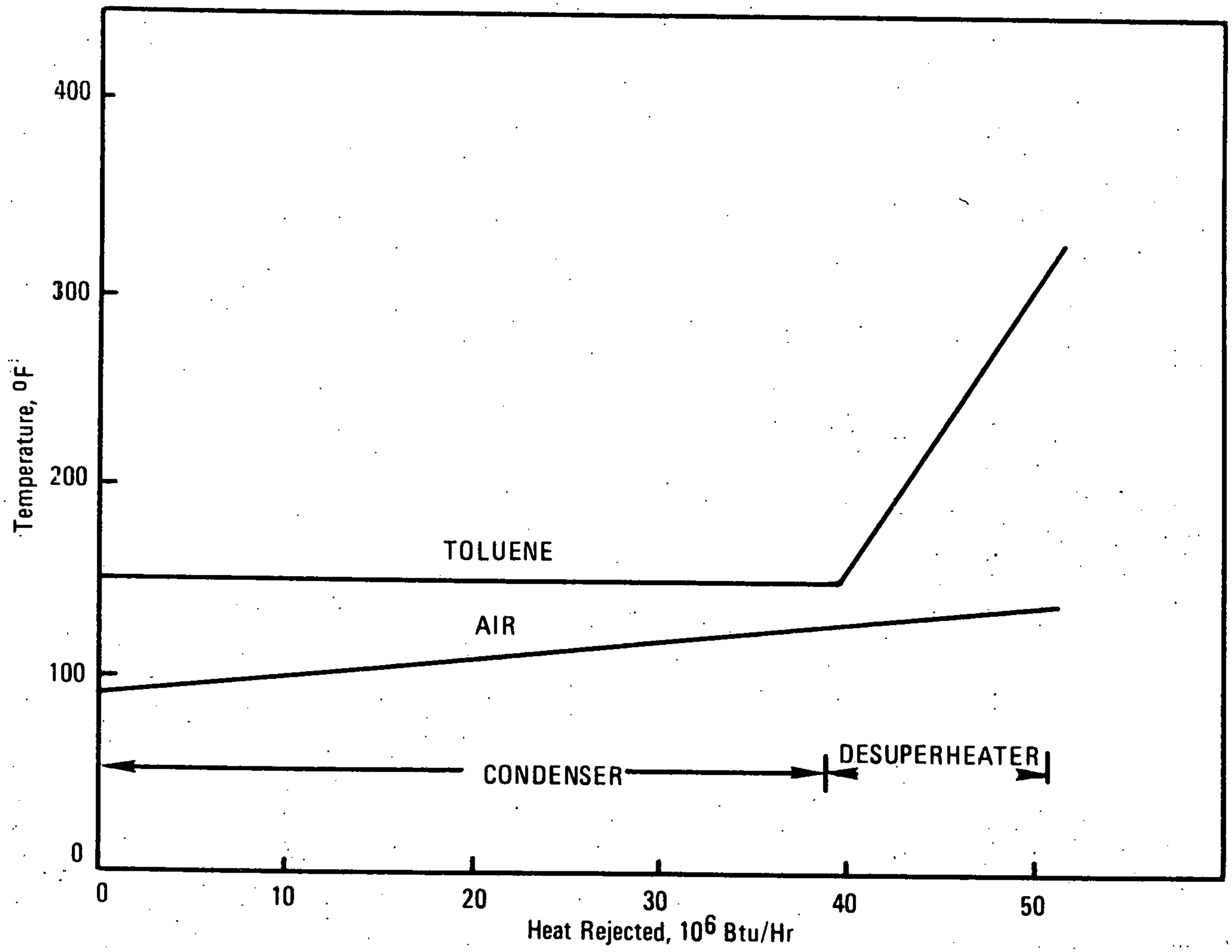


Figure 4-23. Temperature - Heat Transfer Diagram for the Air Cooled Condenser

TABLE 4-18

AIR-COOLED CONDENSER TUBE-GEOMETRY DETAILS

TYPE: Single Pass, Finned-Tube Units. Aluminum Fins on Steel Tubes.
Multiple Units Arranged in Parallel
Each Unit Required Fan for Forced-Draft Circulation
AVP or Two Speed Motors are Used as Drivers

Tube Diameters (inch)	1.25
Wall Thickness (inch)	0.06
Number of Layers	3
Triangular Pitch (inch)	2.625
Fin Outer Diameter (inch)	2.5
No. of Fins Per Inch	12
No. of Tubes Per Unit	164
Width of the Unit (feet)	12

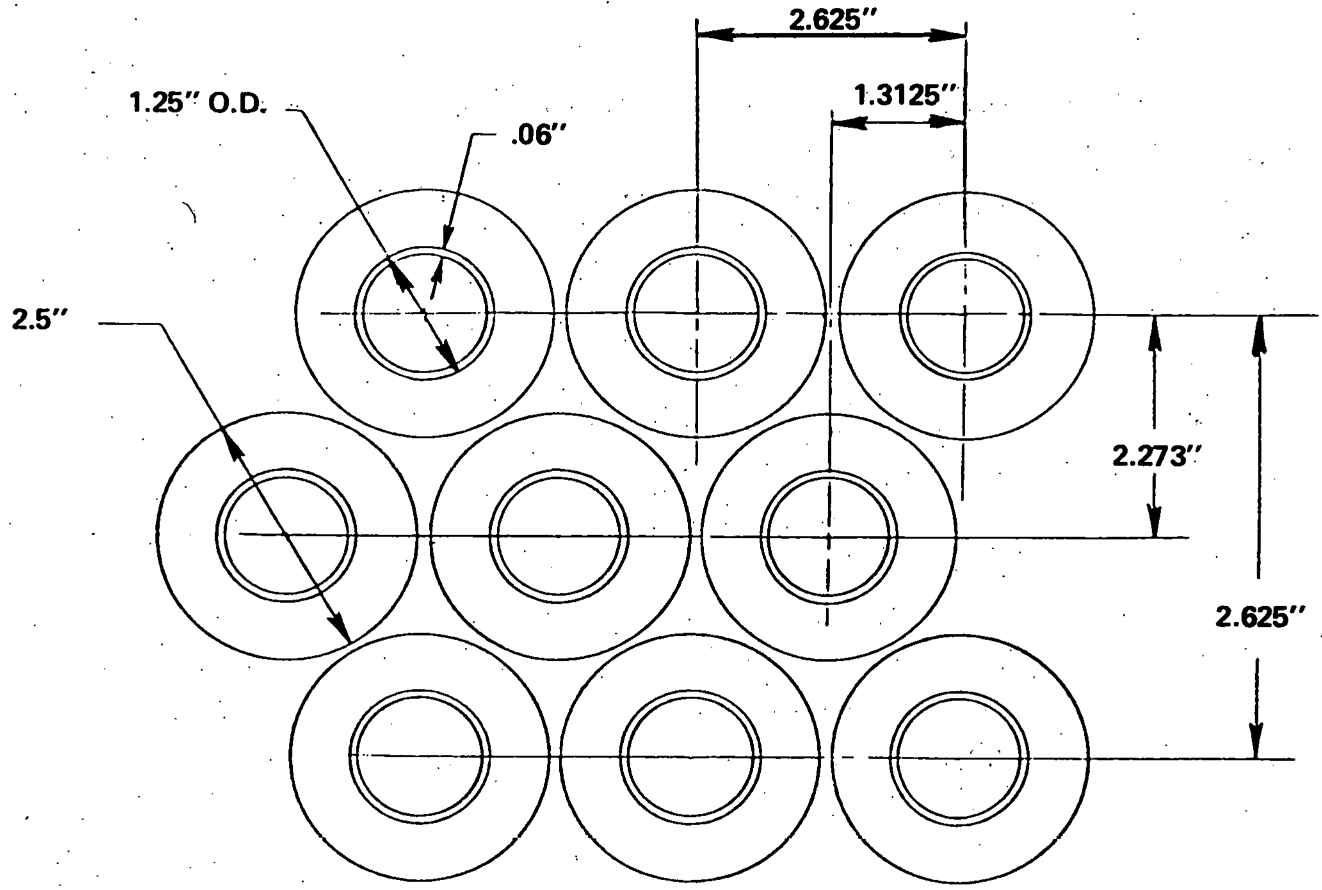


Figure 4-24. Air-Cooled Heat Exchanger Tube Spacing.

TABLE 4-19

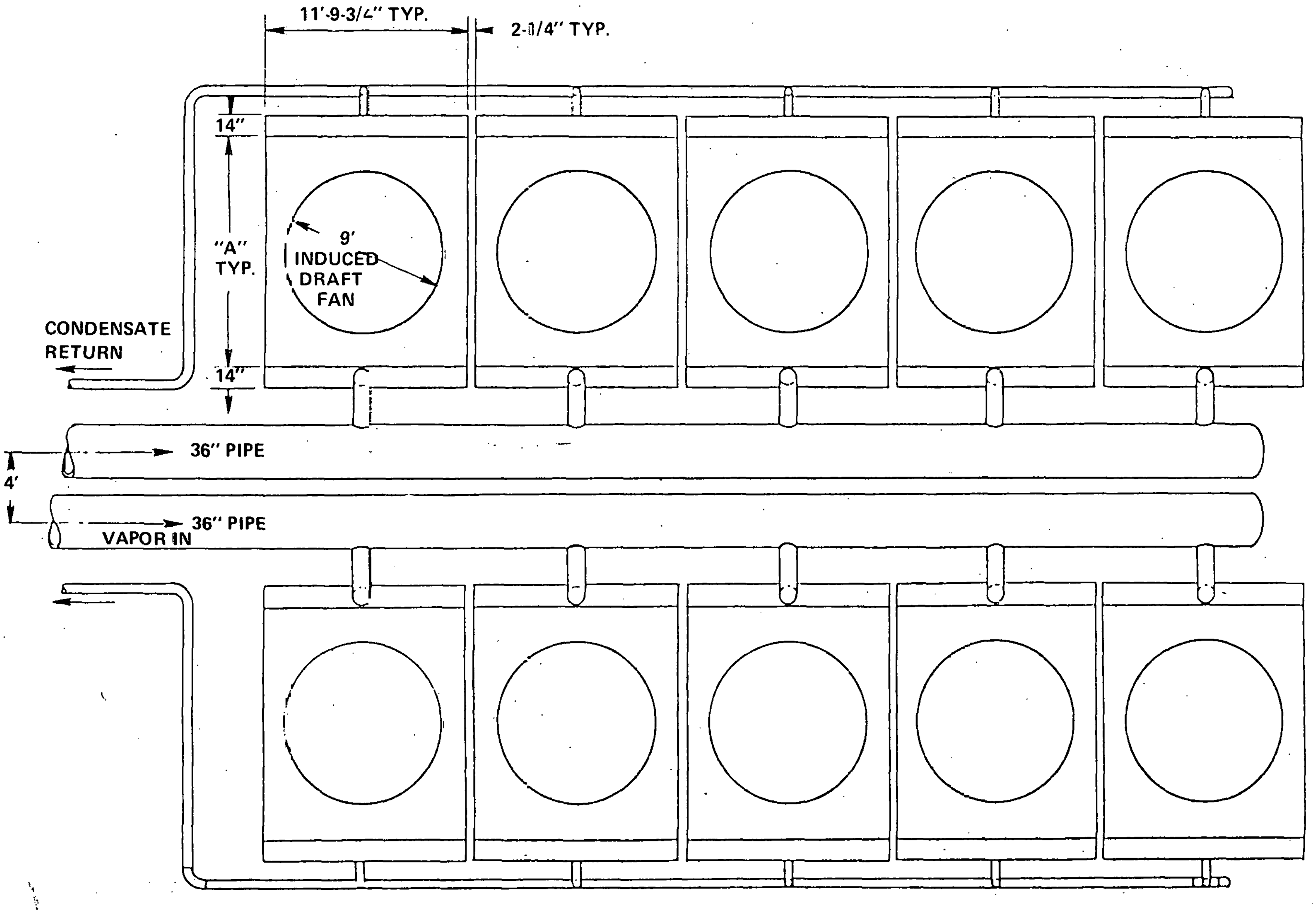
COMPARISON OF AIR-COOLED CONDENSER DESIGNS FOR THE THREE SITES FOR TOLUENE CYCLE

Site	Rayne, LA	Covington, TN	Burney, CA
Duty (Btu/Hr)	5.548×10^{7}	5.1052×10^{7}	4.901×10^{7}
Toluene			
Inlet Temperature (°F)	310.4	310.3	310.2
Outlet Temperature (°F)	145.0	145.0	145.0
Condensing Temperature (°F)	150.0	150.0	150.0
Flow Rate (lbs/hr)	243,610	224,190	215,290
Air			
Inlet Temperature (°F)	90	90	90
Outlet Temperature (°F)	125.4	125.4	125.4
Flow Rate (lbs/hr)	6.53×10^{6}	6.01×10^{6}	5.77×10^{6}
Face Velocity (ft/min)	810	918	922
Heat Transfer Area (ft^2)	184,837	169,735	162,957
Length of the Unit (feet)	13.5	12.35	11.85
No. Of Units in Parallel	10	10	10
Fan Horsepower, hp	241	301	303

TABLE 4-20

COMPARISON OF THE AIR-COOLED CONDENSER DESIGNS WITH OTHER FLUIDS (FOR COVINGTON, TN SITE)

Working Fluids	Fluorinol-50	RC-1	RC-2
Duty (Btu/Hr)	5.19×10^7	5.985×10^7	5.259×10^7
Fluid			
Inlet Temperature (°F)	200.6	275.7	196
Outlet Temperature (°F)	145	145	145
Condensing Temperature (°F)	150	150	150
Flow Rate (lbs/hr)	173,050	530,670	126,440
Air			
Inlet Temperature (°F)	90	90	90
Outlet Temperature (°F)	125.4	125.4	125.4
Face Velocity (ft/min)	1226	940	1047
Flow Rate (lbs/hr)	6.11×10^6	7.045×10^6	6.19×10^6
Heat Transfer Area (ft^2)	129,700	195,590	154,140
Length of the Unit (feet)	11.8	14.2	11.2
No. of Units in Parallel	8	10	10
Fan Horsepower, hp	506	314	381



COVINGTON, TENN.				
	TOLUENE	FL-50	RC-1	RC-2
"A" Dim.	12.35'	11.8'	14.2'	11.2'
No. of Units	10	8	10	10

Figure 4-25. Plan of the Condensers Layout.

SITE	RAYNE, LA.	COVINGTON, TENN.	BURNEY, CA.
"A" Dim.	13.5'	12.35'	11.85'
No. of Units	10	10	10

the number of units required for each design are shown. Shown in Figure 4-26 are three views of a single fan unit. Each unit stands on a frame above the ground and is provided with a motor beneath the heat exchanger to drive the fan. The materials to be used in the air-cooled condenser are as follows: SA214 for the tubes, aluminum for the fins, and SA515 for the box and special cover. The condenser for the Covington site is estimated to weigh 121,840 pounds.

4.5.4 CONTROLS

Shown in Figure 4-27 is a schematic diagram of the bottoming cycle showing the several controls which are necessary for its operation. At the left an exhaust gas diverter valve is shown to bypass the vapor generator when the bottoming cycle is not to be operational. There is a turbine flow control valve between the vapor generator and the vapor turbine. This flow control valve will also act as a stop valve in an emergency situation for the turbine. Shown in the figure are pump and turbine bypass valves. Instrumentation will be provided to read the pressure and temperature of toluene vapor coming from the vapor generator. Also, a reading of the speed of the vapor turbine shaft will be made. The bottoming cycle will be slaved to the prime mover. When the pressure reading at the exit from the vapor generator is too low, the pump bypass valve will be actuated to correct the pressure error. The turbine flow control valve responds to the temperature of the vapor generator outlet. The turbine flow control valve assures that the temperature of the toluene does not exceed a limit of slightly above 550°F. When this temperature begins to rise above 500°F the turbine flow control valve opens to allow more flow through the vapor generator to absorb the energy. This reduces the temperature at the exit of the vapor generator. In the event of an overspeed reading on the vapor turbine shaft the turbine flow control will be closed and the turbine bypass valve will open so that the flow from the vapor generator goes directly to the air-cooled condenser. These two actions keep the vapor turbine from running away in the case of a sheared shaft or in the case of an alternator being driven and the breakers being tripped. After the turbine bypass valve has opened allowing the vapor to go directly to the condenser the control system actuates the exhaust gas diverter valve and diverts the flow around the vapor

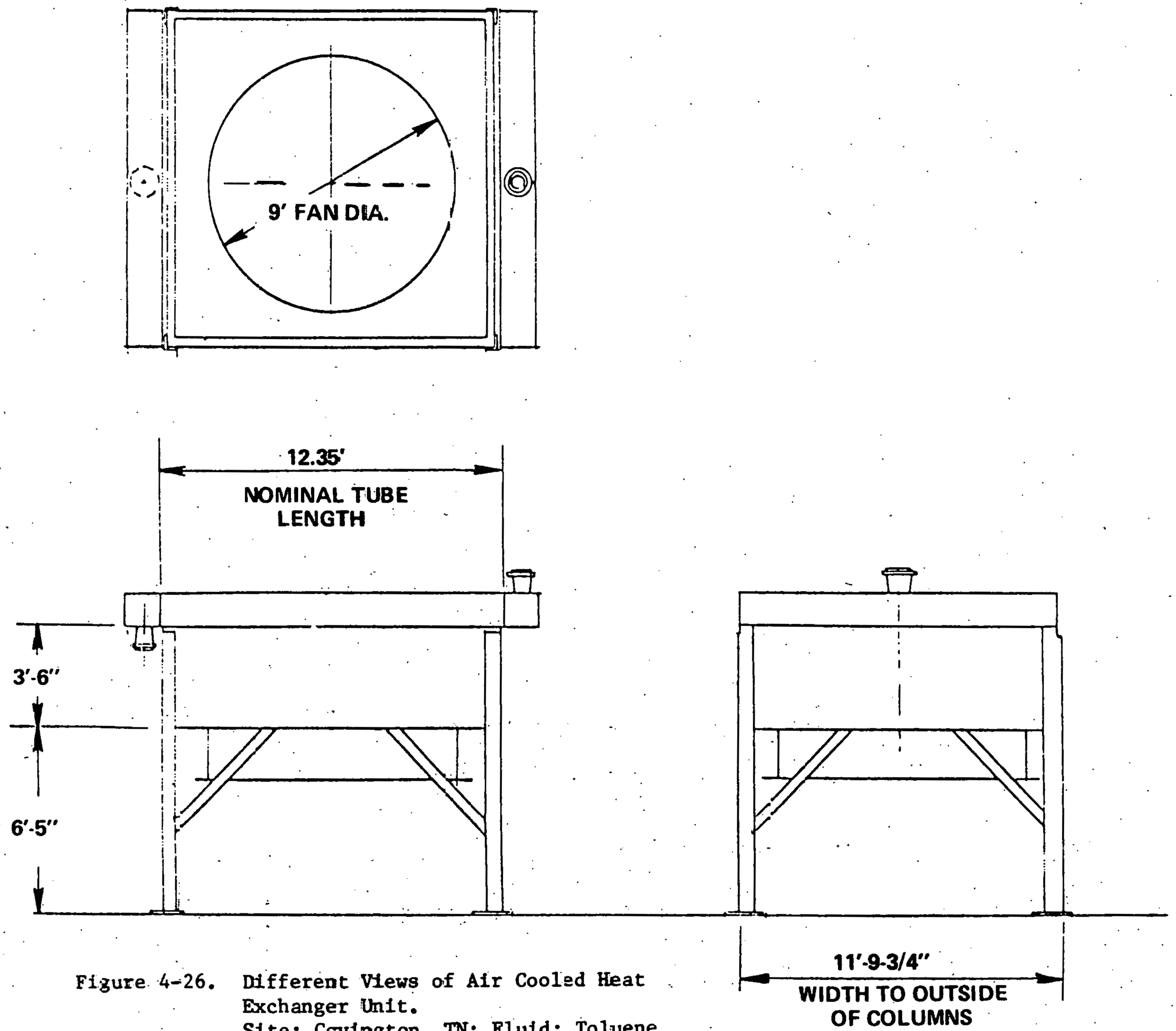


Figure 4-26. Different Views of Air Cooled Heat Exchanger Unit.
Site: Covington, TN; Fluid: Toluene

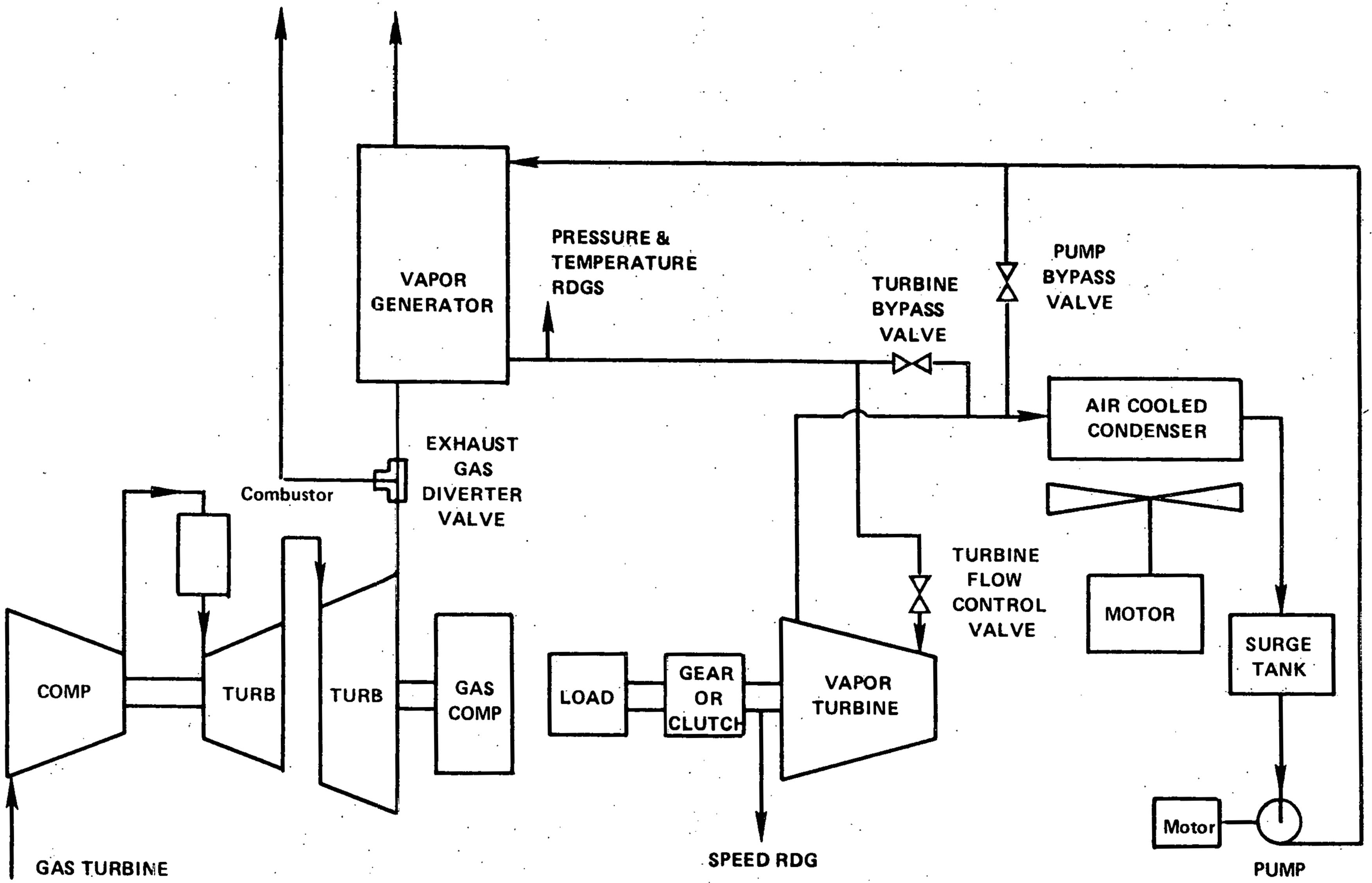


Figure 4-27. Bottoming Cycle Control Schematic.

generator. At this point there is no energy going into the bottoming cycle and the turbine will stop. The control then sets all of the valves in correct position for startup. For the case in which the bottoming cycle drives the generator, as for the Texas Gas Site in Covington, TN, controls will be added to permit the bottoming cycle turbine to respond to the process of synchronizing the generator with a line which it is going to feed.

4.5.5 OTHER COMPONENTS

Other system components for the most part will be commercially available equipment. Specifications covering these components indicating the fluid type (toluene), pressures, temperatures, flow rates, etc., dictated by the system cycle design will be necessary. In the area of piping, experience gained by the refineries in the design and operation of toluene production plants will be utilized. Welded type joints would be used wherever possible through out the toluene vapor or liquid piping system. Where mechanical joints of the flange type are required a Flexitallic (i.e., spiral metal and asbestos) or Grafoil gasket is recommended. Grafoil as manufactured by Union Carbide Corp. would also be utilized for valve stem packing glands. Approximately 9 million pounds of toluene are produced in the USA each year. With proper design specifications and adequate quality control of manufacturing and installation, no major problems are anticipated with system components transporting this fluid.

4.5.6 INSTALLATION OF THE BOTTOMING CYCLE ON THE COMPRESSOR SITES

Shown in Figure 4-28 is the concept of the bottoming cycle apparatus for the three sites that have been selected for this design. Shown on the right hand side of the drawing are the vapor generator connected by an exhaust duct to the turbine at the lower part of the figure, the equipment skid which is to the left of the vapor generator and the air cooled condenser. Dimensions are shown on this drawing of a typical installation. The air-cooled condenser is approximately 60 feet long and 45 feet wide. The vapor generator is approximately 18 ft x 10 ft. And the equipment skid is approximately 18 ft long and 12-1/2 ft wide. Shown in the cross section AA are the exhaust duct for the gas turbine vertically above the turbine and a horizontal exhaust duct leading to the

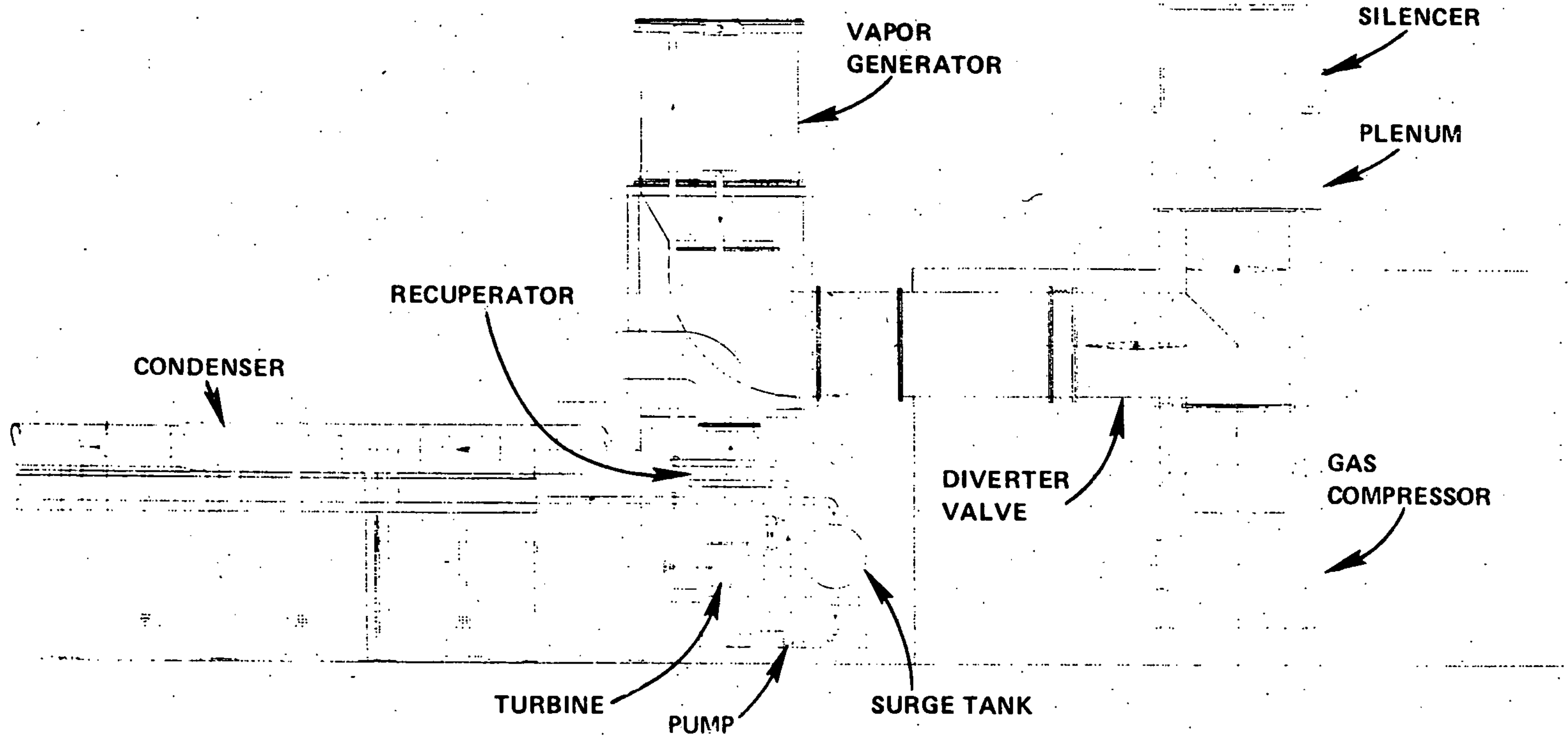


Figure 4-28. Pipeline Bottoming Cycle System Concept:
Vertical Gas Turbine Exhaust (GE-AEPD) (Cont'd).

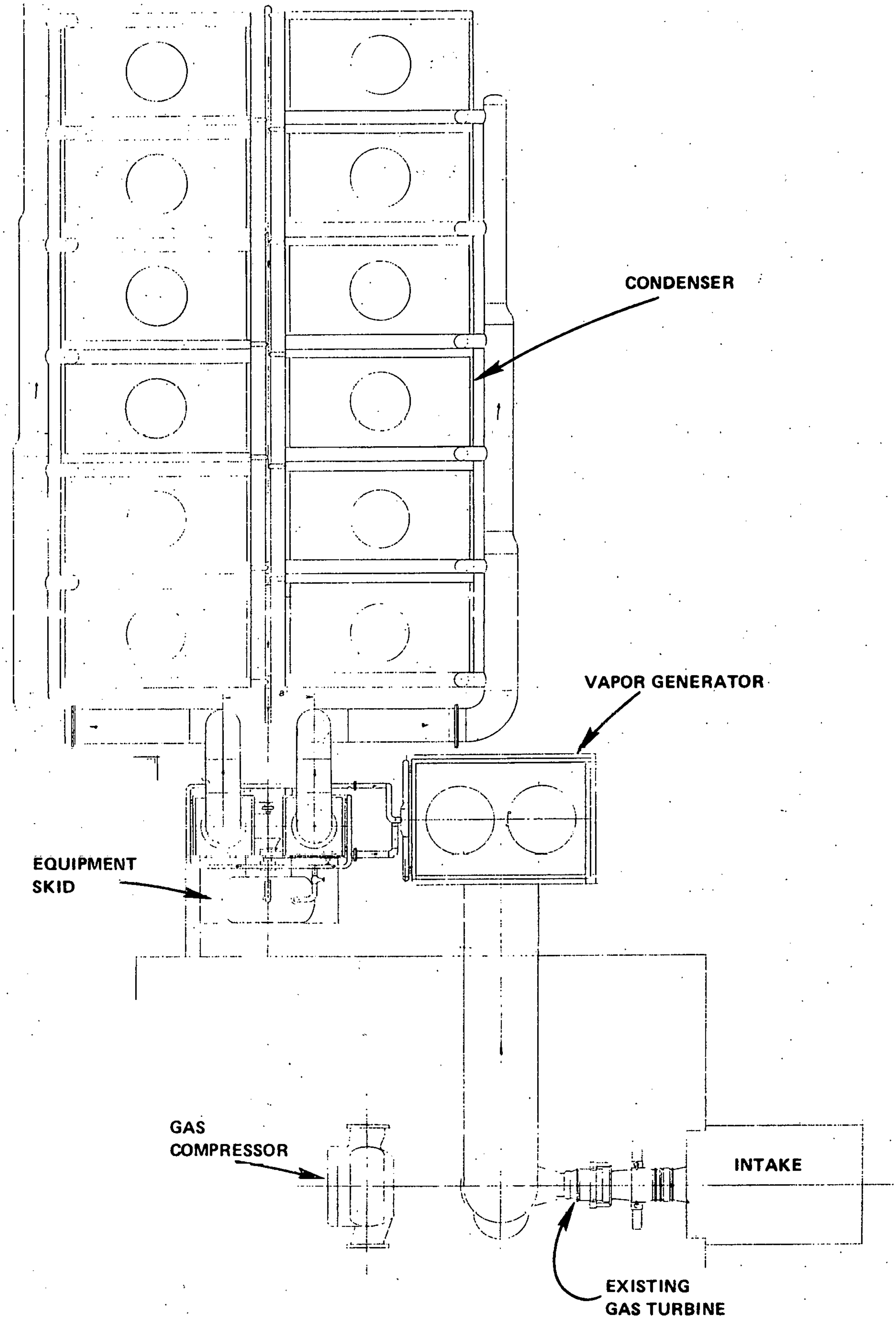


Figure 4-28. Pipeline Bottoming Cycle System Concept: Vertical Gas Turbine Exhaust (GE-AEPD)

base of the vapor generator. The equipment skid is shown to the left of the vapor generator. In this design a recuperator was incorporated. Shown in Figure 4-29 is a typical installation of the vapor generator and the exhaust duct connecting the turbine exhaust hood to the vapor generator. Shown beneath the vapor generator is the diverter valve which when in the closed position allows the exhaust gases to go through the silencer and out through a stack. However when this valve is open the exhaust gas flow turns a corner upward and goes through the vapor generator.

4.5.7 COLUMBIA GULF TRANSMISSION COMPANY INSTALLATION

Shown in Table 4-21 are descriptive information concerning the Rayne site of the Columbia Gulf Transmission Company. Shown in Figure 4-30 is a plan view of the installation at this site. On the left is shown the centerline of the gas turbine; the exhaust hood is shown delivering exhaust gas to the right. The vapor generator is between the exhaust hood of the turbine and the stack. Above the vapor generator is shown the equipment skid and connecting the equipment skid to the air-cooled condenser on the right are the 36-inch-diameter pipes. The gas compressor which is put in parallel with the reciprocating compressors at the site is shown to the right of the equipment skid and directly below the 36-inch-diameter ducts connecting the equipment skid to the bank of condensers.

4.5.8 PACIFIC GAS AND ELECTRIC COMPANY INSTALLATION

Shown in Table 4-22 is descriptive material concerning the Burney Site of the Pacific Gas and Electric Company system. Shown in Figure 4-31 is a plan view of the bottoming cycle installed on the Burney site. The gas compressor existing at the Burney site now is located directly to the right of the gas generator and turbine hood shown at the upper left hand part of this figure. This compressor has an axial inlet and a side outlet. In the present installation, however, the bottoming cycle drives the same compressor as the gas turbine drives. So a new Ingersoll-Rand gas compressor having a side inlet, a side outlet and a shaft at each end has to replace the existing compressor. The piping change can be seen where the gas pipe comes down the compressor centerline and has 3 elbows to take the gas to the side inlet. In this particular design, the bottom-

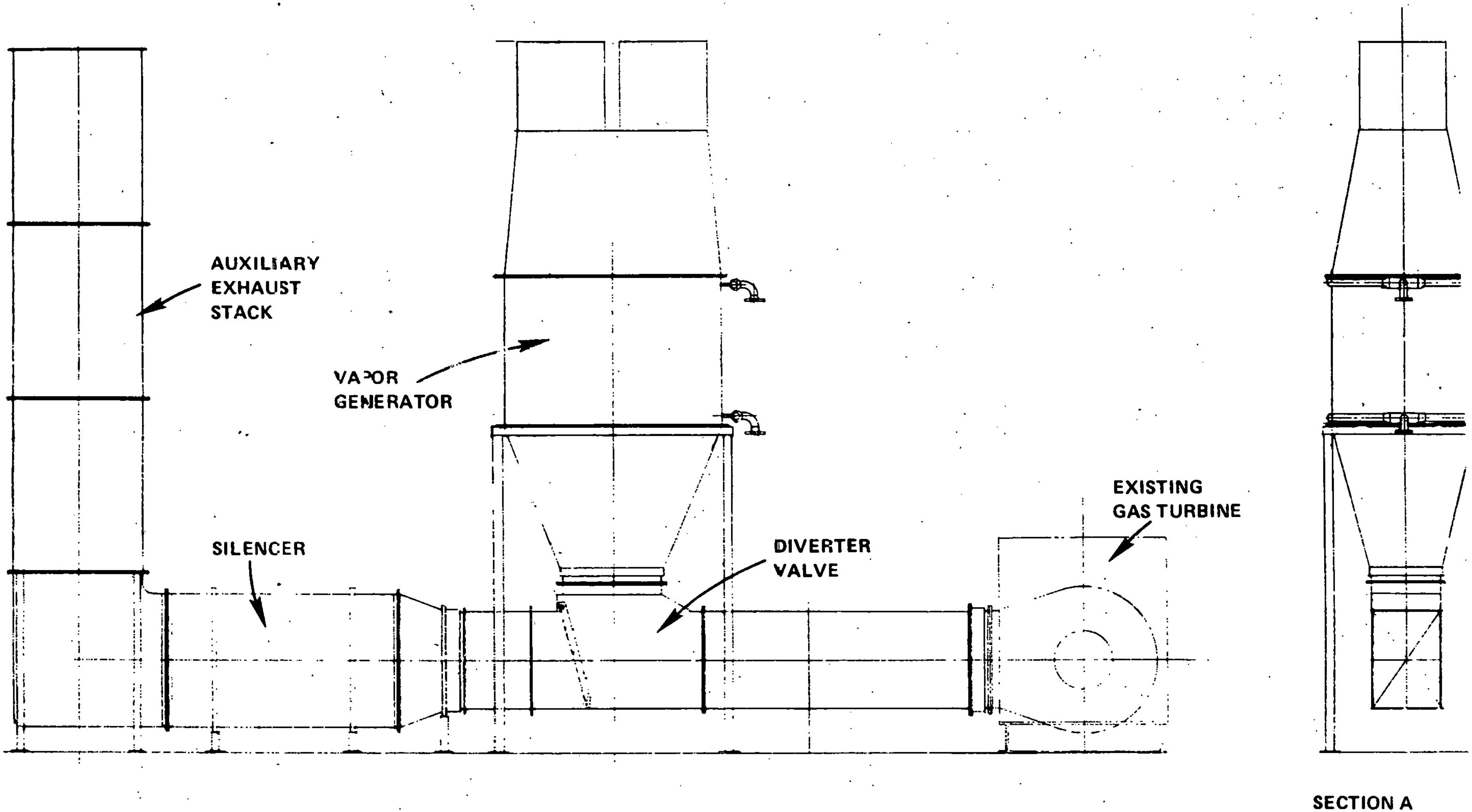


Figure 4-29. Pipeline Bottoming Cycle System Concept: Horizontal Gas Turbine Exhaust (GE-AEPD).

TABLE 4-21

DESCRIPTION OF RAYNE, LA SITE

Pipeline Company	Columbia Gulf
Nearby Airport	Lafayette, LA
Distance (Miles)	16
Annual Average Temperature (°F)	68
Gas Turbine To Be Bottomed	
Type	Simple
Site Power (HP)	13622
Site Heat Rate (BTU/HP-Hr)	9935
Gas Turbine Manufacturer	Rolls Royce
Power Turbine Manufacturer	Cooper Bessemer
Exhaust Temperature (°F)	727
Flow Rate (lb/sec)	164
Gas Compressor	
Type	Centrifugal
Manufacturer	Cooper-Bessemer
Model	RF2BB-30
Other Gas Turbines	2 Each
Type	Simple
Rated Power (HP)	12500
Rated Heat Rate (BTU/HP-Hr)	10060
Reciprocating Engines	7 Each
Rated Power (HP)	2000
Heat Rate (BTU/HP-Hr)	8780
Bottoming Cycle Load	
Compressor in Parallel to Reciprocating	Compressors
Bottoming Cycle Power (HP)	5976
Electrical Load (KWe)	538
Combined Heat Rate (BTU/HP-Hr)	6972

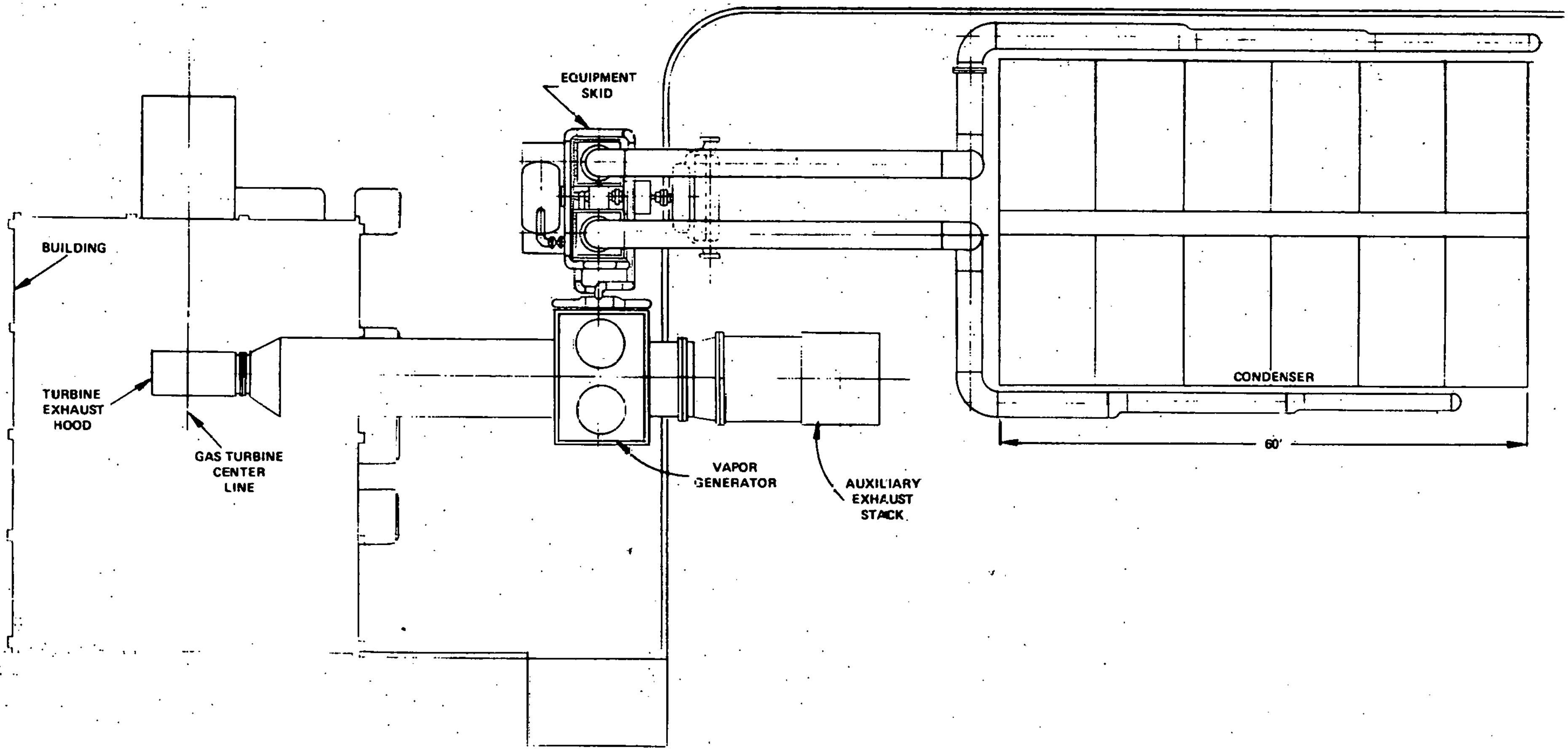


Figure 4-30. Bottoming Cycle Unit 1010 (Columbia Gulf Transmission Co., Rayne, LA Site).

TABLE 4-22

DESCRIPTION OF BURNEY, CA SITE

Pipeline Company	Pacific Gas & Electric
Nearby Airport	Redding, CA
Distance (Miles)	60
Annual Average Temperature (°F)	56
Gas Turbine To Be Bottomed	
Type	Simple
Site Power (HP)	13170
Site Heat Rate (BTU/HP-Hr)	9394
Gas Turbine Manufacturer	General Electric
Power Turbine Manufacturer	Ingersoll Rand
Exhaust Temperature (°F)	713
Flow Rate (lb/sec)	147
Gas Compressor	
Type	Centrifugal
Manufacturer	Ingersoll Rand
Model	CVP-30
Other Gas Turbines	1 Each
Type	Recuperated
Rated Power (HP)	9100
Rated Heat Rate (BTU/HP-Hr)	8750
Reciprocating Engines	None
Rated Power (HP)	-
Heat Rate (BTU/HP-Hr)	-
Bottoming Cycle Load	
Same Compressor as Gas Turbine	
Bottoming Cycle Power, (HP)	5397
Electrical Load (KWe)	476
Combined Heat Rate (BTU/HP-Hr)	6735

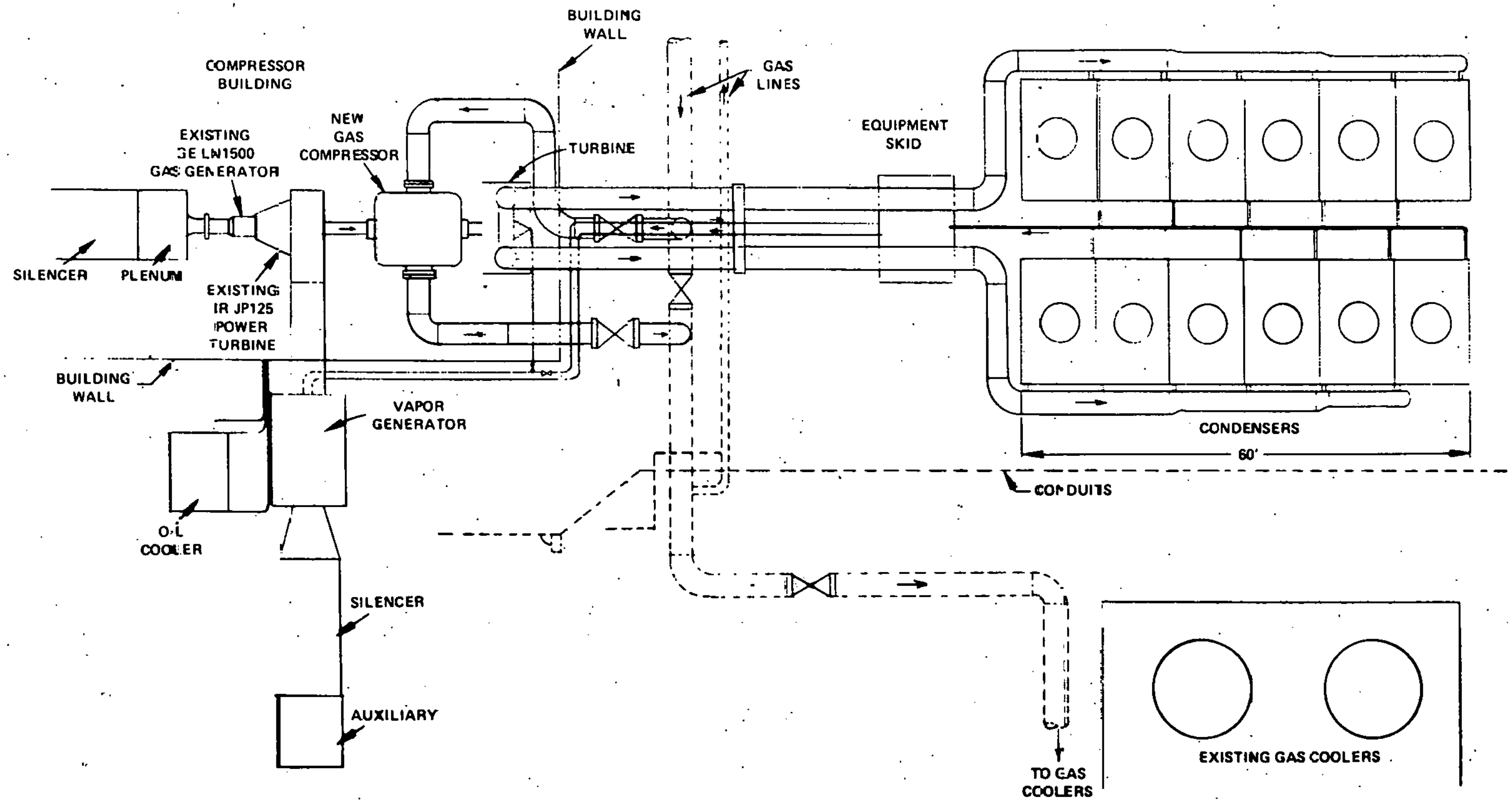


Figure 4-31. Elevation, Proposed Pipeline Bottoming Cycle System Burney Compressor Station

ing cycle turbine was removed from the equipment skid and placed in the compressor building. An evacuated cover is put over the vapor turbine so that any leakage of the working fluid can be drawn out by the exhaust fans. The equipment skid is shown adjacent to the condensers near the right hand side of the sketch. Shown in the lower right hand corner of the sketch are existing gas coolers used to take the heat out of the natural gas after it has gone through the compressor. In this design the gas turbine exhaust duct runs horizontally away from the power turbine and turns a right angle up into the vapor generator. Below the vapor generator in the sketch the silencer and exhaust stack, which can be utilized in the event that the bottoming cycle is not operational, can be seen. In this instance, a clutch is provided after the vapor turbine in order to declutch the vapor turbine from the compressor should the bottoming cycle be inoperative. Shown in Figure 4-32 is a cross section of the installation. Here the location of the expander skid is shown and the location of the equipment skid is also shown. Also the new gas compressor with the side inlets and outlets can be seen. Shown to the left is th vapor generator sitting above the exhaust duct which comes out horizontally from the compressor house. The building that the gas turbine, compressor and vapor turbine are housed in is also shown. In addition a purge duct can be seen placed over the vapor turbine so that all the leakage from the bottoming cycle is carried away. Shown on the right are the condensers with their fans and motors.

4.5.9 TEXAS GAS TRANSMISSION CORPORATION INSTALLATION

Pertinent factors about the Covington, TN Site are shown on Table 4-23. Shown in Figure 4-33 is a plan view of the pipeline bottoming cycle installed in the Covington, TN site of the Texas Gas Transmission Corp. The existing facility is shown in the upper left hand corner. Extending downward from the compressor building in the figure is the exhaust duct leading to the vapor generator. This has been placed 50 feet away from the compressor building for safety. Shown in the bottom portion of the figure is a new building to house the equipment skid and the control room and directly to the right of this building are shown the air-cooled condensers. Shown in Figure 4-34 is a cross section through the compressor building at the power turbine showing the exhaust duct leading to the plenum and silencers above the turbine and extending out to the right to

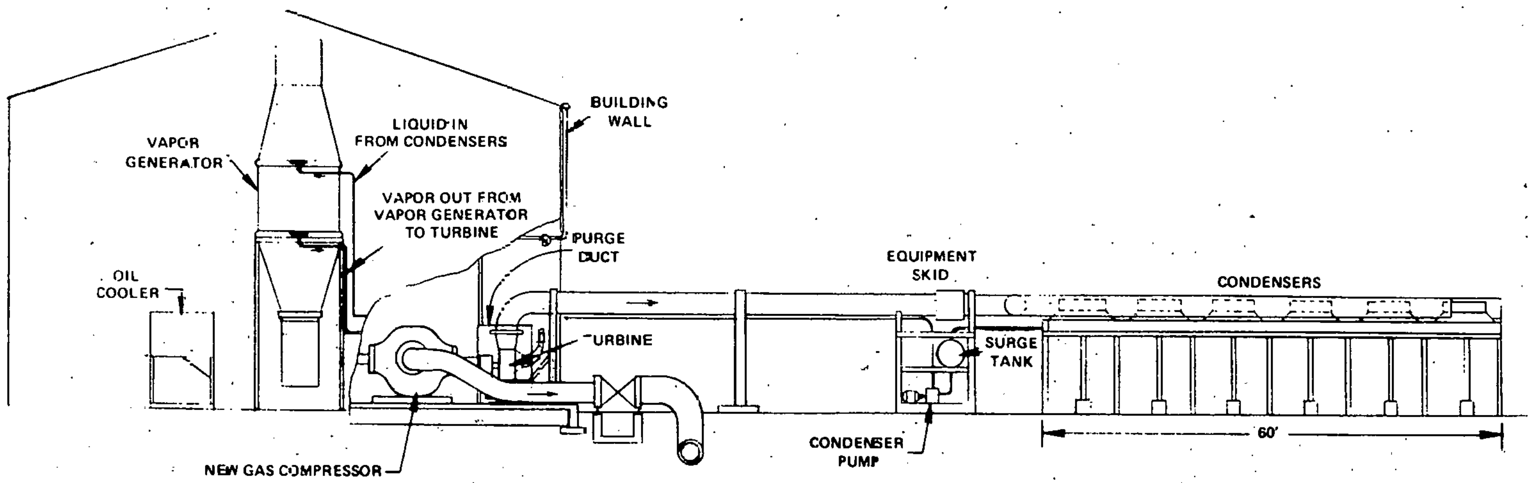


Figure 4-32. Proposed Pipeline Bottoming Cycle System
Burney Compressor Station

TABLE 4-23

DESCRIPTION OF COVINGTON, TN SITE

Pipeline Company	Texas Gas		
Site Location	Covington, TN		
Nearby Airport	Memphis, TN		
Distance (Miles)	35		
Annual Average Temperature (°F)	62		
Gas Turbine To Be Bottomed			
Type	Simple		
Site Power (HP)	13835		
Site Heat Rate (BTU/HP-Hr)	10567		
Gas Turbine Manufacturer	Pratt & Whitney		
Power Turbine Manufacturer	Cooper Bessemer		
Exhaust Temperature (°F)	700		
Flow Rate (lb/sec)	165		
Gas Compressor			
Type	Centrifugal		
Manufacturer	Cooper Bessemer		
Model	RF2B-24		
Other Gas Turbines	None		
Type	-		
Rated Power (HP)	-		
Rated Heat Rate (BTU/HP-Hr)	-		
Reciprocating Engines	5 Each	3 Each	2 Each
Rated Power (HP)	1500	1500	2500
Heat Rate (BTU/HP-Hr)	7740	7450	7310
Bottoming Cycle Load			
Generator - Sell Power to TVA			
Bottoming Cycle Power (HP)	5502		
Electrical Load (KWe)	500		
Combined Heat Rate (BTU/HP-Hr)	7659		

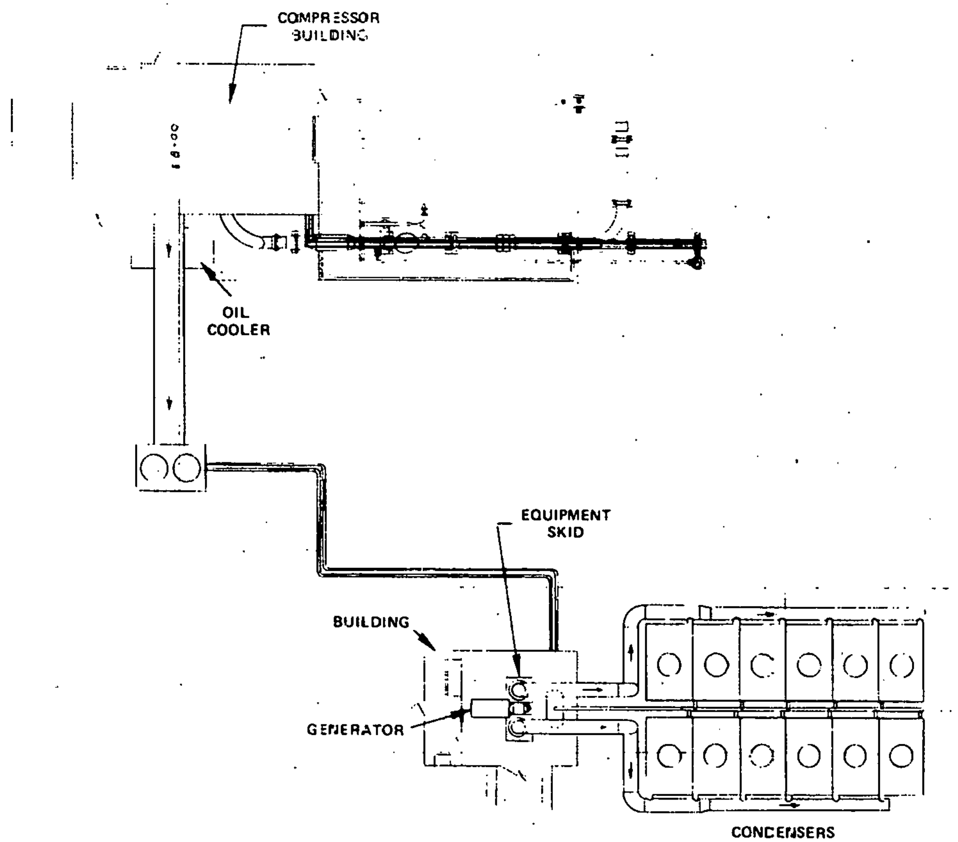


Figure 4-33. Pipeline Bottoming Cycle System, Area Plan (Texas Gas Transmission Corp.)

