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WASTE HEAT ENERGY RECOVERY FOR NATURAL GAS COMPRESSOR STATIONS UTILIZING A GAS-LIQUID WASTE HEAT BOILER

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WASTE HEAT ENERGY RECOVERY FOR NATURAL GAS COMPRESSOR

STATIONS UTILIZING A GAS-LIQUID WASTE HEAT BOILER

By

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ABSTRACT

WASTE HEAT ENERGY RECOVERY FOR NATURAL GAS COMPRESSOR STATIONS UTILIZING A GAS-LIQUID WASTE HEAT BOILER

by

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Waste heat recovery is examined for natural gas compresser stations using three specific prime mover-compresser combinations found in the Consumers Power Company network as base cases. A total of 71.6% of the total recoverable waste energy expended (over 1.57 x 10^{12} BTU per year) at the Consumers Power compresser stations takes the form of hot combustion exhaust gases. A computer simulation of an exhaust gas waste heat boiler is utilized for design of energy recovery equipment. Steam rankine cycles using these exhaust gases as a heat source demonstrate payout periods of approximately four to ten years at 25 percent stream factor with capital costs of $425,000 to $955,000. Organic rankine cycles demonstrate an increase in thermodynamic efficiency but no significant economic advantage due to capital costs in excess of $1,560,000. Process steam production is economically favorable (with payout periods of less than two years) but requires an available outside user.

DEDICATION

I dedicate this thesis to my wife, Cindy, who provided support and encouragment throughout its preparation.

ACKNOWLEDGMENTS

Sincere appreciation is offered to the Consumers Power Company Gas Transmission and Storage Division for their assistance in compiling the necessary data and familiarizing me with energy flow within typical compressor station facilities.

In addition, Dr. Bruce Wilkinson deserves my thanks for his help, confidence and patience and I hereby express my gratitude.

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1. INTRODUCTION

One of the most heavily investigated areas of research today is the general category of energy conservation. Energy conservation is being examined at levels ranging from athletic competition through common residential heating-cooling and finally on to the frontiers of space and man's quest into the unknown. The realization that the immediately available and concentrated solar energy in the form of fossil fuels is quickly diminishing has compelled us to search for new sources of energy. Meanwhile, the available fossil fuel supply must be carefully used for both economic and basic survival reasons.

Interestingly, the energy producers and distributors of the United States are also very concerned about energy conservation. The high market value of energy today means that old or energy-inefficient processing equipment needs to be examined with a new emphasis on economics. Operating costs associated with processing equipment are often dominated by the cost of the energy needed to run it, while in previous years capital costs dominated profitability.

A good example of such processing equipment is natural gas transmission compressor stations. These stations are essential to the transport of natural gas from the production fields to almost all areas of the United States. Natural gas compression requires a large amount of pressure-volume work which is supplied by large prime mover-compressor systems. The prime movers provide the necessary shaft work to drive the natural gas compressor. Attached to the prime movers are large multi-stage reciprocating (positive displacement) compressors or banks of high centrifugal compressors. Natural gas enters these compressors at a low state of compression and must often be stepped up 400 to 600 psi to reachieve pipeline pressure. At storage facilities pressures of up to 1400 psia are achieved prior to storing natural gas in the underground facilities.

As previously stated, the compression of such large amounts of natural gas requires a great deal of pressure-volume work. This work in turn consumes huge amounts of energy. Due to the inefficiencies associated with both the prime movers and the compressors, much of this work is lost to the surrounding environment, most often as heat energy. The quantification of this waste heat and presentation of recovery schemes for such heat is the purpose of the remainder of this thesis.

2. TYPES OF WASTE HEAT SOURCES

There are basically two sources of waste heat at most compressor sites. The first is produced by drive motor inefficiency and the second by natural gas compression inefficiencies.

Drive motors (prime movers) at natural gas compressor sites are commonly very large, low rpm, two-cycle reciprocating engines utilizing natural gas as their energy source. Efficiencies for these two-cycle drivers are in the 25 to 35 percent range (7). Turbocharging or air refrigeration of such drivers boosts their overall efficiency up to approximately 40 percent (2). In addition to reciprocating drivers, less costly, smaller gas turbine drivers are also utilized. Again using natural gas as the fuel source, these drivers produce a very high velocity hot gaseous stream whose energy is converted to shaft work in a reaction turbine. Such drivers are inexpensive to install but operate at low efficiencies of between 16 and 26 percent for unregenerated low compression ratio units (2). Recent technology gas turbines can achieve efficiencies of greater than 30 percent while high compression ratio, regenerated units (state of the art) approach 40 percent efficiency (2).

Both of these drive motor types generate large quantities of waste heat in a variety of forms. The reciprocating drivers produce primarily three waste heat streams. These streams are hot exhaust gases which are often in excess of 700°F, hot lubrication oil (approximately 200°F) and jacket cooling water (approximately 150°F). The gas turbine drivers produce only two streams, the first is a high volume, high temperature exhaust gas stream (often in excess of 800°F), and the second is a hot lubrication oil stream.

In addition to driver inefficiency as a source of waste heat, the compression inefficiency of both reciprocating and centrifugal gas compressors is also a significant energy source. Compression efficiencies of 60 to 70 percent are common for these types of compressors with the waste energy appearing primarily as heat. Between each stage of compression, the natural gas must be cooled back to near ambient conditions in intercoolers or aftercoolers. These coolers are large air to gas heat exchangers (fan forced convection) operating at approximately 200°F. All natural gas requiring multistage compression passes through these coolers, thereby providing a large source of waste heat for recovery.

3. IMPORTANCE OF TEMPERATURE QUALITY

Temperature quality is the single most important factor in determining the _usability_ of any waste energy stream. Low temperature streams are common at compressor stations but have a number of disadvantages in utilization:

1. Lower overall quantity of energy per stream unit.
2. Low condensing temperatures needed in rankine cycles.
3. Lower temperature driving force requires more surface area in heat transfer equipment.
4. More sophisticated working fluids and processing equipment needed in rankine cycles.
5. Rankine cycle efficiencies are quite low.

Low temperature streams not only provide less recovery potential but also require more sophisticated equipment to derive usable benefit from them.

High temperature streams, on the other hand, are also abundant at compressor stations. These streams have the following advantages:

1. Large quantities of waste energy per stream unit.
2. High temperature driving forces.
3. More flexibility in the equipment required and recovery methods available.
4. High condensing temperatures possible.
5. Steam rankine cycles with no unusual processing equipment needed are possible.

6. High cycle efficiencies for rankine cycles.

Fortunately, a major portion of the waste heat available at natural gas compressor stations is of the latter type. This allows high cycle efficiencies and large quantities of waste heat to be recovered. Table 1, presented in Section 6, gives a breakdown of the types and quantities of waste heat available in a typical compressor network.

4. PROBLEMS INHERENT TO COMPRESSOR STATIONS

Natural gas compressor stations pose a few unique problems in designing heat recovery schemes for them. The most important and limiting factor associated with these stations is the timing of their active operation periods, and of course, their overall stream factors. Most natural gas compressor stations operate actively (actively meaning operation as a compressor station as opposed to the "bleeding" of natural gas from storage fields) from mid-summer through late fall. During this period natural gas from production facilities is moved into storage facilities in preparation for the winter heating season. This timing of the active period rules out space heating as a viable alternative. In addition, the sporadic character of operation during this period makes the coordination of energy recovery and consumer usage thereof a difficult problem. Finally, the overall stream factor of any compressor station site seldom exceeds 25 percent. Low stream factors detract from overall profitability due to decreased revenues and the large fixed capital costs associated with any design. These three factors combine to create a high volume source of energy which is sporadic, short-term, and unfortunately, less than optimally timed.

An additional set of complicating circumstances is that natural gas compressor stations are inherently an area where a great deal of fire safety precaution is observed. Due to their potential of explosion and fire they are often placed in outlying areas away from large population concentrations. In addition, heavy industrial energy users are usually not in close proximity to these stations. Most energy recovery schemes are therefore limited to those which are fire safe and are transmittable over moderate distances to potential end users.

Finally, natural gas compressor stations are not large energy consumers in any area other than the production of shaft work. Process steam, hot gaseous streams or heated jacket cooling water is of no value in itself to a natural gas compressor station. For this reason, again, any recovery scheme suggested must produce internally usable shaft work or identify a cooperative, readily available outside user interested in hot process streams.

5. CASE STUDY OF EXISTING COMPRESSOR NETWORK

The Consumers Power Company of Jackson, Michigan, operates a network of eight natural gas compressor stations for the purpose of storing, transporting, and distributing natural gas to its customers throughout Michigan's lower peninsula. During the 1977 operating year these eight stations handled over 565 billion standard cubic feet of natural gas and in the process consumed 2.118 billion SCF of natural gas as fuel (14). Approximately 81 percent of the energy contained in this fuel gas was lost to the environment as waste energy.

The eight compressor station sites are fed via two interstate pipelines: the Panhandle-Eastern Pipeline connecting at the Freedom Station, and the Michigan-Wisconsin Pipeline connecting at the White Pidgeon Station. Through these two lines natural gas from out-of-state is brought into Michigan. In addition to the natural gas piped in from other states, numerous gas production and casing-head gas production fields in Michigan supply the network. Finally, Synthetic Natural Gas from the now "mothballed" Marysville Reforming Plant is an additional source of supply.

Five of the eight compressor stations--Ray, St. Clair, Overisel, Muskegon River, and Northville--operate as both gas transmission and storage facilities. Gas storage is accomplished via the use of dormant oil fields or geologic formations. During the summer months when demand on the interstate pipelines is low, natural gas is pumped into these fields under pressures of 1000 psig or more. By early fall the storage fields are "topped off" in preparation for the winter ahead. During the winter, natural gas demand is met by use of both the interstate gas pipelines and the bleeding of natural gas from the numerous storage fields.

The three remaining sites are transmission-only stations. Two of these three are the previously mentioned White Pidgeon and Freedom stations which are pipeline origin sites. Natural gas entering the pipeline at these stations is at a low state of compression and, therefore, requires recompression in order for it to be transmitted throughout the state. As pipeline origin stations all of the natural gas entering the network from outside the state must pass through one of these stations. For this reason, the Freedom Compressor Station (at the connection with the Panhandle-Eastern Pipeline) is in almost constant operation. The White Pidgeon plant operates some six months of the year. The final site, the BTU Stabilization Plant, is a production and air injection facility. The value of this well-planned system is realized each winter as Michigan's residents seldom see a natural gas shortage, even during peak demand months.

Waste energy is generated by four major types of streams at these compressor sites. Of greatest significance are the waste streams produced by compressor drive motors as hot exhaust gases, hot lubrication oil, and, finally, jacket cooling water. In addition, to the drive motor inefficiencies, waste heat is produced via compression inefficiencies. After each stage of natural gas compression, intercoolers or aftercoolers are necessary to cool the natural gas back to near ambient conditions. Approximately 1.57×10^{12} BTU of waste energy was lost to the environment in the 1977 operating year due to these four types of waste streams.

6. RESEARCH METHODOLOGY AND RESULTS

The approach used for investigation of this project was as follows:

1. Utilizing an existing compressor network, perform an overall energy balance on the compressor stations in an effort to identify possible energy recovery sources.

2. Identify those energy sources which are most promising (i.e., high temperature and large quantity).

3. Identify typical drivers to be used as base cases for discussion.

4. Prepare preliminary cost estimates for the base cases identified and designs for energy recovery.

5. Perform economic analyses in an effort to optimize designs.

6. Apply these analyses to the specific case of the Consumers Power network and, more generally, to the driver type.

Table 1 lists the eight sites in the Consumers Power Network and includes information as to the number of and types of drivers at each site. Also included for the three primary waste stream types are waste stream temperatures, identities, and quantities for the 1977 operating year. Finally, for comparison purposes, a percentage breakdown of the waste energy produced for each of the stations is included (for three major waste stream types).

Of the thirteen driver models listed in Table 1, three primary driver units were isolated as base cases around which all

TABLE 1.--Summary of Driver/Compressor Waste Energy for 1977.

Station	Driver	Type GT or RC	Quantity	Thermal Efficiency	Horsepower Rating Each	Stream Factor *	Total Hp - Hrs x 10^3	Total Fuel Gas (mmcf)	Stack Waste			Jacket Cooling Waste			Lube Oil Waste			Compression Inefficiency Waste		
									Quantity x 10^{10} BTU	Temp. (F°)	% of Total	Quantity x 10^{10} BTU	Temp (F°)	% of Total	Quantity x 10^{10} BTU	Temp. (F°)	% of Total	Quantity x 10^{10} BTU	Temp. (F°)	% of Total
RAY No. 1	Pratt-Whitney GG-12	GT	4	.166	2750	.160	15,507	249.7	17.77	840	15.73	--	--	--	1.28	170	14.75	1.20	190	7.3
No. 2	Pratt-Whitney GG-12	GT	6	.164	2750	.160	28.182	378.6	27.01	840	23.92	--	--	--	1.95	170	22.46	1.79	190	10.9
OVERISEL	Clark TLA-8	RC	4	.356	2700	.308	28 611	220.8	9.57	665	8.48	2.60	154	13.73	.83	170	9.56	2.22	190	13.5
FREEDOM No. 1	Clark BA-5	RC	6	.248	1000	.052	2,188	24.5	1.24	690	1.09	.34	160	1.78	.11	170	1.28	.17	190	1.0
	Clark BA-6	RC	2	.249	1200	.355	6,017	67.2	3.40	690	3.01	.92	160	4.87	.29	170	3.34	.47	190	2.9
No. 2	Clark BA-5	RC	4	.218	1000	.147	4,140	52.7	2.77	690	2.45	.75	160	3.97	.24	170	2.76	.32	190	1.9
	Clark TLA-10	RC	1	.316	3300	.788	17,709	155.6	7.16	700	6.34	1.95	154	10.27	.62	170	7.14	1.37	190	8.4
MUSKEGON RIVER	Clark HBA-10	RC	4	.332	2600	.391	29,742	249.4	11.22	700	9.93	3.05	154	16.09	.97	170	11.17	2.31	190	14.1
	Clark TLA-10	RC	2	.332	3400	.376	19,473	163.3	7.34	700	6.50	1.99	154	10.53	.64	170	7.37	1.51	190	9.2
	Ingersoll-Rand KVG	RC	4	.249	800	.075	1,801	20.1	1.02	N/A	.90	.28	N/A	1.46	.09	N/A	1.04	.14	190	.8
WHITE PIDGEON No. 1	Cooper-Bessmer W-330	RC	2	.366	3900	.262	13,349	99.3	4.36	750	3.86	1.39	170	7.37	N/A	N/A	N/A	1.03	190	6.3
	Ingersoll-Rand 38-KVT	RC	2	.392	2000	.006	177	1.2	.05	N/A	.04	.02	N/A	.09	N/A	N/A	N/A	.01	190	--
No. 2	Cooper-Bessmer V-250	RC	2	.305	2000	.096	2,329	20.8	1.00	N/A	.88	.32	N/A	1.70	N/A	N/A	N/A	.18	190	1.0
	Cooper-Bessmer W-330	RC	2	.366	3900	.148	7,770	57.8	2.53	750	2.24	.81	170	4.30	N/A	N/A	N/A	.60	190	3.8
NORTHVILLE	Clark TLA-8	RC	4	.347	2700	.059	3,757	29.6	1.30	700	1.15	.35	154	1.86	.11	170	1.28	.29	190	1.8
ST. CLAIR No. 1	Pratt-Whitney GG-12	GT	1	.171	2750	.040	920	14.4	1.02	840	.90	--	--	--	.07	170	.81	.07	190	.4
	Pratt-Whitney RC-30	GT	1	.185	3000	.041	1,058	15.3	1.06	840	.94	--	--	--	.08	170	.92	.08	190	.5
No. 2	Pratt-Whitney GG-12	GT	1	.175	2750	.039	899	13.7	.97	840	.86	--	--	--	.07	170	.80	.07	190	.4
	Enterprise HV16C6	RC	3	.295	4320	.136	12,668	114.8	5.10	1000	4.52	1.75	150	9.22	.56	170	6.45	.98	190	6.0
BTU STABILIZATION	Enterprise HV12C-4	RC	2	.338	3450	.354	20,587	169.3	7.06	1000	6.26	2.42	150	12.76	.77	170	8.87	1.59	190	9.8
		--	57	.2782	--	.181	211,884	111.95	112.95	--	100.0	18.94	--	100.0	8.68	--	100.0	16.40	--	100.0

*Stream Factor = $\frac{\text{Actual Engine - Hrs}}{\text{Maximum Engine - Hrs}}$ GT = Gas Turbine RC = Reciprocating Driver Fuel BTU Value = 915 BTU/SCF design and economics calculations were performed. These three cases, listed in Table 2, were chosen because they both encompass the majority of the waste heat available and provide the variety of stream flows and conditions necessary for comparisons. Extrapolation of these cases to similar drivers or waste streams may be carried out, paying careful attention to the assumptions made specifically for these drivers.

Five designs were investigated for the three base cases in the following logic pattern:

1. a. A steam rankine cycle producing shaft work from the waste heat was investigated for each of the three cases. The optimum steam pressure for the individual waste stream maximum temperature was chosen as suggested in Reference (7).

   b. Beginning with the highest BTU output waste heat source, economic optimization of the waste heat recovery with boiler approach temperature as the variable was carried out (at a 25 percent load factor). Load factor sensitivity was then determined at the optimal approach for this driver.

   c. Economics for the remaining two base cases were performed at 25 percent load factor and the optimal approach temperature.

2. Investigation of the Organic Rankine Cycle was performed again utilizing the highest BTU output waste heat source. Once more the load factor sensitivity was investigated for this design. In addition, the lowest BTU output base case was investigated utilizing the Organic Rankine Cycle.

3. Finally, production of 150# steam was investigated for each of the three base cases at 25°F approach temperature and various stream factors.

The following three subsections contain a brief description of each of the designs which are treated in detail in the Appendices.

TABLE 2.--Energy Balance for Driver Base Cases.

	Pratt-Whitney GG-12A	Cooper-Bessmer W-330	Enterprise HV16C-6
Shaftwork BTU/BHP	2,545	2,545	2,545
Stack Waste BTU/BHP	11,844	3,340	4,191
Jacket Cooling BTU/BHP	--	804	1,435
Lube Oil & Misc. BTU/BHP	1,035	265	456
Total BTU/BHP	15,424	6,954	8,627
BHP	2,750	3,900	4,320
	.165	.366	.295
Comments:	Gas Turbine	Two Cycle	Two Cycle
	9 Compressor and 2 Turbine Stages	Eight Cylinder	Sixteen Cylinder
	Single Stage Power Turbine 9000 RPM	Reciprocating Engine	Turbocharged
			Reciprocating Engine

Steam Rankine Cycle

Figure 1 depicts the simple steam rankine cycle concept utilized for the first three designs. High temperature exhaust gases (the waste heat source) are used to generate high pressure ($\approx$ 600 psig) steam in a waste heat boiler. This steam is expanded in a turbine producing shaft work. The low pressure turbine exhaust is condensed and the condensate recompressed to provide feed to the waste heat boiler.

The shaft work produced by the turbine is best utilized to compress natural gas since it is after all, the inefficiency of the drive motors that provided this shaft work initially. Production and immediate on-site utilization of shaft work eliminates the problems of finding a user for waste energy and, in addition, shaft work is produced only when there is an immediate need for it.

In all of the rankine cycle designs, a two driver approach was used. The waste heat from two drivers is utilized in a single turbine in order to reduce equipment costs and provide shaft work at a comparable level to that of an existing gas fueled driver. For example, using the exhaust gases from two Pratt-Whitney GG-12 gas turbine compressors to power a single Rankine Cycle produces an equal amount of shaft work to that of a single gas turbine driven compressor. This system can therefore increase the efficiency of the total system (two gas-fueled compressors and one Rankine Cycle compressor) by 50 percent. This approach could be expanded to more drivers with an increase in the complexity and a decrease in the overall usability of the system.

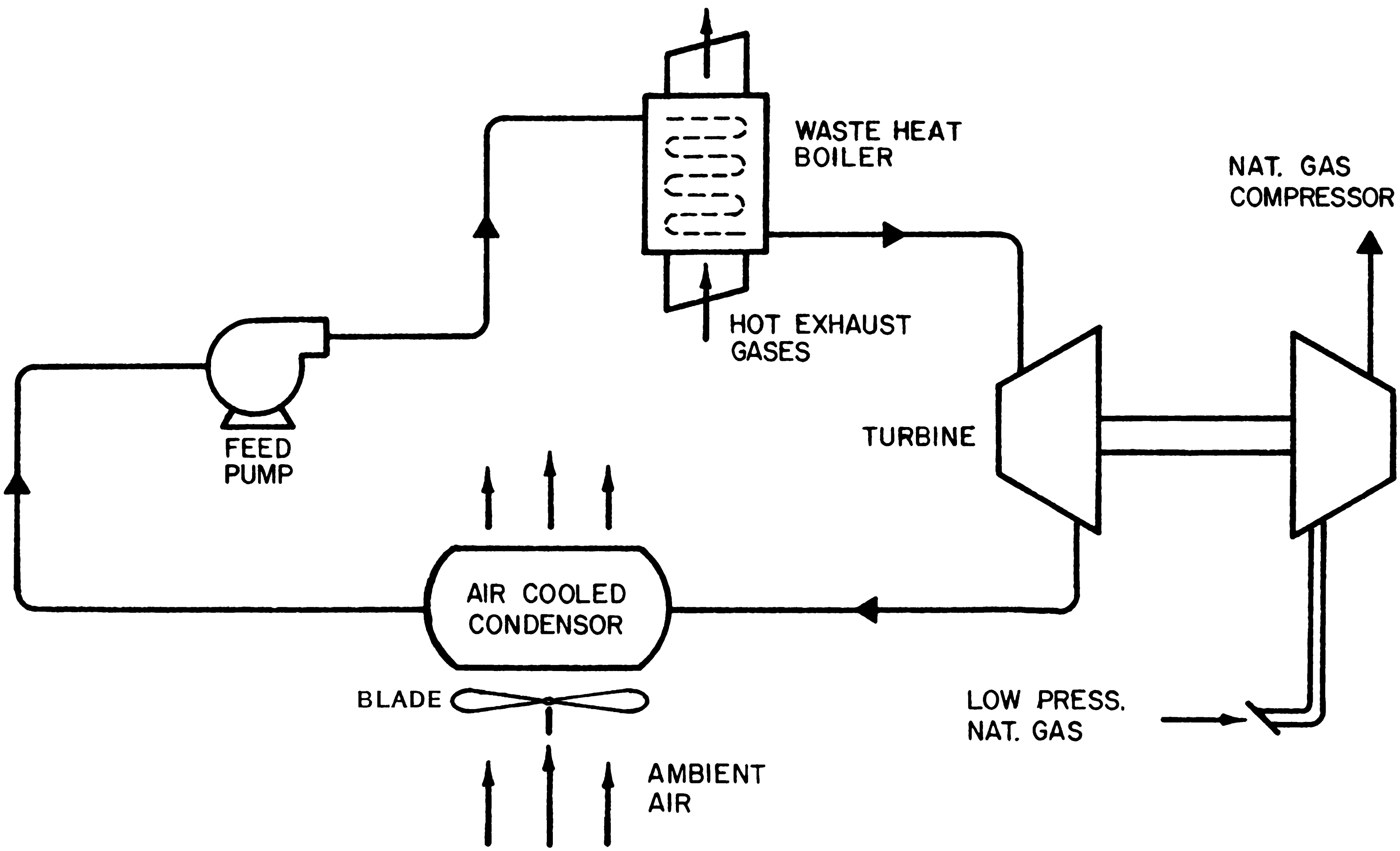

Figure 1.--Process Flow Diagram for Rankine Cycles.

The exhaust gas waste heat boiler is a key part of all designs. A detailed description of the computer simulation used for designs of this boiler and some important design considerations are found in Appendix I. Appendices II and III describe in detail the steam rankine cycle design and economics. Appendix II contains a detailed and more complete description of design considerations for the gas turbine driver (Pratt-Whitney GG-12) base case, while Appendix III summarizes steam rankine cycle designs for the two remaining base cases.

Organic Rankine Cycle

Again, Figure 1 depicts the simple rankine cycle concept utilized for this design. The only difference between the Organic Rankine Cycle and the Steam Rankine Cycle is the substitution of an organic working fluid in place of water. Utilization of an organic working fluid such as tri-flouroethanol allows attainment of a lower condensing temperature, thereby increasing the quantity of usable waste energy.

Some of the advantages and disadvantages of the Organic Rankine Cycle utilizing tri-flouroethanol are as follows (2):

Advantages

1. Low boiling point allowing low condensing temperature.
2. Low heat of vaporization allowing greater super-heating of fluid.
3. Good thermal and chemical stability.
4. Moderate expense for fluid.

5. Fluid is essentially non-toxic, non-flammable.

6. High thermodynamic efficiencies in turbine expansion (i.e., 80 to 85 percent).

7. High overall cycle efficiencies.

8. Lower cycle maximum pressures are possible.

Disadvantages

1. New technology that is not fully proved.

2. Low condensing temperature results in extremely large condensers or requires cooling towers (with 120°F condensing temperature).

3. Large organic fluid flow rates result in larger, more costly pumps, piping, etc.

Again, a two-driver approach was used for the Organic Rankine cycle. This approach takes advantage of the economy of scale and provides almost as much shaft work via waste energy recovery as is produced by the two drivers, effectively doubling their efficiency. Finally, Appendix IV contains the detailed application of this design to the Pratt-Whitney GG-12 gas turbine driver.

Process (150 psia) Steam Production

The final mode of energy recovery examined is the production of low pressure process steam.

The design and utilization of a single waste heat boiler in conjunction with feed and condensate pumps allows any drive motor the option of producing low pressure steam. The dual driver approach is unnecessary in this case, however, it may be desirable in some cases since the operation of drivers often occurs in pairs.

Saturated 150 psia steam was chosen as it is commonly used in chemical and process industry plants. In addition, commercial laundry facilities and commercial or residential heating could use 150 psia steam. Process steam is not a readily usable commodity at the compressor stations since many relatively low temperature liquid and gaseous streams already exist at these sites. The sporadic operation and remote location of most compressor stations (due to safety considerations) also make the sale of low pressure steam a less desirable alternative. Appendix V contains the detailed design and economics for production of 150 psia steam.

Six basic designs are presented in Appendices II through V with payout periods ranging from 1.0 to 20+ years. These six designs cover the spectrum of waste stream conditions found at the Consumers Power compressor stations and can be applied to similar waste streams in other situations. Appendix I contains a description of the computer simulation of a gas-liquid waste heat boiler used in design calculations.

7. DISCUSSION AND RESULTS

Examination of Table 1 (page 13) indicates the following:

1. Compression inefficiency accounts for approximately 10.9 percent of the total recoverable waste energy.

2. Some 12.0 percent of the total recoverable waste energy is rejected as jacket cooling water waste. Comparatively low temperatures of between 150°F and 170°F are common for these streams.

3. Approximately 5.5 percent of the total recoverable waste energy is rejected as lube oil cooling waste. Temperatures of less than 170°F are specified here. Waste energy sources, like numbers 1 and 2 above, are difficult to utilize cost effectively. The low heat load potential and temperature of these streams make them less than desirable as <u>stand-alone</u> sources. They must, therefore, be incorporated into a total system as pre-heaters or auxillary heat sources.

4. Some 71.6 percent of the total recoverable waste energy at the eight compressor stations is rejected as high temperature gases resulting from the combusion of natural gas. These streams are moderately clean, non-corrosive and are therefore ideal candidates for waste heat recovery. The emphasis of the remainder of this study is on the recovery and usage of exhaust gas waste energy since this is a readily available, high potential waste energy source,

Steam Rankine Cycle

The simple four-stage steam rankine cycle was applied to each of the three drivers listed in Table 2. Table 3 lists the maximum steam temperature, optimum approach temperature, and maximum cycle pressure, chosen for each driver case as well as the pertinent economic data. Figure 2 presents the payout period versus stream

TABLE 3.--Summary of Steam Rankine Cycle Design Cases.

	Pratt Whitney GG-12	Cooper Bessmer W330	Enterprise HV16C-6
Steam Pressure (psia)	600	400	1200
Maximum Cycle Temperature (F°)	800	710	960
Optimum Approach Temperature (F°)	25	25	25
Total Fixed Capital Investment (1980)	955,000	425,500	885,000
Payout Period at 25% Stream Factor (years)	3.50	>100	13.0
Payout Period at 90% Stream Factor (years)	1.2	6.50	2.9
Shaft Work Produced (Hp)	2,770	689	1,715

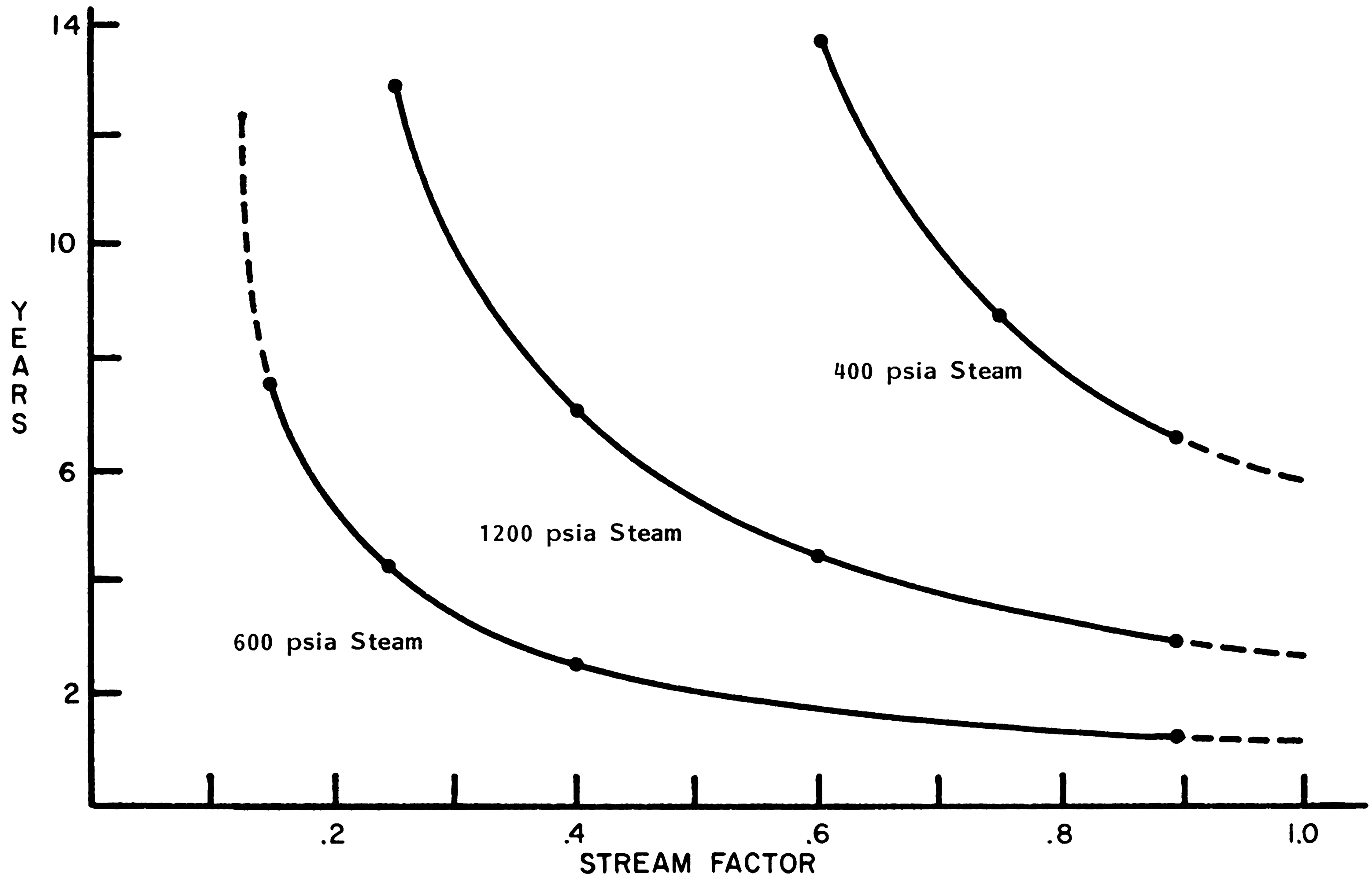

Figure 2.--After Taxes Payout Period versus Stream Factor for Steam Rankine Cycles.

factor for each of the three driver base cases. The payout period is calculated using the yearly cash flow after corporate income taxes (@ 48%) averaged over ten years of operation. All capital costs and operating costs are included as well as interest on working capital. Figure 3 presents the discounted cash flow rate of return (3) versus stream factor for the three base cases. A detailed description of individual economics calculations and assumptions may be found in Appendix II. For simplicity, discussion of the steam rankine cycle for each of the base cases is presented individually in the following three subsections.

a. Pratt-Whitney GG-12 Gas Turbine

The simple four-stage steam rankine cycle approach with direct utilization of shaft work was applied to the low efficiency Pratt-Whitney GG-12 gas turbines with encouraging results. A maximum cycle temperature of 800°F and resultant steam pressure of 600 psig resulted in the production of 2270 Hp. of shaft work for the design point.

Examination of Table 3 in conjunction with Figures 2 and 3 demonstrates that with a total 1980 fixed capital investment of $955,000 (excluding construction loan interest) payout periods of less than 2.6 years are attainable at stream factors of greater than 40 percent. Furthermore, at full capacity the minimum payout period is approximately one year. Finally, at stream factors of less than 15 percent a rapid rise in payout periods of above seven years is evident.

