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DOE/CS/40008--T8

DEVELOPMENT AND PRODUCTION COST ANALYSIS FOR SUBATMOSPHERIC BRAYTON SYSTEM

FOR

BRAYTON-CYCLE HEAT-RECOVERY SYSTEM CHARACTERIZATION PROGRAM

MASTER

DOE/CS/40008--T8

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SUBTASK A.4.2.4 OF PHASE I PROGRAM PLAN
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ALPHA

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DISCLAIMER

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REPORT SUMMARY

This document reports on the cost analysis previously performed for a subatmospheric Brayton cycle system sized to operate off the waste heat of a nominal 200 ton per day glass container furnace. Along with product value and operating costs, the cost data for system components is one of the essential elements required for analyses and prediction of the return-on-investment for a Brayton system. The expected ROI or payback will in turn impact the marketibility of the system. This report fulfills the requirements for documenting the work accomplished, and submitting the information to Garrett AiResearch for use in the Preliminary System Cost Analysis Subtask 4.2.4 of the Phase I Program Plan, 79-16411, Rev. 2 dated December 17, 1979.

This report describes estimated costs in 1978 dollars for both the first development system and for a mature system. Development system costs include design, fabrication and test cost elements for each major component as well as identified tooling costs. Mature system component costs are predicated on a system delivery rate of one hundred per year. Modifications will be made by Garret AiResearch to the mature system costs as component items are changed or when further analysis conducted as a part of this program indicates that a change is necessary.

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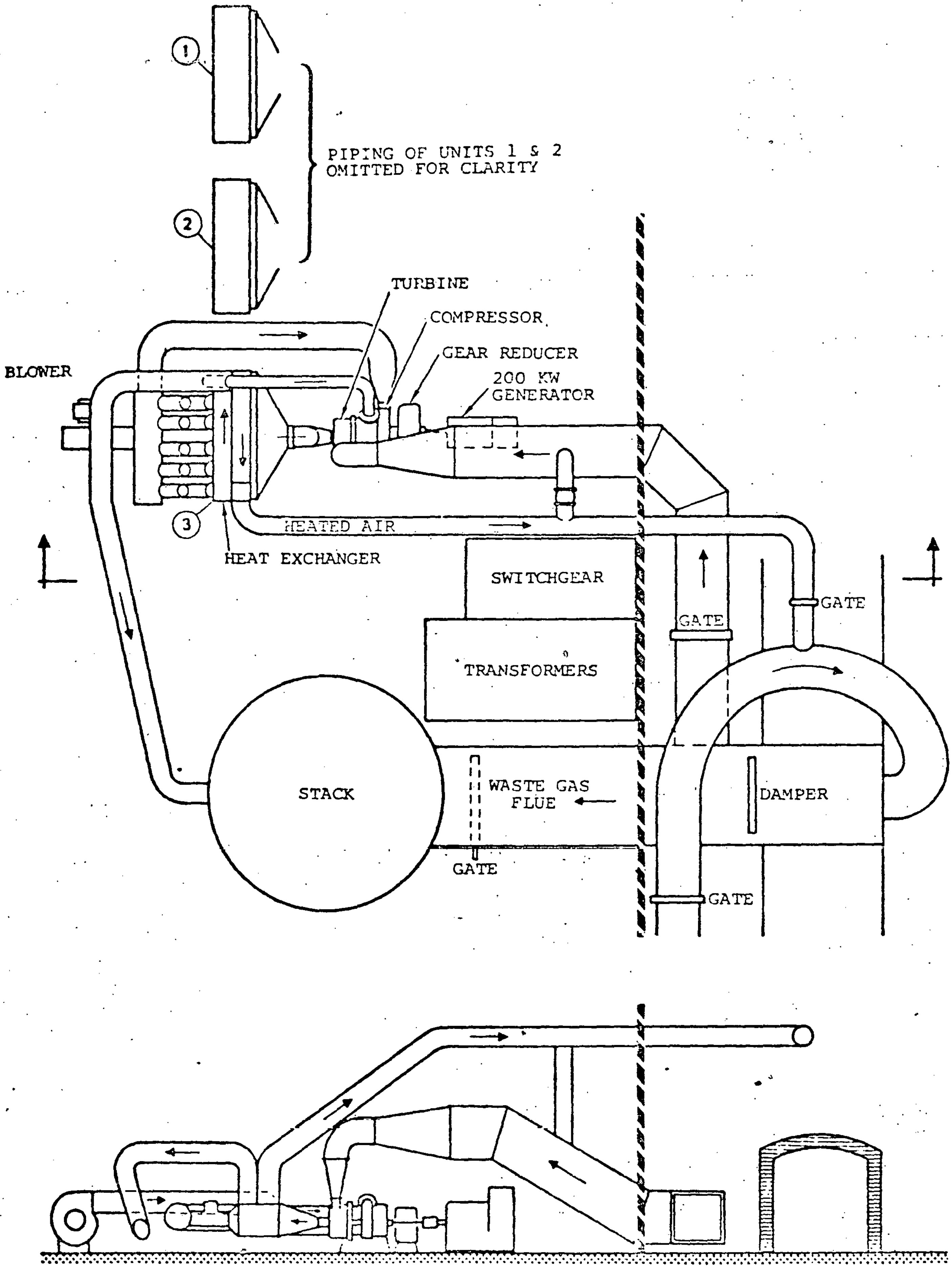
1.0 INTRODUCTION