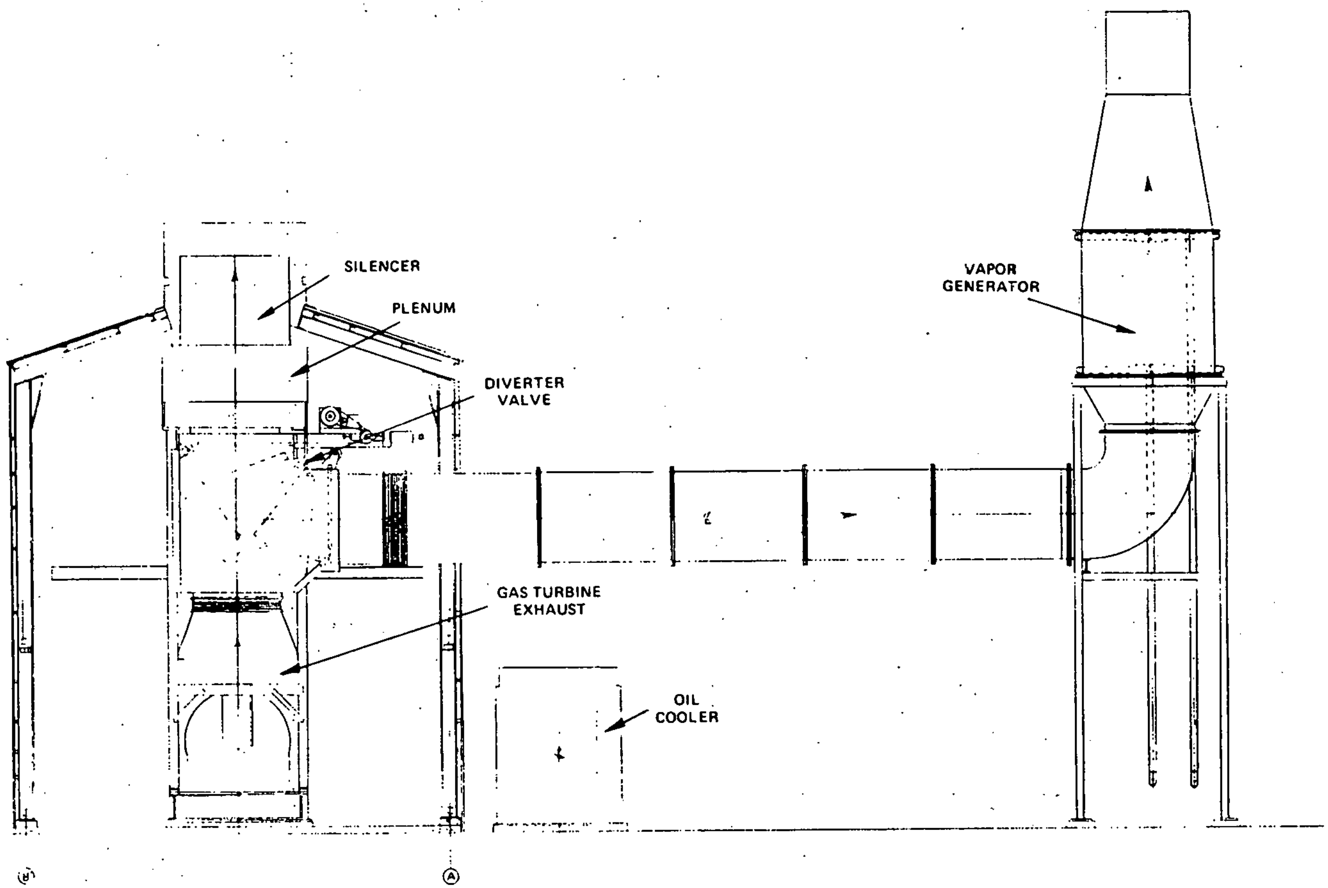


Figure 4-34. Pipeline Bottoming Cycle System, Section Thru Turbine and Vapor Generator (Texas Gas Transmission Corp.)

the vapor generator. The diverter valve is shown at the junction of the vertical and horizontal pipes. This valve is used to select whether the exhaust gas flows through the silencer in the normal way or goes through the vapor generator to operate the bottoming cycle. As can be seen from the drawing, the roof of the compressor building must be raised substantially in order to accommodate the diverter valve above the exhaust power turbine. Shown in Figure 4-35 is a cross section at 90° to the previous drawing showing the vapor generator, the equipment skid and the condenser with its fan motors. Since at this site electricity will be generated, the electric generator is shown to the left of the gearbox into which the turbine is made to drive. Also shown on the equipment skid are the surge tank, the feed pump, the turbine and the recuperators.

4.6 CONCLUSIONS

The following summarizes some of the facts that have come out of the design of the bottoming cycle and the installation of the bottoming cycle on the several sites. There are a number of working fluids which provide comparable performance but no one fluid has all of the following characteristics:

Good Thermodynamic Characteristics
Low Toxicity
Nonflammability
Low Cost
Good Materials Compatibility

All three sites can get more than 25% reduction in heat rate which is significantly more than the original target value of 20%. There are no insurmountable component design problems to solve before the bottoming cycle can be designed. Reasonable installation designs were found for the three types of power utilization, namely pumping gas with a small additional compressor, pumping gas by adding more power to the main compressor and generating electricity for sale to the grid. Bottoming cycles and their installations are feasible. Careful study is required, however, to minimize the installation cost.

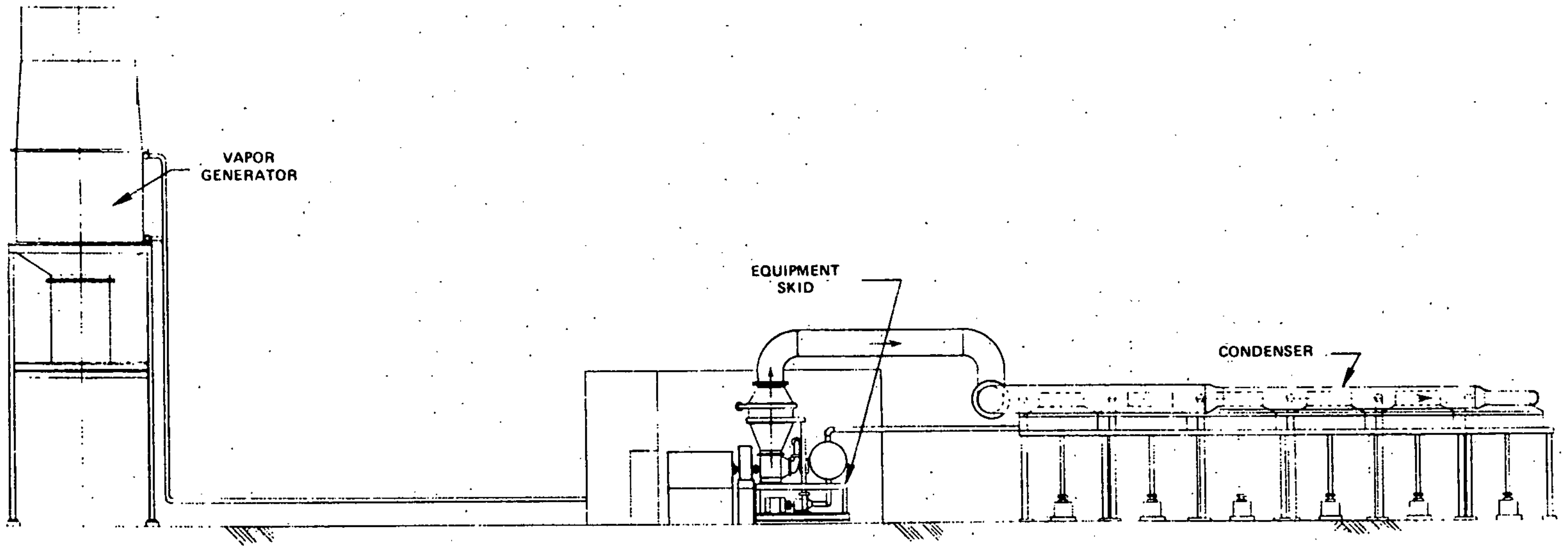


Figure 4-35. Pipeline Bottoming Cycle System, Area Elevation Plan (Texas Gas Transmission Corp.)

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SECTION 5

ECONOMIC ASSESSMENT

5.1 INTRODUCTION

The preliminary system design reported in the previous section was for optimized values of the prime system variables. In this section is presented the economic evaluation of the three sites for which preliminary installation designs were carried out. The economic evaluations are presented for pipeline bottoming cycles composed of commercially available components. The costs associated with bringing the bottoming cycle to a commercially available state were not included in the costs shown. It should be pointed out that the economic analyses in this section do not apply to the demonstration hardware or installation.

The analyses were conducted from the viewpoint of the pipeline company so that an evaluation could be made of the potential for commercialization by means of industry funding. The economic evaluation utilized industry investment return standards as provided by capital investment opportunities considering both risk and return. The value of energy saved and any other benefits were applied based upon the preliminary design. Sensitivity analyses of the effects on the economics of the cost of fuel being utilized by the compressor station, capital cost, investment tax credit, return, depreciation, debt cost, escalation rates and discount rates were carried out.

In the economic assessment a figure of merit is first established and then economic analysis techniques were discussed. The bottoming cycle apparatus and installation cost for the three sites are next discussed. The economic worth and attractiveness of each of the three sites is presented using the aforementioned analysis techniques. The effects of potential institutional changes are explored resulting in some recommendations. The energy consumption during bottoming cycle manufacture and installation is also presented.

5.2 ECONOMIC FIGURE OF MERIT

In order to optimize the Rankine Bottoming cycle (RBC), it was necessary to determine what economic figure of merit would be most useful. A study was made of the several ways in which a bottoming cycle could be used. These can be grouped into four classes:

1. Run prime mover at full power and use the RBC power to augment it. (More work for the same fuel.)
2. Throttle back the prime mover to provide the original total power. (The same work for less fuel.)
3. Shut down less efficient units on the same site to maintain the same station power. (The same work for less fuel.)
4. Generate electricity with the RBC power. (Extra valuable energy for same fuel.)

After considerable thought, it became apparent that the correct figure of merit for financial optimization is the specific present worth cost* of the RBC, in dollars per horsepower. This parameter can be used to optimize a cycle for a particular site or overall system. It is subject to certain limitations, however. For Cases 1 and 4 above, it is completely correct, since all the new power can be used. For these cases, the best RBC is the one which adds power at the lowest cost. Typically, the curve of dollars per horsepower versus amount of power added will have a minimum. At less added power, the fixed costs predominate, and at the highest added power, the law of diminishing returns increases the specific cost per horsepower. For Cases 2 and 3, the situation is more complex. It becomes obvious that the overall Specific Fuel Consumption of the combination of prime movers, RBC, and throttled or shutdown units must provide an improved SFC for the entire package. If it does not, and several cases studied did not, in fact, show improvement, the

* The present worth cost of the RBC is defined as the original installed cost, plus the present worth of O & M costs and replacement of major components.

RBC is not economic no matter how cheap it is. If the combination is better, then the specific dollars per horsepower is the correct RBC financial figure of merit. Based on this analysis, it was recommended and accepted for use in the cycle optimization.

5.3 ECONOMIC ANALYSIS TECHNIQUES

The key problem in the economic analysis of the pipeline bottoming cycle is to evaluate it from the viewpoint of the pipeline owner. Since the pipeline companies are almost all regulated, either by the Federal Energy Regulatory Commission (FERC) or the State Public Utility Commission (PUC), any evaluation must include the effect of this regulation. In the initial phases of this contract, a study was made of a previously developed financial projection model which permits the simulation of pipeline company regulations over a period of many years using FERC rules as well as appropriate tax laws. A sample problem was set up, analyzed, and the results sent to the team's participating pipelines. Based on their review and recommendations, a simpler method, using the incremental cost of services concept, was selected and used for the remainder of this study. Both these methods are described below.

5.3.1 FINANCIAL PROJECTION MODEL FOR REGULATED PIPELINES

The Systems, Science and Software (S^3) Company has prepared, under ERDA contract E(04-03)-1171, a financial projection model for regulated pipeline companies (Reference 16). This computer code permits the economic simulation of a pipeline company over a period of many years. The complete financial description of the company can be simulated, including the debt-to-equity ratio, reinvestment strategies, tax options, depreciation options, effects of FERC regulations, dividend payment decisions, and escalation options.

By modeling the effect of energy conservation techniques with the S^3 computer code, it is possible to find the optimum strategies for investment as well as finding those circumstances which affect the use of energy conservation techniques.

In order to assess the usefulness of this computer code, General Electric has set up a fictional pipeline company, based on data from FERC annual reports (Reference 17). The results of the study of a fictional pipeline company are presented in Appendix A and were sent to the participating pipeline companies for their approval of the methodology used in the economic assessment of the bottoming cycle equipment installation.

The comments received from the three pipeline companies all indicated that they felt that this method was basically too cumbersome, and not the way they would assess the potential purchase of bottoming cycle equipment. Although some of the companies use financial projection models for large capital additions, such as a major new pipeline, the size of the RBC investment was perceived as too small to warrant this complex modeling. The pipeline companies prefer to use the same method they normally use to justify their rates to the FERC when changes in equipment are made in the normal course of business. This method, the "Incremental Cost of Service" method, is also used, in part, to inform customers of necessary rate changes. Hence, the incremental cost of service method was recommended by all as the preferred way to analyze the effect of the bottoming cycle on company economics.

5.3.2 INCREMENTAL COST OF SERVICE METHOD

The incremental cost of service method is a relatively simple way to analyze capital investment decisions. It is essentially a buildup of total annual incremental costs associated with an individual proposal using rate regulations prescribed by the Federal Energy Regulatory Commission. Since the customers of a gas pipeline company are committed to repaying the investment in pipeline facilities, in the form of the rates they pay, they are very interested in the net change in these rates caused by an additional investment made by the pipeline, their supplier. The incremental cost of service, in addition to being the most important tool used for assessing the merits of new investments, is also the format used to convey this rate effect to the FERC staff for approval and to the company's customers.

The pipeline companies use this method because in addition to being required by FERC, it more nearly reflects the methods used to select investments, and more easily shows the immediate rate change effects to pipeline customers.

Key assumptions in this calculation are: first, no change in the company's capital structure occurs as a result of the investment. This means that the same rate of debt-to-equity is used in the calculation as the company presently has. Second, the new investment must earn the FERC approved returns on its own base. This rate is also based on the present company equity and debt structure. Third, all costs and credits are included in the same way as they would be in a formal rate calculation based on the company's entire financial structure. In particular, the rate base is defined as the original cost of the added equipment less average book depreciation and with some smaller corrections.

The "equation" of cost of service for RBC equipment is that the savings, either in fuel not burned or sale of electricity, must be balanced by the costs which the RBC must bear. Table 5-1 shows these elements. The key point is that if the savings of gas or revenue of electricity equals the cost of service, the same profit is earned for the company and the gas price to the customer does not increase.

A small computer code was written to perform this cost of service calculation. (see Table 5-1) In addition to the first year calculation, the future can be predicted, using inflation and escalation on key parameters such as fuel cost. Table 5-2 shows a ten-year projection for Columbia Gulf with introduction of equipment in 1987. The return required decreases after the first year since the investment is depreciated, and the rate base includes the original investment less depreciation. Operating and maintenance expenses increase with inflation. Property tax was assumed constant. State and Federal Income Taxes decrease with the lower return. The fuel savings is subtracted from the true cost of service, which is not the normal way of presenting this type of calculation. Hence the tabulation "cost of service" is actually the difference between the actual cost of service and the savings or revenue received. A positive value here means that the cost of service is greater than the savings while a negative value means that the savings are greater than the cost of service. Typically, the first year cost of service is the highest. This computer code also performs a present worth cost of service calculation, the bottom line showing the levelized annualized cost over years.

TABLE 5-1

MAJOR ELEMENTS IN COST OF SERVICE CALCULATION

COST OF SERVICE EQUALS SUM OF FOLLOWING ELEMENTS

ELEMENT	DESCRIPTION	COMMENTS
Return on Rate Base	Allowed ROI Times Rate Base	Rate Base is Original Cost of RBC Less Average Book Depreciation Plus Miscellaneous Small Corrections Per FERC rules.
Tax Due to Deferred Taxes	Small Cost Based on Deferred Taxes	Difference in Tax Depreciations and Financial Depreciation Leads to an Early Cash Flow Which can Earn Money for Company.
Operating and Maintenance Expenses	Includes Normal Cost for Operations, Repairs, Electricity, Major Parts Replacement, and Insurance	Based on Estimate for Individual Case.
Depreciation	Straight-Line for Book Purposes	Rate Negotiated with FERC.
Property Tax	10% of Installed Cost	
Federal Tax	Tax on Taxable Income Per Federal Tax Rules	48% Rate Used, as Well as Accelerated Depreciation
State Tax	State Income Tax	10% of Federal Tax

- For Rankine Bottoming Cycle, the cost-of-service is balanced by the fuel saving or revenue from sale of electricity.
- If the savings or revenue is less than the cost-of-service, the customer must pay more for gas.
- If the savings or revenue is greater than the cost-of-service, the customer may get a rate reduction.
- In all cases, the company earns its allowed rate of return.

TABLE 5-2

SAMPLE COST-OF-SERVICE CALCULATION

CØLUMBIA GULF - 1987 -AVG. FUEL PRICES -10% FUEL ESCALATIØN 01/20/79

CØST ØF SERVICE
(THØUSANDS ØF DØLLARS)

	1987	1988	1989	1990	1991	1992	1993	1994	1995	19
RETURN	635.	583.	534.	489.	447.	408.	371.	336.	303.	2
TAX DUE TØ DEF. TAX	0.	8.	14.	18.	22.	24.	24.	25.	24.	
Ø & M EXPENSES	237.	256.	276.	293.	322.	348.	376.	406.	438.	4
FUEL SAVINGS	1752.	1983.	2275.	2563.	2820.	3102.	3412.	3753.	4128.	45
DEPRECIATIØN	338.	338.	338.	338.	338.	338.	338.	338.	338.	3.
PRØPERTY TAX	0.	61.	61.	61.	61.	61.	61.	61.	61.	
FEDERAL INCØME TAX	379.	339.	302.	269.	239.	211.	185.	162.	140.	1
STATE INCØME TAX	38.	34.	30.	27.	24.	21.	19.	16.	14.	
TØTAL CØST ØF SERVICE	-125.	-365.	-719.	-1062.	-1367.	-1591.	-2038.	-2409.	-2810.	-32
PRESENT WØRTH FACTØR (.090)	1.0000	0.9174	0.8417	0.7722	0.7084	0.6499	0.5963	0.5470	0.5019	0.4
CUMULATIVE PWF	1.0000	1.9174	2.7591	3.5313	4.2397	4.8897	5.4859	6.0330	6.5348	6.9
PWF CØST ØF SERVICE	-125.	-335.	-605.	-820.	-968.	-1099.	-1215.	-1318.	-1410.	-14
CUM PWF CØS	-125.	-460.	-1066.	-1885.	-2854.	-3953.	-5168.	-6486.	-7896.	-93
ANNUALIZED CØS	-125.	-240.	-386.	-534.	-673.	-808.	-942.	-1075.	-1208.	-13

Assumptions: Escalated Capital Cost of Equipment in 1987 $6,139,870
Fuel Saved 406,000 MCF/year
Fuel Price $4.32/MCF in 1987

A number of pipeline companies observed that they would require an investment in this type of conservation equipment to have a savings or revenue equal or greater than the cost of service in the first year of operation. This investment "hurdle" is a result of not wishing to have their customers pay for the equipment, especially for first-of-a-kind units. In addition, since the cost of service decreases in succeeding years and the savings tend to increase, it is a safe course of action. This requirement is a very stringent condition. Other companies stated that they would not insist on this condition, but would look at the overall profitability over the life of the equipment. Since both types of calculations are made by this computer code, it serves for both types of evaluation.

This code is flexible, and contains provisions for tax credits, although under present FERC rules, the investment tax credit is not included in the calculations, but passes through to the company.

In summary, the incremental cost of service calculations were used for the balance of the economic assessment.

5.4 BOTTOMING CYCLE APPARATUS COST

The bottoming cycle, shown schematically in Figure 5-1, recovers energy from the waste heat of the power turbine in the vapor generator by heating the liquid toluene to superheated vapor. The vapor expands through a turbine,generating power to drive the load through a gearbox. The heat is rejected in the air-cooled condenser and the liquid is dumped into a surge tank. The boiler feed pump is supplied with the liquid toluene from the surge tank which is pumped to the vapor generator completing the cycle. The cost of the bottoming cycle equipment consists of the cost of the vapor generator, the turbine, the condenser, the feed pump, controls and instrumentation, auxiliary equipment and the equipment skid assembly. The equipment skid also contains the surge tank, the feed pump and any other auxiliary equipment.

The cost optimization of the bottoming cycle was done through the system design optimization described in Section 4. This optimization used component cost algorithms to enable the variations of cost with the en-

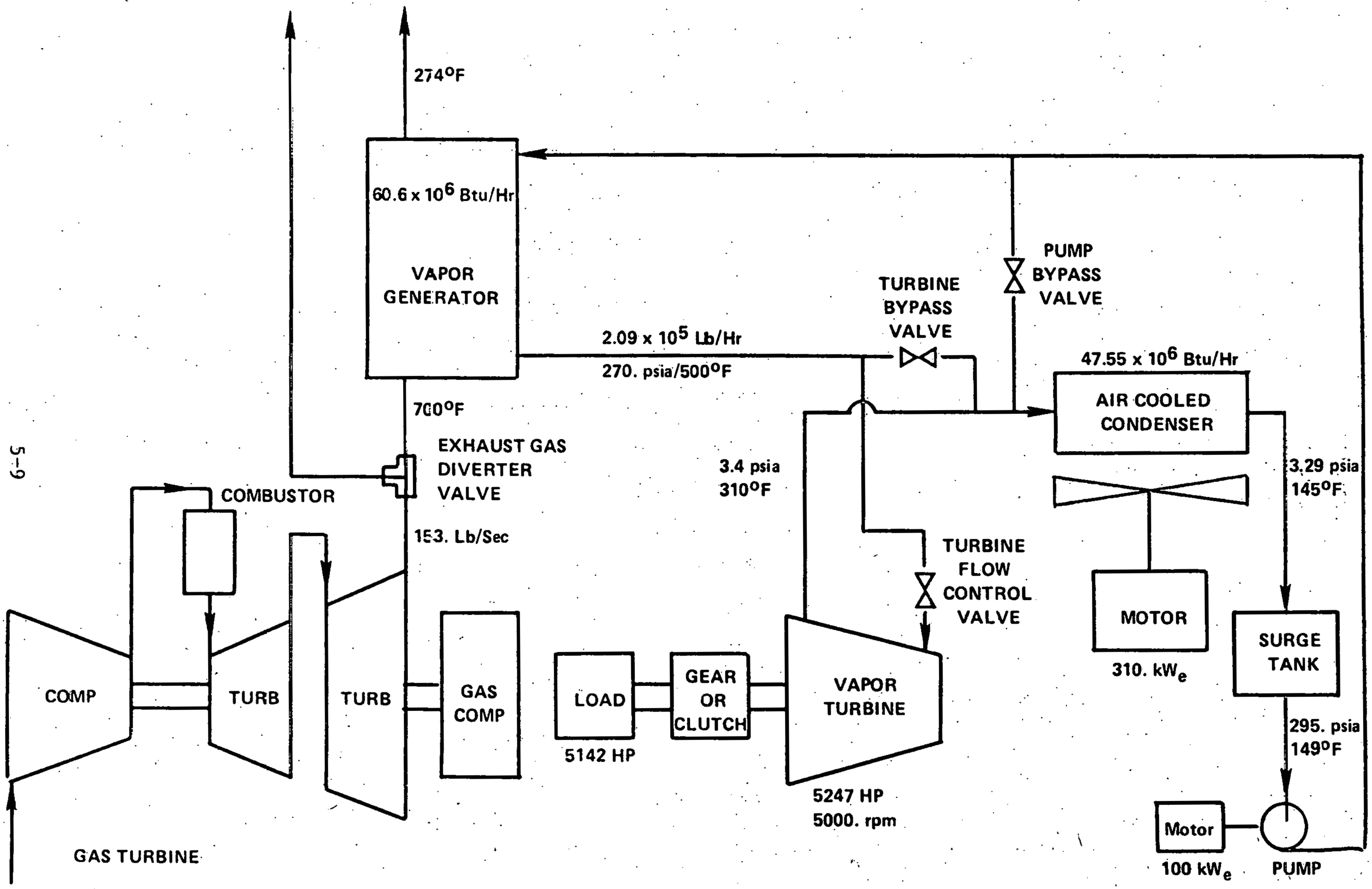


Figure 5-1. Optimized Bottoming Cycle Schematic.

gineering variables to be determined. The cost algorithms applied to mature components, that is, components after they are made available commercially.

After the site-specific designs were completed, the cost of each component was determined for each site. The component cost algorithms were also adjusted to reflect quotations received from vendors where available. Table 5-3 shows the cost of the bottoming cycle components for each site used in the economic analyses.

Subsequent to completion of these analyses, vendor component cost estimates became available for components which were lacking before. These estimates were scaled according to component size to provide a check on the values in Table 5-3. Since in Table 5-3 the component costs are meant to reflect the expected economic benefits of production manufacture, the lower of a group of estimates for a given component were used to approximate production-level costs. (In the demonstration program plan in Section 10 the higher of a group of estimates for a given component were used to approximate first-of-a-kind component costs.) The new estimates would increase the total component costs in Table 5-3 about 29%. As a result of this increase in component costs the installed costs would be increased by about 18%.

5.5 INSTALLATION COSTS

As discussed in Section 3, three sites were selected for the installation of the bottoming cycle, on three different pipeline companies. Since the installation costs vary from site to site and are major elements in the total cost of the system these estimates had to be reviewed very critically. A general cost methodology was developed to evaluate these costs realistically. Initially the bottoming cycle installation costs were prepared by the pipeline companies. These costs were added to the costs of the RBC components arrived at through cost algorithms to arrive at the figures for the overall costs for each site. At this point, the costs for the three site designs were reviewed for consistency. W.T. Wallace, consultant, was employed to prepare independent installation cost estimates for each site whose estimates were checked against pipeline companies costs to arrive at the final values used for the economic analysis. The installation costs are presented in Table 5-4. The specific reasons for the differences among these estimates are discussed below.

TABLE 5-3

BOTTOMING CYCLE EQUIPMENT COSTS

(In Thousands of Dollars)

No.	Component \ Site	Columbia Gulf Trans. Rayne, LA	PG&E Burney, CA	Texas Gas Trans. Corp. Covington, TN
1	Vapor-Generator	184	184	184
2	Condenser	402	388	433
3	Turbine	622	589	600
4	Auxiliary Equipment*	85	79	80
5	Feed Pump	18	17	17
6	Controls	25	25	25
7	Equipment Skid Assembly	55	52	53
8	Diverter Valve	70	70	70
	Total	1461	1404	1462

* Evacuation, Fill & Drain and Fire Protection Systems

TABLE 5-4

SITE SPECIFIC INSTALLATION COST ESTIMATES

(In Thousands of Dollars)

Item	Cost Element \ Site	Columbia Gulf Trans. Co. Rayne, LA	Pacific Gas & Electric Co.* Burney, CA	Texas Gas Trans. Corp. Covington, TN
I	Field Equipment			
	Compressor	320	700	-
	Induction Generator, Transformer Switchgear & Wiring	-	-	264
	Transmission Line	-	-	132
II	Installation Labor			
	Installation Engr.	70	90	100
	Site Construction	797	696	730
	Field Supv. of Contractor	50	65	85
	Startup and Checkout	20	20	20
III	Overhead			
	G&A Costs	61	506	112
	Interest During Constr.	145	155	151
	Contingency	146	182	152
	Total	1609	2414	1746

*Gas Pumping Option.

5.5.1 COLUMBIA GULF TRANSMISSION COMPANY INSTALLATION

In the Columbia Gulf installation the bottoming cycle was to drive a separate compressor on a separate pipeline from the one the prime mover to be bottomed was on. Interaction between the existing system and the RBC system was at a minimum. The low installation cost reflects this.

5.5.2 PACIFIC GAS AND ELECTRIC COMPANY INSTALLATION

The gas pumping system selected for the Burney site required the replacement of an existing compressor with a larger unit with shafts on both sides so that the RBC and prime movers could both drive the same compressor. This new compressor would cost $700,000, significantly higher than the cost of the separate compressor used at the Columbia Gulf site. In addition, PG&E accounting practices appeared to add significantly more burden to the overall plant, although it is probable that the added cost is simply caused by the more difficult installation,and accounted for in a different fashion.

PG&E also estimated the cost of the installation of an electrical generator for use with the RBC unit. This would cost $332,000 less than the gas pumping option, for a site installation cost of $2,082,000.

5.5.3 TEXAS GAS TRANSMISSION INSTALLATION

The Texas Gas Transmission Corp. installation was based upon the generation of electricity. The installation included the required generator as well as a 63,000-volt transmission line to the TVA grid.

5.6 TOTAL CAPITAL COSTS

Shown in Table 5-5 is a summary of the installed costs of bottoming cycles on the three selected sites. The "bottoming cycle" costs refer to mature product costs for the bottoming cycle components after they have become commercially available. It is seen that these costs range between $240-$260 per horsepower. The "field equipment" costs refer to the apparatus which makes up the load for the bottoming cycle. The "installation labor" cost refers to all labor costs required for the installation including engineering, construction, field supervision, startup and check out. The "burden" values refer to indirect costs for installation computed as the individual pipeline companies compute these costs. The "total" cost values

TABLE 5-5

COST BREAKDOWN OF BOTTOMING CYCLE INSTALLATION

Company/Site	Bottoming Cycle	Field Equipment	Installation Labor	Burden	Total
Columbia Gulf At Rayne, LA	$1,462,000 ($240/HP)	$320,000	$937,000	$352,000	$3,071,000
Pacific Gas & Electric At Burney, CA	$1,406,000 ($260/HP)	$700,000	$871,000	$843,000	$3,820,000
Texas Gas Transmission At Covington, TN	$1,463,000 ($261/HP)	$396,000	$935,000	$415,000	$3,209,000

do not contain any non-recurring costs in bringing the bottoming cycle to commercialization. The installed cost of the demonstration bottoming cycle on any site will have non-recurring costs as well as more expensive component costs.

5.7 ENERGY CONSUMPTION DURING MANUFACTURE AND INSTALLATION

The life cycle energy costs of the pipeline bottoming cycle system were estimated in order to determine what portion of the system's total energy output was required to build the system. The results of the analysis indicate that about 0.54% of the total energy output of the pipeline bottoming cycle, over its design life, is required to produce and install the hardware from the raw materials.

A detailed energy breakdown is shown for the vapor generator in Figure 5-2. Since nonmetallic components, with the exception of the concrete foundations, constitute a negligible fraction of the vapor generator weight, their energy requirements have been neglected. This is also true for the analysis of the system as a whole.

The energy used for mining of the raw materials, according to Reference 18, accounts for about 10, 54, and 2 percent of the energy required to produce raw steel, copper, and cement, respectively. The energy required to produce aluminum ore has not been taken into account since most of the aluminum in the U.S. that is made from ore is made from imported ore. For this analysis all the materials are assumed to have been derived directly from the raw materials. In reality, approximately 50% of all raw metal produced is produced from scrap and therefore requires less energy. This means that the values shown in Figure 5-2 for the mining and production of raw steel are probably high.

It was further assumed that 1.6 tons of iron ore were mined for every ton of steel produced. This is somewhat arbitrary since the amount of ore mined per ton of product is quite variable, depending on the quality of the ore.

Transportation estimates are also rather arbitrary since generally a number of different methods and routes could be used. Transportation energies are based on 0.2 kWh/ton-mile for railroad transport and 0.75 kWh/

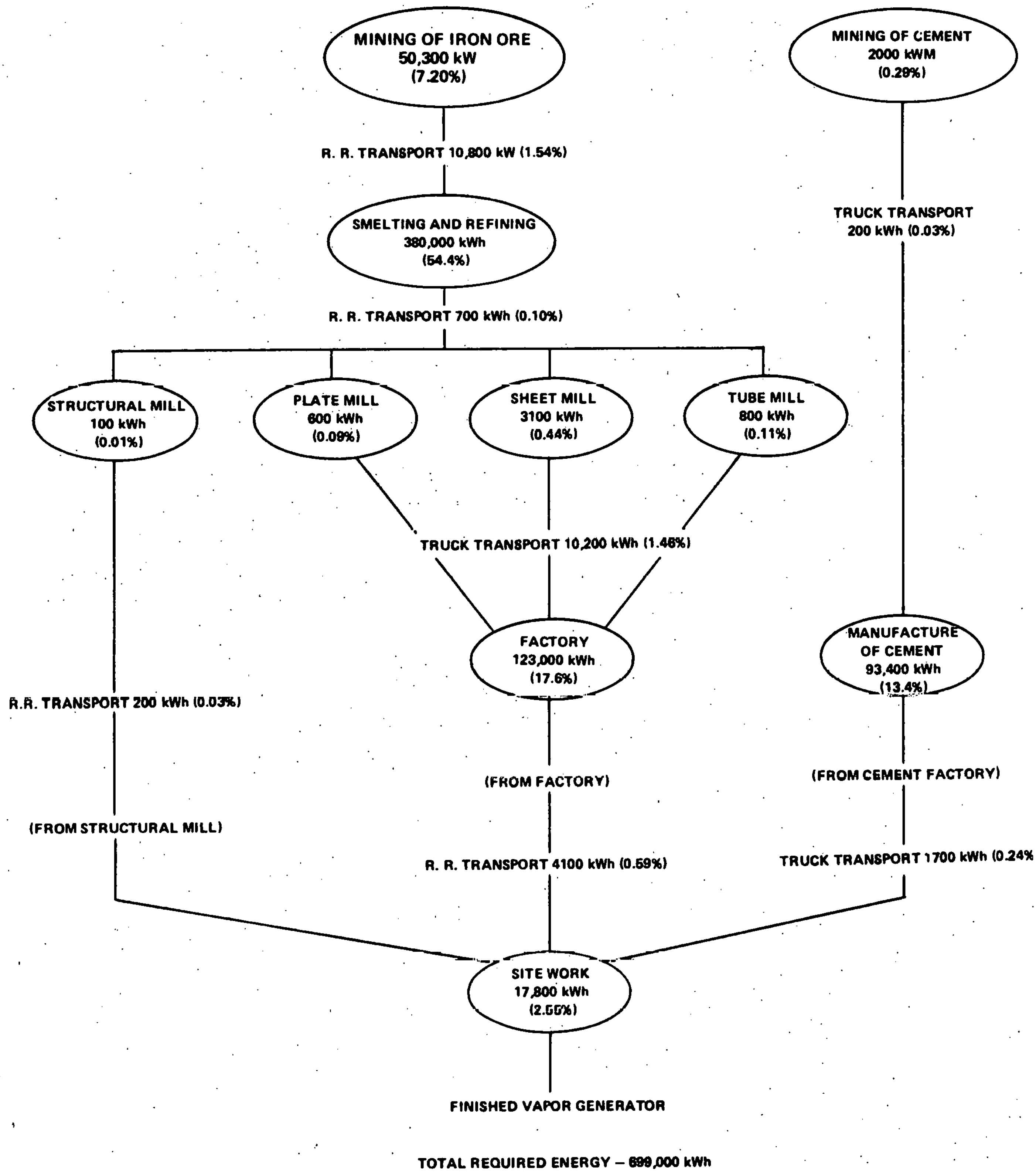


Figure 5-2. Vapor Generator Energy Consumption.

ton-mile for truck from Reference 19. Generally an allowance of 50 or 100 miles is made between manufacturing operations and a 500 mile allowance is made for the transport of iron ore.

Transportation constitutes only a relatively small portion of the total energy requirements, about 4%. Thus one could multiply the transportation energies several times and not seriously affect the results.

By far the largest portion of the energy requirement for the vapor generator and for the bottoming cycle as a whole is used in smelting and refining. Based on Reference 18, the energy requirements, except for aluminum, including mining for steel, aluminum, copper, and cement are 5830 kWh/ton, 51,500 kWh/ton, 32,800 kWh/ton, and 2200 kWh/ton, respectively. These numbers are averages that depend on the amount of scrap added, the particular process used, and the general quality of the equipment. There are also large discrepancies between references. The values used have an average accuracy of around 50%.

The estimates shown in Figure 5-2 for mill work were based on Reference 20, however, their accuracy is difficult to estimate.

The estimate for factory assembly and manufacturing in Figure 5-2 is based upon the assumption that the industry average consumption of energy is about 20 kWh/man-hour or about 0.667 kWh/($ of product).

Site installation energy requirements were determined by assuming that 200 HP would be continuously dissipated on the site while construction was in progress; the entire bottoming cycle apparatus would require about 14 weeks to install and the vapor generator would require about three weeks.

Table 5-6 presents the totals for the entire bottoming cycle apparatus. In all, 177 tons of steel, 19 tons of aluminum, 0.23 tons of copper, and 400 tons of concrete are assumed to be needed per unit. In Table 5-6, except for concrete, the mining energies are included in the energies for ingot production. The iron ore has been assumed to be transported 500 miles by rail. All energies were determined in the same manner as for the vapor generator.

At the Rayne site the bottoming cycle produces 5600 HP and has a

TABLE 5-6

TOTAL ENERGY REQUIRED FOR BOTTOMING CYCLE

Element	Energy Required, MWh			
	Steel	Aluminum	Copper	Concrete
1. Mining	(In 3)	--	--	18
2. Transport of Ore	28	--	--	2
3. Ingot Production	1,030	984	8	848
4. Mill Work	11	1	--	--
5. Transport to Factory	13	--	--	--
	1,082	985	8	868
Total to 5	2,943			
6. Component Manufacture	933			
7. Transport to Site	50			
8. Site Work	85			
Total Energy Requirement	4,010 MWh			

design life of 20 years. The results of the analysis are tabulated below:

Total Energy Requirement	4096 MWh
Site B.C. Turbine Power	4.18 MW
Energy Payback Period	40 Days
% of B.C. Lifetime	0.54

This is typical of the other sites under consideration. The results indicate that the errors in the analysis are of little importance because the total energy requirement is insignificant compared to the energy saved by the bottoming cycle throughout its life.

5.8 ECONOMIC WORTH AND ATTRACTIVENESS

5.8.1 ASSUMPTIONS

The economic attractiveness of retrofitting the Rankine bottoming cycle power plant equipment to the gas turbines located on the pipeline pumping stations is shown by analyzing the return on capital investment by the incremental cost of service approach. The incremental cost of service approach was suggested by the participating pipeline companies in the bottoming cycle study program as the most representative cost analysis approach for assessing the merits of new investments.

The main assumptions made in the economic analysis are that (1) there will be no change in the financial structure of the companies, (2) new equipment earns present return on its own rate base, (3) all costs and credits will be included as they would be in rate base calculations, (4) advantages of accelerated depreciation procedures can be taken, (5) the fuel cost escalations and changes in prices can be projected to the best of the present knowledge and are used in the present analysis and (6) the rate of return on the rate base can be held constant.

The pipeline companies provided the General Electric Company with the information required to make the economic assessment study. It included the values for the overall rate of return on rate base, the FERC allowed book depreciation rate, the double-declining balance tax depreciation rate, and the operating and maintenance expenses for the particular site in question.

The projected rate of return on the investment suggested by the pipeline companies is based on the average cost of capital. The companies have available to them a pool of capital funds, both debt and equity, for investment in various projects and each has a different value for the average cost of capital invested. The rate of return values that have been provided to General Electric Company are the weighted average values for the cost of capital invested in the pipeline.

In the cost of service analysis, depreciation is taken using a straight-line method which assumes a constant annual depreciation over the investment life. In the calculation of income taxes, accelerated depreciation methods are allowed and result in a lower tax in the earlier years of a project. In the analysis performed in this study, the double-declining balance method was used for tax purposes only.

Although investment tax credits are allowed for bottoming cycle equipment, the law does not permit them to be included in the cost of service calculations. At present, this credit passes directly to the company. If it were included in the cost of service, no benefit would accrue to the company, and hence the investment incentive purpose would be defeated. The cost of service would, however, be lower. This matter is discussed in Section 5.9.1 in more detail. Some state Public Utility Commissions do require its inclusion.

The federal income taxes are computed as 48% of the taxable income and state taxes are assumed to be 10% of the Federal Income taxes in the calculations.

5.8.2 PROJECTED FUEL COSTS

According to the most recent Federal law, natural gas is basically categorized as either "new" gas or the "old" gas. According to the details of the bill published in Sept. 28, 1978 Wall Street Journal, the new gas would include any new onshore gas produced 2.5 miles away from, or 1000 feet deeper than, an existing well or from a reservoir that has not produced in commercial quantities. According to the legislation, the ceiling price of new gas will rise to $1.99 a thousand cubic feet immediately after passage of the law from $1.50, the then current price. Through April 20, 1981, this ceiling will climb annually by the rate of inflation plus 3.7 percentage points. Thereafter, it will increase by the inflation rate plus 4.2 percentage points, until decontrol occurred. The price controls on the new gas will be removed January 1, 1985.

The legislation also included a complicated provision for incremental pricing designed to have large industrial users bear most of the burden of higher prices until controls are removed. The aim is to ease the impact on homes, schools and hospitals and also to force industrial users to convert to other forms of energy. It is in effect for new gas, special high-cost gas and liquified natural gas imports.

In addition, there have been many predictions, as in References 21 and 22, of the fuel costs depending on the various escalation rates for the inflation. The report on the "old" and "new" quantity projections of the natural gas by Arthur D. Little Inc. have been converted into the percentage projections and have been plotted vs. the years in Figure 5-3. Inflation rates of 6-10% have been assumed and the price of the natural gas for each category has been evaluated according to the present law. Then, the average price of the combined gas (old and new) for each year in question has been estimated on the weighted average of the new and old gas prices. The projected costs have been plotted as in Figure 5-4 and compare well with other estimates.

In the cost of service calculations of the pipeline bottoming cycle, both the average* and the new gas prices have been used to calculate the dollars saved resulting from the fuel savings. As can be seen later in the following sections, the fuel prices have a great impact on the return on the investments.

*Present FERC rules require the use of average gas prices.

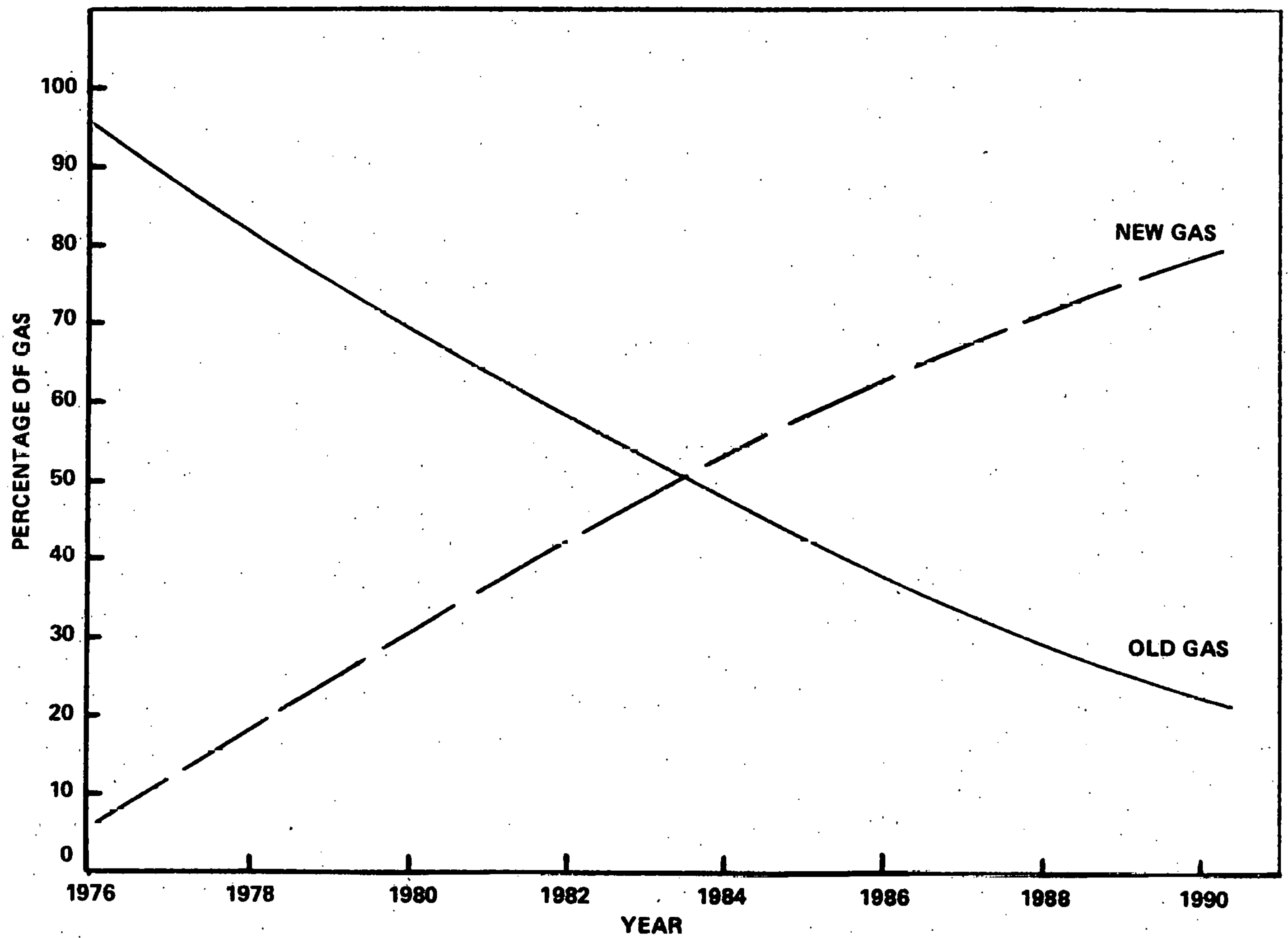
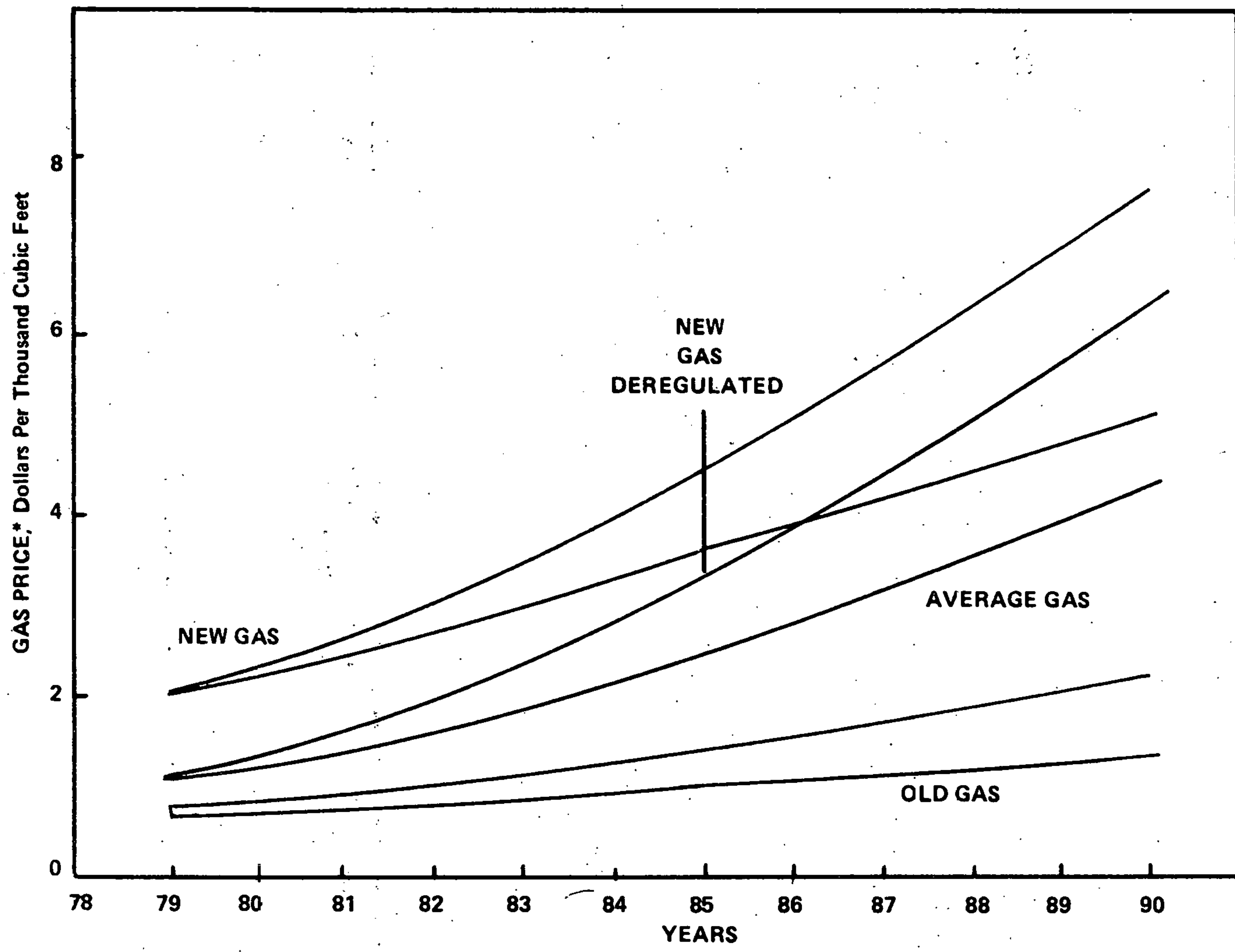


Figure 5-3. Percent New and Old Gas in Service vs Year



*Cost of Gas to Pipeline Company

Figure 5-4. Projection of Natural Gas Prices in the Years Ahead.

5.8.3 COLUMBIA GULF TRANSMISSION COMPANY - RAYNE, LA SITE

The site specifications have been presented earlier in Section 4. The bottoming cycle produces 5976 HP in shaft horsepower from the exhaust of the 12500 HP Rolls-Royce gas turbine. Some of the existing reciprocating engines will be shut off to maintain the constant pumping station power and the necessary power will be supplemented by the generated power from the Rankine bottoming cycle. It is assumed that all the bottoming cycle power can be used. The fuel savings over what would have been required by the shutdown reciprocating units are estimated to be 52.5×10^6 BTU/hr (50,700 ft^3/hr). The electrical costs at the estimated 1979 rates of 22 mills/kW-hr* for running the fans for the air-cooled condenser and the pump are estimated to be $94,700 per year in addition to the operating and maintenance costs of the compressor pumping station at $38,400 per year. The total capital costs for the installation of the new equipment is $3,071,470 as was shown in Section 5.6.

The Columbia Gulf Transmission Company is regulated by the FERC as an interstate pipeline. The investment criteria used by the company is the break-even first year cost of service. The break-even cost of service is the value of the worth of fuel savings when it is equal to the cost incurred by the pipeline company in providing the gas with the new investment in the additional equipment. As the fuel prices escalate, the fuel savings worth in dollars amount increases and the cost of service also escalates at the inflation rates, but at a lower rate than the fuel prices. It then makes sense to make calculations to look for the break-even first year cost of service for installations of the bottoming cycle equipment for various years starting 1979. It is also important to recognize that the calculations made very far into the future will have limited usefulness due to uncertainties involved. The basis for the break-even first year cost of service, though a very stringent criteria for an economic assessment of any project, is due to the fact that since profits are generated for the future automatically, there is no necessity to alarm the consumer with rate increase in inflated future costs.

*This represents an industrial rate, the actual commercial rate at Rayne is 29 mills/kW hr.

The capital costs of Table 5-5 and the O&M costs were escalated with the inflation rates of 4-8% and the discount rate of 9% was used in the present value calculations performed in 1978. The straight-line depreciation rate of 5.5% for the book depreciation and a life of 17.5 years for Double Declining balance tax depreciation were included in the calculations. The amortization factor of 3.7% was used over a debt period of 30 years to calculate the debt cost. Escalation factors of 6-10% were used on the fuel costs.

Figure 5-5 based upon the data of Table 5-5 shows the first year cost of service and savings due to the bottoming cycle by the incremental cost of service method for various years of installation. The figure indicates for the site at Rayne, LA and meets the investment criteria by early 1980's using "new" fuel prices. Also shown is the cost of service calculations using the average fuel prices based on the percentages of two categories of gas in service as required by present law. Average fuel prices are more nearly applicable to this site at present. As the old gas becomes depleted and the new gas comes into service, the investment becomes attractive in the mid 1980's as shown by the figure. If the installed costs of Table 5-5 are increased 18% the bottoming cycle, based upon average fuel prices becomes viable in 1988 for 10% escalation and 1991 for 6% escalation.

The cost of service for 10 years of operation annualized for 5 and 10 years are computed by the present value method and results are presented in Figure 5-6 for comparison with the first year cost of service shown in Figure 5-5. The eonomic gains due to the installation of bottoming cycles appear to be more attractive for the 5- and 10-year cost of service criteria. If the installed costs are increased 18% similar reductions in the viability date as shown by the comparisons of Figures 5-5 and 5-6 will apply.

The computer program was used to study the effect of change in any of the input parameters on the first year cost-of-service. Each parameter was changed by ± 10% of its value in the base case and the change in the first-year cost-of-service from its initial value was calculated. The results of the sensitivity analysis are shown in Figure 5-7 which indicates that cost of gas and the capital cost are two strong factors which influence the determination of the cost-of-service. Any improvement in bringing down the capital cost of the bottoming cycle equipment is going to bring down the cost-of-service, thus improving the economic gains further. The rise

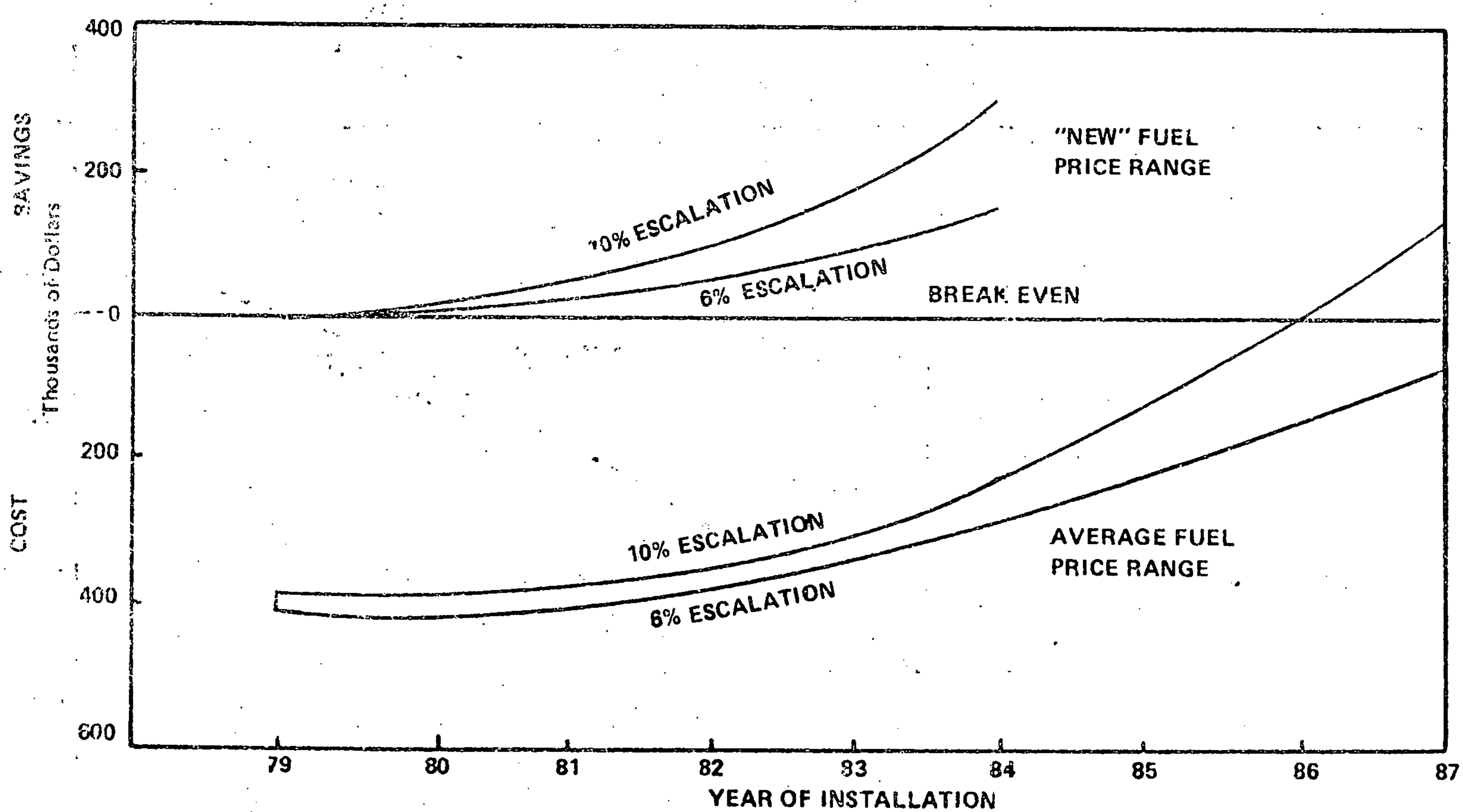


Figure 5-5. First Year Cost of Service for Rayne, LA Site on Columbia Gas Transmission Co.