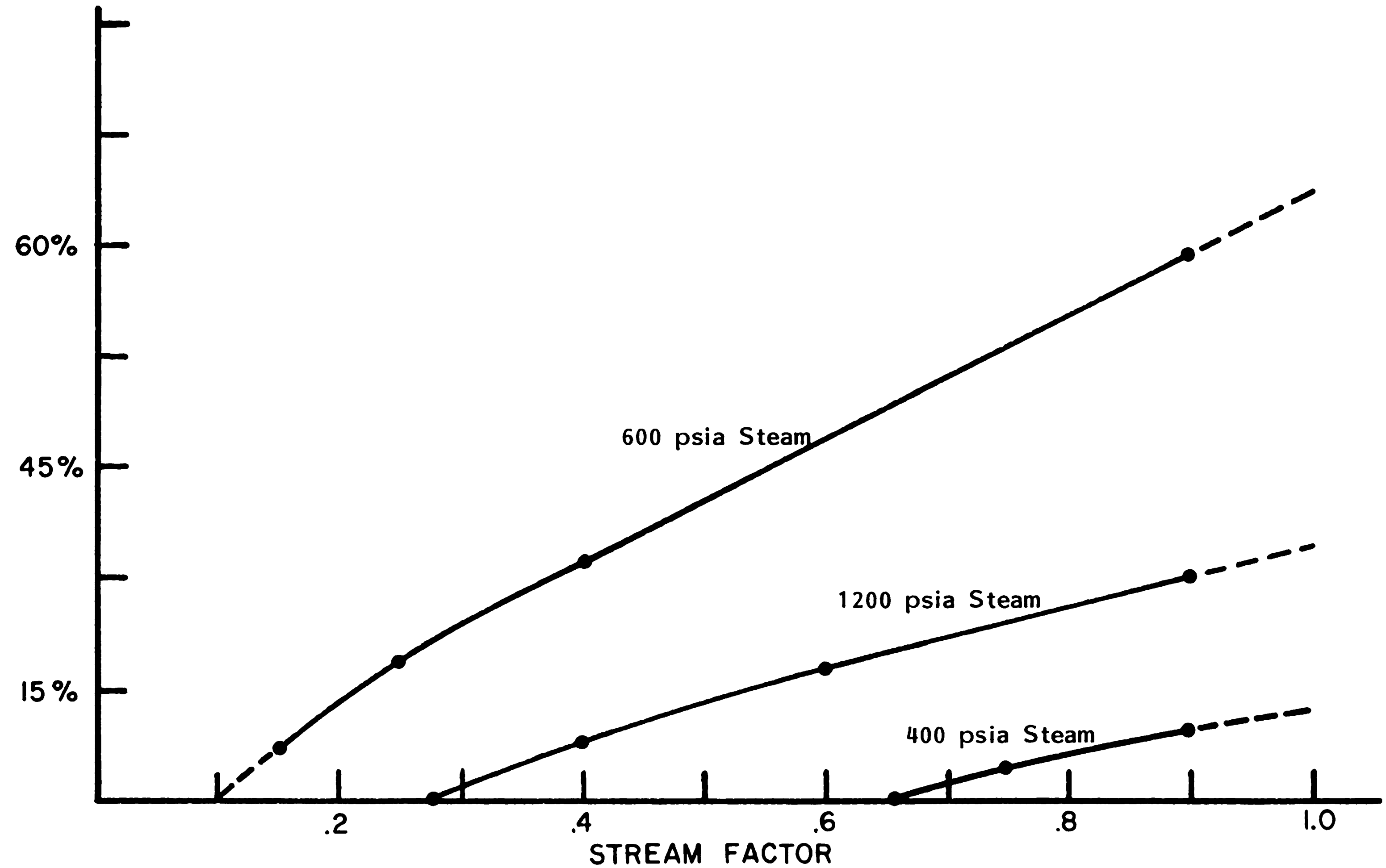

Figure 3.--After Taxes Discounted Cash Flow Rate of Return versus Stream Factor for Steam Rankine Cycles.

Four separate plants operated the Pratt-Whitney GG-12 gas turbines in the 1977 operating year. Inspection of Table 1 indicates that at two of these plants, the Ray Station Plants 1 and 2, stream factors of approximately 16 percent were encountered for the ten drivers. At 16 percent stream factor, payout periods of less than 6.6 years could be expected for any dual driver installation. However, this is the maximum payout period for such an installation assuming no load shifting. Since the Ray Station plants are two parts of the same compressor station, load shifting could allow a two-driver system with a steam rankine cycle to operate in place of the unmodified gas turbine compressors, thereby increasing the stream factor of the two-driver system. The maximum stream factor attainable by this method (based on 1977 loads) is 53 percent, yielding a payout period of two years. This requires all gas compression for the entire year to be performed by one dual driver compressor system retrofitted with a steam rankine cycle which is unlikely.

In addition, the Ray Station is particularly appealing since the ten drivers located in Plants 1 and 2 created some 40 percent of the total recoverable stack waste energy for all of the Consumers Power compressor stations in 1977. The high temperature and large quantity of waste energy generated there certainly provided a probable and profitable source for recovery.

The two remaining plants which operated Pratt-Whitney GG-12 drivers, the St. Clair Plants 1 and 2, both experienced stream factors of four percent or less in the 1977 operating year. Payout periods of greater than 15 years would be expected at such stream factors making installation of a steam rankine cycle unfavorable at these sites.

b. Cooper-Bessmer W-330 Reciprocating Engine

The simple four-stage steam rankine cycle approach with direct utilization of shaft work was applied to the moderately efficient Cooper-Bessmer W-330 drivers with marginally encouraging results. A maximum cycle temperature of 710°F and resultant steam pressure of 400 psig resulted in the production of 689 Hp. of shaft work for the design point.

Examination of Table 3 in conjunction with Figures 2 and 3 demonstrates that payout periods of less than 14 years are attainable with a total 1980 fixed capital investment of $425,500 (excluding construction loan interest) at stream factors of greater than 60 percent. A minimum payout period of six years is attainable at 100 percent stream factor. Finally, unreasonably long payout periods of greater than 14 years are attained at stream factors of 60 percent or less.

Two plant sites, the White Pidgeon Plants 1 and 2, utilized Cooper-Bessmer W-330's in the 1977 operating year. Stream factors of 26.2 and 14.8 percent, respectively, were encountered. With such low stream factors neither plant offers economically feasible operation of a steam rankine cycle. Attempts at load shifting at these two sites appears to be unfeasible since sufficient load does not exist to allow shifting. For these reasons installation of a steam rankine cycle at these sites appears unfavorable.

Although only four Cooper-Bessmer W-330's exist at the Consumers Power compressor stations, numerous other drivers are similar in operation, design and characteristics to the Cooper-Bessmer W-330's. The steam rankine cycle design and economics should be similar for these drivers and with these assumptions in mind are applied to three additional stations.

The first of these drivers is the Clark TLA-10 located at the Freedom Plant 2. This single driver operated at a stream factor of nearly 80 percent and expended 6.3 percent of the total recoverable stack waste in 1977. With such a stream factor a payout period of less than 7.8 years could be expected for the dual driver steam rankine cycle. However, a payout period of greater than 10 years is estimated since dual driver operation is not possible with this driver.

Two Clark TLA-10 drivers operate at the Muskegon River site. At their 1977 stream factor of 38 percent, operation of these drivers using a steam rankine cycle is economically unattractive. However, with load shifting at this plant, a maximum stream factor of approximately 80 percent is attainable yielding a more favorable 7.8 year payout period.

Finally, the four Clark TLA-8 drivers located at the Overisel Compressor Station operated with a 31 percent stream factor during the 1977 operation year. Again, the payout period for these drivers is unreasonably long and even by increasing the stream factor to maximum of approximately 55 percent via load shifting, a payout period of greater than 14 years would be realized. Installation of a steam rankine cycle appears economically unattractive for this site.

c. Enterprise HV16C-4

The simple four-stage steam rankine cycle approach with direct utilization of shaft work was applied to the moderately efficient Enterprise HV16C-4 drivers with marginally encouraging results. A high maximum cycle temperature of 960°F and resultant steam pressure of 1200 psig resulted in the production of 1715 Hp. of shaft work for the design point.

Examination of Table 3 in conjunction with Figures 2 and 3 demonstrates that with 1980 total fixed capital investment of $885,000 (excluding construction loan interest) payout periods of less than five years are attainable at stream factors of over 50 percent. At full capacity, a minimum payout period of 2.6 years is anticipated. Finally, at stream factors of less than 25 percent, unreasonable payout periods of greater than 12 years are realized.

Only two plants operate the Enterprise turbocharged drivers, the St. Clair Plant 2 and the BTU Stabilization Plant. Stream factors of 13.6 and 35.4 percent were encountered, respectively, for these plants during the 1977 operating year. Three 4320 Hp. HV16C drivers (the design case) are in operation at the St. Clair Plant 2 while two smaller HV12C's rated at 3450 Hp. are in operation at the BTU Stabilization Plant.

First considering the design case, at stream factors of 13.6 percent, payout periods of over 14 years would be expected for the three St. Clair plant drivers. Loadshifting could allow a maximum stream factor of 14.1 percent for the dual driver installation. At such low stream factors unreasonably long payout periods result.

Finally, considering the two HV12C's located at the BTU stabilization plant, a maximum stream factor of 35.4 percent results in a payout period of approximately 8.1 years for the dual driver unit. Due to the fact that the smaller HV12C's would not produce exactly the same economic results as the larger HV16C's, some contingency must be applied to this engineering estimation, and slightly higher payout periods would result.

Organic Rankine Cycle

The simple four-stage rankine cycle design was applied in detail to the largest potential, most economically feasible driver (the Pratt-Whitney GG-12). In addition, this design concept was also applied to the least favorable potential waste heat source (the CB-W330). A maximum cycle temperature of 625°F (recommended by Halocarbon Corp. [9]), and maximum pressure of 700 psig, resulted in the production of 4672 Hp. of shaft work for the design point. An optimal approach temperature of 75°F was determined for this design. Figures 4 and 5 present the payout period and discounted cash flow rate of return as a function of stream factor, respectively.

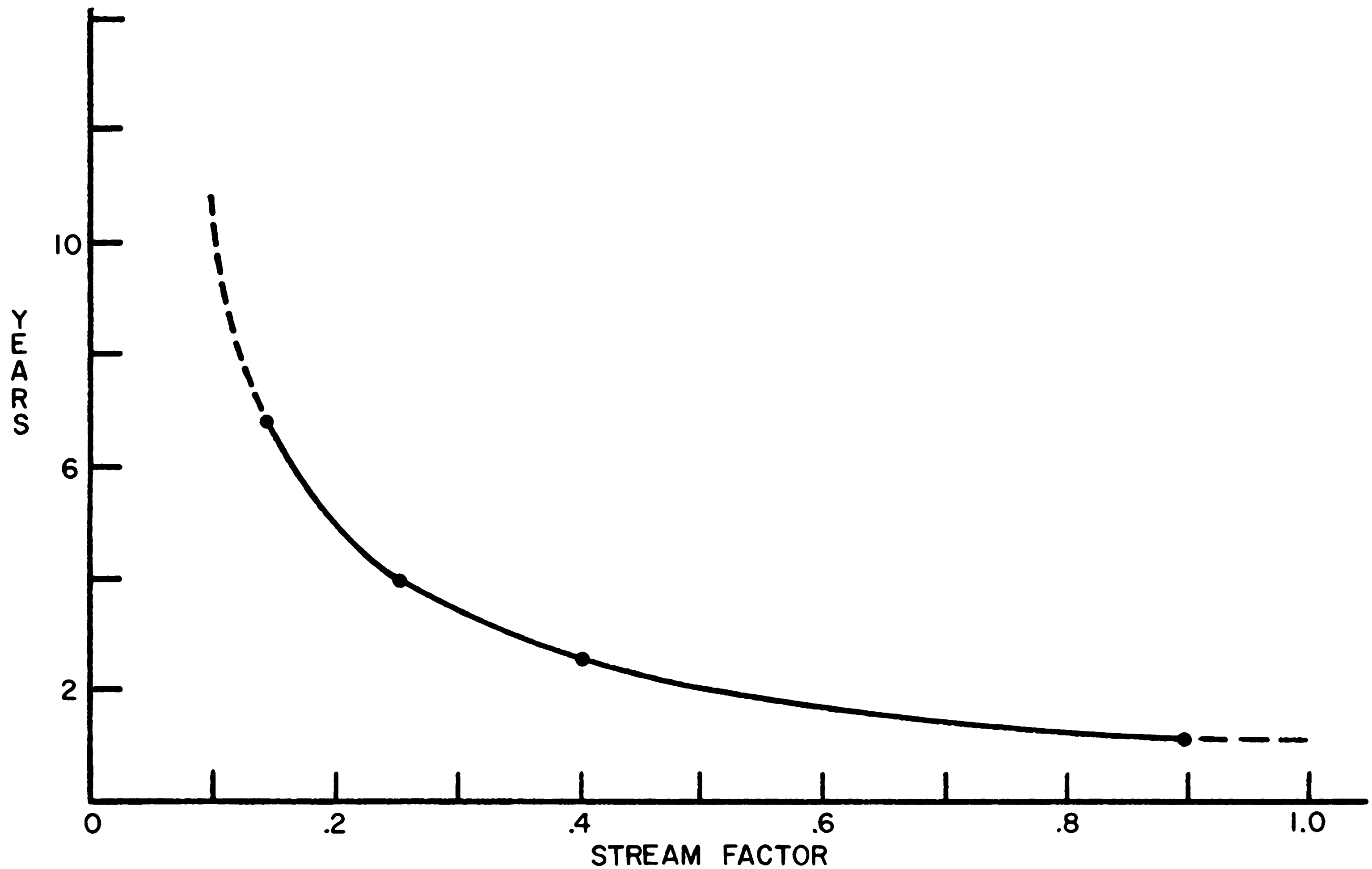

Figure 4.--After Taxes Payout Period versus Stream Factor for Organic Rankine Cycle.

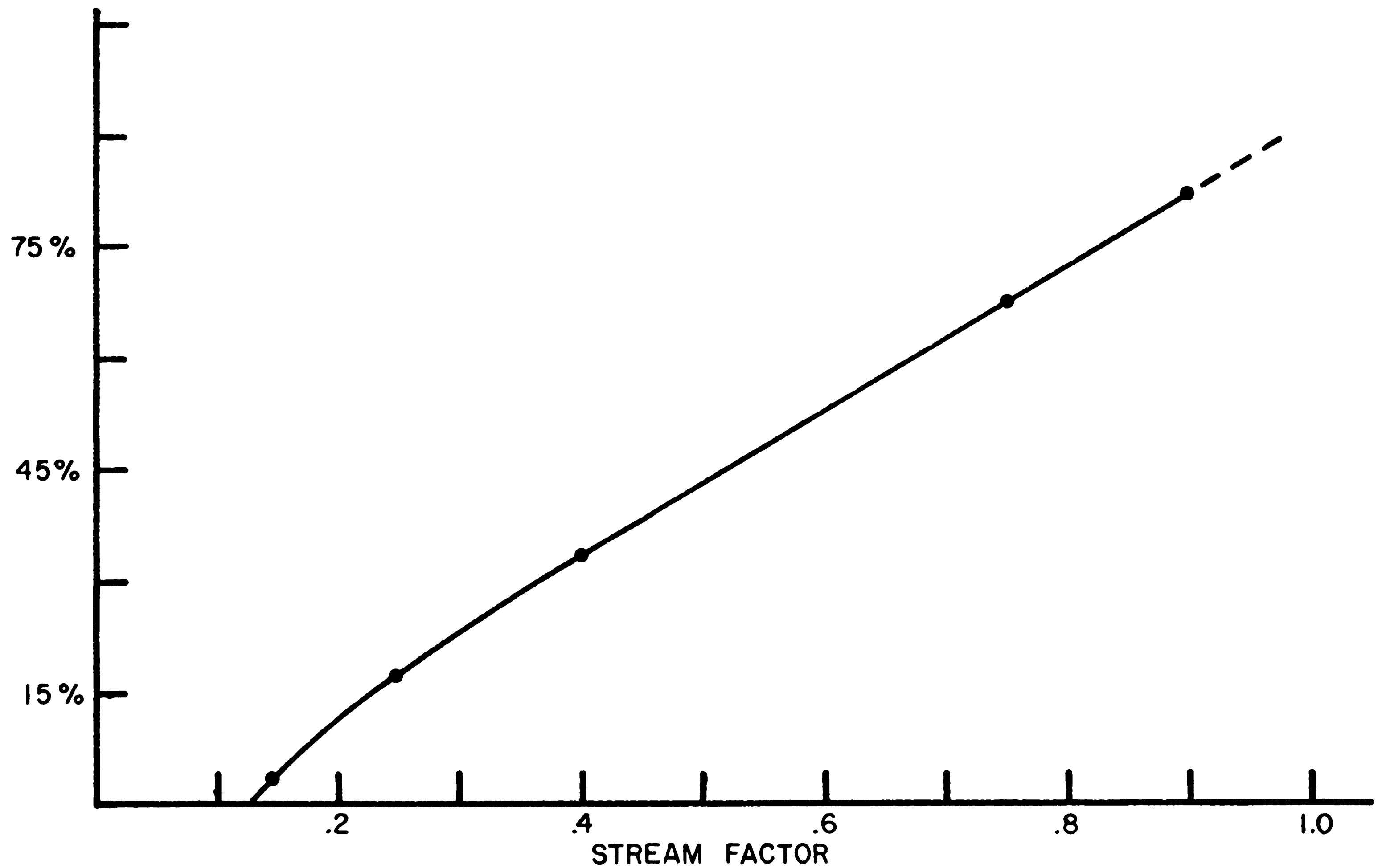

Figure 5.--After Taxes Discounted Cash Flow Rate of Return versus Stream Factor for Organic Rankine Cycle.

Examination of Figures 4 and 5 demonstrates that with a 1980 total fixed capital investment of $1,568,000 (excluding loan interest), payout periods of less than two years are attainable at stream factors of greater than 50 percent. Furthermore, at full capacity, the minimum payout period is approximately one year. Finally, at stream factors of less than 15 percent, a rapid rise in payout periods of greater than six years is evident.

As previously discussed, four plants operated the Pratt-Whitney GG-12 gas turbines in the 1977 operating year. The Ray Station Plants 1 and 2 could expect a payout period of seven years at their 16 percent stream factor for any dual driver installation. With load shifting, a minimum payout period of 1.8 years would result. Once more, the large potential for load shifting and energy recovery at the Ray Station Plants 1 and 2 provide a highly probable site for energy recovery.

Finally, the two remaining plants which operated GG-12 gas turbines (the St. Clair Plants 1 and 2) experienced low stream factors of four percent or less, rendering them unprofitable as recovery sites.

A detailed description of the Organic Rankine Cycle design and economics is presented in Appendix IV.

Process (150 psia) Steam Production

The production of 150 psia process steam was investigated for each of the three base cases at various approach temperatures. Figure 6 presents the payout period versus stream factor for the

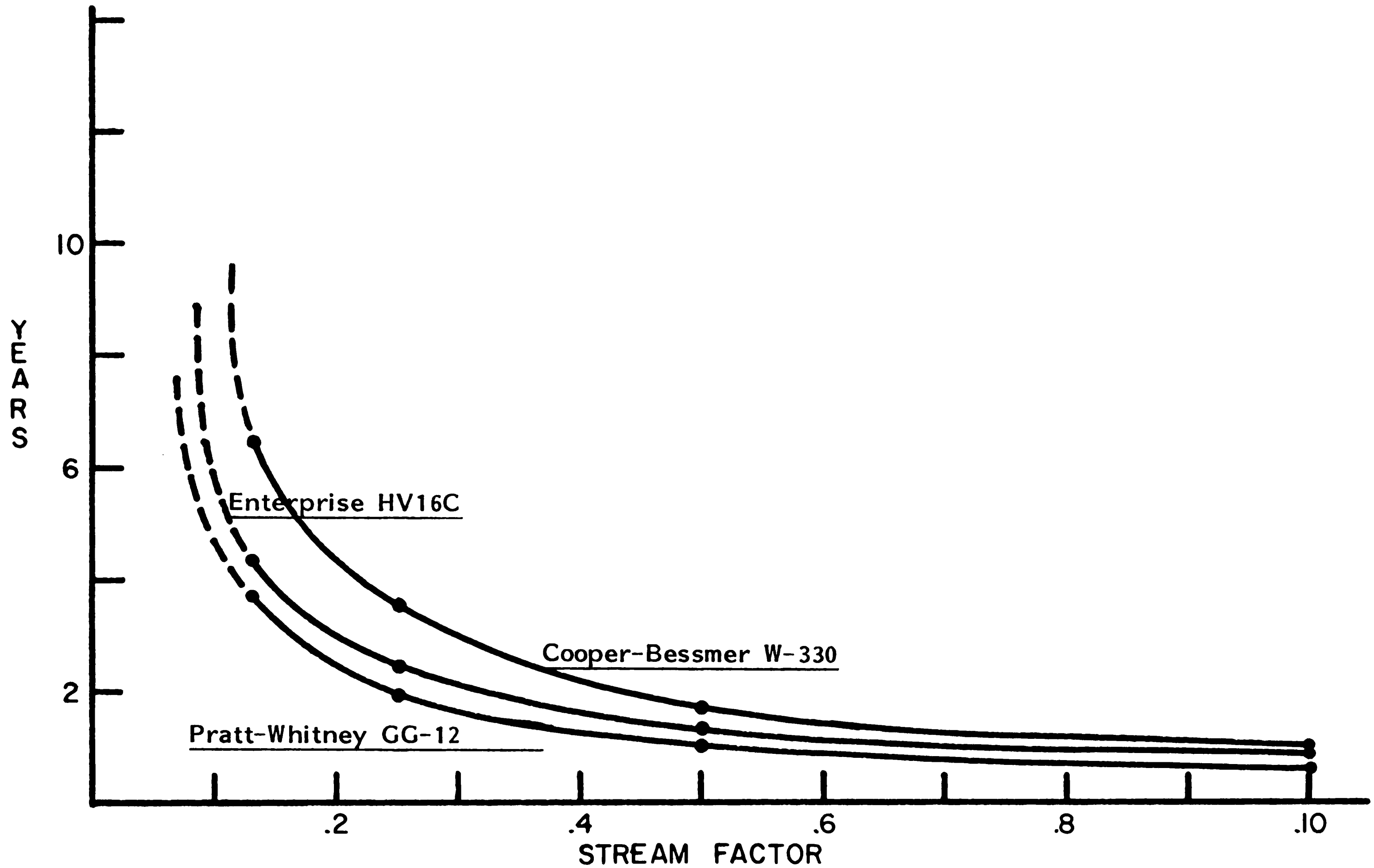

Figure 6.--After Taxes Payout Period versus Stream Factor for Process Steam Production.

three base cases. A detailed description of design and economics for the 150 psia steam production design is presented in Appendix V.

Once more, in the interest of clarity, each of the three base cases is presented individually in the following three subsections.

a. Pratt-Whitney GG-12 Gas Turbine

The production of 150 psia steam via the use of an exhaust gas waste heat boiler was applied to a single Pratt-Whitney GG-12 driver with very encouraging results. A maximum cycle temperature of 360°F resulted in the production of 22,320 pounds per hour of saturated 150 psia steam.

Examination of Figure 6 demonstrates that payout periods of under two years are easily attainable for stream factors of greater than 20 percent. Furthermore, a minimal 1980 total fixed capital investment of $112,800 (excluding construction loan interest) is needed to provide such a payout period. At stream factors of less than 10 percent, less desirable payout periods of over four years are evident. At full capacity, a minimum payout period of 0.45 years is realized.

As stated in previous sections, four plants operated the Pratt-Whitney GG-12 gas turbines during the 1977 operating year. Two of these plants, the Ray Station Plants 1 and 2 encountered stream factors of approximately 16 percent. At such stream factors, payout periods of less than 2.8 years could be expected for any single driver installation. With only minor load shifting, payout periods of well under two years could be realized for these drivers. Finally, with maximum load shifting, a minimum payout period of approximately .9 years is implied.

Again, the extremely low stream factors for the two remaining sites, St. Clair Plants 1 and 2, make it unprofitable to install steam generation equipment at these sites.

b. Cooper-Bessmer W-330
Reciprocating Engine

The production of 150 psia steam via the use of waste heat boilers was applied to the Cooper Bessmer W-330 drivers with encouraging results. A maximum cycle temperature of 361°F resulted in the production of 6085 pounds per hour of saturated 150# steam.

Examination of Figure 6 demonstrates that, with a 1980 total fixed capital investment of $58,100 (excluding interest on construction loan), payout periods of less than two years can be expected at stream factors of 40 percent or greater. A minimum payout period of .9 years occurs at full capacity and at stream factors of less than 20 percent payout periods of over four years result.

Two plant sites operated the Cooper-Bessmer W-330's in the 1977 operating year, the White Pidgeon Plants 1 and 2. With stream factors of 26.2 and 14.8 percent, respectively, payout periods of 3.2 and 6.5 years would be expected. Load shifting does not appear feasible at these sites, due to the minimal work load at these sites, therefore, these payout periods are as good as could be expected.

Again, three additional sites operated drivers which are similar in design characteristics to the Cooper-Bessmer W-330's.

The first of these is the Clark TLA-10 at the Freedom Plant 2. With a stream factor of nearly 80 percent in 1977 a payout period of one year would result. This low payout period combined with the high stream factor make this drive a consistent, economically feasible source for steam production.

Two additional Clark TLA-10 drivers operated at the Muskegon River site with 1977 stream factors of 38 percent. At such a stream factor, a payout period of two years could be expected. Again, load shifting could be utilized at this site making it an even more favorable source of steam production.

Finally, four Clark TLA-8 drivers operated at the Overisel site with a 31 percent stream factor. A payout period of 2.8 years could be expected for any single driver installation here. Once more, load shifting could allow a minimum payout period of less than 1.7 years.

c. Enterprise HV16C Reciprocating Engine

Production of 150 psia steam via the use of an exhaust gas waste heat boiler was applied to the Enterprise HV16C reciprocating driver with encouraging results. A maximum cycle temperature of 361°F resulted in the production of 11,645 pounds per hour of saturated process steam.

Figure 6 demonstrates that with a 1980 total fixed capital investment of $73,000 (excluding interest on construction loan) payout periods of less than 2.2 years can be expected at stream factors of 25 percent or greater. A minimum payout period of .6

years occurs at full capacity and at stream factors of less than 10 percent payout periods of over four years are expected.

Only two plants operated the Enterprise drivers during the 1977 operating year. The St. Clair Plant 2, with HV16C drivers, encountered stream factors of 13.6 percent. At this stream factor, payout periods of approximately 3.5 years would result for a single driver installation. In addition, the BTU Stabilization Plant operated the smaller HV12C drivers at a stream factor of 35.4 percent. This higher stream factor results in an estimated payout period of two years, including a contingency for reduced driver size.

8. CONCLUSIONS

Natural gas compressor stations provide a very large potential source for energy recovery. Clean gaseous waste streams in excess of 700°F are not uncommon and constitute over 70 percent of the available waste energy at many sites. The remaining 30 percent potential is of a less usable, lower temperature (less than 200°F) variety. The specific example of the Consumer's Power network of compressor stations lost over 1570 billion BTU's of energy in the above forms during the 1977 operating year. Only 19 percent of the energy content of 2.2 billion SCF of fuel gas was actually converted into useful work. This single example emphasizes the potential for energy recovery and the need for energy conservation at compressor stations throughout the world.

Low Temperature Waste Heat Streams (< 200°F)

Low temperature waste heat sources are of little value at compressor stations. The need for streams of 200°F or less within a compressor station is minimal and usually the timing of such a need is out of phase with the availability of such streams. For the same reasons, outside users of the heat energy contained in low temperature waste energy streams are probably not interested in them. A potential outside user would need to solve the problems of sporadic supply and low energy quality in order to benefit from such streams.

Steam rankine cycles are not feasible using low temperature waste heat sources. Using such sources as pre-heaters or economizers in steam rankine cycles is also not feasible, unless the cycle is operated with a sub-atmospheric condenser.

Organic rankine cycles can utilize low temperature streams as primary heat sources. However, such streams are usually a supplement to a high temperature waste stream. Organic rankine cycles can also utilize low temperature streams in pre-heaters or economizers due to their lower operating temperatures. However, depending on the system, this might or might not be feasible. Stand-alone usage of low temperature streams is not generally beneficial.

In the Consumers Power network none of the designs presented were able to use these low temperature waste energy streams. Therefore, some 30 percent of the waste heat generated was considered unusable.

High Temperature Exhaust Streams

High temperature driver exhaust streams are readily available at compressor stations. The majority of these streams are in excess of 700°F and they comprise some 70 percent of the available waste energy. Direct usage of such streams for sale to an outside user or usage within the compressor station is again hindered due to the sporacity and timing of normal operation. Again, an outside user would need to work cooperatively with utility personnel in order for mutual benefit to occur. The usage of energy in forms other than as shaft work is almost non-existent at most stations. Electricity, process steam or simply hot gaseous streams are not at a premium and do not therefore justify large capital expenditures. On-site waste energy usage is limited to that energy which is converted to usable shaft work.

Steam Rankine Cycles Via High Temperature Waste

Steam rankine cycles offer a method of energy recovery which is economically favorable and readily usable. The production of shaft work via rankine cycles allows recovered waste energy to be used immediately to reduce the horsepower load on the waste-energy-producing drive motor. There is no significant problem of the timing of recovered waste energy availability, since it will only be produced while it is needed. The net effect of steam rankine cycle shaft work production is an increase in overall drive system thermodynamic efficiency.

The three base cases examined in this study are compared in Table 4 comparing three performance criteria. These criteria are the payout period, capital dollars per installed horsepower and before taxes profit generated per installed horsepower. It is evident from examination of this table that two factors are of prime importance when determining the usability of a waste energy stream for rankine cycle shaft work production: temperature quality and flow rate. High temperatures allow high steam pressures and thereby increase rankine cycle thermodynamic efficiency. High flow rates allow for the effect of "economy of scale" to reduce incremental

TABLE 4.--Exhaust Gas Heat Recovery Design Case Profitability Comparisons.

Driver	Temp. (°F)	Flow Rate (#/hr)	1980 Capital Cost ($/hp)	Stream Factor (%)	Before Taxes Profit* ($/hp)	After Taxes Payout Period** (year)
STEAM RANKINE CYCLE						
PW GG-12	840°	158,400	345	25	62	4.1
PW GG-12	840°	158,400	345	90	395	1.1
CB-W-330	750°	53,950	618	25	-114	N/A
CB-W-330	750°	53,950	618	90	12	6.4
ENT-HV16C	1000°	60,600	516	25	-29	12.1
ENT-HV-16C	1000°	60,600	516	90	174	2.9
ORGANIC RANKINE CYCLE						
PW GG-12	840°	158,400	336	25	66	4.0
PW GG-12	840°	158,400	336	90	387	1.0
CB-W-330	750°	53,950	691	25	-29	N/A
CB-W-330	750°	53,950	691	90	34	6.1

*Before taxes profit includes all credits, operating costs, depreciation and interest on working capital but excludes corporate income taxes.

**Payout period is determined as the average cash flow (after taxes) over a ten year period divided into the 1980 total fixed capital investment.

capital costs. These two factors work together to increase the profitability of waste energy recovery.

It is evident that economy of scale is the predominant influence in reducing incremental capital costs as evidenced by the low capital cost per installed horsepower capacity of the Pratt-Whitney GG-12 steam rankine cycle design. This design realizes a 35 percent reduction in unit capital costs and a 60 percent reduction in payout period when compared to the Enterprise HV16C base case. Clearly, waste <u>stream flow rate</u> is of <u>primary</u> importance when considering base case capital costs and profitability.

In addition to waste stream flow rate overall profitability is strongly effected by the stream factor. Low stream factors decrease revenues (or savings in this case) thereby allowing fixed costs (i.e., capital costs, interest, labor costs) to overwhelm revenues. However, as incremental capital costs decrease, lower and lower stream factors remain profitable. Overall profitability is effected by incremental capital costs and determined by stream factor.

Finally, the temperature of the waste stream in question is of moderate importance. High waste stream temperatures produce higher rankine cycle efficiencies and thereby higher horsepower outputs. The higher the horsepower output is, the lower the incremental capital cost becomes. Again, lower incremental capital costs reduce operating costs and increase profitability. Such an effect is seen in the comparison of the Cooper-Bessmer W-330 and the Enterprise HV16C base cases. At approximately the same waste stream flow rate, these two cases demonstrate a moderate difference in capital costs and subsequently a difference in overall profitability. The Enterprise HV16C case realizes a 16 percent decrease in capital costs and approximately a 50 percent decrease in payout period due to its higher (1000°F) cycle temperature. The Cooper-Bessmer W-300, on the other hand, must contend with both a lower waste stream flow rate and a relatively low 750°F waste stream temperature.

In general, the waste stream flow rate is of primary importance when determining the usability of a waste stream for rankine cycle shaft work production. High waste stream flow rates are necessary to provide a sufficient quantity of waste energy to make recovery economically feasible. In addition, high waste stream temperatures are necessary to both increase overall waste heat availability and increase thermodynamic cycle efficiency. Based on the content of this study, waste streams with overall heat loads of less than 5 million BTU per hour are generally unusable for rankine cycle shaft work production unless a high stream factor of 80 percent or greater can be expected. Waste streams with heat loads in excess of 18 million BTU per hour are economically feasible at 25 percent or greater stream factor and become very economically favorable at stream factors of greater than 60 percent. Individual assessments must be made for waste stream heat loads falling between these two extremes when considering steam rankine cycle shaft work production.

The Consumers Power Company network of compressor stations yields two promising possibilities for steam rankine cycle shaft work production. These two stations are the Ray Station Plants 1 and 2. The Pratt-Whitney GG-12 gas turbine centrifugal compressors located in these two plants produced 40 percent of the total recoverable high temperature stack waste energy in 1977. The low stream factors of 16 percent allow payout periods of approximately 6.6 years for any dual driver installation. With load shifting this stream factor can be increased to a maximum of 53 percent, yielding a payout period of 2.2 years. Sole operation of two pairs of Pratt-Whitney GG-12 drivers with accompanying steam rankine cycles would allow approximately a 27 percent overall stream factor yielding four year payout periods. Table 5 presents pertinent economic information for the Ray Station design alternatives. The last alternative is both economically and technically feasible. It would provide a 50 percent increase in drive system efficiency and could reduce overall maintenance costs. In addition, it would require very little load shifting and would be equivalent to the operation of three pairs of Pratt-Whitney GG-12 compressor-driver combinations.