In order to perform a market analysis of the Brayton waste recovery system, it is necessary to derive a reasonable estimate of the return-on-investment or payback potential. This type of analysis requires estimates of the system product values of electricity and air preheat as well as system operating and component costs for initial and mature systems. This document is restricted to a presentation of the cost data gathered in the previous economic analysis performed by Alpha Glass as required by the program plan subtask A4.2.4. Subsequent analysis will provide assessment of the commercialization potential of the system based on its projected ROI and payback (the program goal for payback is 2 years). This work will require the reevaluation of these system costs based upon the final configuration selected by AiResearch, as well as an update to the product value estimates previously establised.

2.0 ASSUMPTIONS OF SYSTEM CONFIGURATION FOR COSTING PURPOSES

The subatmospheric system configuration used for costing purposes is schematically presented in Figure 2.1. Details of turbine/heat exchanger units 2 and 3 have been omitted for clarity.

The baseline subatmospheric hardware cost configuration consists of three modules, each of which would contain one turbocompressor, a 230 KW generator, plate and fin heat exchanger with automated self-cleaning system, a control system, electrically driven fan and installation ducting. The installation ducting would integrate the furnace flow into the three modules and the air preheat from the three heat exchanger modules into the furnace combustion air system. In a subatmospheric cycle, the components which see the effluent streams are the turbine, heat exchanger and compressor. The three module were sized to operate on a 200-ton/day tank. Each subatmospheric system module operates between ambient pressure and subatmospheric pressure. For design purposes, the waste gas stream was estimated to enter the turbine directly at 1528^{o}F, where it is expanded to 1100^{o}F. The gas enters the heat exchanger and is cooled to 112^{o}F, then compressed to 395^{o}F and exhausted at atmospheric pressure. The heat exchanger used to cool the gases has a 2 to 1 mass flow ratio; some of the flow is dumped with the waste gas stream from the compressor. The other part of the cooling air for the heat exchanger is heated to approximately 900^{o}F and used as an air preheat for the furnace.



SCALE $\frac{1''}{4} = 1'-0''$

FEB.12 1979

Figure 2.1- Subatmospheric System Operating With Glass Plant.

2.1 Turbocompressor Design Definition

The design requirements for the turbine selected for application to the 200-ton glass furnace are as follows as defined by the standard day conditions (60°F) below.

Turbine inlet temperature	1538°F
Turbine inlet pressure	14.70 psia
Turbine flow	237.1 lbs. per min.
Turbine discharge temperature	1102°F
Turbine discharge pressure	4.30 psia
Turbine efficiency	0.880
Compressor inlet temperature	112°F
Compressor inlet pressure	4.09 psia
Compressor flow	240.3 lbs. per min.
Compressor discharge pressure	14.77 psia
Compressor discharge temperature	395°F
Compressor efficiency	0.82

The maximum turbine inlet temperature will occur on a hot day (100°F) and estimated performance is shown below.