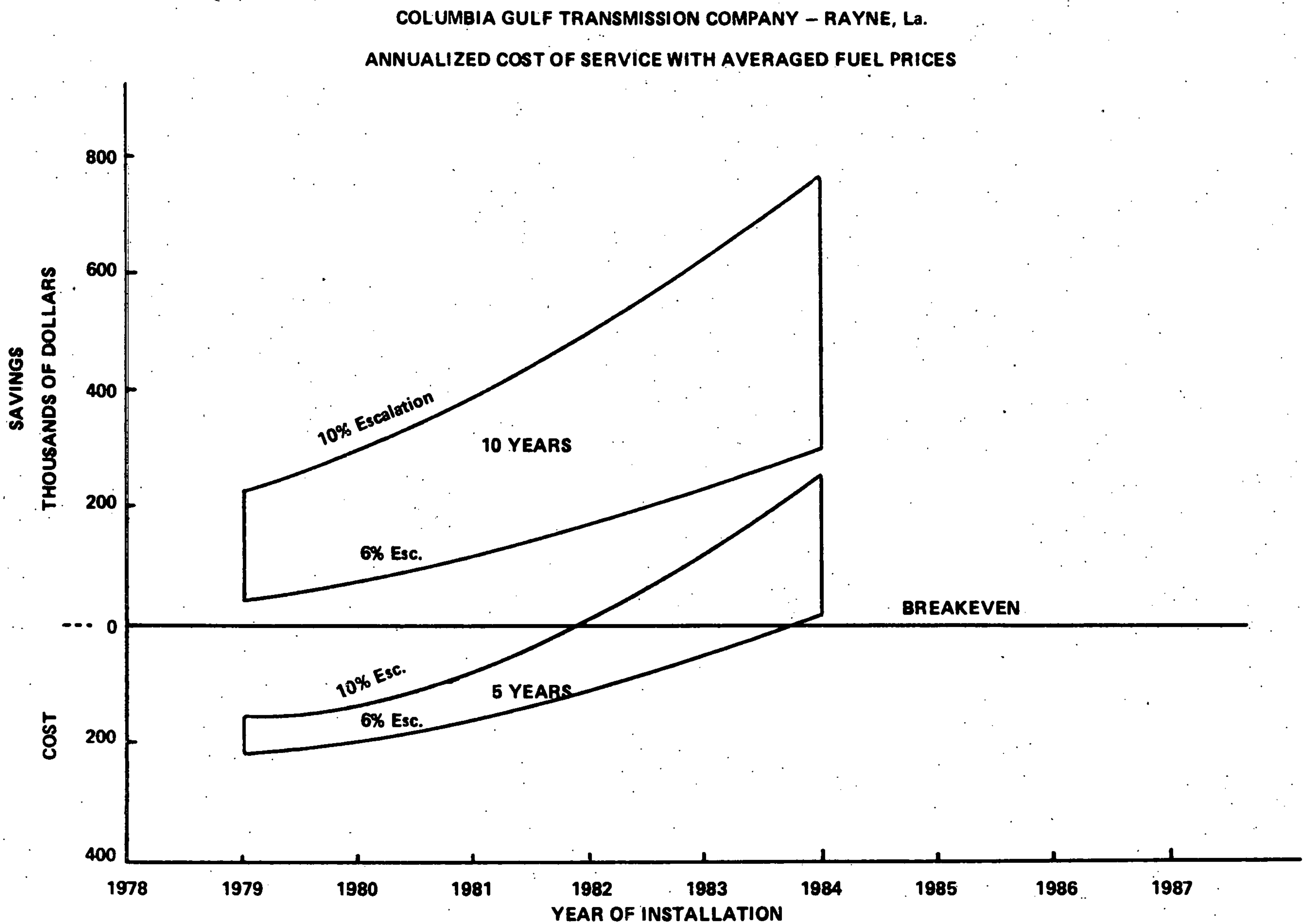


Figure 5-6. Five and Ten Year Costs of Service

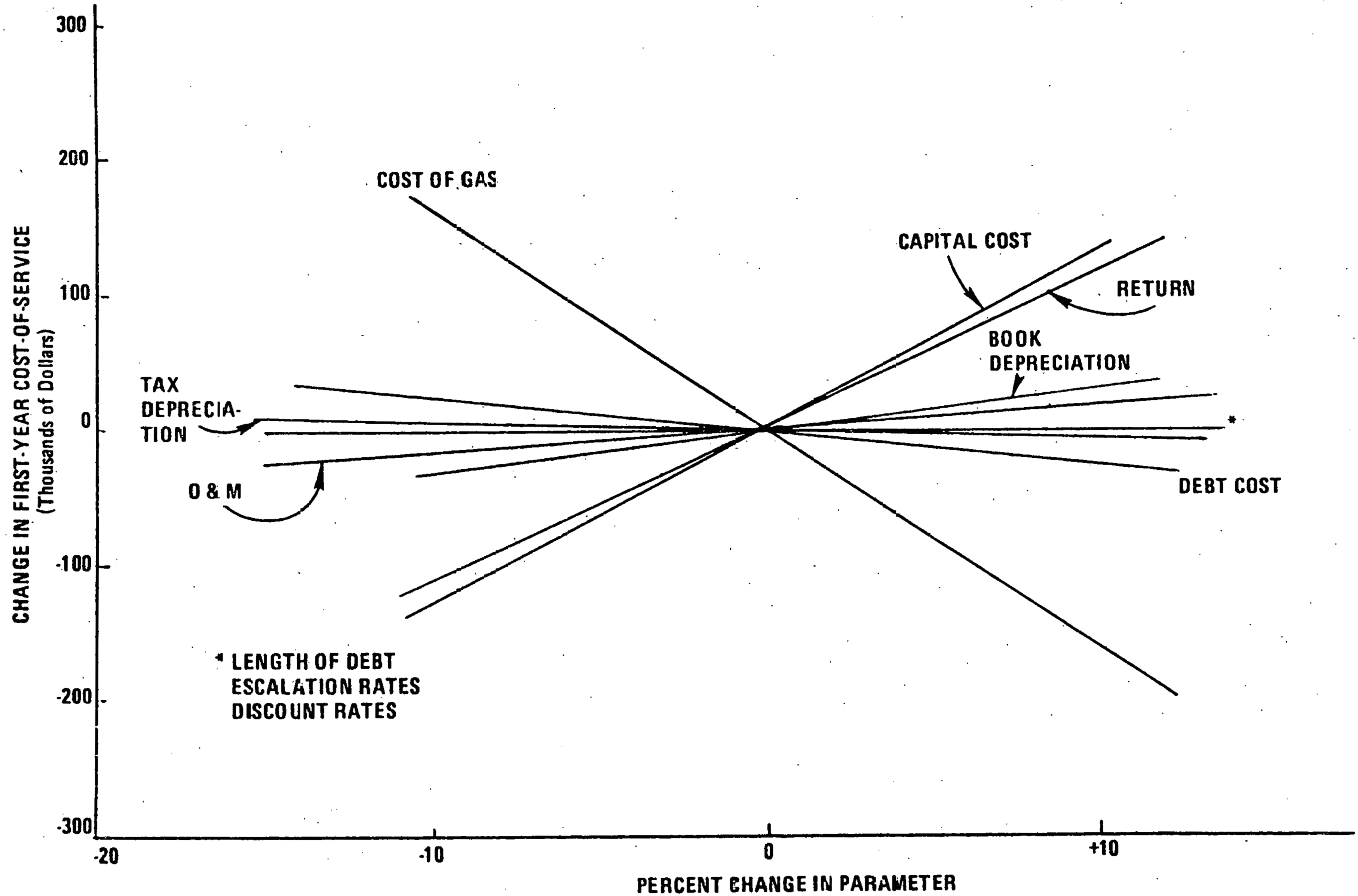


Figure 5-7. Sensitivity Analysis of the First Year Cost-of-Service for the Rayne, LA Site.

in fuel prices makes the bottoming cycle installation more attractive due to the increase in the worth of the fuel savings. Figure 5-7 was used in estimating the effect of an 18% increase in installed cost for the first year cost-of-service results.

5.8.4 TEXAS GAS TRANSMISSION CORPORATION - COVINGTON, TN SITE

The details of the site at Covington, TN were given in Section 4. There are a number of very efficient reciprocating engines at the site and they will have to be shutdown to maintain the station power if the existing nominal 12000 HP Pratt & Whitney engine is to be bottomed. The bottoming cycle gross electrical output is 3692 kW_e and requires 500 kW_e for the fans and the pumps.

The capital cost of the bottoming cycle equipment used in this analysis is $3,209,430. It needs 36,500 ft^3/hr extra fuel to pump the gas with the gas turbine and to generate electricity using the bottoming cycle. This extra fuel represents the difference between what the reciprocating engines would use to pump gas, and what the less efficient turbine requires when it is used. The present electricity cost is assumed to be 19 mills/kW-hr. The site requires $43,000/year[a] as operating and maintenance expense.

The first year cost of service calculations[b] were made assuming that the gas turbine would operate 8000 hours in a year. Figure 5-8 shows the results wherein cost of gas is plotted vs. value of generated electricity. For the operation to be profitable, the price of the natural gas has to be less than $1.45/1000 cu. ft. if the electricity is generated at 50 mills/kW-hr. As the fuel prices climb, the break-even would occur in the cost of service only if the generated electricity also increases proportionately. Although Figure 5-8 appears to show that the unfavorable result of this analysis is primarily due to a low ratio of electricity price to gas price, another important reason is that this site has efficient reciprocating compressors and does not need all the pumping power available. If all the

[a] This value assumes that no full-time manned operation would be required after successful commercialization.

[b] In commenting on this final report, Texas Gas Transmission Corp. stated that a longer period of time would be used for economic assessment. A change to a longer period would not materially affect the results reported here.

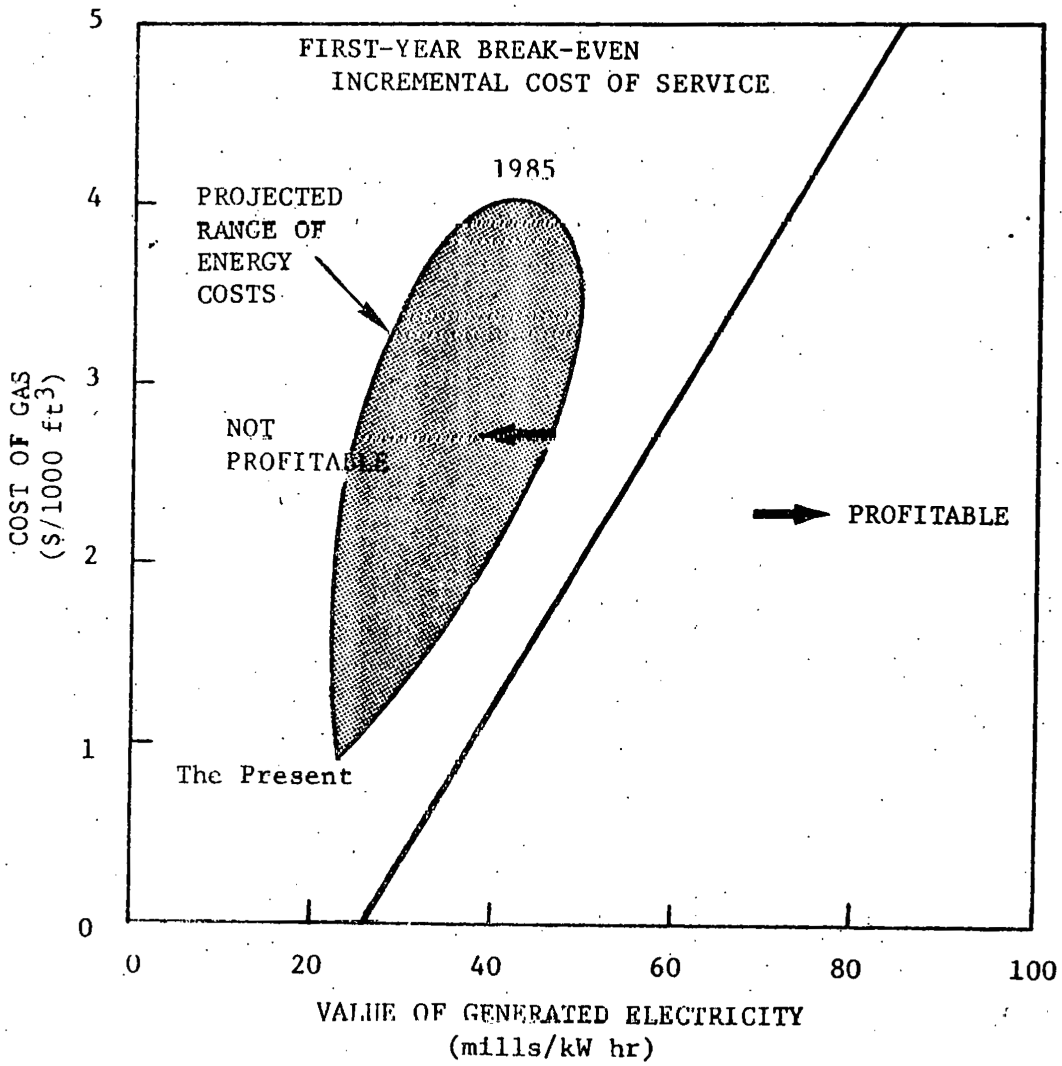


Figure 5-8. Results of First Year Cost of Service Calculations for Texas Gas Transmission Co. Site at Covington, TN.

power could be used to pump gas and the bottoming cycle power were used to generate electricity, the installation would be profitable, on a first year basis, with electric prices approximately 25 mills/kW hr, and lower when multiple years are included. Since 25 mills/kW hr is not unreasonable, the bottoming cycle would be marginally attractive under these conditions.

5.8.5 PACIFIC GAS & ELECTRIC COMPANY - BURNEY, CA SITE

The details of the Burney, CA site of the Pacific Gas & Electric Company were given earlier in Section 4. There are two gas turbines on this particular site and one of the 12500 HP General Electric gas turbines was selected for bottoming. The bottoming cycle generated 5397 horsepower from the waste heat of the gas turbine. The second gas turbine at the station will be shutdown with the installation of the bottoming cycle equipment to maintain the station pumping power. At the site the following economic inputs were estimated with the use of the bottoming cycle: fuel savings, 43 x 10^6 Btu/hr (41,600 ft^3/hr); electricity costs to meet parasitics, $57,000/yr; O&M expenses, $43,400/yr; and the installed bottoming cycle capital cost of $3,820,650.

The same procedure as described in earlier sections to compute the cost of service is adopted in arriving at the break-even point for the installation of the bottoming cycle equipment, including the use of the bottoming cycle installed costs from Table 5-5. The projections are made for both the "new" and "average" gas prices taking into consideration a range of 6-10% escalation in fuel prices. The analysis is done with and without the tax credits. The indications from Figure 5-9 are that, with the use of the new gas prices in the analysis and with the allowed 10% tax credit, the bottoming cycle looks economically viable right now. However, as the number of years over which the cost of service is calculated increases, as shown in Figure 5-10, the advantage of the tax credit becomes smaller. The economic gains, however, are still increasing due to increase in the fuel prices. The "new" fuel price range is more nearly applicable to the Burney, CA site at the present time. If the installed costs of the bottoming cycle in Table 5-5 are increased 18% the first year cost-of-service investment criteria would predict that the investment based upon the "new" fuel price range becomes viable in 1984 for 10% escalation and in 1985 for 6% escalation compared to being viable now as shown in Figure 5-9. If

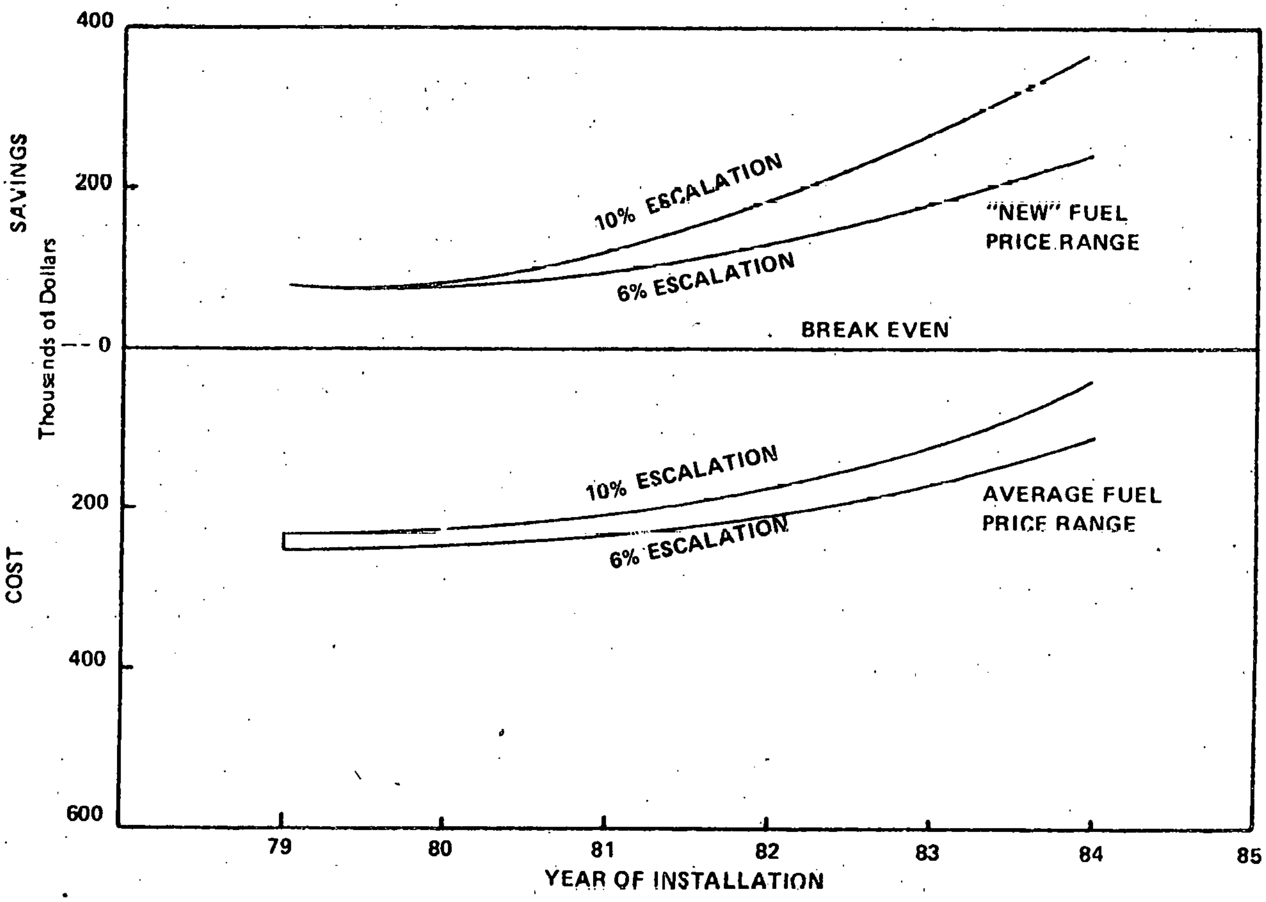


Figure 5-9. First Year Cost of Service Calculations for Burney, CA Site on Pacific Gas and Electric Co. (10% Tax Credit Included).

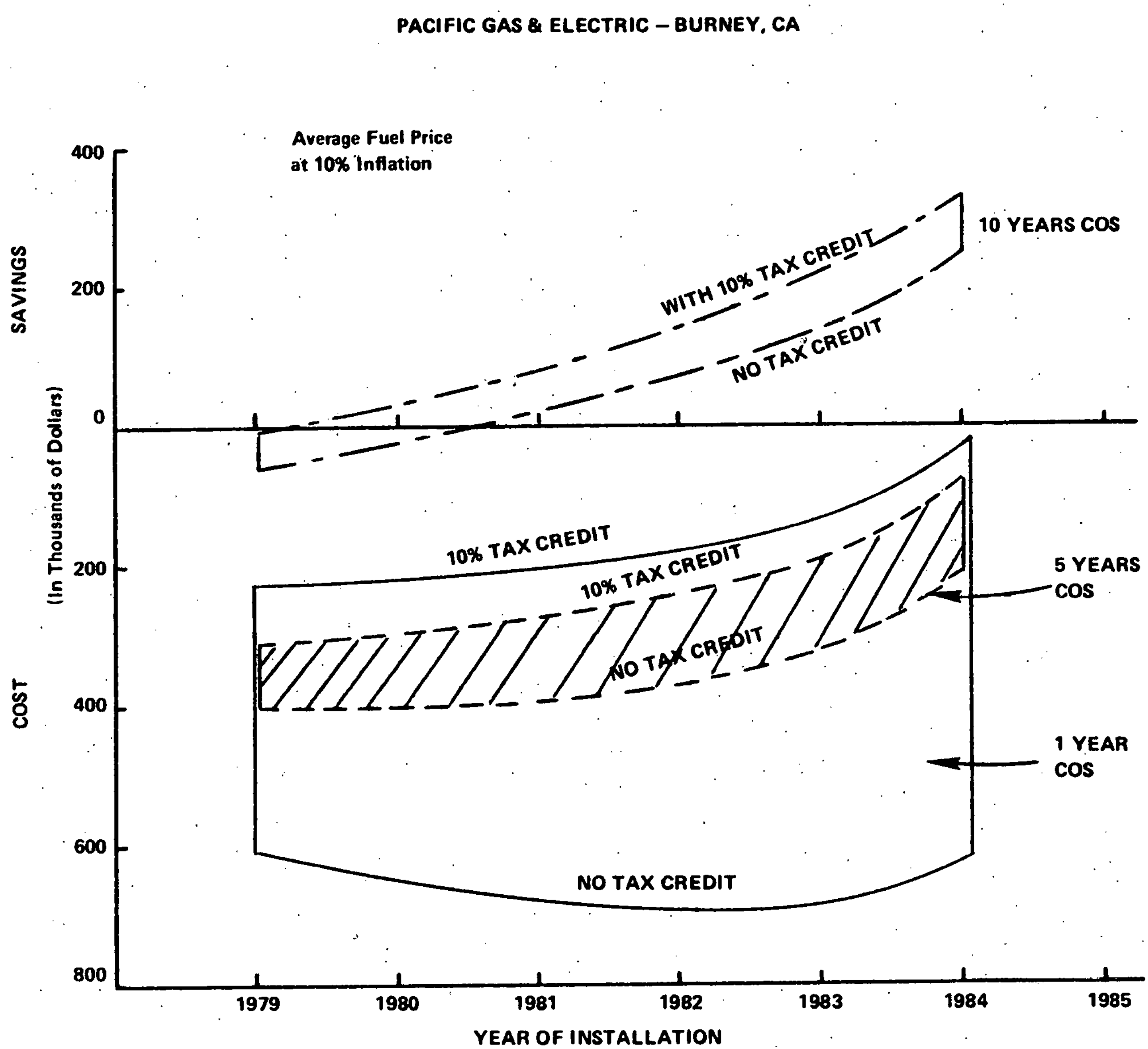


Figure 5-10. Multiple Years Cost of Service With and Without Tax Credit

longer cost-of-service periods were accepted as the criterion, as in Figure 5-10, the investment would be viable somewhat later for a five-year cost-of-service criterion and considerably earlier than stated above for a ten-year cost-of-service criteria. The Pacific Gas & Electric Company is controlled by the Public Utility Commission (PUC) in California. PUC requires that tax credits be included in the cost-of-service. The company has 15 year profitability as the investment criterion which is by no means as severe and stringent a criterion as the first-year break-even cost-of-service. Hence, the economic gains due to the installation of the Rankine bottoming cycle look attractive as shown in Figure 5-11 with the "average" fuel prices and 15 years profitability used. Even at the lower level of inflation, the bottoming cycle is profitable now based upon the 15 year profitability criteria. A check was made for the case of increasing the installed bottoming cycle costs 18%. It was concluded that for even the 6% inflation condition in Figure 5-11 substantial cost savings could be made in 1979. Thus even with an 18% increase in installed bottoming cycle costs the bottoming cycle is viable at the present time under the conditions of Figure 5-11.

Another case was considered for the Burney site which corresponds more closely with prevailing conditions. For this case the Burney site was assumed to be operated at 75% power, to have around-the-clock site attendance, to have available 45 mils/KWh electricity and to be pumping "new gas". It was found that this combination of inputs resulted in a somewhat better investment potential than that shown in Figure 5-11.

Figure 5-12 shows the sensitivity analysis as indicated for the Rayne, LA site (Fig. 5-7). The capital cost and price of gas are the parameters affecting the cost-of-service the most. The results shown in Figure 5-12 were used to determine the effect of an 18% increase in installed costs on the results shown in Figure 5-9.

The Pacific Gas and Electric Company has performed an economic analysis of the use of the bottoming cycle to generate electricity. The analysis was based on the economic life of 15 years beginning in 1980 as the first year of operation. The new electrical output was 25,000,000 kW hr/year which required the cost of electricity generation to be 28.5 mills/kW-hr in 1980 for break-even. The company expects revenues greater than 28.5 mills/kW-hr and hence, the use of this option is attractive economically.

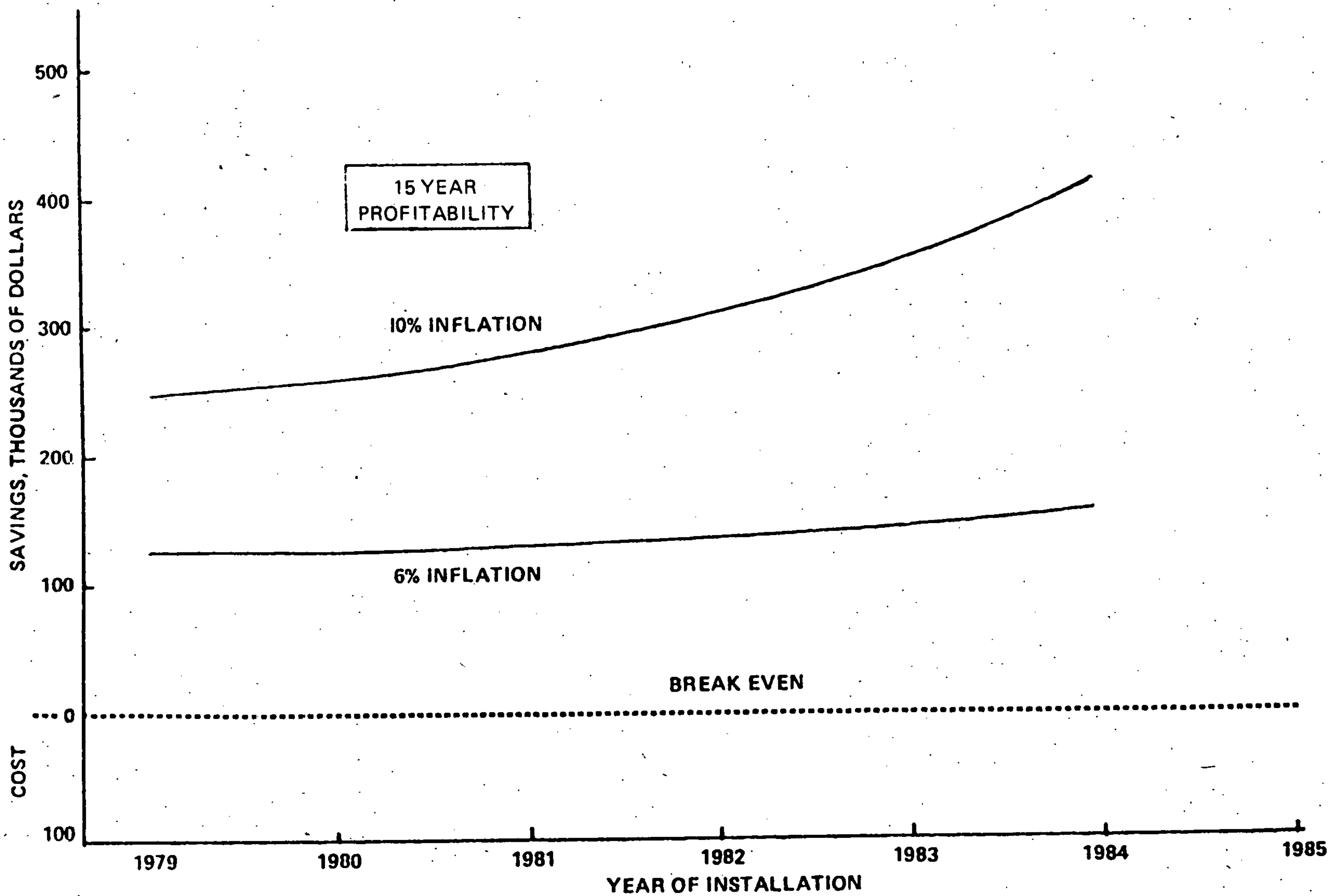


Figure 5-11. Life Cycle Costs for Bottoming Cycle Using Average Gas Prices.

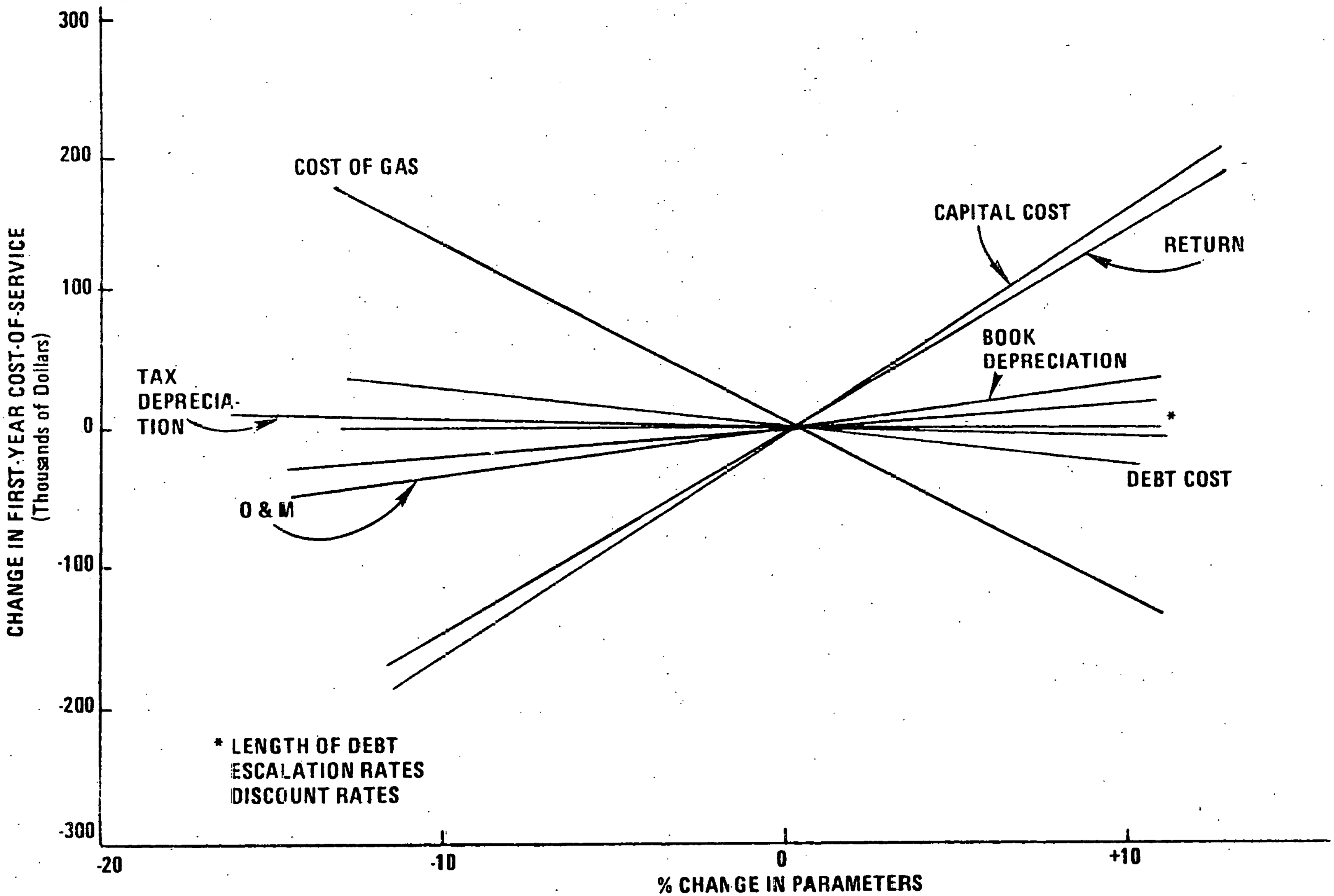


Figure 5-12. Sensitivity Analysis of the First Year Cost of Services for Burney, CA Site on Pacific Gas and Electric Co.

Further, the capital cost required for the generation of electricity is less than that required for pumping gas, primarily because of the $700,000 cost of replacing the present gas compressor shown in Table 5-5.

5.8.6 SUMMARY OF RESULTS

The results of this analysis show that some applications of the bottoming cycle to pipelines are economic now. In the near future, other applications become attractive. With changes in regulations the attractiveness increases. However, for some other applications, it is doubtful if the RBC will ever be attractive.

For the Rayne, LA site under Federal Energy Regulatory Commission rules, using "average" fuel prices, and including an 18% increase in installed costs, the bottoming cycle investment will be viable in the years 1988 and 1991, respectively, depending upon whether a 10% or a 6% rate of inflation is used. For a five-year cost-of-service criterion the investment will be viable in the mid 1980's and for a ten-year cost-of-service criterion in the early 1980's. For the Burney, CA site under California Public Utility Commission rules, using a 10% inflation rate, and including an 18% increase in installed cost the bottoming cycle investment with the allowed 10% tax credit is now viable. If FERC rules are changed to recognize that fuel saved is logically equivalent to "new" gas the use of the bottoming cycle will be economical two to three years earlier for FERC regulated companies.

The combination of FERC rules, a location in territory served by the Tennessee Valley Authority with inexpensive electricty, and surplus pumping power avilable makes the generation of electricity at Texas Gas Transmission's Covington, TN site unattractive. However, in California, under different conditions, electric generation is highly attractive. Obviously, the economics of electrical generation is highly site dependent.

5.9 EFFECT OF POTENTIAL INSTITUTIONAL CHANGES

In addition to being subject to the normal business and tax laws, whether a pipeline operates under Federal or State regulations also has a strong effect on the perceived desirability of new investments. The effects of changing some regulations which appear to impact the selection of the Rankine bottoming cycle are discussed below and in Reference 23.

5.9.1 TAX CREDIT POLICY

Under present tax law, pipeline companies receive a tax credit on new construction for gas transmission. Under the law, this credit can not be used to reduce the rate, but must either be deferred or flowed through to the owners. This set of rules was taken from Reference 24 and was "intended to ensure that the tax money saved would be used for capital investment rather than for the benefit of current customers, and to prevent loss of tax revenues that would result from reduced utility rates and reduced taxable income". From the standpoint of justifying the installation of RBC equipment, this means that tax credits do not impact the cost of service calculation. Although changing these rules would make the first year cost of service more favorable, it would be less favorable to the company, who would now have to reduce rates to the customer.

It is clear that if a RBC investment is not acceptable from a first-year cost of service without the tax credit being included, it is actually less acceptable to the company if, by its inclusion in the rate calculation, the first year cost of service becomes favorable. In summary, the company gets the tax credit in any case, but loses some of it if included in the rate calculation.

For intrastate utilities, regulated by a state Public Utility Commission, the situation is potentially different. Reference 24 describes a recent U.S. Supreme Court decision which allows a state PUC to force a company to include the tax credit in its rate calculation, while the IRS continues to state that, to receive the credit at all, the utilities must not pass the tax credit onto the customer. Obviously some change will occur here, but what will happen and when is unclear.

The only obvious change which might encourage the introduction of RBC equipment would be an additional tax credit for energy conservation devices. The credit could be split between the company and the customer so that an incentive would exist. The potential benefit to the country would have to be balanced against the loss of tax revenue.

5.9.2 FUEL VALUE POLICY

Under present regulations, the cost of gas used for pumping is the average price. This includes gas from older contracts as well as gas from new wells or from liquefied natural gas. When an economic assessment is made, this low price tends to make it difficult to justify the large capital investments required for RBC systems used for saving fuel.

Logically, gas saved through conservation methods should be valued as the most expensive gas purchased. If this change in the rules were implemented, the incentive to use all sorts of energy conserving equipment, including the RBC, would be much greater. Note that electrical generation is not a gas saving procedure, but more akin to a cogeneration process, and thus does not necessarily benefit from increased gas prices.

5.9.3 OTHER REGULATORY POLICIES

There are a number of other regulatory policies which need a more thorough examination. These include the effect of allowing faster tax depreciation (a form of tax credit) and possibly some sort of low cost loan or guarantee from the government for conservation equipment. Most of these schemes have already been examined by DOE for other programs such as cogeneration (Reference 25).

5.9.4 COMPETING INVESTMENTS

In discussions with pipeline companies, it became clear that one problem in gaining acceptance of the RBC system will be competition from other potential investments. The pipeline companies do not have unlimited resources. At the moment, their major concern appears to be acquiring new sources of gas. As a "new" source of gas, the RBC has a modest potential, compared to the opening of a new gas field, the acquisition of liquid natural gas (LNG) from overseas, or getting gas from Mexico or Canada.

This concern of the pipeline companies means that the profitability of the RBC is a necessary but not sufficient condition for making this type of investment. Unfortunately, this is a difficult problem to quantify. Because of a chronic balance-of-payments deficit, it is in the national interest to encourage conservation of domestic gas rather than import foreign gas or LNG. This is not necessarily perceived as being the best option by pipeline companies. Thus, a competition may exist for funds between an investment in conservation equipment and an investment which, dollar for dollar, will produce more gas for sale.

5.9.5 RECOMMENDED CHANGES

Since one of the overall recommendations of this study is that a further study be made of the impact of government regulatory effects on potential market commercialization, the discussion here will be limited and tentative.

First, the application of investment tax credits for energy conservation devices should be modified, with the objective of using this tool to encourage investment in conservation. A balance between increased rates to the customer, tax loss to the government, and incentive to the company must be achieved, while stimulating energy conservation.

Second, the use of a first-year cost-of-service criteria should be reexamined. A possibly less restrictive view of rate calculations for conservation equipment should be encouraged, say a three to five year period, rather than a single year*.

Third, the fuel price used for conservation equipment should be raised to a higher value than the present "average" value.

Last, there is a need for installing unneeded horsepower at a compressor station for fuel conservation purposes. This is not currently allowable under FERC regulations.

*Comments from participating pipeline companies indicate that they have already relaxed this requirement.

From an economic viewpoint, commercialization requires that the pipeline companies perceive that the bottoming cycle is an attractive investment which can compete with other investments for their limited resources. Similarly, the manufacturer of bottoming cycle equipment must perceive that a market exists before he will invest in the necessary development and production facilities.

In the regulated environment in which most pipeline companies operate, it appears that present government regulations, in some instances, deter investment. It is recommended that a study of the impact of government regulatory effects on the potential market commercialization be made.

Since the capital cost is one of the most significant factors in cost effectiveness, a study of potential cost reductions should be made.

5.10 CONCLUSIONS

Figure 5-13 summarizes the conclusions of this assessment in pictorial fashion. These conclusions are discussed below, and validated by the work described in the remainder of this report.

- Differences in application are crucial to market penetration.
  - -- In this study, three different sites were used, and the additional power generated by the bottoming cycle was used either to pump gas or to generate electricity. It was found that there was a strong site and energy use dependence on profitability.

- Economics are highly sensitive to application-dependent system costs and the worth of fuel savings.
  - -- For the retrofit applications studied, site-specific installation costs, which are application dependent, showed significant impact on economics.
  - -- Depending on the mode of station operation and other factors, e.g., cost of fuel, the amount and worth of fuel savings were highly variable.

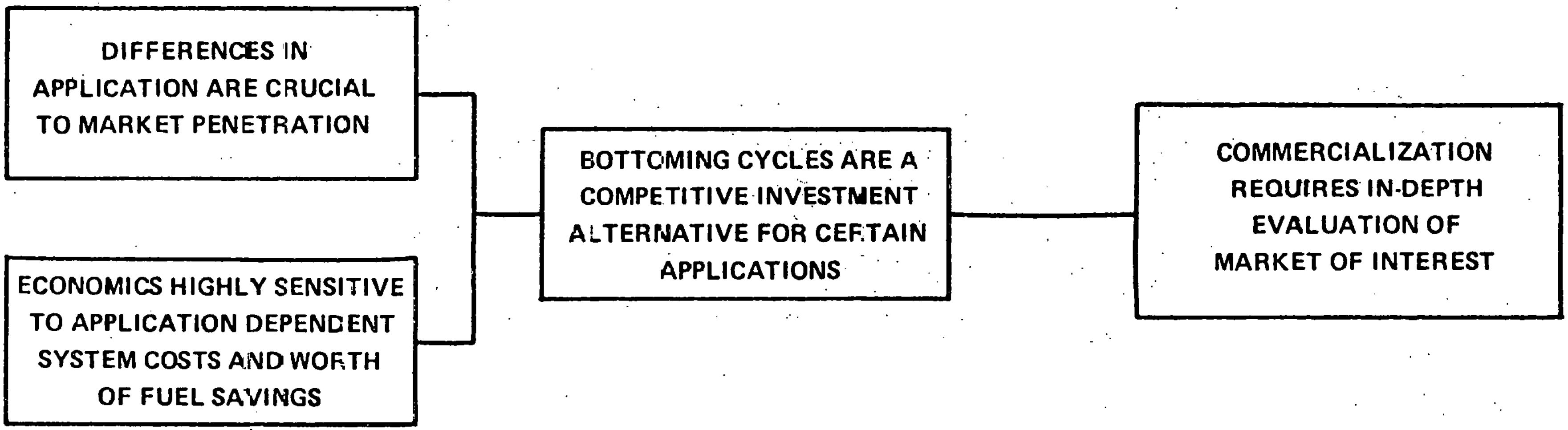


Figure 5-13. Summary of Economic Assessment.

- The economics of all applications studied were highly dependent on the projected cost of natural gas, as well as the rules used to allocate gas costs to pumping use.

- Bottoming cycles are a competitive investment alternative for certain applications.
  - Some of the applications are economically viable now, or in the very near future.
  - With some changes in regulations, more applications will be economic.

- Commercialization requires an in-depth evaluation of the market of interest.
  - The future size of the market requires an evaluation of those potential sites which can be economic.

For each of the three sites studied, different conclusions concerning the economic worth of the bottoming cycle were reached.

5.10.1 COLUMBIA GULF TRANSMISSION CO., RAYNE, LA SITE

With fuel valued at average gas prices and a ten-year cost-of-service investment criterion, the Rayne Site will be economic in the early 1980's if available gas supplies permit all the Rankine cycle power to be used for pumping. With escalating gas prices, this site and application will be even more attractive in the future.

5.10.2 PACIFIC GAS AND ELECTRIC CO., BURNEY, CA SITE

Either pumping gas or generating electricity appears economically attractive at this site. The combination of State PUC regulations, relatively high gas prices, and a relatively long time period for recovery of capital costs all make this site favorable.

5.10.3 TEXAS GAS TRANSMISSION CORP., COVINGTON, TN SITE

The use of the Rankine bottoming cycle power to generate electricity at this site did not appear favorable. The combination of low station throughput, highly efficient reciprocating compressors on this site, and a location in a relatively low-cost electricity area all make this option unfavorable. Unless all the compressors, both turbine and reciprocating, on this site are required for pumping, this does not appear to be a good application for the bottoming cycle.

SECTION 6

ENVIRONMENTAL AND SAFETY ASSESSMENT

6.1 INTRODUCTION

The reason for an environmental and safety assessment is to identify potential changes to the environment that may be brought about due to a specific activity and determine the impact these changes will have on the environment and, subsequently, mankind. The specific activity of interest is the application of an organic Rankine bottoming cycle to a gas pipeline compressor station to conserve energy. The bottoming cycle system is a closed cycle which takes heat from a waste source, converts it to shaft power, and rejects the remainder of the heat to the atmosphere. It has negligible effect on the environment when operating properly. The cycle can be operated utilizing a variety of working fluids, the selection of which is determined from extensive studies based on toxicity, safety, materials compatibility, performance, thermal stability and economics. The optimum fluid would, of course, have all of the above characteristics in the desirable ranges. The choice of toluene as the bottoming cycle system working fluid represents a compromise among the desirable characteristics since it is a flammable hydrocarbon and is considered to be somewhat toxic. Even though the bottoming cycle system is a closed system, it must be assumed that a component or piping failure could release some of the working fluid to the atmosphere. The extent of the failure, methods employed for failure detection, emergency shutdown procedures, etc., would all have an effect on the amount of working fluid released to the atmosphere. The basic question, then, is whether the use of a flammable, somewhat toxic hydrocarbon fluid such as toluene in a closed bottoming cycle system properly designed and operated, constitutes a risk to the environment and mankind which is small in comparison with the benefit. This assessment is an attempt to determine whether the system design is capable of reducing all undesirable characteristics to tolerable levels.

In this section a preliminary analysis of the environmental and safety impacts of a bottoming cycle system are presented. The bottoming cycle components and system design is reviewed for compatibility with Federal, State and Local Codes governing the selected demonstration sites; regulations and codes are examined for possible conflicts and restrictions; potential design and operational problems are discussed; and, finally, conclusions and recommendations are indicated.

6.2 ENVIRONMENTAL AND SAFETY IMPACTS

Under normal operating conditions the bottoming cycle system discharges no contaminants to the environment. The system does, however, add a potentially greater risk to the environment and operating personnel due primarily to the use of a hazardous, somewhat toxic fluid, toluene, as the working fluid. The use of toluene as the working fluid results in a generally negative response to the acceptance of the system from the standpoint of safety. Toluene is a flammable hydrocarbon with a closed cup flash point of 40°F and autoignition temperature of 997°F (Reference 26). It is somewhat toxic and the proposed OSHA occupational standard is 100 ppm, time-weighted average for an 8-hour day and a 40-hour week. Maximum acceptable concentration is 200 ppm. It is one of the more common chemicals and is produced in quantities of nine billion pounds per year. Large numbers of workers are exposed to it daily without serious problems being reported. A comparison of Toluene with natural gas is shown in Table 6-1.

Both hydrocarbon fluids can be classified as hazardous. The explosive limits in air for toluene vapors are slightly less than natural gas. Autoignition temperatures are essentially the same. Since the vapor density of toluene is greater than air it will tend to lay at ground level. A 32 kW_e Toluene Rankine system is operating in spite of this characteristic of Toluene at the Sandia Laboratories as part of a Solar Total Energy System Test Facility. Natural gas, being lighter than air, rises and disperses into the atmosphere. As indicated in the comparison table, fire extinguishing methods will be completely new to pumping station personnel.

According to Reference 27, a gas turbine modified to incorporate a bottoming cycle will not be considered to be modified or reconstructed

TABLE 6-1

COMPARISON OF CHARACTERISTICS OF TOLUENE WITH NATURAL GAS*

Fluid	Toluene	Natural Gas
Synonym	Methylbenzene	Methane, Marsh Gas
Formula	$C_6H_5CH_3$	CH_4
Flash Point, Closed Cup	40°F	Gas
Open Cup	45°F	Gas
Explosive Limits in Air - Lower	1.3% vol.	5.3% vol.
- Upper	7.0% vol.	13.9% vol.
Autoignition Temperature	997°F	999°F
Specific Gravity (H_2O = 1.)	.872	---
Vapor Density (Air = 1.0)	3.14	0.554
Melting Point	-139°F	-294°F
Boiling Point	231°F	-259°F
Water Solubility	Insoluble	0.05g/ml
Suitable Extinguishing Agents	Foam (or dry Chemical)	Shut Off
Hazard	B, D, G, H	D, G, H

CODE

B - Hazardous in contact with oxidizing material

D - Hazardous when heated

G - Explosive or highly combustible

H - May explode in fire

* Reference 26

for the purpose of establishing whether it is a "new" emission source since the addition of the bottoming cycle does not increase the emissions. Therefore, such a modified gas turbine will not come under the Standards of Performance for New Stationary Sources - Gas Turbines.

6.3 SYSTEM COMPATIBILITY WITH REGULATIONS AND CODES

Table 6-2 indicates the bottoming cycle major components and Codes and Federal Regulations which are applicable to the design and operation of the components and the system in general. To ascertain if any particular problems might exist with state or local codes which might affect the three selected demonstration sites, surveys were conducted by each of the partner pipeline companies and the results are as follows for each pipeline company:

6.3.1 TEXAS GAS TRANSMISSION CORP. - COVINGTON, TN SITE

- Tennessee Department of Economic and Community Development may be consulted regarding existing codes and regulations as they apply to system design
- Texas Gas Transmission Corp. - Environmental codes and procedures section, sees no problems, no additional emissions
- State regulations would not require licensed boiler operator. Union might insist
- Insuring Agency-Hardford Steam Boiler knows of no county or local code that would take precedence over state regulations
- State regulations will require annual vaporizer inspection

6.3.2 COLUMBIA GULF TRANSMISSION CO. - RAYNE, LA SITE

- Compliance with "Louisiana Air Control Law"
- State and Local laws extremely flexible
- Nature of the law makes possible problems difficult to foresee
- Columbia Gulf Transmission Co. does not favor toluene working fluid

TABLE 6-2

SYSTEM COMPATIBILITY WITH EXISTING CODES AND FEDERAL REGULATIONS

APPLICABLE REGULATIONS AND CODES

- VAPOR GENERATOR - Designed, constructed and tested per ASME Boiler and Pressure Vessel Code, Section I
- CONDENSER - Designed, constructed and tested per ASME Boiler and Pressure Vessel Code, Section VIII
- SURGE/STORAGE TANK - Designed, constructed and tested per ASME Boiler and Pressure Vessel Code, Section VIII
- PIPING - Designed, fabrication, erection and inspection per ASME Power Piping Code, ANSI B31.1
- ELECTRICAL EQUIPMENT - Designed per National Electric Code - Class I, Division I
- FLUID HANDLING SYSTEMS - National Fire Protection Association Flammable and Combustible Liquid Code
- SAFETY SYSTEM & GENERAL OPERATION - Occupational Safety and Health Administration Regulations; Environmental Protection Agency Regulations

- Suggest further investigation of alternate fluids
- Full time licensed boiler operator may be required
- Annual inspection of vaporizer by state required
- Update Air Control Commission Permit for exhaust emissions at the Rayne Station

If a licensed boiler operator is mandated for a manned site, operational personnel would be trained to qualify. For unmanned sites if a variance could not be obtained this requirement would deter implementation of bottoming cycle systems.

6.3.3 PACIFIC GAS & ELECTRIC CO. - BURNEY, CA SITE

- Certificate of Convenience and Necessity (Ref. CAL PUC 1001)
- County may require environmental impact report (EIR) for 60 KV transmission line
- County building and grading permits
- None of above present major problems to PG&E

No regulations or codes were discovered which would place impractical or unbearable restrictions on the design, construction or operation of a bottoming cycle at any of the chosen demonstration sites. As can be seen from Section 6.3.2 Columbia Gulf Transmission Company is definitely negative with regard to the use of toluene as the bottoming cycle working fluid. This negative attitude is predicated primarily on the fear that regulatory agencies might remove the product from the market, or impose exposure limits so low that they cannot economically be met.

6.4 DESIGN AND OPERATIONAL PROBLEMS

No major design or operational problems are foreseen; however, extensive safety protection requirements for a toluene organic Rankine cycle will be necessary. The insuring agencies covering the pipeline company station facilities will have a major influence on the design, construction and operation of the bottoming cycle system. It is quite possible they will establish the system protection requirements and these, of course, can vary between sites and insurance agencies. Any uncertainty in safety aspects generally results in higher costs for the initial insuring of the facility.

Table 6-3 outlines the minimum protection requirements of a particular insurance company anticipated for a toluene organic Rankine cycle system.

Air pollution control legislation, such as the "Louisiana Air Control Law," defines specific methods for the storage of volatile organic compounds, including the requirements for tank venting and vapor recovery systems. This is, of course, to keep the vapor released to the atmosphere to a minimum. Seals used on rotating machinery, etc., are subject to the approval of the Technical Secretary of the Louisiana Air Control Commission. Abnormal vapor releases to the atmosphere (vapor generator safety relief valve trip, seal failures, etc.) must be reported to the Commission immediately and a written report forwarded to the Commission within seven days of the occurrence. Repeated abnormal releases (failures, etc.) of vapor to the atmosphere can result in a decision by the Commission to shut down the offending facility.

6.5 CONCLUSIONS

Design and operation of an organic Rankine bottoming cycle system for application to gas turbine powered gas compressors must be in accordance with various Federal and State regulations and codes. Safety system requirements will be dictated, to a large extent, by the pipeline company insuring agency and will be largely dependent upon the system working fluid properties. Normal operation of the bottoming cycle system will result in no pollutants being discharged to the atmosphere and, therefore, have no adverse effects on the environment. The bottoming cycle will have no adverse effects on the present emissions from the gas turbine supplying the primary power.

During abnormal operation or in the event of a component failure hydrocarbon vapors can be released to the atmosphere. In such circumstances the hazards associated with the toxicity and flammability of toluene must be measured against the safety systems meant to neutralize them. In spite of the safety systems proposed there is some opposition to the use of toluene in the pipeline industry. Because the hazards exist National Institution for Occupational Safety and Health (NIOSH), Environmental Protection Agency (EPA), or some other governmental agency, may at some time preclude to

TABLE 6-3

TYPICAL MINIMUM PROTECTION REQUIREMENTS FOR TOLUENE ORGANIC RANKINE CYCLE

- Locate vapor generator 50 feet from all other structures
- Provide emergency relief venting for vapor generator and condenser
- Install fixed water spray to protect vapor generator and supports from ground up to 10 feet above toluene
    - Automatic activation
    - Manual activation
- Install CO_2 extinguishing system protecting core of vapor generator
- Install electric wiring and apparatus for use in Class I, Division 2 location
    - Inside or on all bottoming cycle apparatus
    - Within 5 feet of vapor generator
    - Within 25 feet horizontally from all bottoming cycle apparatus from grade level to 3 feet
- Provide interlock to shutoff heat source for loss of power to condenser fans
- Provide in accordance with ASME Code for pressure vessels and ANSI B-31
    - Piping
    - Valves
    - Fittings
    - Relief Valves
- Provide emergency trench-type drain to safe location for
    - Flammable leakage
    - Fire protection water
- Establish controlled area within 50 feet radius of bottoming cycle equipment
    - Limited access
    - Restricted area signs
    - No smoking signs
- Provide vapor generator stack heat sensor for following functions
    - Activate water spray system
    - Activate CO_2 System
    - Shutoff heat source
    - Dump toluene to safe location

use of toluene or some other organic working fluid for bottoming cycles, thus rendering any installed apparatus useless. Therefore, it is recommended that DOE fund a program to evaluate the hazards of toluene and other organic working fluids so that bottoming cycle apparatus can be made safe and when installed not subject to subsequent shut down because the hazards were not evaluated in a timely manner.

