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EXAMINATION OF A MILITARY APPLICATION OF THE SMALL GAS TURBINE WITH WASTE HEAT BOILER

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EXAMINATION OF A MILITARY APPLICATION OF THE SMALL GAS TURBINE WITH WASTE HEAT BOILER

by

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ABSTRACT

The purpose of this study is the determination of the feasibility of the use of the gas turbine engine, with recovery of rejected heat through the use of a waste heat boiler, to meet the requirements of a military advanced base.

A review of the progress in the gas turbine engine field together with rejected heat utilization, the theory of gas turbine engines, the engines available in sizes below 600 HP, performance experience, and discussion of the principle features of this application are presented.

The conclusions that the gas turbine engine for use in the advanced base application is unsuitable, that full utilization of the recovered "rejected heat" is essential to economy of operation, that each particular application requires a specific examination of the requirements of the problem with respect to the equipment capabilities and limitations in lieu of a "universal" solution approach, and the potential suitability for other applications than the military advanced base, are predicated on the results of the study.

PART I.

INTRODUCTION

A. Introduction

The progress and development of the small gas turbine, and the features of high power/weight ratio, reliability, rapid starting in all weather, low maintenance, compactness and simplicity lead quite naturally into examination of suitability for application in many fields. Such examination is of particular interest when it appears that a means of heat recovery may be coupled with the gas turbine. For in this manner, one of the significant detractions, high specific fuel consumption, may be compensated. It is logical that the military application, advanced base small power generating units with waste heat recovery, be investigated to determine the feasibility of use.

Can the "total energy conversion"[1*] concept be applied in this case, while maintaining the high degree of mobility that is necessary in a military advanced base? Can the system be set up quickly, and operated so that the fuel supply required will not overburden the supply system and

*Throughout this thesis, superscript numbers refer to the similarly numbered items in PART VI, LITERATURE CITED, used in support of statements preceding the superscript numbers.

become a logistic problem? Are the loads which make use of the rejected heat such that fullest utilization is obtained?

In approaching these questions, it is necessary that a study be made of what has been accomplished, the principles and theory of the gas turbine, the equipment that is available, and the experiences in similar applications. Then with this background, the practical questions of transportation, installation, field operation, energy utilization and fuel supply can be reviewed. The practicability of application of the gas turbine with heat recovery for advanced base small power generating units will depend on satisfactory answers to these questions.

B. Historical Review

In the historical review of the small gas turbine using a waste heat boiler, it is interesting to look briefly at the general turbine history, and then the influence of other applications. Those of interest, other than aircraft, are the large industrial, marine, small auxilliary, automotive, and units operating with waste heat boilers.

Over 2,000 years ago, in 130 B.C., Hero applied the reaction of steam to drive his aeolipile. John Barber, an Englishman, is credited with the first gas turbine patent in 1791. Progress beyond this stage was not

significant from a production viewpoint until after World War II. There were a number of noteworthy contributions up to this time, some of which have been cited in a chronology in Appendix A.

There is no question that the military necessity for an aircraft engine of higher performance to keep up with those developed and produced by the Germans accelerated the success in the aircraft field in the late thirties and early forties. Aviation development, application and experience has been a major factor in the progress in the stationary power, marine, industrial and automotive fields.[20,21,32]

A comprehensive review of gas turbine advancement is published by the Gas Turbine Power Division of the American Society of Mechanical Engineers.[65,66] A further contribution to current events in the field is found in the publication GAS TURBINE which printed its first edition in January, 1960. The upward trend in interest and use is indicated in Table 1, a comparative summary of units manufactured in 1961 and 1962.

Table 1. Comparison of Models, Manufacturers, and Countries 1961-62[57]

Item	1961	1962
Models	274	319
Manufacturers	46	51
Countries	9	10

The progress in large gas turbines which has been made in recent years is encouraging. The departure from experimentation to a status of proven components of industry is a matter of record. Selection on the basis of comparative analysis with other methods has placed large gas turbine units in operation for pumping, processes and power generation. Little was reported in 1958 on small units for industrial use. Poor fuel economy in comparison with the diesel was the retarding factor, unless the positive characteristics of light weight, compactness and high speed overcame this consideration. About one thousand units were in service, primarily in a military application.[41]

Marine requirements for lighter more compact propulsion and auxiliary drives have inspired a very productive period in the gas turbine history. A current marine bibliography which is of interest to marine engineers and naval architects and of value to general research is available.[36] The marine application, development and progress for generators for both base load and emergency has done much for the advancement of the small gas turbine.[30,34] In the marine application there has been a steady reduction of initial cost.[54] All of the marine generator operating experience has not been as successful as expected. However, the

difficulties encountered have primarily been due to compressor fouling. This in turn has resulted from installation arrangement which was restrictive aboard ship. No conclusions have been drawn pending further operational experience.[43]

In small gas turbines, a good comparison from data available late in 1958 is shown in Table 2.

Table 2. Comparison of Gas Turbines, Diesels, and Gasoline Engines[48]

Manufacturer	Name	Hp.	Lb/hp.	Cost/hp.	Fuel Consumption per hp. hour
GAS TURBINES					
Allison........	GMT-305	225	2.65		.55
AiResearch Mfg. Co..............		30-1000	.40	$18,000 to $25,000*	.65
Boeing.........	502-10C	240	1.36	$62.5	.7-1.0
General Elec...	T-58	800	.34		.69
Clark Bros.....	Mark TA	1220	10.	$80-$110	.91
Continental....	TC 106	160 (Air hp.)			
Lycoming.......	T-53	700	.69	$30	.75
Solar..........	Jupiter	500	1.1	$100	.95
....	Saturn	1000	.75	$50	.65
				*Total Price	
THEIR CHIEF RIVALS					
Chrysler Marine V-8 (Gasoline)...	Imperial Special	225	4.7	$11.10	.61
Aircraft Type (Gasoline)...	Allison V-12	1250	.995		.52
Diesels........	....	20-500	6-10	$16-$30	.4-.55

Information obtained currently from manufacturers has not been in disagreement with that shown in Table 2. Selection for purposes where fuel consumption is not the most important consideration, such as in a mine sweeper specialized power system continues,[8] while the efforts to reduce fuel consumption are increased[55,58] in order that advantage might be gained of other benefits of gas turbines. There have been a large number of contracts for manufacture of small standby or intermittent-use type units for ground support of aircraft, boat propulsion, compressors, pumps, aircraft auxiliaries, and some vehicle propulsion.[59]

The developmental efforts of the automotive industry are most interesting and have great potential effect. Since the report in this field in 1958,[38] the General Motors GMT 305[14,67] has been placed in operation in an ore truck[39] and the Chrysler engine[61,67] has made its second transcontinental trip.[4] These engines and the Ford model[45,67] use combinations of regenerators, recuperators, intercoolers, free power turbines, and high and low pressure compressors to obtain the greatest possible heat recovery, lowest possible specific fuel consumption (SFC, lbs/bhp-hr), and most favorable torque-speed relationship. The attainment of a low initial cost, operationally economical engine,

competitive with diesel and gasoline reciprocating units, is dependent upon materials and manufacturing research and mass production techniques that are available in the automotive industry. It is apparent that a great future for the gas turbine will open up should the automotive industry make the step from research to mass production of the small gas turbine engine.[48]

The idea of waste heat recovery using a boiler is not new.[16,31] The benefit[28] and the features[3] of this means of economy with the gas turbine operation are described in current articles. The development of the concept has been stimulated by the gas industry of the United States. The Southern Gas Association sponsored a seminar on industrial and commercial use of gas turbines in Houston, Texas, 15-16 June 1961.[52] The "total" energy conversion approach to air conditioning has been shown to be economically favorable for gas turbine driven compressors with either steam absorption or steam turbine driven compressors in a hypothetical 1800 ton installation, when compared with all electric-driven centrifugal compressors, condensing steam turbine driven centrifugal compressors, and steam turbine-centrifugal compressor (non-condensing) and absorption system operation on 15 psig back pressure steam.[1] The

extension of this to the "systems" concept for lighting, heating, cooling and power[37,56] has been put into operation in the Park Plaza Shopping Center, Little Rock, Arkansas,[10,19] the first in the United States. Another installation of this "systems" concept, but utilizing high frequency generation for lighting, is in an office building which will be occupied in July of 1962 at the Northern Illionis Gas Company offices, Glen Ellyn, Illinois.[60,64]

The Northern Illionis Gas Company will make a second gas driven "energy package" installation in their general offices near Aurora in 1963. This is a further development of the Glen Ellyn system in that it consists of four complete modules. Each has its own gas turbine, 420 cycle generator, boiler, and 60 cycle converter. The system will be fully automatic. The gas company's objective: to provide the customer complete utility service while the fuel savings pay off equipment costs in a period of approximately four and a half years.[17]

C. Statement of the Problem

The Bureau of Yards and Docks at the present time uses internal combustion engines (generally diesel) as prime movers for small electric power generators. These

units are for use in advanced bases, i.e., naval bases set up ashore in or near combat areas for support of fleet operations. The mission may be that of a naval operating base, ship repair base, supply base, air base, ammunition storage depot, communications station, construction base, etc. The nature of the installations is temporary and the utility requirements varied. Power, 60 cycle 3 phase AC, is required. Direct current may be required. Heat for comfort is generally provided by small oil burning space heaters. Refrigeration is supplied by either electric motor or internal combustion engine driven compressors. Water supply if not available from other reliable sources must be distilled from sea or brackish water. Present units are diesel operated. Air conditioning has not been an advanced base requirement. However, developments in communications equipment and operations procedures, and air conditioned aviation flight clothing indicate the probability of the requirement in future advanced bases. The present method is the use of the foregoing components as may be required in the specific advanced base installation, such requirements being determined in type and number by the mission of the base.

The question under study is whether a small gas turbine utilizing a waste heat boiler can be successfully applied to meet the requirements of power, water, refrigeration, air conditioning, heat, and process steam, or any combination thereof.

A gas turbine approximately, 150 HP, as prime mover for a 60-75 Kw 3 phase, 60 cycle, AC generator, will be the basis for the problem study. A waste heat boiler is required capable of producing steam for provision of 200 gallons per hour of potable water, 10 tons of refrigeration (food storage, part of which is assumed to require a temperature of approximately $0^{o}F$, the remainder $35\text{-}40^{o}F$), 5-10 tons of air conditioning, and heating and process steam the requirements for which are not predetermined. Direct current power requirements were not established for this study and are assumed to be not in excess of 20-25 Kw. An additional examination of the problem is desired using a gas turbine of 300 HP with the same loads as above.

The degree of success of the application must be measured by the availability of suitably sized equipment with a satisfactory operating record, ability to provide an arrangement with a minimum installation in order that

mobility may be retained, uniformity and sufficiency (that is demand consistent with the supply) of the requirement for the steam supplied from the recovered "rejected heat", and finally, by favorable comparison of the fuel required to support the gas turbine combination with that to support a comparable diesel engine driven (and fired) arrangement.