Turbine inlet temperature	1580°F
Turbine inlet pressure	14.70 psia
Turbine flow	234.6 lbs. per min.
Turbine discharge temperature	1150°F
Turbine discharge pressure	4.59 psia
Turbine efficiency	0.885
Compressor inlet temperature	158°F
Compressor inlet pressure	4.39 psia
Compressor flow	237.8 lbs. per min.
Compressor discharge temperature	441°F
Compressor discharge pressure	14.77 psia
Compressor efficiency	0.820

The final design that was selected and used for cost estimating purposes was the result of four evolutions of conceptual designs to reduce program as well as production costs of the commercialized system. The evolution of designs and the one selected for costing (final design) is shown in Figure 2.2.

Fabrication facility costs for producing high precision exotic metal products like the turbine have changed dramatically from the program inception. The modifications in design reflect smaller component parts and increased rotational speeds of the machines to achieve lower costs in the current vendor environment. Vendors capable of producing critical components for the prime turbine manufacturer increase at a more rapid rate if component parts decrease in size and weight producing a more competitive environment. Having an increased subcontractor selection not only decreases product cost but minimizes the chance for production delays due to long hardware lead times that could be caused by the cues in the few shops capable of fabricating the large components.

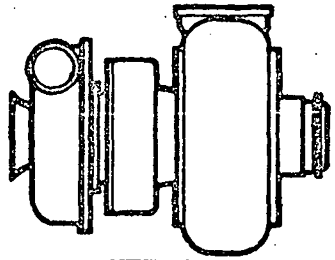
The gear system went through four evolutions of designs. The first system utilized a shaft speed alternator having no gears and no lube oil systems. The second and third system designs utilized an undeveloped right angle gear drive system. The selected baseline design for costing purposes uses an existing gear drive system from the Garrett 831-800 industrial turbine. This evolution has reduced not only development costs but also component costs because the production units will benefit from existing production volume.

The air bearing system was reduced in the design evolution from a 7-1/2 diameter statement-of-the-art bearing to 4-1/2" diameter bearing, just 1/8" larger than an existing bearing from the current Garrett turbocompressor system used in fuel cells. The change to the smaller air bearing size greatly reduced the development risk that was associated with the larger air bearing size and should result in lower production costs.

The current design of the turbocompressor system is one which should be capable of being developed within forecast cost and commercially produced in quantity to satisfy investment and marketing goals. Alpha made an independent cost analysis of the turbine development cost that was required for the initial

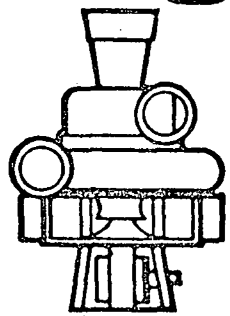
TURBINE EVOLUTION IN ALPHA GLASS
DOE BRAYTON CYCLE WASTE HEAT RECOVERY

TURBINE WHEELS	
2 STAGE	21"
COMPRESSOR WHEEL	23.4"
GEAR BOX	NONE
SPEED	15,000
PRESSURE RATIO	3.8
BEARINGS	AIR



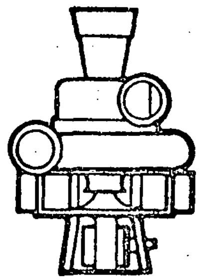
PROPOSAL TURBINE

TURBINE WHEEL	18.75
COMPRESSOR WHEEL	20.9
GEAR BOX	RIGHT ANGLE
SPEED	19,000
PRESSURE RATIO	3.0
BEARINGS	AIR



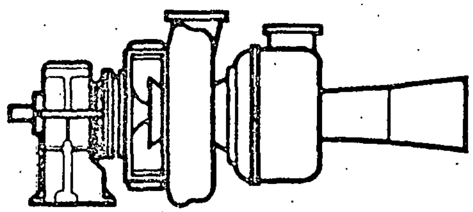
MARCH BASELINE

TURBINE WHEEL	15.0
COMPRESSOR WHEEL	13.5
GEAR BOX	RIGHT ANGLE
SPEED	—
PRESSURE RATIO	3.85
BEARINGS	AIR



JUNE BASELINE

TURBINE WHEEL	13.55
COMPRESSOR WHEEL	12.174
GEAR BOX	831-800
SPEED	30,000
PRESSURE RATIO	3.75
BEARINGS	AIR



FINAL DESIGN

Figure 2.2

demonstration unit which resulted in similar forecast costs to those independently proposed by Garrett AiResearch.