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SECTION 7

TECHNOLOGICAL FEASIBILITY

7.1 INTRODUCTION

The organic Rankine cycle is being developed in small sizes (600 to 800 HP) (References 28 and 29) by the Office of Coal Utilization of DOE. Working fluids and design details vary appreciably among these systems. In assessing the results of these and other studies that have been conducted in the area of bottoming cycle technology, preliminary conclusions have been reached that Rankine heat recovery cycles offer a cost effective means of achieving energy conservation. The intention of the DOE Pipeline Bottoming Cycle Demonstration Program is to assess the potential for widespread commercialization of bottoming cycle heat recovery systems. This program will make use of existing as well as advanced technology resulting from current programs of other government and private sources. Due to the small size of current developmental bottoming cycle apparatus, the applicability of this technology must be reexamined for the pipeline application where the size is of the order of 3000 HP - 6000 HP.

Because the exhaust gas temperatures of most gas turbines are below 1000^{o}F organic working fluids are preferred. The reason for this is that such working fluids have substantially lower latent heats of vaporization compared to water. This situation results in a reduction in the unavailable energy compared to steam because the vapor generator condition line (temperature versus heat input) lies close to the gas turbine exhaust gas condition line (temperature versus heat output). Organic working fluids are diverse in numbers and properties. Each fluid has a certain level of toxicity and flammability. Also each fluid has a temperature at which it is no longer chemically stable. Finally certain working fluids or their products of chemical breakdown can cause corrosion of the containment materials. It is also possible for the containment materials to catalyze decomposition of the working fluid in certain temperature ranges.

The use of organic working fluids spawns new uncertainties in component and system design. Although the thermodynamic and transport properties are generally available, the effects of scale on heat exchangers and turbomachinery must be reckoned with. Also because of toxicity, flammability or high cost the leakage of the working fluid from the containment must be eliminated or at least minimized. This imposes a stringent set of requirements upon the design of shaft seals, valve stems and flanged joints. It also means that the containment joints made by welding or brazing must be of a higher quality and inspected more thoroughly than similar such joints for steam apparatus, where a certain level of make up is permissible.

It is then the purpose of this section of the report to establish the degree of technological feasibility of the bottoming cycle when applied to the conservation of the waste heat from gas turbines used to pump natural gas in the nation's pipelines. Since the object of the DOE Pipeline Bottoming Cycle Demonstration Program is to effect a fuel saving in the pumping of natural gas, the feasibility of the bottoming cycle apparatus must be established before projections of fuel savings are made. The degree of feasibility of the pipeline bottoming cycle will establish any research and development efforts that are needed to commercialize this apparatus.

Based upon the results of the preliminary design and the environmental and safety assessment of that design a discussion of the technological feasibility of building the demonstration system are presented. The working fluid stability and compatibility with component materials and the adaptability of the bottoming cycle apparatus to the conventional pipeline gas compressor prime movers are discussed. An assessment of the component research and development currently being conducted on organic Rankine cycles in both the public and private sectors including a discussion of the status and the expected date when the components will be available are provided. The research and development work required for specific components before the demonstration systems can be built are also identified. This section includes a discussion of the availability of resource materials required for the production of bottoming cycle systems. The availability of both the raw materials and the processed materials is compared to the expected demand for bottoming cycle systems.

The selected working fluid, toluene, is compared with other possible working fluids. These fluids include mixtures of trifluoroethanol and water, methylpyridine and water and penta-hexafluorobenzene.

7.2 WORKING FLUID

The working fluid must have only an acceptable impact upon the environment and safety. In addition it must be chemically stable up to a temperature which will not be exceeded in the system and it must be compatible with the containment materials.

7.2.1 ENVIRONMENTAL IMPACT

The one aspect of the bottoming cycle system which causes some concern regarding environmental impact is the choice of toluene as the working fluid. Although it was known that toluene is flammable and toxic it was chosen as the working fluid for several reasons. Toluene is stable to 700^{o}F and is compatible with ordinary metallic containment materials. Although toluene is toxic, it is one of the more common chemicals. Large numbers of workers are exposed to it daily without serious problems. Recently, a bottoming cycle system with toluene as working fluid was tested at Sandia Laboratory (Reference 30).

Because of the concern about toluene, the selection of working fluid was reconsidered. The fluids listed in Table 7-1 have been selected by others as a result of their evaluations. Fluorinol is a working fluid selected by the Thermo Electron Corporation for a bottoming cycle system they are developing. The other two fluids, RC1 and RC2, were selected by the Monsanto Research Corp. after a comprehensive study to identify optimum working fluids for automotive Rankine engines.

All of these fluids are toxic to some extent and their toxicity levels are reproduced in Table 7-2. Most of the data on the toxicity levels were given in the NIOSH Registry of Toxic Effects (Reference 13) but some values were taken from Reference 12 and Reference 14. It is difficult to find comparable toxicity data for all the fluids so a comparative procedure was used. The first column, which is the lethal oral dose for 50 percent of the mice, indicates that Fluorinol is more toxic

TABLE 7-1

ADDITIONAL FLUIDS CONSIDERED

Fluorinol 50	50 mol%	222 Trifluoroethanol
	50 mol%	Water
RC2*	35 mol%	2 Methylpyridine
	65 mol%	Water
RC1*	60 mol%	Pentafluorobenzene
	40 mol%	Hexafluorobenzene

* Reference 12

TABLE 7-2

TOXICITY RANKING (DESCENDING ORDER)

	orl mus LD50	ihl rat LC50	ihl rat LC Lo
F-50	432 mg/kg	897 ppm/6H	-
Toluene	-	-	4000 ppm/4H
RC-2	916 mg/kg	8000 ppm/4H	5435 ppm/4H
RC-1	-	6000 ppm/4H	-
	-	16000 ppm/4H	-
Freon 22	-	-	250000 ppm/4H

than pyridine. The second column, which is the lowest reported lethal concentration for inhalation by rats, indicates that toluene and pyridine are equally toxic. The third column, which is the lethal concentration for 50 percent of rats by inhalation, indicates from Reference 14 that Fluorinol is more toxic than RC1 and from Reference 12 that RC2 is more toxic than RC1.

According to Reference 15 although toluene is listed in Suspected Carcinogens, Reference 31, that does not automatically mean that it must be avoided. In fact many substances which are common in everyday life and are in nearly every household in the United States are so listed. Listing occurs in two circumstances: (1) if a substance has reported neoplastigenic (tumor-forming) or carcinogenic effects or (2) if a substance is merely of oncological (tumor study) interest. It should be emphasized that no neoplastigenic nor carcinogenic effects are listed for toluene, implying that toluene is listed merely for oncological interest. Reactant grade toluene is specified for the proposed system. Such a grade of toluene contains less than 0.01% benzene, a carcinogen. The Occupational Safety and Health Administration has promulgated an 8-hour time-weighted average exposure limit of 1 ppm of benzene vapor. The 8-hour maximum allowable concentration of toluene corresponds to a concentration of benzene of less than 0.01 ppm, a value that is 1% of the promulgated benzene standard. Therefore, toluene can be considered safe carcinogenically if the 8-hour time-weighted-average concentration is kept below 100 ppm, Reference 15.

Shown in Table 7-3 is a comparison of the four working fluids. The performance is close enough that it should not be the principal factor in the fluid selection. Toluene and RC1 appear to be most compatible with conventional containment materials. RC2 failed the materials compatibility test in the Monsanto evaluation, and some problems have been reported with Fluorinol, although Thermo Electron has been using this fluid for several years. All the fluids are stable to over 625°F and are therefore suitable for this application. Toluene is the most flammable and RC1 is the only one that is nonflammable. Fluorinol and RC2 are mixtures of chemicals and water and although they will burn in the presence of a flame they are not considered as flammable as toluene. According to the ranking in Table 7-2, Fluorinol is most toxic, RC1 is least toxic, and toluene and RC2 are about half as toxic as Fluorinol. From the last row, which compares fluid costs, toluene and RC2 are inexpensive and RC1 and Fluorinol are expensive.

TABLE 7-3

FLUID EVALUATION

	Toluene	F-50	RC-2	RC-1
Performance	5067 HP	4796 HP	5002 HP	4923 HP
Materials Compatibility	good	good in absence of air	incompatible with Al or steel	good tested in 4130 steel
Thermal Stability	> 750°F	> 625°F	> 670°F	> 700°F
Safety	flammable	ignitable	ignitable	nonflammable
Toxicity	100 ppm TWA 200 ppm max.	toxic	comparable to toluene	least toxic
Cost $/lb.	0.10	7. now < 3. future	0.50	30. now < 5. future

7.2.2 SAFETY ASPECTS

A bottoming cycle is a system of conventional components, heat exchangers, pumps, turbine, valves, piping and controls. It takes heat from a waste heat source, such as a gas turbine exhaust, converts some of it into shaft power and rejects the rest of the heat to the atmosphere. Since it is a closed cycle, it has negligible effect on the environment when it is operating properly. However, in any mechanical system, there is always the possibility of failure of some component. If there is a leak in the heat exchanger tubes, turbine, or other system piping, some of the working fluid will be discharged into the atmosphere until the system is shut down and the fault corrected.

Toluene must be handled like other flammable liquids. Ventilated areas are required away from excessive heat and ignition sources. However, it is felt that the safety hazards can be reduced to acceptable levels by proper design practices and operating procedures.

Sundstrand was to place five Rankine cycle systems using toluene in the field this year, including three bottoming cycles for reciprocating engines and two for industrial waste heat applications. The operating experience with these units should be useful in the evaluation of possible safety and environmental problems in using toluene as a Rankine system working fluid.

Early in the study, water was considered as a working fluid, but toluene was selected because it could produce 22% more bottoming cycle power than steam for this application. Water is used as the bottoming cycle fluid in STAG plants, and several gas/steam combined cycles are operating in pipeline applications.

The basic question is whether the implementation of the Rankine bottoming cycle will have a significant adverse effect on the human environment. Although the working fluid is toxic and flammable, the bottoming cycle is a closed system and with proper design and operating procedures the risk of an accidental spill should be small. The five Sundstrand units being installed at various sites should provide a practical test of whether toluene is an acceptable working fluid. The fact that they are being installed is a good indication of their technological feasibility.

7.2.3 THERMAL STABILITY

Toluene is thermally stable up to a temperature of 750°F. The thermal stability temperatures of the other fluids considered are shown below:

Toluene	750°F
Penta-hexafluorobenzene	700°F
2 Methylpyridine	670°F
Trifluoroethanol	625°F

For toluene the thermal stability temperature is about the same as or higher than the exhaust temperature from the gas turbine (713-740°F). The system control must sense the vapor generator exit temperature and actuate the turbine flow control valve so that ample flow is available to preclude this temperature from ever approaching the thermal stability limit. The thermal stability temperatures for the remainder of the working fluids are lower than toluene. The measured working fluid temperature at the exit of the vapor generator for a 600 KW Fluorinol-water bottoming cycle is 600°F, only 25 degrees lower than the stability temperature.

7.2.4 MATERIALS COMPATIBILITY

Toluene is not reactive with ordinary materials. Since it is an excellent solvent, there is a possible reaction with nonmetallics such as insulation, plastics, paint and asphalt. Penta-hexafluorobenzene is presumed to be compatible with containment materials; in tests reported in Reference 12 it had no adverse effect upon 4130 steel, a low alloy product. Fluorinol does not readily react with containment materials, however, in the presence of air it may be slightly corrosive. Additives also can cause the Fluorinol to be more corrosive. Methylpyridine is not compatible with aluminum and steel.

7.3 ASSESSMENT OF CURRENT ORGANIC RANKINE CYCLE R&D

The Department of Energy (DOE) is currently sponsoring four Rankine bottoming cycle system developments and there are others under private development, for example Solar Division of International Harvester. The General Electric Company also has been involved with Rankine bottoming cycle systems for Steam and Gas Turbine (STAG) plants, total energy systems and locomotive diesel engines.

The following discussion is directed toward a description of the Rankine bottoming cycle technology available for use in the program. The DOE sponsored Rankine bottoming cycle systems under development by Thermo Electron Corporation, Mechanical Technology Incorporated and Sundstrand are discussed along with a description of a private industry development by Therma Technology Incorporated and Solar Division of International Harvester.

7.3.1 THERMO ELECTRON CORPORATION

DOE is supporting the demonstration of a combined diesel-organic Rankine cycle power plant. The Organic Rankine Cycle System (ORCS), being developed at Thermo Electron Corporation (TECO) will be installed on a 2500 KW diesel engine power plant at the New England Electric System substation in Lynn, Massachusetts. The ORCS will provide additional power of 440 KW (590 HP), an 18 percent increase. The demonstration phase of the program was scheduled to start during late 1979. A schematic cycle diagram is shown in Figure 7-1.

The working fluid selected for this application is Fluorinol-85, a mixture of 85 mole percent trifluoroethanol and 15 mole percent water. Thermo Electron has been testing organic Rankine cycle systems with Fluorinol 85 and other fluorinated alcohols for six years, and has demonstrated thermal stability of the fluid with boiler outlet temperatures to 600°F, compatibility with inexpensive materials of construction, and confirmation of the thermodynamic and physical properties of the fluid.

The vapor generator is a once-through, 3 parallel pass, forced convection unit with tubes finned on the gas side. Once-through boilers can have lower costs because there are no drums, and the working fluid inventory can be reduced. There is a potential problem of parallel channel flow instability but this can be handled by putting an orifice at the entrance of each tube with a pressure drop greater than the two-phase pressure drop in the boiler tubes. The working fluid conditions at the exit of the vapor generator are 700 psia and 550°F.

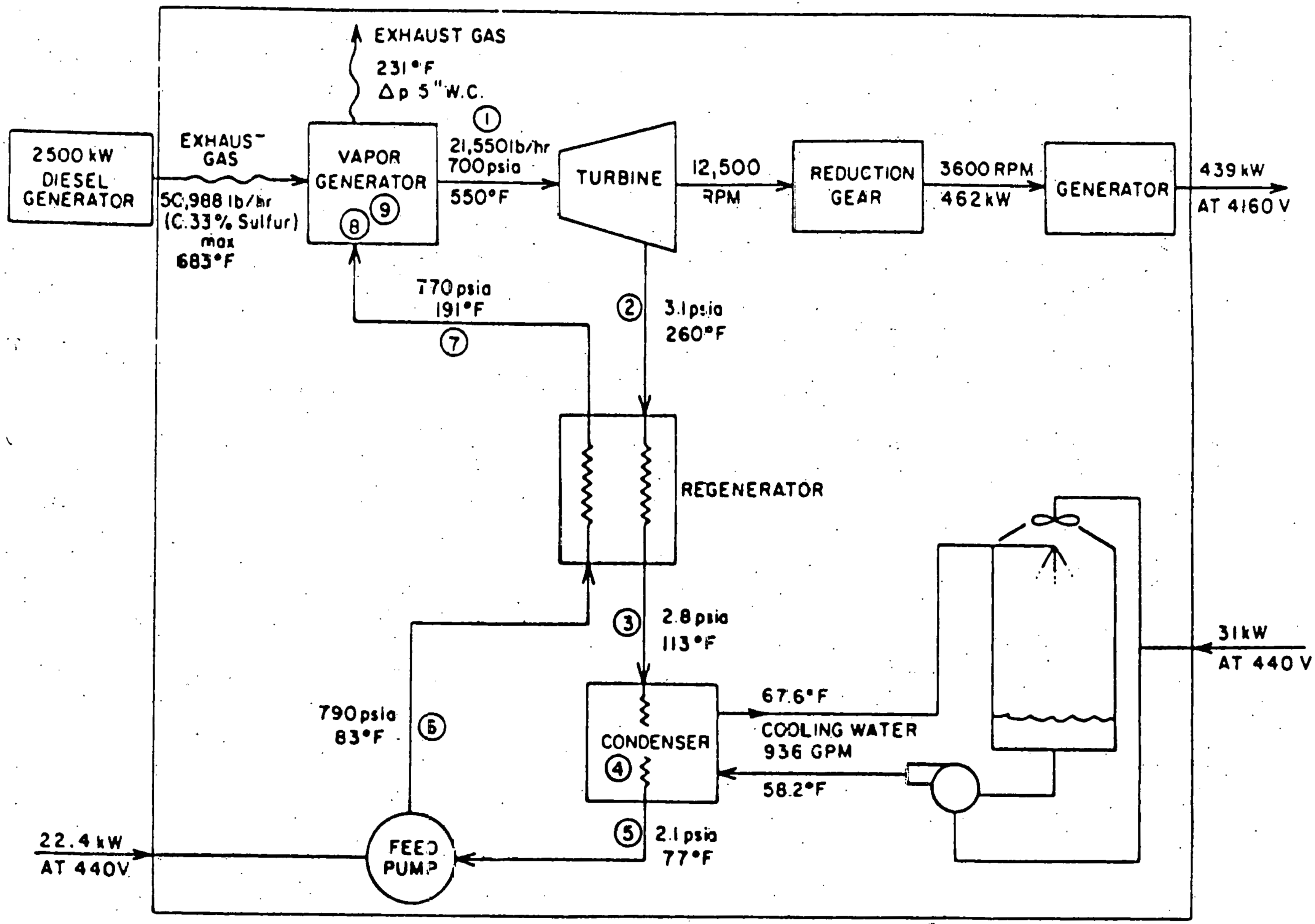

Figure 7-1. Thermo Electron Corporation Combined Diesel-Organic Rankine Cycle Engine With Cooling Tower.

The turbine for this system is a six stage axial design with a design speed of 12,500 RPM and a predicted efficiency of 81 percent. Leakage around the shaft is prevented with low speed double face seals and balance pressurization of the sealing fluid. The gearbox is a commercially available gear reduction with a speed ratio of 3.47.

A regenerator is used to transfer some of the heat from the turbine exhaust into the pressurized liquid before it enters the vapor generator. This reduces the amount of heat to be rejected in the condenser, and also reduces the heat added to the working fluid in the boiler and increases the efficiency of the power conversion system. The regenerator also raises the vapor generator temperature at the low temperature end to minimize acid corrosion. The regenerator is of shell and tube construction with stainless steel tubes and the shell is fabricated from low carbon steel. To prevent leakage the tubes and returns are furnace brazed in a hydrogen atmosphere. Welded construction is specified for the shell.

The condenser is water cooled and is of shell and tube construction. A horizontal baffle separates the desuperheating and condensing section from the lower subcooling section. The water headers are made of low carbon steel and the tubes and all other parts are of stainless steel.

The ORCS will drive an induction generator, which is essentially an induction motor driven above synchronous speeds. When the turbine produces net power the generator electrical output will increase from zero at 3600 RPM to full load at about 3630 RPM. Synchronizing equipment is not required with an induction generator.

In addition to the combined diesel organic Rankine cycle power plant for a municipal power plant Thermo Electron Corporation is under DOE contract to develop a diesel organic Rankine compound engine for long-haul trucks (Reference 32). The organic Rankine cycle system has been installed in a Mack F-model sleeper cab. This required preparation of the Rankine bottoming cycle subsystem for vehicle installation modification of the first prototype truck and installation of bottoming cycle subsystem into the truck. A single-vehicle test program was to be completed within 1979. This program was to include system functional check out tests, calibrated diesel engine tests with the organic bottoming cycle subsystems bypassed, compound engine baseline system tests with chassis-mounted dynamometer and road test checkout at Thermo Electron Corporation. Included also in the single-vehicle test program are functional checkout of chassis, driveability, braking and handling; dynamometer testing; noise evaluation; and fuel economy evaluation at Mack Truck.

7.3.2 MECHANICAL TECHNOLOGY INC.

DOE is also supporting the development of a steam/Freon binary cycle by Mechanical Technology Inc. (MTI). The system was installed in the municipal power plant at Rockville Centre, New York in 1978 and will use the exhaust heat from one or both of two diesel engines, each rated at 7675 BHP. The bottoming cycle provides 500 KW (670 HP).

The binary cycle is shown schematically in Figure 7-2. Diesel exhaust gas at 520°F is ducted into a low pressure waste heat boiler. Steam generated at 430°F and 40 psig is expanded in a single stage, radial inflow turbine and is exhausted at 1 psig. The low pressure steam vaporizes Freon in a shell and tube vapor generator. The Freon vapor at 190°F and 75 psig is expanded through a second single-stage, radial inflow turbine to a pressure of 2 psig. A single, integrated gearbox couples both turbines to a conventional 1800 RPM synchronous generator.

The two fluids, steam and Freon, are inexpensive, readily available, nonflammable, compatible with most materials, and carry no "psychological" barriers to ready acceptance. Consideration may have to be given the effect of Freon leakage on the ozone layer should a large number of systems become operational. Otherwise Freon is highly acceptable since it is widely used in domestic refrigerators and air conditioners. The water-Freon binary cycle has thermo-dynamic flexibility which allows this cycle to be applied to exhaust heat systems of 1000°F and higher without the problem of fluid stability. The upper Freon loop in Figure 7-2 can be added to recover some of the low temperature heat in the exhaust stack at the price of additional complexity.

The steam generator unit is of extended surface, finned tube construction. Forced circulation is utilized and the unit imposes a back pressure of about four inches of water on the diesel exhaust manifold.

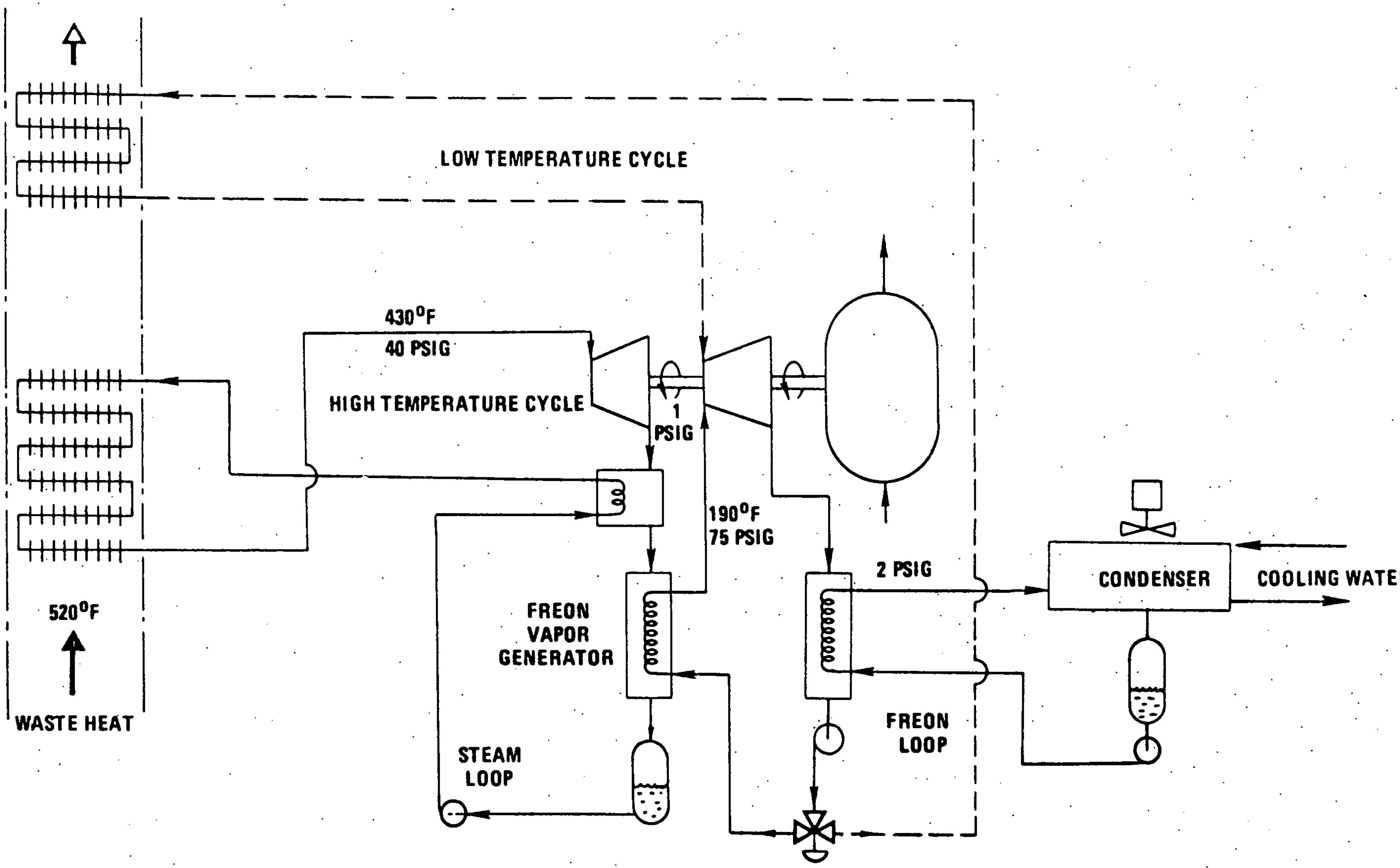

Figure 7-2. Mechanical Technology Incorporated Two Fluid Rankine Cycle Schematic

The heat exchanger skid contains the steam condenser/Freon vaporizer and the Freon condenser. The steam condenser receives slightly superheated steam turbine exhaust at the shell side flange. The tube side of this exchanger comprises a forced circulation evaporator and vapor separator to provide saturated Freon at the separator discharge. The condenser utilizes conventional refrigeration design practice and is water-cooled.

The radial inflow steam turbine operates at 42,200 RPM near optimum specific speed and therefore good efficiency. The steam turbine output is speed reduced to 9540 RPM and combined at the Freon turbine output shaft. The gearing allows both turbines to be independently optimized at maximum aerodynamic efficiency. The turbine rotors are mounted on oil film bearings. The steam turbine has 5 shoe preloaded pivoted pad bearings and the Freon turbine has 4-axial-groove bearings.

7.3.3 SUNDSTRAND CORPORATION

Another Rankine bottoming cycle system is being developed by Sundstrand Corp. which generates 600 KW (800 HP) using waste heat as the energy source. Toluene is the working fluid for this bottoming cycle, and was selected because of its good thermodynamic properties and Sundstrand's experience with this fluid. The system is shown schematically in Figure 7-3. It contains a vaporizer, a turbine, two pumps, a regenerator and a water cooled condenser.

The vaporizer is an existing heat recovery boiler design and uses a centrifugal separator to remove liquid from the vapor stream at the outlet of its natural circulation boiler. A small superheater is included to provide control margin; the bulk of the heat transfer area is in the preheater or economizer.

The regenerator is of a low cost design meeting the American Society of Mechanical Engineers Code but not requiring cleaning or the ability to withstand physical damage because it is sealed into the system. The condenser is also of low cost design and shares the same shell with the regenerator and the hotwell.

The turbine is a single stage supersonic impulse design and runs at 10,100 RPM. It is supported by journal bearings that are lubricated by

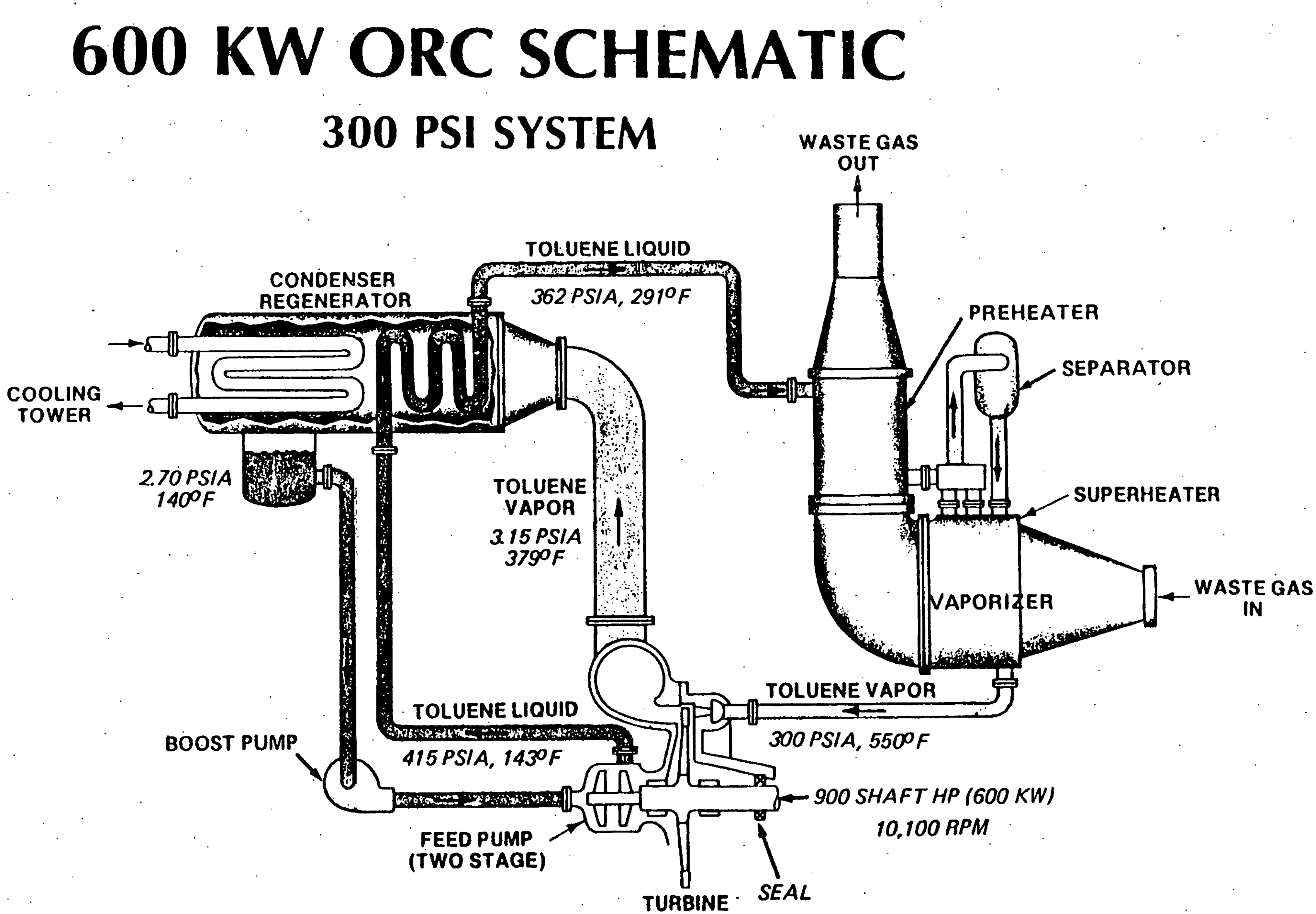

Figure 7-3. Sundstrand Corporation 600 KW RBC Schematic.

the working fluid. A double carbon face seal is used on the turbine shaft. The sealant fluid is liquid toluene. The feed pump is a two-stage centrifugal design mounted on the turbine shaft. The turbine was selected because it is simple mechanically, inexpensive, and takes advantage of the noncondensing properties of the organic fluid by operating at a pressure ratio of 100 to 1. The disadvantage of this type of turbine is that it is not particularly efficient.

The development unit has been checked out and initial performance data have been obtained. The development unit has over 3800 hours of operation. As part of a field test program three units were to have been installed as bottoming cycle systems for dual fuel or diesel engines in municipal power plants; two additional units were to have been installed in industrial waste heat applications.

7.3.4 THERMA TECHNOLOGY INC.

Therma Technology Inc. offers Rankine bottoming cycle apparatus to the petroleum, chemical and natural gas industries. They have several heat exchanger companies and working arrangements with a system engineering company and a turboexpander company. They have designed a bottoming system or power recovery package which uses either normal butane or propane as the working fluid.

Shown in Figure 7-4 is a schematic diagram of the power recovery system offered by Therma Technology Inc. It consists of the same components seen in other Rankine bottoming cycles. The bottoming cycle components are claimed not to be new; all the components of the hydrocarbon system are claimed to have been proven by years of experience. It is claimed that the Western Division makes tube and shell heat exchangers similar to the vapor generator and the recuperator and that Happy Division makes air-cooled heat exchangers like the condenser. These heat exchangers, it is claimed, are made for the refinery, petrochemical and gas transmission industries. The radial inflow turbine is claimed to be similar to those provided by Mafi-Trench Corp. to the gas processing industry in the middle 60's to reduce the pressure and temperature of fluids without wasting the energy. Several hundred units are claimed to be in operation.

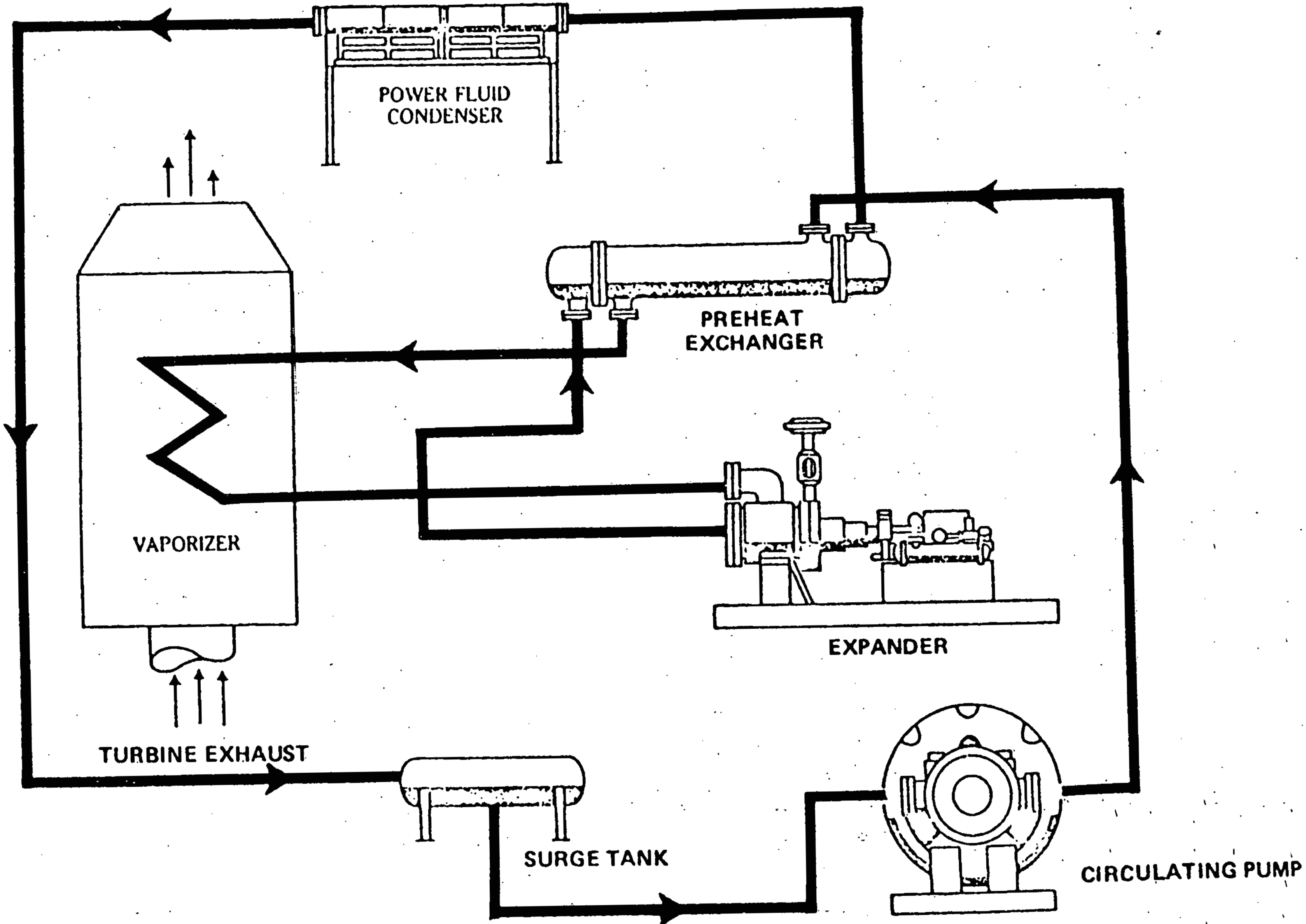

Figure 7-4. Therma Technology Inc. Power Recovery Package.

7.3.5 SOLAR DIVISION OF INTERNATIONAL HARVESTER

The Rankine bottoming system the Solar Division has been developing uses a two-pressure level steam system rather than an organic Rankine system. The two-pressure level system ameliorates one objection to steam, namely, that because of the large latent heat of vaporization of water the condition line for the steam in the vapor generator lies too far away from the exhaust gas condition line resulting in a large amount of unavailable energy. The second objection to steam, namely, turbine efficiency values in the range of 67 to 77% in the power range of 1900 to 4100 HP will be ameliorated by designing new high-pressure and low-pressure steam turbines for optimized throttle conditions using the latest turbomachinery flow path design technology. The two turbines have different rotative speeds so as to optimize each turbine design. The high-pressure turbine has two axial stages and the low-pressure, six.

A once-through vapor generator is used on the bottoming cycle to eliminate the steam drum and level control. A once-through boiler requires very pure water. Solar intends to use stainless steel for all containment, reduce makeup to a minimum and use a pair of low cost deionizing units interchangeably instead of a conventional high cost water treatment system. To help reduce makeup water the performance of the shaft seals on the turbine will be improved.

The condenser will be water cooled and have special features to avoid contamination of the condensate. A double tube sheet will be used with the tubing running through it. Thus, if there are leaks at the salt water end the coolant will not leak outside the containment. Solar Division expected to deliver the first units in 1980 for use on pipelines, offshore platforms, marine, and small electric utility base load applications (Reference 33).

7.3.6 GENERAL ELECTRIC COMPANY

The General Electric Co. has developed a series of gas turbine/Rankine cycle heat recovery systems commonly referred to as combined cycles. General Electric has chosen the acronym, STAG, (Steam And Gas turbine) for this combined cycle since steam has been the Rankine cycle working fluid. An example of a simple-cycle gas turbine combined with a steam turbine is shown schematically in Figure 7-5. GE STAG power systems are offered in

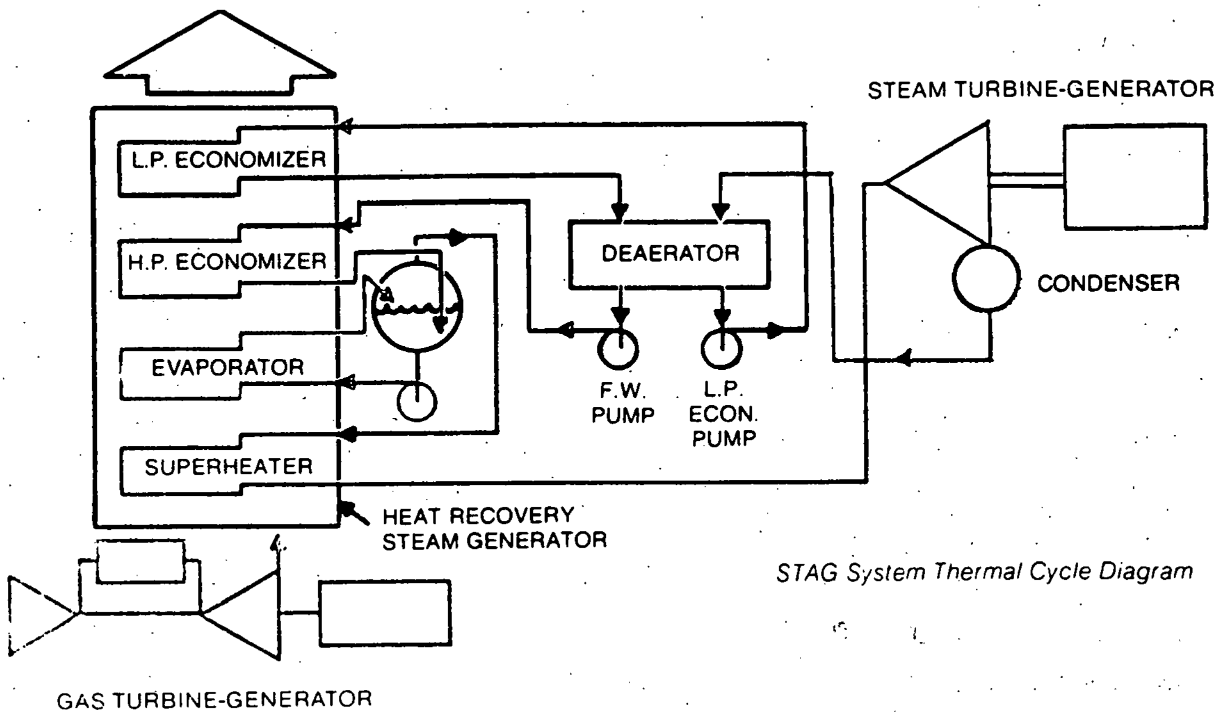

Figure 7-5 General Electric STAG Power System.

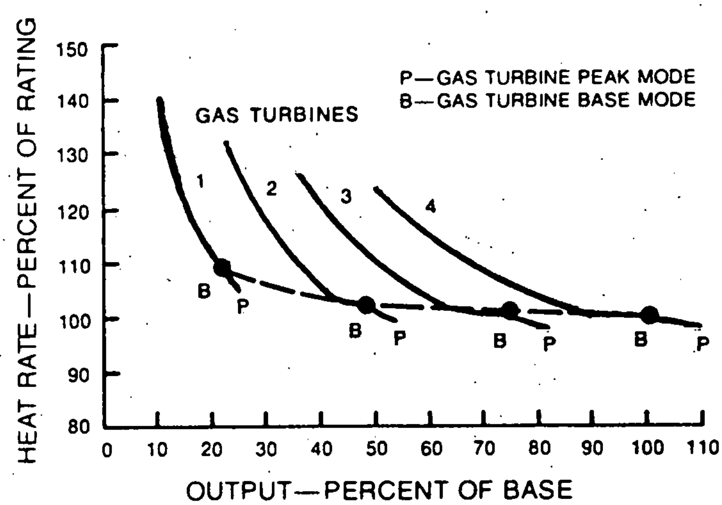

Figure 7-6. STAG Part Load Performance.

three basic sizes: STAG 100, 400, and 600. STAG systems offer excellent heat rates and reliability with simple nonextraction steam turbines and single low-pressure cycles. Bypass stacks are included to provide flexibility and maximum heat recovery steam generator reliability.

The STAG cycle has the unique capability to operate at a greatly reduced load while sustaining excellent heat rate characteristics. The STAG power systems' heat rate of 8100 Btu/KWhr can be maintained across the load range. By varying the operation of each of the several generator components, it is possible to operate over a wide load range with only minor changes in plant efficiency. This is shown by the dashed line on the STAG 400 part load performance curve shown in Figure 7-6.

The combined cycle has been accepted not only for production of electricity, but also for applications such as natural gas pumping. Shown in Figure 7-7 is a schematic diagram of the second unit installed at the Ceredo, W. Virginia station of the Columbia Gas Transmission Corp. This plant has a combined cycle efficiency of 35.7% (Reference 32). Improvements are being made in gas turbine performance and recent calculations indicate that a modern gas/steam combined cycle would have an efficiency of about 40% (Reference 34). There have been over 50 STAG systems installed in electric utilities.

For applications with an exhaust gas temperature lower than 1000°F some of the organic fluids are attractive as bottoming cycle working fluids. A schematic diagram of a Rankine bottoming cycle (RBC) is shown in Figure 7-8. The cycle includes a vapor generator or boiler, a turbine or expander, a condenser, a pump and possibly a regenerator, if the fluid is superheated after the expansion process. Except for the different working fluid and smaller size, Rankine bottoming cycle systems are very similar to the STAG* systems.

Many of the design problems that have been solved for steam are applicable to organic fluids. Mechanical design of the heat recovery boiler and the condenser can be based on the technology developed for STAG plants. Although the fluid design of the turbine is different for organic working fluids, the design principles are the same and the mechanical design of bearings, shafts, disks and casings can all be based on steam turbine

* General Electric Trademark: Steam and Gas Turbine

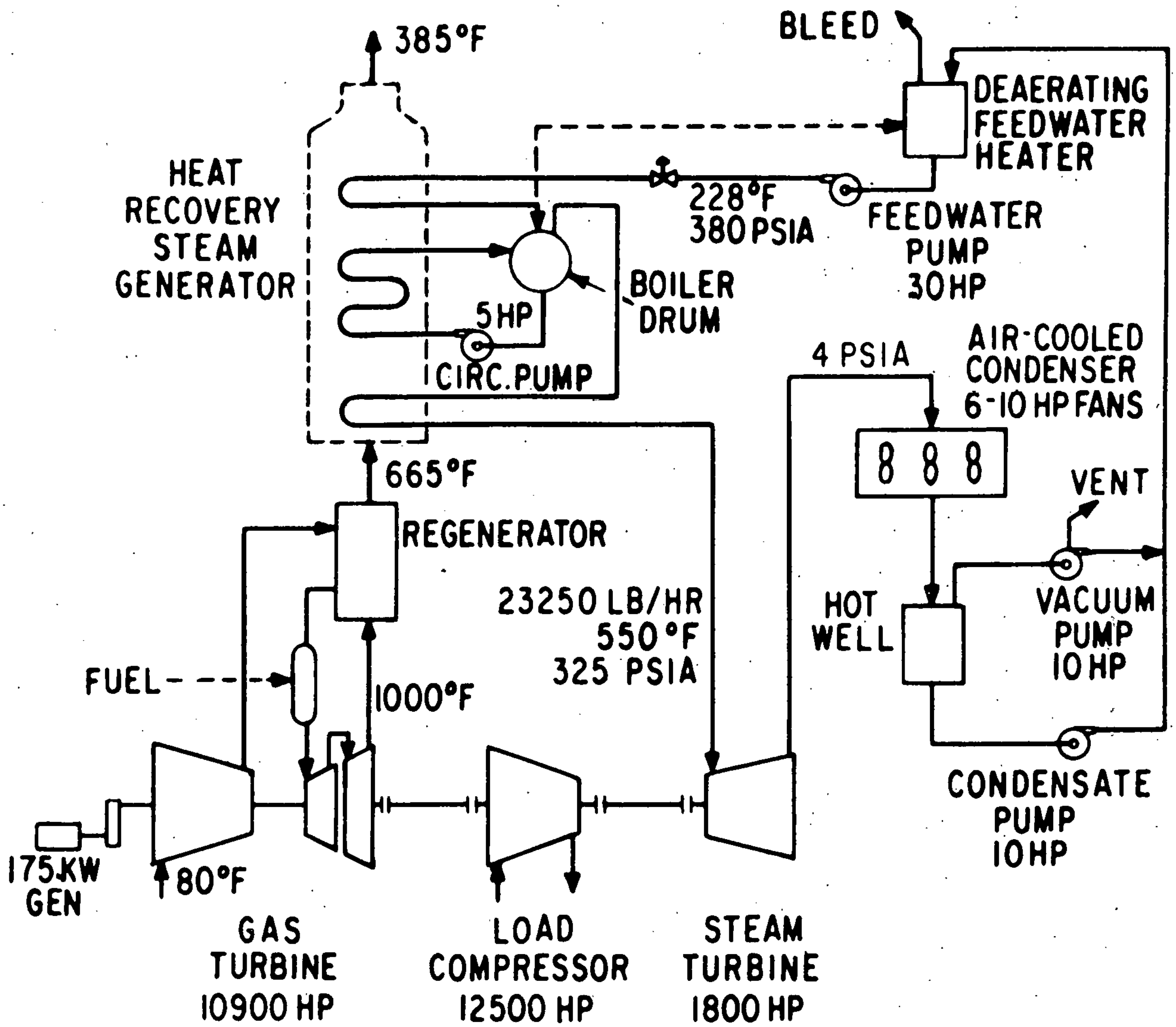

Figure 7-7. Schematic Diagram - Combined Cycle Unit No. 9 with Thermal Efficiency of 35% at Ceredo Station.

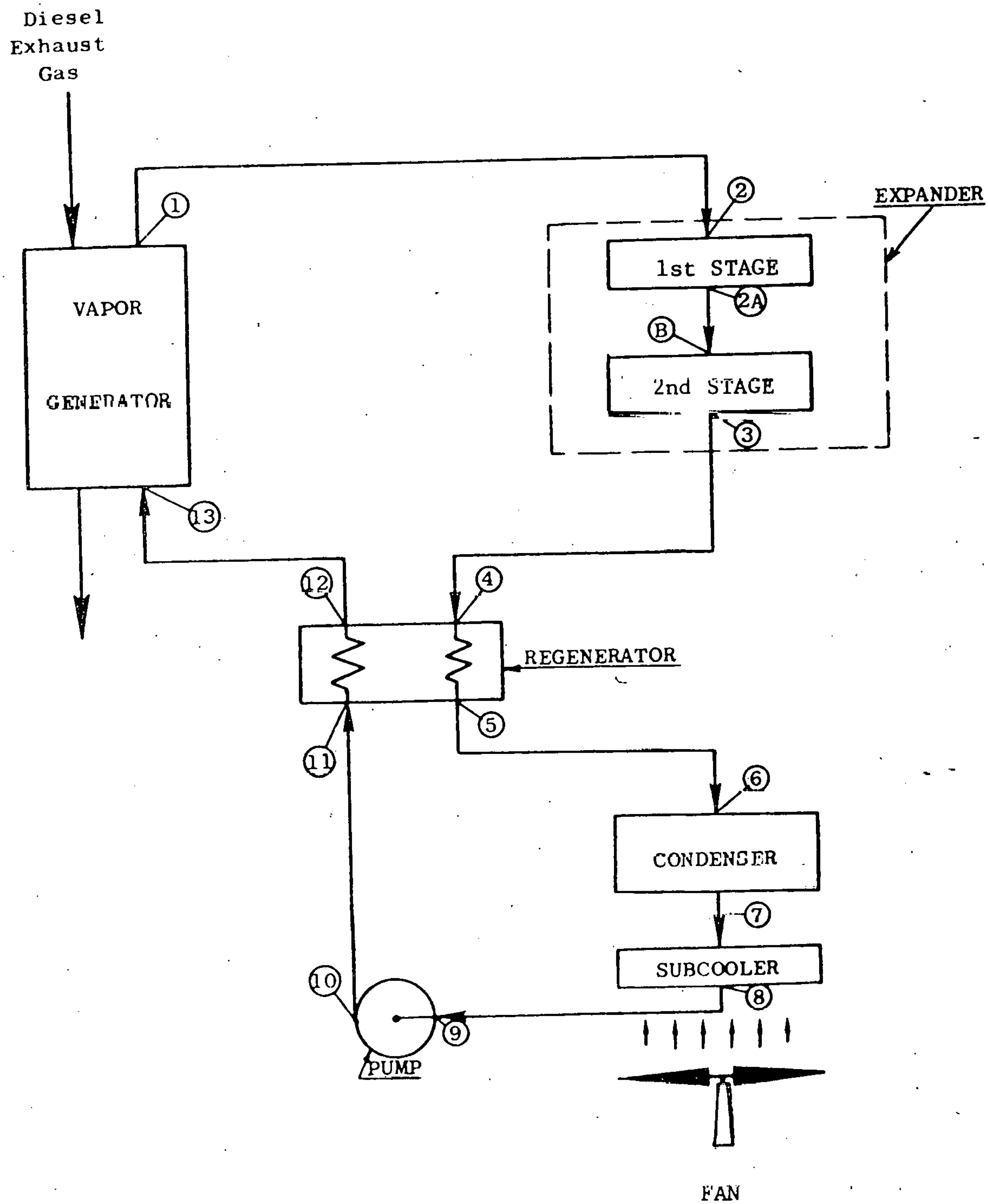

Figure 7-8. Schematic Diagram of a Rankine Bottoming Cycle.

experience. Any problems due to flammability or toxicity of the organic fluids can be minimized by good design practices.

One of the organic Rankine systems developed by General Electric was a total energy system and another was a low temperature Rankine system for solar energy conversion. The first provided electric power, heating and air conditioning for a motor home. It has an expander, condenser, pump, auxiliary equipment and a fired boiler. It used pyridine water mixture as the working fluid. The development of this organic Rankine system included assembly and test of a preprototype, prototype and preproduction models. Production of this unit was curbed by the onset of the current world fuel situation. The schematic diagram for the low temperature Rankine system is shown in Figure 7-9. It was built and tested driving an air conditioning system. After several thousand hours of test and evaluation the system is still operating satisfactorily on solar energy. This system was the forerunner of systems now in development to run three and ten ton air conditioning systems for use with solar collectors.

7.4 STATUS OF RANKINE BOTTOMING CYCLES

The three bottoming cycles being developed under DOE contracts have been discussed above and are summarized in Table 7-4. The power levels of these systems, 600 to 800 HP, are much smaller than the 5000 HP systems that would be used to bottom pipeline gas turbines. However the technology is applicable but needs to be scaled up to larger sizes. As shown in Table 7-4, these three systems are to be operational in 1979, in demonstration tests that will be run this year.

Some companies that serve the chemical and petroleum industries are proposing to use their technology in heat exchangers and radial inflow expanders for bottoming cycle systems of the size required for pipeline gas turbines. Petro-chemical plants use large hydrocarbon heat exchangers for heating, cooling, evaporating and condensing. They use expansion turbines to recover energy from high pressure gas and to provide refrigeration. Butane or propane is recommended by them as the working fluid; a bottoming cycle using these working fluids would produce

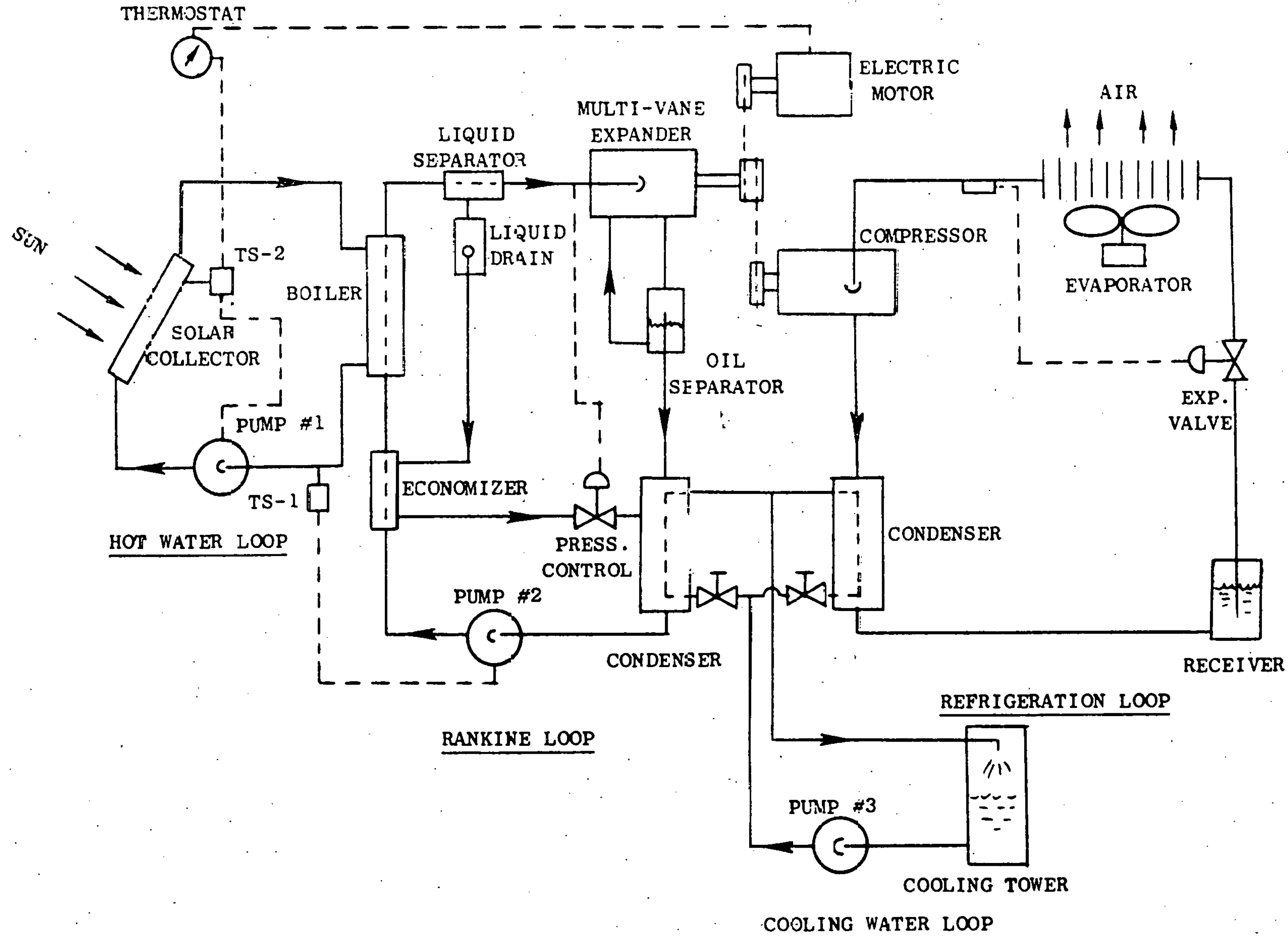

Figure 7-9. Schematic of Demonstration Rankine Powered Air-Conditioner.