Organic Rankine Cycle Via High Waste Streams

Organic rankine cycles also offer an economically and technically feasible means for waste energy recovery. Similar to the steam rankine cycles previously discussed, these cycles provide shaft work for immediate usage in natural gas compression. The primary difference between Organic rankine cycles and their steam counterparts is in the size of equipment needed to support the cycles and in the overall cycle heat duty. Extremely large flow rates of

TABLE 5.--Summary of Ray Station Design Economics.

Degree of Load Shifting	Plant Number	No. of Drivers	No. of Steam Turbines	Steam Factor %	Total 1980 Fixed Capital Investment ($1000's)	1981 Cash Flow ($1000's)	Payout Period (years)	Discounted Cash Flow Rate of Return (%)
None	1	4	2	16.0	1910	235	6.6	7.6
None	2	6	3	16.0	2865	352	6.6	7.6
Partial	1	2	1	27.0	955	199	3.9	20.0
Partial	2	2	1	27.0	955	199	3.9	20.0
Maximum	1	0	0	0	0	0	N/A	N/A
Maximum	2	2	1	53.0	955	393	2.2	43.0

organic working fluids demanded by the large heat duty also demand the large processing equipment. An Organic rankine cycle coupled with a driver (such as the Pratt-Whitney GG-12) can produce a quantity of shaft work equal to that produced by the driver, thereby doubling its efficiency.

One factor is of primary importance to the economic feasibility of an organic rankine cycle: the waste stream flow rate. High waste stream heat duty allows the economy of scale to reduce incremental capital costs, thereby decreasing operating costs. Since most exhaust gas waste streams are in excess of 625°F, and therefore exceed the maximum safe operating temperature of the organic rankine cycle (using Flourinol 85), temperature is of less importance than it was in the prior example. For example, an 840°F exhaust gas stream allows a steam rankine cycle to operate at higher pressures and thereby higher thermodynamic efficiencies than a 625°F exhaust stream would (at the same heat duty). An organic rankine cycle, on the other hand, would operate at the same thermodynamic efficiency utilizing either exhaust stream, due to its 625°F maximum cycle temperature restriction.

The organic rankine cycle, discussed in Appendix IV, is compared to the steam rankine cycle design in Table 4. Both designs utilize the same waste energy stream as their heat source. However, the horsepower output of the organic rankine cycle is approximately double that of its steam counterpart. Due to the low condensing temperature and high working fluid flow rate, a higher total capital cost is characteristic in the organic rankine cycle. However, due to the higher efficiency, the cost per installed horsepower output is nearly equal to that of the steam rankine cycle.

The payout period at 25 percent stream factor is 4.0 years. Although this is slightly better than the steam rankine cycle, it is not significantly different. The same trend is seen at 90 percent stream factor with the organic rankine cycle indicating a 1.1 year payout period.

Utilization of the organic rankine cycle for energy recovery from high temperature, high flow rate waste energy streams shows no significant advantage over the conventional steam rankine cycle. The added system complexity and investment in equipment for the organic rankine cycles is not justified economically. The added risk in an unproved technology is also a strong deterrent, when steam rankine cycles can be used cost effectively.

For the lower temperature, low flow rate streams (such as CB W-330 exhaust gas) the organic rankine cycle again offers no significant advantage over the conventional steam rankine cycle. Although the organic rankine cycle almost doubles the horsepower output possible when compared to steam rankine cycles, the organic's are burdened with large capital investments which severely detract from the economics.

In the Consumers Power network, two plants are again of interest for waste heat recovery, the Ray Station Plants 1 and 2. These plants operated their Pratt-Whitney GG-12 gas turbines at a

16 percent stream factor in 1977 indicating a seven year payout period for any dual driver organic rankine cycle retrofitted combination. Again, with minimal load shifting, two pairs of retrofitted drivers could operate at 20 percent stream factor and effectively double their thermodynamic efficiency. At such a stream factor a payout period similar to that seen in the steam rankine cycle would be realized (approximately four years).

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APPENDICES

APPENDIX I

WASTE HEAT BOILER COMPUTER SIMULATION

APPENDIX I

WASTE HEAT BOILER COMPUTER SIMULATION

Introduction

One of the more costly and difficult to design pieces of equipment in all of the design schemes presented here is the exhaust gas waste heat boiler. This one component accomplishes all of the energy recovery via the transfer of heat energy from the hot exhaust gases to the working fluid of choice. In an effort to more accurately design the large number of waste heat boilers considered, an iterative computer simulation of a shell and finned-tube heat exchanger was utilized. The following detailed description of that simulation is identical for all of the waste heat boiler designs in theory and method. Therefore, a general discussion is provided in this Appendix while specifics are presented in the individual design sections (i.e., heat transfer coefficients, areas, tube pitches, etc.).

Discussion

Description of Exhaust Gas Waste Heat Boiler

The waste heat boiler (see Figure 7) is quite simply a cross counter-flow shell and tube heat exchanger utilizing the hot exhaust gases on the shell side and the working fluid of choice on the tube side (2). Serrated fins are used on the shell side to both increase

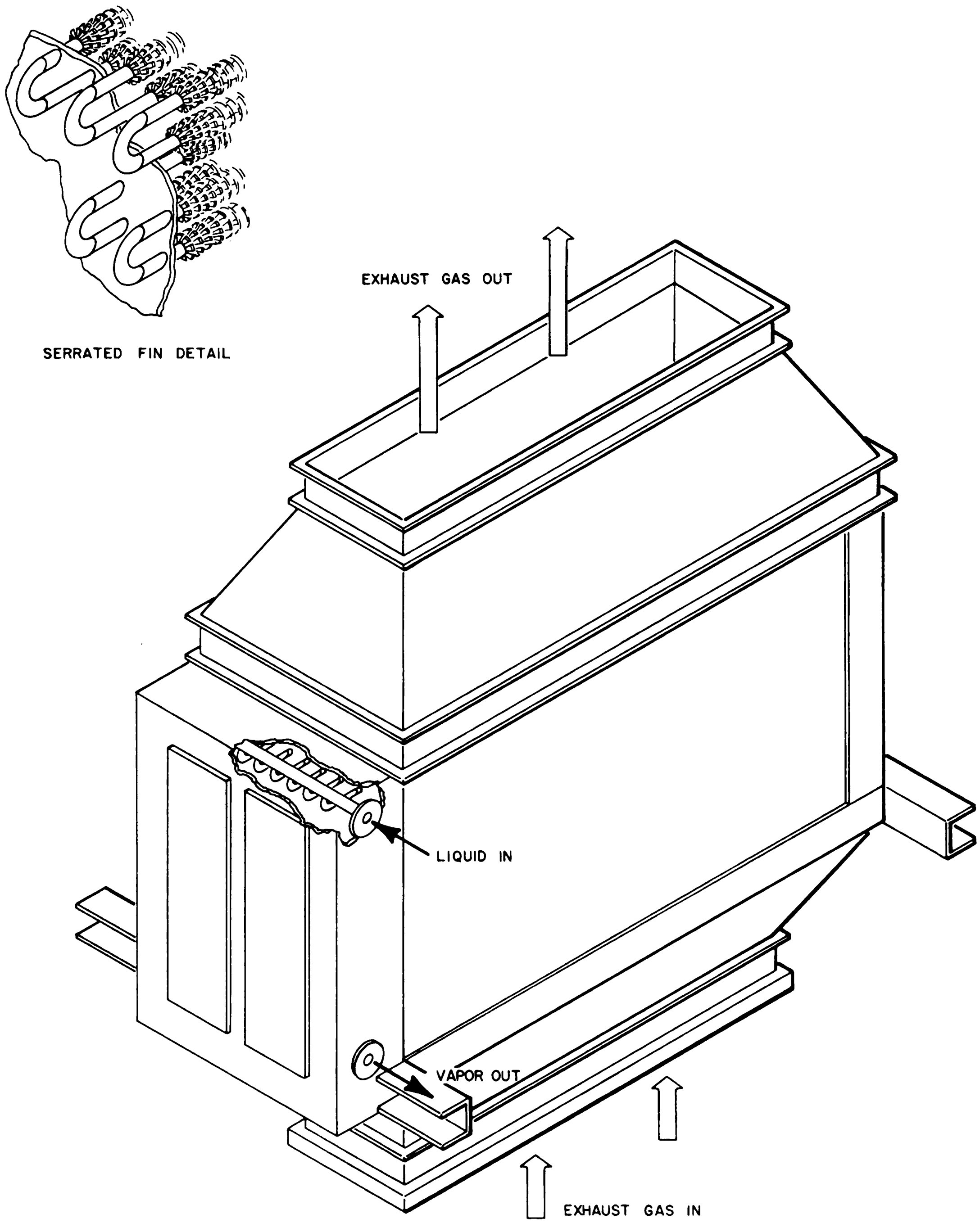


Figure 7.--Diagram of Exhaust Gas Waste Heat Boiler.

the apparent shell side surface area, and to increase the shellside heat transfer coefficient via the introduction of additional turbulence. In applications where exhaust gases contain large amounts of particulates and soot, the serrated fins are reported to act as self-cleaning fins thereby reducing fouling on the shell side of the heat exchanger (2).

The tube side consists of rows of parallel tubes perpendicular to the flow of exhaust gases. Triangular tube pitch is used again to provide maximum turbulence on the shell side. The tubes are simply carbon steel 40 BWG using 180° elbows on each end to provide flow between individual banks. A small baffle or dummy tube is included in each bank due to the assymetry that triangular pitch imposes.

A carbon steel or galvanized plenum surrounds the tube banks and acts as the shell. Manifolds are provided at each end of this plenum to serve as connection points to the driver exhaust inlet and outlet. Insulation is applied to this plenum to act as both a heat insulator and a noise deadening medium. The exhaust muffler for each driver is replaced by the waste heat boiler, which is similar in accoustic concept to the muffler, thereby accomplishing the same task.

Basic Program Structure

In order to provide flexibility and ease of use, the heat exchanger simulation was written as the general case of a shell and finned-tube heat exchanger in cross counter flow. A main program containing all energy balance, heat transfer coefficient and pressure drop calculations was written as a single module. This module is stored in the computer's memory and can be accessed at will. The module does require three subroutines to be loaded at program execution. These three subroutines contain the exchanger dimensional parameters, the shell side gas physical properties and the tube side liquid physical properties.

The exchanger dimensional parameters cover a wide spectrum from the type of tube pitch to the diameter of the fin to be used. Basically, by fixing the parameters contained in this subroutine, the program can calculate all of the remaining dimensional characteristics that describe the heat exchanger.

The physical property subroutines contain heat capacity, density, viscosity and thermal conductivity information. Where possible, these properties are expressed as a function of temperature thereby allowing the program to continually update their values. This continual updating of physical properties is an important feature of the iterative method used in this simulation.

A macroscopic view of the program provides the following logical sequence:

1. Fix all heat exchanger dimensional parameters (i.e, number of tube rows, tubing I.D., etc.)
2. Obtain physical property information from physical property subroutines.
3. Calculate heat transfer coefficients, amount of heat transferred and amount of surface area needed for heat transfer.
4. Calculate pressure drop (shell and tube sides) associated with #3 and return to #2.

5. Continue as above until heat transfer task is complete then print output.

6. Check to see if shellside pressure drop is too high or too low. If so, readjust dimensional parameters to correct pressure drop and start over at #1.

7. End program.

Detailed Flow Chart and Design Equations

Figure 8 is a detailed flow chart of the computer simulation that when used in conjunction with the program listing presented in Table 6 aids in understanding the following discussion. For ease of reading the numbers in the far left column of the listing will be used as locaters when discussing a line or group of lines of code. For general overview of the program utilization of Figure 8 exclusively is recommended.

Lines 2 through 25 accomplish the initialization of all variables. In line 26 the subroutine Param is called to update all of the heat exchanger dimensional characteristics and to set the maximum shell side pressure drop, characteristic temperatures, material mass flow rates and temperature change per iteration. This subroutine is called only once during execution of the program. Lines 29 through 41 determine which tube pitch option was chosen in the subroutine Param.

Three options exist:

1. IOPT = 1, Square tube pitch

2. IOPT = 2, Triangular tube pitch with dummy tubes or baffles)

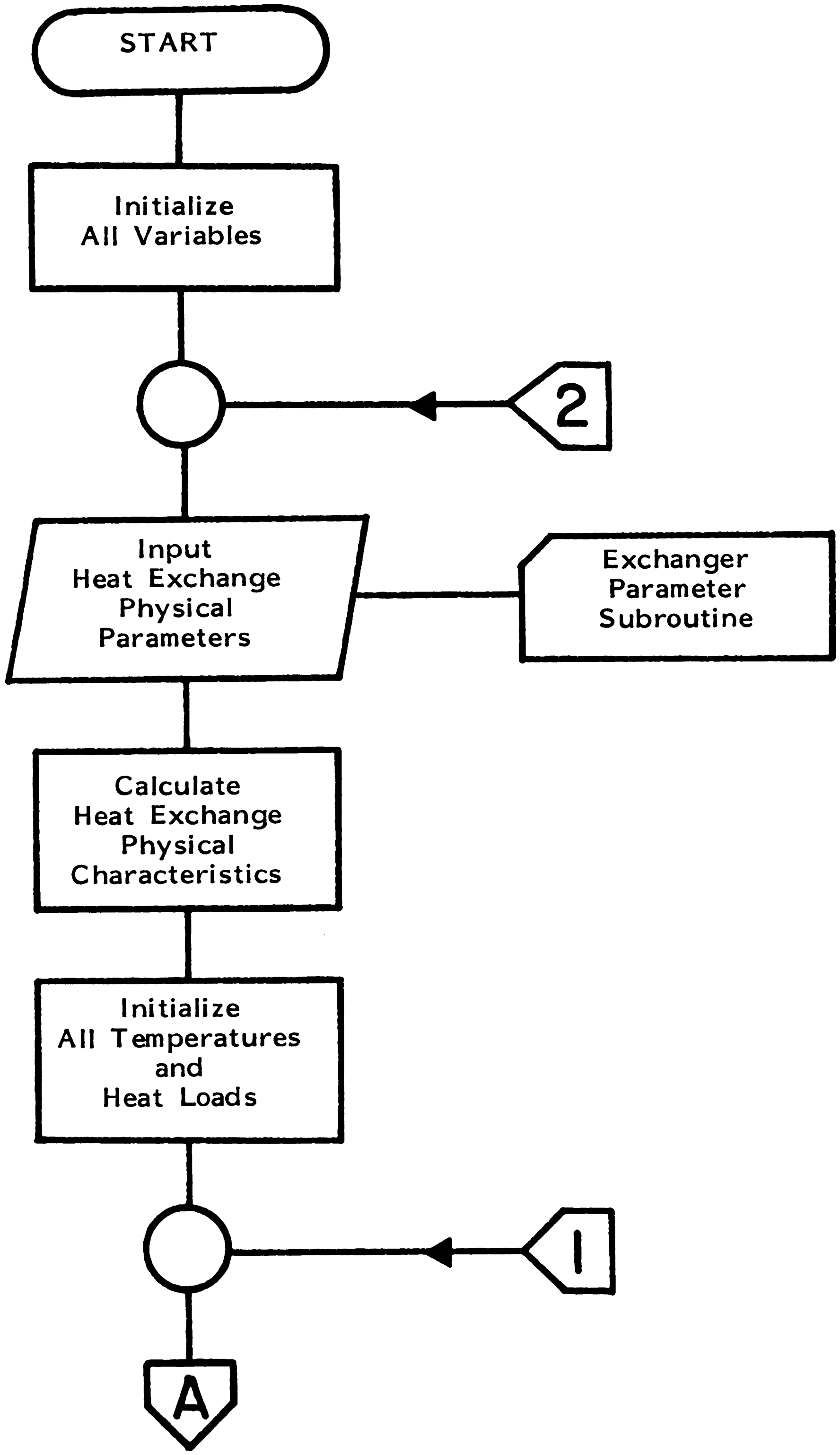
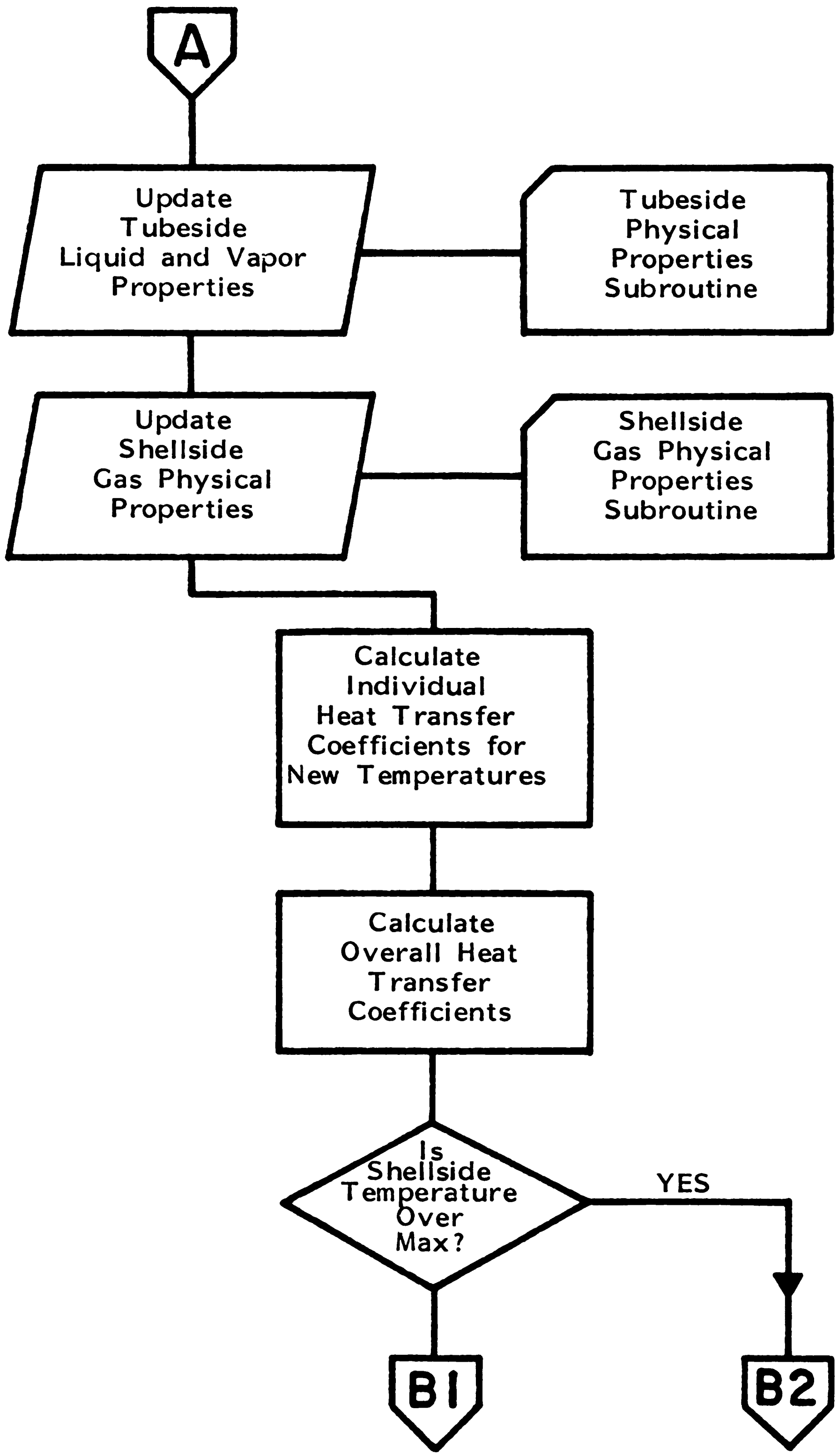
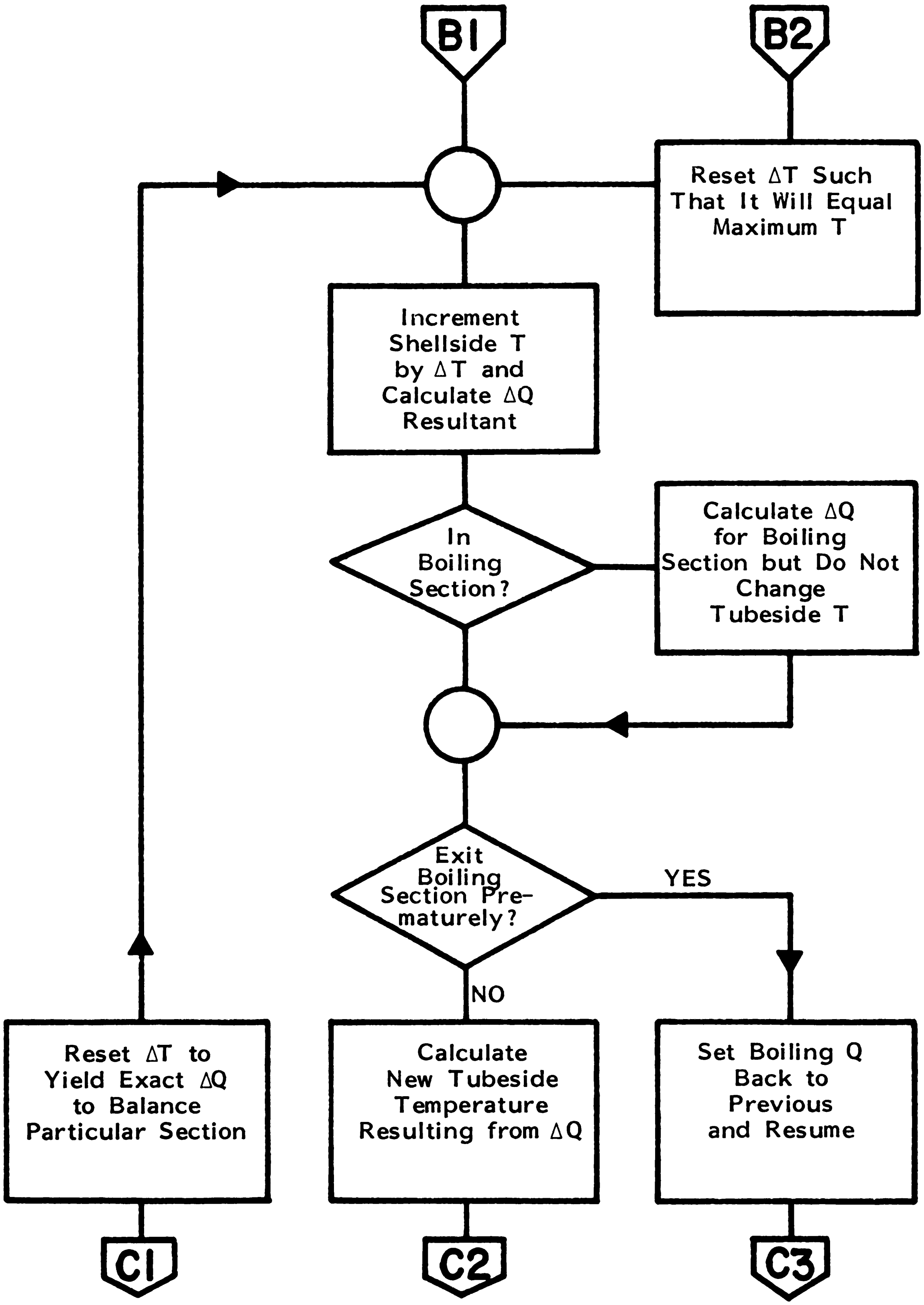
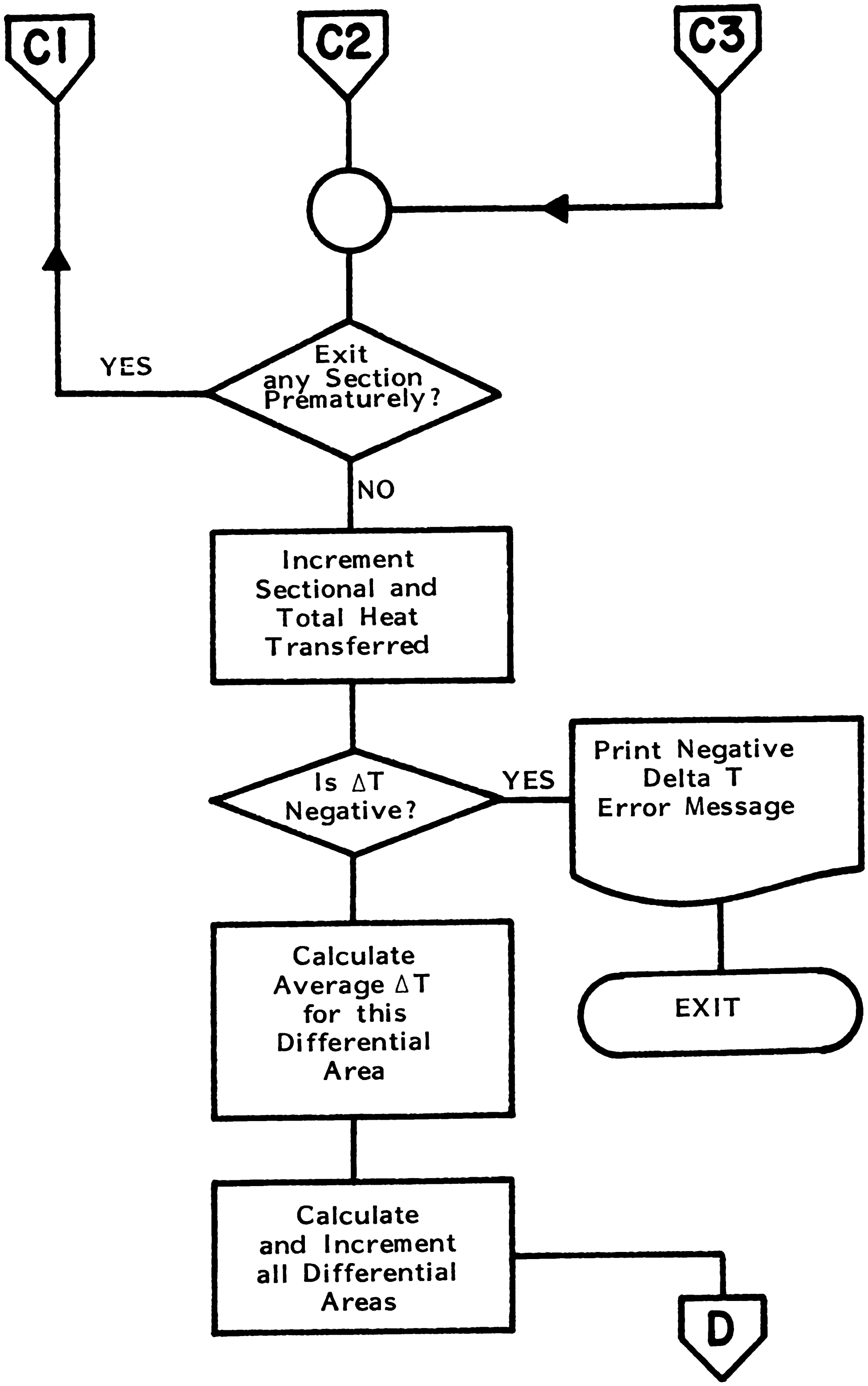
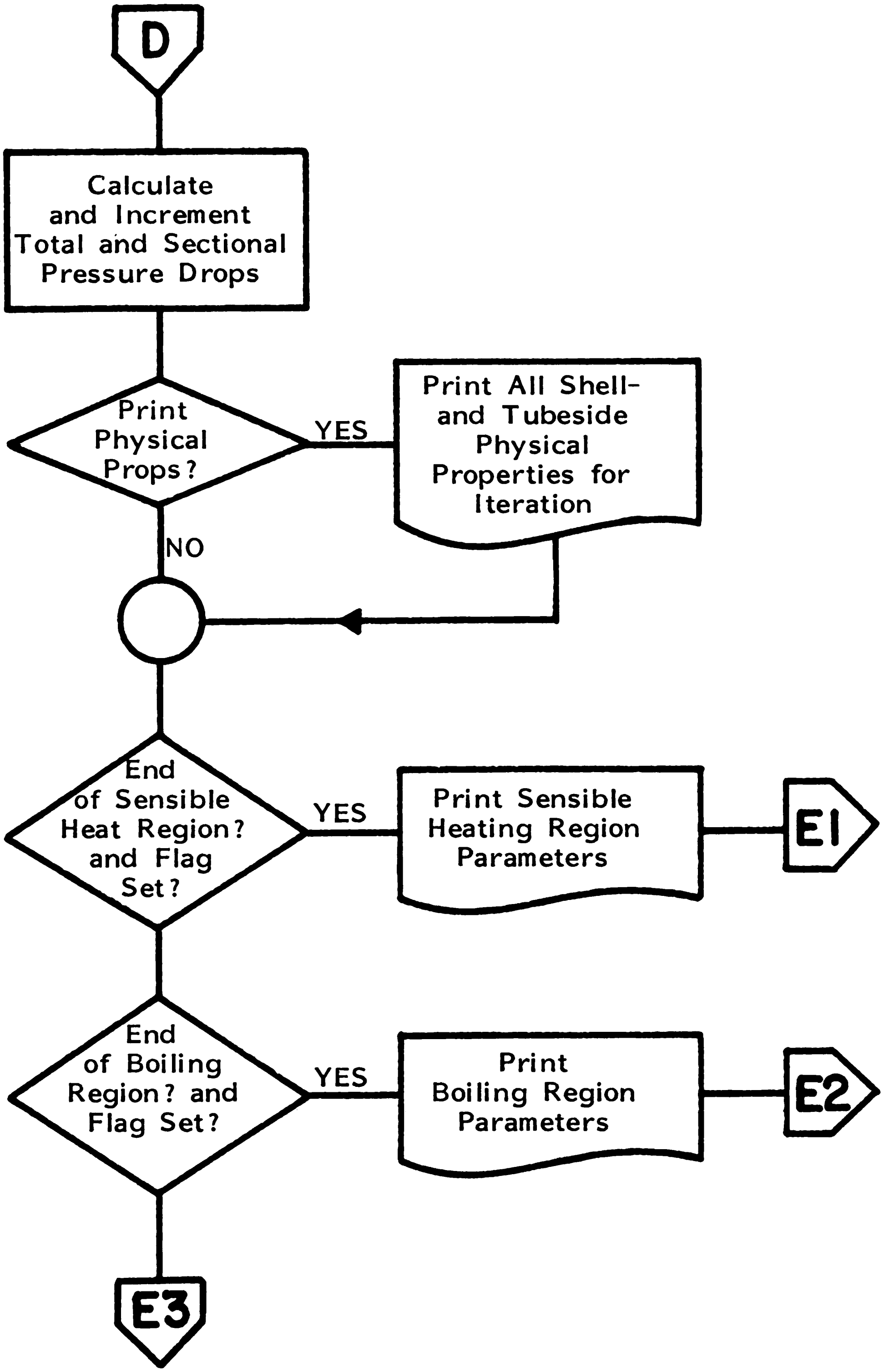


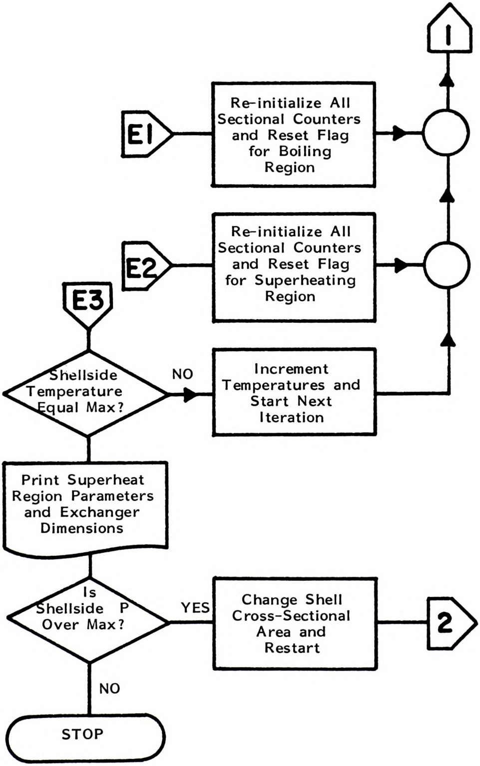
Figure 8.--Waste Heat Boiler Simulation Flow Chart.