D. Method of Procedure

The procedure followed in the development of this examination was a review of the small gas turbine and waste heat boiler field together with related subjects; correspondence with industry to determine what equipment was available; correlation of the requirements of the problem with the available means to solve the problem; selection of one of the units and the necessary auxillaries for a typical analysis; review of the information and results obtained; and conclusions on the basis of the findings.

The review included study of the history and theory of gas turbines and waste heat boilers, applications, operational experience, and developmental efforts and their meaning to the future. Related subjects were free piston engines, combined cycle operation, large gas turbines, etc. Manufacturers contacted are listed in Appendices B through D.

PART II.

THEORY

A. Principles of Gas Turbine Operation

An understanding of the principles of operation of the gas turbine is necessary for the proper study of application problems. No effort will be made to present a comprehensive reproduction of the theory as this information is readily available in a number of references. A particularly good reference specifically addressed to the small gas turbine field has been written by Arthur W. Judge.[20] The material presented is related to the performance capabilities and limitations, and equipment characteristics in order that the available equipment may be analyzed with regard to the proposed application.

The gas turbine consists in its simplest form of a compressor, combustion chamber, and a turbine to drive the compressor and for power output. Figure I shows the simple gas-turbine cycle.

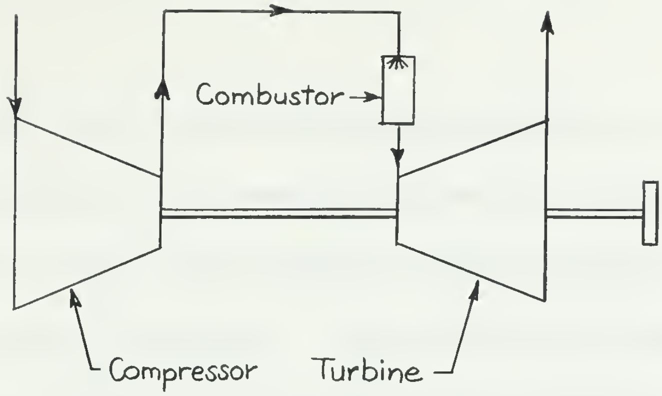

Figure I. The Simple Gas Turbine

Typical temperatures and pressures in the cycle are, for example, air at atmospheric pressure and 60°F to 80°F, raised to 300°F to 400°F with a pressure ratio of 3 or 4 to 1 in the compressor, 1550°F entering the turbine, and 1150°F leaving for a single stage turbine, exhausting to atmosphere from duct exits at approximately 850°F to 1000°F.

Gas turbines operate on an open cycle or a closed cycle. In the former case the working medium is air to which the products of combustion are added and then the total rejected. In the latter, the working medium, a separate fluid, is enclosed and recirculated. An example of the closed cycle gas turbine is the U. S. Army ML-1 Mobile Nuclear Power Plant.[7,42,44] This cycle will not be discussed

as it has not been considered in the examination of the problem.

Turbines used may be impulse, reaction or combination using both impulse and reaction elements. Turbines commonly used are axial flow although radial inflow is used in some of the smaller designs. Compressors are more generally of the centrifugal type in the smaller gas turbines. Cost, durability and less possibility of damage from foreign matter in the air supply influence this selection. However, the axial flow compressor is more efficient.

The need for better torque speed characteristics than that of the simple gas turbine and for improved control, in particular in traction applications, has resulted in the addition of a free power turbine.

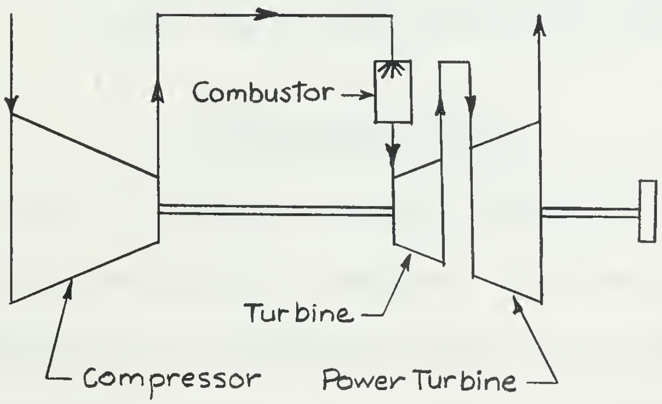

Figure II. The Free Turbine Engine

The compressor-turbine unit can operate at its best performance speed while the power turbine operates at its speed, starting loads are not as great, acceleration to operating speed is faster, and at constant compressor speed, the power turbine can be operated over a wide range. Figure III is a comparison of the torque-speed curves for single shaft and free turbine.

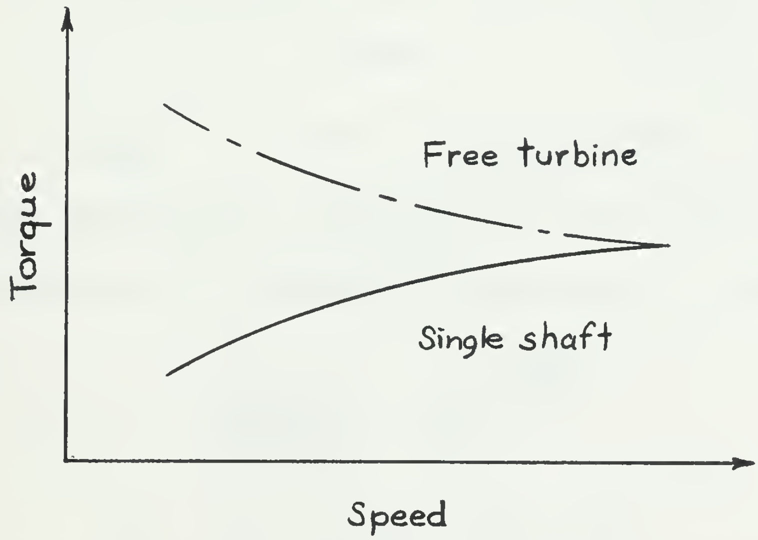

Figure III. Comparison of Torque-Speed Curves for Single Shaft and Free Turbine Engines

B. Theoretical Cycle

The Joule or Brayton cycle, Figure IV, is the basic cycle of the gas turbine engine. It consists of adiabatic compression of atmospheric air, 1-2, heat addition at constant pressure, 2-3, and expansion adiabatically to atmospheric pressure, 3-4.

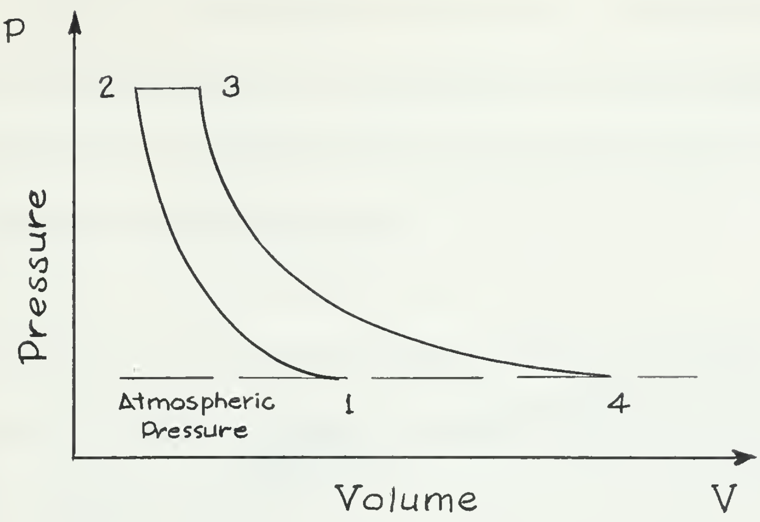

Figure IV. Joule or Brayton Cycle

The theoretical efficiency of this cycle, assuming 100% efficiency of turbine and compressor, is given by the equation

$$T.E. = 1 - \left(\frac{T_4 - T_1}{T_3 - T_2}\right) = 1 - \left(\frac{1}{R}\right)^{\frac{(k-1)}{k}}$$

T.E. = Theoretical Efficiency

T_n = Temperature at the corresponding point on Figure IV

R = Compression or pressure ratio

k = $\frac{C_p}{C_v}$ ratio of specific heats

This equation is developed in Appendix E.

If the medium is air $k = \frac{0.241}{0.1725} = 1.396$ and

$$T.E. = 1 - \left(\frac{1}{R}\right)^{0.2846}$$

Figure V shows the variation of theoretical efficiency and specific fuel consumption with pressure ratio. The specific fuel consumption curve applies only to the ideal constant pressure cycle.

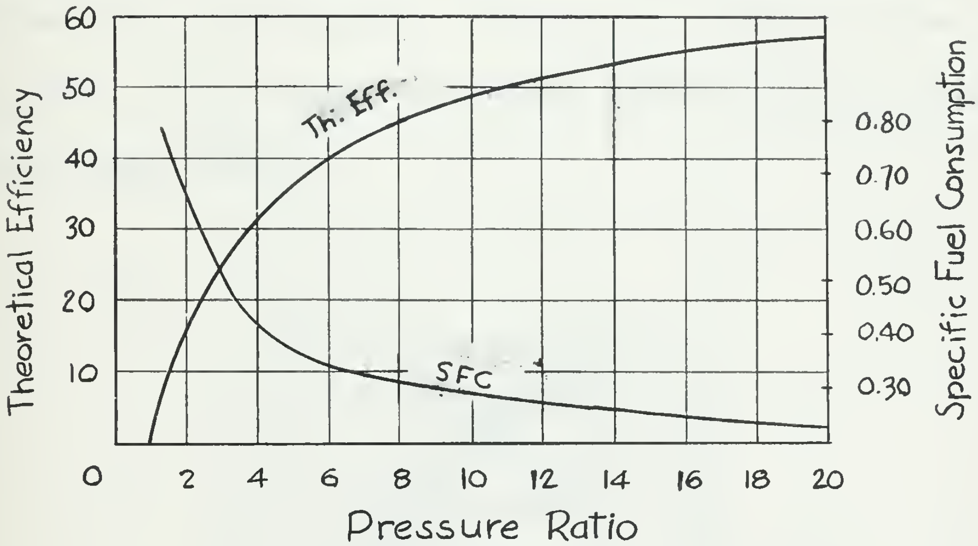

Figure V. Theoretical Efficiency and Pressure Ratio[20]

An examination of the equation $T.E. = 1 - \left(\frac{T_4 - T_1}{T_3 - T_2}\right)$ shows the effect on efficiency of variation of turbine inlet temperature and the benefit of a low exhaust temperature.

C. Actual Efficiency

The difference in actual and ideal efficiencies results from: a departure from the assumed isentropic compression and expansion; variation in specific heat during

the cycle; friction losses; loss of energy in the gases during flow through the turbine and compressor; and losses in the combustion chamber. The actual efficiency is dependent on compressor, turbine and combustion chamber efficiencies. The influence on over-all efficiency of the compressor and turbine shaft efficiencies and pressure ratio is shown in Figure VI. Turbine and compressor efficiencies are equal in the cases shown.