TABLE 7-4

DOE BOTTOMING CYCLE RESEARCH AND DEVELOPMENT

Manufacturer	Sundstrand	Thermo Electron	MTI
Working Fluid	Toluene	Fluorinol 85	Steam/Freon
Power, HP	805	590	670
Turbine Inlet Cond. psia/°F	300/550	700/550	55/430 90/190
Turbine	10,100 RPM 1 Stg. Supersonic	12,500 RPM 6 Stg. Axial	42,200 RPM radial / Radial 9540 RPM
Boiler	Natural Circulation	Once-through Forced Convection	Forced Circulation
Load	Generator	Generator	Generator
Seal Type	Double Face Seals	Double Face Seals	
Available	Field Test 1979	Demonstration 1979	Installation 1978

about 14% less power output than the toluene cycle, but some practical advantages are claimed for it, such as smaller discharge pipe size.

It is also proposed by others that the bottoming cycle using propane or butane be operated supercritically using a once-through vapor generator. The vapor generator is a standard waste heat unit in use for twenty years. Similar units are used in the petro-chemical industry to reclaim waste heat in a heater or vaporizer. The materials of construction are carbon steels.

The expanders for propane or butane systems may have small enough pressure ratios so that a single-stage radial inflow turbine will suffice. Efficiencies are in the range of 80 to 84% according to one manufacturer. Similar turbines are used in the petro-chemical industry and are known as turboexpanders or gas expanders (References 35 and 36). Such turbines are used for the expansion of hydrocarbon fluids from higher pressures, generating power from energy that would otherwise be wasted. An example is chilling hydrocarbons, such as propylene from 125 psia and 100^{o}F to 20 psia and -35^{o}F. Other examples include separation processes for ethane and propane or methane and ethane. The turboexpander may also be used in the liquefaction of natural gas. Seals have been produced for such turbines using a sealant gas or liquid and the bearings are oil lubricated. A number of seal arrangements are shown in Reference 37.

The estimated efficiency for the pipeline bottoming cycle (5000 to 6000 HP) turbine is 86%. This represents the value expected when the turbine becomes a mature product in production. To assure this efficiency value the turbine will be subjected to a well-balanced aerodynamic design of the blading constrained only by rotor stresses due to steady and vibratory loads. The number of stages chosen will be adequate to permit moderate aerodynamic loading on the blading, which is conducive to high efficiency, and a rate of annulus-area growth commensurate with moderate rates of diffusion on the blading. The blade chords selected in the front stages will be adequate for close tolerances on blade profiles to minimize flow separation and errors in flow incidence angles and to provide adequate blade Reynolds numbers to minimize profile drag. Also considered in blade-chord selection will be sufficiently high aspect ratios to limit secondary flow losses. In the last stages the chords will be selected to provide adequate section modulus to prevent excessive vibratory stress levels. Surface roughness will be controlled to minimize profile drag. The blade height will be selected to minimize the turbine leaving loss within stress limitations. Consideration will be given to the use of an exit diffuser to reduce further the turbine leaving loss. The blading will be designed taking the three dimensionality of the flow into account, resulting in twisted nozzle vanes and buckets throughout. In this way the incidence angles will be optimum for all radial positions. The blading profiles will be generated using two dimensional design methods, resulting in the minimization of diffusion losses on the blading. Bucket covers will be used on all stages where the resulting stresses permit to minimize tip clearance losses. Interstage seals will be placed at the minimum possible diameter and utilize the best tooth configuration to minimize interstage leakage.

In contrast the turbine selected for the bottoming cycle demonstration will have an efficiency of 77% so as to minimize the time and cost associated with the demonstration program. It is felt that this level of efficiency will be adequate to demonstrate the attractiveness of the pipeline bottoming cycle. This is true because the economics of the bottoming cycle with a production turbine can be obtained by adjusting the results obtained for the demonstration turbine for the correct efficiency and cost of the production version. As the market develops the level of the turbine efficiency will rise, improving bottoming cycle output.

The demonstration turbine will be adapted from available steam turbine designs. This will shorten the design and fabrication time for the demonstration turbine and reduce the cost. Such a turbine will be adequate for demonstrating the technological feasibility of the bottoming cycle system. In addition in approximately one year's running time all operational problems can be identified and corrected. Data can also be compiled on reliability and maintainability of the bottoming cycle system, including the turbine.

The recuperator in a propane or butane bottoming cycle would be a standard tube-in-shell heat exchanger made from carbon steel. The air cooled condenser is similar to units now used to cool natural gas after compression. These could be supplied by a number of domestic companies and made of carbon steels.

7.5 REQUIRED RESEARCH AND DEVELOPMENT

No component research and development is required before a pipeline bottoming cycle demonstration can be accomplished. Design verification tests may be required on the bearings and seals. All heat exchanger components can be purchased from vendors as special designs of equipment that they presently manufacture.

Since no 5000 HP, 500^{o}F organic Rankine bottoming cycle for a gas turbine has been built, the demonstration system will verify component matching, establish integrated system performance and supply information on operating problems, reliability and maintenance requirements.

7.6 ADAPTABILITY OF CONVENTIONAL EQUIPMENT TO BOTTOMING CYCLES

Most of the bottoming cycle components can be obtained from available commercial sources. Energy Users News lists 98 heat exchanger manufacturers, many of which could design and manufacture vapor generators. The same source lists 29 condenser manufacturers, of which many could produce air-cooled condensers. Refinery equipment suppliers stock numerous models of hydrocarbon pumps. The pumps for the bottoming cycle will have to be provided with cavitation protection. Toluene is a petroleum derivative which is produced commercially in large quantities.

Turboexpanders for hydrocarbon fluids are available from several sources. Some manufacturers make radial inflow expanders with variable inlet vanes for use in the petroleum and chemical industries and also for low temperature propane power systems that extract power from geothermal heat sources. The largest installed power is in the hydrocarbon process industry where the expanders drive compressors. Both radial inflow machines and axial flow expanders are made commercially in various sizes by more than one manufacturer.

7.7 ADAPTABILITY OF BOTTOMING CYCLES TO PIPELINE GAS TURBINES

The organic bottoming cycles are designed to match the available levels of gas turbine exhaust gas temperatures. Previously in Table 4-12 it was shown that heat rate improvement values greater than 25% are readily attainable on simple cycle gas turbines. There are a variety of ways to use the additional power generated by the bottoming cycle. The gas turbine can be run at part load or smaller, less efficient prime movers can be shut down; in each case the pumping is done with less fuel consumption. Some pipelines are part of an electric utility system, in which case it may be feasible to generate electricity with the bottoming cycle. The ideal case is when the pipeline wants to increase its pumping capacity; then the bottoming cycle can provide additional pumping power with the same fuel rate as before. Each installation requires a resolution of the most advantageous way to use the additional power.

7.8 MATERIALS AVAILABILITY

The materials that would be used for the bottoming cycle components are shown in Table 7-5. Very little high alloy steel is required and most of these materials are readily available. In the event that the bottoming cycle is generally applied to the pipeline industry it must be determined whether the raw materials needed for the bottoming cycles will constitute a drain on the available raw materials or process materials. In order to obtain an estimate of this, the total amount of the horsepower which would be potentially available for bottoming cycles was determined. From the materials required for the pipeline bottoming cycle reported in the preliminary design task report (Reference 38), the actual amount of materials for the bottoming cycles were estimated based on the total amount of bottoming cycle horsepower. It is estimated that if all the available gas turbines in the pipeline industry were bottomed there would be 1,855,000 installed HP in bottoming cycle equipment. Shown in Table 7-6 are the elements present in the materials of which this bottoming cycle equipment would be made. Also shown are the required weights in short tons of the materials for the total potential bottoming cycles. The U.S. production of these elements is shown in the next column along with the world resources and this was taken from Reference 39. Finally, the last column shows the percent of the U.S. production of these elements which would be necessary in order to manufacture the bottoming cycles for all potential sites in the U.S. There are two exceptions in the last column, indicated by note 3, in which the U.S. production was so small that the imports would have to be included. Looking at the value shown in the last column most of the materials that are required for the bottoming

TABLE 7-5

COMPONENT MATERIALS

Turbine and Pumps	
Casing and Stators	2.25 Cr-Mo Steel
Wheels and Blades	B5F5 Wrought Steel Alloy
Vapor Generator	
Tubes, Fins	SA178A
Tube Supports	A36
Headers	SA108A
Casing	Carbon Steel
Condenser	
Tubes	SA214
Fins	Aluminum
Box	SA515

TABLE 7-6

AVAILABILITY OF MATERIALS

ESTIMATED WEIGHTS OF STRATEGIC MATERIALS IN PIPELINE BOTTOMING CYCLE SYSTEMS

ASSUMING ALL POSSIBLE BOTTOMING CYCLE POWER (1,855,000 HP) IS INSTALLED

Element	Required Weight Short Tons	U.S. Production[38] Short Tons	World Resources[38] Short Tons	Percent of U.S. Prod.
Aluminum	5908	5.0×10^6	4.7×10^9	0.11
Chromium	90.9	(a)	797×10^6	0.0061[c]
Copper	395.1	3.0×10^6	1.5×10^9	0.013
Magnesium	25	124,000		0.02
Manganese	383	(b)	1.2×10^9	0.019[c]
Molybdenum	4	52,500	7.2×10^6	0.0076
Nickel	72	3,000	99.6×10^6	2.4
Vanadium	0.6	5,000	18.8×10^6	0.012

Notes

(a) No domestic production since 1961. Imports in 1970; 1,500,000 short tons.

(b) Very little domestic production. Imports in 1970; 2,000,000 short tons.

(c) Percent of imports.

cycle are in great enough supply as not to be affected by the installation of bottoming cycle apparatus on pipelines. However, there are two exceptions to this general statement;namely, the amount of aluminum and nickel required. It appears that the exports from foreign countries would have to be included in the source supply in order to make the percent of the available resources a small enough number so as not to impact upon the available materials in the area of aluminum and nickel.

7.9 CONCLUSIONS

Toluene was selected for the working fluid in spite of certain levels of toxicity and flammability because toluene is a well-known, well-documented substance having acceptable levels of toxicity and flammability. It has adequate thermal stability, superior performance, a low cost and is readily available. In addition toluene is being carried along as the working fluid in one of the organic Rankine systems being developed by the DOE.

Organic Rankine systems of less than 900 HP are being developed by Thermo Electron, Sundstrand, Mechanical Technology, Inc., and others. A number of these smaller systems have or shortly will be demonstrated in power plants and industry.

It is in connection with these smaller systems that research and development is being carried out. There is no organic bottoming cycle research and development known to be in progress in sizes as large as 5000 HP. There are, however, steam bottoming cycles in operation in large sizes in both pipeline compressor service and in power plants. The first steam bottoming cycles of large size went into service in 1968 on both power plants and on pipelines.

Much of the apparatus needed for large size organic bottoming cycles is in service in petrochemical plants in the form of hydrocarbon process equipment. For this reason no research and development is required. However, design verification testing may be needed for the turbine bearings and seals. These need to be tested in full size, at expected speeds and under the expected operating conditions.

Since much of the kind of apparatus needed for the organic bottoming cycle is in general use in the petrochemical field, this apparatus can be adapted to use in large organic bottoming cycles. In the demonstration system an organic turbine will be designed utilizing steam turbine technology, adapted to organic use by employing new bearings and seals. It is estimated that this turbine will have an efficiency of 77%. It represents the most timely and lowest cost approach to the demonstration. By the time the bottoming cycle is in full production an efficiency of 86% will have been achieved. Where economically feasible, consideration could be given to replacing the low-efficiency turbines by the higher-efficiency turbines as they become available.

The installation designs for the three sites studied brought to light no insurmountable problems. Therefore, the bottoming cycle can be adapted to gas turbines in the field without much difficulty. Heat rate reduction exceeded the DOE target of 20% on all three sites by 30% or more.

Because the large size organic bottoming cycle operates at low temperature, no high temperature alloys are required. All elements needed for the component parts are produced in large quantities domestically except chromium and manganese. Historically, these two elements have been imported in sufficient quantities for domestic use. The amounts of elements needed for the bottoming cycle are small in comparison to current production or imports (in the case of chromium and manganese) except nickel. Bottoming cycles may require up to 2.4% of production.

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SECTION 8

OPERATIONAL RELIABILITY AND MAINTAINABILITY

8.1 INTRODUCTION

System reliability and maintainability are major factors in the design and operation of a bottoming cycle applied to a pipeline pumping station. These factors can have a major influence on the economics of initial installation and operation of the bottoming cycle system and, therefore, be deciding factors when pipeline companies are faced with the decision as to apply or not apply such a system to a particular pumping station.

It should be noted, before proceeding, that bottoming cycles(or combined cycles as they are sometimes called) are not completely new to the pipeline industry. Four combined cycle (steam) systems were installed on pipelines between 1968 and 1970, two at one site and two at another. Although some initial design and operating problems were encountered with the first unit installed on one site, successful, reliable operation was obtained for the two units on the site through the use of improved components and controls (References 40 and 41).

Obviously, the combined cycle drive system is more complex than the standard gas turbine/compressor system. This factor inherently indicates a potential decrease in the combined cycle reliability. It further indicates the potential for increased maintenance of a planned and unplanned nature. These negative features, along with other pros and cons for the bottoming cycle, must be carefully evaluated by a Gas Transmission Company before deciding to install the bottoming cycle system at a particular pumping station.

In this section reliability and availability values for the gas turbines/compressor units presently being considered for bottoming cycle demonstrations are indicated and discussed. Reliabilities of similar types of Rankine cycle systems are reviewed. Potential reliability problems with the proposed bottoming cycle designs are evaluated and the status of research on these problems presented. An estimate of the bottoming cycle system availability is provided and the general overall system acceptability from a reliability standpoint indicated. Estimated maintenance requirements for the bottoming cycle system are presented and a comparison between these requirements and the requirements without a bottoming cycle made. The severity of new maintenance problems, site-sensitive problems, and/or procedures are discussed and the general acceptability of the bottoming cycle system from a maintenance standpoint reviewed. Finally, general conclusions and recommendations in the areas of reliability and maintainability as regards the bottoming cycle system are presented.

8.2 RELIABILITY OF EXISTING GAS COMPRESSOR PRIME MOVERS

Since the bottoming cycle system is dependent upon the gas turbine to which it is applied for its energy input in a pipeline compressor station, it seems appropriate to first determine the reliability of the gas turbine/gas compressor system. Since the gas turbine and natural gas compressor are coupled the reliability of this system must be examined. Data taken from turbocompressor operation at pumping stations operated by Texas Gas Transmission Company, Columbia Gulf, and Pacific Gas and Electric Company are shown in Tables 8-1, 8-2 and 8-3, respectively. In order to fully understand the data presented in these tables a definition of terms is necessary.

Reliability	Probability that a unit (system) will function for a period of time under specific conditions.
Availability	Fraction of a given period of time that a unit can function (whether it does or not).
Utilization Factor	Ratio running hours divided by total installed hours for a machine.

All of the above are normally expressed as a percentage. Reliability does not normally include planned outages for inspection and maintenance work and is based on forced outages. Availability includes forced and planned outages. Pipeline companies maintain records of data on a

TABLE 8-1

TURBINE UNIT RELIABILITY DATA

Site: Covington, TN
Pipeline: Texas Gas Transmission Company

Year	Average Reliability %	Average Availability %	Utilization Factor %
1973	99.95	99.09	98.45
1974	99.83	98.17	97.07
1975	99.65	98.59	90.05

- Aircraft derivative gas turbine unit
- Standby spare units available
- 8 hours for complete unit change
- Frequency of changes: 1-2 times per year
- Planned outage: Nil
- No field overhauls

TABLE 8-2

TURBINE UNIT RELIABILITY DATA

Site: Rayne, LA
Pipeline: Columbia Gulf Transmission Company

Year	Average Reliability %	Average Availability %	Utilization Factor %
1973	99.93	99.75	36.04
1974	99.77	99.52	18.17

- Aircraft derivative gas turbine unit.
- Standby units available.
- 16 hours for complete unit change.
- Factory overhauls.

TABLE 8-3

TURBINE UNIT RELIABILITY DATA

Site: Burney, CA
Pipeline: Pacific Gas and Electric Co.

Year	Average Reliability %	Average Availability %	Utilization Factor %
1974	97.6	93.2	82.0
1975	98.3	95.3	75.1
1976	97.0	89.0	74.0
1977	99.5	95.4	82.7

- Aircraft derivative gas turbine unit (LM1500)
- Gas generator removed for factory overhaul
- Pcwer turbine and compressor maintenance at site

monthly basis and average this data for yearly values. Availability is of the greatest importance to the equipment user since it can have a large effect on economics.

Table 8-1, based on data from the Texas Gas Transmission Company, Covington, Tennessee compressor station, is typical of gas turbine/gas compressor units in which the gas turbine is of the aircraft derivative type. There are no planned outages and station overhaul work on the gas turbine is nil. If an operating problem is encountered, the complete gas generator and power turbine, if necessary, are removed and replaced with a spare unit. The removed unit is then returned to the manufacturer's overhaul depot for repairs. Removal and installation work is accomplished in an 8-16 hour period. This fast turnaround results in the high availability indicated in Table 8-1.

Table 8-2 was compiled from data obtained from Columbia Gulf, Rayne, Louisiana compressor station and indicates again the high reliability and availability obtained from the aircraft derivative units. The low utilization factor is due to the decrease in gas supplies available at this particular station and the need to operate only one of the three available turbines. The horsepower demand at this location requires only one of the three available units to operate. With the installation of a bottoming cycle on one of these units the utilization factor would be expected to increase to 96-98%.

Table 8-3 indicates data obtained from Pacific Gas and Electric Company for their Burney, California compressor station. The turbine compressor reliability is somewhat less and the availability considerably less when compared to the two previous compressor stations. The difference in planned outage times for inspections is the major reason for the difference in availability.

Table 8-4 is based on data taken at the Texas Gas Transmission Corporation's Lake Cormorant compressor station where two recuperated industrial gas turbines are employed to drive gas compressors. This table clearly points out the difference in availability between industrial and aircraft derivative gas turbines. The decrease in availability of the

TABLE 8-4

COMPRESSOR STATION RELIABILITY DATA

Site: Lake Cormorant
Pipeline: Texas Gas Transmission Company

A. General Electric Model M3912R, Frame 3, Recuperated Gas Turbine

Year	Average Reliability %	Average Availability %	Average Utilization Factor %
1973	99.97	96.98	95.81
1974	99.98	88.46	78.21
1975	99.97	79.35	46.75
Three-Year Average	99.97	88.26	73.59

B. General Electric Model M3112R, Frame 3, Recuperated Gas Turbine

Year	Average Reliability %	Average Availability %	Average Utilization Factor %
1973	99.79	96.81	95.88
1974	99.96	94.02	82.27
1975	99.80	76.55	64.23
Three-Year Average	99.85	89.13	80.79

industrial gas turbines is due to the planned outages for inspections and overhauls made in the field.

8.3 RELIABILITY OF SIMILAR RANKINE SYSTEMS

Since bottoming cycles applied to pipeline compressor stations and employing organic Rankine cycle components are nonexistent at this time, reliability data is not available. However, some insight into what reliabilities are practically obtainable can be achieved by looking at similar systems.

In 1967, the United Fuel Gas Company, now the Columbia Gas Transmission Co., installed a 10,500 HP gas turbine/steam turbine combined cycle (Unit #8) as a compressor drive in their Ceredo, West Virginia station. In 1970 a system of improved design (Unit #9) and rated at 12,500 HP was installed in the same station. The improved design for Unit #9 has resulted in practically troublefree operation since startup in 1970 (References 40 and 42). The unit is presently used for peaking service. Reliability and availability data is not readily available but operating personnel are well satisfied with bottoming cycle component performance.

A STAG combined cycle power system of General Electric Company utilizes a bottoming cycle to generate electrical power and, although the working fluid is water (steam), the Rankine cycle components are similar to those proposed to drive a gas compressor. Table 8-5 indicates the forced outage rate, planned outage hours and availabilities for such a system (Reference 43). The gas turbine has a lower availability percentage than the aircraft derivatives shown in Tables 8-1 and 8-2. The plant availability values in Table 8-5 are about the same as for industrial gas turbines as shown in Table 8-4 and significantly lower than the aircraft derivative gas turbine in Tables 8-1 and 8-2. It is evident from this data that the Rankine cycle systems have inferior availability percentages compared to the aircraft derivative gas turbines.

Table 8-6 provides statistics on a typical fossil fired steam plant, (Reference 42). This data indicates that the boiler is, by far, the most unreliable component in the Rankine system. This is very likely to be true also in the proposed pipeline pumping bottoming cycle system.

TABLE 8-5

STAG COMBINED POWER PLANT AVAILABILITY
(FUEL #2 DISTILLATE)

Component	Forced Outage Rate (Hours)		Planned Outage (Hours)
Gas Turbine	0.05	(427.1)	217.9
Steam Generator	0.02	(170.8)	During GT Outage
Steam Turbine	0.01	(85.0)	During GT Outage
Gas Turbine Availability	92.6%		
Plant Availability	89.0%		

Based on 8760 Hours/Year

TABLE 8-6

TYPICAL DATA
FOSSIL FIRED STEAM PLANT - EDISON ELECTRIC INSTITUTE SURVEY 1966-1975

		Forced Outage Rate	Operating Availability
1.	Turbine Availability	1.3%	92.3%
2.	Condenser Availability	0.0%	96.3%
3.	Boiler Availability	3.9%	87.6%
4.	Generator Availability	0.3%	95.3%
5.	Other Equipment	0.4%	96.8%
	Overall Plant Availability (Weighted Averages)		84.4%

8.4 PROPOSED BOTTOMING CYCLE SYSTEM RELIABILITY

A reliability diagram for the combined system is shown in Figure 8-1. As can be seen from this diagram the reliability of a system is the product of the reliabilities of its parts and the system reliability is the product of the component reliabilities, (Reference 44). The addition of a bottoming cycle to a gas turbine prime mover which drives a gas compressor presents some potential reliability problems and reduces the reliability and availability of the particular unit to which it is applied. Even though the bottoming cycle system can be divorced from the gas turbine via the gas diverter valve, the diverter valve has become a part of the gas turbine/compressor system and, therefore, adds another factor in the reliability expression.

To insure that the bottoming cycle system will not place a penalty on the original system the diverter valve must possess a high degree of reliability. From the standpoint of design the valve must be rugged. Oxidation and corrosion from the hot gas environment, along with thermal distortion produced by cyclic operation, must be thoroughly investigated in the design stages. Vibration created by mechanical or acoustical sources can produce metal fatigue failures in a very short time. The source of power for actuation should be from a highly reliable source or redundancy provided. The gas leakage rate through the valve must be kept to a minimum so that maintenance can be performed on the vapor generator while the turbine continues to operate. Diverter valves of the required size are commercially available.

The vapor generator operates at the highest temperature and pressure of all the bottoming cycle system components and can be subjected to thermal shock (primarily at startup) and many thermal cycles depending on system operation. An extensive amount of tube-to-header welded joints must be made during the unit's fabrication. The large number of joints coupled with thermal cycling operation or thermal shocks makes the potential for vapor or liquid leaks high. In a steam generator some amount of leakage is permissible and always occurs. With an organic working fluid such as toluene this is not permissible. Therefore, the vapor generator design must be thoroughly analyzed from the standpoint of thermal expansion, thermal and pressure stresses, and thermal cyclic fatigue.

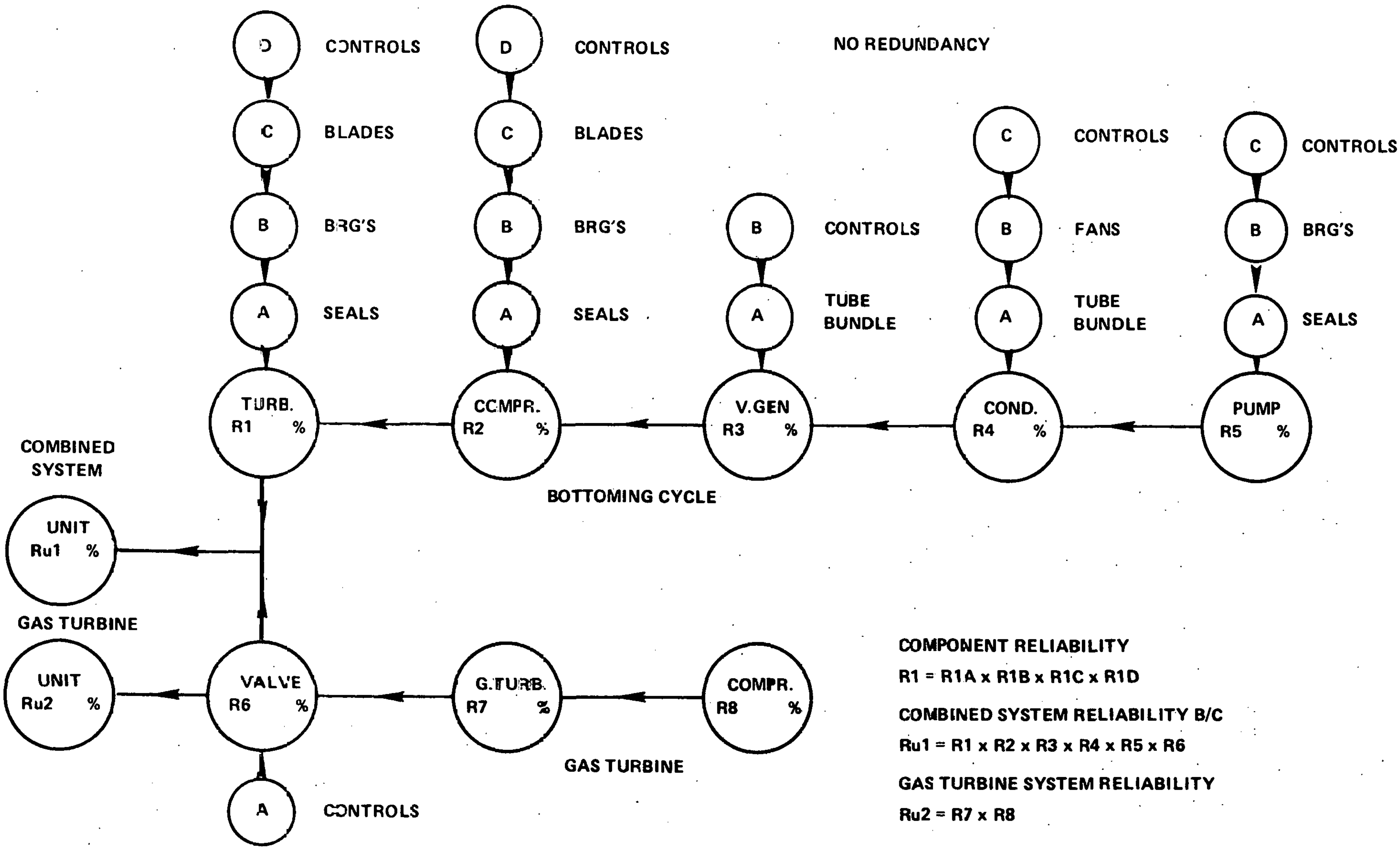

Figure 8-1. Reliability Diagram.

Quality control of materials and joining (welding) must surpass normal steam boiler construction practices. Provisions for normal and emergency draining of the working fluid from the unit to a holding tank must be provided. Problems resulting in forced outage will require longer downtimes for repairs than a normal steam system since a fluid such as toluene will require system draining and purging prior to making weld joint repairs, etc. Planned outages should include an annual boiler inspection by a licensed inspector.

Reliability obtainable from the condenser, compressor and/or generator is expected to be relatively high and no potential problems are foreseen with these components.

The organic vapor turbine with the possible exception of shaft seals is expected to have reliabilities equal to a steam turbine.

Other miscellaneous components such as the boiler feed pump, valving, piping, control components, etc., will be commercially available equipment. Special attention must, however, be paid to pump shaft seals, valve stem packing materials and gasketed joints to insure compatibility with the working fluid at the system operating temperature and pressure. High reliability in the area of system seals will be obtained by using the same type of pump shaft seal, valve stem packing and flange gaskets as used in refineries for toluene and by welding all pipe joints except those to be opened for maintenance.

As a part of the safety control system, sensors capable of detecting liquid or vapor leaks to the atmosphere will be necessary. This will be particularly true in the vapor generator and the sensors must be capable of withstanding the hot gas environment of this unit. Their reliability will be of prime importance to proper operation of the bottoming cycle system.

The application of a bottoming cycle to generate electrical power for use on a utility grid poses potential problems with the reliability of the electrical grid. The ability of the bottoming cycle to generate or not generate power would not be expected to create any major problem to the utility. However, the failure of protective electrical switchgear to remove the bottoming cycle electrical system from the grid in the event of short circuits or other failures which could overload sections of the utility grid will be of concern to the utility. The necessary controls are available on the commercial market; however, design must emphasize reliability from a control component and system standpoint.

Because of the size of the pipeline bottoming cycle system, the first experimental verification of potential reliability problems will be obtained in the demonstration tests. However, some applicable data is expected from tests of the three Rankine bottoming cycle systems being sponsored by DOE, especially the Sundstrand unit because it uses toluene as the working fluid. These three systems, however, are an order of magnitude smaller in size. To supplement the availability data of the three small units, a fault and criticality analysis will be a part of the final design. An estimate of the bottoming cycle component availabilities based on planned and forced outages which appear reasonable when compared to other similar types of Rankine cycle systems is indicated in Table 8-7. Availabilities for the bottoming cycle system and the combined systems are also indicated.

8.5 MAINTENANCE REQUIREMENTS

The estimated general maintenance requirements for the bottoming cycle system are indicated in Table 8-8. The required maintenance times and periods were based on comparisons of bottoming cycle components with similar equipment utilized in gas compressor stations. These requirements do not include the maintenance normally performed on the gas turbine or gas compressor.

The gas turbine/gas compressor maintenance depends to a large extent on the gas turbine utilized. Aircraft derivative units are removed and replaced as a unit, major maintenance work is conducted at an overhaul depot. A spare unit is kept at or near the compressor station so that replacement can be made quickly. Removal and replacement is normally made in 8-16 hours. The industrial gas turbine units, however, are inspected and overhauled in the field. These units, therefore, require planned outages for inspections and repairs. Major inspections are normally conducted at 30-40,000 hour intervals and require planned outages of 3-4 weeks duration, (Reference 45). The gas compressor requires considerably less maintenance than the gas turbine and inspections and maintenance are performed during gas turbine changeover or planned outages.

TABLE 8-7

ESTIMATED PIPELINE BOTTOMING CYCLE AVAILABILITY FOR MATURE PRODUCT

Cause	Forced Outage Hrs.		Planned Outage Hrs.		Maint. Outage Hrs.		Operating Availability	
	FFP[(a)]	PLBC[(b)]	FFP[(a)]	PLBC[(b)]	FFP[(a)]	PLBC[(b)]	FFP[(a)]	PLBC[(b)]
Turbine	97	72	417	168	147	100	92.3	96.1
Vapor Generator	298	200	519	336	247	123	87.6	92.4
Condenser	7	25	238	168	74	74	96.3	96.9
Generator (Compressor)	22	25	299	243	82	82	95.3	96.0
Other	34	30	164	133	78	78	96.8	97.2
Unit	458	352	603	336	290	210	84.4	89.7

ESTIMATED AVAILABILITY OF GAS TURBINE PLUS BOTTOMING CYCLE

	Gas Turbine Availability	Bottoming Cycle Availability	Combined Availability
Aircraft Gas Turbine, %	99.0	89.7	88.7
Industrial Gas Turbine, %	88.7	89.7	78.4

(a) Fossil fired plant for generating electricity with a steam turbine. 1976 EEI data for 200-389 MW plants.

(b) Pipeline bottoming cycle estimates.

TABLE 8-8

SUMMARY OF MAINTENANCE REQUIREMENTS FOR BOTTOMING CYCLES

VAPOR GENERATOR	Annual inspection of tube bundle by state licensed inspector, 1 week
VAPOR TURBINE	Daily routine checks, lube systems, bearing temperatures, seals, vibration sensors, controls, etc., 1 hour/day. Annual 8000 hour disassembly and inspection, replacement of seals, etc., 1 week
CONDENSER FIN-FAN UNIT	Daily routine checks of fan motors, gear boxes, filter screens, etc., 1/2 hour/day. Annual inspection fan blades tube bundle, etc., 2 days
VAPOR GENERATOR FEED PUMP	Daily routine check. Shaft seal, drive motor, pressure and flow annual inspection impellor, seal replacement, etc., 2 days
TURBINE GEARBOX AND DRIVE TRAIN	Daily routine check lube system, bearing temperatures and vib sensors, 1/2 hour/day. Annual inspection gear teeth, shafts, bearings, 3 days
DIVERTER VALVE	Daily routine check of actuation system components, periodic system actuation
FIRE PROTECTION SYSTEMS	Daily routine inspections, 1 hour/day; periodic system actuation as prescribed by insuring agency, 1 hour/6 months

The bottoming cycle system working fluid will have a major influence on maintenance and repair procedures particularly where the working fluid system must be opened to the atmosphere to effect repairs. Because toluene is flammable and somewhat toxic, the portion of the system to be worked on will be cooled down, drained, isolated via valving from the rest of the system, and purged with nitrogen before repairs are initiated. Careful planning will assure that the method of draining the toluene into a storage tank and the method of purging the system will result in the minimum release of contamination to the atmosphere.

8.5.1 VARIATIONS IN MAINTENANCE PROCEDURES

The bottoming cycle system coupled to a gas turbine/gas compressor will, of course, require more maintenance than the prime system by itself. Since the aircraft derivative gas turbines require very little maintenance, installation of a bottoming cycle at a site which has only aircraft derivative gas turbines can mean an appreciable increase in onsite maintenance for the combined system. If the compressor station has industrial gas turbines and/or reciprocators, in addition to the aircraft derivative gas turbines, the site maintenance work force will probably be larger and the additional maintenance work required by the bottoming cycle system can be offset by shutting down less efficient turbines or reciprocators and applying the maintenance work normally required on these units to the bottoming cycle system. With regard to the types of maintenance skills required, the commercially available equipment, upon which the bottoming cycle system design is predicated, should ensure that no new maintenance craft skills are required. Certain bottoming cycle equipment will not be familiar without training for the maintenance personnel; however, servicing techniques will be acquired with training. The procedures and safety precautions required when working on the natural gas systems in their stations are similar to those that must be taken when working on the bottoming cycle system due primarily to the flammability of the fluid. Some variations in maintenance and safety procedures, however, will be required since the bottoming cycle system working fluid is a liquid in some portions of the system and a vapor in others. This is not expected to be a severe problem; however, safety training programs should be conducted on this subject.

The bottoming cycle system will require at least one planned outage for approximately one week each year during which time the vaporizer will be inspected. Gas turbine inspection and general annual inspections and repairs of all other components of the bottoming cycle system can be made at this time. During this time the gas turbine/gas compressor can also be shutdown and planned maintenance work conducted or the gas turbine may be allowed to operate with the diverter valve in the bypass position. Since planned outages for the aircraft derivative gas turbines will be of short duration, the diverter valve would permit the gas turbine to operate before the bottoming system is ready to start. Although there may be some differences in maintenance procedures between the complete system and the gas turbine alone, primarily due to the toluene in the bottoming cycle system, more maintenance work will be necessary. It is not expected that the difference will be large.

Generally speaking, the acceptability of the bottoming cycle from a maintenance standpoint will be largely dependent upon the acceptance of toluene as the working fluid by the industry. The fact that air pollution control laws have placed definite limits (Reference 46) on the hydrocarbon vapors which can be discharged to the atmosphere will place an extra load on the maintenance department when a system containing toluene or other organic fluid is installed.

8.6 CONCLUSIONS

A bottoming cycle applied to a pipeline gas turbine driving a gas compressor can be designed, constructed and operated in a manner which provides high reliability and a minimum of maintenance. The type of gas turbine to which the bottoming cycle is applied (aero-derivative or land-based unit), the detail design of the bottoming cycle components and system, and the working fluid utilized are major factors in determining the reliability and maintainability of the combined cycle. The combined cycle is indirectly less reliable than the prime system and more maintenance problems can be expected, resulting in a lower system availability. The reliability of the diverter valve is critical to maintaining gas turbine/compressor reliabilities at their current levels. Since steam bottoming cycle equipment for the most part is commercially available and, to some extent, similar to equipment now employed on pipeline pumping stations, no new maintenance skills appear to be required. Depending on the pumping station to which the bottoming cycle is applied and its method of operation, maintenance work may increase or stay approximately the same.

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SECTION 9

INDUSTRIAL POTENTIAL ASSESSMENT

9.1 INTRODUCTION

The technological feasibility section concluded that the bottoming cycle is a technologically feasible system and components such as heat exchangers, pumps and expanders could be obtained from existing manufacturers. The economic assessment concluded that the bottoming cycle is a competitive investment alternative for certain applications.

The purpose of the industrial potential assessment is:

1. To identify the potential market for bottoming cycles on gas transmission pipelines.
2. To provide a projection to the year 2000 of bottoming cycle use within the industry.
3. To project the potential energy savings from the use of bottoming cycles in the pipeline industry by the year 2000.

In this assessment estimates are made of the potential gas saving that could be realized through the year 2000 by adoption of bottoming cycles. The estimated potential gas saving depends upon the projections of gas prices, power requirements, and gas supply. The estimates are made for two natural gas supply projections. For the low supply projection the estimated throughput in the year 2000 does not exceed the present capacity of the system. Two cases were considered. In the first case no additional power was installed. In the second case 50% of the difference between the installed power and that actually used in 1978 was installed on new sites, assuming that some new equipment will be required to reach new gas supplies.

For the high supply projection, it was estimated that additional pumping power of 3.1×10^6 HP would be required between 1983 and 2000. Two growth options were considered; one in which all the growth is on new sites and the other in which half of the growth is on existing sites.

9.2 GAS SUPPLY PROJECTIONS

Predicting the future supply of natural gas is not easy since the transport of natural gas has been a no growth business for several years and some sources predict little or no growth for the future. However, some trade publications indicate increased activity in the natural gas industry, as illustrated by the examples in Table 9-1. The American Gas Association has made two projections for the natural gas industry, based on a low-supply scenario, and a high-supply scenario (Reference 47). The assumptions used for its projections are given in Table 9-2. A summary of the data on present gas production and projections for the year 2000 are shown in Table 9-3, and were obtained from the sources listed. The values that were used for this study are shown at the right hand side, and are 19.2 Tcf/yr (10^{12} ft^3/yr) for 1978 gas production, 23 Tcf/yr for 1978 system capacity and 32.5 Tcf/yr for the high supply scenario by the year 2000.

The annual natural gas low-supply projection used in the present study is shown in Figure 9-1 and was extrapolated to 22.9 Tcf/yr by the year 2000 using the data of Reference 48. For this projection the supply never exceeds the present system capacity of 23 Tcf/yr. The annual high-supply projection used in the present study is in Figure 9-2 and was extrapolated to 32.5 Tcf/yr by the year 2000, using the data of Reference 49. This projection exceeds the present system capacity in 1983.

9.3 POWER REQUIREMENTS

The relationship between pipeline throughput and installed power is shown in Figure 9-3. The points shown at the left side of the plot are based on data for 1955 through 1973, taken from the listed reference. From previous surveys of the pipeline industry described in Section 3, the amount of installed power was estimated at 13 x 10^6 HP and from the previous subsection the throughput capacity was estimated at 23 Tcf/yr. From the intersection of these two values the relationship of throughput to installed power was extrapolated to 32.5 Tcf/yr using the slope of the relationship from 1962 to 1970. The power requirement of 16.1 x 10^6 HP in the year 2000 indicates that additional power of 3.1 x 10^6 HP will need to be installed from 1983 to 2000.

TABLE 9-1

INDICATIONS OF INCREASED ACTIVITY IN THE NATURAL GAS INDUSTRY

*(50)	1979 US Gas Pipeline Mileage Let by Diameter	6664	
(50)	US Gas Pipeline Mileage Let by Unidentified Diameters	266	
(50)	Total	6930	$3,750,000,000

(51)	Projected 1979 US Gas Pipeline Completion, Miles	3027
(51)	1979 Increases in All Pipelines Over 1978, %	44

(52)	Fraction of New Wildcat Completions of Gas and Oil Well Exploration:	1960's	10%
		1978	16.5%

(52) "The Number of Gas Well Completions Has Never Been Higher. . ."

(47) "To Transport Present and Future Discoveries to Market, Two Fully Looped Pipeline Systems are Necessary," (By the year 2000) (Looping Refers to Putting in Parallel Pipelines all Pumped with the same Gas Compressors to Increase Capacity by Lowering Pressure Drop Between Stations)

- Present to 1990 First Looped Alaskan Pipeline
- Similar Project 1993 to 2000

* Numbers in parantheses designate references at the end of this report

TABLE 9-2

AGA ECONOMIC DEMAND MAJOR ASSUMPTIONS*

- Demand Based on Price Competition Among Fuels
- Inflation 5.5% Per Year, 1978 to 1990
- Oil Price Inflation 7% Per Year, 1978 to 1990
- No Gas Use in Entirely New Gas Markets, Before 1990
- Gas Prices at Wellhead Simulated According to NG Pricing Act, Deregulated in 1985
- Low Supply Scenario: Conventional Natural Gas Production Plus Supplemental Supplies from the Following:
  - Canada
  - Mexico
  - SNG
  - LNG
- High Supply Scenario: Same as Low Supply Scenario Plus Supplement Supplies from the Following:
  - Coal Gasification
  - New Technologies

    Devonian Shale

    Tight Sands

    Biomass

    Peat Gasification

    Geopressured Gas

    Gas from Coal Seams

    Gas from Biological Waste

- Alaska

*Reference 48

TABLE 9-3

AGA NATURAL GAS SUPPLY PROJECTION DATA

ITEM	SOURCES*	DATA	VALUE USED
1978 Gas Production:	(53)	20 to 20.8 Tcf/yr	
	(49)	19.2 Tcf/yr	19.2
	(47)	20 Tcf/yr	
	(48)	20.4 Tcf/yr	
Present System Capacity:	(49)	23 Tcf/yr	23
Supplemental Production from New Technologies by 2000:	(53)	12 to 15 Tcf/yr	
	(52)	12.1 Tcf/yr	
Supply by 2000:	(53)	32 to 36 Tcf/yr	
	(49)	32.5 Tcf/yr	32.5
	(47)	30 to 36 Tcf/yr	
	(52)	31 to 35 Tcf/yr	

* Numbers in parentheses designate reference at the end of this report.

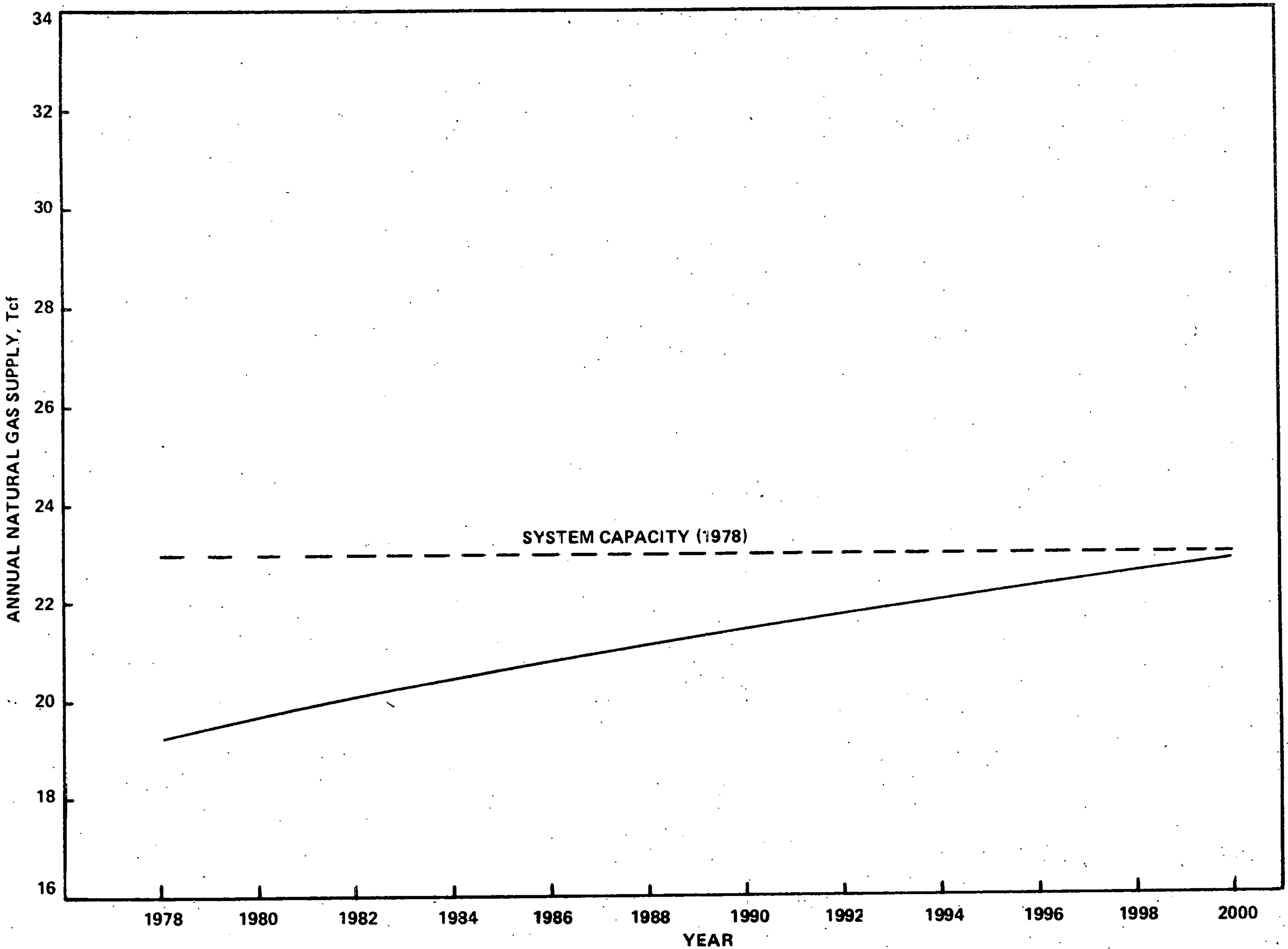

Figure 9-1. Total U.S. Natural Gas Low Supply Projection.

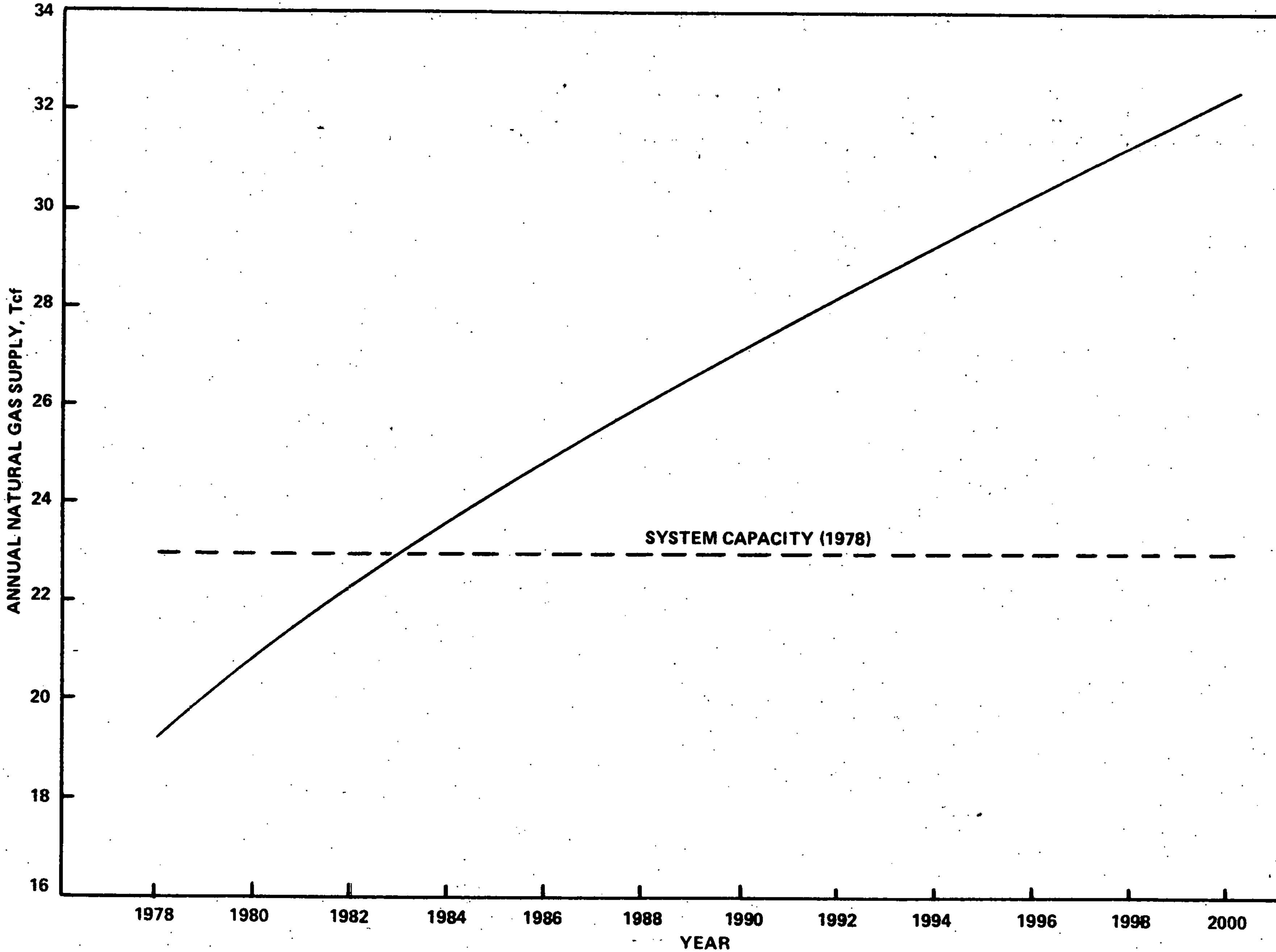

Figure 9-2. Total U.S. Natural Gas High Supply Projection.

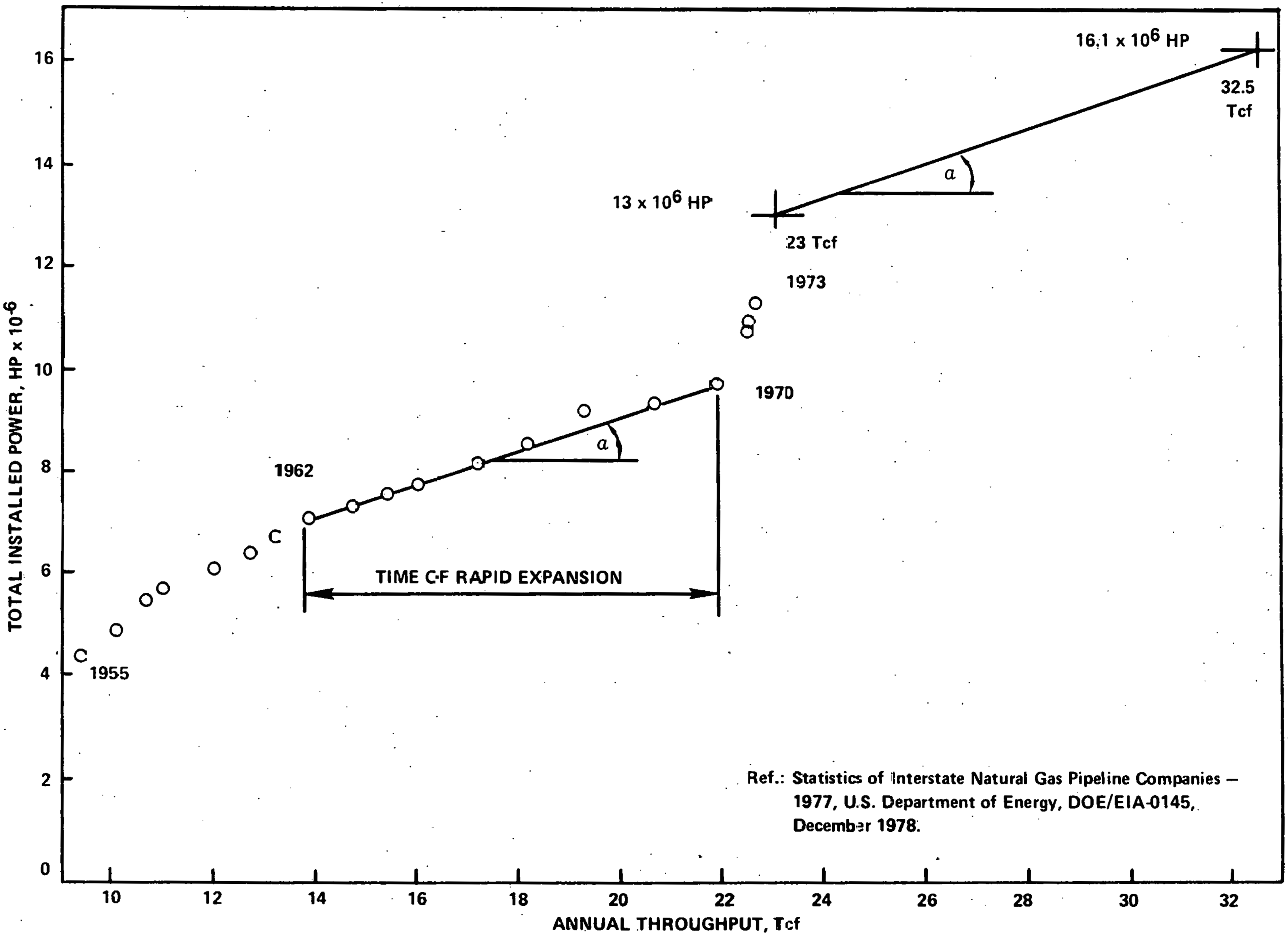

Figure 9-3. Assumed Relation Between Throughput and Installed Power.

9.4 OPTIONS FOR LOW-SUPPLY PROJECTIONS

For the low-supply projection it is estimated that there presently is about 4.7×10^6 HP not being used because of lower than capacity throughput. This is based on an estimated throughput of 19.8 Tcf compared to a capacity of 23 Tcf. In the first case it is assumed that no additional installed power is required. However, gas can be conserved by installing bottoming cycles on the existing powerplants and operating the powerplants at a reduced power setting at which the power of the combined system equals the power of the gas turbines or reciprocating engine without the bottoming cycle. For the second case it is assumed that one half the presently unused 4.7×10^6 HP is supplied by building new sites and using equipment as described for the high supply projection in which 50% of growth is on new sites and 50% of growth is on existing sites.

The options available in the low-supply projection with no additional power (the first case) are to bottom the existing powerplants, namely, reciprocating engines, recuperated gas turbines or simple-cycle gas turbines. Using a computer code that was developed for the economic assessment, incremental cost of service calculations were made for the three options. This code is described in the Economic Assessment Section and the main features of the code are presented again in Table 9-4. Important inputs to the code are the capital cost of the bottoming cycle, the amount of gas saved by the bottoming cycle and the value of the gas saved.