2.2 Heat Exchanger Design Considerations And Final Definition

The performance requirements of the heat exchanger are defined on the standard day design point ($60^{\circ}F$). Good engineering practice allows for an effectiveness of 0.95 with a thermal conductance ratio of 2. The resulting design point requirements are defined below.

$W_{hot\ gas}$	237.1 lbs
W_{air}	480 lbs per min
$T_{hot\ gas\ in}$	$1102^{\circ}F$
$T_{air\ in}$	$66^{\circ}F$
$T_{hot\ gas\ out}$	$112^{\circ}F$
$P_{hot\ gas\ in}$	4.30 psia
$\Delta P_{hot\ gas}$	0.19 psi
$P_{air\ in}$	15.20 psia
ΔP_{air}	4.0 inches H_2O
Effectiveness	0.955

Structural requirements for the heat exchanger must be met on a hot day when the expected hot gas inlet temperature is expected to reach $1150^{\circ}F$. Hot day expected performance is defined below.

$W_{hot\ gas}$	234.6 lbs. per min.
W_{air}	446 lbs. per min.
$T_{hot\ gas\ in}$	$1150^{\circ}F$
$T_{air\ in}$	$106^{\circ}F$
$T_{hot\ gas\ out}$	$158^{\circ}F$
$P_{hot\ gas\ in}$	4.59 psia

$\Delta P_{hot\ gas}$	0.20 psi
$P_{air\ in}$	15.16 psia
ΔP_{air}	3.70 inches H_2O
Effectiveness	0.950

The initial heat transfer design for the program was based on tests run by Alpha at Latchford Glass prior to program initiation. These tests indicated that a plate fin heat exchanger did not plug with the glass effluent passing through it. A core section from an industrial recuperator being built by Garrett was used for these tests. It was not until March of 1978 that further testing showed that plugging would occur at higher metal and gas temperatures. From the analysis of the available space that was ordinarily available in a glass plant adjacent to the furnace reversing valves, it was determined that the cube as well as the foot print of the heat exchangers, such as a shell and tube, require large volumes and high metal content due to the lower radiation characteristics at temperature of 1500^oF and below. Therefore, configurations that are not compact are more expensive and will not fit into most glass furnace applications. The design of a compact non-plugging heat exchanger became important for commercialization of the system for both cost and application considerations.

Figure 2.3 presents the family of heat exchanger types and configurations considered in the design evolution. Each of the heat exchangers were evaluated for their cost, cube, life and susceptability to plugging. The ability of the configurations to operate in a positive pressure and subatmospheric mode were also evaluated. Industry sources were then contacted to determine the probable operability of these configurations in a corrosive glass effluent environment.

None of the heat exchanger manufacturers in the glass industry had experience in building heat transfer systems to operate in the particular temperature range with the space constraints required for the Brayton cycle application. Most glass industry metallic recuperators are built for the fiber plants whose exhaust gases are at a much higher temperature. The only tubular heat exchanger built for the glass industry for the subject temperature