A
Update Tubeside Liquid and Vapor Properties
Tubeside Physical Properties Subroutine
Update Shellside Gas Physical Properties
Shellside Gas Physical Properties Subroutine
Calculate Individual Heat Transfer Coefficients for New Temperatures
Calculate Overall Heat Transfer Coefficients
Is Shellside Temperature Over Max?
YES
B1
B2


B1
B2
Reset ΔT Such That It Will Equal Maximum T
Increment Shellside T by ΔT and Calculate ΔQ Resultant
In Boiling Section?
Calculate ΔQ for Boiling Section but Do Not Change Tubeside T
Exit Boiling Section Prematurely?
YES
NO
Reset ΔT to Yield Exact ΔQ to Balance Particular Section
Calculate New Tubeside Temperature Resulting from ΔQ
Set Boiling Q Back to Previous and Resume
C1
C2
C3


C1
C2
C3
YES
Exit any Section Prematurely?
NO
Increment Sectional and Total Heat Transferred
Is ΔT Negative?
YES
Print Negative Delta T Error Message
EXIT
Calculate Average ΔT for this Differential Area
Calculate and Increment all Differential Areas
D


D
Calculate and Increment Total and Sectional Pressure Drops
Print Physical Props?
YES
Print All Shell- and Tubeside Physical Properties for Iteration
NO
End of Sensible Heat Region? and Flag Set?
YES
Print Sensible Heating Region Parameters
E1
End of Boiling Region? and Flag Set?
YES
Print Boiling Region Parameters
E2
E3


1
E1
Re-initialize All Sectional Counters and Reset Flag for Boiling Region
E2
Re-initialize All Sectional Counters and Reset Flag for Superheating Region
E3
Shellside Temperature Equal Max?
NO
Increment Temperatures and Start Next Iteration
Print Superheat Region Parameters and Exchanger Dimensions
Is Shellside P Over Max?
YES
Change Shell Cross-Sectional Area and Restart
2
NO
STOP

3. IOPT = 3, Triangular tube pitch without dummy tubes (results in asymmetrical rows and is therefore undesirable).

Lines 42 and 43 calculate the inside diameter and inside surface area (per foot) of the tubes. Lines 44 and 45 calculate the fin surface area. Two options are provided here as follows:

1. If IFIN = 1, then solid circular fins are utilized and the fin area per foot,

$$FA = \frac{\pi \times FPF\ [D2 + 2 \times FH)^2 - D2^2]}{2}$$

2. If IFIN = 2, then serrated fins are utilized and the fin area per foot, $FA = 2\pi \times FPF \times FH \times D2$

   Where FA = Fin area per foot (ft^2/ft)

   FPF = Fins per foot

   D2 = Outside diameter of tubing (ft)

   FH = Fin height (ft)

Line 46 calculates the bare tube area per foot as

$$ABT = \pi(D2 - D2 \times FPF \times FTH)$$

Where ABT = Area bare tube (ft^2/ft)

FTH = Fin thickness (ft)

Line 47 calculates the projected perimeter as

$$PP = 4 \times FH \times FPF + 2$$

Where PP = Projected perimeter (ft).

TABLE 6.--Waste Heat Boiler Simulation Program Listing.

```
          PROGRAM HTEX(INPUT,OUTPUT)
          DUMM1=0.
          DUMM2=0.
        5 CONTINUE
          SIGNAL=0.
          DUMM1=DUMM1+1.
        6 CONTINUE
          TA=0.$TA2=0.$TA3=0.$HIGT=0.$PDRPS=0.$PDRPT=0.
          QT=0.
          QB=0.
          TQ=0.
          PDPS2=0.
          TTA=0.
          TTA2=0.
          TTA3=0.
          TPDRT=0.
          TPDRS=0.
          FTPT=0.
          FTPS=0.
          ITER=0
          IFLAG=2
          ICONT=0
          IFIN=0
          DTMIN=1000000000.
          PI=3.141592654
C
C         INPUT PARAMETERS
C
          CALL PARAM(IFIN,IPRNT,IEQN,IOPT,GS,GT,DHVAP,T
TP,TL,FTH,D2,ENTT,ENR
         1PP,FPF,FH,TWTH,TKF,TBOIL,TEMTL,TEMSL,TEMSH,DE
LT,HDO,HDI,DUMM1,DUMM
         22,PDMAX,SIGNAL)
C
C
C         HEAT EXCHANGER CHARACTERISTICS
C
C
C........ SECTION BELOW DETERMINES TUBE PITCHES DEP
ENDING ON TRIANGULAR
C............OR SQUARE OPTION.
          IF(IOPT.LE.2)GOTO15
          GOTO16
```

```
   15 CONTINUE
      TPL=TTP
      ENTT2=ENTT
      ENTPP=ENRPP*ENTT
      IF(IOPT.EQ.2)TPL=SQRT(TTP**2-(TTP/2.)**2)
      GOTO17
   16 CONTINUE
      TPL=SQRT(TTP**2-(TTP/2.)**2)
      ENTT2=ENTT-1.
      ENTPP=(ENTT+ENTT2)*ENRPP/2.
   17 CONTINUE
C........ CALCULATION OF I.D.
      D1=D2-2.*TWTH
C........ CALCULATION OF INSIDE SURFACE AREA.
      AIT=D1*PI
C........ CALCULATION OF FIN AREA.
      AF=PI/4.*(((D2+2.*FH)**2)-(D2**2))*FPF*2.
      IF(IFIN.EQ.2)AF=2.*PI*FH*FPF*D2
C........CALCULATION OF BARE TUBE AREA.
      ABT=PI*D2-PI*FPF*FTH*D2
C........CALCULATION OF PROJECTED PERIMETER.
      PP=4.*FH*FPF+2.
C........CALCULATION OF EQUIVALENT DIAMETER.
      DE=2.*(AF+ABT)/(PI*PP)
C........CALCULATION OF EXCHANGER WIDTH.
      W=ENTT*TTP
C........CALCULATION OF NET VOLUME.
      VNET=W*TL*TPL-.5*(ENTT+ENTT2)*PI*D2**2*TL/4.
      VNET=VNET-.5*(ENTT+ENTT2)*PI*D2*FH*FPF*TL*FTH
C........CALCULATION OF VOLUMETRIC EQUIVALENT DIAME
TER.
      DEV=8.*VNET/((ENTT+ENTT2)*(AF+ABT)*TL)
C......CALCULATION OF SHELLSIDE CROSSFLOW.
      ACS=W*TL-ENTT*D2*TL-ENTT*2.*FH*FPF*TL*FTH
C........CALCULATION OF TUBESIDE FLOW-CROSSECTION.
      ACT=ENTPP*D1**2*PI/4.
C
C
C     HEAT TRANSFER COEFFICIENT CALCULATION
C
C
C........INITIALIZE TEMPERATURES AND ENERGY REQUIRE
D FOR VAPORIZATION.
```

```
      TTSL=TEMSL
      TTTL=TEMTL
      TSL=TEMSL
      TTL=TEMTL
      QBCK=DHVAP*GT
      QCK=QBCK
C........BEGIN ITERATIONS BY SETTING THE PHYSICAL P
ROPERTIES AT THEIR
C............RESPECTIVE VALUES.
   30 CONTINUE
      ITER=ITER+1
      CALL PROPT(TTL,CPT1,VIT1,THERT1,ROET1)
      CALL PROPT2(TTL,CPT2,VIT2,THERT2,ROET2)
C........DETERMINE THE WEIGHT FRACTION OF VAPOR ON
THE TUBESIDE AND
C............CALCULATE THE TUBESIDE PHYSICAL PROPER
TIES.
      FRAC=QB/QBCK
      IF(FRAC.GT.1.)FRAC=1.
      CPT=FRAC*CPT2+(1.-FRAC)*CPT1
      VIT=FRAC*VIT2+(1.-FRAC)*VIT1
      THERT=FRAC*THERT2+(1.-FRAC)*THERT1
      ROET=FRAC*ROET2+(1.-FRAC)*ROET1
C........CALCULATE THE SHELLSIDE PHYSICAL PROPERTIE
S.
      CALL PROPS(TSL,CPS,VIS,THERS,ROES)
C........DETERMINE THE MULTIPLIER FOR HF2 CALCULATI
ON BELOW DEPENDENT
C............UPON SQUARE OR TRIANGULAR TUBE LAYOUT
C............SIDE HEAT TRANSFER COEFFICIENT EQUATIO
N THAT APPLIES IF
C............SQUARE PITCH IS CHOSEN).
      EK=.30
      IF(IOPT.GE.2)EK=.45
C........TUBESIDE REYNOLDS NUMBER CALCULATION.
      RET=D1*GT/(VIT*ACT)
C........SHELLSIDE REYNOLDS NUMBERS WITH RESPECT TO
C............VOLUMETRIC EQUIVALENT DIAMETER AND OUT
SIDE DIAMETER.
      RES=DE*GS/(VIS*ACS)
      RER=DEV*GS/(VIS*ACS)
      RERS=D2*GS/(VIS*ACS)
C........CALCULATION OF TUBESIDE HEAT TRANSFER COEF
-FICIENT USING
```

```
C............CORRELATION SUGGESTED BY SIEDER AND TA
TE.
      HT=.023*(RET**.8)*((CPT*VIT/THERT)**.3333)*TH
ERT/D1
      IF(TTL.GE.TBOIL.AND.QB.LE.QBCK)HT=10000.
C........CALCULATION OF TUBE WALL COEFFICIENT.
      HW=TKF*(D2-D1)/(ALOG(D2/D1)*TWTH*D1)
C........CALCULATION OF TOTAL INSIDE COEFFICIENT(IN
CLUDING TUBE WALL
C............RESISTANCE,BUT EXCLUDING OUTSIDE RESIS
TANCES).
      HTI=1./(1./HT+1./HDI+1./HW)
C
C........NOTE-FIN SIDE COEFFICIENT IS CALCULATED BY
C............CORRELATIONS FOR COMPARISON SAKE.
C
C........CALCULATION OF FIN SIDE HEAT TRANSFER COEF
FICIENT USING
C............CORRELATION SUGGESTED BY JAMESON.
      SL1=.7293877575
      B1=-1.075296549
      AJS=ALOG10(RES)*SL1+B1
      AJS=10.**AJS
      HF=AJS*THERS*((CPS*VIS/THERS)**.3333333333333
3)/DE
C........CALCULATION OF FIN SIDE HEAT TRANSFER COEF
FICIENT AS SUGGESTED
C............BY SCHMIDT.
      HF2=EK*(RERS**.625)*(((AF+ABT)/(D2*PI))**(-.3
     1**.333333)*THERS/D2
C........CALCULATION OF FIN SIDE HEAT TRANSFER COEF
FICIENT AS SUGGESTED
C............BY WARD AND YOUNG.
      HF3=.364*(RERS**.68)*(CPS*VIS/THERS)**.33333*
     2)*(FTH/(D2+2.*FH))**.30*THERS/D2
C........CALCULATION OF TOTAL OUTSIDE HEAT TRANSFER
      HF=1./((1./HF)+(1./HDO))
      HF2=1./((1./HF2)+(1./HDO))
      HF3=1./((1./HF3)+(1./HDO))
C........CALCULATION OF FIN EFFICIENCY( AS SUGGESTE
D BY PERRY).
      FNEFF=FH*SQRT(HF/(TKF*(FTH/2.)))
      FNEF1=TANH(FNEFF)/FNEFF
```

```
          FNEFF=FH*SQRT(HF2/(TKF*(FTH/2.)))
          FNEF2=TANH(FNEFF)/FNEFF
          FNEFF=FH*SQRT(HF3/(TKF*(FTH/2.)))
          FNEF3=TANH(FNEFF)/FNEFF
C........CALCULATION OF TOTAL OUTSIDE COEFFICIENT W
ITH NORMALIZATION
C............TO INSIDE AREA.
          HFI=(FNEF1*AF+ABT)*HF/AIT
          HFI2=(FNEF2*AF+ABT)*HF2/AIT
          HFI3=(FNEF3*AF+APT)*HF3/AIT
C........CALCULATION OF OVERALL HEAT TRANSFER COEFF
ICIENT WITH RESPECT
C............TO INSIDE AREA.
          UI=1./(1./HFI+1./HTI)
          UI2=1./((1./HFI2)+(1./HTI))
          UI3=1./((1./HFI3)+(1./HTI))
C
C
C         ENERGY  BALANCE
C
C
C........SET DELTA T FOR SHELLSIDE, THEREBY INCREME
NTING THE AMOUNT OF
C............HEAT TRANSFERRED IN THIS ITERATION.
          DT=DELT
C........CHECK TO SEE IF OVER MAXIMUM SHELLSIDE TEM
PERATURE.
          IF(TEMSH.LT.(TSL+DT))DT=TEMSH-TSL
     90   CONTINUE
          TSH=TSL+DT
C........CALCULATE THE DIFFERENTIAL AMOUNT OF HEAT
TRANSFERRED( USING
C............EULERS METHOD).
          Q=GS*CPS*DT
C........CHECK TO SEE IF ARE IN BOILING REGION.
          IF(TTL.EQ.TBOIL.AND.QB.LT.QBCK)GOTO100
C........IF NOT IN BOILING REGION SKIP OVER ENERGY
BALANCE FOR BOILING
C............SECTION.
          GOTO110
C........ENERGY BALANCE FOR BOILING SECTION.
    100   CONTINUE
          QB=QB+Q
```

```
C........CHECK TO SEE IF OUT OF BOILING SECTION PRE
MATURELY.
      IF(QB.GT.QBCK)QB=QB-Q
      TTH=TTL
      GOTO111
 110  CONTINUE
C........CALCULATE HIGH TEMPERATURE IF NOT BOILING
SECTION.
      TTH=TTL+Q/(GT*CPT)
 111  CONTINUE
C........CHECK TO SEE IF PREMATURELY EXITED ANY SEC
TION OF ENERGY
C............BALANCE CALCULATIONS. IF SO RESET DELT
A T SO THAT ENERGY
C............BALANCE COMES OUT EXACT FOR EACH SECTI
ON.
      IF(TBOIL.LT.TTH.AND.IFLAG.EQ.2)GOTO91
      IF(QCK.EQ.0.0.AND.IFLAG.EQ.1)GOTO93
      IF(QCK.LT.0.AND.IFLAG.EQ.1)GOTO92
      GOTO93
  91  CONTINUE
      DT=(TBOIL-TTL)*GT*CPT/(GS*CPS)
      GOTO90
  92  CONTINUE
      DT=QCK/(CPS*GS)
      GOTO90
  93  CONTINUE
      QCK=QBCK-QB
C........INCREMENT TOTAL HEAT TRANSFERRED AND HEAT
TRANSFERRED IN
C............THIS SECTION.
      QT=QT+Q
      TQ=TQ+Q
C........CHECK FOR NEGATIVE DELTA T. IF SO, EXIT PR
OGRAM.
      DTCK1=TSH-TTH
      DTCK2=TSL-TTL
      IF(DTCK1.LT.DTMIN)DTMIN=DTCK1
      IF(DTCK2.LT.DTMIN)DTMIN=DTCK2
      IF(DTCK1.LE.0.0.OR.DTCK2.LE.0.0)GOTO250
C........CALCULATE THE DELTA T AVERAGE.
      DTLM=((TSH-TTH)+(TSL-TTL))/2.
C
```

```
C
C      CALCULATION  OF  AREA
C
C
C........CALCULATE DIFFERENTIAL AREA THAT RESULTS F
ROM DIFFERENTIAL
C............AMOUNT OF HEAT TRANSFERRED, AND INCREM
ENT ALL TOTAL AREAS
C............AND ALL AREAS FOR INDIVIDUAL SECTIONS.
       DA=Q/(DTLM*UI)
       DA2=Q/(DTLM*UI2)
       DA3=Q/(DTLM*UI3)
       TA=TA+DA
       TTA=TTA+DA
       TA2=TA2+DA2
       TTA2=TTA2+DA2
       TA3=TA3+DA3
       TTA3=TTA3+DA3
C
C
C      PRESSURE DROP CALCULATION
C
C
C........SET WHICH AREA THE PRESSURE DROP ON THE SH
ELLSIDE WILL BE
C............CALCULATED WITH RESPECT TO.
       DD=DA
       IF(IEQN.EQ.2)DD=DA2
       IF(IEQN.EQ.3)DD=DA3
C........CALCULATE THE DIFFERENTIAL HEIGHT THAT THI
S ITERATION
C............CORRESPONDS TO(FOR SHELLSIDE PRESSURE
DROP CALCULATION).
       DH=DD*TPL*FNRPP/(ENTPP*AIT*TL)
C........INCREMENT THE HEIGHT OF THE EXCHANGER.
       HIGT=HIGT+DH
C........CALCULATE THE DIFFERENTIAL PRESSURE DROP O
N THE SHELLSIDE AS
C............SUGGESTED BY ROBINSON AND BRIGGS.
       DELPS=(.256/144.)*(RERS**(-.264))*(FTH/(D2+2.
      2(FPF*D2))**(-.396)) *((GS**2)*DH/(D2*ROES*(AC
S**2)*4.16923E8))
C........CALCULATE THE DIFFERENTIAL PRESSURE DROP O
-N THE SHELLSIDE AS
```

```
C............SUGGESTED BY GUNTER AND SHAW.
      FFO=ALOG10(RER)*(-.1320771677)-2.093593641
      FFO=10.**FFO
      DLPS2=FFO*((GS/ACS)**2)*DH/(8.365E8*DEV*ROES)
     1TPL/TTP)**.6)
C........CALCULATE THE DIFFERENTIAL PRESSURE DROP O
N THE TUBESIDE AS
C............SUGGESTED BY KERN.
      DELPT=((.00673+.60/(RET**.32))/144.)*(GT**2)*
     165E8*D1*ROET*(ACT**2))
C........INCREMENT THE TOTAL PRESSURE DROP AND THE
PRESSURE DROP FOR
C............THIS SECTION, FOR BOTH THE SHELL AND T
UBE SIDES.
      PDRPS=PDRPS+DELPS
      TPDRS=TPDRS+DELPS
      PDPS2=PDPS2+DLPS2
      PDRPT=PDRPT+DELPT
      TPDRT=TPDRT+DELPT
C........CALCULATE ENERGY NEEDED TO OVERCOME PRESSU
RE DROP.
      FTPT=FTPT+DELPT/ROET*144.
      FTPS=FTPS+DELPS/ROES*144.
C
C
C     PRINT STATEMENTS AND FORMATS
C
C
C........CHECK TO SEE IF THIS ITERATION MUST BE PRI
NTED.
      ICONT=ICONT+1
      IF(ICONT.EQ.IPRNT)GOTO60
      IF(ITER.EQ.1)GOTO60
      GOTO65
   60 CONTINUE
      ICONT=0
      PRINT 513
      PRINT 511,ITER
      PRINT 552,RET,RES,RERS
      PRINT 503,RER
      PRINT 510
      PRINT 551,TSL,CPS,VIS,THERS,ROES
      PRINT 555,FNEF1,FNEF2,FNEF3
```

```
      PRINT 556,HF,HF2,HF3
      PRINT 557,HFI,HFI2,HFI3
      PRINT 554,HDO
      PRINT 510
      PRINT 550,TTL,CPT,VIT,THERT,ROET
      PRINT 558,HT,HW,HDI
      PRINT 510
      PRINT 559,UI,UI2,UI3,DTLM
      PRINT 560,DA,DA2,DA3
      PRINT 561,DELPT,DELPS,Q
   65 CONTINUE
C........CHECK TO SEE IF OUT OF THE SENSIBLE HEATIN
G REGION.
      IF(TTH.EQ.TBOIL.AND.IFLAG.EQ.2)GOTO75
      GOTO76
   75 CONTINUE
C........CALCULATE THE LOGMEAN DELTA T FOR THIS SEC
TION.
      DTLMF=((TSH-TTH)-(TTSL-TTTL))/ALOG((TSH-TTH)/
C........CALCULATE THE DESIGN COEFFICIENTS FOR THIS
      UD1=TQ/(TTA*DTLMF)
      UD2=TQ/(TTA2*DTLMF)
      UD3=TQ/(TTA3*DTLMF)
      PRINT 519
      DO 300 I=1,8
      PRINT 520
  300 CONTINUE
      PRINT 515,UD1,UD2,UD3,DTLMF
      PRINT 516,TTA,TTA2,TTA3
      PRINT 517,TPDRT,TPDRS
      PRINT 518,TQ
      DO 301 I=1,8
      PRINT 520
  301 CONTINUE
      PRINT 519
C........REINITIALIZE ALL SECTIONAL PARAMETERS AND
RESET IFLAG.
      TQ=0.
      TTA=0.
      TTA2=0.
      TTA3=0.
      TPDRT=0.
      TPDRS=0.
```

```
      TTSL=TSL
      TTTL=TTL
      IFLAG=1
      GOTO80
   76 CONTINUE
C........CHECK TO SEE IF OUT OF THE BOILING REGION.
      IF(QB.EQ.QBCK.AND.IFLAG.EQ.1)GOTO78
      GOTO80
   78 CONTINUE
C........CALCULATE THE LOGMEAN DELTA T FOR THIS SEC
TION.
      DTLMF=((TSH-TTH)-(TTSL-TTTL))/ALOG((TSH-TTH)/
C........CALCULATE THE DESIGN COEFFICIENTS FOR THIS
      UD1=TQ/(TTA*DTLMF)
      UD2=TQ/(TTA2*DTLMF)
      UD3=TQ/(TTA3*DTLMF)
      PRINT 513
      DO 305 I=1,8
      PRINT 520
  305 CONTINUE
      PRINT 515,UD1,UD2,UD3,DTLMF
      PRINT 516,TTA,TTA2,TTA3
      PRINT 517,TPDRT,TPDRS
      PRINT 518,TQ
      DO 306 I=1,8
      PRINT 520
  306 CONTINUE
      PRINT 519
C........REINITIALIZE ALL SECTIONAL PARAMETERS AND
RESET IFLAG.
      TTA=0.
      TTA2=0.
      TTA3=0.
      TQ=0.
      TPDRT=0.
      TPDRS=0.
      TTSL=TSL
      TTTL=TTL
      IFLAG=0
   80 CONTINUE
C........CHECK TO SEE IF SHELLSIDE TEMPERATURE HAS
REACHED THE
C............MAXIMUM, IF SO EXIT THE ITERATIVE LOOP
.
```

```
      IF(TSH.GE.TEMSH)GOTO70
C........RESET THE LOW TEMPERATURES AND CONTINUE TH
E ITERATIONS.
      TSL=TSH
      TTL=TTH
      GOTO30
   70 CONTINUE
C........CALCULATE THE NUMBER OF TUBE ROWS NEEDED I
N THE EXCHANGER.
      ENLT=HIGT/TPL
C........CALCULATE THE LOGMEAN DELTA T FOR THIS SEC
TION.
      DTLMF=((TSH-TTH)-(TTSL-TTTL))/ALOG((TSH-TTH)/
C........CALCULATE THE DESIGN COEFFICIENTS FOR THIS
      UD1=TQ/(TTA*DTLMF)
      UD2=TQ/(TTA2*DTLMF)
      UD3=TQ/(TTA3*DTLMF)
      BTUS=FTPS*GS*.0012854
      BTUT=FTPT*GT*.0012854
      IF(IFLAG.EQ.2)PRINT 509
      IF(IFLAG.EQ.1)PRINT513
      IF(IFLAG.EQ.0)PRINT514
      DO 310 I=1,8
      PRINT 520
  310 CONTINUE
      PRINT 515,UD1,UD2,UD3,DTLMF
      PRINT 516,TTA,TTA2,TTA3
      PRINT 517,TPDRT,TPDRS
      PRINT 518,TQ
      DO 311 I=1,8
      PRINT 520
  311 CONTINUE
      PRINT 519
      PRINT 510
      PRINT 510
      PRINT 510
      PRINT 512
      DO 320 I=1,8
      PRINT 520
  320 CONTINUE
      PRINT 510
      PRINT 551,TSH,CPS,VIS,THERS,ROES
      PRINT 550,TTH,CPT,VIT,THERT,ROET
```

```
      PRINT 504,GS,GT
      PRINT580,DTMIN
      PRINT 510
      PRINT 512
      IF(IOPT.GE.2)GOTO200
      GOTO201
200   CONTINUE
      PRINT 500,TTP,TPL
      GOTO202
201   CONTINUE
      PRINT 501,TTP
202   CONTINUE
      PRINT 505,D1,D2,TWTH
      PRINT 502.DE,DEV
      PRINT506,FH,FTH,FPF
      PRINT 507,AIT,AF,ABT
      PRINT 508,ACS,ACT
      PRINT 510
      PRINT 564,W,TL,HIGT
      PRINT 565,ENTT,ENRPP,ENLT
      PRINT 562,TA,TA2,TA3
      PRINT 563,PDRPT,PDRPS,IEQN
      PRINT 570,PDPS2
      PRINT 571,BTUT,BTUS
      PRINT 510
      PRINT 567,QT
      DO321 I=1,8
      PRINT 520
321   CONTINUE
      PRINT 519
      GOTO251
250   CONTINUE
      PRINT 599,DTCK1,DTCK2
251   CONTINUE
      IF(PDRPS.GT.PDMAX)GOTO150
      GOTO151
150   CONTINUE
      PCK=PDRPS-PDMAX
      ENTT=((PDRPS/PDMAX)**.4405)*ENTT
      ENTT=AINT(ENTT)
      IF(PCK.LT.0.4)ENTT=ENTT+1.0
      SIGNAL=1.0
      GOTO 5
```

```
 151  CONTINUE
      IF(DUMM1.EQ.DUMM2)GOTO260
      GO TO 5
 260  CONTINUE
 500  FORMAT(//,* TRIANGULAR PITCH TUBES WITH TRANV
. PITCH =*,F8.6,* FT.
     1 AND LONGT. PITCH =*,F8.6,* FT.*)
 501  FORMAT(//,* SQUARE PITCH TUBES WITH TRANV. AN
D LONGT. PITCH =*,F8.
     16,* FT.*)
 502  FORMAT(/,* EQUIVALENT DIAM. = *,F10.6,* FT.
     1LENT DIAM. = *,F10.6,* FT.*)
 503  FORMAT(/,* RE SHELLSIDE (W.R.T. VOLUMETRIC EQ
UIV. DIAM. ) = *,F10.
     14)
 504  FORMAT(/,* MASS FLOWRATE SHELLSIDE = *,F12.2,
     1S FLOWRATE TUBESIDE = *,F12.2,* LB. PER HR. *
 505  FORMAT(/,* TUBE I.D. =*,F8.6,* FT.    TUBE O.
D. =*,F8.6,* FT.     T
     1UBE WALL THICKNESS =*,F8.6,* FT.*)
 506  FORMAT(/,* FIN HEIGHT =*,F8.6,* FT.    FIN TH
ICKNESS =*,F8.6,* FT.
     1  WITH*,F8.4,* FINS PER FOOT*)
 507  FORMAT(/,* AREA INSIDE TUBE =*,F8.6,* SQ. FT.
     1 =*,F8.6,* SQ. FT. PER FT.    AREA BARE TUBE
     2 FT.*)
 508  FORMAT(/,* AREA CROSS-SECT SHELL =*,F10.6,*SQ
. FT.    AREA CROSS-S
     1ECT TUBE =*,F10.6,* SQ. FT.*)
 509  FORMAT(/////,* .............................
.....  SENSIBLE HEATI
     1NG REGION  ................................
.. *)
 510  FORMAT(//)
 511  FORMAT(////,*........................  ITERA
TION NUMBER IS *,I5,*
     1    ........................ *,////)
 512  FORMAT(* ...................................
...  END OF ITERATION
     1S  ................................... *)
 513  FORMAT(/////,* .............................
..........  BOILING R
     1EGION  .....................................
.. *)
```

```
 514  FORMAT(/////,* ..............................
.......  SUPERHEATING
     1 REGION  ....................................
.. *)
 515  FORMAT(////,* OVERALL DESIGN COEFF. = *,F10.4
,5X,F10.4,5X,F10.4,5X
     1,* DELTA T LOGMEAN = *,F10.4)
 516  FORMAT(/,* ACCUM. AREA = *,10X,F10.4,5X,F10.4
,5X,F10.4,* SQ. FT.*)
 517  FORMAT(/,* ACCUM. PR. DRP. TUBESIDE = *,F10.4
,5X,* ACCUM. PR. DRP.
     1 SHELLSIDE = *,F10.4,* PSI *)
 518  FORMAT(/,* HEAT LOAD THIS REGION = *,F12.2,*
BTU PER HR. *,////)
 519  FORMAT(* ....................................
....................
     1......................................... *)
 520  FORMAT(* . *,94X,* . *)
 550  FORMAT(/,* TUBESIDE .... TEMP =*,F8.2,* HT. C
AP. =*,F8.6,* VISC. =
     1*,F8.4,* TH. COND. =*,F8.6,* DENSITY =*,F8.4)
 551  FORMAT(/,* SHELLSIDE ..... TEMP. =*,F8.2,* HT
. CAP. =*,F8.6,* VIS.
     1 =*,F8.6,* TH. COND. =*,F8.6,* DENSITY =*,F8.
4)
 552  FORMAT(/,* RE TUBESIDE =*,F12.4,* RE SHELLSID
E =*,F12.4,* RE SHELL
     1SIDE( W.R.T. OD.) =*,F12.4)
 554  FORMAT(/,* FOULING COEFF. OUTSIDE = *,F10.3)
 555  FORMAT(/,* FIN EFFICIENCY =*,12X,F10.6,5X,F10
.6,5X,F10.6)
 556  FORMAT(/,* FIN HT. TRAN. COEFF. =*,6X,F10.6,5
X,F10.6,5X,F10.6)
 557  FORMAT(/,* FIN SIDE (W.R.T. IN. AREA) =*,F10.
 558  FORMAT(/,* TUBESIDE COEFF. = *,F10.6,* TUBEWA
LL COEFF. = *,F10.6,
     1* FOULING COEFF. INSIDE = *,F10.3)
 559  FORMAT(/,* OVERALL COEFF. (INSIDE) =*,F10.6,5
X,F10.6,5X,F10.6,*
     1DELTA T =*,F10.5)
 560  FORMAT(/,* DIFFL. AREA =*,12X,F10.6,5X,F10.6,
 561  FORMAT(/,* DIFFL. PR. DRP. TUBESIDE =*,F10.6,
     1LLSIDE =*,F10.6,* DIFFL. HEAT TRANSED. =*,F10
.2)
```

```
 562  FORMAT(/,* TOTAL AREA =*,F12.4,5X,F12.4,5X,F1
2.4)
 563  FORMAT(/,* PR. DRP. TUBESIDE =*,F10.6,* PR. D
RP. SHELLSIDE =*,F10.
     16,* W.R.T. EQUATION NO. *,I2)
 564  FORMAT(/,* EXCHANGER DIMENSIONS (W,L,H)......
     11C.5,* FT.*)
 565  FORMAT(/,* WITH *,F8.2,* TUBES PER TRAN. ROW
AND*,F8.2,* ROWS PER
     1PASS,   *,F8.2,* LONGITTUDINAL ROWS HIGH*)
 567  FORMAT(/,* TOTAL HEAT LOAD = *,F12.2,* BTU PE
R HOUR *)
 570  FORMAT(/,* PR. DRP. SHELLSIDE (GUNTER AND SHA
W) = *,F10.6,* PSI *)
 571  FORMAT(/,* ENERGY REQUED. FOR PR. DRP. TUBESI
DE = *,F12.2,* BTU PE
     1R HR.     ENERGY REQUED. FOR PR. DRP. SHELLSI
DE = *,F12.2,* BTU PE
     2R HR.*)
 580  FORMAT(/,* MINIMUM DELTA T ITERATED = *,F10.3
 599  FORMAT(/////,* DELTA T 1 = *,F10.5,* DELTA T
2 = *,F10.5,* PROGRAM
     1 TERMINATED DUE TO THERMODYNAMICALLY IMPOSSIB
LE DELTA T. *)
      END
      SUBROUTINE PROPS(TEMSH,CPS,VIS,THERS,ROES)
C........SUBROUTINE CONTAINING SHELLSIDE PHYSICAL P
ROPERTIES.
      CPS=1.78604E-5*TEMSH+.25556
      VIS=3.4470039E-2+6.150348E-5*TEMSH-1.257735E-
      THERS=1.81742E-5*TEMSH+.0127591
      ROES=.0602497827-8.175549E-5*TEMSH+4.37484E-8
      RETURN
      END
      SUBROUTINE PROPT(TEMTH,CPT,VIT,THERT,ROET)
C........SUBROUTINE CONTAINING THE PHYSICAL PROPERT
IES FOR THE
C............TUBESIDE LIQUID.
      CPT=1.049011178
      ROET=62.4
      T=(TEMTH-32.)/1.8
      VIT=242./(2.1482*((T-8.435)+SQRT(8078.4+(T-8.
435)**2))-120.)
```

```
      THERT=2.97619E-4*TEMTH+.33345
      RETURN
      END
      SUBROUTINE PROPT2(TTL,CPT2,VIT2,THERT2,ROET2)
C........SUBROUTINE CONTAINING THE PHYSICAL PROPERT
IES FOR THE
C............TUBESIDE VAPOR.
C........ 400 PSI STEAM
      ROET2=1.598607213-(.0021254274*TTL)+(.0000010
038698*TTL**2)
      CPT2=1.727892263-(.0029896996*TTL)+(.00000183
      T=(TTL-32.)/1.8
      R=ROET2/62.4
      VIT2=(353.*R+676.5*(R **2)+102.1*(R**3)+.407*
T+80.4)*.000242
      THERT2=((103.51+.4198*T-.00002771*(T**2))*R+2
.1482E14*(R**2)/(T**4
     1.2)+(17.6+.0587*T+.000104*(T**2)-.0000000451
      RETURN
      END
      SUBROUTINE PARAM(IFIN,IPRNT,IEQN,IOPT,GS,GT,D
HVAP,TTP,TL,FTH,D2,EN
     1TT,ENRPP,FPF,FH,TWTH,TKF,TBOIL,TEMTL,TEMSL,TE
MSH,DELT,HDO,HDI,DUMM
     21,DUMM2,PDMAX,SIGNAL)
C........SUBROUTINE CONTAINING THE INPUT PARAMETERS
      IOPT=2
      GS=53950.
      TEMTL=212.
      TBOIL=444.6
      DHVAP=780.43
      TEMSH=750.
      IFIN=2
      IPRNT=1000
      IEQN=2
      TTP=.1666666667
      TL=8.
      FTH=.002916667
      D2=.08333333333
      ENRPP=1.
      FPF=96.
      FH=.03125
      TWTH=.00691667
      TKF=26.
      DELT=5.
      HDO=1000.
      HDI=1000.
      PDMAX=.2888
      RETURN
      END
```

Line 48 calculates the fin side equivalent diameter as

$$DE = \frac{2\ (AF + ABT)}{\pi \times PP}$$

Where DE = Equivalent diameter (ft).