The capital costs of the bottoming cycles were estimated by scaling the cost of one of the systems described in the preliminary design Section 4. For the reciprocating engines the average site power was estimated to be 8000 HP and a bottoming cycle system would be about 1120 HP at full load or 985 for the throttled application. The cost of a 985 HP bottoming cycle system was scaled from the cost of the larger system using the power law with an exponent of 0.6. The cost is $\$1.04 \times 10^6$. Similarly the size of the recuperated gas turbine was taken as 8640 HP which is a typical size for the recuperated gas turbines on the sites examined in Section 4. The bottoming cycle power for the throttled condition for this application was estimated to be 1870 HP. The capital cost is $\$1.53 \times 10^6$. For the simple cycle gas turbine the typical size was

TABLE 9-4

MAJOR ELEMENTS IN COST OF SERVICE CALCULATIONS

Cost of Service =	Return on Rate Base
	+ Operating & Maintenance Expenses (Including Fuel Used)
	+ Depreciation (Financial)
	+ Property Taxes
	+ Federal Taxes
	+ State Taxes

- For Rankine Bottoming Cycle, the cost of service is balanced by the fuel saving or revenue from sale of electricity.
- If the savings or revenue is less than the cost of service, the customer must pay more for gas.
- If the savings or revenue is greater than the cost of service, the customer may get a rate reduction.
- In all cases, the company earns its allowed rate of return.

taken to be 12500 HP, based on the site selection study, Section 3, and the bottoming cycle power for the throttled condition was estimated to be 3050 HP. The capital cost was estimated to be $\$2.05 \times 10^6$.

The amount of gas saved per year by the bottoming cycle was estimated from the difference in heat rate of the prime mover, with and without the bottoming cycle, the total power, and the hours of operation per year. The data are tabulated in Table 9-5. The value of gas saved was taken as new gas prices, as described in the Economic Assessment Section. The values used are plotted in Figure 9-4. The escalation of fuel values was taken as 8%.

The average incremental cost of service was calculated over a five year period, and the installation date was varied from 1980 to 1995, with capital costs escalated 8% per year from 1978 prices. The incremental cost of service is the difference between the actual cost of owning and operating new equipment and the savings due to the new equipment. A positive value means that the cost is greater than the savings, while a negative value means that the savings are greater than the cost. The average five year incremental cost of service for a system installed in 1990 (near the middle of the time span between now and the end of the century) was calculated to be $178,000 for a bottoming cycle on a reciprocating engine, $231,000 for a bottoming cycle on a recuperated gas turbine, and $319,000 for a bottoming cycle on a simple cycle gas turbine. For both the reciprocating engine and the recuperated gas turbine the cost of owning and operating the bottoming cycle is positive, i.e., greater than the value of the gas saved, but for the simple cycle gas turbine the value is negative, i.e., the value of the gas saved is greater than the cost. Therefore bottoming the simple-cycle gas turbine is the only option that is economically attractive for the low-supply projection when no additional power is installed.

9.5 OPTIONS FOR HIGH SUPPLY PROJECTION

For the high supply projection it was estimated that additional power of 3.1×10^6 HP would be required between 1983 and 2000. This additional power could be obtained by purchasing new gas turbines or reciprocating engines or by adding bottoming cycles to new or existing equipment. Cost

TABLE 9-5

GAS SAVED WITH BOTTOMING CYCLE

	Heat Rate Btu/HP Hr	Heat Rate With B.C.* Btu/Hp-Hr	Power HP	Hrs. Operated Hrs./Year	Gas Saved Ft^3/Year
Recip.** Engine	7400	6490	8000	7446	5.237×10^7
S.C.*** Gas Turbine	10060	7860	12500	7446	1.978×10^8
Recup.**** Gas Turbine	8750	7440	8640	7446	8.143×10^7

*Bottoming Cycle

**Reciprocating

***Simple Cycle

****Recuperated

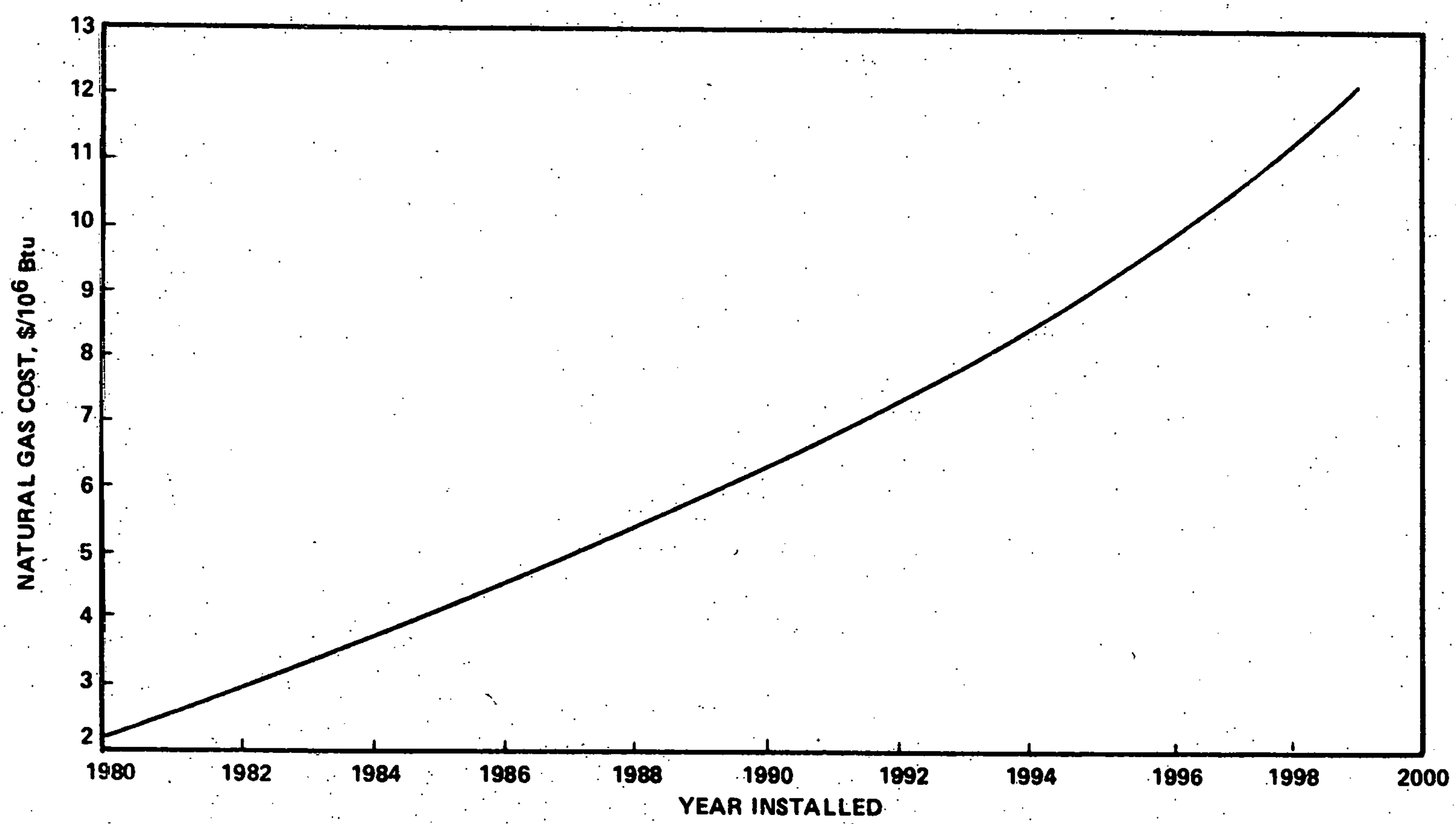

Figure 9-4. Estimated New Gas Prices.

of service calculations were made for the various options using the computer code described previously. The results depend upon typical sizes selected, the capital costs and the number of hours of operation each year. These data are presented in Table 9-6. Since the power generated is in addition to the existing power, the "fuel saved" input value is the gas consumed by the system to generate the additional power with a negative sign because it is gas used and not saved.

For each option a typical size was selected and a capital cost estimated. These are shown in Table 9-6. Then the cost of service results were normalized to a basis of dollars per thousand cubic feet of gas pumped. In this calculation typical efficiency values were used for the gas turbine driven centrifugal compressors and reciprocating engine driven reciprocating compressors (viz. 0.83 and 0.75, respectively). The results are shown in Table 9-7 for seven options. These results indicate that for each type prime mover it is more economic to bottom existing systems than to buy all new equipment. The selection of the type of equipment to use for growth power on any one pipeline depends upon duty cycles expected, amount of power required, off design characteristics, type of existing equipment at a site, etc. and on the economics of various options.

Two growth options were considered; one in which all of the growth is on new sites and the other in which half of the growth is on existing sites. Undoubtedly, a substantial portion of the growth will come about by laying new pipelines to or from the existing pipelines. The increased capacity of existing pipelines can be obtained by adding more installed power or by laying new pipe in parallel to existing pipe without increasing installed power (looping).

For the assumption of all of the growth on new sites the amount of power for each option was selected in proportion to the inverse of the cost-of-service, and is shown in Table 9-8. Although the total additional power is 3,100,000 HP, the amount that is provided by bottoming cycles is 293,800 HP.

For the assumption of 50% of the growth on existing sites all of the options were used, the amount of power for each option is proportional to the inverse of the cost of service shown in Table 9-7. The amount of power for each option for 50% growth on existing sites is shown in Table 9-9.

TABLE 9-6

COST OF OPTIONS FOR GROWTH POWER

	Size HP	Cost $ x 10^{-6} (1978)	Hrs. Operated Hrs./Year
New Gas Turbine	13000	4.927	7446
New Gas Turbine with B.C.	16770	7.261	7446
New Recip. Engine	8000	4.84	7446
New Recip. Engine with B.C.	9120	5.955	7446
Existing S.C. GT with B.C.	5625*	2.959**	***
Existing Recip. Eng. with B.C.	1120*	1.12**	***
Existing Recup. GT with B.C.	2560*	1.846**	***

* Size of Bottoming Cycle

** Cost of Bottoming Cycle

*** These options do not use additional fuel, ∴ hrs. of operation do not apply.

TABLE 9-7

COST OF SERVICE

Options for Growth	Cost of Service \$/1000 Ft^3
Existing S.C. Gas Turbine with B.C.	0.0128
Existing Recup. G.T. with B.C.	0.0175
New Gas Turbine with B.C.	0.024
New Gas Turbine	0.0266
Existing Recip. Engine with B.C.	0.0269
New Recip. Engine with B.C.	0.0345
New Reciprocating Engine	0.0355

TABLE 9-8

DISTRIBUTION OF GROWTH POWER AMONG OPTIONS

ALL GROWTH ON NEW SITES

Options	Total HP	Bottoming Cycle HP
New Gas Turbines with B.C.	947,000	212,900
New Gas Turbines	854,000	
New Reciprocating Engines with B.C.	659,000	80,900
New Reciprocating Engines	640,000	
	3,100,000	293,800

TABLE 9-9

DISTRIBUTION OF GROWTH POWER AMONG OPTIONS

50% OF GROWTH ON EXISTING SITES

Options	Total HP	Bottoming Cycle HP
Existing S.C. Gas Turbines with B.C.	391,900	391,900
Existing Recup. G.T. with B.C.	286,500	286,500
New Gas Turbines with B.C.	209,200	47,000
New Gas Turbines	188,700	
Existing Recip. Engines with B.C.	186,700	186,700
New Recip. Engines with B.C.	145,500	17,900
New Recip. Engines	141,500	
	1,550,000	930,000

Although the total power is 1,550,000 HP the fraction that is bottoming cycles is 930,000 HP. When existing systems are bottomed all of the additional power is assumed to be from the bottoming cycles. Therefore the bottoming cycle market is much larger for this option. The other 50% of the growth power on new sites is half of the values shown in Table 9-8.

9.6 POTENTIAL CONSERVATION OF GAS

For purposes of calculating the maximum gas saving from the adoption of bottoming cycles, it was assumed that bottoming cycles would be installed beginning in 1983, after completion of the demonstration program, and that all of the existing simple-cycle gas turbines would be bottomed in twenty years or by 2003 (100% penetration). After that time consideration could be given to replacement of equipment. For the low-supply projection gas saved was calculated for two assumptions: 1. with no additional installed power and 2. newly installed power by the year 2000 being 50% of the difference between presently installed power and presently needed power. For the low-supply projection with assumption 1, no additional installed power, it was concluded from the incremental cost-of-service calculations that it was economically attractive to bottom the simple-cycle gas turbines only. It was estimated that there are about 2.6×10^6 HP from simple-cycle gas turbines on existing gas pipelines. If these were all bottomed at reduced power the bottoming cycle power was calculated to be 632,000 HP. The gas saving was calculated as the difference in heat rate between the bottomed and the unbottomed systems multiplied by the system horsepower and the hours of operation per year which was taken as 85% of the maximum or 7446 hours per year. The total cumulative gas saved through the year 2000 was calculated to be 51×10^6 BOE (barrels of oil equivalent) or 0.296 trillion cubic feet of natural gas. In 1978 approximately 20 trillion cubic feet were pumped on U.S. domestic pipelines. Table 9-10 shows the potential gas saving for the present century by use of bottoming cycles under two options.

For the low-supply projection with assumption 2, 50% of the difference between presently installed power (13×10^6 HP) and presently used power (8.3×10^6 HP) installed as new power by the year 2000, the total cumulative fuel saving is 70×10^6 BOE (or 0.406 Tcf). The full 51×10^6 BOE (or 0.296 Tcf) is saved because conservation on existing sites is not af-

TABLE 9-10

POTENTIAL GAS SAVED BY USE OF BOTTOMING CYCLES*

PROJECTION	CUMULATIVE GAS SAVED THROUGH 2000 10^6 BOE	TCF
LOW-SUPPLY		
CONSERVATION ON EXISTING SITES	51	0.296
	51	0.296
50% OF DIFFERENTIAL BETWEEN 1978 INSTALLED AND ACTUAL POWER NEW SITES	19	0.110
CONSERVATION ON EXISTING SITES	51	0.296
	70	0.406
HIGH-SUPPLY		
GROWTH ON NEW SITES	25	0.145
CONSERVATION ON EXISTING SITES	51	0.296
	76	0.441
50% GROWTH ON EXISTING SITES	82	0.476
50% GROWTH ON NEW SITES	12.5	0.072
CONSERVATION ON EXISTING SITES	32	0.186
	126.5	0.734

* All existing simple-cycle gas turbines assumed to be bottomed in twenty years beginning in 1983.

fected by the growth on new sites. In addition the newly installed power saves 19 x 10^6 BOE (0.110 Tcf) in a way similar to that explained for the high-supply projection, 50% of growth on new sites discussed below.

For the high-supply projection gas saved was calculated for two assumptions: 1. all of the growth on new sites, and 2. half of the growth on existing sites. For the high-supply projection with assumption 1, all of the growth on new sites, the amount of each option used is shown in Table 9-8. The growth power was installed at a constant rate from 1983 to the year 2000. The gas saved by the bottoming cycles was calculated as the difference in heat rate between a bottomed and an unbottomed system multiplied by the power and the hours of operation per year. For these assumptions the gas saved through the year 2000 was calculated to be 25 x 10^6 BOE (barrels of oil equivalent) or 0.145 trillion cubic feet. The existing simple-cycle gas turbines could also be bottomed at reduced power to achieve an additional gas saving of 51 x 10^6 BOE, for a total gas saving of 76 x 10^6 BOE or 0.441 trillion cubic feet.

For the high-supply projection with assumption 2, half of the growth is on existing sites, the amount of each option used is shown in Table 9-9. Because the amount of bottoming cycle power is much larger than for the other projections, it was assumed that one fourth of the desired yearly capacity could be installed in the first year, one half could be installed in the second year and the full capacity in the third and subsequent years. For these assumptions the gas saved through the year 2000 was calculated to be 82 x 10^6 BOE or 0.476 trillion cubic feet for the growth on existing sites. For growth on new sites the fuel saving is half of the value when all of the growth was assumed to be on new sites or 12.5 x 10^6 BOE or 0.072 trillion cubic feet. Although the growth on existing sites used some of the simple cycle gas turbines to be bottomed at full power, there are about 62% of the simple cycle gas turbines that can be bottomed at reduced power to save about 32 x 10^6 BOE or 0.186 trillion cubic feet. Therefore the total potential gas saving through the year 2000 for this projection is 126.5 x 10^6 BOE or 0.734 trillion cubic feet. This is equivalent to 0.734 x 10^{15} BTU or 0.734 quads.

During the remainder of this century while bottoming cycles are being applied the rate of fuel saving increases rapidly. Shown in Table 9-11 are estimates of the fuel saving rates potentially available at the

TABLE 9-11

POTENTIAL FUEL SAVING RATE IN THE YEAR 2000*

	RATE OF FUEL SAVING	
PROJECTION	BPDE	TCF/Y
LOW-SUPPLY		
CONSERVATION ON EXISTING SITES	16900	0.0357
	16900	0.0357
50% (INSTALLED HP 1978 ACTUAL HP) NEW	5800	0.0123
CONSERVATION ON EXISTING SITES	16900	0.0357
	22700	0.0480
HIGH-SUPPLY		
GROWTH ON NEW SITES	7600	0.0162
CONSERVATION ON EXISTING SITES	16900	0.0357
	24500	0.0519
50% GROWTH ON EXISTING SITES	26700	0.0565
50% GROWTH ON NEW SITES	3800	0.0081
CONSERVATION ON EXISTING SITES	10500	0.0223
	41000	0.0869

* Assuming capture of entire economically viable bottoming cycle market.

end of the present century when most of the bottoming cycle systems are assumed to be installed. Based upon the historical throughput values from Reference 48 and typical fuel usage by the gas pipelines in Reference 55 projected gas pipeline fuel consumption values at the end of the century for the low- and high-supply projection were estimated for the case without the bottoming cycle. Using these values of projected fuel consumption the rate of fuel saving at the end of the century for the low-supply projection is estimated to lie between 4.8 and 6.4%. For the high-supply projection the fuel savings at the end of the century will lie between 4.9 and 8.2%. Reference to Table 9-10 indicates that for all cases studied putting bottoming cycles on existing sites saves the most gas. Except for the case of 50% growth on existing sites and 50% growth on new sites conservation on existing site is the largest part of the gas saving. For the other three cases very little gas is saved compared to the potential on existing sites because the only economically viable retrofit bottoming cycle is on simple-cycle gas turbines, accounting for only installed prime mover power of 2.6×10^6 HP compared to a total of 13×10^6 HP. If bottoming cycles could be put on the remaining 10.4×10^6 HP of existing prime movers economically, considerably more gas could be saved.

9.7 POTENTIAL BOTTOMING CYCLE MARKET

The potential bottoming cycle market through the year 2000 is 530,000 HP for the low-supply projection, assuming that 632,000 HP could be installed in twenty years beginning in 1983 and no addition in installed power. If additional power is installed the value becomes 753,000 HP. For the high-supply projection 294,000 HP could be installed for growth on new sites, for a total potential market of 824,000 HP of bottoming cycle power. The potential bottoming cycle market for the high-supply projection with half of the growth on existing sites is 856,000 HP growth on existing sites, 147,000 HP growth on new sites and 331,000 HP in the reduced-power mode on existing sites for a total potential market of 1,334,000 HP of bottoming cycle power. Table 9-12 illustrates the potential bottoming cycle markets under the two different supply projections and two sets of assumptions. The number of bottoming cycle units of various power levels varies over the approximate range from 170 to 500 for the supply projections studied.

TABLE 9-12

POTENTIAL MARKET FOR BOTTOMING CYCLES *

PROJECTION	BOTTOMING CYCLE POWER, H.P.
LOW-SUPPLY	
CONSERVATION ON EXISTING SITES	530,000
	530,000
50% OF DIFFERENTIAL BETWEEN 1978 INSTALLED AND ACTUAL POWER NEW SITES	223,000
CONSERVATION ON EXISTING SITES	530,000
	753,000
HIGH-SUPPLY	
GROWTH ON NEW SITES	294,000
CONSERVATION ON EXISTING SITES	530,000
	824,000
50% GROWTH ON EXISTING SITES	856,000
50% GROWTH ON NEW SITES	147,000
CONSERVATION ON EXISTING SITES	331,000
	1,334,000

* All existing simple-cycle gas turbines assumed to be bottomed in twenty years beginning in 1984.

9.8 CONCLUSIONS

The potential fuel (natural gas) saving by employing bottoming cycles on pipeline compressor prime movers was determined for the low- and high-supply projections, each with two separate cases. The following conclusions based upon the assumption discussed were obtained during the studies:

- For the low-supply projection for the case in which no additional power was added to the system (Assumption 1), 530,000 HP of bottoming cycle apparatus was installed, resulting in a cumulative saving of 51 x 10^6 barrels of oil equivalent (BOE) or 0.296 trillion cubic feet (Tcf) by 2000 A.D. and a rate of saving in 2000 A.D. of 16,900 barrels per day equivalent (BPDE) of 0.0357 trillion cubic feet/year (Tcf/y).

- For the low-supply projection for the case in which one-half the difference between installed and actually used power in 1978 was installed on new sites (Assumption 2), 753,000 HP of bottoming cycle apparatus was installed, resulting in a cumulative saving of 70 x 10^6 BOE or 0.406 Tcf by 2000 A.D. and a rate of saving in 2000 A.D. of 22,700 BPDE or 0.0480 Tcf/y.

- For the high-supply projection for the case in which all growth power was put on new sites (Assumption 1) 824,000 HP of bottoming cycle apparatus was installed, resulting in a cumulative saving of 76 x 10^6 BOE or 0.441 Tcf by 2000 A.D. and a rate of saving in 2000 A.D. of 24,500 BPDE or 0.0519 Tcf/y.

- The high-supply projection for the case in which 50% of the growth was on new sites and 50% on old sites (Assumption 2), 1,334,000 HP of bottoming cycle was installed, resulting in a cumulative saving of 126 x 10^6 BOE or 0.734 Tcf by 2000 A.D. and a rate of saving in 2000 A.D. of 41,000 BPDE or 0.0869 Tcf/y.

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SECTION 10

DEMONSTRATION PROGRAM PLAN

10.1 INTRODUCTION

The organic Rankine system, in sizes from 500 to 900 HP, is being developed by the Department of Energy to save energy. In an attempt to utilize this principle in larger sizes to save energy the Department of Energy has established a pipeline bottoming cycle research, development and demonstration program which strives to achieve maximum energy efficiency and to achieve optimized pipeline applications. Specific objectives of this program include the determination of the viability of the bottoming cycle system, the demonstration of that viability, and the measurement of the energy savings attainable from such systems.

So far in this program three sites have been selected as potential sites for demonstration of the pipeline bottoming cycle system as mentioned in Section 3. Following that, in Section 4, preliminary designs were made of pipeline bottoming cycle systems suitable for each of the three sites. The pipeline companies which are hosts to these sites then performed preliminary installation studies of the bottoming cycle equipment on each of their respective sites. Following the preliminary design a series of assessments were made concerning the pipeline bottoming cycle system. The bottoming cycle system was evaluated from an economic standpoint in Section 5, the environmental and safety aspects of the system were studied in Section 6, the technical feasibility of the pipeline bottoming cycle system was assessed in Section 7 and, finally, a study was made of the reliability and maintainability of the bottoming cycle system in Section 8. Following this the industrial potential for using pipeline bottoming systems was assessed in Section 9. The bottoming cycle which was studied throughout the program consisted of about a 5500 HP organic Rankine system (turbine efficiency,86%) driven from the waste heat provided by any one of three aircraft-derivative gas turbines having about 12,000 HP and approximately the same flow rate and exhaust gas temperature. It is expected that this turbine efficiency of 86% will be attained by the time the bottoming cycle goes into production. However, the efficiency of the demonstration system turbine, which will be adapted from steam turbine designs, will be 77%. In the preliminary design it was determined that reasonable designs could be made for all three sites that had been studied. However, because of the narrow range occupied by the exit temperatures from the three gas turbines studied and the small range of engine mass flows, one bottoming cycle design would be suitable for all three engines. In addition, the preliminary design indicated that it was feasible to make reasonable installations on the three sites. In these installations each site had a different means of using the bottoming cycle energy. In the one site the bottoming cycle drove a centrifugal compressor which was put in parallel with reciprocating compressors driven by reciprocating gas engines. In another site the bottoming cycle was made to drive the same gas compressor as was being driven by the gas turbine on which the bottoming cycle was placed. In the third instance, the bottoming cycle drove a generator which provided power for a nearby utility grid.

In carrying out the four assessments on the bottoming cycle studied in the preliminary design it was found that in some cases the bottoming cycle was economic now because of the value of natural gas on that particular pipeline and the investment criteria utilized to determine the viability of a new investment. It was also found that on another site the bottoming cycle would be viable in the mid-to-late 80's even though some reciprocating engines driving reciprocating gas compressors had to be shut down to provide a block of power that could be generated using the bottoming cycle system. No large impediments were discovered in studying safety and environmental aspects of the bottoming cycle except for the fact that the selected fluid, toluene, has a certain level of flammability and toxicity. The bottoming cycle was found to be technically feasible and the only design verification tests required were of the bearings and seals.

The bottoming cycle placed upon a gas turbine was found to reduce the availability of the combination below the very high value associated with aircraft-derivative gas turbines. However, it was felt that high availability could be attained after the apparatus had been in the field for awhile based on the availabilities of combined steam and gas turbine power plants.

Since the equipment which makes up the bottoming cycle was seen to be available, and since, in some instances studied, the bottoming cycle would be viable economically in the 1980's, there is a need for a demonstration bottoming cycle system to be built and demonstrated. The risk of utilizing the bottoming cycle system in the pipeline industry would be reduced if a demonstration under actual pipeline operating conditions were made. The support of the demonstration of the pipeline bottoming cycle system by the government would also reduce the risk to the manufacturer of the equipment to a level where it would be feasible to carry out. The bottoming cycle demonstration would make it possible for both the pipeline industry and the manufacturer to assess operating problems which might arise at the beginning of operation of the bottoming cycle system. In addition the operation of equipment would verify the economic viability of the system and prove out the predicted performance for the equipment. Also, operation of the equipment would establish whether environmental and safety regulations applicable to the site and generally applicable to the country could be observed in the operation of the bottoming cycle equipment. In addition, the operation of the bottoming cycle would determine the reliability of the equipment and it would provide information on the maintenance. Since a demonstration of the bottoming cycle equipment seems desirous and necessary to obtain useful information about the system, a plan is needed to carry through the bottoming cycle design, delineated in the preliminary design, to hardware development and an operational test.

In this section recommendations for the various phases of the demonstration program are made. Next, statements of work for each of the various steps in the program plan are provided. The various tasks which must be accomplished in order to carry out the demonstration are assembled in a reasonable schedule and the costs of the effort to carry out these tasks are estimated. Finally, recommendations are made as to the means for program participation by the manufacturer, the government and the pipeline industry.

10.2 RECOMMENDATIONS FOR DEMONSTRATION

The technical feasibility of the pipeline bottoming cycle has been established in Section 7. There are no barrier component design problems which must be solved before the bottoming cycle equipment could be made. Equipment similar to that for petro-chemical plants can be adapted to pipeline bottoming cycle heat exchangers and turbines, though the latter may require design verification testing of the bearings and seals. In addition to this, it has been seen in Section 4 that the bottoming cycle system can reduce the heat rate of the combined gas turbine and bottoming cycle systems considerably more than 20% at full throttle conditions for the gas turbine. This exceeds the target value that was set in the performance of the subject contract. The bottoming cycle was shown to have a reliability of approximately 90% which is somewhat lower than the gas turbine alone but is in the range that would be expected for combined gas and steam turbine powerplants.

The maintainability of the bottoming cycle equipment with gas turbine has been shown in Section 8 not to be excessive as compared to the gas turbine alone, although it is expected that some additional maintenance may be required. Therefore, it can be seen that the pipeline bottoming cycle system of approximately 5000-6000 HP is completely technically feasible at this time and, therefore, should be demonstrated to provide the pipeline industry with the kind of information that is required for them to make selections of the equipment for their own use.

Two of the three sites which were studied for the economic assessment indicated that the bottoming cycle as a system would be economically viable in the 1980's. One site has been shown to be economically viable at the present time pumping natural gas. Another site has been shown to be economically beneficial in generating electricity to be put on the company's electrical grid. There was one site which was not economically feasible at the present time. The reason for this is mainly that extra power was not required at the site, raising the fuel consumption some, and the price paid for the power was too low to make the system economically viable. Therefore, from the studies that have been made it can be seen that the pipeline bottoming cycle is economically viable now, in some instances. It should be more viable in other instances later in the coming decade. Since no bottoming cycle in the 5000-6000 HP range utilizing an organic fluid at a temperature of approximately 500°F has been demonstrated as part of a gas turbine bottoming cycle system, it is recommended that a demonstration be carried out on such a system to prove its economic viability. Although a demonstration system turbine efficiency of 77% is predicted, this value is commensurate with minimum changes in tooling. To minimize the cost of the demonstration apparatus, in full production the value of 86% will be attained. Thus, since no research and development is required it is recommended that the final design of the bottoming cycle system be started as soon as possible.

As is to be expected, economically the pipeline bottoming cycle system is sensitive to capital costs, the value of fuel and to the amount of utilization that the bottoming cycle will see in a given year. Thus, the economic analysis makes it clear that a site which pumps "new" gas, on a new pipeline and which needs all the power that can be generated would be the most economically viable choice for the bottoming cycle system. The pipeline industry is expected to expand in the near future according to References 47, 50, 51, and 52, opening the way for a number of additional pipeline companies to be considered for the demonstration site. This will open up a number of sites for which demonstration apparatus could be built. Although this may result in the demonstration being put on a new rather than existing site, the improved economic aspects stemming from pumping "new", higher-price gas, the need for all the power the bottoming cycle makes available and the elimination of retrofitting expense will ease the establishment of the demonstration site. Such a demonstration will benefit the pipeline industry in that operational experience under actual pipeline operating conditions will be obtained. The appropriate adjustments in the economics will have to be made to apply the results to retrofit applications. As a result it is recommended that a study be made of those pipelines and those sites which would have the best economics in light of the potential growth of the pipeline industry which now seems to be indicated by reports of activity in this industry.

In the studying of the economics of the bottoming cycle for use by the pipeline industry it has been found in Section 5 that a number of Federal Energy Regulatory Commission rules can affect the cost of service of a bottoming cycle system adversely. One such area is the subject of tax credits. According to the present law a tax credit may not be passed on to the utility's customers, and thereby reduce the cost of service but must go to the stockholders to stimulate investment. If a tax credit for conservation apparatus were permitted to reduce the cost of service but not be passed on to the customers, the return on investment would be higher for the conservation apparatus, making the investment more attractive to the investor. Since the conservation apparatus investment is small compared to the entire pipeline this action would not necessarily make the return on the entire pipeline investment exceed the limit thereby causing the pipeline to lower the utility rate. Not all pipelines are governed by the Federal Energy Regulatory Commission. Some, like the Pacific Gas and Electric Company, who have been contributing to the study, being an intrastate pipeline company are governed by the Public Utilities Commission of the state in which they operate. It is customary for the Public Utility Commission to permit any tax credit that is granted to go into the rate calculation. Thus, the economic viability of pipeline bottoming cycle systems can vary among companies depending upon which financial rule a company is required to follow. Another area which should be investigated is the value chargeable for the fuel that is used to drive the prime mover. For example, there are allowable prices for so-called "new" gas and "old" gas. The former commands a higher price than the latter. However, in evaluating economics of a pipeline which is operating on old gas, the saving of gas by means of a bottoming cycle system, in effect, is creating something that could be called "new" gas. It may be beneficial for the regulatory rules to be changed to permit this evaluation to be put on the gas that is saved since it would increase the economic viability of a pipeline bottoming cycle. Two additional items which should be studied more thoroughly as to their impact upon the economic viability of the pipeline bottoming cycle system are a faster tax depreciation and low cost loans. In the regulatory environment, there is now in place present government regulations which deter investment in conservation equipment by demanding unrealistically short payback periods. As a consequence of the adverse effects which some of the regulatory laws have on the economic evaluation of conservation equipment, it is recommended that an investigation of the regulatory impact on pipeline bottoming cycle applications be made. In the execution of the present contract, toluene has been selected as the working fluid for the organic bottoming cycle for several reasons. First of all, toluene gives about the highest performance of any of the organic fluids studied for gas turbine bottoming cycles, which are the subject of this contract. Some objections have been raised to the use of toluene because of its toxicity and flammability. However, no organic working fluid excels in each one of the six criteria used in fluid selection; namely,performance, thermal stability, flammability, toxicity, cost and compatibility with containment materials. Toluene was selected because in addition to its good performance it is well documented as a low-cost fluid compatible with materials of construction, it has acceptable levels of toxicity and flammability and it has a high enough thermal stability temperature. Of course, some of the pipelines feel that because of the flammability and toxicity of toluene some governing agency such as EPA or OSHA might remove toluene from use and thus make the bottoming cycle system designed to use toluene inoperable. This suggests that an investigation be made of thermal stability temperature, materials compatibility characteristics, fire-safety aspects, health and environmental aspects, cost, and performance of a number of organic fluids for possible use in organic systems. It is recommended that an evaluation of organic working fluids for a natural gas pipeline Rankine bottoming cycle installation be made early in the demonstration program of the bottoming cycle system.

At present many of the pipelines have installed apparatus in excess of present needs due to reduced throughput. The pipelines need to be able to install apparatus in excess of power requirements which will conserve fuel.

Since the selected working fluid has a certain level of flammability and toxicity it will be necessary to provide seals on the bottoming cycle system which will virtually keep any toluene from leaving the system. The sealing problem becomes much less difficult if toluene is also used for the bearing lubricant. All of the technology exists for sealing the toluene bearings and seals as has been demonstrated by the operation of a toluene system at Sandia Laboratories according to Reference 47. However, it is necessary to make sure that the actual design of the bearings and seals for the bottoming cycle system will be serviceable under the operating conditions which will exist for the subject bottoming cycle apparatus. These conditions for which the bearing and seal design must be verified include not only the thermodynamic conditions of the toluene but also the geometric configurations of the bearing and seal in relationship to the rotating speed of the shaft on which it will be placed. Not only the shaft seal will be verified but also the packing for the valve stems and the gasket material for flange joints. An additional operational problem which might occur in the subject design is that the seal coming from the turbine could operate at subatmospheric pressures which would subject the turbine and the system in which it is placed to the influx of air. Air as a noncondensible would have to be removed from the system with a vacuum pump in order to keep from blanketing the surface of the condenser and preventing the condensation of the toluene vapor. This situation of operating at subatmospheric pressure will occur if the rotating seal is made to protrude through the turbine case at the low pressure end. However, if it seems advisable to keep the seal operating at pressures above atmospheric it would be possible to locate the seal at the high pressure end of the turbine where the vapor pressure will exceed the ambient pressure by a large amount. Thus, because of these uncertainties it is deemed necessary to carry out a design verification test on the design for the bearing and seal which are selected in the final design for the pipeline bottoming cycle system.

Finally, it is recommended that the bottoming cycle be operated for approximately one year under actual pipeline operating conditions for a number of reasons. First of all, any seasonable effects of temperature could be observed by testing for this period of time. Also data is needed on the economic viability, and economics is very dependent upon the performance that the bottoming cycle can obtain. The operation of the bottoming cycle system is expected to provide useful information on maintenance requirements and reliability of the apparatus and to uncover any operational, safety or environmental problems.

10.3 STATEMENT OF WORK

The General Electric Company has planned and recommends a research, development and demonstration program for the DOE to undertake. A complete statement of work for the proposed demonstration program plan is given in Appendix C which includes the various tasks. Under the heading of "Directed Studies" in Phase II, the program plan addresses the issues of pipeline site determination, regulatory changes necessary for the acceptance of the bottoming cycle apparatus and their impacts, and evaluation of organic working fluids. The General Electric Company is already under contract at present to carry out the tasks of site determination and regulatory impacts study. The work statement also delineates various tasks and subtasks involved in the demonstration plant design and construction phase of the program, as well as the operation of the demonstration apparatus under actual pipeline operating conditions.

10.4 PROGRAM SCHEDULE

The schedule showing the milestones for Phases II, III and IV of the demonstration program for the pipeline bottoming cycle apparatus are shown in Figure 10-1. Phase II consists of pipeline site determination, regulatory impacts study and a study to evaluate the organic working fluids as described earlier. The General Electric Company,at present, is on contract with DOE to complete Task I and Task II of the Phase II section of the program. The program is funded by DOE to identify and describe the market segments for the application of the bottoming cycle and to analyze the impact of the selected economic/regulatory modifications on the possible installations of the bottoming cycle apparatus.

The program schedule shows June 1979 as the start date for Phase II and October 1979 for Phase III (demonstration plant design and construction) of the program. During Phase III, the major bottoming cycle components will be designed, fabricated and tested. The organic vapor gas turbine will be designed, developed and tested by the Mechanical Drive Turbine Division of GE. The vapor turbine and other components of the bottoming cycles will be scheduled so as to complete the design, fabrication and verification testing in a time period of two years. Phase III of the program will be completed by the first quarter of 1982 at which time the actual demonstration phase (Phase IV) will start. The complete pipeline bottoming cycle demonstration program shall be completed in the time frame of four years.

The tasks in Phases II,III and IV are further subdivided into subtasks and the schedules for the subtasks are given in Appendix D, showing the milestones for each subtask involved in the various phases of the program.

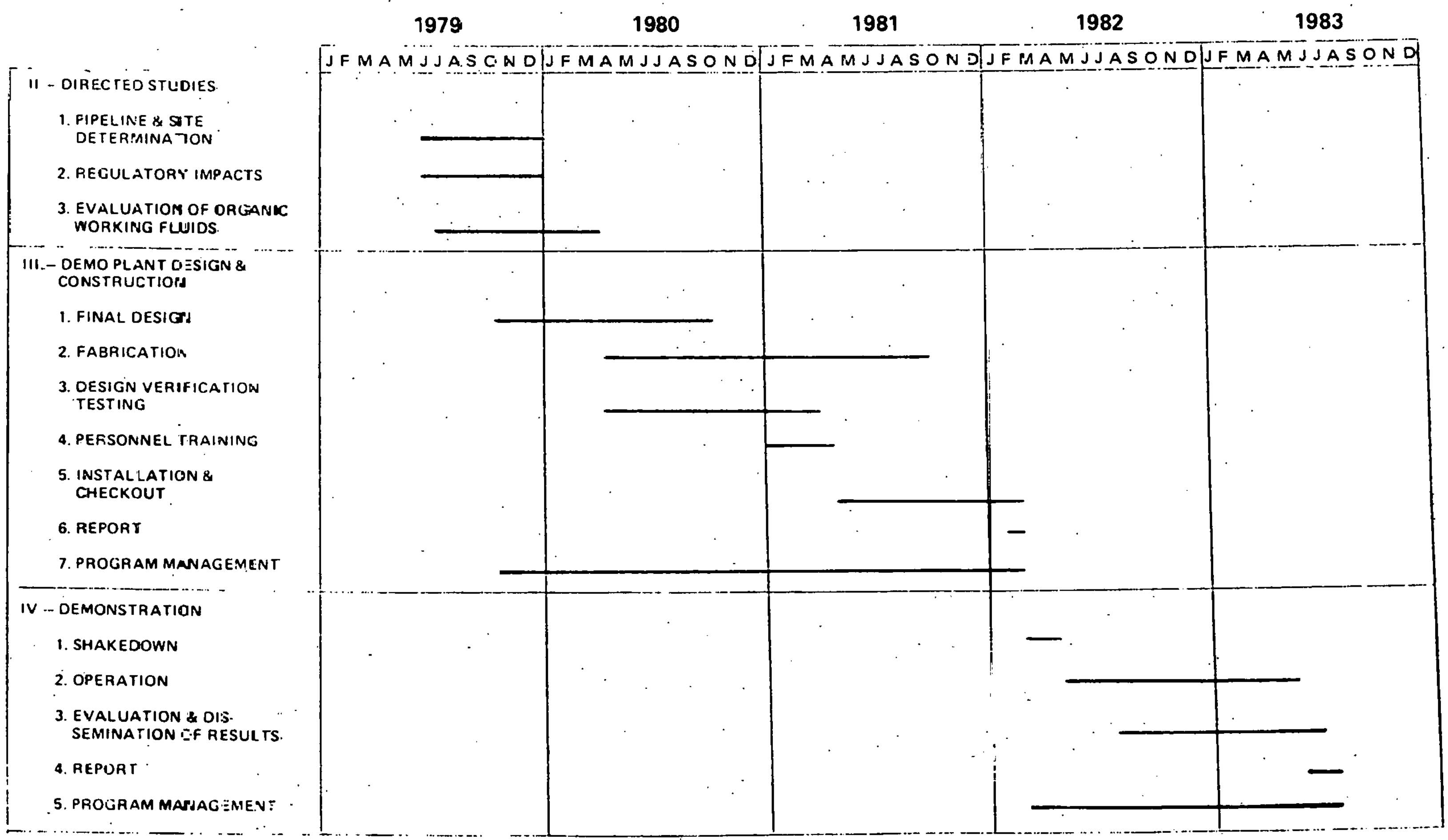

Figure 10-1. Pipeline Bottoming Cycle Program Schedule

10.5 COST ESTIMATES

The cost details for performing Phase II in accordance with the outlined program plan are presented in Table 10-1. The first two tasks in Phase II (demonstration pipeline site determination, and regulatory impacts study) have been funded by the DOE.

The cost details for performing Phase III and Phase IV are presented in Table 10-2.

Like the data for Phase II, the cost estimates to complete each task are divided among the system manager (General Electric Advanced Energy Programs Department), the turbine manufacturer (General Electric Mechanical Drive Turbine Department), the vendors and the pipelines. In order to minimize program costs and to shorten the schedule, the cost is included for a turbine with an efficiency of 77%. This turbine can be built with no research and development from steam turbine designs. A design-verification test of the bearings and seals will be required to assure that these elements using toluene perform satisfactorily. This turbine will have the high reliability inherent in General Electric steam turbines. It will permit system data to be gathered during pipeline operations that can be extrapolated to a production system having an efficiency of 86%.

In the Phase III, Design and Construction costs shown in Table 10-2 a system similar to the one selected for the Rayne, LA site was used since it seemed best suited for the demonstration. The costs shown are larger than the values used in the economic analyses in Section 5. Final Design costs include preliminary and final design of the systems and turbine and extensive control systems studies. Higher component cost values were used for most of the demonstration components rather than production component costs, reflecting first-of-a-kind component costs. The design verification testing of the bearings and seals of the toluene systems is included in the cost estimate. However, the installation costs, exclusive of the gas compressor, were the same as for the Rayne, LA site. The gas compressor cost was higher.

Budgetary cost estimates to complete the three phases of the pipeline bottoming cycle program are given below in 1979 dollars:

TABLE 10-1

ESTIMATED COST FOR PHASE II (in thousands of 1979 dollars)

Phase/Task Description	System Manager	Turbine Manufacturer	Vendors	Pipeline Co.	Total
Phase II - Directed Studies					
1. Demonstration Pipeline Site Determination	50.0	0.0	0.0	0.0	50.0
2. Regulatory Impact on Pipeline Bottoning Cycle Applications	49.0	0.0	0.0	0.0	49.0
3. Evaluation of Organic Working Fluids for Pipeline Rankine Bottoming Cycle Installation	140.5	23.8	150.7	0.0	315.0
Phase II Totals	239.5	23.8	150.7	0.0	414.0

TABLE 10-2

PIPELINE BOTTOMING CYCLE DEMONSTRATION PROGRAM

ESTIMATED COST (IN THOUSANDS OF 1979 DOLLARS)

PHASE/TASK DESCRIPTION	GENERAL ELECTRIC SYSTEM MANAGER	GENERAL ELECTRIC TURBINE MANUFAC-TURER	VENDORS	PIPELINE COMPANY	TOTAL
PHASE III - DESIGN & CONSTURCTION	350.0	722.5	876.3	0.	1948.8
1. FINAL DESIGN	350.0	722.5	876.3	0.	1948.8
2. FABRICATION	571.4	571.0	1424.5	0.	2566.9
3. VERIFICATION TESTING	58.0	486.3	15.0	0.	559.3
4. TRAINING	36.0	7.9	0.	0.	43.9
5. INSTALLATION & CHECKOUT	31.0	31.7	0.	1289.0	1351.7
6. REPORT	19.0	7.9	0.	0.	26.9
7. PROGRAM MANAGEMENT	143.4	0.	0.	0.	143.4
PHASE III TOTALS	1208.8	1827.3	2315.8	1289.0	6640.9
PHASE IV - DEMONSTRATION					
1. SHAKE DOWN	22.0	21.1	0.	20.0	63.1
2. OPERATION	73.0	140.2	0.	100.0	313.2
3. EVALUATION & DISSEMINATION OF RESULTS	43.0	7.9	0.	10.0	60.9
4. REPORT	22.0	2.6	0.	5.0	29.6
5. PROGRAM MANAGEMENT	23.0	0.	0.	0.	23.0
PHASE IV TOTALS	183.0	171.8	0.	135.0	489.8
TOTAL ESTIMATED COST FOR PHASES III & IV	1391.8	1999.1	2315.8	1424.0	7130.7

Phase II	- Directed Studies	$ 414,000
Phase III	- Design and Construction	$ 6,640,900
Phase IV	- Demonstration	$ 489,800
	Total Cost Estimate	$ 7,544,700

Not shown are funds for removal of the bottoming cycle apparatus and restoring the site which may be required in negotiation with a particular pipeline if the pipeline was not satisfied with performance. Retrofitting a high-efficiency turbine when developed surely would be desireable but would depend upon as yet unknown circumstances in the future; as a result funding for this activity was not included.

10.6 COST PARTICIPATION

Under a new task presently underway, Investigation of Demonstration Site and Commercialization of Potential of New and Planned Pipelines, the General Electric Company is investigating other potential sites for demonstration of the pipeline bottoming cycle under actual gas pipeline operating conditions. It would be expected that the pipeline agreeing to furnish the site for the demonstration would participate in the funding of that demonstration. The form and extent of that participation should be based on the value to the pipeline of the annual fuel saving from the demonstration system and will be determined when the final pipeline selection is made. Those companies supplying components for the demonstration system (including General Electric) can be expected to evaluate cost participation on the basis of the market potential for the components they supply, based on the economic attractiveness of the bottoming cycle, as shown by the demonstration system. Suppliers of off-the-shelf components, or components requiring very little modification should not be expected to financially participate in the program.

10.7 CONCLUDING REMARKS

Based upon the findings of the succeeding tasks, recommendations are made for a demonstration program plan including Phase II - Directed Studies, Phase III - Design and Construction and Phase IV - Demonstration. A single milestone chart is presented for the entire program in the main text with the detailed schedules in the appendix. The work statements for the program are also presented in the appendix. The total cost of the program is broken down by phase, by task and by performing team member. Cost Participations are expected in the funding of the program.

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SECTION 11

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APPENDIX A

COMMITMENT LETTERS

Letters of commitment to the Pipeline Bottoming Cycle Program by the Pipeline Companies.

November 16, 1977

Mr. W. H. Hurlebaus, Manager
Advanced Energy Program
Space System Division
General Electric Company
P. O. Box 15132
Cincinnati, Ohio 45215

Dear Mr. Hurlebaus:

Thank you for your letter of September 10, 1977 to William F. Morse inviting our corporation to participate on your team for the Department of Energy's "Pipeline Bottoming Cycle Study". The Columbia Gas System, Inc. recognizes that a bottoming cycle can potentially save fuel in pipeline operations by recovering and using the waste heat in the exhausts of existing compressor station engines. However, it also recognizes that the development and demonstration of an operating system is needed to prove the fuel savings and to evaluate economic merit. The program described in the Department of Energy (DOE) request for proposal No. EC-77-R-03-1381 (Pipeline Bottoming Cycle Study) and which has been awarded to your company will help fulfill this need. Therefore, we are agreeable to be a subcontractor in Phase I of a four-phase program.

Phase I will define the preliminary design of a heat recovery system and its economical and institutional feasibility. Our participation in Phase I will include providing information to you to assist in selecting a demonstration site, developing a preliminary system design and cost for installation at the site and assessing the relative merits of the design. In addition, we will assist in planning for future phases during which the system will be developed, constructed, installed and demonstrated at the selected site.

Our participation in Phase I will be a joint effort between the Research Department of the Columbia Gas System Service Corporation and the Columbia Gulf Transmission Company. Our Research Department furnishes research services exclusively to the parent company (The Columbia Gas System, Inc.) and to its eighteen operating company affiliates. The Columbia Gulf Transmission Company is one of the affiliates. It operates over 3500 miles of transmission pipelines with a design capacity of about 2.1 billion cubic feet per day. It also currently operates 11 on-shore compressor stations plus a compressor station on a platform in the Gulf of Mexico.

We understand that the Texas Gas Transmission Company and Pacific Gas Electric Company have also agreed to participate in the study. From the compressor stations offered as candidate sites by these companies and the

Columbia Gulf Transmission Company, four will be recommended to the DOE for further consideration for demonstrating the heat recovery system. The DOE will make the final selection of a demonstration site. After the demonstration site has been selected by DOE, only the pipeline company operating that site will participate in the remainder of the Phase I program.

Based on the information gathered to date, we believe the compressor station at Rayne, Louisiana is the most promising of the 11 on-shore compressor stations to demonstrate the heat recovery system. However, we recognize that from your evaluation of the information on these stations which we have already supplied, General Electric may wish to recommend a different Columbia site to DOE. We reserve the right to approve your evaluation and recommendation before submittal to DOE.

The information which you receive to assist in selecting a demonstration site will be provided without charge. However, if DOE selects a Columbia Gulf Transmission Company compressor station as the demonstration site, the cost for our participation in the remainder of the Phase I program under subcontract is $16,244. The necessary Contract Pricing Proposal is attached. The subcontract program will be managed and directed by the Research Department of the Columbia Gas System Service Corporation. The Columbia Gulf Transmission Company will provide the necessary engineering support. The details of the subcontract can be resolved after the final selection of the demonstration site.

When the results of Phase I are available, they will be reviewed by us and we will determine if subsequent phases warrant our participation in Phase II (research and development testing), Phase III (detailed design, fabrication and installation) or Phase V (start-up operation and data analysis). It is further understood that should we decide to participate in Phases II, III, and IV, cost sharing by Columbia will be necessary. The cost sharing arrangement will be negotiated and defined during Phase I.

We are also aware that if we participate in Phase III and IV, we must consider purchasing the installed heat recovery system from DOE after completion of Phase IV. This does not, however, commit us to a purchase. The criteria for making a decision to purchase the system will be determined during Phase I. Should Columbia participate in Phase IV and then decide it is not in our best interest to purchase the system, it will be removed from our compressor station and the site will be restored to its original condition at no cost to Columbia. It is also understood that Columbia may be required to reimburse DOE for the value of any gas saved during the Phase IV demonstration program.

We are very interested in working with the General Electric Company in this important program and look forward to the selection of one of our compressor stations as the demonstration site.

Very truly yours,

R. P. Rowen

March 30, 1977

General Electric Company
P. O. Box 15132
Cincinnati, Ohio 45215

ATTENTION: Mr. W. H. Hurlebaus, Manager
Evendale Operations

Gentlemen:

This letter is to advise you that Pacific Gas and Electric Company will participate in your efforts to respond to ERDA RFP EC-77-R-03-1381, Pipeline Bottoming Cycle Study.

We understand that the ERDA Program is directed to the eventual demonstration of a pipeline bottom cycle heat recovery system and that the objective is to assess the potential for widespread commercialization of bottom cycle heat recovery systems throughout the pipeline industry.

The intent will be to utilize existing, as well as advanced technology, in selection, design and demonstration and the program will provide operating data and experience to pipeline owners and operators without incurring the risk associated with development of a new technology.

Pacific Gas and Electric Company (PGandE) operates a 36-Inch diameter intrastate natural gas transmission pipeline which connects with the facilities of Pacific Gas Transmission Company. We anticipate that one or more of PGandE's four compressor stations, utilizing aircraft derivative and industrial gas turbines, could provide a suitable site for study and evaluation of a bottoming cycle system. In addition, PGandE operates a 34-Inch natural gas transmission pipeline system utilizing reciprocating engine-compressor stations which would also be suitable for a bottoming cycle study. PGandE favors the stated goals of the ERDA Program and we are willing to provide you with assistance within our capabilities, in the following areas to support the Phase I study, should you be chosen by ERDA as a contractor in this Program:

- Identify prospective field demonstration sites which have a broad application to pipeline operators.

- Describe system operating parameters and typical duty cycles.

- Define the problems posed by a demonstration program on an operating pipeline.

- Assist in the correlation of data identifying potential heat recovery utilization for pipeline prime movers.

General Electric Company
Page 2
March 30, 1977

- Provide an assessment of industry acceptance of the bottom cycle and attitudes regarding adoption of such a system.

Our participation will be limited to provision of information only as it relates to the PGandE system. Further, while we intend to cooperate closely with G.E. in your conduct of the Phase I work, the burden of conduct, accuracy and timeliness, rests entirely with General Electric.

Should G.E. be successful in obtaining an ERDA contract for Phase III and IV, site installation and operation, we would be willing to consider some form of participation in cost sharing of the demonstration installation if, in our opinion:

a. Phases I and II of the program clearly demonstrate expectation of meeting performance and economic targets; and,

b. Economic conditions internal and external to PGandE clearly justify proceeding at the time when/if Phases III and IV develop; and, if the California Public Utilities Commission grants PGandE any required authorizations to perform the contemplated installations and operations, including rate authorization to recover related costs not otherwise reimbursed.

c. That projected compressor down time during Phases III and IV is acceptable to PGandE and PGandE has control over work schedules.

d. PGandE receives interim funding during Phases III and IV rather than upon completion and operation of the bottoming cycle system, and any payments by PGandE be from documented savings resulting from the bottoming cycle.

We have attached five resumes of key personnel and maps of the pipeline system with a brief description of the compressor equipment, for your use in your response to ERDA.

Thank you for offering PGandE the opportunity to participate with you in this study.

Sincerely,

Attachments

March 30, 1977

United States Energy Research
and Development Administration
Procurement Operations
20 Massachusetts Avenue, N.W.
Washington, D. C. 20545

Gentlemen:

The Gas Transmission Services Division of Texas Gas Transmission Corporation is pleased to join the General Electric Company in cosponsoring a proposal to the Energy Research and Development Administration for Phase I - Pipeline Bottoming Cycle Study.

We are acutely aware of the importance of natural gas to the economy of our country. As an interstate pipeline serving a major section of the United States, we have witnessed the decline of available gas reserves and have encountered the growing difficulty of acquiring new reserves to serve our customers' needs.

The development of a practical heat recovery cycle for use at pipeline compressor stations would improve their use of natural gas as fuel, thus helping to extend our limited supplies. The success of this program will benefit the many customers who are dependent upon natural gas.

The Gas Transmission Services Division of Texas Gas has always been interested in using our natural gas as efficiently as possible. We will continue these efforts in order to help make this design program a success.

You are assured of our full support throughout the proposed project.

Yours very truly,

Robert J. Koch

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APPENDIX B

S^3 PIPELINE FINANCIAL PROJECTION MODEL(16)

The S^3 Financial Projection Model is the software development of Systems, Science and Software under ERDA Contract E(04-03)-1171 for the regulated pipeline companies. The computer code was designed as a general business financial planning model with capabilities to generate various financial data under different conditions. Many financial accounting projection reports are available from this computer code, including:

Report 10: Statement of Income - Profit & Loss Projection

Report 20: Statement of Changes in Financial Position - Cash Flow Projections

Report 30: Statement of Financial Position - Balance Sheet. All of which are the general type of report which reflect the year-to-year overall financial condition of the company.

Report 38: "Capital Investment Planning and Energy Conservation Impact Projects", which enables the investigation of the effects of energy conservation measures, such as the bottoming cycle, on company profitability. It is this report, specifically, as well as the internally calculated DCF*, which makes the S^3 model valuable.

The model is versatile enough to completely simulate the financial standing of the Company under different criteria. The schematic diagram of the general system design of the S^3 Financial projection model is extracted from Reference 16 and is presented in Figure B-1.