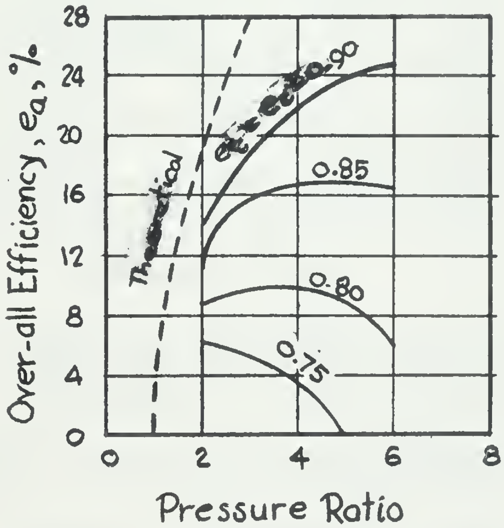

Figure VI. Effect of Pressure Ratio and Turbine and Compressor Efficiency on Over-all Efficiency[21]

The fact that in the reciprocating engine, the combustion, compression and power output all occur in the cylinder, without transfer of the medium is significant in

explanation of the higher thermal efficiency of this type engine than that of the gas turbine.

The influence of turbine inlet temperature and pressure ratio is shown in Figure VII for turbine efficiency = 0.85, compressor efficiency = 0.84.

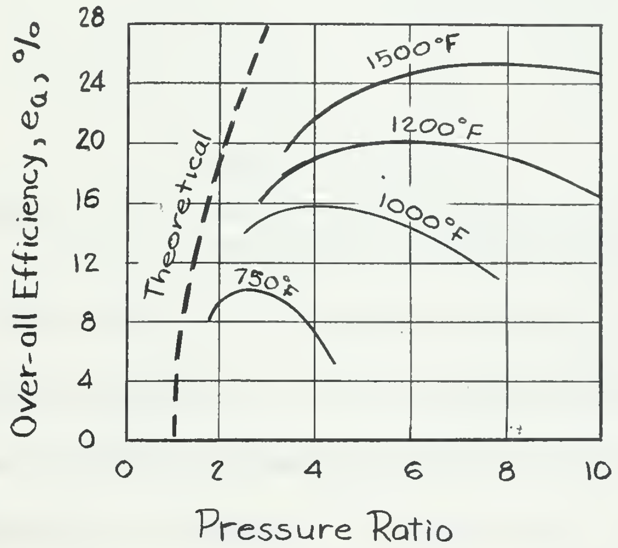

Figure VII. Effect of Pressure Ratio and Turbine Inlet Temperature on Over-all Efficiency[21]

Operation at variation of ambient inlet temperature of the compressor effects the over-all efficiency and output of the gas turbine through the changes in the air density. At higher ambient temperatures lower density air moves through the turbine lowering the efficiency. At lower ambient temperatures the air density is increased and the efficiency is increased. The effect of ambient air temperature on thermal efficiency and turbine output is shown in Figure VIII.

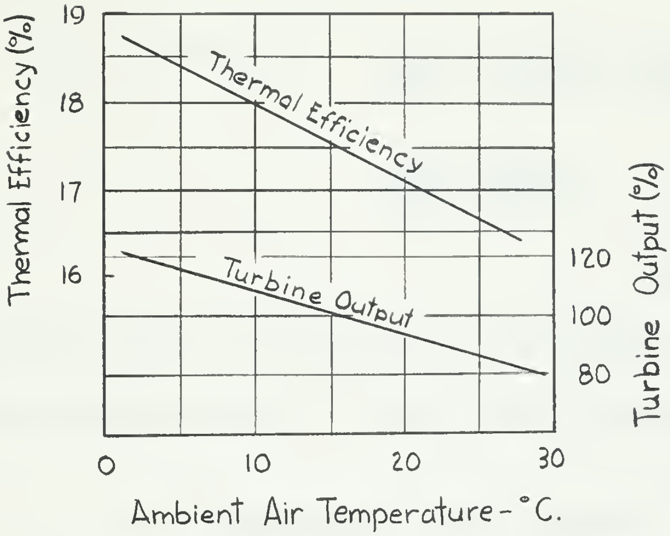

Figure VIII. Effect of Ambient Air Temperature on Output and Efficiency[20]

The effect on efficiency of part load operations is indicated in Figure IX. This curve is based on simple cycle engine of 2700 HP maximum output with a 1000°F turbine inlet temperature. However, it shows the general relationship between part load and efficiency.

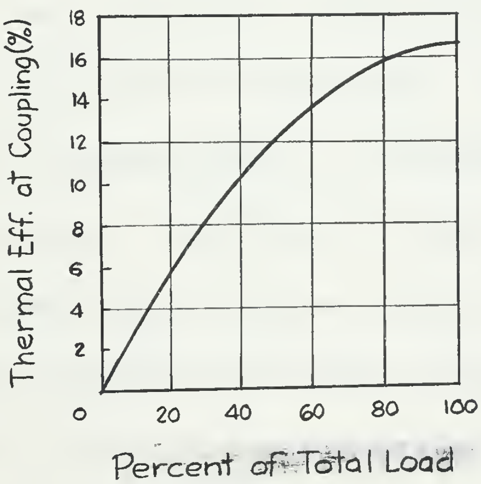

Figure IX. Thermal Efficiency of a Simple Gas Turbine at Various Loads[20]

A practical measure of the over-all cycle efficiency may be obtained by using the equation

$$\text{Cycle Efficiency} = \frac{\text{bhp} \times 33{,}000 \times 60}{\text{Heat input} \times 778}$$

$$= \frac{\text{bhp} \times 2545}{\text{Heat input}}$$

where heat input is in BTU/hr and is obtained by the product of the pounds of fuel consumed per hour and the heating value of the fuel.

D. Performance Improvement

Improvement of performance through higher turbine inlet temperatures, improvement in the component efficiencies, and increase in the pressure ratio is limited by materials, manufacturing technique, costs, and, by the design, when considering available equipment.

Other means which are successful but which add weight, cost, and complexity, reducing some of the advantages of the simple cycle will be reviewed. These are methods of recovery of the rejected heat. The methods can be divided into two categories. The first is that in which the energy recovered is used within the gas turbine engine cycle. The second is that in which the energy recovered is used to produce steam, hot water or air for use external to the gas turbine cycle.

The means in the first group are regeneration (recuperation); regeneration and intercooling; regeneration and reheating; and regeneration, intercooling and reheating. Table 3 is a comparison of the different cycle arrangements. The data in Table 3 are based on a constant net output (useful output = 1.0). They are compared graphically in Figure X. The cycles are shown diagramatically in Figures XI through XIV.

Table 3. Effect of Different Open Cycle Arrangements on Each Major Element[21]

Item	1*	2*	3*	4*	5*
Input in fuel	4.95	3.75	3.43	3.55	3.11
Turbine rating	3.95	2.95	2.80	2.88	2.55
Compressor power	2.95	1.95	1.80	1.88	1.55
Useful output	1.00	1.00	1.00	1.00	1.00
Efficiency at 1200°F,%	20.2	26.6	29.2	28.1	32.2
Gas Temperature, °F at:					
Turbine inlet	1,200	1,200	1,200	1,200	1,200
Leaving reheater	-	-	-	1,200	1,200
Turbine exhaust	635	790	695	920	865
Leaving Regenerator	-	455	350	560	520
Air Temperature, °F at:					
Compressor inlet	70	70	70	70	70
Leaving intercooler	-	-	70	-	70
Leaving compressor	490	340	230	440	405
Entering combustor	490	680	575	800	750
Pressure, psia:					
Compressor inlet	14.7	14.7	14.7	14.7	14.7
Compressor discharge	88.2	51.5	73.5	73.5	102.9
Turbine inlet	88.2	50.2	71.7	71.7	100.4
Turbine exhaust	14.7	15.1	15.1	15.1	15.1

*1 Simple cycle
*2 Simple cycle with regeneration
*3 Simple cycle with regeneration and intercooling
*4 Simple cycle with regeneration and reheating
*5 Simple cycle with regeneration, intercooling, and reheating

Tabulation based on:
Turbine Efficiency 85%
Compressor Efficiency 84%
Combustion Efficiency 100%
Ambient air temperature 70°F
Regenerator pressure drop 5%

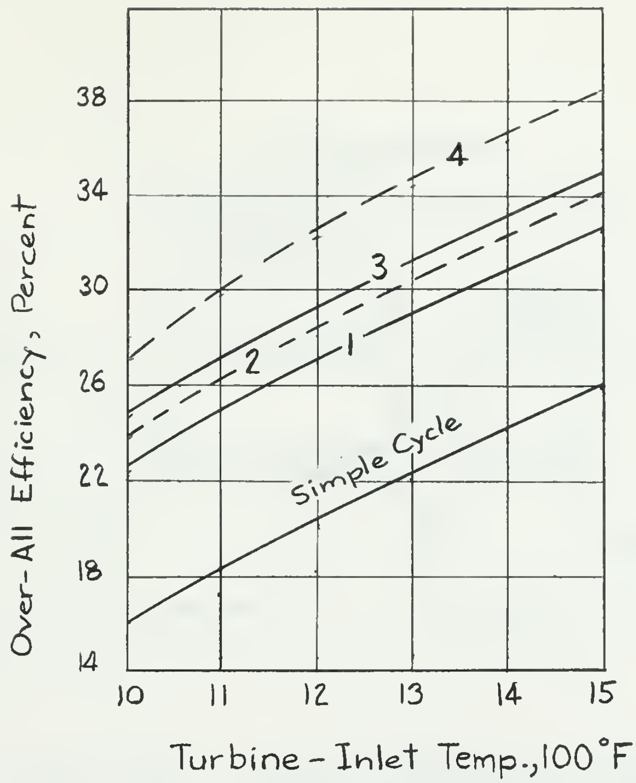

Figure X. Effect of (1) regeneration, (2) reheat and regeneration, (3) intercooling and regeneration, and (4) reheat, intercooling and regeneration upon over-all efficiency[21]