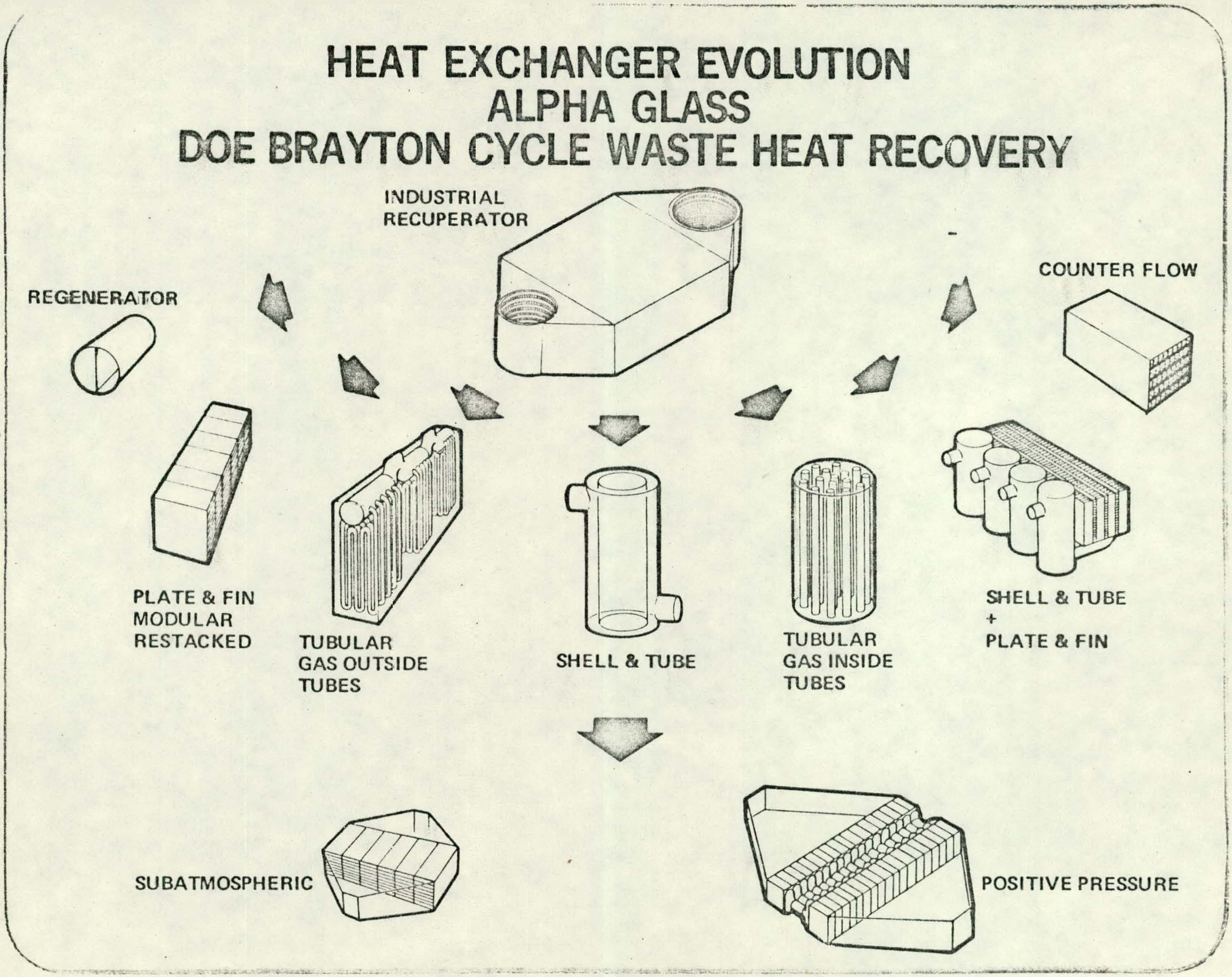

HEAT EXCHANGER EVOLUTION
ALPHA GLASS
DOE BRAYTON CYCLE WASTE HEAT RECOVERY
INDUSTRIAL RECUPERATOR
REGENERATOR
COUNTER FLOW
PLATE & FIN MODULAR RESTACKED
TUBULAR GAS OUTSIDE TUBES
SHELL & TUBE
TUBULAR GAS INSIDE TUBES
SHELL & TUBE + PLATE & FIN
SUBATMOSPHERIC
POSITIVE PRESSURE

Figure 2.3

range was used as an air cooler for an electrostatic precipitator operation, and it plugged. Cleaning systems were investigated for all forms of heat exchangers with concern to operability and cost factors.

Many methods were evaluated for heat exchanger cleaning as is shown in Figure 2.4. The two most common techniques for cleaning heat exchangers are chains and soot blowers. Chains are used for cleaning internal surfaces of shell and tube heat exchangers, and soot blowers are primarily used for cleaning external surfaces of tube and bundle type heat exchangers. Owens Illinois uses air blowers to remove condensates from tube type heat exchangers that operate at two of their glass container plants which are converting waste heat to steam.

None of these systems in their current form could be used to clean a plate and fin heat exchanger. Ultrasonics were investigated for cleaning and was discarded due to the high energy required to get any penetration into the plate and fin matrix. At the lower end of the audible spectrum (250 Hz), a Swedish horn has been used to clean a variety of configurations of heat exchangers. This horn's energy requirements were minimal, and it was felt that it could achieve penetration depths of 2 to 3 feet. Another approach which has been applied in industry is an air cannon. It has been applied to plate and fin heat exchangers, as well as other types of configuration. These cleaning devices were evaluated for their effectiveness and cost in cleaning the plate and fin heat exchanger, and the design approach selected was a single pass plate and fin cross flow heat exchanger with a high velocity air cleaning device integrated into the heat transfer.