Because of its versatile nature, S^3 model was initially used in the economic assessment of pipeline bottoming cycle study on a fictional

*DFC: Discounted Cash Flow

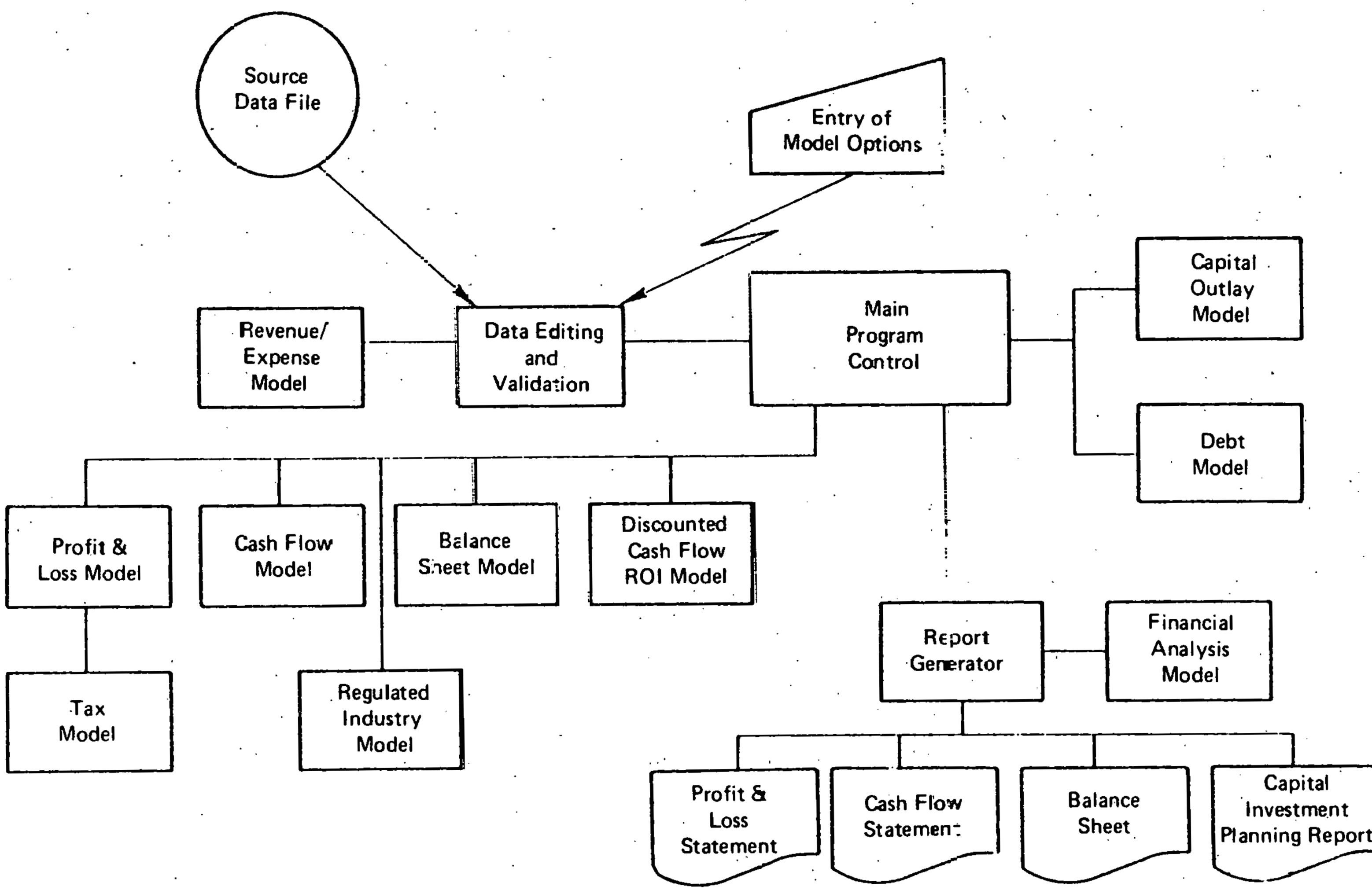

Figure B-1. Schematic Diagram of the General System Design of the S^3 Financial Projection Model.

pipeline company set up by General Electric Company for the purpose of assessing the usefulness of the code. The financial data of the fictional pipeline company was gathered from FERC annual reports*. Two cases were then analyzed using the computer code. One was a base case, using assumptions of fuel price, etc. for twenty years. The second case had simulated bottoming cycle equipment installed. A comparison of the results was made and is reported below.

A fictional gas pipeline company was described by both financial and operational data for the years 1974, 1975, and 1976. Table B-1 shows this data for the first year of simulated operations, 1975. From this starting point, twenty years of operations were simulated under a set of groundrules, shown in Table B-2. The most significant assumption was the use of constant throughput, based on no new gas being available. The cost of gas was assumed to increase faster than inflation through 1984, and then increase at 6%, slightly lower than the assumed general inflation.

It was also assumed that the FERC rate base would be increased at about 6% per year by means of additional capital investment, essentially the replacement of depreciated equipment. No new equity funding was used, and therefore all expenditures were financed by means of mortgage loans and short-term notes.**

A limit of 12% was placed on the rate of return permitted under FPC rules. The effect of this limit is to lower the price at which gas is sold so that the return satisfies the 12% limit. The computer code automatically lowers the difference between gas cost and sale price (nominal tariff) to meet this requirement.

After 20 years, this fictional pipeline company had achieved good financial success. The discounted cash flow rate of return was 23%. The book value of the capital stock had increased from a nominal $61 per share to $264, and a total of $169 per share in dividends had been paid.

To simulate the addition of bottoming cycle equipment, it was assumed that the entire system would be bottomed in the years 1983, 1984, and 1985.

*"Statistics of Interstate National Gas Pipeline Companies-1975", Federal Power Commission, FPC-S-257.

**Not a practical assumption under present FERC rules.

TABLE B-1

BASIC DATA FOR FICTIONAL PIPELINE COMPANY

(1975-1994)

OPERATIONAL DATA		
Annual Throughput (Sales)	365.4×10^9 Ft^3	
Type of Power	Gas Turbines	
Total Power	237,500 HP	
Number of Units	19	
Power Per Unit	12,500 HP	
Fuel Used, Annual	15.1×10^9 Ft^3	
FINANCIAL ASSUMPTIONS		
Price of Gas (1975)	$1.232/MCF	
Sale Price of Gas (1975)	$1.367/MCF	
Short Balance Sheet (1975)	(Millions of Dollars)	
Assets		
Current Assets	5.3	
Property, Plant, Equipment	113.1	
Investments	37.4	
Differed Charges	0.6	
Total Assets	156.4	
Liabilities		
Debt, Long Term	90.4	
Differed Taxes and Credits	0.4	
Capital Stock	43.6	
Retained Earnings	22.0	
Total Liabilities	156.4	
Short Form Profit & Loss (1975)	(Millions of Dollars)	
Revenues		
Net Sales of Nat. Gas	49.9	
Other Sales	1.1	
Total Revenue	51.0	
Expenses		
O&M	3.9	
G&A	2.4	
Cost of Fuel	18.6	
Taxes, Non Income	2.8	
Total Expenses	27.7	
Gross Operating Income		23.3
Interest Expense	5.9	
Financial Depreciation & Amortization	8.3	
Investment Income, Net	-3.5	
Net Income Before Taxes		12.7
Income Taxes		4.3
Net Income		8.4

TABLE B-2

GROUNDRULES USED IN USING S^3 MODEL

Throughput: Constant	
Escalation and Inflation:	
Operating Expenses	7% per year
G&A Expenses	7% per year
Non-Income Taxes	8% per year
Fuel Cost	9.5% per year through 1984
	6% per year thereafter
FERC Rate Base Assumptions	
Maximum Rate of Return (on FERC Rate Base)	12%
Planned Increase in Rate Base	∿ 6% per year
Financing: All increases financed through loans (no new investment capital)	
Interest Rate on Mortgage Loans	8%
Interest rate on Line-of-Credit Loans	9%

Table B-3=shows the assumptions used. The effect is to lower the fuel used and increase the amount available for sale, at the expense of a significant additional capital investment. From 1983 on out to the end of the simulation, the model was modified to account for the higher earnings and larger loan balances. See Figures B-2, B-3, and B-4.

The results shows a modest, but significant improvement in profitability. Table B-4 shows some of the more important measures of profitability. Figure B-5 shows graphically where the total fuel cost savings of 129 million dollars went. Because of the FERC rate limits, much of the savings goes to the customers in the form of lower gas prices. The bottoming cycle case lowered the 1994 sale price of gas from $5.461/MCF to $5.429/MCF.*

The comments received from the three pipeline companies all indicated that they felt that this method was basically too cumbersome, and not the way they would assess the potential purchase of bottoming cycle equipment. Their suggestion of the use of "cost of service" method to predict the economic assessment of the pipeline bottoming cycle equipment convinced the General Electric Company to switch from S^3 financial code to an internally developed computer code which was described in Section 5.3.2.

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*In inflated dollars.

TABLE B-3

BOTTOMING CYCLE EQUIPMENT

Total Gas Turbine Power Before Bottoming	237,500 HP
Cost of Bottoming Cycle Equipment (1977 Dollars)	$350/HP
Bottoming Cycle Addition	25%
Fuel Savings, Assuming Gas Turbine Throttled Back	22%

ADDITION SCHEDULE

Year	BC HP Added	Fuel Used MMCF Per Year	Cost of Bottoming Cycle 1977 $*	Cost of Bottoming Cycle Escalated @ 7%
1983	19800	15,100	6,927,000	12,735,000
1984	19800	14,000	6,927,000	13,627,000
1985	19800	12,900	6,927,000	14,580,000
1986	0	11,800	0	0

*$350/HP

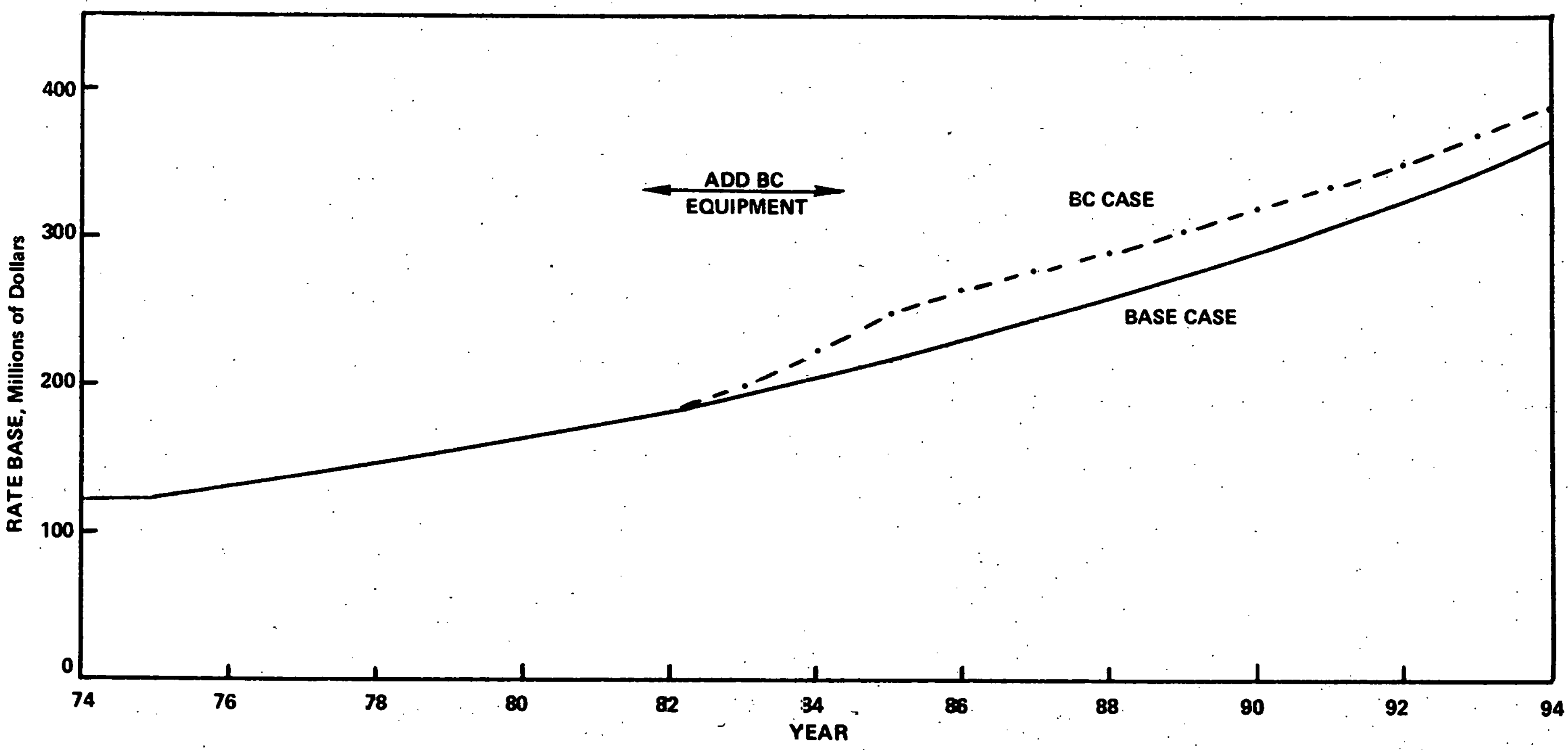


Figure B-2. Effect of Bottoming Cycle on FERC Rate Base.

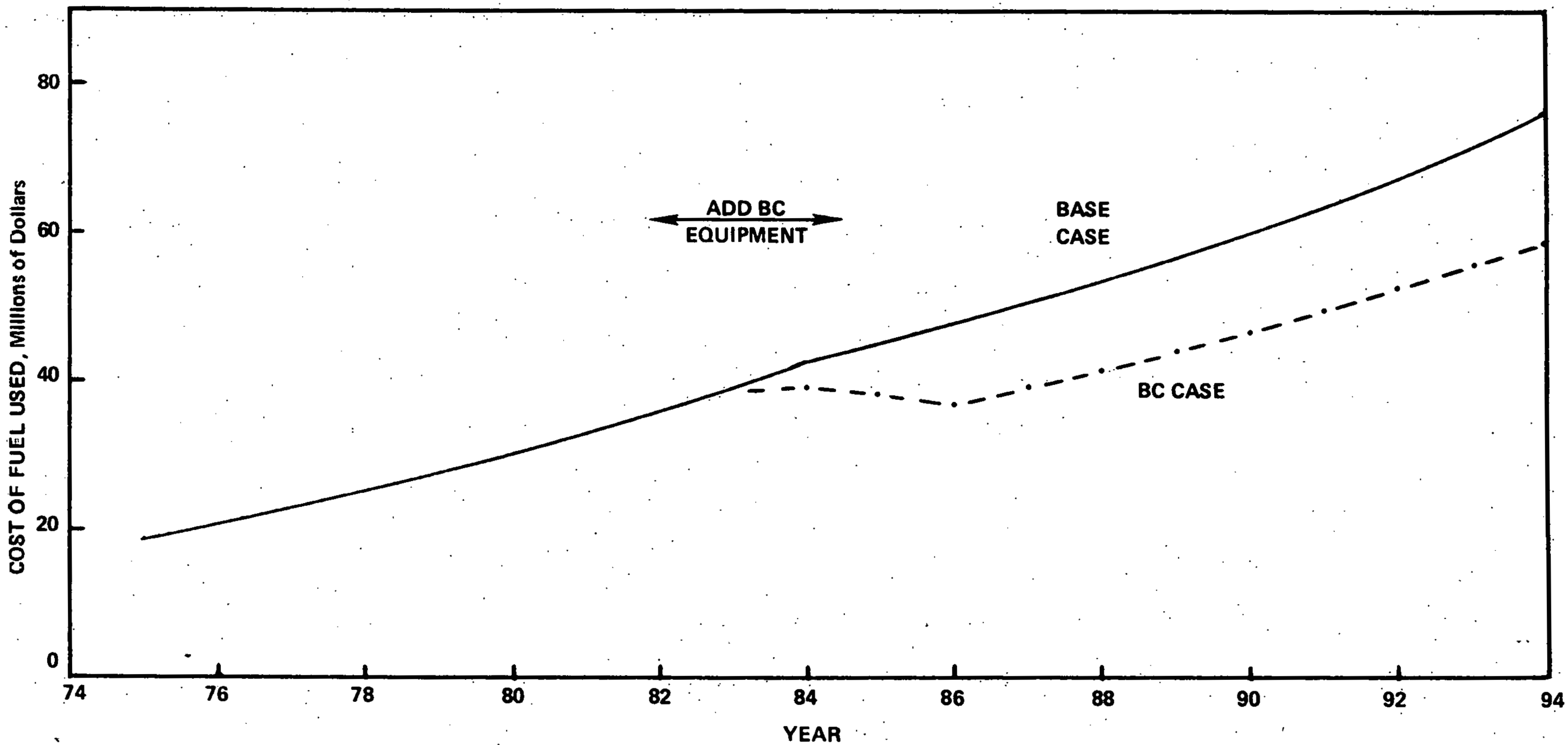


Figure B-3. Effect of Bottoming Cycle on Fuel Cost.

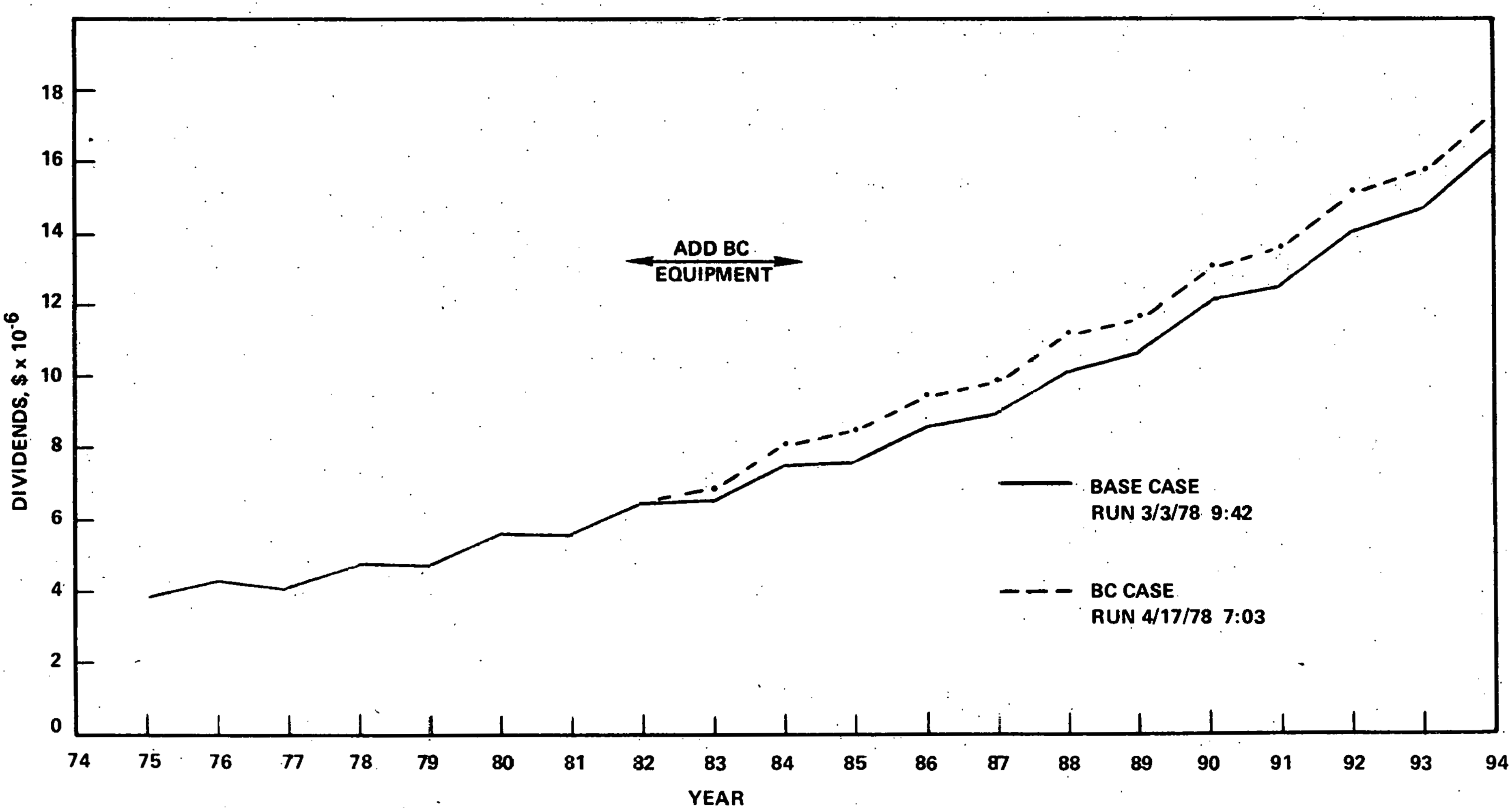


Figure B-4. Effect of Bottoming Cycle on Dividends. (Max Dividends = 45.5% of Net Profit).

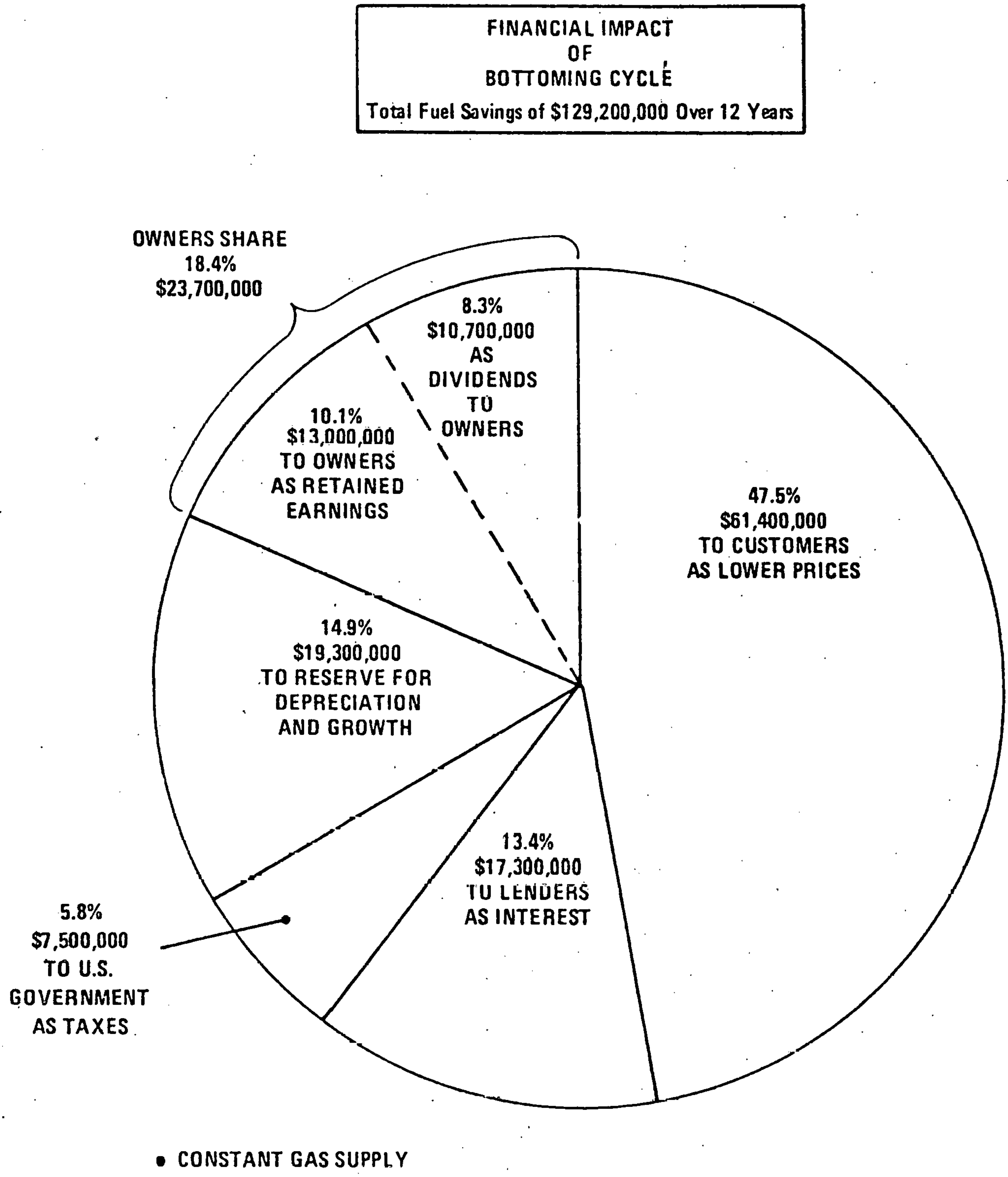
TABLE B-4

OVERALL IMPACT OF BOTTOMING CYCLE

(20 Years Operation)

Parameter	Base Case	Bottoming Cycle Case	Difference	%
Net Income (M$)	372.1	395.8	+ 23.7	+ 6.4
Revenue (M$)	2018.2	1956.8	- 61.4	- 3.0
Total Fuel Used (MMCF x 10^3)	302.2	268.9	- 33.3	-10.0
Cost of Fuel Used (M$)	877.8	748.6	-129.2	-14.7
Book Value of Investment* ($)	264.1	277.1	+ 13.0	+ 4.9
Dividends Paid* ($)	169.3	180.1	+ 10.8	+ 6.4
20 Year DCF-ROI (%)	22.88	23.21	+ 0.33	+ 1.4

*Per nominal share worth $61 in 1974.



- CONSTANT GAS SUPPLY
- BOTTOMING CYCLE FINANCED BY LOANS

Figure B-5. Financial Impact of Pipeline Bottoming Cycle.

APPENDIX C

STATEMENT OF WORK

The paragraphs in the following pages consist of the work statement and describe the work breakdown in the form of tasks and subtasks in Phase II, Phase III and Phase IV of the Pipeline Bottoming Demonstration Program.

The Advanced Energy Programs and the Mechanical Drive Turbine Department of the General Electric Company along with the participation of a pipeline company will complete the three phases of the program within a period of 48 months and the work statement in the following pages represents a realistic means of achieving the program goals.

PIPELINE BOTTOMING CYCLE PROGRAM PLAN

STATEMENT OF WORK

Phase II - Directed Studies

Task 1 - Investigation of Demonstration Site and Commercialization of New and Planned Pipelines

The General Electric Company as Contractor shall carry out the work of identifying and evaluating new and planned pipelines offering potential as demonstration sites and commercial markets for bottoming cycle systems and developing recommendations regarding site suitabilities by performing the following tasks.

Subtask 1.1 - Identify Additional Potential Demonstration Hosts

Perform further investigations of pipeline compression stations suitable for use as demonstration sites for a pipeline bottoming system, with emphasis on new or planned pipeline construction.

The primary criteria for selecting additional potential demonstration hosts shall include, but not necessarily be limited to, the following:

- A site with a large-size, simple-cycle gas turbine,
- A site with a projected near-capacity throughput;
- A site with high cost (or "New") gas.

Data sources such as Statistics of Interstate Natural Gas Pipeline Companies, U.S. Department of Energy; the technical and trade literature and pertinent internal General Electric data shall be utilized. The data shall be used to determine pipeline companies that have existing and planned sites meeting the above criteria.

Although existing pipelines and sites shall be thoroughly investigated, emphasis shall be placed upon identifying proposed compressor stations which will meet the above criteria. Information on required compressor power for proposed sites shall be matched with available simple-cycle gas turbine units having bottoming cycles fitted. Expected heat rates of the combined systems shall be compared with heat rates attainable with available simple-cycle and recuperated gas turbines. The time frame and the environment in which the compressor station is to be built shall be considered in the selection.

The culmination of the subtask shall be a list of pipeline companies having a high probability of having or constructing a site suitable for the pipeline bottoming cycle demonstration and possessing high commercialization potential.

Subtask 1.2 - Analyze Sites Selected

Presentations shall be made to the pipeline companies which have or will construct sites meeting the demonstration host selection criteria of Subtask 1.1 above. The presentations shall include, but not necessarily be limited to, the following:

- Pipeline bottoming cycle performance;
- Bottoming cycle design features;
- Bottoming cycle installation drawings and specifications;
- Energy savings;
- Economic, technical feasibility, environmental and safety, and operational reliability and maintainability assessments;
- Proposed demonstration program plan;
- Outline of pipeline company tasks;
- Need for cost sharing;
- Eventual need for a commitment.

Data shall be solicited from the pipelines which shall include, but not necessarily be limited to, the following:

- Prime mover and details affecting combined system performance;
- Pipeline capacity and projected throughput and cost of natural gas;
- Cost of bottoming cycle installation;
- Pipeline financial data for incremental cost of service code;
- Special problems.

The contractor shall use the solicited data to evaluate the energy conservation potential of the site by assessing the performance, economy, and practicality of the installation. The results of the evaluation and other pertinent available data shall be presented to the pipeline company for review and consideration.

Subtask 1.3 - Recommendations for Pipeline Demonstration Sites

On the basis of the foregoing analyses, recommendations concerning the suitability of the investigated sites for bottoming demonstrations shall be made. Factors considered shall include, but not necessarily be limited to the following:

- Potential contribution to pipeline energy conservation, including commercialization benefits;
- Interest of the pipeline companies;
- Tentative cost sharing provisions;
- Performance and economics;
- Description of bottoming cycle apparatus to be installed on the site.

Subtask 1.4 - Reporting

The contractor shall provide, in addition to the reports required by the present contract, a topical report on the Demonstration Pipeline and Site Determination.

Task 2 - Investigation of Regualtory Impacts on Pipeline Bottoming Cycle Applications

The Contractor shall perform the effort described below in addition to the work described in DOE Contract No. EC-77-C-03-1381 and previously issued modifications. These changes are brought about by considering the impacts of current regulations on the application of pipeline bottoming cycle systems. This effort is a refinement of work performed under the current contract as well as prior work*, and will involve close coordination with other DOE contractors as specified by the DOE Program Manager.

A major result of this study will be to further the understanding of the impact of current pipeline regulations on energy conservation investments. This shall be done by an approximate quantification of the costs and benefits associated with specific regulatory modifications.

Subtask 2.1 - Description and Analysis of Market Segments & Environment

The Contractor shall define market segments for the organic Rankine bottoming cycles as applied to the gas pipeline market. These segments shall be defined by appropriate characteristics such as:

- Prime mover (type and size)
- Type of installation (retrofit, replacement,or new)
- Type of RBC power use (gas pumping or electric generation)
- Type of regulation (FERC, PUC)

Segments that shall be emphasized in this task are those including:

- Gas turbine (aircraft derivative) and reciprocating prime movers
- Retrofit and replacement installation
- FERC regulation

Since this Task provides the basis for describing the baseline implementation rates, those market segments shall be identified and described in the detail required for subsequent tasks.

The period of interest for this analysis shall be 1981 through 1995.

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*Banks, W.F., "Federal Regulations of the Pipeline Industry A Summary Review," Systems, Science and Software, SSS-R-77-3024 Revision 1, 31 May 1977 (Under ERDA Contract No. E(04-03)-1171.

For the gas pipeline industry, as made up by the described market segments, the expected number of units installed as a function of time shall be described under the present regulatory environment and economic forecasts. This baseline implementation rate shall include allowances for gas pipeline industry growth, if such allowance is deemed appropriate. The total bottoming cycle power and fuel saved for the period of interest shall also be determined.

The investment decision criteria appropriate to the types of regulation analyzed shall be determined. The criteria described shall be consistent with the market penetrations developed for the baseline case. In the analysis of decision criteria, an understanding of financial criteria shall be emphasized so that in subsequent tasks, the Contractor can analyze the sensitivity of the decision to specific financial or economic changes.

The Contractor shall utilize the information and results of the market penetration and economic assessment tasks of the present contract in completing this task; updating the existing information as required.

The Contractor shall determine the probable implementation rates and numbers of each segment, and calculate the fuel savings. The Contractor will also determine the decision criteria appropriate to each segment, based on current regulatory constraints.

In the performance of this task, the Contractor shall work closely with the other DOE Contractors, as specified by the DOE Program Manager, in order to insure adequacy of form and content for Subtask 2.2 analysis.

Subtask 2.2 - Analysis of the Impact of Selected Economic/ Regulatory Modifications

This task will utilize the results of Subtask 2.1 in analyzing the impact of regulatory/economic changes on the implementation of RBC units and on fuel savings.

Subtask 2.2.1 - Selection and Definition of Regulatory Changes

The Contractor shall monitor, become familiar with the work of, and consult with the other DOE Contractors working on this task. In particular, the Contractor will review the results to insure their adequacy in form and content for use in Subtask 2.2.2.

Subtask 2.2.2 - Calculation of Changes in Implementation Quantities and Rates

Based on methods used in the economic and market assessment of the current contract and utilizing the output of Subtask 2.2.1, implementation rates under each regulatory modification shall be estimated for the period 1981-1995. The economic or financial criteria or "hurdle rate" as established in the current contract shall be the primary basis for the description of implementation rates.

The output of the subtask shall be an estimate of the number of units that could be in service vs. time, based on the "hurdle rates." An appropriate presentation such as graphs or curves shall be utilized to illustrate the implementation estimates. It is noted that the type and value of the hurdle rates will vary according to whether the pipeline is FERC or state PUC regulated. FERC regulated pipelines shall be emphasized in this subtask.

Based on the energy savings due to bottoming cycles that were determined for the baseline case, the change in energy savings resulting from each regulatory modification that is analyzed shall be determined.

Subtask 2.2.3 - Costs of Regulatory Changes

As a result of Subtask 2.2.2, a preliminary estimate of the maximum increase that would be expected in adoption of bottoming cycle units and energy savings due to regulatory changes will be known. However, costs are associated with the regulatory changes. This subtask will address these considerations.

The output of Subtask 2.2.1 will be utilized to estimate costs, such as those due to decreases in Government revenues, changes in the pipeline tariff structure, and others that may be appropriate. Then, based on the implementation rates of Subtask 2.2.2, a gross cost value shall be determined. This cost estimating exercise shall draw primarily from the expertise gained in the market segment descriptions and penetration rates, but shall also require significant input from the expertise gained in the regulatory descriptions and modifications so that gross values of cost can be determined.

The Contractor shall primarily address the calculation of costs, relying on the associated DOE contractors to provide the input based on their expertise in regulatory descriptions and modifications.

Subtask 2.3 - Conclusions and Summary Report

The conclusions arising from Subtasks 2.1 and 2.2 will be summarized and incorporated in a final report which will document the work performed under this study. The report shall be submitted to the DOE Technical Contract Monitor in draft form for review and comment. A final report will be submitted within 30 days of receipt of DOE comments.

This final report is primarily the responsibility of the associated DOE Contractors, and the Contractor shall only provide consultation and aid in incorporating those sections for which he was responsible.

Task 3 - Evaluation of Organic Working Fluids for a Natural Gas Pipeline Rankine Bottoming Cycle Installation

Subtask 3.1 - Pipeline RBC Installation Site Selection

During this task AEP in conjunction with the pipeline representative will select a site for a potential RBC demonstration system installation. This selection task will draw heavily upon the results of the AEP Pipeline Bottoming Cycle Study Program and will be directed toward the bottoming of an aircraft derivative gas turbine engine.

Subtask 3.2 - Fluid Selection and Preliminary Evaluation

The basis for the selection and evaluation of candidate fluids will be established by detailing the specifications against which these fluids will be evaluated. These specifications will be determined on the basis of the total system operational and required performance characteristics. Fluids to be used in the proposed program will include toluene, propane, butane, methane (as reference) and up to 6 others that will be determined by screening additional candidates. The initial selection of additional fluids considered of potential interest for this program will be used on MRC's knowledge of fluids properties as functions of chemical structure and their background in the evaluation of fluids for Rankine cycle engines for automotive engines and for the utilization of solar energy. In consideration of the fluids strong guidance by the chemical structure-thermal stability correlations that have been developed will be adhered to and fluids will also be excluded that would be expected to pose problems because of materials incompatibility, or because of adverse health or environmental effects. Most of the information for the initial screening will be obtained from data compilations in the literature, by computerized literature searches, from trade literature and from the Monsanto Company's computerized files of the thermophysical properties of fluids. The initial screening will be of the pass/fail type. Candidate fluids that pass this phase of evaluation will be subjected to experimental testing.

Subtask 3.3 - Fluid Stability

During this task thermochemical stability tests will be conducted with the candidate fluids in metal sample vials to simulate the use environment and exposure. These will be fabricated from the metals that will contact the RBC working fluid at elevated temperatures. The tests will be conducted at the maximum working temperature. Test durations will be 1, 5, 9 and 13 days. Compounds that remain stable at the maximum working temperature will also be tested at 100°F above this temperature. Testing at the latter temperature will be conducted to detect potential problems with fluids that show no degradation during the 13-day tests at the maximum working temperature.

The thermochemical stability tests will be pass/fail type determinations in the evaluation of candidate fluids. Only those fluids that are completely stable at the maximum working temperature and at 100°F above this temperature, or that under go only minimal degradation at the latter temperature, will be considered to have passed the thermochemical stability evaluation with the static test system.

The two leading candidate fluids, evolving from all phases of the testing and evaluation program, will be subjected to 1000-hr dynamic loop testing at the working temperature. The gaseous atmospheres of this test system, the fluids, and the internal surfaces of the test loop coils will be subjected to similar examinations and analyses as described for the static thermochemical stability tests.

Subtask 3.4 - Materials Compatibility

The candidate materials that will contact the fluid in the RBC system will be identified by GE. Welded low carbon, low alloy steel is the preferred material in the hot, high-pressure section of the Rankine engine. Its compatibility with the candidate fluids will be evaluated in conjunction with the thermochemical testing of fluids. All other materials that will contact the working fluid at temperatures lower than those prevailing in the high-pressure section will be tested for compatibility at 300°F. The compatibility of materials exterior to the gas turbine/RBC system installation, such as insulation, floor tiles and paint, will be determined with the candidate fluids. Compatibility tests will also be conducted with any elastomers, plastics, metals and solder materials likely to be in contact with the fluids. Materials that absorb large quantities of a fluid, discolor extensively, dissolve in or react with the fluid, or catalyze the thermochemical degradation of the tested fluid will be rated as incompatible with that fluid.

Subtask 3.5 - Fire Safety Evaluation

The fire safety evaluation will consist of three types of activities:

- Ignitability evaluation of fluids by tests that simulate potential accident situations for the Rankine engine working fluids
- Conception and recommendation of protective measures
- Establishment of codes and regulations that apply to the use of fluids in locations where the Rankine engines will be placed into operation.

Flash ignition temperature, fire point, autoignition temperature, manifold ignition temperature will be determined and a hot compartment ignition test will be conducted for each candidate working fluid. A total system analysis will be performed considering all conceivable modes by which the working fluid could cause accidental fires at the RBC installation. Means for minimizing fire accident risk will be defined and recommended. The existence or absence of state and local fire codes at the specified RBC installation site will be established to assure that the selected fluid will be in compliance with regulations.

Subtask 3.6 - Health and Environmental Safety Evaluation

The following activities will be performed to provide maximum possible assurance that the selected working fluid will not pose human health environmental problems:

- Evaluation of the candidate fluids for acute toxicity and mutagenicity
- Establishment of biodegradation characteristics
- Establishment of handling and disposal guidelines.

Level 1 acute cytotoxicity and metagenicity tests will be conducted for all the candidate working fluids with Level 2 and Level 3 testing as deemed necessary. The threshold limiting values for exposure for most candidate fluids will be available in the literature. For the fluids for which the data have not been reported, the values will be determined. Since fluid biodegradation studies are quite involved and time-consuming such studies will be conducted only with the prime candidate fluid if that fluid has a low vapor pressure at ambient temperature. Codes and other legal regulations pertaining to the transportation, handling and disposal of the candidate fluids will also be established.

Subtask 3.7 - Rankine Cycle Evaluation

Criteria developed by David Miller of Monsanto Corporation in the search and selection of optimum working fluids for automotive Rankine engines will be adapted to the proposed program. A fluid parameter developed during that study, called the I-factor, will be used as the main thermodynamic screening parameter. I-factors which express the tendency of isentropically expanding vapors to converge with or diverge from saturated vapor lines will be calculated for the candidate fluids using available computer programs. Complete Rankine cycle efficiency calculations will be conducted for the fluids that have a high ranking on the basis of the I-factor calculations and their applicability to the pipeline bottoming cycle system will be determined.

Subtask 3.8 - Final Report

A final report will be prepared detailing the results and conclusions of the fluids evaluation for the natural gas pipeline bottoming cycle installation. This report will incorporate the previously prepared task reports into a final document to be submitted for approval.

Phase III - Design and Construction

Task 1 - Design

Subtask 1.1 - Design Studies

Beginning with the analysis of the selected site performed in Phase II, Subtask 1.2, a design study shall be carried out to determine the best design point conditions for the selected site. The result of this study shall be a heat balance diagram for the bottoming cycle at design conditions. From this diagram shall come the design requirements for the several components of the system. Included among the components shall be all auxiliary equipment needed for the safe and proper operation of the bottoming cycle system including fire protection systems and the necessary sensors, a safe dump-tank for storage of working fluid in the event of a fire, sealant and lube sytems and a non-condensible removal system. The requirements for any enclosures or buildings shall be explored.

System design criteria shall next be compiled. These criteria shall include, but not necessarily be limited to, the following: State and Local regulations, EPA and OSHA standards and insurance company requirements. In addition the following codes shall be used where appropriate:

- The ASME Boiler Code
- The ASME Unfired Pressure Vessel Code
- The ANSI Power Piping Code
- The TEMA Heat Exchanger Code

- The National Electric Code
- The NFPA Flammable and Combustible Liquid Code

Special attention shall be given to the requirements and selection of seals. The number of shaft seals shall be minimized and consideration shall be given to whether such seals shall be placed where the working fluid pressure is normally above or below atmospheric pressure. The number of flanges in the working fluid piping also shall be minimized and types of gaskets or seals for the required flanges shall be identified. Valves with both bellows and packing gland shall be considered and the type to be used shall be selected. A list of approved materials of construction for the several components comprising the working fluid and combustion gas envelopes shall be compiled. These materials shall be selected for the appropriate temperature ranges resulting from the anticipated operating conditions of the bottoming cycle system. Considered also in the selection of materials shall be the thermal stability of the working fluid, including the catalytic effect of materials on decomposition, corrosion, the stress levels anticipated and the cost of the materials.

A control philosophy for the bottoming cycle system shall be established. This control philosophy shall include all anticipated operational modes, including emergency operating conditions. It shall also include protection of the working fluid from thermal decomposition. The control instrumentation shall also be considered. In order to make the design of the bottoming cycle widely applicable to employment on various sites and in conjunction with the several similar gas turbines selected for this program, the designs for the several sites considered in this program shall be reviewed for site-related or prime-mover-related differences. The appropriate design compromises shall be delineated so as to make the demonstration hardware widely applicable. The output of these design studies shall be the required information to initiate subtasks 1.3 and 1.5 (Architect/Engineer Siting Study and Design and Build Specifications).

Subtask 1.2 - Preliminary Turbine Design

Beginning with the analysis of the selected site performed in Phase II, Subtask 1.2, a preliminary turbine design shall be initiated to determine the best design for the selected site. The preliminary turbine design task shall support the overall design Subtask 1.1 of Phase III to arrive at the design point conditions of the bottoming cycles. The preliminary turbine design shall be carried out by the Mechanical Drive Turbine Division of GE. The design task shall provide the MDTD component and GE sufficient time to investigate the methods to incorporate the standard turbine components and practices tailored to site specific parameters. Under a separate Task 1.6 of Phase III, the final turbine design shall be carried out to meet the schedule as per the design/build specifications that shall be generated as a result of this task and Subtask 1.1.

Subtask 1.3 - Subcontract to Architect/Engineer

The statement of work, including a milestone chart, for an Architect/Engineer shall be prepared. An Architect/Engineer shall be selected and a subcontract for services shall be placed.

Subtask 1.4 - Architect/Engineer Siting Study

The Architect/Engineer shall carry out a siting study in enough detail so that installation costs can be estimated and the work schedules established. The study shall be based upon the work of Phase III, Subtask 1.1 (Design Studies), Phase II, Task 1 (Demonstration Pipeline Site Determination), and Phase I, Task 2 (System Preliminary Design (present contract)). In this study by the Architect/Engineer the emphasis shall be on minimizing the installation cost within the constraints, EPA and OSHA standards and insurance company requirements.

Subtask 1.5 - Architect/Engineer Costs and Schedules

Based upon the Architect/Engineer Siting Study of the previous subtask the Architect/Engineer shall estimate the installation cost and prepare work schedules for the installation activities.

Subtask 1.6 - Final Turbine Design

Based upon Phase III, Task 1, Subtask 1.1 (Design Studies), Subtask 1.2 (Preliminary Turbine Design), Phase II, Task I (Demonstration Pipeline and Site Determination) and Phase I, Task 2 (System Preliminary Design), the design specifications for the organic turbine design for the bottoming cycle shall be provided to MDTD component of GE.

After completion of the site specific heat balance, GE shall initiate turbine final design in order to meet the schedule requirements. The results and the knowledge acquired in the Subtask 1.2 of Phase II shall be used to meet the schedule. The design will incorporate many standard turbine components and practices tailored to site specific parameters. The basic turbine design will cover as wide an application range as is practical for variations in heat balance parameters. The turbine design will incorporate areas including thermo-dynamic layout, bearings and seals, valving, mechanical design and rotor dynamic analysis and turbine controls and instrumentation. This Subtask 1.6 will result in a design to be fabricated in Subtask 2.1 of Task 2.

Subtask 1.7 - Design and Build Specifications

Based upon Phase III, Task 1, Subtask 1.1 (Design Studies), Subtask 1.2 (Preliminary Turbine Design), Phase II, Task 1 (Demonstration Pipeline and Site Determination), and Phase I, Task 2 (System Preliminary Design), the design specifications for the several components of the bottoming cycle system shall be written. These specifications shall be used to secure bids for the construction of system components by vendors. The specifications shall completely define the functions of the several components, the range over which they shall be designed to operate, and the design conditions which are consistent with the design point of the system. The design specifications shall be prepared for the following components:

- 1.7.1 Vapor generator including fire protection systems and sensors.
- 1.7.2 Condenser, including fans and motors.
- 1.7.3 Diverter valve.
- 1.7.4 Load apparatus, including all auxiliaries.
- 1.7.5 Feed pump and drive, including lube and sealant system.
- 1.7.6 Power package, including surge tank, non-condensible extraction, dump tank and mounting skid.
- 1.7.7 Control system, including instrumentation, cabinet and annunciator panels and equipment.

The level of quality control required in fabrication of the components shall be determined and incorporated in the design specifications.

Subtask 1.8 - Component Bids

The design specifications written in the previous subtask shall be sent out for bids. The bidders for each component shall include, where possible, more than three vendors. Prior to the selection of vendors for the bidders list a set of qualifying criteria for vendors shall be drawn up. The vendors for the list shall be chosen because they meet the established criteria.

The bids shall be evaluated for compliance with all specifications on delivery, price, capability and recent performance. A Purchase Order shall be placed with the selected vendor for each component. A make or buy decision on the turbine shall be made by the prime contractor.

Subtask 1.9 - Design Components

Based upon the design specifications the vendors shall design the several components and provide price and delivery. Installation drawings for the several components with certified dimensions shall be provided. Component designs shall be made for the following components:

1.9.1 Vapor generator, including fire protection systems and sensors.

1.9.2 Condenser, including fans and motors.

1.9.3 Diverter valve.

1.9.4 Load apparatus, including all auxiliaries.

1.9.5 Feed pump and drive, including lube and sealant systems.

1.9.6 Power Package, including surge tank, non-condensible extraction, dump tank, and mounting skid.

1.9.7 Control system, including instrumentation, cabinets and annunciator panels and equipment.

Subtask 1.10 - Architect/Engineer Site Design

Based upon Phase III, Task 1, Subtask 1.3 (Siting Study), and Subtask 1.9 (Design Components) the Architect/Engineer shall make the final site design for the installation of the bottoming cycle and load components. In this design the vendor drawings with certified dimensions shall be utilized. The final layouts and construction details shall be approved by the prime contractor and the host pipeline company. The site design shall include, but not necessarily be limited to, the following:

- Arrangement of the apparatus.
- Foundations and structures for the bottoming cycle apparatus and the load machinery.
- Piping, ducting, wiring and electrical equipment.
- Site improvements.
- Construction and installation drawings and specifications.

Subtask 1.11 - Secure Installation Bids

The prime contractor shall with the assistance of the Architect/Engineer compile a list of approved construction contractors. Qualifications for listing shall be delineated prior to the selection of the construction contractor. Bids shall be sent out to bidders on the approved list. The returned bids shall be evaluated for compliance with specifications, price, delivery, capability, and recent performance. A Purchase Order shall be placed with the selected vendor.

Task 2 - Fabrication

Subtask 2.1 - Orgainic Vapor Turbine Fabrication

The organic vapor turbine and accessories including the lube and sealant systems shall be fabricated under Subtask 2.1 by Mechanical Drive Turbine Division (MDTD) of GE in accordance with the design made in Subtask 1.6. The turbine fabrication shall be inspected to assure full compliance with design requirements.

Subtask 2.2 - Fabrication of Other Bottoming Cycle Components

The Subtasks 2.2 through 2.7 involve the fabrication of the bottoming cycle components by the vendors. The bottoming cycle components shall be fabricated according to the specifications generated in Subtask 1.7 - (Design and Build Specifications) and the delivery schedule established by the vendor in his bid, Subtask 1.8 (Component Bids). The components received shall comply with the quality control specifications delineated in Subtask 1.7 - (Design and Build Specifications). The components shall be inspected to assure compliance with the design specifications. Components not in compliance with the specifications or quality control requirements shall be rejected unless found to be acceptable by a designated representative of the prime contractor. The following components shall be fabricated according to the above requirements:

2.2 Vapor generator, including fire protection system and sensors.

2.3 Condenser, including fans and motors.

2.4 Diverter valve.

2.5 Load apparatus, including all auxiliaries.

2.6 Feed pump and drive, including lube and sealant systems.

2.7 Control system, including instrumentation, cabinets, and annunciator panels and equipment.

Simultaneously under a separate Subtask 2.8, GE will fabricate power package, including surge tank, non-condensible extractor, dump tank and mounting skid.

Task 3 - Verification

It is expected that design verification testing will be required on the bearings and seals for the toluene system. Rotating test units will be required for the bearing and seal test facilities. The seal test unit besides testing rotating seals at approximately the design rotative speed of the proposed equipment will also test the packing of a typical valve in the system and the gasket or seal material for typical flanged joints.

Subtask 3.1 - Test Unit and Facility Modification Design

A test plan to verify the design of the bearings and the rotating, valve stem and flanged joint seal shall be prepared. A test unit shall be designed for the existing test facilities, including test instrumentation, to carry out the planned tests. The test unit shall contain a full-size bearing and rotating seal as well as typical valve and flanged joint. The configuration(s) of the bearings, rotating seal, valve packing and flange gasket or seal shall be designed. Following this the required modifications of the existing facilities shall be designed in sufficient detail to permit fabrication.

Subtask 3.2 - Test Fabrication and Modification

The test unit shall be fabricated according to the design drawings and specifications and the instrumentation shall be installed.

Subtask 3.3 - Facility Preparation

The existing test facilities shall be modified for the new test unit and the test unit shall be installed. The test unit instrumentation shall be connected to the appropriate readout.

Subtask 3.4 - Verification Testing

The verification testing shall be carried out according to the test plan written in Subtask 3.1.

Subtask 3.5 - Test Documentation

A report shall be prepared documenting the bearings, rotating, valve stem and flanged-joint seal design; the test unit design, the facility modifications, the instrumentation and the results of the verification tests.

Task 4 - Training

Subtask 4.1 - Operational Manual Preparation

A manual shall be written which describes the operation of the bottoming cycle with the gas turbine to which it is attached, including emergency procedures. The manual shall also describe the bottoming cycle apparatus and its maintenance and repair. Pertinent information obtained from the bottoming cycle checkout shall be placed in the manual.

Subtask 4.2 - Hands-Off Training

The contractor shall carry out the training of pipeline personnel utilizing the Operational Manual. The training shall include operation, maintenance and repair of the bottoming cycle apparatus.

Task 5 - Contractor Installation and Checkout

Subtask 5.1 - Installation

The construction contractor shall install the bottoming cycle and load apparatus according to the plan and schedule of the Architect/Engineer developed in Subtask 1.10 (A/E Site Design). The prime Contractor shall provide technical support in the installation of the bottoming cycle and load apparatus. The installation shall be inspected by the Architect/Engineer and/or by a representative of the Pipeline Company. The installation shall include, but not necessarily be limited to, the following:

5.1.1 Vapor generator, including fire protection system and sensors

5.1.2 Condenser, including fans and motors

5.1.3 Diverter valve

5.1.4 Load apparatus, including all auxiliaries

5.1.5 Power package, including the following:

- Turbine and accessories, including lube and sealant systems
- Feed pump and drive, including lube and sealant systems
- Surge tank, non-condensible extraction, dump tank and mounting skid

5.1.6 Controls, including instrumentation, cabinets, and annunciator panel and equipment

Subtask 5.2 - Checkout

The construction contractor shall check out all equipment and apparatus listed above to assure that the bottoming cycle and load apparatus will function as intended on startup. The checkout of the above equipment and apparatus shall be approved by the Architect/Engineer and/or the representative of the Pipeline company before the apparatus is accepted from the Construction Contractor.

Task 6 - Report

The Contractor shall submit a report which documents the pertinent activites and data of Phase III, Construction. The report shall document the Design Studies, the A/E Siting Study, the Bottoming Cycle Drawings and Specifications, the Component Designs, the Design Verification Results and the Results of the Bottoming Cycle Checkout.

Phase IV - Demonstration

Task 1 - Shakedown

Subtask 1.1 - Startup

Initially a test plan shall be prepared for the startup testing, including objectives, procedures to be followed, description of test equipment (including accuracy), operating conditions and schedule. Startup testing shall then be carried out in accordance with the test plan. The results shall be documented in a report.

Subtask 1.2 - Hands-On Training

The pipeline operating personnel shall be given hands-on training during the startup testing. The Contractor's training personnel shall oversee the hands-on training.

Task 2 - Operation

The running of the bottoming cycle apparatus during the demonstration shall be carried out by the pipeline company personnel under actual pipeline operating conditions. During operation performance data shall be measured, operating problems shall be defined, maintanence required shall be noted, the personnel required shall be documented and operating costs shall be determined.

Task 3 - Evaluate and Disseminate Results

Subtask 3.1 - Evaluation

The data generated during operation shall be used to determine bottoming cycle off-design performance and turn-down ratios. In addition operating problems shall be analyzed, maintenance requirements determined, cost of service computed, and required numbers of operating personnel delineated. The findings under Task 3.1 (Evaluation) shall be documented.

Subtask 3.2 - Dissemination of Information

The Contractor, in cooperation with the pipeline company, shall disseminate the results of the operational testing and other pertinent information about the bottoming cycle apparatus, through the trade associations, such as the American Gas Association, and through the pertinent engineering societies.

Task 4 - Report

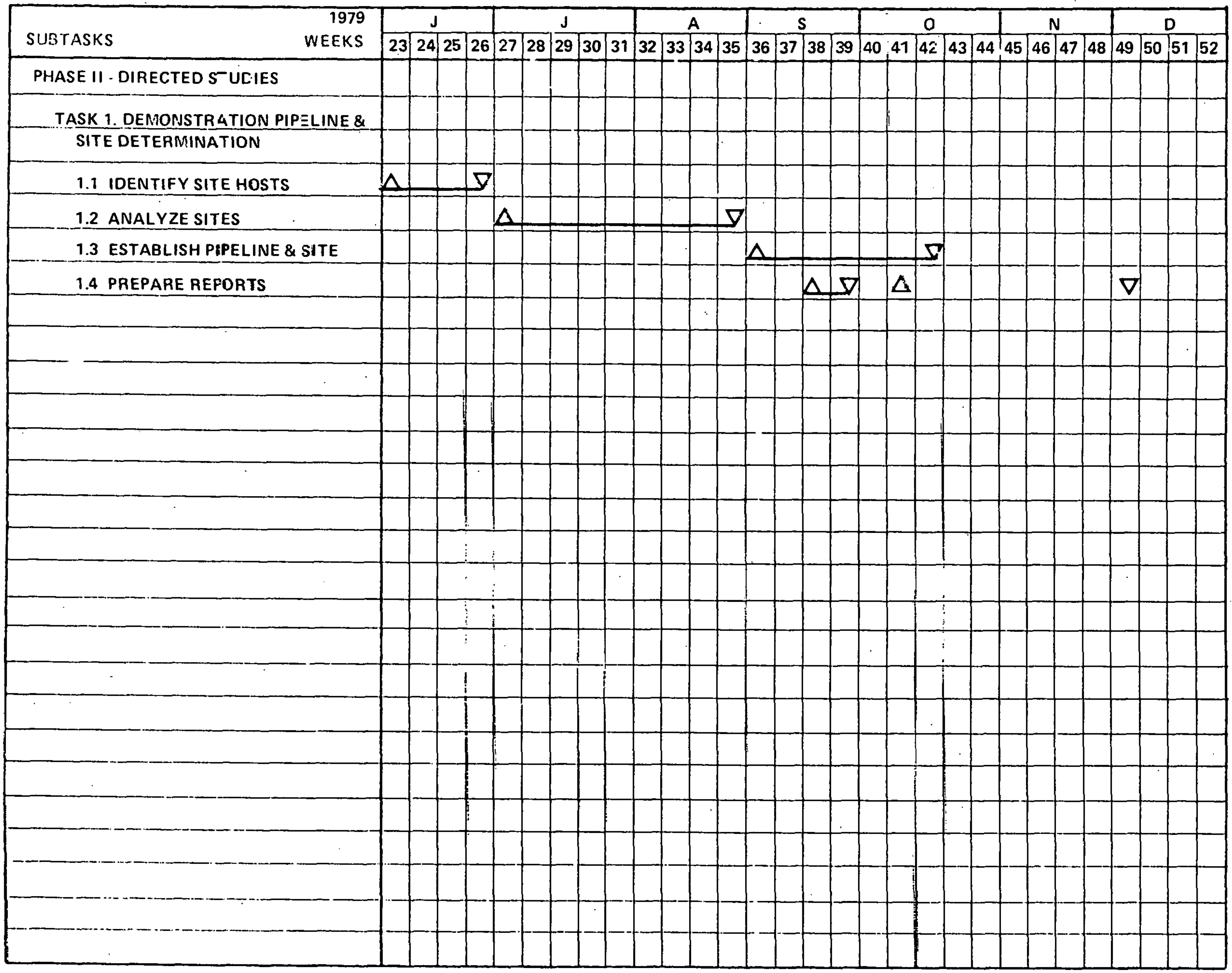
The Contractor shall prepare a report on the Phase IV activities. The report shall include the results of the apparatus shakedown activities, the findings during operational testing concerning system performance, operating problems, the cost of service, maintenance cost, frequency, and procedures.