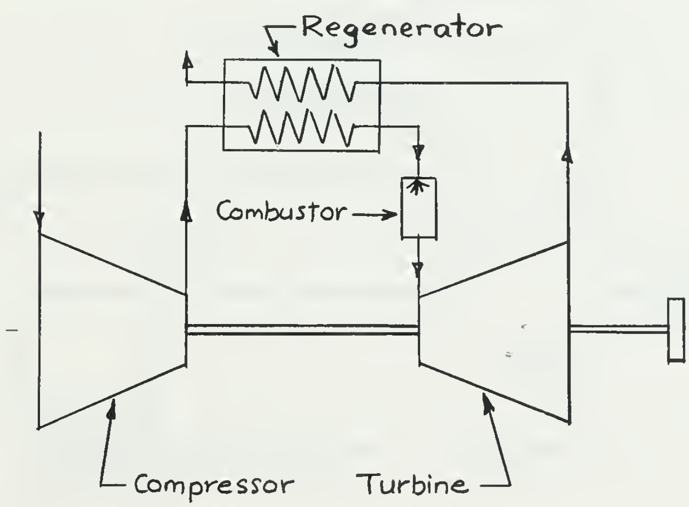

Figure XI. Simple Cycle with Regeneration

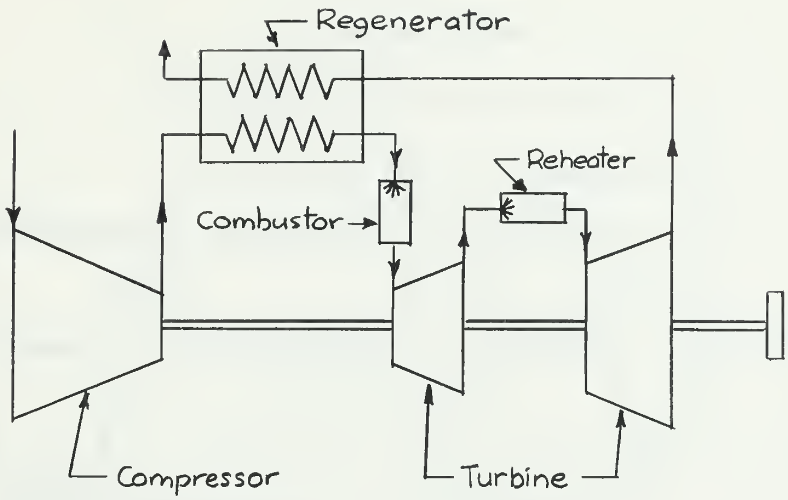

Figure XII. Simple Cycle with Reheat and Regeneration

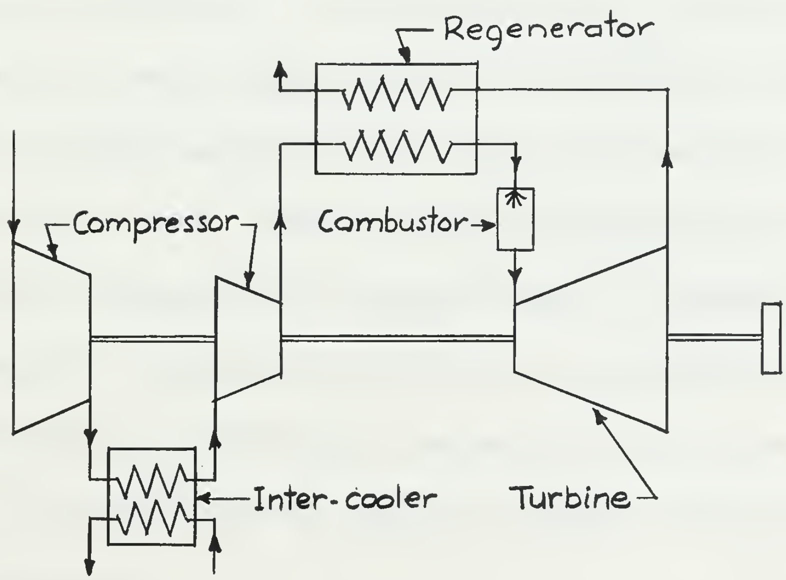

Figure XIII. Simple Cycle with Inter-cooling and Regeneration

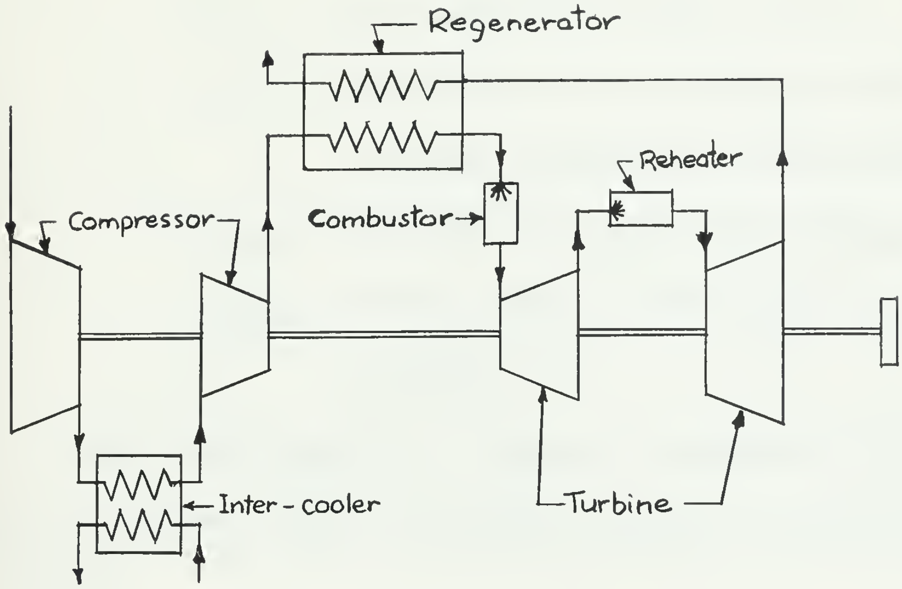

Figure XIV. Simple Cycle with Reheat, Inter-cooling and Regeneration

In the energy recovery for use external to the gas turbine cycle, the effect on efficiency, or "utilization" of rejected heat has been approached in the following manner. For each gas turbine manufactured, data are available on the air mass flow rate, turbine exhaust temperature, and specific fuel consumption at varying loads. Assuming that the heat transfer equipment available on the market is capable of reducing the exhaust temperature to values in the order of 350°F to 550°F, the heat energy available may be calculated from

$$Q = mc_p\ (T_{gte} - T_{whbe})$$

Q = BTU/hr

m = air mass flow rate, lbs/hr

c_p = Specific heat of exhaust gases at average temperature and pressure

T_{gte} = Temperature of turbine exhaust gases

T_{whbe} = Temperature of waste heat boiler exhaust

The waste heat boiler exhaust temperature should be maintained above $320^{\circ}F$ to prevent moisture condensation in the stack.

This approach may be extended to obtain the required heat recovery boiler inlet temperature, the heat to be added by supplemental firing, and the additional fuel rate to meet a given process or heating load.[2]

In gas turbine operation, the rate of fuel consumption and the type of fuel will determine the air required for combustion, while the characteristics of the equipment set the amount of air supplied. From these quantities the excess air may be calculated. A typical calculation of theoretical air required for diesel fuel, and calculation of excess air percentage for a gas turbine engine are found in Appendix F. The amount of remaining

oxygen in the turbine exhaust will place an upper limit on the extent to which supplemental firing can be used. An excess air of 10-15% is generally the minimum.

The utilization of heat recovered may than be evaluated from

$$\% \text{ Utilization} = \frac{2545 \times \text{bhp} + \text{heat recovered}}{\text{Total heat input}}$$

where the heat recovered is the total recovered from the gas turbine exhaust and the heat added by supplemental firing, and the total heat input is the fuel consumed by the gas turbine times its heating value plus the fuel consumed in supplemental firing times its heating value.[2,33]

In addition to the limitation on supplemental firing of maintenance of the proper amount of excess air, and the dependence on the availability of waste heat boiler equipment which will give a favorable exit temperature for recovery of rejected heat, there are other factors which influence utilization. Gas turbine equipment is subject to power loss as a result of increase in back pressure. Allowance should be made on the shaft power available due to the back pressure loss attributable to the waste heat boiler. Generally, heat recovery equipment will cause a back pressure of approximately 4" H_2O. When determining the energy

which will be available through rejected heat recovery, consideration must be given to the conditions of part load gas turbine operation, and variation in entering ambient air temperature.

PART III.

FINDINGS

The data obtained on equipment are from manufacturers, journals, technical articles, and texts. They are of a general nature and due consideration has been given to this limitation in their use in the study of the problem.

A. Gas Turbine Equipment

The problem hypothesis is based on a unit to supply 60-75 Kw. This places the horsepower required in the 100 to 150 HP range. The restriction to size of this order does not permit a reasonable study and therefore data have been collected on units in the 100-600 HP range. From the 1962 GAS TURBINE SPECIFICATIONS[57] a list, Appendix G, of eighteen manufacturers of approximately thirty-three models in the 100-600 HP class was extracted. Of the eleven manufacturers who were contacted on this particular application, listed in Appendix B, Clark, General Electric, Westinghouse, and Brown-Boveri do not manufacture units in the size range of interest. Data obtained and tabulated in Table 4 are from the manufacturers except where otherwise noted.

Table 4. Small Gas Turbine Performance Data

Model	Rating	Exhaust Temp ^{o}F	Air flow lbs/sec	SFC lbs/hp-hr	Reduced RPM	Wt. lbs.	Cost	Shafts
Production:								
Vector	100	1200	2.4	1.5FL .91NL	3600	125	$6,000	1
Ruston TE	430	950	10.3	1.25FL 1.5 75% 2.05 50%	1200 1500 1800	3700	$40,000	1
Boeing 502-21	200	1075	3.69	1.14FL	--	335	$14,000	2
Solar T350	370	980	6.4	.9FL	--	195	--	1
Allison T63-A-5	212	--	--	.74	6019	138	--	2
GMT 305	225	300-500	--	.55	3400	760	--	2
Austin[50]	250	--	--	1.06	--	900	$7,500	1
Develop-mental:								
Ford 704[45]	300	740	2.7	.56FL .48 50% .58 25%	4600	650	--	2
Chrysler[67]	140	--	2.0	.46FL	--	450	--	2
AiResearch 331-50	350	--	--	.695	--	--	--	2