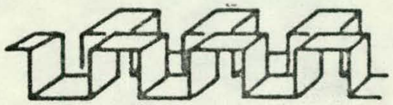
The fin configuration consists of five fins per inch, 0.145" high, offset in the flow direction 0.25". The plates and fins as costed were constructed of 0.010" thick 347 stainless steel.

The subatmospheric heat exchanger is divided into two modules, each consisting of three core sections (see Figure 2.5 for SAS configuration). These cores are 25" x 25" in the flow directions with a no-flow height of 10'. The purpose of utilizing small core sections was to provide for

CLEANING SYSTEM EVOLUTION
ALPHA GLASS
DOE BRAYTON CYCLE WASTE HEAT RECOVERY

1976 TESTS

900^{o}F TEST
NONE REQUIRED – NONE PROPOSED

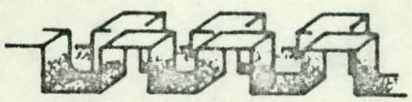


FEB 1978 TESTS

256 HRS OPERATION

MARCH 1978 TESTS

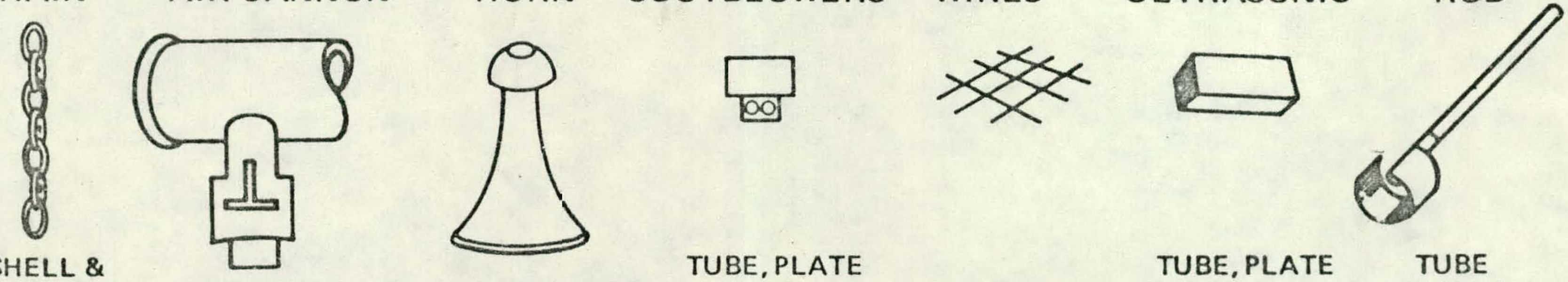
1200^{o}F TEST
MATERIAL BUILD-UP



APRIL – SEPTEMBER

HEAT EXCHANGER CLEANING

CHAIN	AIR CANNON	HORN	SOOTBLOWERS	WIRES	ULTRASONIC	ROD
SHELL & TUBE	PLATE & FIN	TUBE, PLATE & FIN	TUBE, PLATE & FIN	TUBE	TUBE, PLATE & FIN	TUBE INSIDE