Line 49 determines the width of the heat exchanger as

$$W = ENTT \times TTP$$

Where W = Width (ft)

ENTT = Number of transverse tubes

TTP = Transverse tube pitch (ft)

Lines 50 and 51 determine the net volume as

$$VNET = VTOT - VTUBE - VFIN$$

Where VTOT = Total plenum volume (ft^3)

VTUBE = Total volume of tubing (ft^3)

VFIN = Total volume of fins (ft^3)

Line 52 calculates the volumetric equivalent diameter as

$$DEV = \frac{4 \times VNET}{ENTT \times TL \times (AF + ABT)}$$

Where TL = Tube length (ft)

DEV = Volumetric equivalent diameter (ft)

Lines 53 and 54 complete the heat exchanger dimensional characteristics section and compute the shell side and tube side area of cross section to flow as:

$$ACS = W \times TL - ENTT(D2 \times TL - 2 \times FH \times FPF \times TL \times FTH)$$

and

$$ACT = \frac{ENTPP \times D1^2 \times \pi}{4}$$

Where ACS = Shell cross sectional area (ft^2)

ACT = Tube cross sectional area (ft^2)

ENTPP = Number of transverse tubes per tube pass

D1 = Inside diameter of tube (ft)

Line 61 is the beginning of the major iterative loop. Line 62 increments the iteration number. Lines 63 and 64 call the two tube side physical property subroutines for the liquid and vapor and update all tube side physical properties. If the tube side materials are partially liquid and vapor a weighted average set of physical properties is calculated in Lines 65 through 70. Finally, the shell side physical properties are updated in Line 71.

The tube side Reynold's number is calculated in Line 74 as:

$$RET = \frac{D1 \times GT}{VIT \times ACT}$$

Where RET = Tube side Reynold's Number

FT = Tube side Mass Flow Rate (lbs/hr)

VIT = Tube side Viscosity (#/Ft-Hr)

Three shell side Reynold's numbers are calculated in Lines 75 through 77 as follows:

$$RES = \frac{DE \times GS}{VIS \times ACS}$$

$$RER = \frac{DEV \times GS}{VIS \times ACS}$$

$$RERS = \frac{D2 \times GS}{VIS \times ACS}$$

Where RES = Reynolds number wrt equivalent diameter

RER = Reynolds number wrt volumetric equivalent dia.

RERS = Reynolds number wrt tube outside diameter

GS = Shell side mass flow rate (lbs/hr)

VIS = Shell side viscosity (#/Ft-Hr)

Lines 78 through 81 determine the inside heat transfer coefficient, tube wall coefficient and overall inside heat transfer coefficients, respectively, as:

$$HT = .023\ RET^{.8} \left[\frac{CPT \times VIT}{THERT}\right]^{1/3} \times \frac{THERT}{D1}$$

$$HW = \frac{TKF\ (D2 - D1)\ TWTH \times D1}{\ln [D2/D1]}$$

$$HTI = \frac{1}{\left(\frac{1}{HW} + \frac{1}{HT} + \frac{1}{HDI}\right)}$$

Where HT = Tube side coefficient calculated using Sieder-Tate correlation $\left(\frac{BTU}{Hr\ ft^2\ F°}\right)$

HW = Tubewall coefficient $\left(\frac{BTU}{Hr\ Ft^2\ F°}\right)$

TKF = Thermal Conductivity of tubing material

$\left(\frac{BTU}{Hr\ Ft\ F^\circ}\right)$

TWTH = Tubewall thickness (ft)

HTI = Overall heat transfer coefficient with respect to the inside area $\left(\frac{BTU}{Hr\ Ft^2\ F^\circ}\right)$

HDI = Inside tube fouling coefficient $\left(\frac{BTU}{Hr\ Ft^2\ F^\circ}\right)$

Lines 82 through 105 determine the finside (outside) heat transfer coefficient and combine this value with a fin efficiency. Subsequently, an overall heat transfer coefficient with respect to the inside surface area is calculated by incorporating the inside heat transfer coefficient with that of the fin side. Three separate calculations of the fin side coefficient are performed using three separate correlations. Because of the predominant effect of the fin side coefficient on the overall heat transfer coefficient it was deemed necessary to compare these three correlations and pick the most conservative of the three for design purposes.

The first of the three correlations appears in Lines 82 through 86, and is suggested by Jameson as follows:

$$HF = \frac{AJS \times THERS}{DE} \times \left[\frac{CPS \times VIS}{THERS}\right]^{1/3}$$

Where LOG (AJS)= .7294 x RES - 1.075

HF = Jamesons' Finside heat transfer coefficient $\left(\frac{BTU}{Hr\ Ft^2\ F°}\right)$

The second correlation suggested by Schmidt appears in Line 87 as:

$$HF2 = \frac{EK \times THERS}{D2} \times RERS^{.625} \times \left[\frac{AF + ABT}{\pi \times D2}\right]^{-.375} \times \left[\frac{CPS \times VIS}{THERS}\right]^{1/3}$$

Where HF2 = Schmidts' Finside heat transfer coefficient $\left(\frac{BTU}{Hr\ Ft^2\ F°}\right)$

EK = Performance factor for square or triangular pitch tubes (.30 or .45, respectively)

The final correlation is that suggested by Ward and Young and appears in Line 89 as:

$$HF3 = \frac{.364\ THERS}{D2} \times RERS^{.68} \times \left[\frac{CPS \times VIS}{THERS}\right]^{1/3} \times \left[\frac{D2 + 2FH}{D2}\right]^{.45} \times \left[\frac{FTH}{D2 + 2FH}\right]^{.30}$$

Where HF3 = Ward and Youngs' Finside heat transfer coefficient $\left(\frac{BTU}{Hr\ Ft^2\ F°}\right)$

Lines 91 through 93 are analogous to Line 81 in that they incorporate a fouling coefficient into the oustide heat transfer coefficients. This fouling coefficient is labeled HDO and is provided by subroutine Param earlier in the program.

The fin efficiency is calculated using the method suggested by Perry as:

$$FNEF_n = \frac{\mathrm{TANH}\left[FH \times \sqrt{\frac{HF_n \times 2}{TKF \times FTH}}\right]}{FH \times \sqrt{\frac{HF_n \times 2}{TKF \times FTH}}}$$

Where $FNEF_n$ = Fin efficiency

HF_n = Individual fin heat transfer coefficient
$n = 1, 2, 3$

Since the calculation of the fin efficiency requires the outside heat transfer coefficient as a variable and three separate outside heat transfer coefficients were determined, three fin efficiencies must be calculated.

The three outside heat transfer coefficients are combined with the fin efficiencies and the normalized to the tube inside are in Lines 100 through 102. The equation is of the form:

$$HFI_n + \frac{HF_n \times [FNEF_n \times AF + ABT]}{AIT}$$

Where HFI_n = Outside heat transfer coefficient with respect to inside area $\left(\frac{BTU}{Hr\ Ft^2\ F°}\right)$

Finally, the tube side heat transfer coefficient and fin side heat transfer coefficients are combined to form the overall heat transfer coefficient with respect to the inside area as:

$$UI_n = \frac{1}{\left[\frac{1}{HFI_n} + \frac{1}{HTI}\right]}$$

Where UI = The three overall heat transfer coeffients $\left(\frac{BTU}{Hr\ Ft^2\ F°}\right)$

$n = 1, 2, 3$

The next major section of the program is found in Lines 106 through 140 and is comprised of all energy balance calculations for each section of the heat exchanger. Since there is a material that undergoes a phase change in the exchanger three theoretical sections result in the calculations, a sensible heating region, a boiling region and a vapor superheating region. These three regions are examined individually by the program since heat exchange parameters will vary in each region. Each region parameters are reported upon completing the heat exchange task for that region. It is important to note that this section of the program, as was true with the previous section, is a portion of the major iterative loop. Figure 9 presents a pictorial description of the method and variables used.

Line 106 sets the current change in temperature on the shell side equal to that amount specified in subroutine Param (earlier in the program). This variable, called DT, may be modified later in the energy_balance section so as to predetermine the amount of heat energy needed to fulfill a particular section's exact energy requirement (as is done in Line 107). Line 107 assures that incrementation of the shell side low temperature (TSL) by the change in temperature

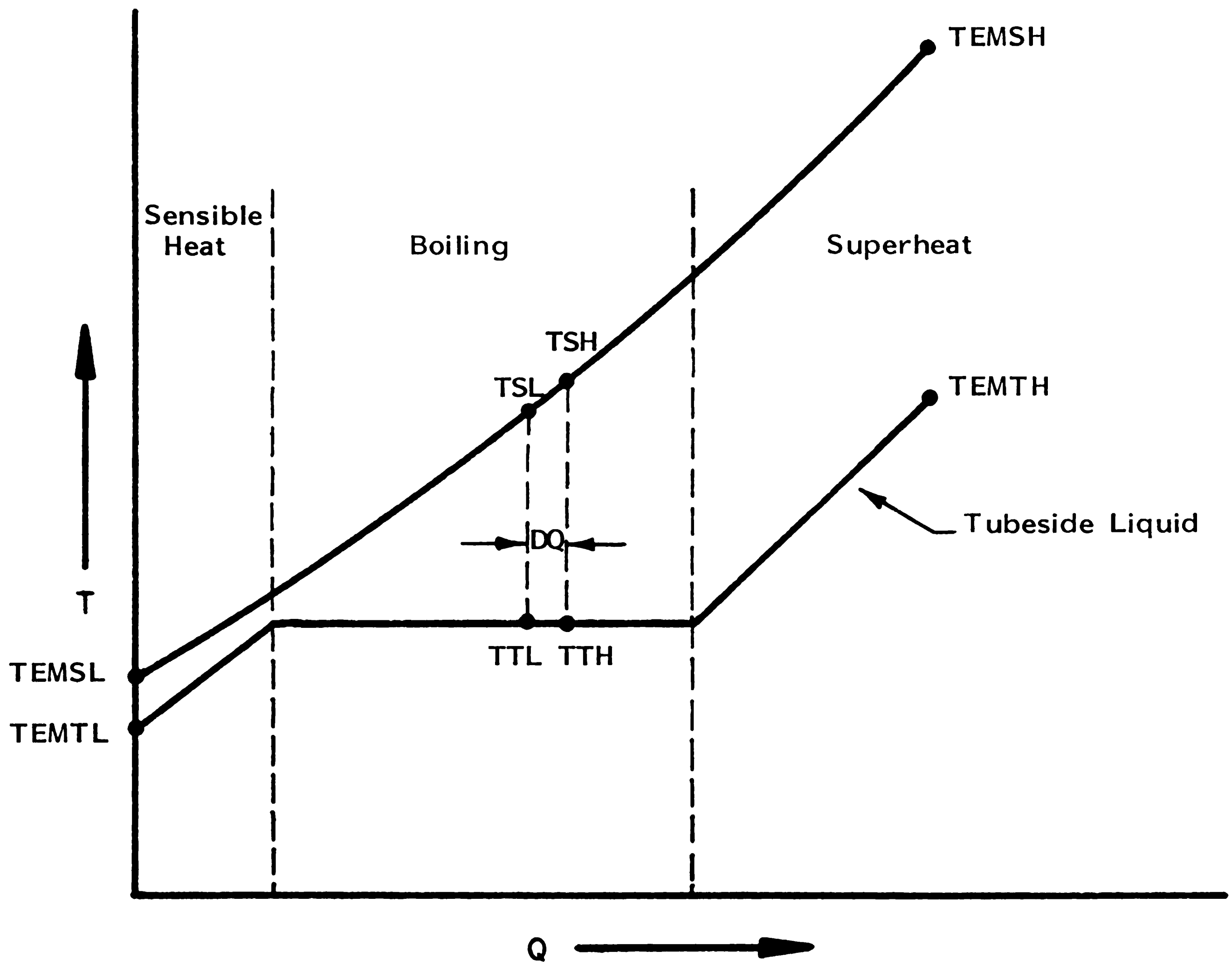


Figure 9. --Diagram of Q versus T for Waste Heat Boiler Simulation.

(DT) does not exceed the maximum shell side temperature (TEMSH). Line 110 calculates the differential quantity of heat transferred via the change in the shell side temperature as:

$$Q = GS \times CPS \times DT$$

Where Q = Differential quantity of heat transferred (BTU/Hr)

GS = Shell side Mass flow rate (#/Hr)

DT = Shell side temperature change (F°)

CPS = Shell side heat capacity $\left(\frac{BTU}{Lb\ F^{\circ}}\right)$

Note: CPS is updated each iteration by subroutine PROPS.

Line 111 determines if the program is currently performing calculations for the boiling region of the exchanger. If so, Lines 113 through 117 perform that energy balance by incrementing a boiling region energy counter (QB) and keeping the tube side temperatures equal to the boiling point of the liquid used. If not in the boiling section, the tubeside temperature is calculated in Line 119 as follows:

$$TTH = TTL + \frac{Q}{GT \times CPT}$$

Where TTH = Tube side temperature at end of iteration

TTL = Tube side temperature at beginning of iteration.

Lines 121 through 124 determine whether the incrementation of the shell side temperature by DT has caused a premature (non exact) exit from either of two sections on the tube side (i.e., sensible heat or boiling sections). If so, DT is reset to cause an exact correspondence of the energy transferred from the shell side and that transferred to the tube side.

For the sensible heating region this is accomplished in Lines 125 through 127 as:

$$DT = \frac{GT \times CPT\ [TBOIL - TTL]}{GS \times CPS}$$

Where DT = Revised shell side temperature increment (F°)

TBOIL = Tube side Boiling Point (F°)

TTL = Current iteration's tube side low temperature (F°)

Or the boiling region accomplishes this in Lines 128 through 130 as:

$$DT = \frac{QCK}{CPS \times GS}$$

Where QCK = WBCK - QB

QBCK = Energy required to vaporize tube side liquid (BTU/Hr)

QB = Total energy used in boiling section up to and including this iteration (BTU/Hr)

Lines 133 through 140 increment the total amount of heat energy transferred and the total amount of energy transferred in this section. In addition, a negative temperature difference is checked for and finally the average temperature difference is calculated as:

$$DTLM = \frac{(TSH - TTH) + (TSL - TTL)}{2}$$

Where DTLM = Average Temperature difference (F°)

TSH = Shell side iteration high temperature (F°)

TSL = Shell side iteration low temperature (F°)

TTH = Tube side iteration high temperature (F°)

TTL = Tube side iteration low temperature (F°)

The next section of the program (again a part of the major iterative loop) increments all heat transfer areas necessary to accomplish the transfer of the amount of heat energy which has been specified by the energy balance section. Three incremental areas are set as:

$$DA_n = \frac{Q}{DTLM \times UI_n}$$

Where DA = incremental area wrt IU (Ft^2)

DA2 = incremental area wrt UI2 (Ft^2)

DA3 = incremental area wrt UI3 (Ft^2)

n = 1, 2, 3

and then Lines 144 through 149 increment the total areas and individual section cumulative areas by the amount DA, DA2, or DA3, as required.

The final section of the major iterative loop is contained in Lines 150 through 169 and is comprised of the pressure drop calculations for both the shell and tube sides. Again due to the importance of accuracy in the shell side pressure drop calculations, two separate correlations are used and reported. Lines 150 through 152 determine which heat transfer correlation (shell side) has been chosen as the basis to perform the pressure drop calculations. Since the three shell side heat transfer correlations can produce three different areas necessary for heat transfer, it must be determined prior to running which correlation will be of interest.

The incremental tube side area, DP, is converted to an incremental height on the shell side by Line 153 as:

$$DH = \frac{DD \times TPL \times ENRPP}{ENTPP \times AIT \times TL}$$

Where DH = Incremental shell side height (Ft)

AIT = Area inside tubes (Ft^2/Ft)

TL = Tube length (Ft)

ENTPP = Number of tubes per pass

ENRPP = Number of rows per pass

TPL = Longitudinal tube pitch (Ft)

DD = Incremental tube side area for this iteration (Ft^2)

The shell side pressure drop for this incremental height is then calculated first as suggested by Ward and Young:

$$DELPS = \frac{4.16923 \times 10^8}{144} \times .256\ RERS^{-.264} \times \left[\frac{FTH}{D2 + 2FH}\right]^{-.377} \times \frac{GS^2 \times DH}{D2 \times ROES \times ACS^2} \times \frac{1}{[FPF \times D2]}^{-.396}$$

Where DELPS = Pressure drop Ward & Young ($\#/In^2$)

$$4.16923 \times 10^8 = gc \times 3600^2 \text{ (conversion to correct units)}$$

And then as suggested by Gunter and Shaw as:

$$DLPS2 = \frac{.96\ RER^{-.145} \times GS^2 \times DH}{(144)\ 4.16923 \times 10^8 \times DEV \times ACS^2 \times ROES} \times \left[\frac{DEV}{TTP}\right]^{.4} \times \left[\frac{TPL}{TTP}\right]^{.6}$$

Where RER = Reynold's number with respect to volumetric equivalent diameter

DEV = Volumetric equivalent diameter (Ft)

DLPS2 = Pressure drop Gunter and Shaw ($\#/In^2$)

Finally, the tubeside pressure drop is calculated as suggested by Kern:

$$DELPT = \frac{.6\ RET^{-.32} + 6.73 \times 10^{-3}}{144\ (8.316 \times 10^8)} \times \frac{GT^2 \times DD \times D1 \times ROET \times ACT^2}{AIT \times ENTPP}$$

Where DELPT = Tube side pressure drop (psi)

RET = Tube side Reynolds number

DD = Incremental area this iteration (Ft^2)

Lines 163 through 169 complete this section of the program by incrementing the respective pressure drop register by the amount determined in this iteration.

Lines 170 through 407 contain primarily output information and format statements in standard Fortran IV coding convention. Line 256 is the end of the major iterative loop which started at Line 61. The code in this large section is broken up into three output areas. One area is used for each heat transfer section of the heat exchanger (sensible heating, boiling and super heating). The sectional data is printed upon exiting a particular section of the heat exchanger and then the program continues to the next section until the heat transfer task is completed. In addition, a logmean temperature difference is calculated for each section as:

$$DTLMF = \frac{[TSH - TTH] - [TTSL - TTTL]}{l_n \left(\frac{TSH - TTH}{TTSL - TTTL}\right)}$$

and a design heat transfer coefficient is subsequently calculated as:

$$UD_n = \frac{TQ}{TTA_n \times DTLMF}$$

Where DTLMF = Lobmean temperature difference (F°)

TSH = Shell side high temperature

TTSL = Shell side low temperature

TTH = Tube side high temperature

TTTL = Tube side low temperature

TQ = Amount of heat transferred (BTU/Hr)

TTA_n = Heat transfer area (Ft^2)

UD_n = Design coefficient $\left(\frac{BTU}{Hr - Ft^2 - F°}\right)$

n = 1, 2, 3

Logmean temperature differences and the individual design coefficients are presented for informational purposes and could be used for estimation purposes when designing a similar heat exchanger.

Finally, Lines 323 through 331 perform the calculations needed to determine whether the maximum allowable shell side pressure drop has been exceeded. If so, the program will change the number of transverse tube rows, thereby changing the area of cross sectional flow on the shell side. By changing the area of cross-sectional flow, a resultant change in both the shell side Reynolds number and the shell side pressure drop will occur. The algorithm used for this determination was empirically determined by examination of a number of computer printouts and appears as:

$$DNTT = [(\frac{PDRPS}{PDMAX})]^{.4405} \times [ENTT']$$

Where ENTT = New estimation of number of transverse tube rows

ENTT' = Number of tube rows previous

PDRPS = Shell side pressure drop this trial (psi)

PDMAX = Maximum allowable shell side pressure drop (psi)

This algorithm converges in approximately three iterations for the heat exchangers discussed here and could require modification in the exponent for other systems.

In conclusion, Line 331 terminates the final loop which consists of the shell side pressure drop iterative algorithm and concludes the program.

APPENDIX II

STEAM RANKINE CYCLE FOR PRATT-WHITNEY GG-12

GAS TURBINE DRIVERS

APPENDIX II

STEAM RANKINE CYCLE FOR PRATT-WHITNEY GG-12 GAS TURBINE DRIVERS

Summary

A simple four stage rankine cycle utilizing water as the working fluid was chosen for the Pratt-Whitney GG-12 gas turbine drivers. The cycle (described in Figure 10) requires a maximum cycle temperature of 800°F and a maximum pressure of 600 psia. A steam flow rate of 29,760 lbs/hr is attainable when waste heat is used from a pair of the Pratt-Whitney drivers. The horsepower rating of such a rankine cycle is 2770 Hp resulting in a 50 percent increase in the base horsepower output of the 2750 Hp individual drivers. The large increase in horsepower output is due to the relatively low efficiency of the Pratt-Whitney turbines coupled with their large output of high temperature exhaust gases. Each driver exhausts 158,400 lbs. of clean 840°F exhaust gases per hour and operates at 16 percent thermal efficiency.

The total 1980 fixed capital investment necessary for the design case is $955,000. A payout period of approximately one year is estimated for a 90 percent stream factor increasing to 4.1 years for a 25 percent stream factor. These payout periods represent discounted cash flow rates of return (after taxes) of 73.8 and 18.1 percent, respectively.

The annual savings in natural gas usage at 90 percent stream factor is $1,343,900 at 1981 gas prices ($3.87 per million BTU). This large savings is due to the inefficiency of the Pratt-Whitney drivers and is more fully discussed in the economics section of this Appendix.

Design Basis

The basis for this design is the simultaneous operation of a pair of Pratt-Whitney GG-12 gas turbine drivers rated at 2750 Hp each. Each driver operates at 100 percent load capacity when in operation. However, stream factor is considered to be variable. Each driver produces 158,400 #/hr. of hot exhaust gases which are considered to be primarily combustion products of natural gas (Consumers Power Company). The exhaust outlet temperature of the driver is 840° maximum. Each driver can accept a waste heat boiler attached as a portion of its exhaust system with no greater than 10" water ($\approx$ .36 psi) pressure drop on the shell side.

Each compressor station is assumed to be remotely located, but with all necessary electrical supply readily available. In addition, the sites are assumed to be in a climate with a maximum air temperature of 100°F (dry bulb) and a maximum of 80°F wet bulb. No additional cooling water capacity is assumed available at any site. Finally, additional Cooper-Bessmer centrifugal compressors and transfer cases are assumed to be available for shaft work usage.

For a detailed description of economic assumptions, see the Economic Analysis section.

Process Description

Figure 10 is a schematic diagram of the four stage rankine cycle used in this design. Table 7 contains all process stream conditions for this cycle.

TABLE 7.--Thermodynamic Conditions for 600 psia Steam Rankine Cycle.

Stream Number	Pressure (psia)	Temperature (F°)	Enthalpy (BTU/#)	Entropy (BTU/#-F°)
1	14.7	192°	160.17	
2	600	192°	162.65	
3	600	800°	1408.3	1.6351
4	14.7	255°	1171.5	1.787
5	14.7	212°	180.17	.3121

Beginning with Stream #1, 29,760 lbs/hr of subcooled liquid condensate at 192°F and 14.7 psia are compressed to 600 psia and 192°F in a 60 GPM, 29.0 Hp high pressure liquid pump. The resultant stream 2 is then split into two streams of 14,880 #/hr each and flow through the two waste heat boilers in counter flow. Each waste heat boiler has an inside surface area of 1980 square feet. Each waste heat boiler is fed with an exhaust gas stream of 158,400 #/hr of hot gases which enter the boiler at 840°F and exit at 407°F. The boiler heat loads are 1.853×10^7 BTU/hr each. The two vapor streams leaving the waste heat boiler are joined to form stream 3 which is 800°F

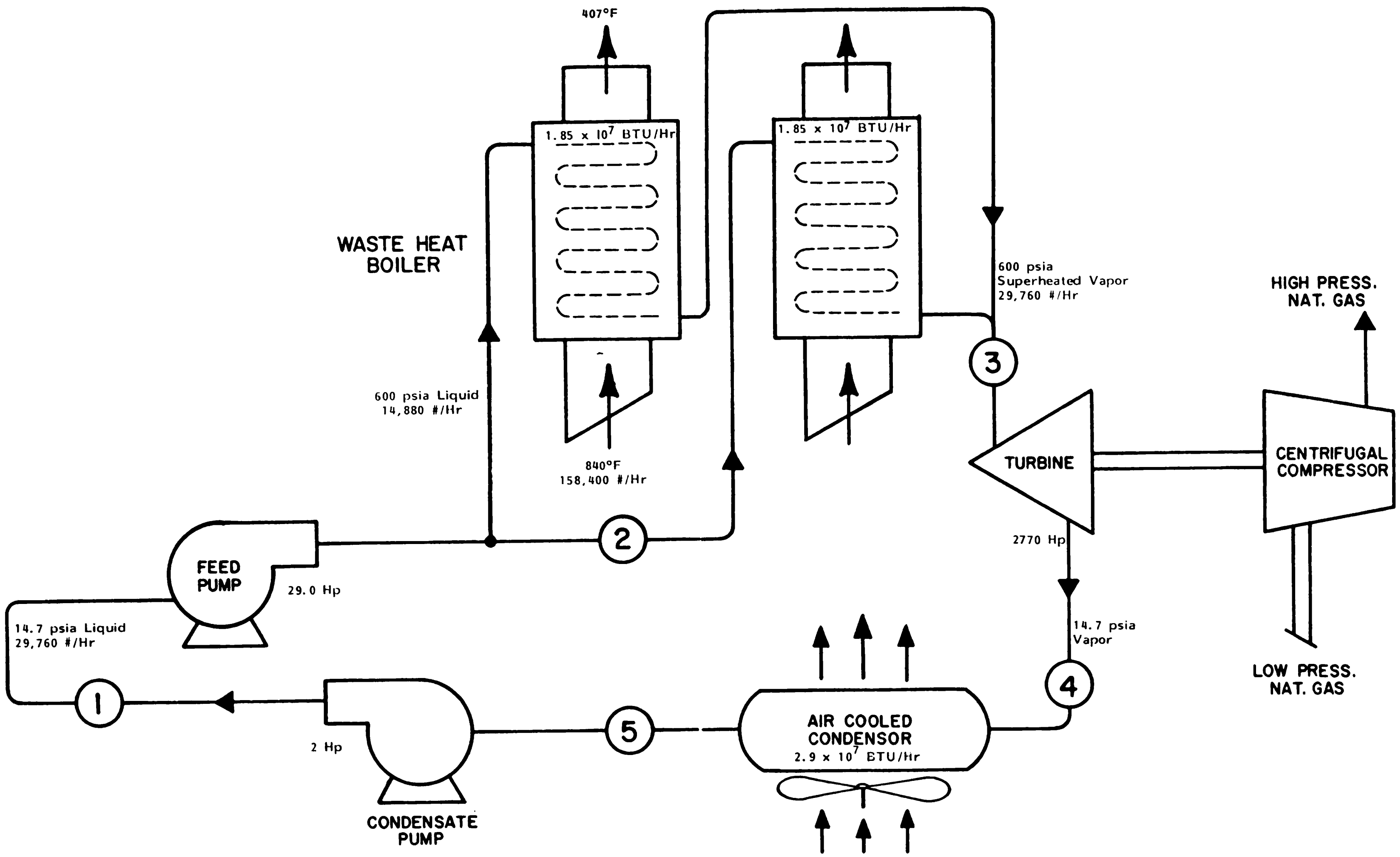


Figure 10.--Process Flow Diagram for 600 psia Steam Rankine Cycle.

600 psia super heated vapor. Stream 3 is expanded in a six-stage steam turbine (η = .70), producing 2770 Hp of shaft work. Stream 4 exits the turbine exhaust, superheated 255°F vapor at 14.7 psia and with no condensate present. Stream 4 is then condensed in a 2998 square foot air cooled condensor with four 15 Hp fan motors. The condensor heat load is 2.92×10^7 BTU/Hr. Finally, condensate from the supplied condensate drums is pumped from stream 5 to stream 1 through a 2 Hp condensate pump completing the cycle. It is assumed that the liquid will subcool 20°F between the condensate drums and the waste heat boilers.

Any shaft work produced by the cycle above can be utilized directly by attachment to a centrifugal natural gas compressor through the proper transfer gearbox. Since the centrifugal compressors usually operate in pairs and can be set up to operate in quads, the additional shaft work can be easily utilized to provide a real and immediate savings over additional compressor driver operation.

Equipment Description

Waste Heat Boilers

Two identical waste heat boilers are utilized in this design. Each is constructed of serrated fin tubes surrounded by a plenum to create a shell and tube heat exchanger (see Appendix I). Both serrated fin tubes and triangular tube pitch are used to maximize the limiting outside heat transfer coefficient. Cleaning is not considered to be a problem in this design since the waste heat exhaust gases are exceptionally clean. For diesel or conventionally fueled drivers,

exhaust soot could well be a significant problem and this could alter the design for such drivers.

A total of 1980 square feet of inside tube surface area is necessary in each waste heat boiler to collect the 18.27 million BTU/Hr of waste heat available for this design. Subcooled water at 192°F flowing at 14,880 lbs/Hr is circulated through the tubeside in counter flow to the hot exhaust gases. Superheated vapor at 800°F and 600 psia is produced with a tube side pressure drop of .54 psi (see Appendix I for calculation method).

On the shell side 158,400 lbs/hr of hot exhaust gases flow in counter flow through the 60 square foot (cross-sectional) shell, entering at 840°F and exiting at 407.4°F. A total shell side pressure drop of .26 psi is encountered.

The minimum approach temperature of 25°F occurs immediately prior to the boiling section of the heat exchanger. Design heat transfer coefficients of 75.5, 92.0 and 114.2 BTU/hr Ft^2 F° are found in the sensible heating, boiling and superheating regions of the heat exchanger, respectively.

Each waste heat boiler requires an adapter manifold to connect the 24" by 36" exhaust outlet to the 7.5 by 8 foot waste heat boiler. In addition, a mounting frame and 2 to 3 inches of insulation must be included. Finally, provision for safety valves and control sensors must be made for safety reasons.

Table 8 is a complete listing of all pertinent specifications for each boiler. Refer to the Economics Section for costing information.

TABLE 8.--Waste Heat Boiler Design Parameters for 600 psia Steam Rankine Cycle (GG-12 Gas Turbine Driver Case).

TUBESIDE:

- Fluid-Deionized Water
- Inlet Temperature - 192°F
- Outlet Temperature - 800°F
- Inlet Pressure - 600 psia
- Pressure Drop - .54 psi
- Flow Rate - 14,880 lbs/hr
- Tube Material - 12 BWG-Carbon Steel
- Tube OD - 1 Inch
- Tube Length - 8 Feet
- Tube Pitch - 1 Inch Triangular
- Fin Height - 3/8 Inch
- Fin Density - 96 Fins/Foot
- Fin Type - Aluminum Circular Fin Press Fit
- Tubes per Pass - 45
- Total Number of Passes - 25
- Dummy Tubes - 1 per Pass

HEAT DUTY:

- Sensible Heating Region - 4.60 MM BTU/Hr
- Boiling Region - 10.89 MM BTU/Hr
- Superheating Region - 3.04 MM BTU/Hr
- Total - 18.53 MM BTU/Hr

LOGMEAN TEMPERATURE DIFFERENCE:

- Sensible Heating Region - 82.9°F
- Boiling Region - 114.3°F
- Superheating Region - 128.2°F

SHELLSIDE:

- Fluid - Natural Gas Combustion Products
- Inlet Temperature - 840°F
- Outlet Temperature - 407°F
- Inlet Pressure - .36 psia
- Pressure Drop - .26 psi
- Flow Rate - 158,400 lbs/hr
- Shell Material - 12 Gauge Carbon Steel
- Shell Dimensions - (W,L,H) 7.5' x 8.0' x 3.6'
- Free Cross-Sectional Area - 23.7 Ft^2
- Insulation - 2" Rock Wool
- Baffles - None
- Number of Passes - 1

OVERALL HEAT TRANSFER COEFFICIENT:

- Sensible Heating Region - 75.5 BTU/Hr Ft^2 F°
- Boiling Region - 92.0 BTU/Hr Ft^2 F°
- Superheating Region - 114.2 BTU/Hr Ft^2 F°

HEAT TRANSFER AREA:

- Sensible Heating Region - 736 Ft^2
- Boiling Region - 1036 Ft^2
- Superheating Region - 208 Ft^2
- Total - 1980 Ft^2

TABLE 9.--Turbine Design Parameters for 600 psia Steam Rankine Cycle (GG-12 Turbine Driver Case).

Fluid - Deionized Water	
Turbine Hp Output - 2770 Hp	
Inlet Conditions:	Temperature - 800°F
	Pressure - 600 psia
Outlet Conditions:	Temperature - 255°F
	Pressure - 14.7 psia
	No Condensation
Flow Rate - 29,760 lbs/hr	
Enthalpy Change - 236.8 BTU/lb	
Thermal Efficiency - 70 percent	
Number of Stages - 6	
Type - Axial Flow-Horizontally Split Case (2)	

Turbine

The two high pressure steam lines from each waste heat boiler combine to form a total of 29,760 lbs/hr of superheated vapor at 600 psia. This material is expanded through the six stage axial turbine resulting in 2770 Hp of shaft work. A turbine efficiency of 70 percent was chosen considering the use of water as a working fluid.

The turbine shaft work is routed through a transfer gearbox to synchronize it with the shaft work produced in the driver reaction turbine. Turbine design is considered to be at a level equivalent to that proposed in reference (2). Table 10 describes the turbine parameters.

Pumps

Two pumps are necessary in this design. The first is a 60 GPM high pressure liquid pump for compressing liquid to 600 psia in

TABLE 10.--High Pressure Pump Design Parameters for 600 psia Steam Rankine Cycle (GG-12 Gas Turbine Driver Case).

Inlet Pressure - 14.7 psia
Outlet Pressure - 600 psia
Inlet Temperature 192°F
Flow Rate - 59.6 GPM
Efficiency - 70 percent
Hp Rating - 29 Hp
RPM - 3500
Fluid - Deionized Water
Fluid Density - 62.4 lbs/Ft^3
Type - Centrifugal - Single Shaft
Stages - 22
Model # - Gould #3934 (2)

the rankine cycle. A 29 Hp centrifugal pump and electric drive motor are needed here. A Gould model #3934 or equivalent could be used.

The second is simply a condensate pump to move the saturated liquid from the condenser drums back to the auxillary buildings. A 60 GPM 2 Hp pump and drive motor are needed here.