APPENDIX D

PROGRAM SCHEDULE

On the following pages the program schedules for the Phase II, Phase III and Phase IV of the Department of Energy Pipeline Bottoming Cycle Program are shown in greater detail. The master schedule for the entire program was shown in Figure 10-1 of this report. The following schedule charts indicate the milestones for the entire program by task in each phase of the program. The entire program is shown to last for 48 months starting with the "Directed Studies" task in Phase II in June 1979 through the report on the demonstration of the Bottoming Cycle apparatus on a participating pipeline in Phase IV in May 1983.

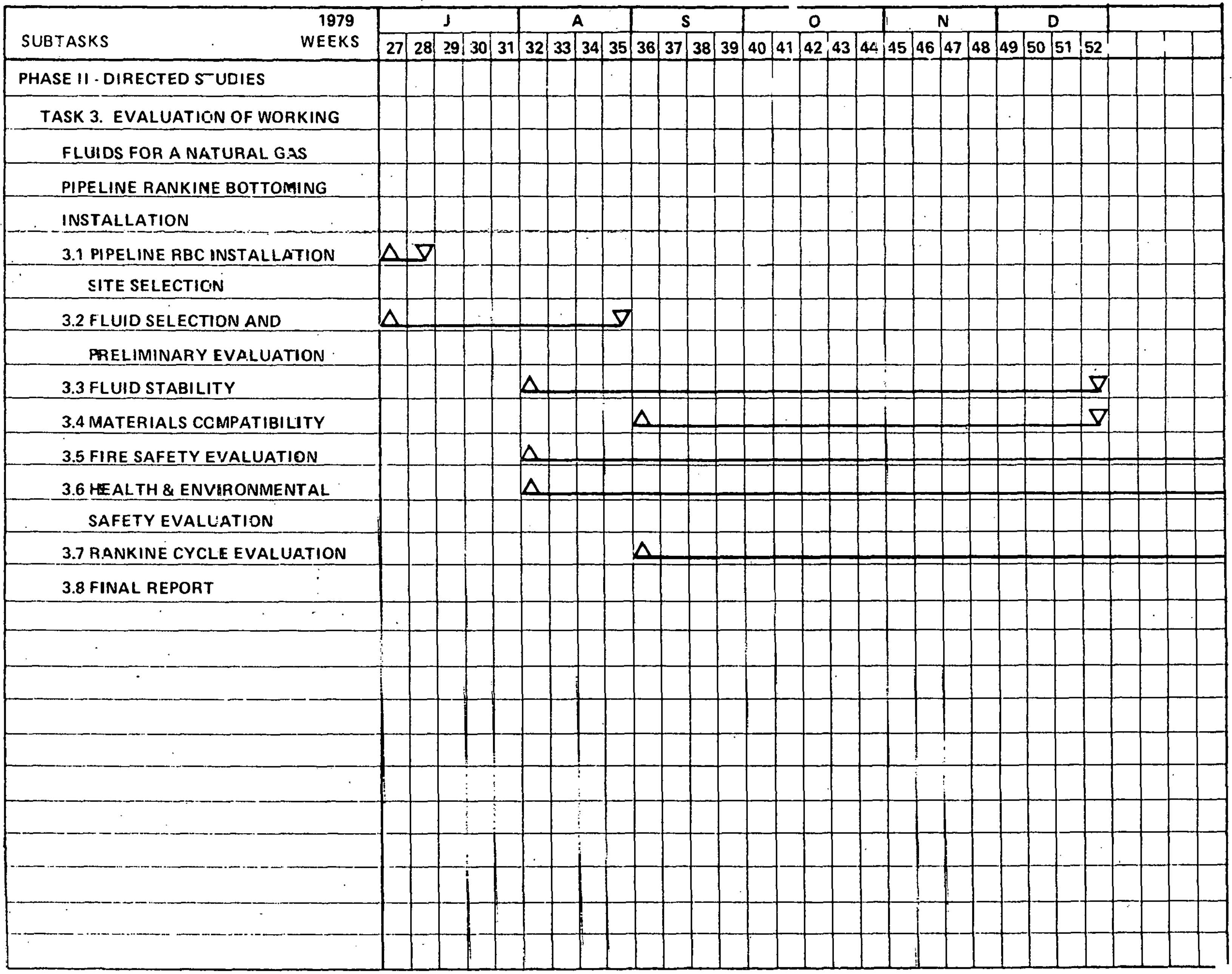
DOE PIPELINE BOTTOMING CYCLE PROGRAM SCHEDULE



DOE PIPELINE BOTTOMING CYCLE PROGRAM SCHEDULE

SUBTASKS 1979	J				J					A				S				O					N				D			
WEEKS	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52
PHASE II - DIRECTED STUDIES																														
TASK 2. REGULATORY IMPACTS ON PIPELINE BOTTOMING CYCLE APPLICATIONS																														
2.1 DESCRIPTION & ANALYSIS OF MARKET SEGMENTS & ENVIRONMENT	△							▽																						
2.2 ANALYSIS OF THE IMPACT OF SELECTED ECONOMIC/ REGULATORY MODIFICATIONS	△																	▽												
2.3 CONCLUSIONS & SUMMARY REPORT																		△		▽								▽		

DOE PIPELINE BOTTOMING CYCLE PROGRAM SCHEDULE



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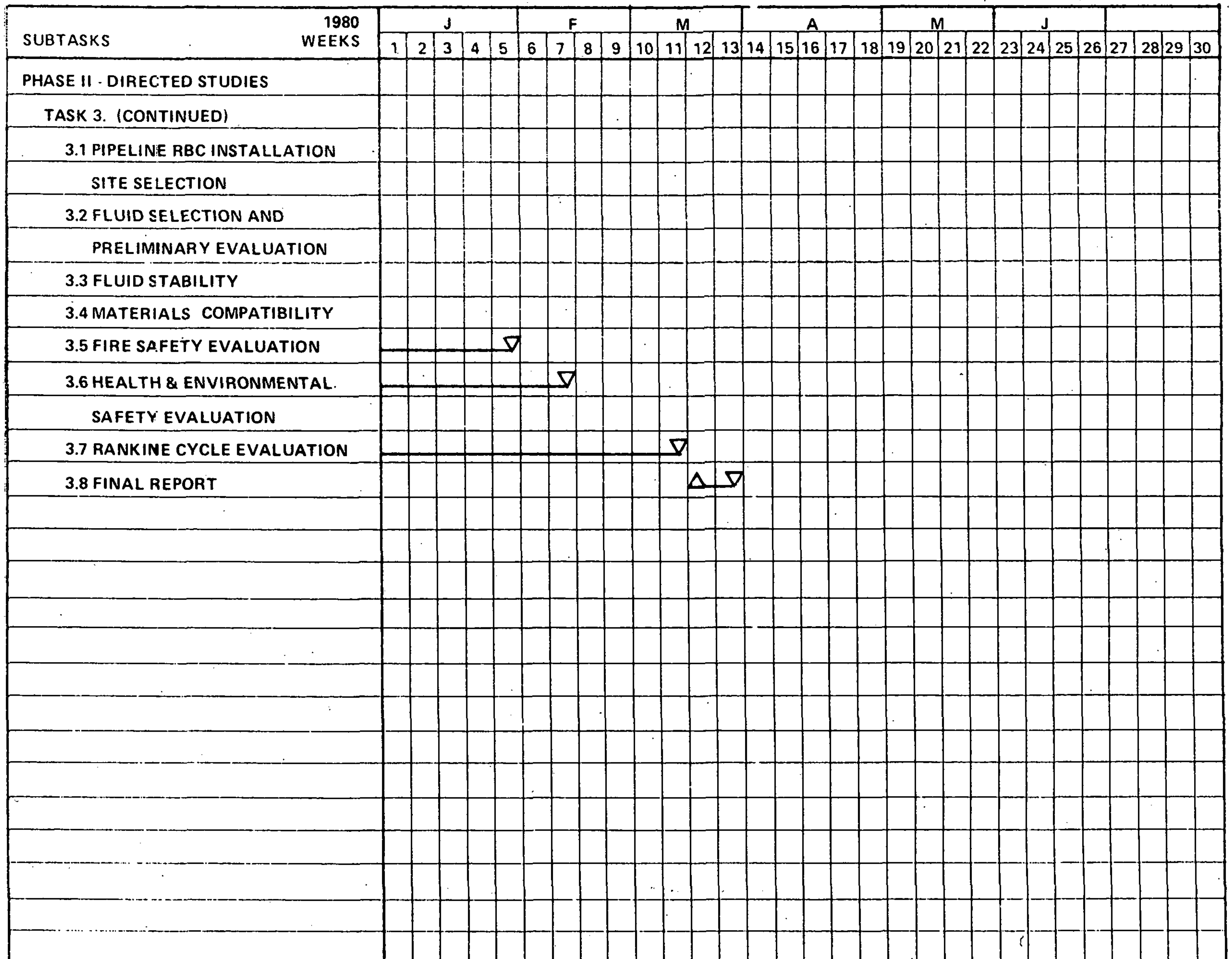

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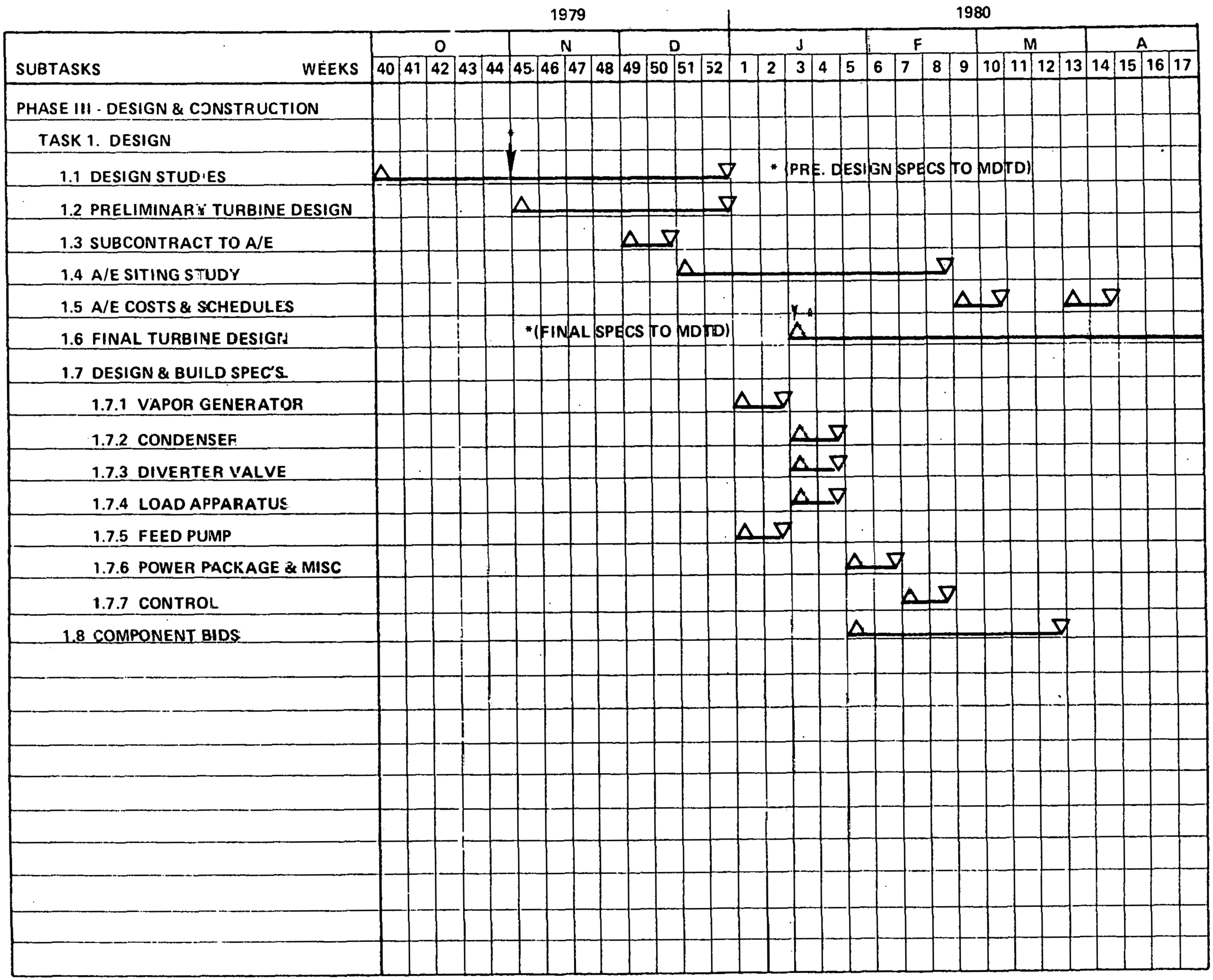

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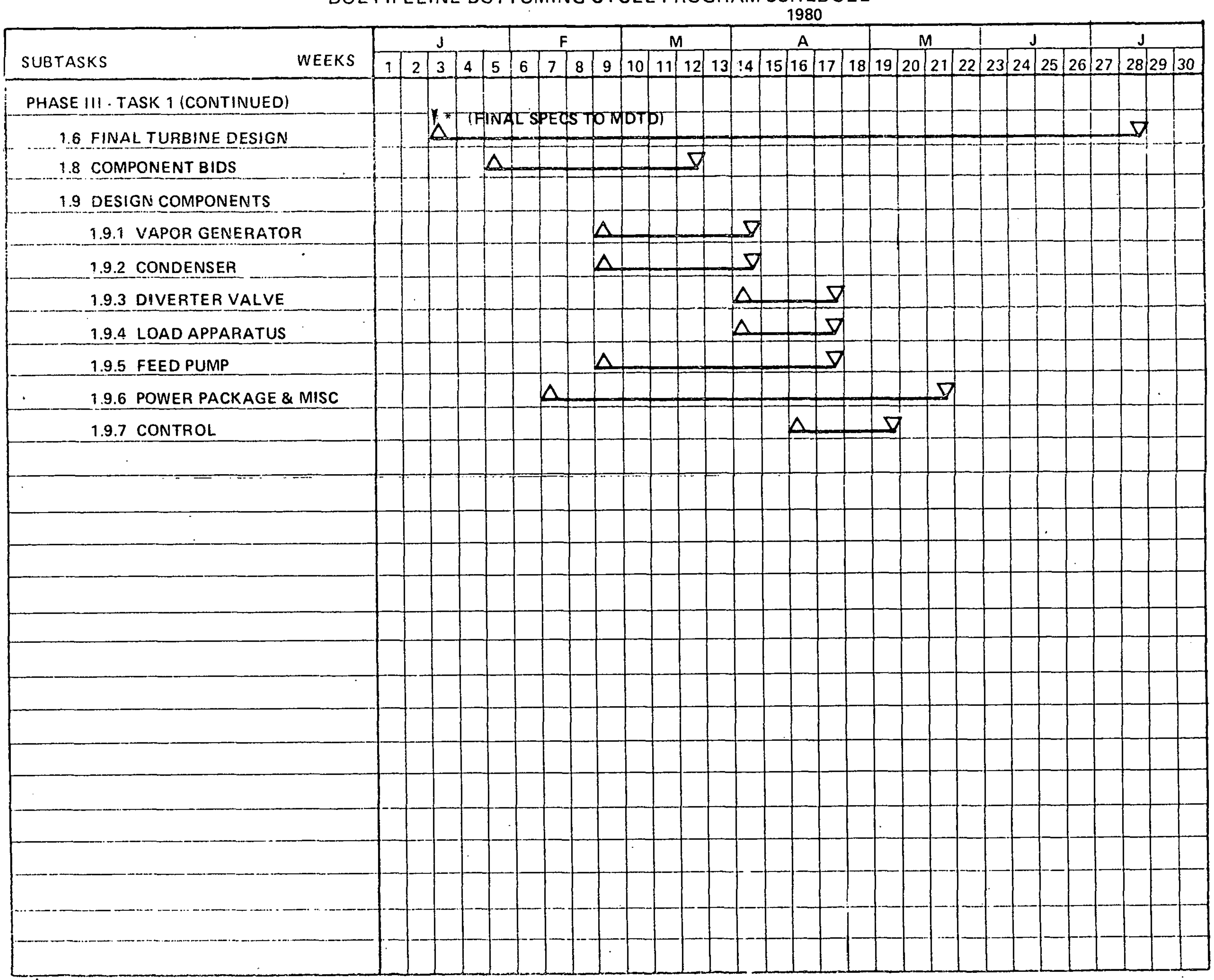
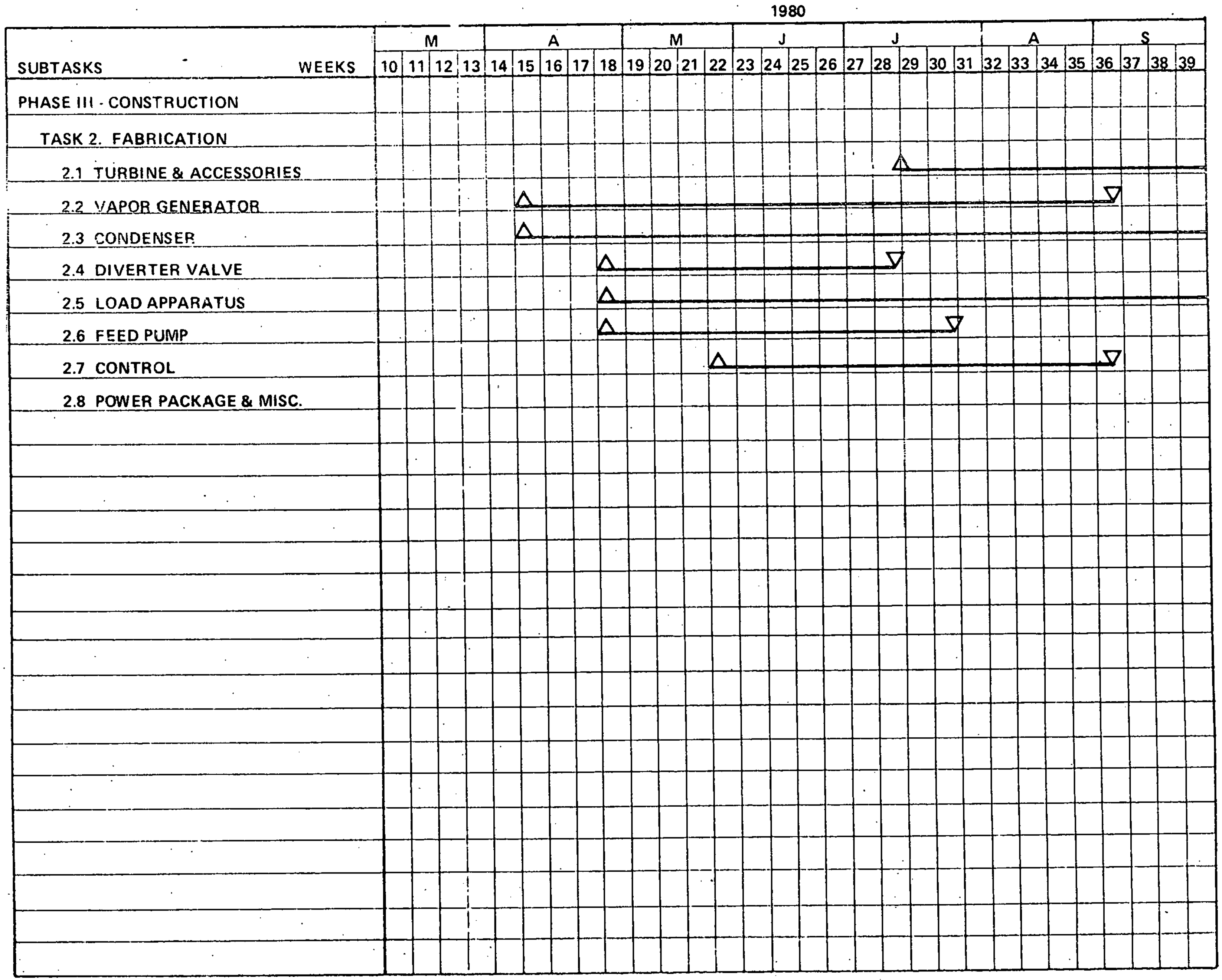


DOE PIPELINE BOTTOMING CYCLE PROGRAM SCHEDULE
1980

SUBTASKS	A					M				J				J					A				S				O			
WEEKS	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	36	38	39	40	41	42	43
PHASE III - TASK 1 (CONTINUED)																														
1.9 DESIGN COMPONENTS																														
1.10 A/E SITE DESIGN																														
1.11 SECURE INSTALLATION BIDS																														

DOE PIPELINE BOTTOMING CYCLE PROGRAM SCHEDULE



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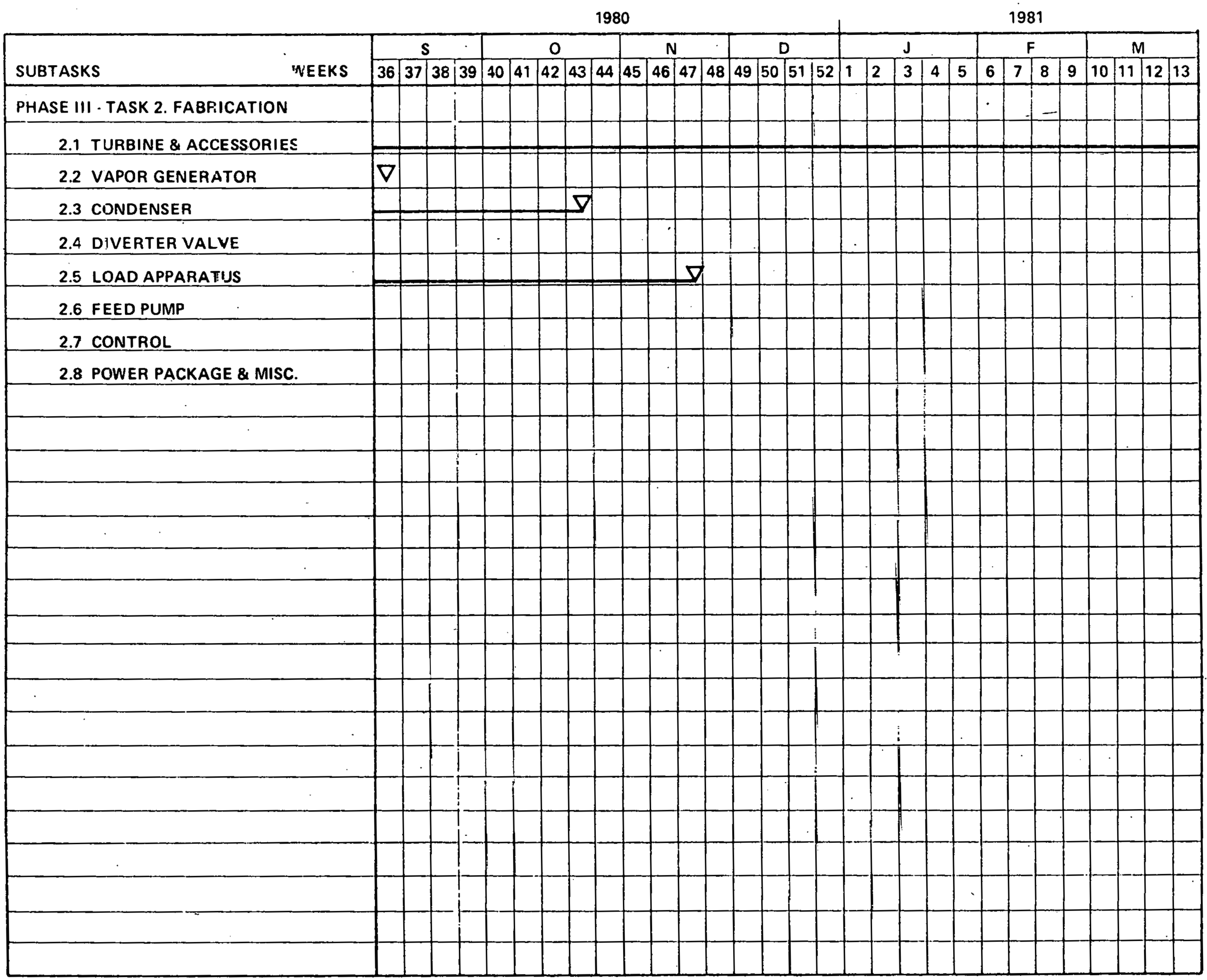

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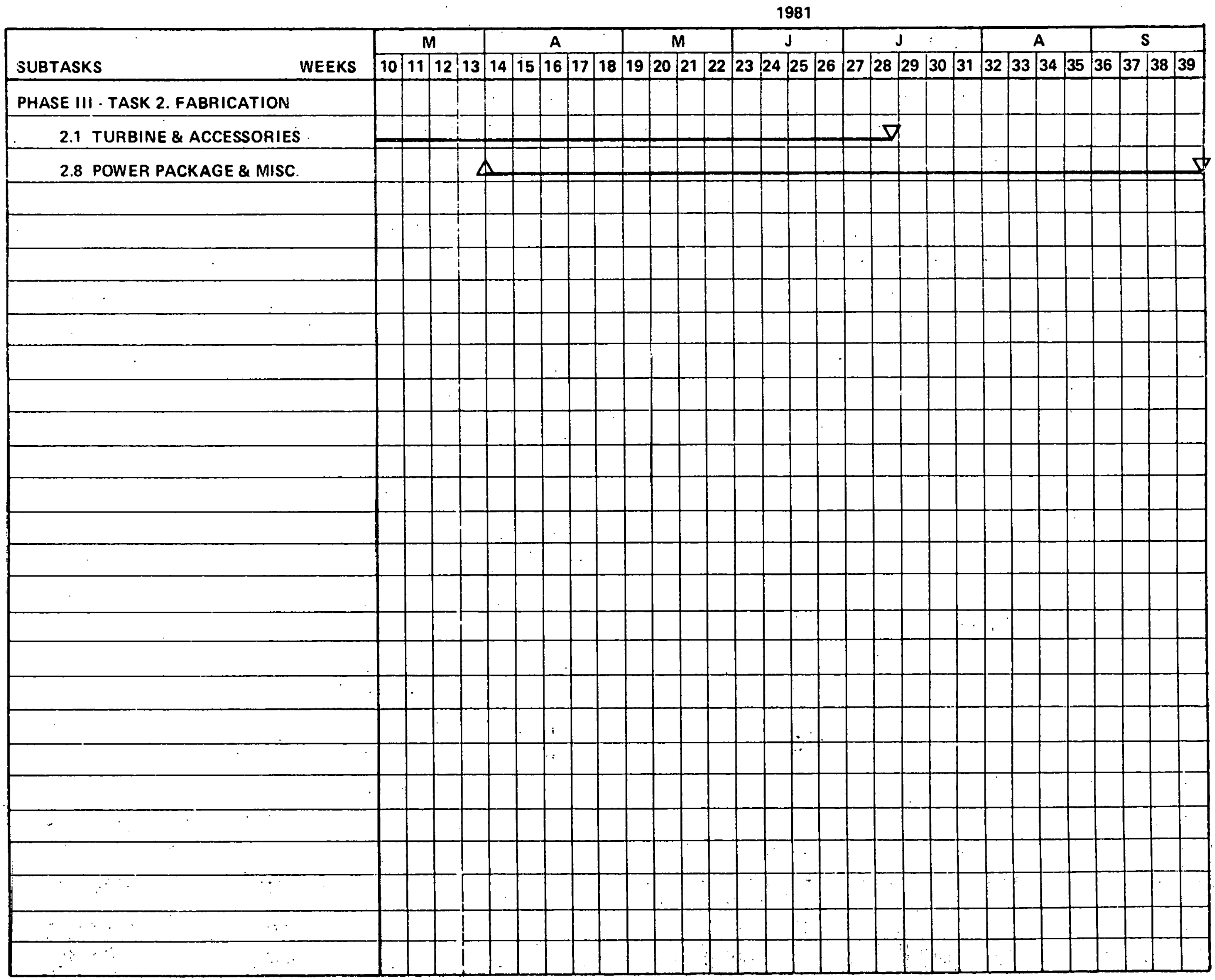

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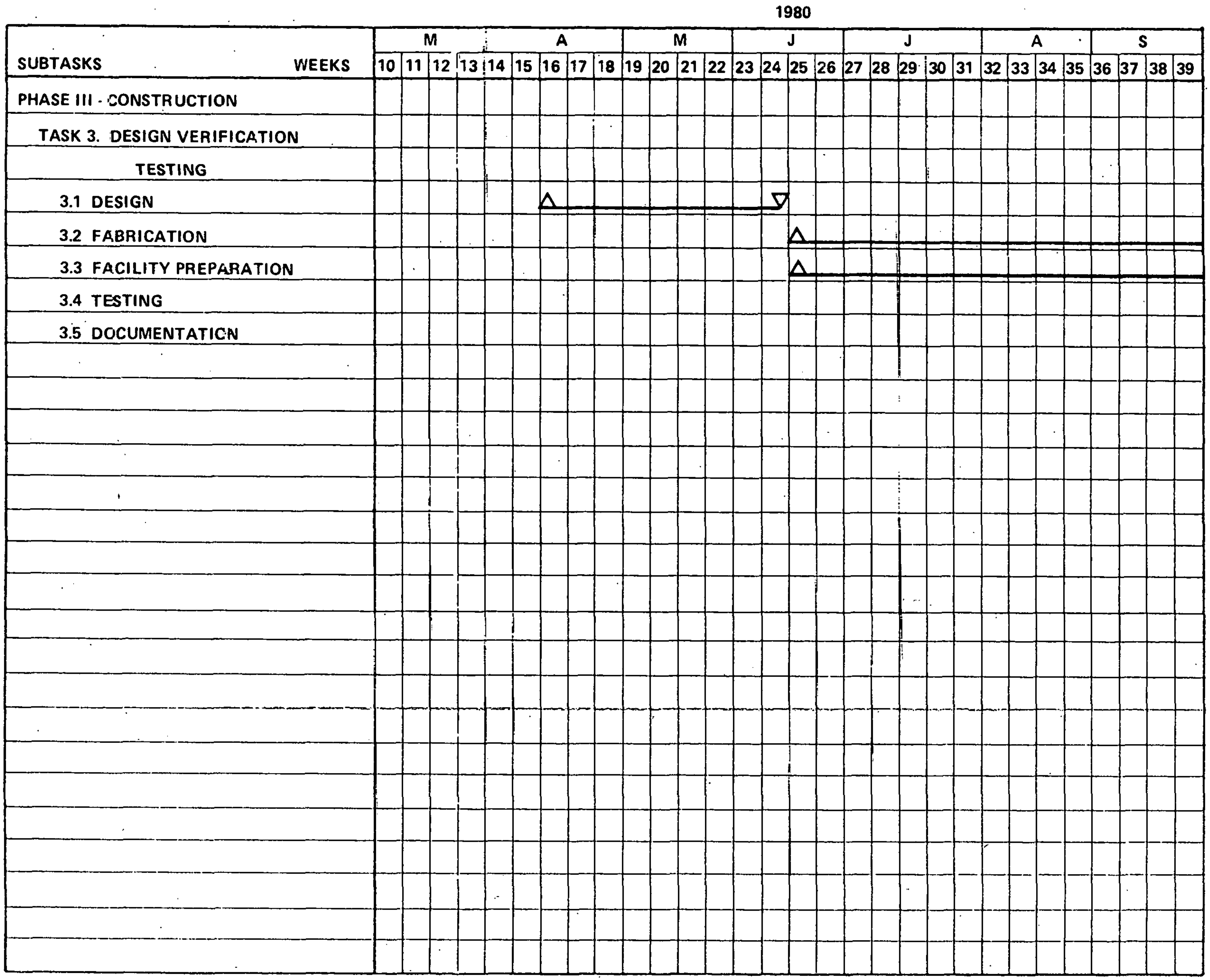

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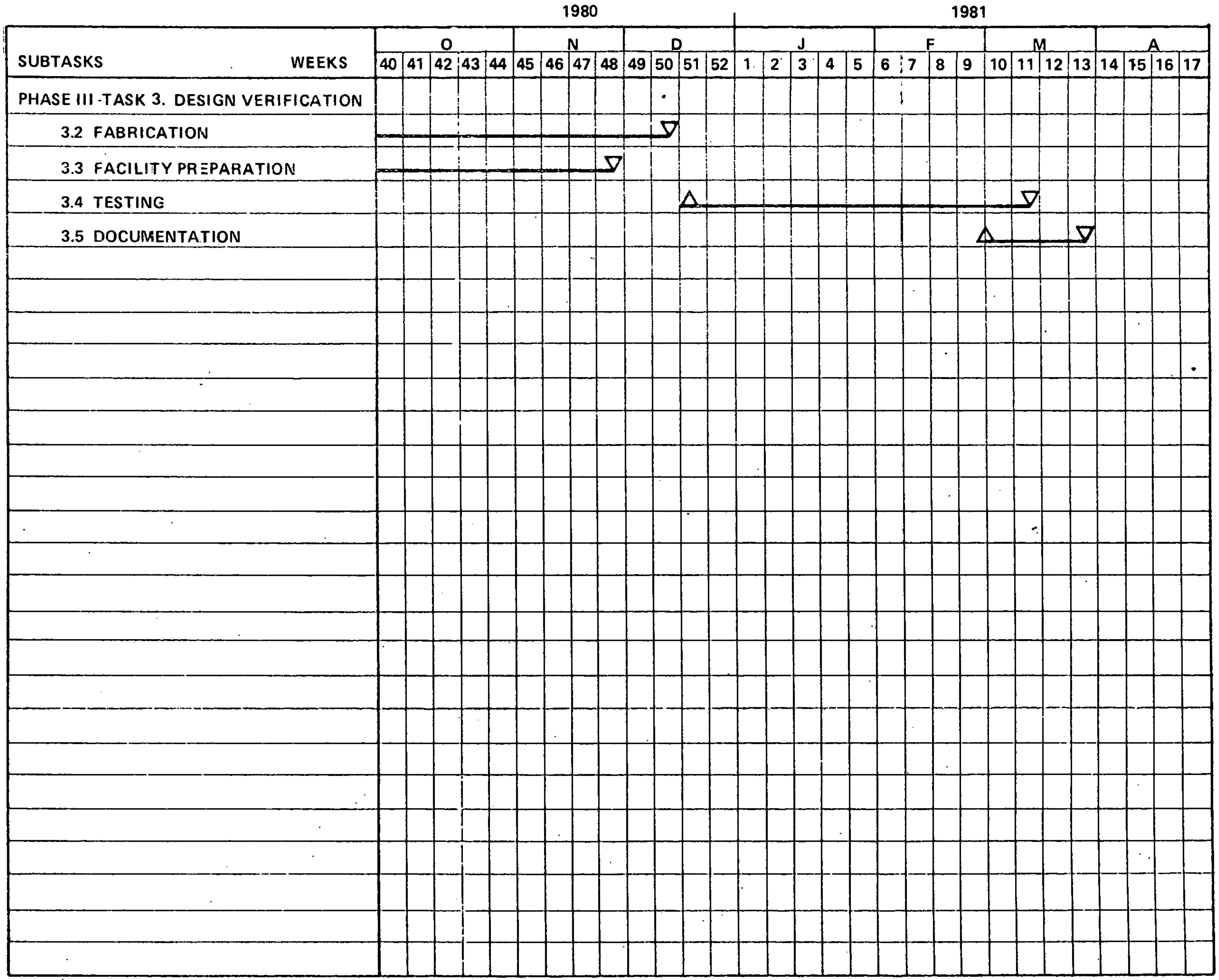
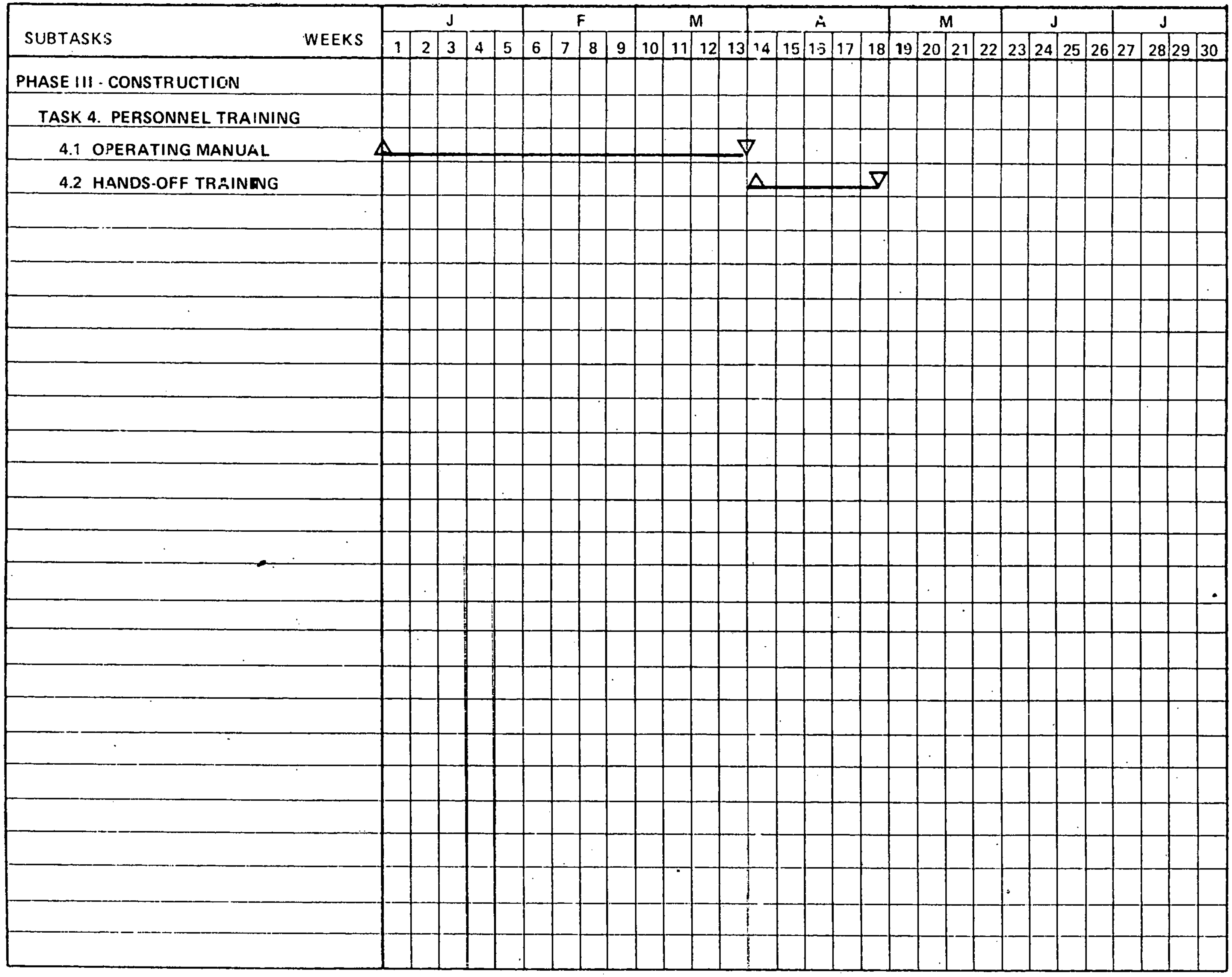


DOE PIPELINE BOTTOMING CYCLE PROGRAM SCHEDULE
1981



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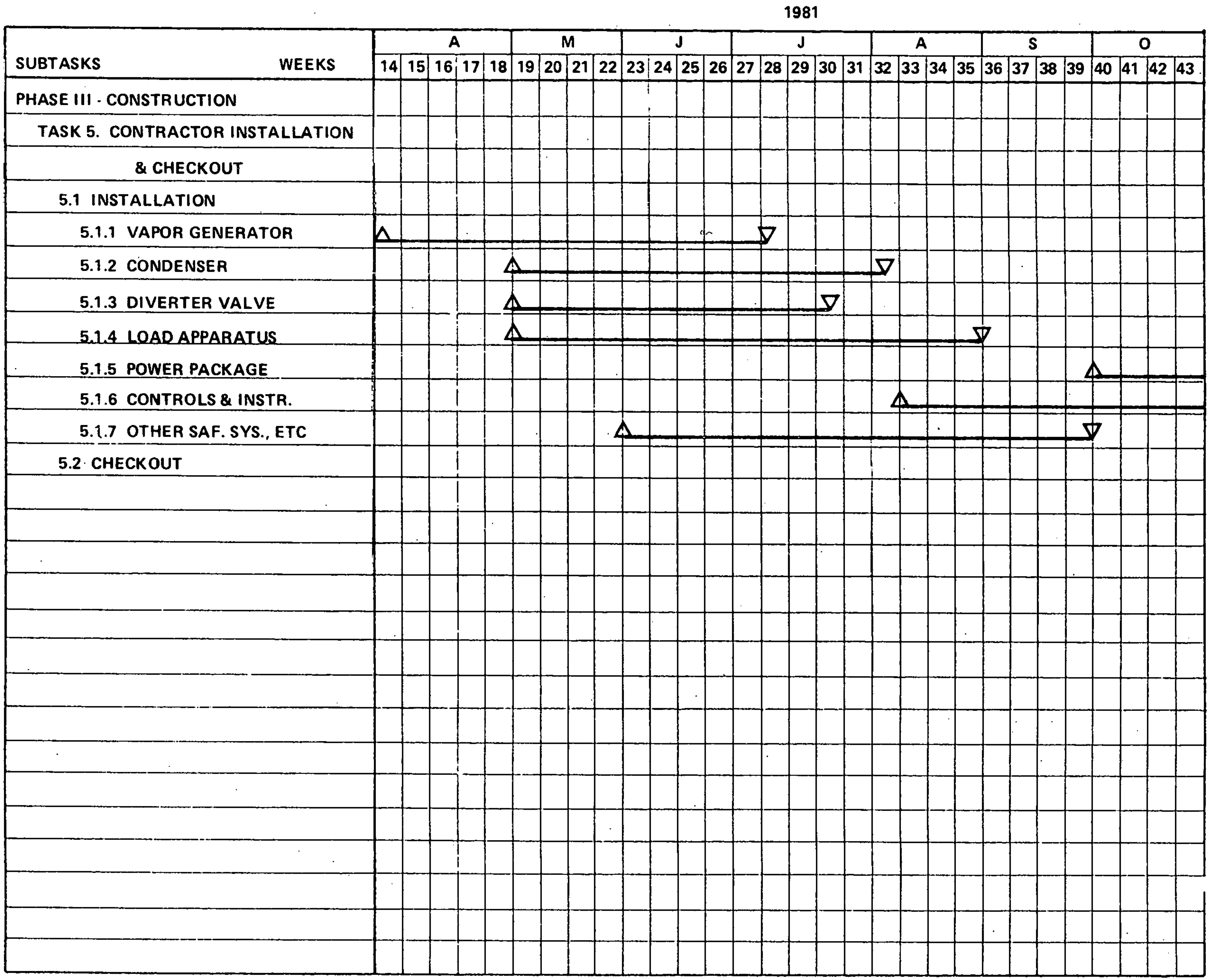

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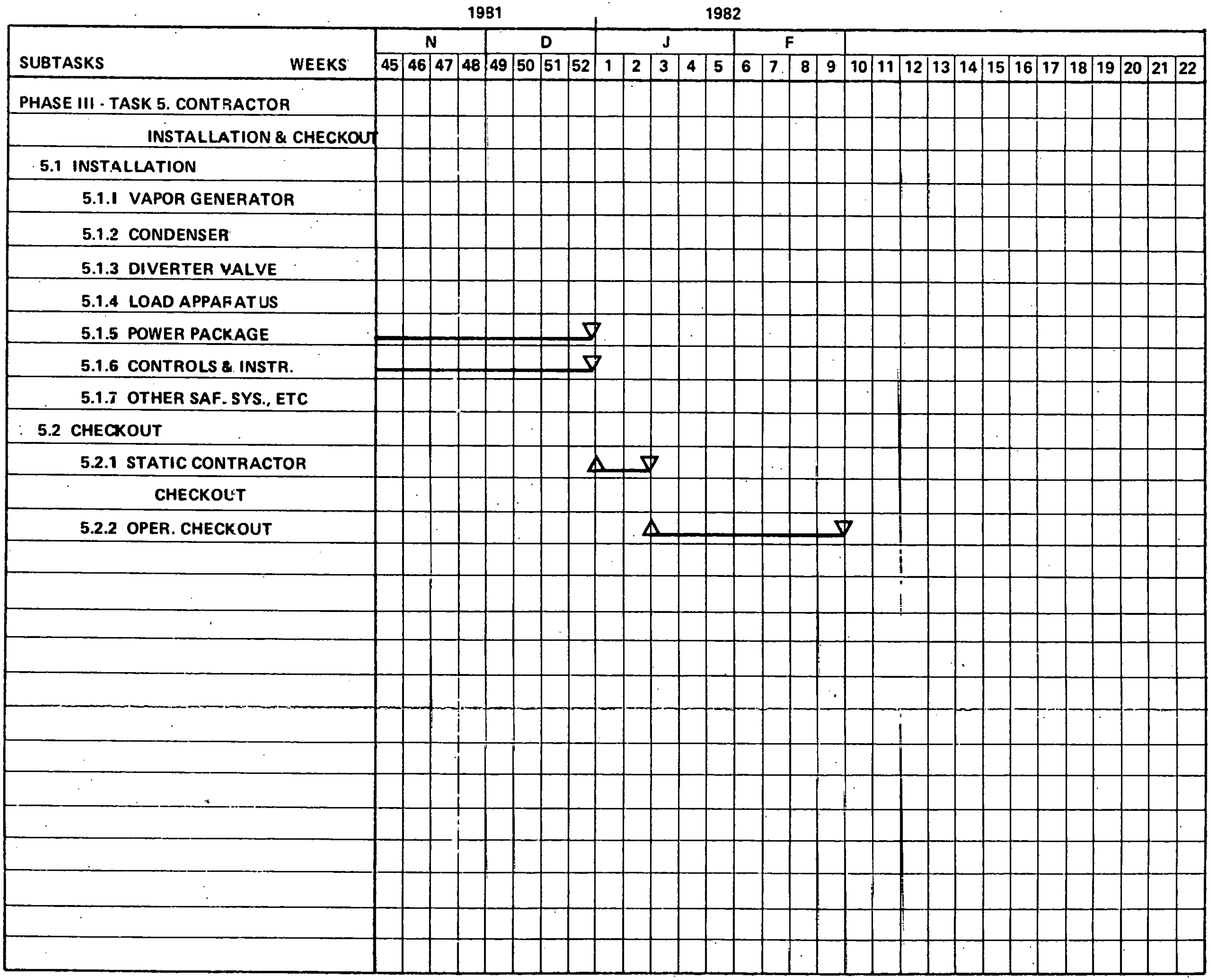
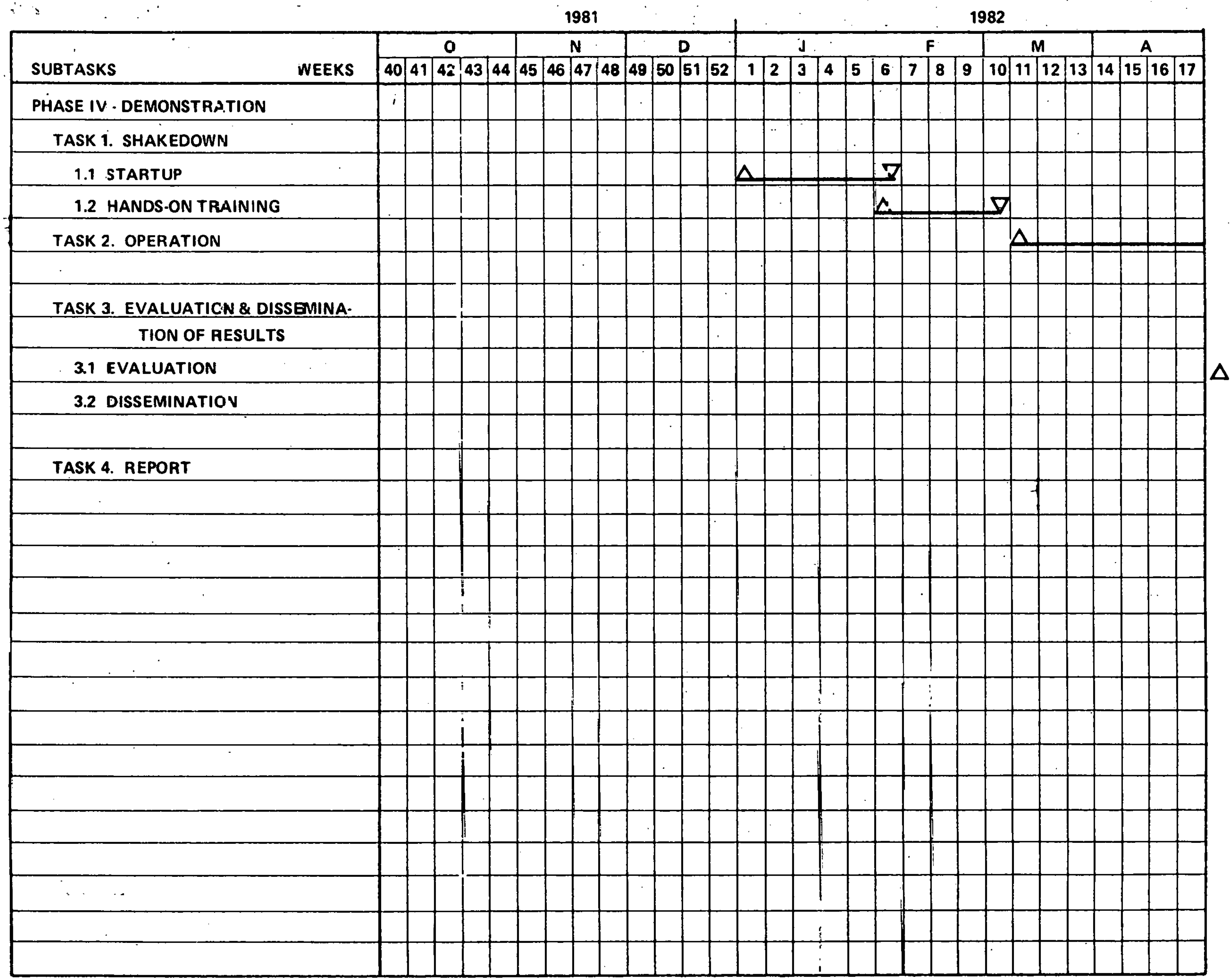


DOE PIPELINE BOTTOMING CYCLE PROGRAM SCHEDULE

	1981													1982															
	O					N				D				J					F				M				A			
SUBTASKS WEEKS	40	41	42	43	44	45	46	47	48	49	50	51	52	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
PHASE III - CONSTRUCTION																														
6. REPORT														△				▽												
7. PROGRAM MANAGEMENT THRUOUT PHASE III																														

DOE PIPELINE BOTTOMING CYCLE PROGRAM SCHEDULE

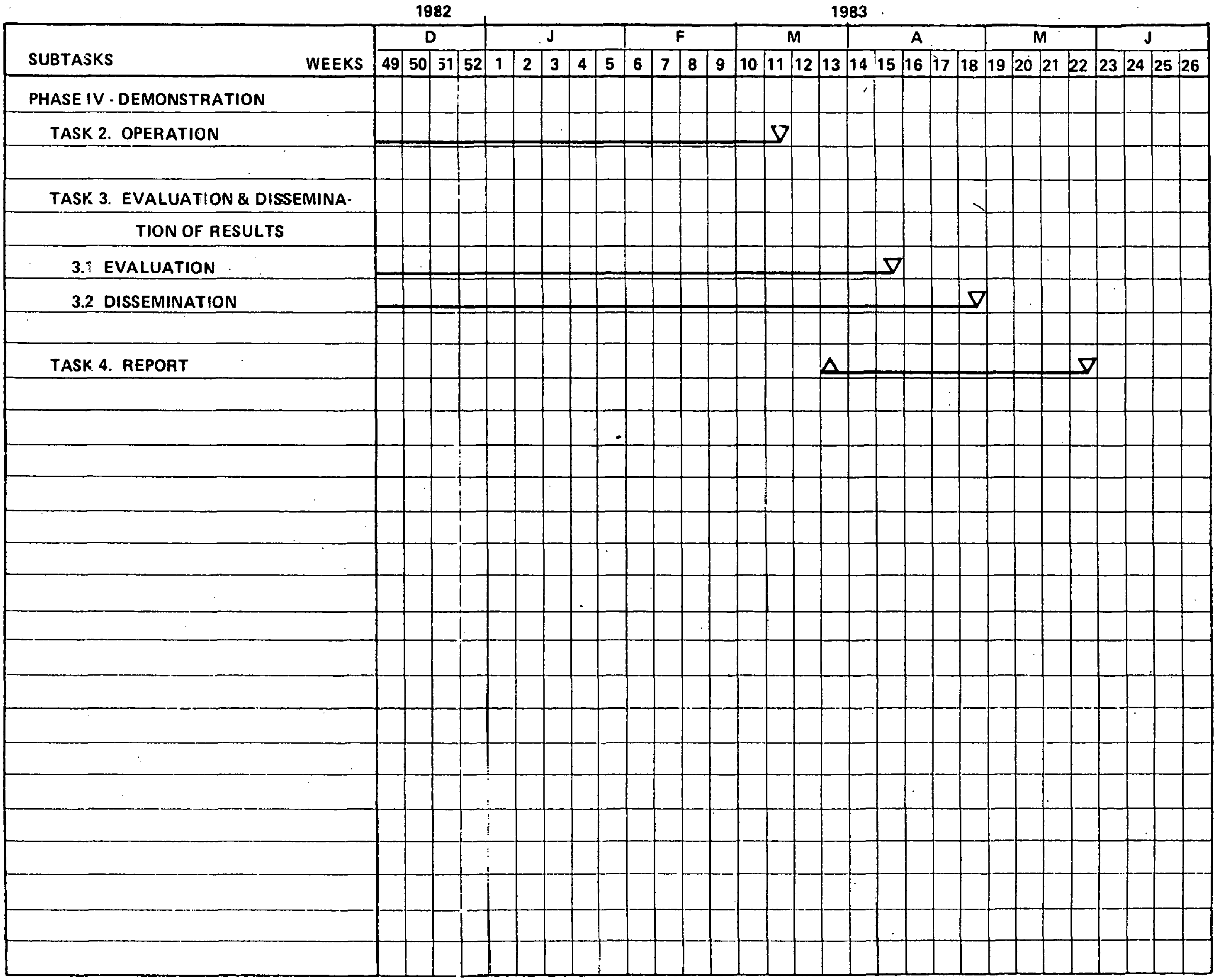


DOE PIPELINE BOTTOMING CYCLE PROGRAM SCHEDULE

1982

	M				J				J					A				S				O					N			
SUBTASKS WEEKS	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48
PHASE IV - DEMONSTRATION																														
TASK 2. OPERATION																														
TASK 3. EVALUATION & DISSEMINATION OF RESULTS																														
3.1 EVALUATION																														
3.2 DISSEMINATION																														
TASK 4. REPORT																														

DOE PIPELINE BOTTOMING CYCLE PROGRAM SCHEDULE



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