ENERGY USE

LOW	LOW	LOW	MEDIUM	LOW	HIGH	LOW

PREFERRED APPROACH: HIGH VELOCITY AIR CANNON

Figure 2.4

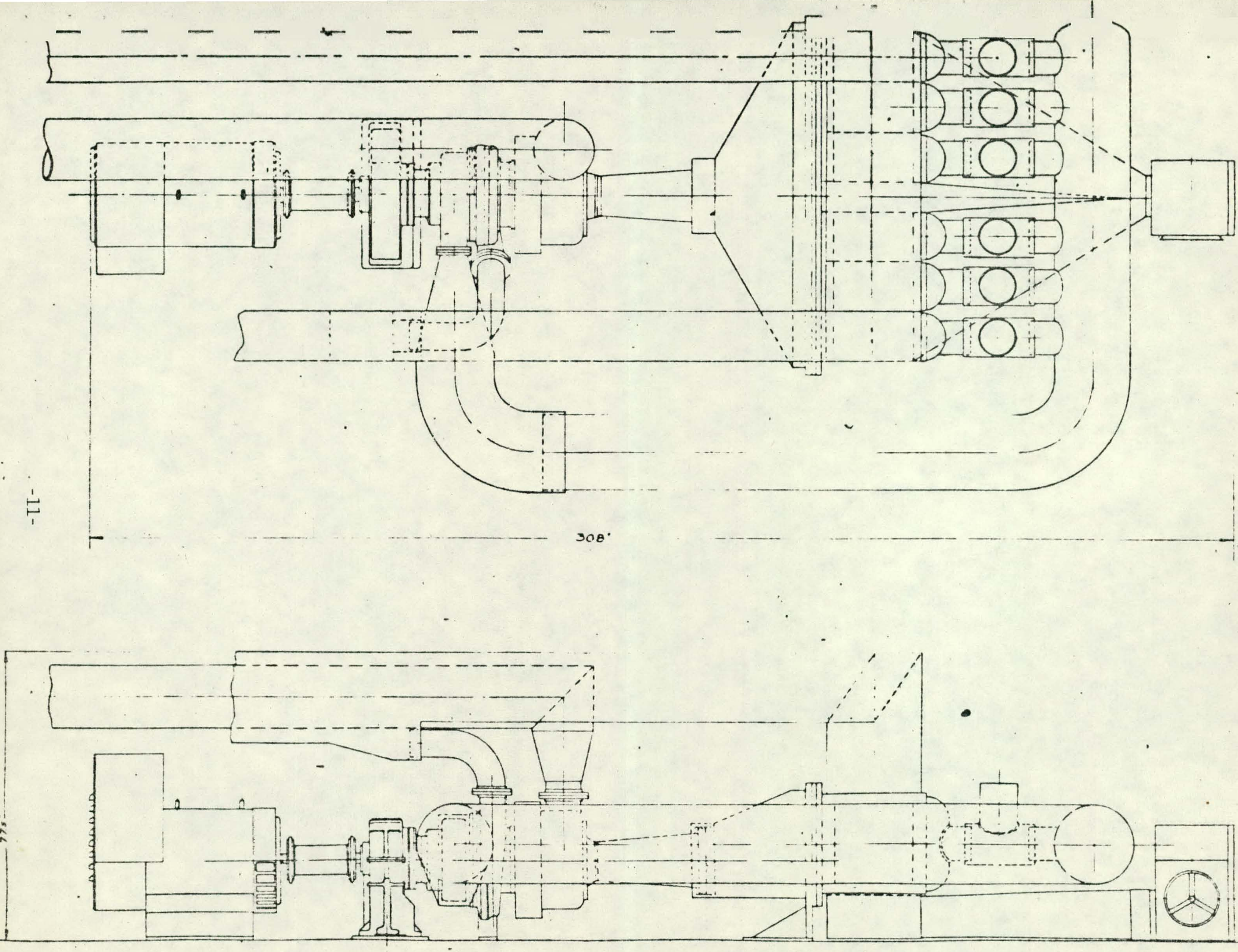


Figure 2.5 - SAS Single Unit Configuration

isolation of a small section of the total heat exchanger core for periodic cleaning. In this manner a complete subatmospheric system heat exchanger has six separate cores, each of which can be cleaned sequentially during furnace reversals. The turbine discharge flow is ducted to the full face area of the heat exchanger. At the discharge side of the heat exchanger, each core has its own individual pan, which is then manifolded into a common duct. Each of the core pans has its own individual cleaning valve. Cleaning of the individual core section is accomplished by opening the cleaning valve and allowing ambient air to flow through the individual core creating momentary high velocity stream breaks away the loosely bonded effluent particles. The actuation time of the valve is approximately 0.2 second allowing about five pounds of air to pass through the core section. In this manner, although very high velocity is obtained in the core, minimal disturbance to the turbine discharge pressure should occur.