Condensor (Air Cooled)

An air cooled condenser was chosen as part of the Rankine cycle to eliminate the need for additional cooling water supply. The condenser for this design is a 2998 square foot (inside area) finned tube heat exchanger with the saturated vapor on the tube side and ambient air on the outside of the tubes. Four 15 Hp fans are needed to circulate air around the tubes. A total of 29.5 million BTU's of sensible and latent heat energy (combined) are transferred in this condenser. The condenser occupies an area of 24' by 24'. The condenser is designed to operate with a 100°F maximum dry bulb and 80°F maximum wet bulb temperature. Table 11 lists the condenser specifications.

TABLE 11.--Air Cooled Condenser Design Parameters for 600 psia Steam Rankine Cycle (GG-12 Gas Turbine Driver Case).

U Overall - 130 BTU/Hr. Ft^2 F° (based on outside tube surface area)

Q Overall - 29.5 x 10^6 BTU/Hr

ΔT_{lm} - 71.8°

Vapor Flow Rate - 29,760 #/Hr

Vapor Inlet Temperature - 255° Superheated Vapor

Fluid Outlet Temperature - 212°F Saturated Liquid

Operating Pressure - 14.7 psia

Design Pressure - 150 psia (Tube side)

Tubing - 1" O.D. 12 BWG Carbon Steel

Tube Pitch - 2-3/8" Triangular

Number of Tube Rows - 4

Fins - 5/8" High Circular Aluminum-Press Fit

Fin Density - 8 per Foot

Tube Length - 24 Feet

Face Area - 597 Sq. Ft.

Total Heat Transfer Area - 3008 Sq. Ft.

Fan Motors - 15 Hp Explosion Proof Electric (4 needed)

Air Face Velocity - 595 Ft./Min.

Air Inlet Temperature - 100°F

Air Outlet Temperature - 173°F

Air Flow Rate - 449 #/Sec.

Economic Analysis for GG-12 Gas Turbine Rankine Cycle

It is of extreme importance to realize that an alternatives approach was used for these economic evaluations. Natural gas savings are a large part of the profitability of any of these designs and therefore the computation of these savings is critical. In the alternatives approach, the natural gas savings is computed as the quantity of natural gas which would be required to power the driver motor in question to provide an amount of shaft work equal to that produced in the rankine cycle. This approach obviously makes the inefficient drive motors appear more favorable as energy recovery sources, which is correct. It is incorrect, however, to assume that total driver system economics would parallel the rankine cycle portion of the economics. An evaluation of a total system incorporating a drive motor and a rankine bottoming cycle is a totally separate problem. High drive motor efficiency and low net waste energy output would be primary objectives in an overall high efficiency driver system utilizing rankine bottoming cycles as supplementary energy sources (3). Such a system would need to be evaluated on a cost per horsepower produced basis, over a ten-year period (including capital costs, maintenance, and overall fuel costs) as opposed to a net savings basis as was used in this study. Given the opportunity to make a choice of drive motors one would obviously not choose one that is inefficient solely to provide waste energy for a rankine cycle to utilize.

All economics for this design are performed assuming a 3rd Quarter 1980 installation and a 1st Quarter 1981 start-up. The total fixed capital investment includes the cost of equipment, installation and a small auxilliary building housing the turbine and pump assemblies. Interest for any construction loan is not included. In addition, costs associated with start-up and debugging are not included.

Ten year straight line depreciation is used for all economic evaluations and zero salvage value is assumed. Labor is assumed to be available at $20,800/man-year with any additional supervisory labor already in existence. All dollars are based on 1980 valuations. Corporate income taxes are assumed to be at 48 percent of gross operating profit. No tax credits for energy recovery investments are included.

Electricity is assumed to be available at 8 cents per kilowatt-hour. Natural gas is assumed to increase in value as described in Figure 11. All remaining fixed costs are considered as factored estimates based on the total fixed capital investment. Maintenance is at a rate of 4 percent of the TFCI per year. A credit for maintenance savings realized by not needing to operate a driver to perform the shaft work obtained from the rankine cycle is also included. This credit is calculated as the number of Hp-Hrs work obtained, times total 1977 maintanence costs per Hp-Hr for the Consumers Power Company network. This method assumes an equal maintenance cost per Hp-Hr for reciprocating drivers and gas turbines which is probably incorrect but is nevertheless a good approximation.

Annual plant overhead and insurance are assumed to be at 2.6 percent and 1.5 percent of the TFCI, respectively. Finally, interest on working capital is assumed to be at a 12 percent per annum rate.

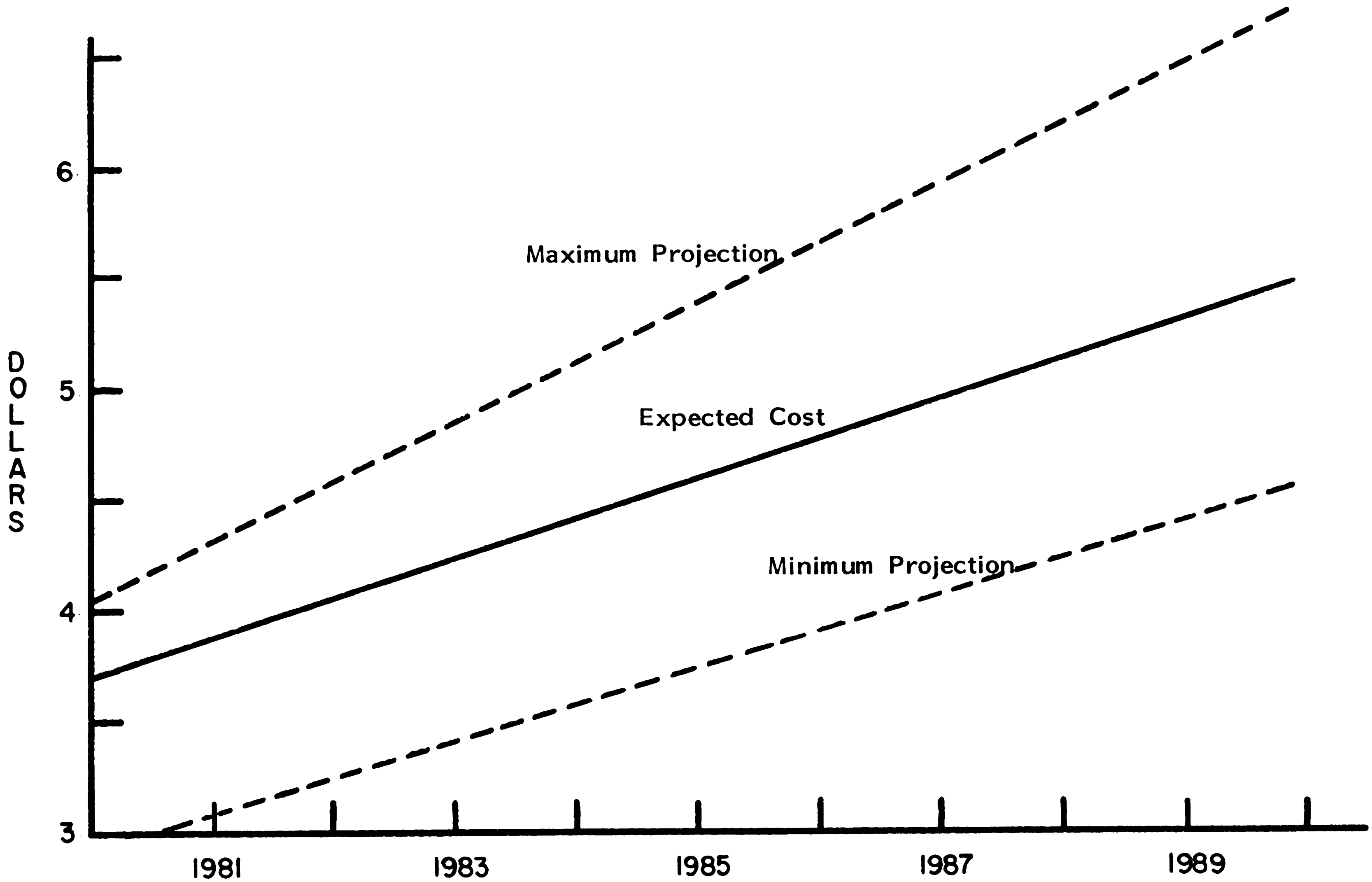


Figure 11.--Natural Gas Cost Escalation in Dollars per MMBTU.

The 1980 total cost for all equipment, instrumentation, electrical and piping for the rankine cycle described is $955,000 (1, 3, 10). This figure includes installation of all equipment but excludes the cost of any construction loans. A complete breakdown of individual equipment and installation costs is found in Table 12. In addition, a breakdown of operating costs associated with this design at 25 percent factor is presented in Table 13. A payout period (after taxes) of 4.1 years is expected for this design at 25 percent stream factor. This corresponds to an 18.1 percent discounted cash flow rate of return. Ten year cash flow summaries are presented in Table 14.

Parametric Studies

The basis for optimization in the steam rankine cycle design was the variation of the waste heat boiler approach temperature. Variation of this temperature will simultaneously change the size of the waste heat boiler and the quantity of steam produced. The relative cost difference of the waste heat boiler versus the value of steam produced initiates the optimization. Four approach temperatures were investigated for the design case resulting in four individual economic evaluations. Approach temperatures of 10, 15, 25 and 40 degrees fahrenheit were chosen.

Figure 9 (see Appendix I) is a chart describing Q (total quantity heat transferred) versus T for the PW GG-12 steam waste heat boiler, As can be seen on this chart the minimum approach temperature occurs at the point where boiling begins for the working fluid.

TABLE 12.---Individual Equipment Cost Breakdown for 600 # Steam Rankine Cycle (GG-12 Gas Turbine Driver Case).

Equipment Type	Equip. Cost $	Install. Cost $	Material Cost $	Indirect Cost $	Total Cost $
Turbine	122,000	36,000	33,000	73,000	264,000
Reduction Gearbox	12,000	3,500	3,500	7,000	26,000
Feed Pump	32,500	22,000	23,000	29,500	107,000
Condensate Pump	12,000	3,500	3,500	7,500	26,500
Waste Heat Boilers	103,000	30,500	30,000	62,000	225,500
Air Cooled Condensor	64,000	19,000	18,500	38,500	140,000
Aux. Building	16,500	7,000	--	25,000	48,500
SUB TOTALS	362,000	121,500	111,500	242,500	837,500

Total Delivered Equipment Cost	362,000
Total Controls, Electrical, Piping and Painting Cost (Material Costs)	111,500
Total Installation Labor Cost	121,500
Total Direct Cost	595,000
Total Indirect Cost (Engineering, Procurement, Supervision	242,500
Total Direct and Indirect Cost	837,500
Contractor's Fees	33,500
Contingency	84,000
Total Fixed Capital Investment (All 1980 Dollars)	$955,000

References 1, 2, 3, 10.

TABLE 13.--Operating Cost Breakdown for 600 # Steam Rankine Cycle
(GG-12 Gas Turbine Driver Case)

Total Fixed Capital Investment - $955,000

Stream Factor - 25 Percent

Utilities	Quantity of Units	1981 $/Unit	1981 $/Year
Electricity	163,300 KWH	.08	13,000
Natural Gas	96,458 MM BTU**	3.87	(373,300)

Labor	Quantity of Units	1980 $/Unit	1980 $/Year
Direct Labor	1 Man Year	20,800	20,800
Overheat and Fringe Benefits		20,800	20,800

Capital Related	1981 $/Year
Maintenance @ 4%year (includes credit*)	10,500
Plant Overhead @ 2.6%/year	24,800
Insurance @ 1.5%/year	14,300
Depreciation @ 10%/year	95,500
Interest on Working Capital @ 12%/year	2,600
Total Operating Costs (Excluding Depreciation and Natural Gas Credit)	106,800
Total Operating Profit (before Taxes)	171,000
Working Capital (Accounts Payable)	21,400

*Credit of $10.02 per Hp-capacity for original driver replacement.

**Calculated as fuel required to produce 2770 Hp in GG-12 driver @ 25% stream factor.

TABLE 14.--After 48% Tax Cash Flow Stream Factor Sensitivity for 600 psia Steam Rankine Cycle (GG-12 Gas Turbine Driver Case).

Stream Factor	15%	25%	40%	90%
1980	(955,000)	(955,000)	(955,000)	(955,000)
1981	109,900	184,400	296,200	668,900
1982	118,400	198,800	319,300	721,000
1983	124,600	209,000	335,700	758,000
1984	129,700	217,700	349,700	789,700
1985	134,900	226,400	363,700	821,300
1986	140,000	235,000	377,500	852,500
1987	145,300	243,800	391,600	884,300
1988	150,800	253,000	406,400	917,700
1989	156,000	261,700	420,400	949,400
1990	183,400	293,000	457,400	1,005,500
Discounted Cash Flow Rate of Return (after Taxes)	6.4	18.1	32.6	73.8
Payout Period (Years) (after Taxes)	6.9	4.1	2.6	1.1

For the 40° approach case the minimum occurs twice, once as stated above and once as the vapor is superheated to its maximum temperature of 800°F.

Table 15 summarizes the pertinent economic information for each of the four cases. In each of the cases a 25 percent stream factor was assumed. Labor was considered fixed at one man-year for all cases. Capital-related costs (i.e., maintenance, overhead, etc.) were allowed to fluctuate as a factored estimate of the total fixed capital investment. The major contributor to changes in the total fixed capital investment is the waste heat boiler cost.

TABLE 15.--Summary of Optimization Economics (Boiler Approach Temperature Parameter)

	Approach Temperature			
	10	15	25	40
Total Fixed Capital Investment (1000's)	1,047	1,011	955	894
Dollar Cost per Installed HP-Capacity	363	355	345	338
Discounted Cash Flow Rate of Return (%)	17.0	17.6	18.1	16.0
Payout Period (Years)	4.5	4.4	4.2	4.7
Natural Gas Savings per Year (MM BTU)	100,476	99,375	96,458	82,733

Stream Factor = 25%

As can be seen in Table 15, the cost in dollars per installed horsepower of capacity decreases as a weak function of approach temperature. In addition, a weak maximum in the discounted cash flow rate of return occurs at the 25° approach temperature. It is noteworthy that the maximum return on investment and minimum installed cost do not necessarily coincide for the steam rankine cycle. This is due to the difference in the relative values of shaft work produced and invested capital for this case and this is once more heavily dependent upon the alternatives approach being used. As the efficiency of the driver motor in question increases, this maximum rate of return should occur at higher approach temperatures. Subsequently, the more efficient reciprocating drivers discussed in later design cases should exhibit higher optimal approach temperatures. However, on the basis of this optimization 25° was chosen as the optimal approach temperature for all of the steam rankine cycle designs.

In addition to determining the optimal approach temperature the sensitivity of this design to the station stream factor was investigated. As the stream factor changes, only the variable costs change iwth it. Only two items were considered as variable costs in this design, electricity costs and natural gas credits. All other costs are either capital related fixed costs or labor costs which were considered fixed for purposes of this investigation.

Table 14 presents ten-year cash flow summary, plus the discounted cash flow rate of return and payout period at various stream

factors. Figures 12 and 13 present the discounted cash flow rate of return versus stream factor and the payout period versus stream factor, respectively. As is evident from these figures, stream factor becomes a critical parameter economically as it approaches 40 percent and strongly detracts from the economics at values less than 40 percent. Unfortunately, many natural gas compressor stations in the Consumers Power network operate at less than 25 percent stream factor if no load shifting is specified.

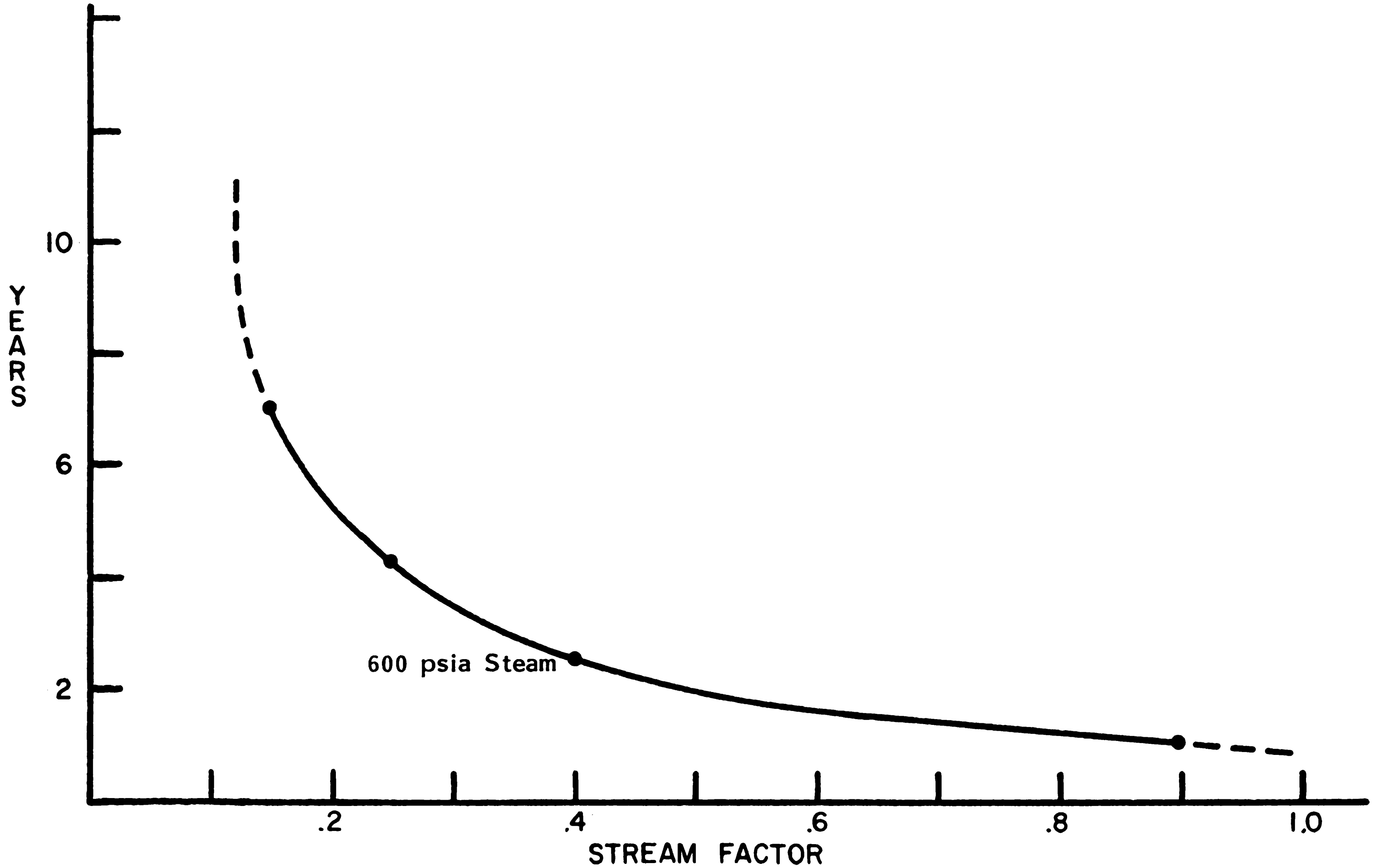


Figure 12.--Payout Period versus Stream Factor for 600 psia Steam Rankine Cycle (GG-12 Gas Turbine Driver Case).

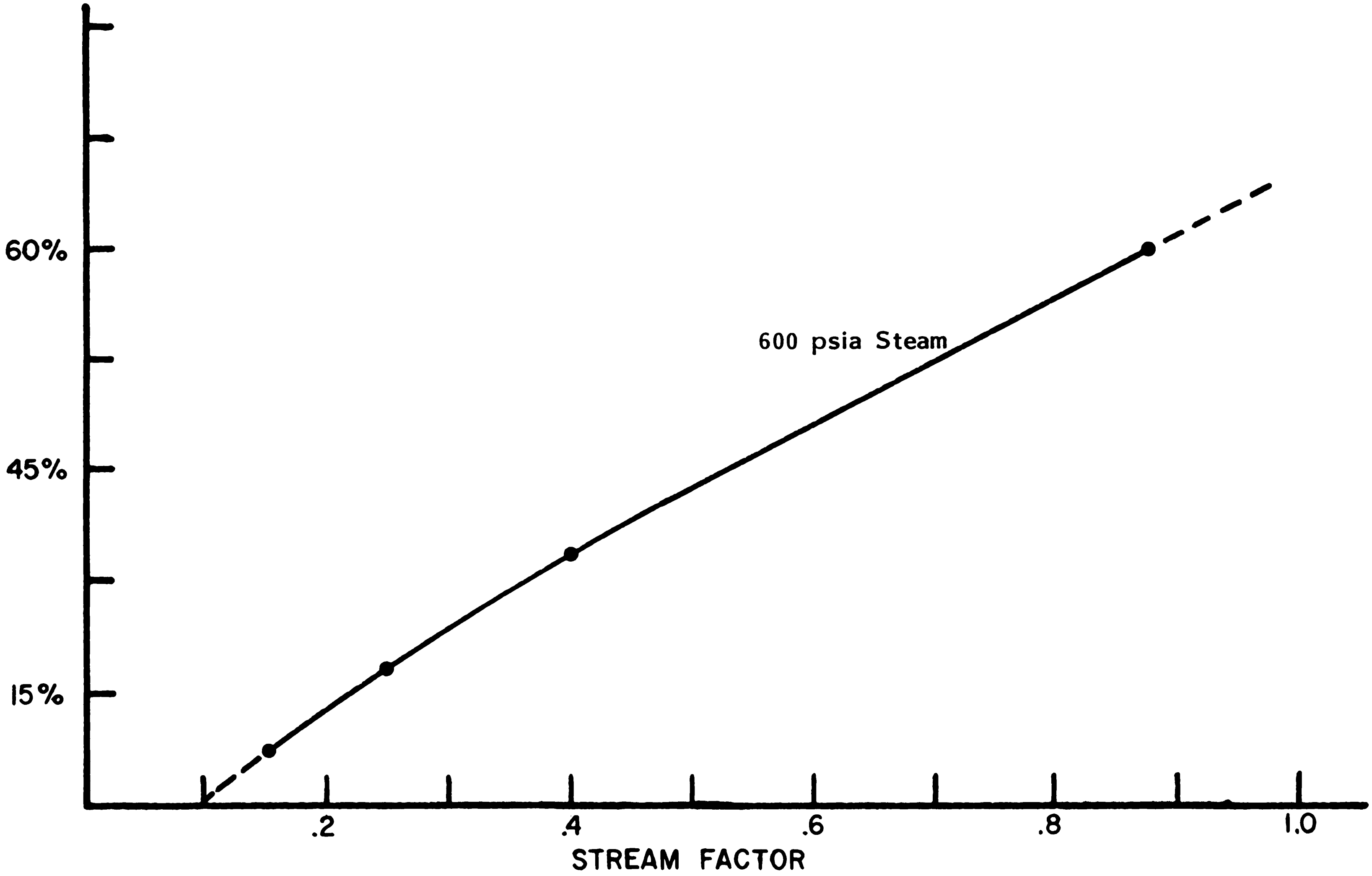


Figure 13.--After Taxes Discounted Cash Flow Rate of Return versus Stream Factor for 600 psia Steam Rankine Cycle (GG-12 Gas Turbine Driver Case).

APPENDIX III

STEAM RANKINE CYCLE CONCEPT APPLIED TO

TWO ADDITIONAL DRIVER TYPES

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STEAM RANKINE CYCLE CONCEPT APPLIED TO TWO ADDITIONAL DRIVER TYPES

Introduction

The following Appendix contains the application of the steam rankine cycle concept to two additional driver types, the Cooper Bessmer W-330 and the Enterprise HV16C. Both are two cycle reciprocating drivers unlike the previously discussed Pratt-Whitney GG-12 gas turbines (see Appendix II). However, though the drivers are not the same type as that discussed in Appendix II, the application of basic design concepts remains the same. The waste heat boiler only requires a stream of hot exhaust gases to power the rankine cycle and each of these cases fulfills this requirement. Post waste heat boiler design, equipment and concepts are the same for all of the cases.

With this similarity in mind, a briefer, concise description of the two additional design cases is possible. It is assumed that the reader has read and understands the basic concepts of the steam rankine cycle design (see Appendix II). A brief summary is provided for each driver case. In addition, all of the necessary tables and figures have also been provided. Each table has an analogous table in Appendix II for comparison purposes.

Summary of Cooper Bessmer W-330 Steam Rankine Cycle

A simple four stage rankine cycle utilizing water as the working fluid was chosen for the Cooper-Bessmer W-330 reciprocating drivers. The cycle (described in Figure 14) requires a maximum cycle temperature of 710°F and a maximum pressure of 400 psia. A steam flow rate of 8524 lbs/hour is attainable when waste heat is used from a pair of the Cooper-Bessmer drivers. The horsepower rating of such a rankine cycle is 690 Hp resulting in an 8.8 percent increase in the horsepower output of the 3900 Hp individual drivers. The relatively small increase in horsepower output is due to the high efficiency of the Cooper-Bessmer drivers coupled with their smaller output of medium temperature exhaust gases. Each driver exhausts 53,950 lbs/hr of clean 750°F exhaust gases and operates at 37 percent thermal efficiency.

Table 16 presents the thermodynamic conditions for the individual streams described in Figure 14. Tables 17 through 20 present the individual design parameters for necessary equipment.

The total fixed capital investment necessary for the design case is $425,500. A payout period of 6.4 years is estimated for a 90 percent stream factor increasing to 13 years at approximately 60 percent stream factor. These payout periods represent discounted cash flow rates of return of 8.0 and -3.4 percent, respectively. Table 23 presents ten year cash flow summary as a function of stream factor for a more comprehensive description.

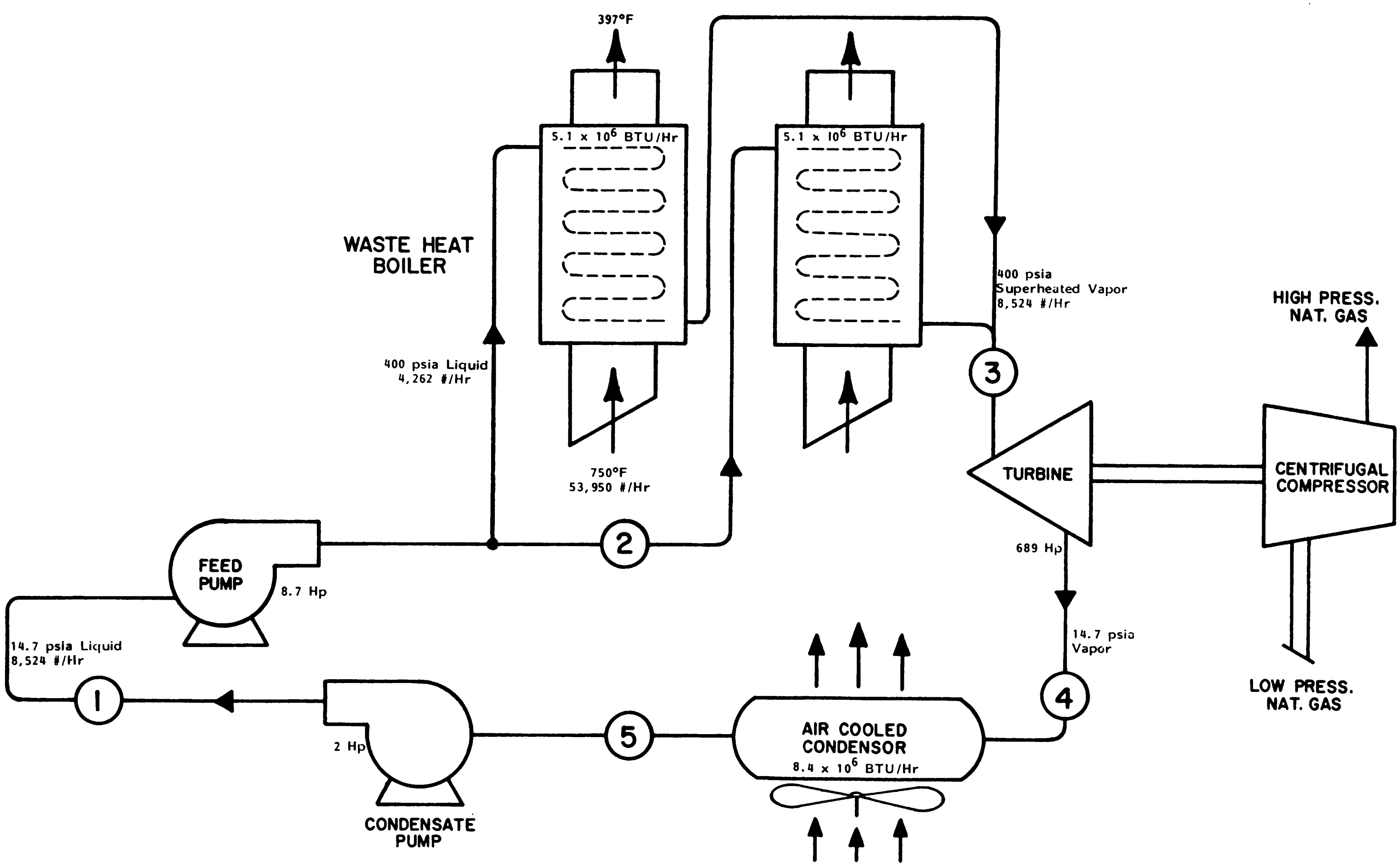

Figure 14.--Process Flow Diagram for 400 psia Steam Rankine Cycle.

TABLE 16.--Thermodynamic Conditions for 400 psia Steam Rankine Cycle (W-330 Driver Case).

Stream Number	Pressure (psia)	Temperature (F°)	Enthalpy (BTU/#)	Entropy (BTU/#-F°)
1	14.7	192	160.17	
2	400	192	160.8	
3	400	710	1368.8	1.645
4	14.7	240	1163.1	1.775
5	14.7	212	180.17	.3121

An annual natural gas savings of $138,600 is realized at 90 percent stream factor and 1981 gas prices. Table 21 presents the individual equipment costs, installation costs, etc., for the design case. Finally, Table 22 is a breakdown of the yearly operating costs at 25 percent stream factor.

Summary of Enterprise HV16C Steam Rankine Cycle

A simple four stage rankine cycle utilizing water as the working fluid was chosen for the Enterprise HV16C reciprocating drivers. The cycle (described in Figure 15) requires a maximum cycle temperature of 960°F and a maximum pressure of 1200 psia. A steam flow rate of 14,810 lbs/hr is attainable when waste heat is used from a pair of the Enterprise drivers. The horsepower rating of such a rankine cycle is 1715 Hp resulting in a 19.9 percent increase in the horsepower output of the 4320 Hp individual drivers. The relatively

TABLE 17.--Waste Heat Boiler Design Parameters for 400 psia Steam Rankine Cycle (W-330 Driver Case).

TUBESIDE:

- Fluid - Deionized Water
- Inlet Temperature - 192°F
- Outlet Temperature - 710°F
- Inlet Pressure - 400 psia
- Pressure Drop - .48 psi
- Flow Rate - 4262 lbs/hr
- Tube Material - 14 BWG-Carbon Steel
- Tube O.D. - 1 Inch
- Tube Length - 8 Feet
- Tube Pitch - 2 Inches Triangular
- Fin Height - 3/8 Inch
- Fin Density - 96 Fins per Foot
- Fin Type - Aluminum Serrated Circular Fin-Press Fit
- Tubes per Pass - 14
- Total Number of Passes - 23
- Dummy Tubes - 1 per Pass

HEAT DUTY:

- Sensible Heating Region - 1.12 MM BTU/Hr
- Boiling Region - 3.23 MM BTU/Hr
- Superheating Region - .69 MM BTU/Hr
- Total - 5.14 MM BTU/Hr

LOGMEAN TEMPERATURE DIFFERENCE:

- Sensible Heating Region - 80.2°F
- Boiling Region - 108.4°F
- Superheating Region - 124.9°F

SHELLSIDE:

- Fluid - Natural Gas Combustion Products
- Inlet Temperature - 750°F
- Outlet Temperature - 396.7°F
- Inlet Pressure - .36 psig
- Pressure Drop - .26 psi
- Flow Rate - 53,950 lbs/hr
- Shell Material - 12 Gauge Carbon Steel
- Shell Dimensions (W,H,L) - 2.33' x 3.23' x 8'
- Free Cross-Sectional Area - 7.37 Ft^2
- Insulation - 2" Rock Wool
- Baffles - None
- Number of Passes - 1

OVERALL HEAT TRANSFER COEFFICIENT:

- Sensible Heating Region - 75.4 BTU/Hr-Ft^2-F°
- Boiling Region - 94.1 BTU/Hr-Ft^2-F°
- Superheating Region - 113.9 BTU/Hr-Ft^2-F°

HEAT TRANSFER AREA:

- Sensible Heating Region - 186 Ft^2
- Boiling Region - 326 Ft^2
- Superheating Region - 49 Ft^2
- Total - 561 Ft^2

TABLE 18.--Turbine Design Parameters for 400 psia Steam Rankine Cycle (W-330 Driver Case).

Fluid - Deionized Water
Turbine Hp Output - 689
Inlet Conditions:
    Temperature - 710°F
    Pressure - 400 psia
Outlet Conditions:
    Temperature - 240°F
    Pressure - 14.7 psia
    No Condensate Present
Flow Rate - 8524 #/Hr
Enthalpy Change - 205.7 BTU/#
Thermal Efficiency - 70 Percent
Number of Stages - 6
Type - Axial Flow - Horizontally split case
RPM - 7950

Reference 2

TABLE 19.--High Pressure Pump Design Parameters for 400 psia Steam Rankine Cycle (W-330 Driver Case).