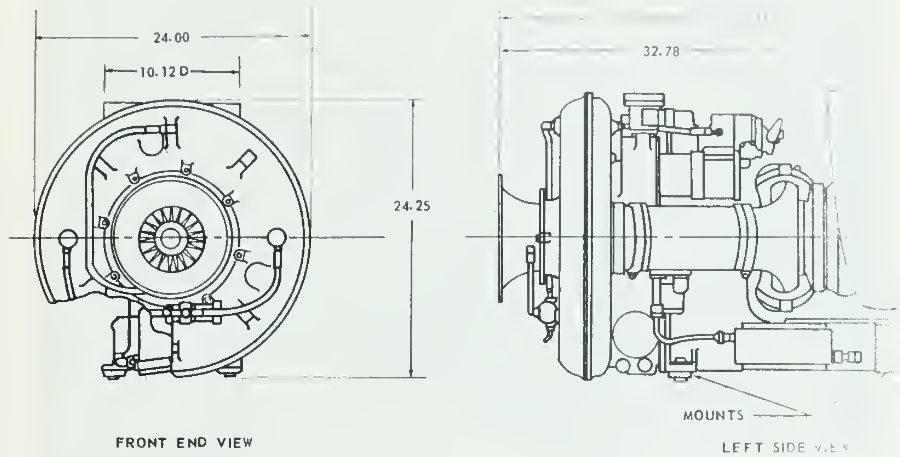

Figure XV. Boeing Model 502-21

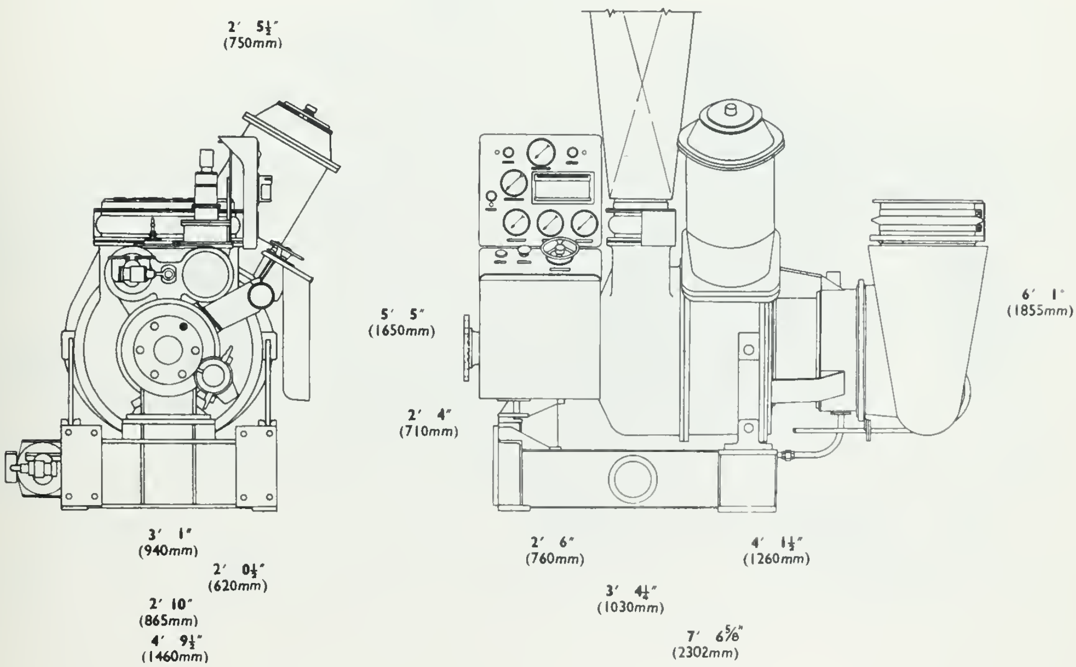

Figure XVI. Ruston TE

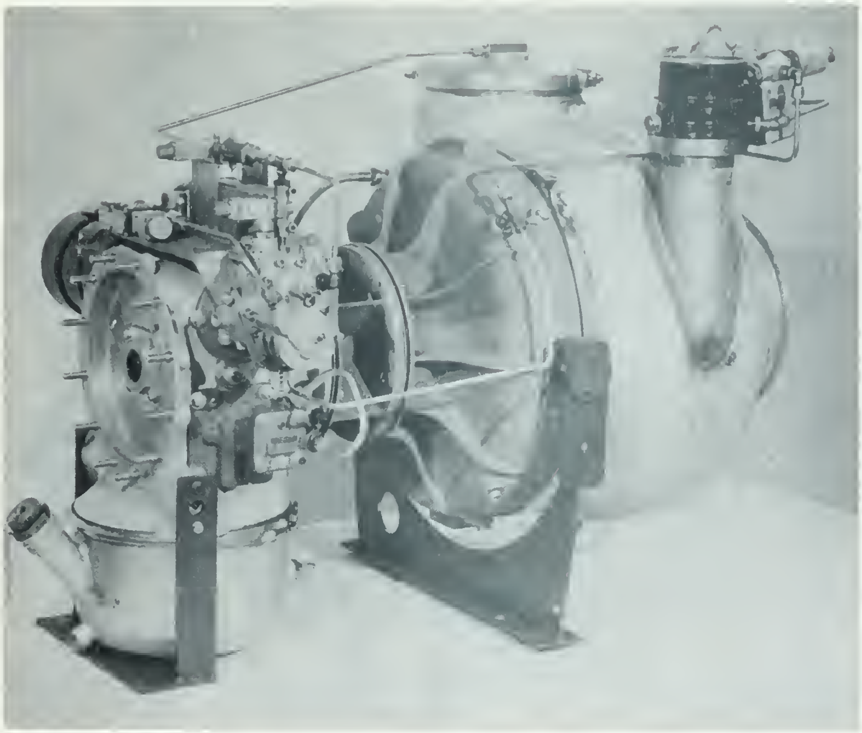

Figure XVII. Solar T-350

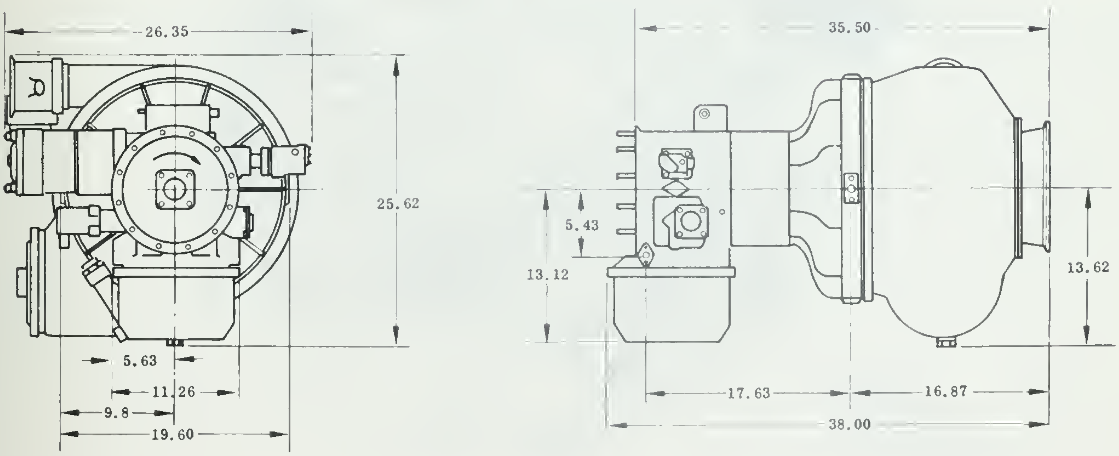

Figure XVIII. Solar T-350 Dimensions

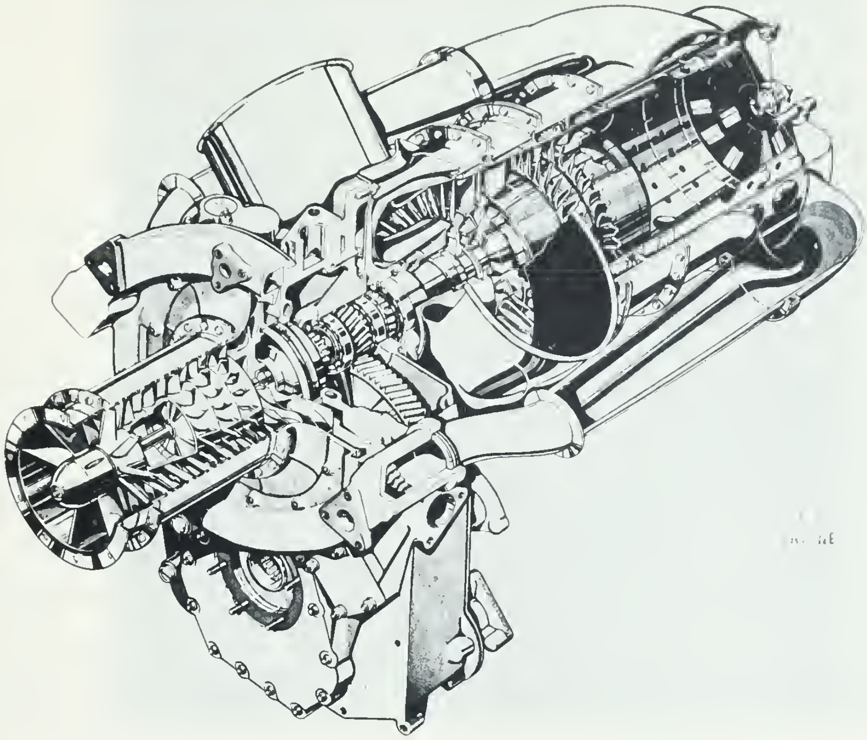

Figure XIX. Allison T63-A-5

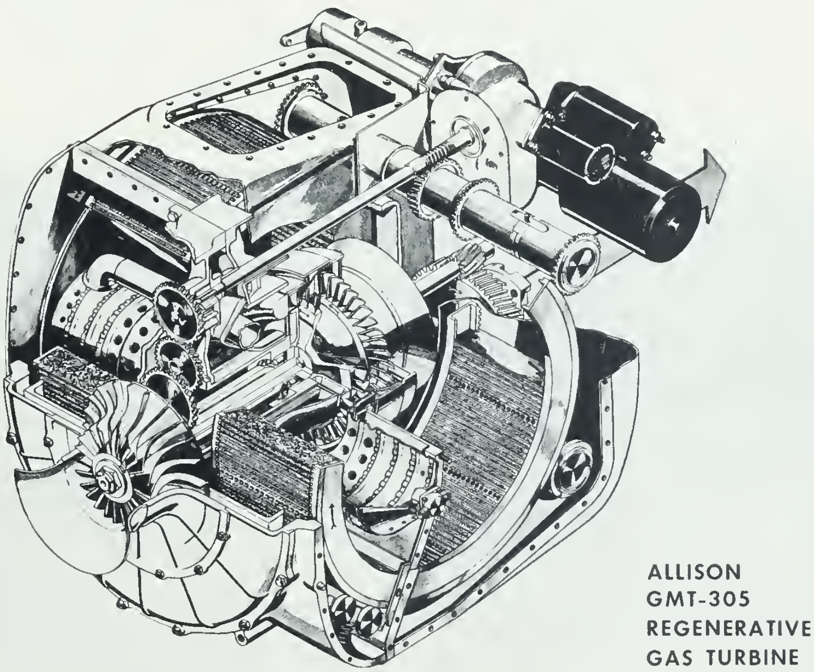

Figure XX. GMT-305

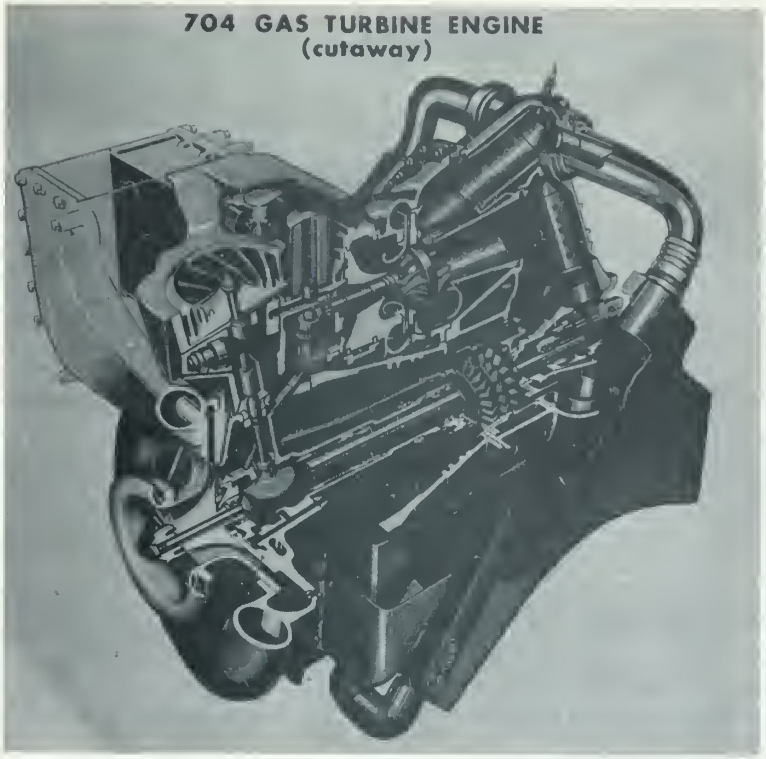

Figure XXI. Ford 704

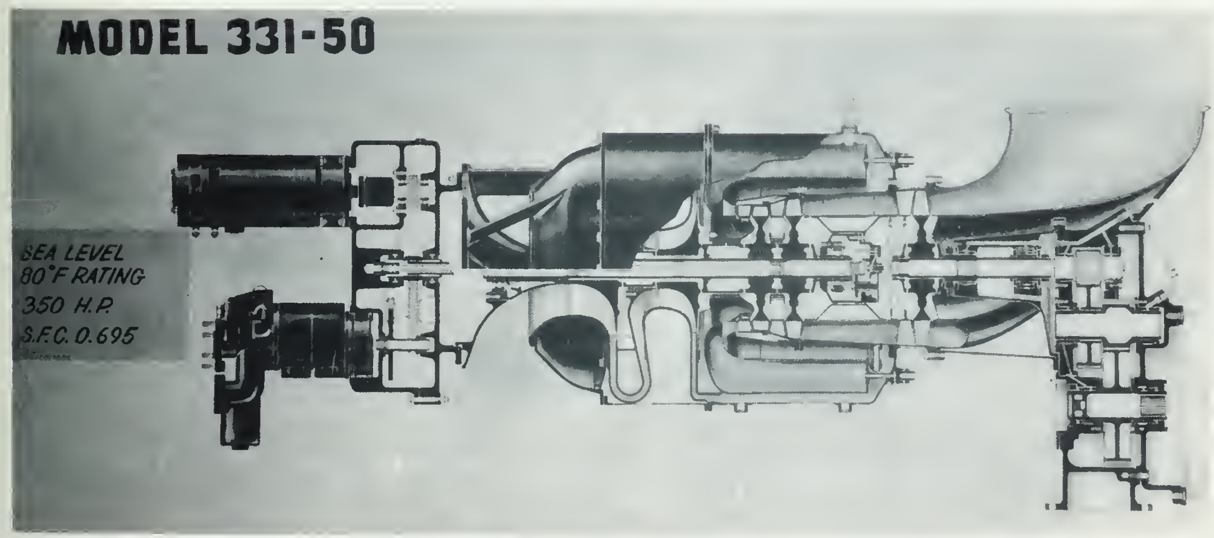

Figure XXII. AiResearch 331-50

B. Absorption Refrigeration and Air Conditioning Equipment

Inquiries were sent to five manufacturers, listed in Appendix C. Of these, the Frigidaire Division of General Motors does not manufacture absorption equipment, and the others do not manufacture steam operated units in the 5-10 ton range. The lowest size manufactured by each company is shown in Table 5.

Table 5. Minimum Available Sizes of Absorption Air Conditioning Equipment

Manufacturer	Size Tons	Minimum Recommended Chilled Water Temp.
Carrier	5	44^{o}F (Nominal rating)
York	50	40^{o}F
Arkla	25	45^{o}F
Trane	100	41^{o}F

The application of absorption equipment is limited to loads in excess of 25 tons for air conditioning and process refrigeration where chilled water temperature not less than 40-45^{o}F is acceptable.

C. Evaporator Equipment

Information was requested from twenty manufacturers, obtained from the THOMAS REGISTER under listing of vapor

compression, flash, single and multiple effect evaporators. Of thirteen replies, nine companies indicated that they fabricated on a custom basis and had no information, fabricated only components for other manufacturers, or were not in this line. One of the companies, Mechanical Equipment Company, Inc., manufactures only self-contained diesel powered units.