3.0 COST FOR DEVELOPMENT SYSTEM

The hardware costs for an initial SAS system with three turbines and spares for the turbines are as follows. Projected costs include design, fabrication and test:

1st demonstration SAS turbine with gear box		$234,586
2nd demonstration SAS turbine with gear box		$234,586
3rd demonstration SAS turbine with gear box		190,222
Spares for turbine		160,522
Electric generators (3)		40,000
SAS heat exchanger		
Design	48,000	
Fab.	106,970	
Test	8,740	
		163,710
Controls		
Design	16,800	
Fab.	7,000	
Test	22,190	
		45,990
Wrap-up		
Design	54,847	
Fab.	25,296	
Test	3,475	
		83,618
Core cleaner		
Design	12,104	
Fab.	19,416	
		31,520
Component tests		73,500
System tests		39,973
Site preparation		26,624
System installation		67,922
System shakedown		17,997
Total		$1,410.770

3.1 Back-Up Pricing Data For Developmental Heat Exchanger

	Items	Quantity	Unit Price	Amount
(a)	Modules	18 (3 units x 6)	5,942	106,956
(b)	Structural steel	5000	0.51	2,550
(c)	Cleaning valves	18	664	11,952
	Tooling for valves			4,000
(d)	Pressure tank (rework)	1		2,138
(e)	15" dia connector	5	300	1,500
(f)	12" dia connector	5	275	1,375
(g)	Miscellaneous hardware			11,000
(h)	Tooling for ht exchanger			32,000

The following shows material and fabrication labor costs for each subatmospheric heat exchanger module. There are 2 modules with 3 core sections for each turbine.

Items	Quantity	Unit Price	Amount
Core	3	1,655	4,965.00
409 sheet .049	57	1.50/lb	85.50
304 tubing 12" dia	24 lbs	3.50/lb	84.00
304 tubing 6" dia	7.5 lbs	3.50/lb	26.25
Misc.	10 lbs	5.00/lb	50.00
Labor			732.00
			$5,942.75

3.2 Back-Up Pricing Data For Wrap-Up Hardware On Development System

Item	Estimate Cost
Turbine inlet duct 15" dia, 316, 10'	1,498
Compressor discharge duct, 12" dia, 304, 10'	1,428
Fan discharge duct, 36" dia, 304, 30'	4,271
Inlet plenum	4,427
Outlet plenum	3,377
Fan, 700°F exhauster, 50 hp, 21,000 ACFM	4,200
Pressure relief valve and flow sensor	1,500
Isolation valve, turbine discharge, 24"	306
Isolation valve, heat exchanger inlet, 48"	540
Isolation valve, fan discharge, 36"	320
Flue flow sensor	100
Combustion blower check valve	266
Filter	250
Control lines & fittings	400
Flex hose	200
Electronic control cables	600
Turbine mount	250

4.0 MATURE SYSTEM COST

The cost of a matured subatmospheric Brayton system was obtained for the purposes of estimating the system ROI. The costs which are shown in Figure 4-1 are in 1978 dollars and were obtained through quotations as in the case of off-the-shelf type components; analyses of predicted materials and labor combined with the learning curve effects for such components as the heat exchanger; or comparison to other similar type equipment. The components costs presented are based on production runs of 100 units per year. The markup to the base engineering costs is representative of the industry margins for similar type component units.

Figure 4-1
COST ESTIMATES FOR SUBATMOSPHERIC SYSTEM

Item	Purchase or Fab Cost	Basis of Estimate	Markup	Sell Price	Comments
Turbocompressor and Gear Box	50,000	Garrett Quote	1.6	80,000	x 3 = 240,000
Generator and Generator Control	5,000	$20/KW	1.6	8,000	x 3 = 24,000
Switch Gear	4,000	Onan Quote	1.6	6,400	19,200
Heat Exchanger 1 pass 1700 lbs.	6,800	$4/1b.	1.6	10,880	32,640
Inlet Plenum	520	260 lbs @ $2/1b	1.6	832	2,496
Outlet Plenum	520	260 lbs @ $2/1b	1.6	832	2,496
Heat Exchanger Support Structure	680	10% of heat exchanger	1.6	1,088	3,264
Modulating Valve	1,500		1.6	2,400	7,200
Cleaning System Valves	3,000		1.6	4,800	14,400
Control Unit	6,000		1.6	9,600	55,800
Blower	4,000		1.6	6,400	19,200
Installation	100,000			100,000	
Hardware					393,696