Inlet Pressure - 14.7 psia
Outlet Pressure - 400 psia
Inlet Temperature - 192°F
Flow Rate - 17 GPM
Efficiency - 45 Percent
Hp Rating - 8.7 Hp
RPM - 3500
Model # - Gould # 3934
Stages - 22
Fluid - Deionized Water
Fluid Density - 62.4 lbs/Ft^3
Type - Centrifugal - Single Shaft

Reference 2

TABLE 20.--Air Cooled Condenser Design Parameters for 400 psia Steam Rankine Cycle (W-330 Driver Case).

U Overall - 130 BTU/Hr Ft^2 F° (Condensing Steam)

Q Overall - 8.379 x 10^6 BTU/Hr

ΔT_{lm} - 70.7°F

Vapor Flow Rate - 8524 #/Hr

Vapor Inlet Temperature - 240°F Superheated Vapor

Fluid Outlet Temperature - 212°F Saturated Liquid

Operating Pressure - 14.7 psia

Design Pressure - 150 psia

Tubing - 1 Inch O.D. 12 BWG Carbon Steel

Tube Pitch - 2-3/8 Inch Triangular

Number of Tube Rows - 4

Fins - 5/8 Inch Circular Aluminum-Press Fit

Fin Density - 8 per Foot

Tube Length - 20 Feet

Face Area - 181 Ft^2

Total Heat Transfer Area - 912 Ft^2

Fan Motor - 11 Hp Explosion Proof (two required)

Air Face Velocity - 595 Ft/Min.

Air Inlet Temperature - 100°F

Air Outlet Temperature - 171°F

Air Flow Rate - 132 #/Sec.

TABLE 21.--Individual Equipment Cost Breakdown for 400 psia Steam Rankine Cycle (W-330 Driver Case).

Equipment Type	Equip. Cost $	Install. Cost $	Material Cost $	Indirect Cost $	Total Cost $
Turbine	66,000	20,000	18,000	40,500	144,500
Reduction Gearbox	7,500	2,500	2,000	4,500	16,500
Feed Pump	7,000	4,500	5,000	6,500	23,000
Condensate Pump	7,000	2,500	2,000	4,500	16,000
Waste Heat Boilers	32,000	9,500	9,500	19,500	70,500
Air Cooled Condenser	26,000	7,000	7,000	14,000	54,000
Aux. Building	16,500	7,000	--	25,000	48,500

Total Delivered Equipment Cost	162,000
Total Controls, Electrical, Piping and Painting Cost	43,500
Total Installation Labor Cost	53,000
Total Direct Cost	258,500
Total Indirect Cost	114,500
Total Direct and Indirect Cost	373,000
Contractors' Fees	15,000
Contingency	37,500
Total Fixed Capital Investment (All 1980 Dollars)	425,500

References 1, 2, 3, 10

TABLE 22.--Operating Cost Breakdown for 400 psia Steam Rankine Cycle (W-330 Driver Case).

Total Fixed Capital Investment - $425,500

Stream Factor - 25 Percent

Utilities	Quantity of Units	1981 $/Unit	1981 $/Year
Electricity	45,081 KWH	.08	3,600
Natural Gas	9,944 MMBTU**	3.87	(38,500)
Labor			
Labor	1 Man-Year	20,800	20,800
Overhead and Fringe Benefits		20,800	20,800

Capital Related	
Maintenance @ 4%/year (includes credit)*	10,500
Plant Overhead @ 2.6%/year	11,000
Insurance @ 1.5%/year	6,400
Depreciation @ 10%/year	42,500
Interest on Working Capital @ 12%/year	1,600
Total Operating Costs (Excluding Depreciation and Gas Credits)	74,700
Total Operating Loss (includes Depreciation)	(78,700)
Working Capital (Accounts Payable)	14,900

*Credit of $10.02 per Hp capacity for original driver replacement.

**Calculated as fuel required to produce 689 Hp @ 25% stream factor in Cooper-Bessmer W-330 driver.

TABLE 23.--After 48% Tax Cash Flow Projections for 400 psia Steam Rankine Cycle (W-330 Driver Case).

Stream Factor	25%	60%	75%	90%
1980	(425,500)	(425,500)	(425,500)	(425,500)
1981	(36,200)	12,600	33,500	48,700
1982	(34,400)	17,100	39,200	52,300
1983	(32,400)	21,900	43,900	56,000
1984	(30,500)	26,400	46,900	59,500
1985	(28,600)	31,000	49,800	63,000
1986	(26,700)	35,500	52,800	66,600
1987	(24,800)	40,000	55,700	70,100
1988	(22,800)	43,700	58,800	73,900
1989	(20,900)	46,000	61,800	77,400
1990	(6,000)	61,600	77,800	94,000
Discounted Cash Flow Rate or Return	0	0	3.4%	8.0%
Payout Period (Years)	N/A	12.7	8.2	6.4

large increase in horsepower output is due primarily to the large output of relatively high temperature exhaust gases. Each driver exhausts 60,880 lbs. per hour of clean 1000°F exhaust gases and operates at 29 percent thermal efficiency. Table 24 presents the thermodynamic conditions for the individual streams described in Figure 15. Tables 25 through 28 present the individual design parameters for necessary equipment.

The total fixed capital investment necessary for the design case is $885,000. A payout period of 2.9 years is estimated at 90 percent stream factor increasing to 13 years at 25 percent stream factor. These payout periods represent discounted cash flow rates of return (after taxes) of 29.4 and -3.0 percent, respectively. Table 31 presents the ten year cash flow summary as a function of stream factor for a more comprehensive description.

An annual natural gas savings of $525,200 is realized at 90 percent stream factor and 1981 gas prices. Table 26 presents the individual equipment costs, installation costs, etc., for the design case. Finally, Table 30 is a breakdown of the yearly operating costs at 25 percent stream factor.

TABLE 24.--Thermodynamic Conditions for 1200 psia Steam Rankine Cycle (HV16C Driver Case).

Stream Number	Pressure (psia)	Temperature (F°)	Enthalpy (BTU/#)	Entropy (BTU/#-F°)
1	14.7	192	160.17	--
2	1200	192	165.19	--
3	1200	960	1476.0	1.6155
4	14.7	275	1181.3	1.8000
5	14.7	212	180.17	.3121

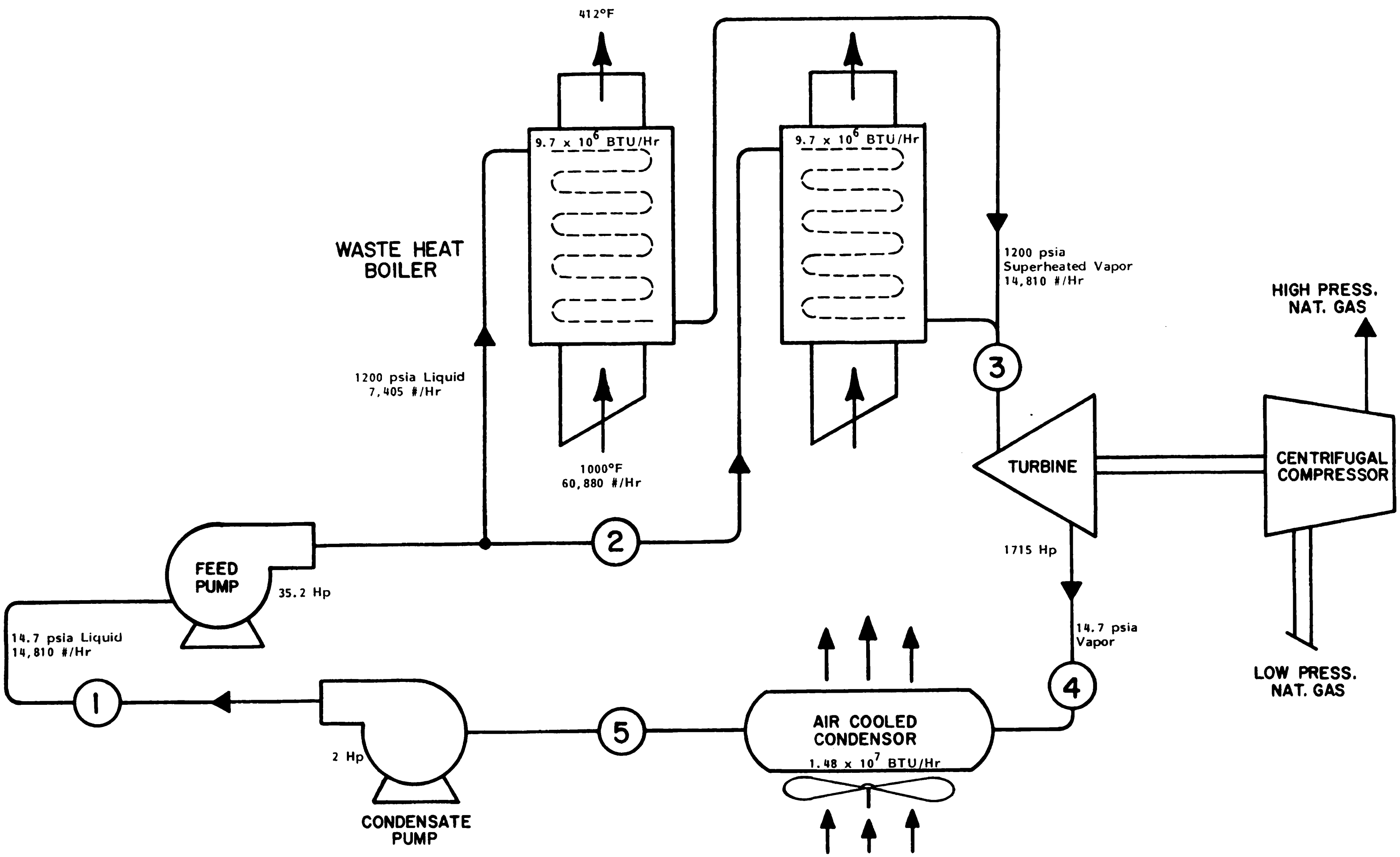

Figure 15.--Process Flow Diagram for 1200 psia Steam Rankine Cycle (HV16C Driver Case).

TABLE 25.--Waste Heat Boiler Design Parameters for 1200 psia Steam Rankine Cycle (HV16C Driver Case).

TUBESIDE:

- Fluid - Deionized Water
- Inlet Temperature - 192°F
- Outlet Temperature - 953.2°F
- Inlet Pressure - 1200 psia
- Pressure Drop - .48 psi
- Flow Rate - 7405 Lbs/Hr
- Tube Material - 12 BWG-Carbon Steel
- Tube O.D. - 1 Inch
- Tube Length - 8 Feet
- Tube Pitch - 2 Inch Trinagular
- Fin Height - 3/8 Inch
- Fin Density - 96 Fins per Foot
- Fin Type - Aluminum Serrated Circular Fin-Press Fit
- Tubes per Pass - 20
- Total Number of Passes - 30
- Dummy Tubes - 1 per Pass

SHELLSIDE:

- Fluid - Natural Gas Combustion Products
- Inlet Temperature - 1000°F
- Outlet Temperature - 412.1°F
- Inlet Pressure - .36 psig
- Pressure Drop - .26 psi
- Flow Rate - 60,880 Lbs/Hr
- Shell Material - 12 Gauge Carbon Steel
- Shell Dimensions - 3.33' x 4.24' x 8.0
- Free Cross-Sectional Area - 10.53 Ft^2
- Insulation - 2" Rock Wool
- Baffles - None
- Number of Passes - 1

HEAT DUTY:

- Sensible Heating Region - 3.01 MM BTU/Hr
- Boiling Region - 4.54 MM BTU/Hr
- Superheating Region - 2.16 MM BTU/HR
- Total - 9.71 MM BTU/Hr

LOGMEAN TEMPERATURE DIFFERENCE:

- Sensible Heating Region - 84.2°F
- Boiling Region - 111.6°F
- Superheating Region - 136.3°F

OVERALL HEAT TRANSFER COEFFICIENT:

- Sensible Heating Region - 75.0 BTU/Hr-Ft^2-F°
- Boiling Region - 94.2 BTU/Hr-Ft^2-F^2
- Superheating Region - 117.5 BTU/Hr-F2-F°

HEAT TRANSFER AREA:

- Sensible Heating Region - 477 Ft^2
- Boiling Region - 432 Ft^2
- Superheating Region - 135 Ft^2
- Total - 1044 Ft^2

TABLE 26.--Turbine Design Parameters for 1200 psia Steam Rankine Cycle (HV16C Driver Case).

Fluid - Deionized Water
Turbine Horsepower Output - 1715
Inlet Conditions:
- Temperature - 960°F
- Pressure - 1200 psia

Outlet Conditions:
- Temperature - 275°F
- Pressure - 14.7 psia
- No Condensate Present

Flow Rate - 14,810 #/Hr
Enthalpy Change - 294.7 BTU/#
Thermal Efficiency - 70 percent
Number of Stages - 6
Type - Axial Flow - Horizontally Split Case
RPM - 7950

Reference 2

TABLE 27.--High Pressure Pump Design Parameters for 1200 psia Steam Rankine Cycle (HV16C Driver Case).

Inlet Pressure - 14.7 psia
Outlet Pressure - 1200 psia
Inlet Temperature - 192°F
Flow Rate - 30 GPM
Efficiency - 60 Percent
Hp Rating - 35.2 Hp
RPM - 3500
Model # - Gould # 3934
Stages - 22
Type - Centrifugal - Single Shaft
Fluid - Deionized Water
Fluid Density - 62.4 lbs/Ft^3

Reference 2

TABLE 28.--Air Cooled Condenser Design Parameters for 1200 psia Steam Rankine Cycle (HV16C Driver Case).

U Overall - 130 BTU/Hr-Ft^2-F° (Condensing Steam)

Q Overall - 1.483 x 10^7 BTU/Hr

ΔT_{lm} - 70.7 F°

Vapor Flow Rate - 14,810 Lbs/Hr

Vapor Inlet Temperature - 275°F

Fluid Outlet Temperature - 212°F

Operating Pressure - 14.7 psia

Design Pressure - 150 psia

Tubing - 1 Inch O.D. 12 BWG Carbon Steel

Tube Pitch - 2-3/8 Inch Triangular

Number of Tube Rows - 4

Fins - 5/8 Inch Circular Aluminum-Press Fit

Fin Density - 8 per Foot

Tube Length - 24 Feet

Face Area - 320.5 Ft^2

Total Heat Transfer Area - 1615 Ft^2

Fan Motors - 16 Hp (two required)

Air Face Velocity - 595 Ft/Min

Air Inlet Temperature - 100°F

Air Outlet Temperature - 172°F

Air Flow Rate - 229 #/Sec

TABLE 29.--Individual Equipment Cost Breakdown for 1200 psia Steam Rankine Cycle (HV16C Driver Case).

Equipment Type	Equip. Cost $	Install. Cost $	Material Cost $	Indirect Cost $	Total Cost $
Turbine	100,500	29,500	27,000	59,500	216,500
Reduction Gearbox	10,000	3,000	2,500	6,000	21,500
Feed Pump	59,000	40,000	41,000	53,500	193,500
Condensate Pump	10,500	3,000	3,000	6,000	22,500
Waste Heat Boiler	85,000	25,000	24,500	52,000	186,500
Air Cooled Condenser	39,500	11,500	11,500	24,000	86,500
Aux. Building	16,500	7,000	--	25,000	48,500
SUB TOTALS	321,000	119,000	109,500	226,000	775,500

Total Delivered Equipment Cost	321,000
Total Controls, Electrical, Piping, Painting Cost (Material)	109,500
Total Installation Labor Cost	119,000
Total Direct Cost	549,500
Total Indirect Cost (Engineering)	226,000
Total Direct and Indirect Cost	775,500
Contractors Fees (@ 4% TD + I)	31,500
Contingency (@ 10% TD + I)	78,000
Total Fixed Capital Investment (All 1980 Dollars)	885,000

References 1, 2, 3, 10.

TABLE 30.--Operating Cost Breakdown for 1200 psia Steam Rankine Cycle (HV16C Driver Case).

Total Fixed Capital Investment - $885,000

Stream Factor - 25 Percent

Utilities	Quantity of Units	1981 $/Unit	1981 $/Year
Electricity	102,067 KWH	.08	8,200
Natural Gas	37,909 MMBTU**	3.87	(145,900)
Labor			
Labor	1 Man Year	20,800	20,800
Overhead and Fringe Benefits		20,800	20,800

Capital Related	
Maintenance @ 4%/year (includes credit)*	18,800
Plant Overhead @ 2.6%/year	23,000
Insurance @ 1.5%/year	13,300
Depreciation @ 10%/year	88,500
Interest on Working Capital @ 12%/year	2,400
Total Cost (Excluding Depreciation and Gas Credit)	107,300
Total Operating Loss (includes Depreciation)	(49,900)
Working Capital (Accounts Payable)	21,460

*Credit of $10.02 per Hp-capacity for replacement of original driver.

**Calculated as fuel required to produce 1715 HP @ 25% stream factor in Enterprise HV16C driver.

TABLE 31.--After 48% Tax Cash Flow Projection for 1200 psia Steam Rankine Cycle (HV16C Driver Case).

Stream Factor	25%	40%	60%	90%
1980	(885,000)	(885,000)	(885,000)	(885,000)
1981	39,400	102,000	159,600	246,000
1982	46,600	108,000	168,800	259,500
1983	54,200	114,300	178,000	273,700
1984	61,400	120,200	187,000	287,200
1985	68,600	126,200	196,000	300,700
1986	75,800	132,100	205,000	314,100
1987	83,000	138,100	214,000	327,600
1988	89,600	144,500	223,500	341,800
1989	93,300	150,500	232,500	355,300
1990	118,600	178,000	262,900	390,300
Discounted Cash Flow Rate or Return	-3.0%	7.2%	17.0%	29.4%
Payout Period (years)	12.1	6.7	4.3	2.9

APPENDIX IV

ORGANIC RANKINE CYCLE FOR PRATT-WHITNEY GG-12 GAS TURBINE DRIVERS

APPENDIX IV

ORGANIC RANKINE CYCLE FOR PRATT-WHITNEY GG-12 GAS TURBINE DRIVERS

Summary

A simple four stage rankine cycle utilizing Flourinol-85 as the working fluid was chosen for the Pratt-Whitney GG-12 gas turbine drivers. An organic working fluid was utilized in place of the conventional steam cycle to investigate the economic and efficiency advantages of this material. The lower condensing temperature chosen for this design coupled with the low heat of vaporization for the organic working fluid provide a higher overall thermal efficiency. The cycle, described in Figure 16, requires a maximum cycle temperature of 625°F and a maximum pressure of 700 psia (2). An organic flow rate of 165,140 lbs per hour is attainable when waste heat is used from a pair of the Pratt-Whitney drivers.

The horsepower rating of such a cycle is 4672 Hp resulting in an 85 percent increase in the base horsepower output of the two 2750 Hp individual drivers. The large increase in horsepower output is due to both the relatively large output of hot exhaust gases coupled with the high efficiency of the rankine cycle. Each driver exhausts 158,400 lbs per hour of clean 840°F hot exhaust gases and operates at 16 percent thermal efficiency. The cycle itself has a

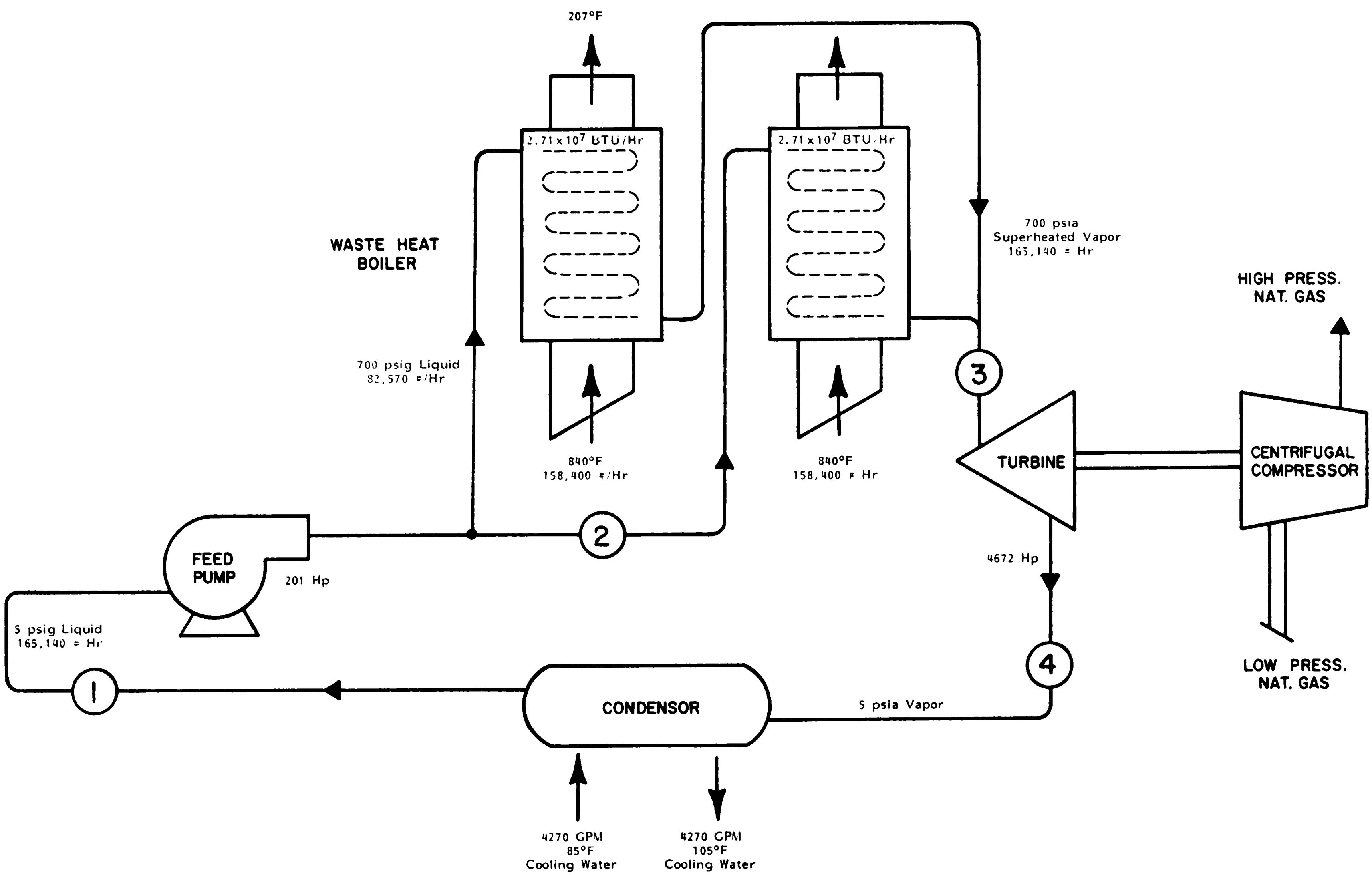

Figure 16.--Process Flow Diagram for Organic Rankine Cycle (GG-12 Driver Case).

120°F condensing temperature and operates at 21.3 percent overall thermal efficiency.

The 1980 total fixed capital investment necessary for the design case is $1,568,000. A payout period of 4.0 years is estimated at 25% stream factor. This payout period represents a 19.0 percent after taxes discounted cash flow rate of return. The annual savings in natural gas usage is 162,745 million BTU or $629,800 at 1981 gas prices and 25% stream factor. This credit increases substantially to $2,267,000 at 90 percent stream factor. Finally, at a stream factor of 90 percent, a payout period of approximately 1.1 years is expected.

Design Basis

The basis for this design is the simultaneous operation of a pair of Pratt-Whitney GG-12 gas turbine drivers rated at 2750 Hp each. Each driver operates at 100 percent load capacity when in operation, however, stream factor is considered to be variable. Each driver produces 158,400 lbs/hr of hot exhaust gases which are considered to be primarily combustion products of natural gas. The exhaust outlet temperature of the driver is 840°F maximum. Each driver can accept a waste heat boiler attached as a portion of its exhaust system with no greater than 10 inches (approximately 0.36 psi) pressure drop on the shell side.

Each compressor station is assumed to be remotely located, but with all necessary electrical power supply readily available. In addition, the sites are assumed to be in a climate with a maximum air

temperature of 100°F (dry bulb) and a maximum 80°F wet bulb. A minimal additional cooling water capacity is assumed available at any site. The additional property needed to facilitate construction is assumed to be currently available. Finally, additional Cooper-Bessmer centrifugal compressors and transfer cases are assumed to be available for shaft work utilization.

For a detailed description of economic assumptions see Economic Analysis section.

Process Description

Figure 16 is a schematic diagram of the four stage organic rankine cycle used in this design. Table 32 contains all process stream thermodynamic conditions for this cycle.

Beginning with stream 1, 165,140 lbs per hour of saturated liquid condensate at 120°F and 5 psia are compressed to 700 psia and 120°F in a 435 GPM, 201 Hp high pressure liquid pump. The resultant stream 2 is then split into two streams of 82,570 lbs per hour each and flow through the two waste heat boilers in counter flow. Each waste heat boiler has a tube side surface area of 3050 square feet. The shell side of each waste heat boiler is fed with 158,400 lbs/hr of hot exhaust gases which enter the boiler at 840°F and exit at 207°F. The boiler heat loads are 27.1 million BTU per hour each. The resultant streams are joined to form stream 3 which is 700 psia 625°F superheated Flourinol 85 vapor. Stream 3 is expanded in a six stage axial flow turbine (η = .80) producing 4672 Hp of shaft work. Stream 4 exists the turbine exhaust, superheated 335°F vapor at

TABLE 32.--Thermodynamic Conditions for Organic Rankine Cycle.

Stream Number	Pressure (psia)	Temperature (F°)	Enthalpy (BTU/#)	Entropy (BTU/#-F°)
1	5	120	38.0	.069
2	700	120	41.1	--
3	700	625	368	.475
4	5	336	296	.499

5 psia with no condensate present. Stream 4 is then condensed in a 3343 square foot water cooled condenser with a heat load of 42.6 MMBTU per hour. Finally, condensate from the supplied condensate drums is fed back to stream 1.

Cooling water for the water cooled condenser is supplied from forced draft cooling tower at a rate of 4270 GPM. Cooling water enters the condenser at 85°F maximum and returns to the cooling tower at 105°F maximum.

The shaft work produced by the above cycle can be utilized directly and immediately through attachment to a centrifugal natural gas compresser. This attachment must include a proper transfer gearbox and control equipment. Since the centrifugal compressers usually operate in pairs and can be set up to operate in quads, the large quantity of shaft work obtained can be used to provide a real and immediate savings over additional gas powered compresser driver operation. This large output of shaft work can in fact nearly double the output of a pair of the Pratt-Whitney GG-12 drivers.

Equipment Description

Waste Heat Boilers

Two identical waste heat boilers are utilized in this design. Each is constructed of serrated fin tubes surrounded by a plenum to create a shell and tube heat exchanger (see Appendix I). Both serrated fin tubes and triangular tube pitch are used to maximize the limiting outside heat transfer coefficient. Cleaning is not considered to be a problem in this design since the waste heat exhaust gases are exceptionally clean. For diesel or conventionally fueled drivers, exhaust soot could well be a significant problem and this could alter the design for such drivers.

A total of 3050 square feet of tube side surface area is necessary in each waste heat boiler to transfer the 27.1 MMBTU per hour of waste heat available for this design. Subcooled Flourinol 85 at 120°F flowing at 82,570 lbs/hr is circulated through the tube side in counter flow to the hot exhaust gases. Superheated vapor at 700 psia and 625°F is produced with a tubeside pressure drop of .35 psi (see Appendix I for calculation method).

On the shell side 158,400 lbs per hour of hot exhaust gases flow in counter flow through the 73.6 square foot (cross-sectional) shell, entering at 840°F and exiting at 207°F. A total shell side pressure drop of .26 psi is encountered.

The minimum approach temperature of 75°F occurs immediately prior to the boiling section of the heat exchanger. Design heat transfer coefficients of 70, 75, and 80 BTU/Hr-F^2-F° are found in the sensible heating, boiling and superheating regions, respectively.

Each waste heat boiler requires an adaptor manifold to connect the 24" x 36 exhaust outlet to the 9.2 x 8 foot waste heat boiler. In addition, a mounting frame and two to three inches of insulation must be included. Finally, provisions for safety valves and control sensors must also be made for safety reasons.

Table 33 is a complete listing of all pertinent specifications for each boiler. Refer to Economics Section for costing information.

Turbine

The two high pressure vapor lines from each waste heat boiler combine to form a total of 165,140 lbs/hr of superheated Flourinol 85 at 700 psia. This vapor is expanded through a six stage axial turbine resulting in 4672 Hp of shaft work. A conservative turbine efficiency of 80 percent was chosen in spite of recommendations in Reference 2. In addition, a vertically split case is specified for more reliable sealing. The turbine shaft work is routed through a transfer gearbox to synchronize it with the shaft work produced by the driver reaction turbine. Turbine design is considered to be at a level equivalent to that proposed in Reference 2. Table 34 describes the turbine design parameters.

High Pressure Pump

A 435 GPM high pressure centrifugal pump is needed to compress the saturated Flourinol 85 liquid from 5 psia to 700 psia. A total of 5 stages are needed to accomplish this compression each with a horsepower load of 40.2 Hp. A Gould model #VIC or equivalent pump may be used. The total horsepower load of the high pressure

TABLE 33.--Waste Heat Boiler Design Parameters for Organic Rankine Cycle (GG-12 Driver Case).

TUBESIDE:

Fluid - Flourinol 85
Inlet Temperature - 120°F
Outlet Temperature - 625°F
Inlet Pressure - 700 psia
Pressure Drop - .35 psi
Flow Rate - 82,570 #/Hr
Tube Material - 14 BWG Carbon Steel
Tube O.D. - 1 Inch
Tube Length - 8 Feet
Tube Pitch - 2 Inch Triangular
Fin Height - 3/8 Inch
Fin Density - 96 Fins per Foot
Fin Type - Aluminum Serrated Circular-Press Fit
Tubes per Pass - 55
Total Number of Passes - 32
Dummy Tubes - 1 per Pass

SHELLSIDE:

Fluid - Natural Gas Combustion Products
Inlet Temperature - 840°F
Outlet Temperature - 207°F
Inlet Pressure - .36 psig
Pressure Drop - .26 psi
Flow Rate - 158,400 lbs/hr
Shell Material - 12 Gauge Carbon Steel
Shell Dimensions - (W,L,H) 9.2' x 8.0' x 4.6'
Free Cross-Sectional Area - 29.0 Ft^2
Insulation - 2" Rock Wool
Baffles - None
Number of Passes - 1

HEAT DUTY:

Sensible Heating Region - 14.92 MM BTU/Hr
Boiling Region - 4.29 MM BTU/Hr
Superheating Region - 7.84 MM BTU/Hr
Total - 27.05 MM BTU/Hr

OVERALL HEAT TRANSFER COEFFICIENT:

Sensible Heating Region - 70 BTU/Hr-Ft^2-F°
Boiling Region - 75 BTU/Hr-Ft^2-F°
Superheating Region - 80 BTU/Hr-Ft^2-F°

LOGMEAN TEMPERATURE DIFFERENCE:

Sensible Heating Region - 94.9°F
Boiling Region - 163.3°F
Superheating Region - 216.9°F

TABLE 34.--Turbine Design Parameters for Organic Rankine Cycle (GG-12 Driver Case).

Fluid - Flourinol 85
Turbine Horsepower Output - 4672 Hp
Inlet Conditions:
Temperature - 625°F
Pressure - 700 psia
Outlet Conditions:
Temperature - 336°F
Pressure - 5 psia
Superheated Vapor
Flow Rate - 165,140 lbs/hr
Enthalpy Change - 72.0 BTU/Lb
Thermal Efficiency - 80 Percent
Number of Stages - 6
Type - Axial Flow - Vertically Split Case
RPM - 6500

Reference 2

TABLE 35.--High Pressure Pump Design Parameters for Organic Rankine Cycle (GG-12 Driver Case).

Inlet Pressure - 5 psia
Outlet Pressure - 700 psia
Inlet Temperature - 120°F
Flow Rate - 435 GPM
Efficiency - 70 Percent
Hp Rating - 201 Hp
RPM - 1150
Number of Stages - 5
Model Number - Gould Model VIC

Reference 2

pump is 201 Hp operating at 70 percent efficiency. Table 35 describes the design parameters for this pump.

Water Cooled Condenser and Forced Draft Cooling Tower

The lower condensing temperature of Flourinol 85 as compared with steam makes it expedient to use a closed cycle forced draft cooling tower as opposed to the air cooled condensers used in the steam rankine cycles. The small temperature driving force encountered results in exceedingly large air cooled condensers at the low condensing temperatures required for Flourinol 85. It is therefore economically more sensible to utilize a forced draft cooling tower as a source of cooling water and a water cooled condenser within the rankine cycle itself.

The water cooled condenser is simply a shell and tube heat exchanger mounted such that the tubes are horizontal. A condensate drum is an integral part of the shell and feeds directly to the system high pressure pump. A total of 165,140 lbs/hr of superheated vapor enter the shellside at 336°F, exiting the condensate drum at 120°F as saturated liquid. Water flows through the tube side at a flowrate of 4270 GPM in counter flow to the vapor. Cooling water from the forced draft cooling tower is supplied at 85°F maximum and returns at 105°F maximum. The condenser heat load is 42.6 million BTU per hour. The total tube side heat transfer area is 3343 square feet. Table 36 provides a complete description of the water cooled condenser.

TABLE 36.--Condenser Design Parameters for Organic Rankine Cycle (GG-12 Driver Case).