The Aqua-Chem 26 stage flash 100,000 gal/day unit in operation at the Southern California Edison Company's Mandalay Station, Oxnard, California, cost $250,000 installed, or $2.50/gal/day. The experimental nature of the installation accounts for the high cost. Current information obtained from engineers working in this field indicates that the cost of flash distillation equipment alone is in the order of 80¢ per gallon per day of installed capacity. This figure has been used for the flash equipment cost in Table 6. In addition, data on the MECO unit (self-contained, diesel powered) and the developmental[13] Aqua-Chem diesel heat recovery unit are included in Table 6 for comparison.

Table 6. Comparison of Typical Available Evaporator Equipment

Manufacturer	Rate gpd	Lbs. Distillate per 1000 BTU	Wt. lbs.	Cost*
Maxim M 120C	12,000	5.0	1,995	$14,650
M 80C	8,000	5.0	1,665	$13,225
Aqua-Chem				
Flash-2 stage	13,000	1.58	--	$10,400
Vapor Compression				
S 600D	14,400	16	21,500	--
S 200D	4,800	16	9,100	--
MECO	10,000	15-18	13,500	$27,500
	4,800	15-18	8,400	$15,980
Aqua-Chem				
Diesel heat recovery	4,000	--	7,600	$20,000

*Approximate

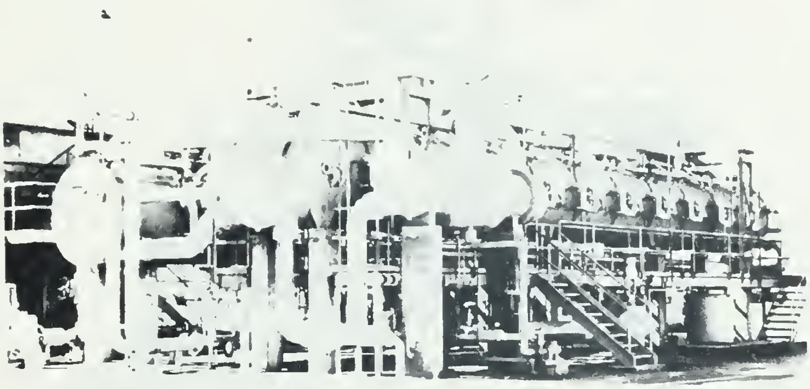

Figure XXIII. Aqua-Chem 26 Stage, 100,000 gpd Pilot Flash Evaporator

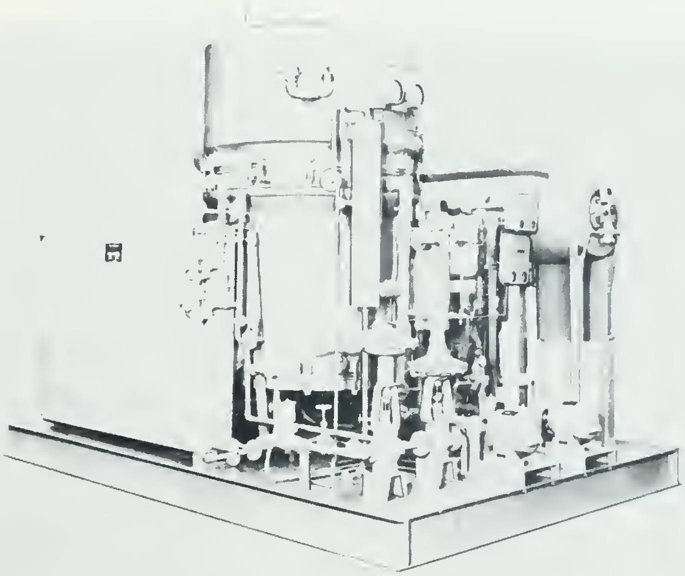

Figure XXIV. Aqua-Chem Vapor Compression Unit

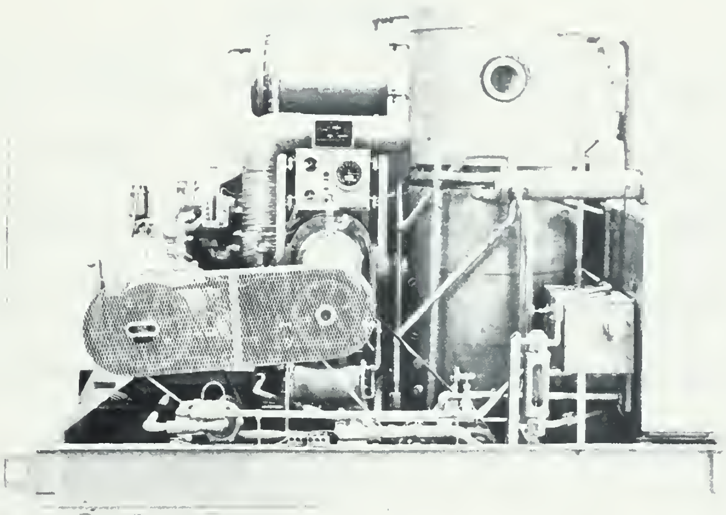

Figure XXV. MECO Diesel Powered Self-contained 200 GPH Unit

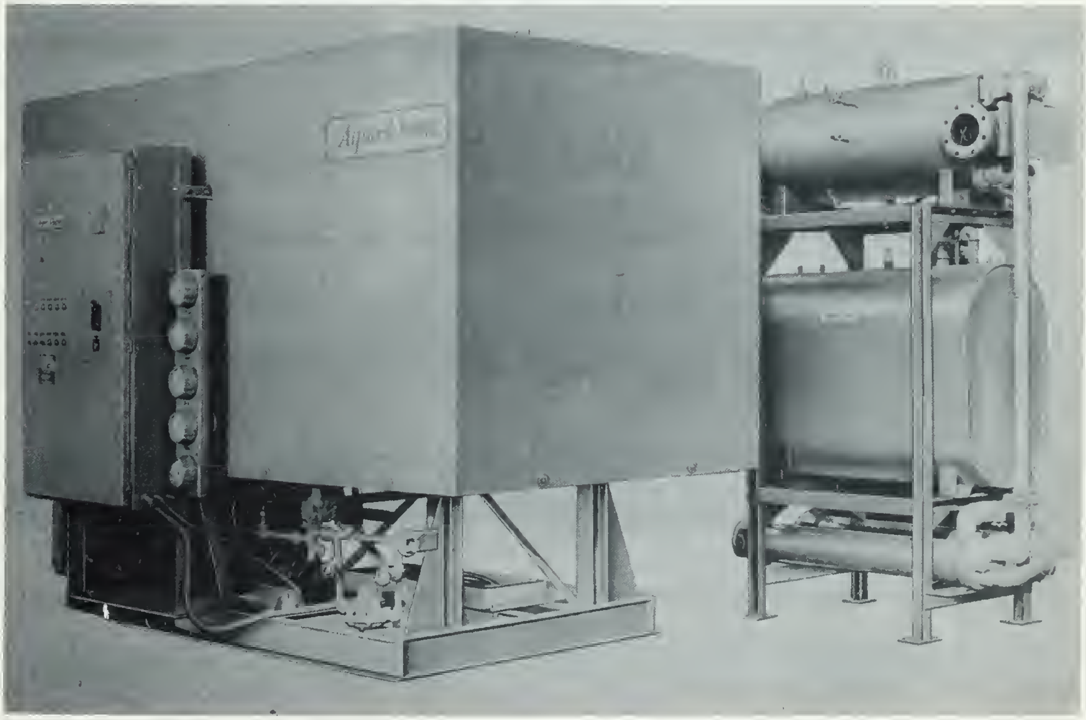

Figure XXVI. Aqua-Chem Diesel Engine Exhaust Heat Recovery, 200 GPH, Unit

PART IV.

DISCUSSION

A. Gas Turbine

The gas turbine engine is available in the size range that is required. There are other factors which must be considered before selection for application. The military specification for gas turbine driven generators lists as criteria of paramount importance, in the order of their priority: (a) reliability, (b) economy of fuel, (c) lightweight, (d) and compactness.[68]

Reliability has been defined as a percent:[25]

$$\text{Percent reliability} = \frac{(\text{installed hr - forced-outage hr.}) \times 100}{\text{installed hour}}$$

The reliability of the gas turbine has been proven by experience. In the case of a small unit, the 300 Kw generator installed in the U.S.S. Gyatt,[5] reliability of 100% was reported in 1958 with a total of 275 operating hours.[34] Since that time, the operating hours have increased to 500, and the Bureau of Ships will install twelve similar 300 Kw emergency ships service generators aboard destroyers (DLG 16-24).[46] Good cold weather starting performance

compared with reciprocating engines may be added to the favorable characteristics of gas turbine engines.

Economy of fuel, specific fuel consumption, in the gas turbine engine ranges from 0.55 lbs/hp-hr in the GMT 305 to 1.5 lbs/hp-hr at full load for the Vector. Diesel engine specific fuel consumption is in the order of 0.38 lbs/hp-hr to 0.55 lbs/hp-hr.[20,48] Component efficiencies and turbine inlet temperature limit the thermal efficiencies to particularly low values in small gas turbine engines.[30,38] In some cases the multifuel capability of the gas turbine can overcome the higher fuel consumption rate if low cost fuel is available. It is not expected that a variety of fuels would be available in an advanced base to permit advantage to be taken of this factor.

The gas turbine engine is much lighter and smaller than the gasoline and diesel engines of comparable output. This is a feature which is of considerable importance in the transportation required to initially outfit a base. It could overcome to some extent initial cost if economy of operation could be obtained. An average of the specific weights from Table 4 shows approximately 3.0 lbs/hp for ten models, with a low of 0.5 lbs/hp and a high of 8.6 lbs/hp.

Gasoline engines are on the order of 4-6 lbs/hp while diesel engines are 6-10 lbs/hp.[20]

Other features of the gas turbine engine add to its attractiveness and should be discussed. The engine has fewer moving parts than reciprocating engines, resulting in lower friction losses and better mechanical efficiency. The simplicity of the engine also contributes to low maintenance, accessibility and ease of overhaul.[35] A 160 HP engine in a minesweeper can be removed and disassembled for inspection in six hours. A crew of four can remove four engines from the same type boat, install replacement engines and have them operating in a total of twenty-eight hours.[34] Installation, i.e., foundation structure, is minimal due to the balance characteristics and light weight. No cooling water is required, and lubrication is required for only the main shaft bearings, reduction gears and reduction gear bearings. Lube oil consumption is low.

The initial cost of gas turbines in comparison with diesels is high. The Boeing 502-21, which is close to the size of interest in the study is approximately \$60/hp, Table 4, compared with \$16-30/hp for diesels, Table 2.

The time for acceleration of gas turbines is long. The Chrysler engine has overcome this feature, and at the

same time the poor braking performance when decelerating, by installation of a variable second stage nozzle mechanism at the power turbine entry.[61]

Noise in gas turbine operation cannot be ignored, but it is not a major problem.[4,27,43]

The limitation of the turbine inlet temperature was cited in PART II, Theory, and in the discussion of economy of fuel. It also has a bearing on the power output. In a simple cycle arrangement with a heat exchanger, an increase from 1400°K to 2000°K would raise the horsepower from 150 to 300 bhp per pound of air per second.[23]

Inspection of Table 4 shows the SFC improvement in the developmental engines. Advancement has resulted from better component efficiencies, and use of the cycle modifications described in PART II, Theory. The results are encouraging, but they are not complete. Cost figures are not included. These benefits were obtained at the expense of increased cost of engineering, materials, and manufacture. Furthermore, features of simplicity of the simple cycle engine are lost. The developmental engines are not now available for use and therefore can only be viewed with reference to what might be possible at some future date.

For each of the current applications of the gas turbine engine, there have been specific reasons for the selection. The gas turbine has not been used as a "cure-all" to replace other prime movers. When selection has been made, particular advantages have outweighed the disadvantages.

In the case of products pumping in the Great Lakes Pipe Line Company's Tulsa pumping station, the prime factor was availability of clean economical fuel.[62] Other considerations included flexibility, simplicity, ease of maintenance, and adaptability to automatic control.[18]

Selection has been made for power generation and power "peaking" installations for reasons of low initial cost, reduced operating personnel, lower maintenance requirements, rapid starting and ease of automation.[25,28,49,63,11] Combined installations, gas turbines and steam turbines have also been successfully employed with increased thermal efficiency.[3,29] Package peaking units (11,000 Kw) are being produced by General Electric at $85 per Kw.[53]

In marine applications, the gas turbine has been selected for propulsion of high speed craft where weight saving was paramount; for propulsion units to obtain performance in excess of cruising power, again where weight and

and space reduction, rapid starting, high availability and reliability were determining factors; and for emergency generating units where reduction of weight above the main deck was a consideration.[34,35,43,54]

The petroleum refining industry selected gas turbines in one case in which the cost of gas turbine installation was comparable with the alternate full condensing steam turbine with the associated cooling water, boiler and auxilliary equipment. The availability of excess catalytic-cracking plant gas, which contributed fuel economy, and ability to place control responsibility on the process operator rather than on another division, led the company to selection of a prototype installation. The developmental experiences although not without problems have not resulted in an unfavorable attitude on the part of the operating company toward further installation of units of this type and capacity.[9]

Specifically pertinent to the small gas turbine application, there are large numbers of units which are in operation where high power/weight ratios give the advantage over other prime movers. This is particularly the case for auxilliary power units in aircraft for air conditioning,

compressed air, and electric power. Ground support aircraft power units for starting, heat, air conditioning and electric power are in wide use. These are intermittent operation units, however. The compactness and high portability are the primary reasons for their success. This may be said also for any emergency service type equipment such as small pumps and electric generators. Lightweight and high availability and reliability are much more important than fuel consumption over a short period of time in this application.

In 1958, there were 238 generators, gas turbine driven, from 200-600 Kw, in service or being installed in ninety-six naval and merchant ships. Of the total, 218 engines were for pulse generators. The remainder were for emergency, ship's service or standby generation.[34] Two of the latter group were installed in U. S. Naval vessels. The experience in one case, 500 operating hours in the <u>U.S.S. Gyatt</u> has been mentioned. The other engine is an 1130 HP unit which has accrued 130 operating hours.[12] Both of these engines are single shaft and have been generally satisfactory. The 218 engines driving pulse generators have accrued a total of 112,200 operating hours, and no

record of any major difficulties has been found. The nature of pulse generation and performance requirements as compared with those for power generation of the type considered in the study are not known.

The developmental problems and design features of the British gas turbine generators included in the total of the preceding paragraph are described by A. W. Pope. One small unit, 125 Kw, for emergency use with anticipated efficiency of 9.2%, SFC 2.3 lbs/Kwh is illustrative of the limitations in a small unit design. The 500 Kw base load unit obtained a 17% thermal efficiency, 0.82 lb/bhp-hr, 450 Kw in tests. This unit weighed 2.5 tons. The 350 Kw emergency and standby unit, 1.25 tons (gas turbine only), performed in tests at 10.75% thermal efficiency.[30]

Recent operating reports on the 500 Kw design, installed in the H.M.S. Llandaff and at the National Gas Turbine Establishment, England, have shown experience to be disappointing on one hand but beneficial to advances in the field on the other. The H.M.S. Llandaff has experienced compressor fouling and some engine failures. The compressor fouling emphasizes the necessity for careful installation arrangements of ducting. The second 500 Kw prototype

experience at NGTE has contributed to improvement in bearing design, correction of overheating during starting, improvement of safety devices, and correction of compressor fouling by cleaning methods and air filtration methods. The NGTE 500 Kw unit has now accrued 4500 successful operating hours. Fifteen similar units are being installed in destroyers and frigates. One of the installations, in the H.M.S. Ashanti has completed 2000 hours. Compressor fouling has not been the problem that it was in H.M.S. Llandaff but the engine did experience cracks in the L.P. turbine.[43]

In review of the small gas turbine driven generators, it should be noted that the British 500 Kw unit is two-shaft, the Boeing pulse generators are two-shaft, the Solar 522-J engines driving pulse generators are two-shaft, but the 300 Kw Gyatt generator and the 750 Kw Oklahoma City generator are single shaft. Furthermore the military specification for gas turbine driven generators which covers all sets up to 2000 Kw, AC, specifies single shaft machines.[68] In testing the GMT 305, a two-shaft engine, as a generator prime mover, the U. S. Naval Engineering Experiment Station was unable to maintain satisfactory speed under variations in load.[14] Although the records of performance of the other

two-shaft engines in service do not indicate a similar experience in speed control, the absence of information on operating requirements of pulse generators precludes a conclusive finding.

The poor speed control of the two-shaft compared with the single-shaft engine is in part due to the difference in inertia of the rotating components. The single-shaft unit inertia is obtained from the compressor, turbine, shaft and generator. The two-shaft unit inertia is in the lighter free power turbine, shaft and generator. When the engine includes a heat exchanger, speed control is influenced by thermal inertia. The desired fineness of speed control in the two-shaft engine may be obtained by load-sensing governors, and by use of variable-inlet vanes in the compressor and/or turbine.[12] Added cost over that of the single-shaft engine must be expected for the controls.

The best performance of the gas turbine engine is attained at or near full-load. Reference to Figure IX shows how the efficiency drops in a typical engine operating at part load. The use of a 150 HP engine to drive a 60 or 75 Kw generator results in operation below full load of the gas turbine even though the generator may be at full load. The

condition is worse in the case of a 300 HP prime mover for the same size generator.