U Overall - 540 BTU/Hr-Ft^2-F° (Condensing Organic Vapor)

Q Overall - 42.6 MM BTU/Hr

ΔT_{lm} - 23.6°F

Area - 3343 Ft^2

TUBESIDE:

Fluid - Treated Water
Flow Rate - 4270 GPM
Inlet Temperature - 85°F
Outlet Temperature 105°F
Pressure - 15.5 psia
Design Pressure - 150 psia
Tubing - 1" O.D. 12 BWG Carbon Steel
Tube Length - 16 Feet
Number of Tubes - 798
Pressure Drop - .5 psi
Cross-Sectional Area - 4.35 Ft^2

SHELLSIDE:

Vapor - Flourinol 85
Flow Rate - 165,140 Lbs/Hr
Inlet Temperature - 336°F (Superheated)
Outlet Temperature - 120°F Saturated Liquid
Operating Pressure - 5 psia
Shell Diameter - 4.2 Feet
Free Cross Sectional Area - 9.2 Ft^2
Shell Material - 12 Gauge Carbon Steel
Insulation - 2" Rock Wool
Condensate Drum - 200 Gallon Capacity

A closed system forced draft barometric cooling tower is the source of cooling water for condensation. The tower is similar to those commonly seen at power plants, however, on a somewhat smaller scale. The tower supplies a minimum of 4270 GPM of 85°F water for use by the condenser. A base area of approximately 700 square feet is estimated along with 74 horsepower of fan capacity to perform this task. Total additional water consumption is 80 GPM (evaporation and blowdown losses) which is minimal and could be supplied easily by a small aquifer. The cycle concentration is set to eight cycles usage to minimize blow down losses. A complete description of the cooling tower design specifications is found in Table 37.

TABLE 37.--Cooling Tower Design Parameters for Organic Rankine Cycle (GG-12 Driver Case).

Type - Forced Draft Barometric Cooling Tower
Base Area - 710 Ft^2 (25' x 30' approx.)
Number of Cells - 1
Maximum Dry Bulb Temperature - 100°F
Maximum Wet Bulb Temperature - 80°F
Mean Wet Bulb - 40°F
Water Flow Rate - 4270 GPM
Inlet Temperature - 105°F (Max)
Outlet Temperature - 85°F (Max)
Fan Horsepower - 74 Hp
Evaporation Loss - 70 GPM
Drift Loss - 1.5 GPM
Blow Down Rate - 8.5 GPM
Concentration Cycles - 8

Reference 2

Economic Analysis

All economics for this design are performed assuming a 3rd Quarter 1980 installation and a 1st Quarter 1981 start-up. The total fixed capital investment includes the cost of equipment, installation and a small auxillary building housing the turbine, condenser and pump assemblies. Interest for any construction loans is not included. In addition, costs associated with start-up and debugging are not included.

Ten year straight-line depreciation is used for all economic evaluations and zero salvage value is assumed. Labor is assumed to be available at $20,800 per man year with any additional supervisory labor in existence. All dollars are based on 1980 valuations. Corporate income taxes are assumed to be at 48 percent of gross operating profit. Should government wave all or part of these taxes a substantial addition to the profitability of these ventures would occur.

Electricity is assumed to be available at 8 cents per kilowatt hour. Cooling water is assumed available in small quantities at 5 cents per thousand gallons. Natural gas is assumed to increase in value as described in Figure 11.

All remaining fixed costs are considered as factored estimates based on the total fixed capital investment. Maintenance is at a rate of 4 percent of the TFCI per year. In addition, a maintenance credit for the savings realized by not needing to operate a gas powered driver to provide the shaft work obtained from the organic rankine cycle is also included. Plant overhead and insurance are

assumed to be at 2.6 and 1.5 percent of TFCI per year, respectively. Finally, interest on working capital is assumed to be at 15 percent per year.

Once more an alternatives approach is used for economic evaluation. Therefore, natural gas savings (which is the only revenue source) is computed as the quantity of natural gas which would be required to power the drive motor in question to obtain an equal quantity of shaft work to that which is provided by the rankine cycle. This is a critical assumption and must be considered carefully before applying this economic analysis to similar drivers with varying efficiencies. It is generally true that the less efficient the base driver is, the more favorable the economics for the rankine cycle used in conjunction with that driver.

The total cost for all equipment, instrumentation, electrical and piping for the rankine cycle described is $1,568,000. This figure includes installation of all equipment but excludes the cost of any construction loans. A complete breakdown of individual equipment and installation costs is found in Table 38. The operating profit of this design including depreciation is $307,700 per year at 25 percent stream factor. A complete breakdown of the operating costs is found in Table 39. A payout period of 4.0 years is anticipated for this design at 25 percent stream factor. This corresponds to a 19 percent discounted cash flow rate of return. Finally, ten year cash flow summary as a function of stream factor is presented in Table 40.

TABLE 38.--Individual Equipment Cost Breakdown for Organic Rankine Cycle (GG-12 Driver Case).

Equipment Type	Equip. Cost $	Install. Cost $	Material Cost $	Indirect Cost $	Total Cost $
Turbine	146,000	43,000	39,500	87,000	315,500
Reduction Gearbox	14,500	4,500	4,000	8,500	31,500
Feed Pump	85,500	52,500	54,000	73,000	265,000
Waste Heat Boilers	135,000	39,500	39,000	81,500	295,000
Condenser	50,000	30,000	35,000	44,000	159,000
Cooling Tower	60,000	13,000	13,500	15,500	102,000
Aux. Building	16,500	7,000	--	25,000	48,500
Working Fluid	44,000	--	--	17,000	61,000
SUB TOTALS	551,500	230,000	185,000	351,500	1,277,500

Total Delivered Equipment Cost	551,500
Total Controls, Electrical Piping, Painting Cost	185,000
Total Installation Labor Cost	230,000
Total Direct Cost	966,500
Total Indirect Cost	351,500
Total Direct and Indirect Cost	1,318,000
Contractors Fees	52,500
Contingency @ 15%	197,500
Total Fixed Capital Investment (1980 Dollars)	1,568,000

References 1, 2, 3, 10.

TABLE 39.--Operating Cost Breakdown for Organic Rankine Cycle (GG12 Driver Case).

Total Fixed Capital Investment - $1,568,000

Stream Factor - 25 Percent

Utilities	Quantity of Units	$/Unit	$/Year
Electricity	461,925 KWH	.08	37,000
Make-up Cooling Water	9460/1000 Gal	.15	1,500
Natural Gas	162,745 MM BTU**	3.87	(629,800)
Labor			
Labor	1 Man Year	20,800	20,800
Overhead and Fringe Benefits		20,800	20,800

Capital Related	
Maintenance @ 4%/year (includes credit)*	18,500
Plant Overhead @ 2.6%/year	40,500
Insurance @ 1.5%/year	23,500
Depreciation @ 10%/year	157,000
Interest on Working Capital @ 12%/year	3,500
Total Operating Costs (Excluding Depreciation and Gas Credit)	166,100
Total Operating Profit (includes Depreciation)	306,700
Working Capital	33,200

*Credit of $10.02 per Hp-capacity for original driver replacement.

**Calculated as fuel required to produce 4672 HP @ 25% stream factor in Pratt-Whitney GG-12 driver.

TABLE 40.--After 48% Taxes Ten Year Cash Flow Summary vs Stream Factor for Organic Rankine Cycle (GG-12 Driver Case).

Year	25%	40%	75%	90%
1980	(1,568,000)	(1,568,000)	(1,568,000)	(1,568,000)
1981	317,000	513,500	972,000	1,168,500
1982	332,900	539,500	1,020,800	1,227,100
1983	349,800	565,600	1,069,600	1,285,700
1984	365,900	591,600	1,118,400	1,344,200
1985	382,000	617,600	1,167,200	1,402,800
1986	398,000	643,700	1,216,000	1,461,400
1987	414,000	669,700	1,264,500	1,519,900
1988	431,000	695,700	1,313,600	1,578,500
1989	447,100	721,700	1,362,500	1,637,000
1990	496,600	781,000	1,444,500	1,728,800
Payout Period (Years)	4.0	2.4	1.3	1.0
Discounted Cash Flow Rate of Return	19.0%	33.7%	64.2%	76.6%

Stream Factor Sensitivity

Stream factor is the single most important variables in determining the profitability of organic rankine bottoming cycles. As stream factor increases above 50 percent, payout periods of less than two years could be expected for the design case. At above 90 percent stream factor, payout periods of approximately 1.1 years could be expected. Unfortunately, most compressor stations are in operation less than three months of the year. Load shifting to increase this stream factor for a selected group of drivers that have been retrofitted with organic rankine bottoming cycles is a promising alternative. With load shifting, less natural gas would be pumped over longer periods of operation thereby increasing the stream factor for the more efficient retrofitted drivers. Table 40 presents the payout period and discounted cash flow rate of return as a function of stream factor.

Summary of Cooper-Bessmer W-330 Organic Rankine Cycle

A simple four stage organic rankine cycle utilizing Flourinol-85 as the working fluid was chosen for the Cooper-Bessmer W-330 reciprocating driver. The cycle described in the following tables and similar in concept to that presented earlier, requies a maximum cycle temperature of 625°F and a maximum pressure of 700 psia (recommended by Halocarbon Inc.). A vapor flow rate of 47,226 lbs/hr is attainable when waste heat is used from a pair of the Cooper-Bessmer drivers. The horsepower rating of such a rankine cycle is 1335 Hp resulting in a 17 percent increase in the combined

horsepower output of the 3900 Hp individual drivers. Each driver exhausts 53,950 pounds per hour of clean 750°F exhaust gases and operates at 37 percent thermal efficiency.

Table 32 (page 146) presents the thermodynamic conditions for the organic rankine cycle concept described in Figure 15 (page 124). Tables 41 through 45 present the individual design parameters and equipment sizes for each major component of the design.

The total fixed capital investment for this design is $923,000. A payout period of 6.1 years is estimated for a 90 percent stream factor increasing to 9.8 years at 60 percent stream factor. These payout periods represent discounted cash flow rates of return of 9.2 and zero percent, respectively. Table 45 presents a ten-year cash flow summary as a function of stream factor. An annual natural gas savings of $283,300 is realized at 90 percent stream factor and 1981 gas prices. Table 46 presents the individual equipment costs, installation costs, etc., for the design case. Table 47 is a breakdown of the 1981 operating costs at 25 percent stream factor.

TABLE 41.--Waste Heat Boiler Design Parameters for Organic Rankine Cycle (W-330 Driver Case).

TUBESIDE:

- Fluid - Flourinol 85
- Inlet Temperature - 120°F
- Outlet Temperature - 625°F
- Inlet Pressure - 700 psia
- Pressure Drop - 4.9 psi
- Flow Rate - 23,613 lbs/hr
- Tube Material - 12 BWG Carbon Steel
- Tube O.D. - 1 Inch
- Tube Length - 8 Feet
- Tube Pitch - 2 Inch Triangular
- Fin Height - 3/8 Inch
- Fin Density - 96 Fins per Foot
- Fin Type - Aluminum Serrated Circular Fin-Press Fit
- Tubes per Pass - 20
- Total Number of Passes - 32
- Dummy Tubes - 1 per Pass

HEAT DUTY:

- Sensible Heating Region - 4.27 MM BTU/Hr
- Boiling Region - 1.23 MM BTU/Hr
- Superheating Region - 2.24 MM BTU/Hr
- Total - 7.74 MM BTU/Hr

LOGMEAN TEMPERATURE DIFFERENCE

- Sensible Heating Region - 79.3°F
- Boiling Region - 109.4°F
- Superheating Region - 140.9°F

SHELLSIDE:

- Fluid - Natural Gas Combustion Products
- Inlet Temperature - 750°F
- Outlet Temperature - 207°F
- Inlet Pressure - .36 psig
- Pressure Drop - .26 psi
- Flow Rage - 53,950
- Shell Material - 12 Gauge Carbon Steel
- Shell Dimensions - 3.33 x 4.52 x 8.0
- Free Cross-Sectional Area - 10.53 Ft^2
- Insulation - 2" Rock Wool
- Baffles - None
- Number of Passes - 1

OVERALL HEAT TRANSFER COEFFICIENT:

- Sensible Heating Region - 70 BTU/Hr-Ft^2-F°
- Boiling Region - 75 BTU/Hr-Ft^2-F°
- Superheating Region - 80 BTU/Hr-Ft^2-F°

HEAT TRANSFER AREA

- Sensible Heating Region - 770 Ft^2
- Boiling Region - 150 Ft^2
- Superheating Region - 200 Ft^2
- Total - 1120 Ft^2

TABLE 42.--Turbine Design Parameters for Organic Rankine Cycle (W-330 Driver Case).

Fluid - Flourinol 85
Turbine Horsepower Output - 1335 Hp
Inlet Conditions:
    Temperature - 625°F
    Pressure - 700 psia
Outlet Conditions:
    Temperature - 336°F
    Pressure - 5 psia
Flow Rate - 47,226 lbs/hr
Enthalpy Change - 72.0 BTU/lb
Thermal Efficiency - 80 Percent
Number of States - 6
Type - Axial Flow Horizontally Split Case
RPM - 6500

Reference 2

TABLE 43.--High Pressure Pump Design Parameters Organic Rankine Cycle (W-330 Driver Case).

Inlet Pressure - 5 psia
Outlet Pressure - 700 psia
Inlet Temperature - 120°F
Flow Rate - 125 GPM
Efficiency - 70 Percent
Hp Rating - 58 Hp
RPM - 1150
Number of Stages - 5
Model Number - Gould Model VIC

Reference 2

TABLE 44.--Condenser Design Parameters for Organic Rankine Cycle (W-330 Driver Case).

U Overall - 540 BTU/Hr-Ft^2-F° (Condensing Organic Vapor)

Q Overall - 12.2 MM BTU/Hr

ΔT_{lm} - 23.6°F

Area - 960 Ft^2

TUBESIDE:

Fluid - Treated Water

Flow Rate - 1225 GPM

Inlet Temperature - 85°F

Outlet Temperature - 105°F

Pressure - 15.5 psia

Design Pressure - 150 psia

Tubing - 1 Inch OD 12 BWG Carbon Steel

Tube Length - 12 Feet

Number of Tubes - 305

Pressure Drop - .5 psi

Cross Sectional Area - 1.66 Ft^2

SHELLSIDE:

Vapor - Flourinol 85

Flow Rate - 47,226

Inlet Temperature - 336° (Superheated)

Outlet Temperature - 120°F (Saturated Liquid)

Operating Pressure - 5 psia

Shell Diameter - 2.6 Feet

Free Cross Sectional Area - 3.52 Ft^2

Shell Material - 12 Gauge Carbon Steel

Insulation - 2" Rock Wook

Condensate Drum - 60 Gallon Capacity

TABLE 45.--Cooling Tower Design Parameters for Organic Rankine Cycle (W-330 Driver Case).

Type - Forced Draft Barometric Cooling Tower

Base Area - 205 Ft^2 (15' x 14' Approx.)

Number of Cells - 1

Maximum Dry Bulb Temperature - 100°F

Maximum Wet Bulb Temperature - 80°F

Mean Wet Bulb - 40°F

Water Flow - 1225 GPM

Inlet Temperature - 105°F (Max.)

Outlet Temperature - 85°F (Max.)

Fan Horsepower - 22 Hp

Evaporation Loss - 21 GPM

Drift Loss - .5 GPM

Blow Down Rate - 2.5 GPM

Concentration Cycles - 8

Reference 2

TABLE 46.--Individual Equipment Cost Breakdown for Organic Rankine Cycle (W-330 Driver Case).

Equipment Type	Equip. Cost $	Install. Cost $	Material Cost $	Indirect Cost $	Total Cost $
Turbine	86,500	25,500	23,000	51,000	186,000
Reduction Gearbox	8,500	2,500	2,000	5,000	18,000
Feed Pump	68,500	46,500	47,500	62,000	224,500
Waste Heat Boilers	88,500	26,000	25,500	54,000	194,000
Condenser	16,500	10,000	11,500	15,000	53,000
Cooling Tower	20,000	4,500	4,500	5,000	34,000
Aux. Building	16,500	7,000	--	25,000	48,500
Working Fluid	13,000	--	--	5,000	18,000
SUB TOTALS	318,000	122,000	114,000	222,000	776,000
	Total Delivered Equipment Cost				318,000
	Total Controls, Electrical, Piping, Painting Costs (Materials)				114,000
	Total Installation Labor Cost				122,000
	Total Direct Cost				554,000
	Total Indirect Cost (Engineering)				222,000
	Total Direct and Indirect Cost				776,000
	Contractor's Fees (@ 4% TD + I)				31,000
	Contingency (@ 15% TD + I)				116,000
	Total Fixed Capital Investment (All 1980 Dollars)				$923,000

References 1, 3, 10

TABLE 47.--Operating Cost Breakdown for Organic Rankine Cycle (W-330 Driver Case).

Total Fixed Capital Investment - $923,000

Stream Factor - 25 Percent

Utilities	Quantity of Units	$/Unit	$/Year
Electricity	132,100 KWH	.08	10,600
Make-up Cooling Water	2,700 M Gal.	.15	500
Natural Gas	20,330 MM BTU**	3.87	(78,700)
Labor			
Labor	1 Man Year	20,800	20,800
Overhead and Fringe Benefits		20,800	20,800

Capital Related	
Maintenance @ 4%/year (includes credit)*	23,500
Plant Overhead @ 2.6%/year	24,000
Insurance @ 1.5%/year	13,900
Depreciation @ 10%/year	92,300
Interest on Working Capital @ 12%/year	2,800
Total Operating Costs (Excluding Depreciation and Natural Gas Credit)	116,900
Operating Loss (includes Natural Gas Credit and Depreciation)	(38,200)
Working Capital	23,400

*Credit of $10.02 per Hp-capacity for replacement of original driver.

**Calculated as fuel required to produce 1315 HP @ 25% stream factor in Cooper-Bessmer W-330 driver.

TABLE 48.--After 48% Tax Cash Flow Projections for Organic Rankine Cycle (W-330 Driver Case)

Year	25%	60%	75%	90%
1980	(923,000)	(923,000)	(923,000)	(923,000)
1981	(38,200)	56,500	94,700	115,800
1982	(34,500)	65,700	100,700	123,000
1983	(30,500)	74,900	106,800	130,300
1984	(26,500)	84,700	113,100	137,900
1985	(22,600)	93,200	119,100	145,100
1986	(18,700)	98,000	125,200	152,300
1987	(14,800)	102,800	131,200	159,600
1988	(10,800)	107,900	137,500	167,200
1989	(6,900)	112,700	143,500	174,400
1990	20,300	140,900	173,000	205,000
Discounted Cash Flow Rate of Return	0	0	5.3%	9.2%
Payout Period (Average Cash Flow) Years	N/A	9.8	7.4	6.1

APPENDIX V

DESIGN FOR PROCESS STEAM PRODUCTION

APPENDIX V

DESIGN FOR PROCESS STEAM PRODUCTION

Introduction

One of the simplest and least costly energy recovery methods available is the production of 150 psia process steam in the waste heat boiler. This method requires a minimal capital investment and no additional labor or supervision. Maintenance and overhead are very limited and variable costs are nearly insignificant.

Process steam could be used by a variety of users from space heating applications to plant steam for commercial laundry facilities. In addition, a number of oil fields exist in close proximity to many natural gas storage facilities. Process steam could be used for secondary and tertiary oil recovery in these fields. Finally, natural gas often requires preheating before throttling into distribution pipelines. This could be accomplished with process steam. Each of these ideas is feasible. However, the single factor retarding their implementation is the timing and sporacity of operation of almost all natural gas compresser stations.

Most natural gas compresser stations actively operate from mid-summer through the fall in preparation for winter gas demand. During this period of time, they operate at high load factors, thereby creating substantial amounts of waste heat. However, as

winter approaches compression activities diminish and storage fields are bled into the major distribution pipelines. Process steam made during the active period for most stations would be useless for space heating and gas preheating, since it would be available during the wrong time period. Sporacity of operation would be a problem for any process steam user since provisions for making process steam would have to be made for the compresser stations' dormant periods. In summary, a prospective user would need to work cooperatively with the steam producer to make energy recovery successful for both.

Process Description

Table 41 lists the driver types examined and the pertinent process information for each design case. As is evident steam flow rates from 6,085 to 22,320 lbs/hr are attainable depending on the driver type used. A maximum temperature of 358°F is attained at the saturation point of 150 psia steam.

The cycle begins with 192°F sub-cooled water at 14.7 psia (20° of sub-cooling is assumed in transport) entering high pressure pump. The pump compresses this condensate to 150 psia and delivers it to the waste heat boiler. In the waste heat boiler hot exhaust gases are run in counter flow to the condensate, first heating it to 358°F then vaporizing it to create saturated 150 psia steam. Process steam is then transmitted to the end user who condenses it in his process and returns the condensate to a small condensate pump. This completes the closed cycle and allows reuse of deionized water.

Figure 17 describes this cycle in detail and presents the thermodynamic conditions for each stream.

Equipment Description

A single waste heat boiler is used in this design since there is no real advantage for operating this design in pairs as with previous designs. The boiler is simply a bank of serrated fin tubes surrounded by a plenum to create a simple shell and tube heat exchanger (see Appendix I). Both serrated fin tubes and triangular tube pitch are utilized to introduce turbulence and maximize the limiting outside heat transfer coefficient. Cleaning is not considered to be a problem in this design due to the exceptionally clean natural gas exhaust. For diesel or conventionally fueled drivers soot could be a problem and tube configuration may need to be altered.

Table 49 lists the waste heat boiler design parameters for each of the three driver types examined. These drivers are briefly described in Table 2 and for simplicity the discussion is restricted to the Pratt-Whitney GG-12 driver for the remainder of this description.

A total of 1350 square feet of inside tube surface area is necessary to transfer the 19.72 MM BTU/Hr of waste heat available to this design. Sub-cooled water at 192°F flowing at 22,320 lbs/hr and 150 psia is circulated in counter flow to the hot exhaust gases. Saturated vapor at 358°F and 150 psia is produced with a tube side pressure drop of .42 psi.

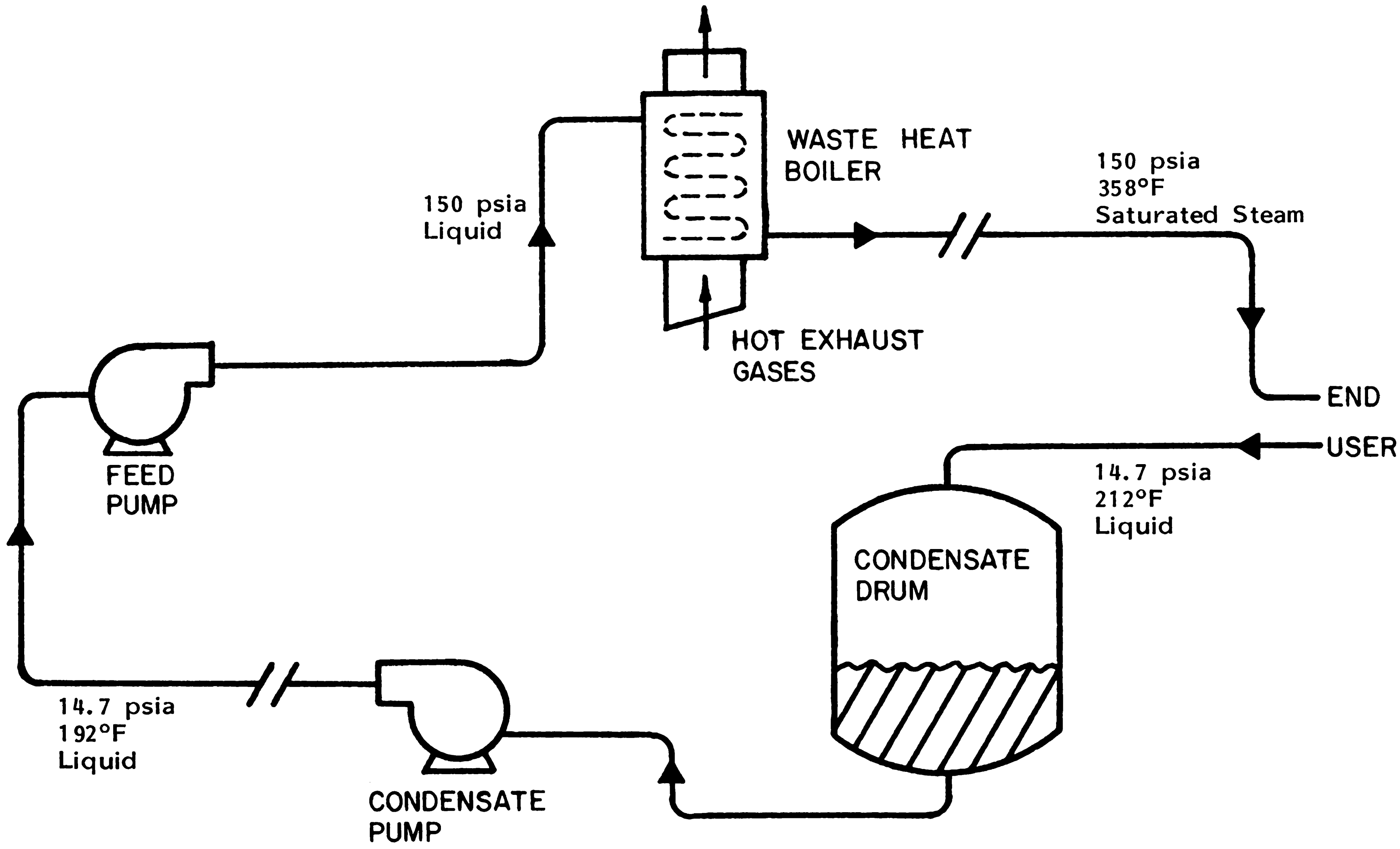

Figure 17.--Flow Diagram for 150 psia Process Steam Production

TABLE 49.--Waste Heat Boiler Design Parameters for 150 psia Process Steam.

	Pratt-Whitney GG-12 Driver	Cooper Bessmer W-330 Driver	Enterprise HV16 C Driver
TUBESIDE:			
Fluid	De-ionized Water	De-ionized Water	De-ionized Water
Inlet Temperature	192°F	192°F	192°F
Outlet Temperature	358°F	358°F	358°F
Inlet Pressure	150 psia	150 psia	150 psia
Pressure Drop	.41 psi	.58 psi	.29 psi
Flow Rate	22,320 Lbs/Hr	6,085 Lbs/Hr	11,645 Lbs/Hr
Tube Material	12 BWG-Carb. Stl.	Same	Same
Tube O.D.	1 Inch	Same	Same
Tube Length	8 Feet	Same	Same
Tube Pitch	2" Triangular	Same	Same
Fin Height	3/8 Inch	Same	Same
Fin Density	96 Fins/Ft	Same	Same
Fin Type	Serrated Aluminum-Press Fit Circular		
Tubes per Pass	37	12	16
Total Number of Passes	21	20	22
Dummy Tubes	1 per Pass	Same	Same
SHELLSIDE:			
Fluid	Natural Gas Combustion Products	Same	Same
Inlet Temperature	840°F	750°F	1000°F
Outlet Temperature	372.7°F	374.8°F	368.8°F
Inlet Pressure	.28 psia	Same	Same
Pressure Drop	.28 psia	.28 psia	.25 psia
Flow Rate	158,400 Lbs/Hr	53,950 Lbs/Hr	60,880 Lbs/Hr
Shell Material	12 Gauge Carbon Steel	Same	Same
Shell Dimensions (W,L,H)	6.2' x 8' x 3'	2' x 8½ x 2.8'	2.78' x 8' x 3.2'
Free Cross Sectional Area	19.5 Ft^2	6.3 Ft^2	8.4 Ft^2
Insulation	2" to 3" Rock Wool	Same	Same
Baffles	None	None	None
Number of Passes	1	1	1
HEAT DUTY:			
Sensible Heating Region	.45 MM BTU/Hr	.12 MM BTU/Hr	.23 MM BTU/Hr
Boiling Region	19.27 MM BTU/Hr	5.25 MM BTU/Hr	10.06 MM BTU/Hr
Total	19.72 MM BTU/Hr	5.37 MM BTU/Hr	10.29 MM BTU/Hr
LOGMEAN TEMPERATURE DIFFERENCE:			
Sensible Heating Region	29.4°F	30.3°F	27.6°F
Boiling Region	155.5°F	140.4°F	193.2°F
OVERALL HEAT TRANSFER COEFFICIENT:			
Sensible Heating Region	86.2 Hr-Ft^2-F°	85.8 Hr-Ft^2-F°	83.9 Hr-Ft^2-F°
Boiling Region	105.9 Hr-Ft^2-F°	102.9 Hr-Ft^2-F°	100.2 Hr-Ft^2-F
HEAT TRANSFER AREA			
Sensible Heating Region	176 Ft^2	47 Ft^2	101 Ft^2
Boiling Region	1170 Ft^2	364 Ft^2	519 Ft^2
Total	1346 Ft^2	411 Ft^2	620 Ft^2

On the shell side 158,400 lbs/hr of hot exhaust gases flow in counter flow through the 19.5 Ft^2 (cross-sectional) shell, entering at 840°F and exiting at 373°F. A total shell side pressure drop of .28 psi is encountered.

The minimum approach temperature of 25°F occurs immediately prior to the boiling section of the heat exchanger. Design heat transfer coefficients of 86.0 and 106.0 BTU/Hr-Ft^2-F° encountered in the sensible heating and boiling regions, respectively.

The waste heat boiler requires an adaptor manifold to connect the 24" x 36" exhaust outlet to the 6.2 x 8 foot waste heat boiler. In addition, a mounting frame and two to three inches of insulation must be included. Finally, provisions for safety valves and control sensors must be made for safety reasons. Refer to the Economics Section for costing information.

Pumps

Two pumps are needed in this design. The first is a high pressure single shaft centrifugal pump. A 6.5 Hp pump and electric drive motor are needed and are assumed to operate at 60 percent efficiency. The flow rate in this pump is approximately 45 GPM. The maximum discharge pressure is 150 psia.

The second pump is a low pressure condensate return pump. A 45 GPM centrifucal pump is needed with a maximum discharge of approximately 20 psia. Again, an operating efficiency of 60 percent is assumed. Table 42 presents the pertinent design parameters for the high pressure pump for each of the driver base cases. Refer to the Economic Analysis section for costing information.

TABLE 50.--High Pressure Pump Design Parameters for 150 psia Steam Production.

	Pratt-Whitney GG-12 Driver	Cooper Bessmer W-330 Driver	Enterprise HVI6C Driver
Inlet Pressure	14.7 psia	Same	Same
Outlet Pressure	150 psia	Same	Same
Inlet Temperature	192°F	Same	Same
Flow Rate	45 GPM	12 GPM	23 GPM
Efficiency	60 Percent	Same	Same
Hp Rating	6.5 Hp	1.8 Hp	3.4 Hp
Fluid	De-ionized Water	Same	Same
Fluid Density	62.4 Lbs/Ft^3	Same	Same
RPM	1750	Same	Same
Type	Centrifugal-Single Shaft	Same	Same
Stages	3 to 5	Same	Same

Reference 2

Economic Analysis

All economics for these three designs are performed assuming a 3rd Quarter 1980 installation and a 1st Quarter 1981 start up. The total fixed capital investment includes the cost of equipment and installation thereof. Interest on construction loans is not included. Costs associated with start-up and debugging are not considered to be significant due to the design simplicity and are therefore ignored.

Ten year straight line depreciation is used with an assumed zero salvage value. No additional operating or supervisory labor is expected to be necessary. Corporate income taxes are assumed to be at 48 percent of gross operating profit. All dollars are at 1980 valuations.

Electricity is assumed to be available at 8 cents per kilowatt hour. All other fixed costs are considered as factored estimates of the total fixed capital investment. Maintenance is at a rate of 4 percent of TFCI while plant overhead and insurance are at 2.6 and 1.5 percent of TFCI, respectively. Interest on working capital is assumed at 15 percent per annum. Finally, process steam is valued at 2.5 dollars per 1000 lbs. produced (3).

Once more discussing only the Pratt-Whitney GG-12 design for simplicity, a total fixed capital investment of $112,800 is necessary for all equipment, piping, electrical and installation costs. Total yearly operating costs including depreciation are $21,200 with depreciation accounting for $11,300 of this amount. A total of $122,200 in revenues is generated via the sale of 150 psia steam. A payout period of 1.8 years is expected for 25 percent steam factor operation. Table 51 presents the individual equipment and installation costs for all three driver base cases. Table 52 presents the yearly operating costs for each of these base cases.

TABLE 51.--Individual Equipment Costs for 150 psia Process Steam Production.

	Pratt-Whitney GG-12 Driver	Cooper Bessmer W-330 Driver	Enterprise HV16C Driver
Waste Heat Boiler	$41,400	$20,300	$26,000
High Pressure Feed Pump	2,000	1,400	1,600
Condensate Pump	600	600	600
Total Delivered Equipment	44,000	22,300	28,200
Controls, Piping, Electricity	13,800	7,300	9,100
Installation Labor	13,900	7,400	9,100
Total Direct Cost	71,700	37,000	46,400
Indirect Cost (Engineering)	27,200	14,000	17,600
Total Direct and Indirect Cost	98,900	51,000	64,000
Contractor's Fee	4,000	2,000	2,600
Contingency	9,900	5,100	6,400
Total Fixed Capital Investment (All 1980 Dollars)	$112,800	$58,100	$73,000
Working Capital	1,600	800	1,000

References 1, 2, 3

TABLE 52.--Operating Cost Breakdown for 150 psia Process Steam Design (25 Percent Stream Factor).

	Pratt-Whitney GG-12 Driver	Cooper Bessmer W-330 Driver	Enterprise HV16C Driver
Total Fixed Capital Investment	112,800	58,100	73,000
150 psia Steam Produced	22,320 Lbs/Hr	6,085 Lbs/Hr	11,645 Lbs/Hr
Utilities:			
Electricity	600	200	300
Steam Credit	(122,200)	(33,300)	(63,800)
Capital Related:			
Maintenance	4,500	2,300	2,900
Plant Overhead	2,900	1,500	1,900
Insurance	1,700	900	1,100
Depreciation	11,300	5,800	7,300
Interest on Working Capital	200	100	100
Total Operating Costs (excluding Depreciation and Steam Credit)	9,900	5,000	6,300
Total Operating Profit	101,000	22,500	50,200
Cash Flow 1981	63,800	17,500	33,400
Payout Period	1.8 years	3.3 years	2.2 years
Working Capital	1,600	800	1,000