B. Waste Heat Boilers

No manufacturers of waste heat boilers were contacted for equipment data. No problem is anticipated with this component as it is a relatively common item. Erie City 10,000 lb/hr equipment was used in the Park Plaza Shopping Center.[10] Among others, Foster-Wheeler, Cleaver-Brooks, International Boiler Works Company and Maxim are known to manufacture the equipment. Given the exhaust temperature, the weight rate of air-flow, and the limiting back-pressure for a gas turbine, specific output data may be obtained.

From the Boeing Company, it was learned that the 502-21 can be used with a waste heat boiler to produce 150 psig steam at 2400 lbs/hr. It is estimated that such a boiler would cost $4-5,000 and weigh 5-6,000 lbs.

From the Bureau of Yards and Docks information was obtained on a waste heat boiler, with provision for supplemental firing, to produce 125 psig steam at 5000 lbs/hr. The weight of the equipment including boiler, boost burner, piping, ducting, etc., is 10,000 lbs. At $2 per pound of steam, the cost is estimated at $10,000.

In principle the process uses heat transfer apparatus to recover the heat rejected from gas turbine engine operation. A 1000 HP engine rejects approximately 8,000,000 BTU/hr of recoverable energy.[15] An example of waste heat recovery calculation cites for a 1265 HP simple cycle engine, 11,300,000 BTU/hr recoverable of 20,600,000 BTU/hr input by use of a standard waste heat boiler.[2] Another example of this kind shows, for a 7600 HP simple cycle gas turbine, recovery of 40,000,000 BTU/hr.[33] From the examples, dependent on the boiler equipment performance, a recovery of 40-55% of the energy input can be anticipated. The recovery can raise the over-all cycle efficiency to 57-70%. Additional over-all cycle efficiency can be obtained by supplemental firing.

The potential of the combined use of the gas turbine and waste heat boiler has been recognized in a number of applications. The Warren Petroleum Company has installed three Solar Saturn 1000 HP engines for electrical generation with a waste heat boiler for process steam. The combination provides 2100 Kw and 65 psig steam at 17,350 lbs/hr. The U. S. Lines' <u>S.S. Pioneer Moor</u> uses a Solar Saturn 1000 HP unit for a 600 Kw generator exhausting into

a waste heat boiler to produce 150 psig steam at 500 lbs/hr. No information is available on the performance of these units. Figure XXVII shows an arrangement of Solar equipment.

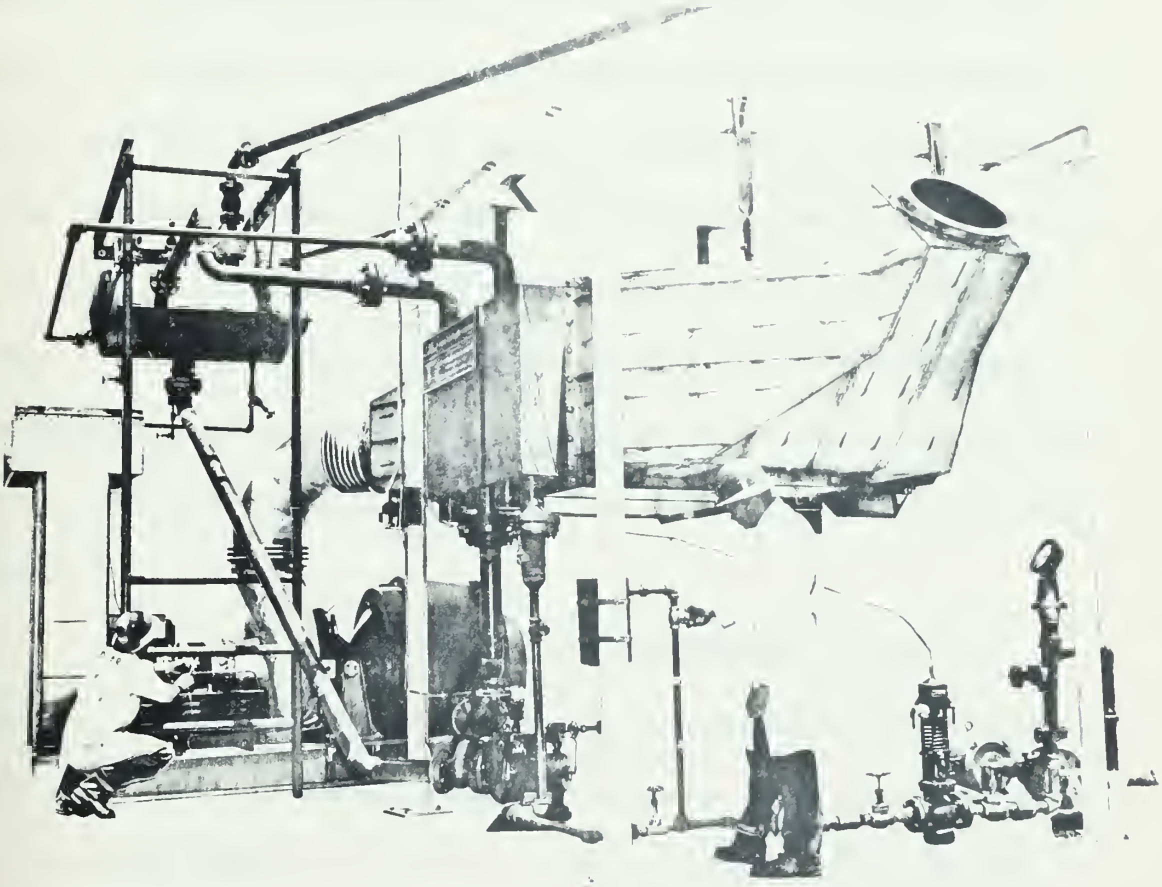

Figure XXVII. Solar Gas Turbine Engine With Waste Heat Boiler

The Esso Petroleum Company's Fawley, England, refinery has had an English Electric Company EM27 in service utilizing a waste heat boiler for two years. The unit, 2625 HP, has operated 17,000 hours with a 99.7% availability. The over-all efficiency with the boiler has been 60%.[51]

The Park Plaza Shopping Center, Little Rock, Arkansas, uses the Clark TA gas turbine to drive a 900 Kw generator which exhausts into a 10,000 lb/hr boiler. The generator provides power and lights, and the 15 psig steam is used to operate two 250 ton absorption air conditioning units, and for heating. A by-pass arrangement is provided in the turbine exhaust ducting which holds the steam pressure at 15 psig. Inlet air in the installation is filtered and washed. The operation in the first year has demonstrated the necessity of operating the turbine at or near rated load and of constant steam loads, for the best utilization of the system.[10]

If the boiler load is not maintained close to capacity, then exhaust heat must be by-passed to the atmosphere, and the gas turbine operates under such conditions at its low efficiency. In the case of a 150 HP unit, already operating at part load efficiency (even though it may be at full electrical load), the loss of the heat load (required to give any degree of economy comparable to other prime movers) results in very poor over-all performance. Under a comparable situation where a 300 HP turbine is used to provide the 60 or 75 Kw electrical output, loss of the heat load represents operation at the worst condition.

The waste heat boiler installation and portability features are not good. The experience reviewed does not show any degree of portability. Even though a certain amount of packaging could be accomplished, the duct connections and piping to the loads would add to the installation requirements.

C. Refrigeration and Air Conditioning

The use of absorption air conditioning is limited by the size of equipment available. The minimum size is 25 tons. The requirement is for only 5-10 tons. In order that use could be made of steam, the possibility of small steam turbines was considered.

Small steam turbines are inefficient. Information obtained from a turbine manufacturer for a 5 HP 1800 rpm turbine with exhaust pressure of 50 psig, 15 psig, and 0 psig indicated steam rates in the order of 600, 350, and 250 lbs/hr, respectively. If a condenser is used to obtain the lower exhaust pressures and the better steam rate, the weights, installation requirements and cost could not compete with an electric motor for the same power output.

Electric motor driven five ton air conditioning compressors are inexpensive, relative portable, and readily installed. The same is true for the ten ton air conditioning unit.

Refrigeration requirements for coil temperatures lower than 40°F prevent the use of absorption equipment. If cold storage temperatures of 35-40°F are to be maintained, the coil temperature will be less. If 0°F is required for frozen food storage, then some method of refrigeration other than absorption is required.

Again, the features of an electric motor driven compressor for refrigeration are comparable to those for air conditioning.

D. Evaporator Equipment

Equipment in the size range of the requirement is available in flash and vapor compression design. The costs for the 4800 gpd size are reasonable close. The economy of operation of the different designs is not.

The flash evaporator process is one of evaporation under a vacuum. The water temperature is raised in a heat exchanger and introduced into a vacuum chamber where it flashes and condenses on the condensing tubes. The brine flows into the next stage where the vacuum is higher and the process is repeated. The coolant in the condensing tubes is the sea or brackish water which ultimately will be distilled. Distillate is produced at the rate of 1.58 lbs

per 1000 BTU in a two-stage design. Increase in the number of stages increases the output. A 26 stage unit yields 6.2 lbs per 1000 BTU. Scaling in the flash process is low. Flash evaporators are capable of delivery of water to meet purity requirements as required ranging from that of boiler feed water to that of a potable supply. Capacities of Aqua-Chem models range from 4000 gpd to 1,000,000 gpd, and equipment uses varied heat sources, such as steam boilers, extraction steam or gas turbine and diesel engine heat. The best economy for flash evaporators is obtained in the integrated power plant arrangement, where the total equipment operation produces water and power. The multiple stage units are large in size and weight which decreases the portability.

Vapor compressor units are available as package components. Feed water is introduced into the evaporator through a heat exchanger, where distillate heat is recovered. Heat is added to start the cycle from an auxilliary source (e.g., steam supply or heat of the diesel engine used to drive the vapor compressor). The feed water is vaporized, taken through a separator into the vapor compressors, and then into the steam chest where it condenses and is recovered

as distillate. The latent heat of the vapor is transferred through the walls of the heat transfer tubes in the steam chest back into the feed water forming more vapor. The heat of blow-down and non-condensable gases, as well as that of the distillate, is partly recovered by heat exchangers. Auxilliary heat required during the process is only that necessary to maintain the heat balance. Aqua-Chem sea water vapor compression units range from 42 to 2100 gph. The 16 lbs. of distillate per 1000 BTU production rate is good. Scaling has been a problem in the past in this equipment. Modifications to provide feed water control, blow-down regulation and acid injection have been effective in combating the scale problem. A program is underway to incorporate these improvements into the 85 gph and 200 gph units which are currently in military supply stocks.[13]

The Maxim Baskevap equipment is available in sizes from 6000-12,000 GPD. The design offers lower weight, but the economy is poorer than vapor compression models. Values of 0.9 lbs. distillate per 1000 BTU for sea-water cooled and 5.0 lbs. distillate per 1000 BTU for condensate cooled operation are given by the manufacturer.

The Mechanical Equipment Company, Incorporated, New Orleans, Louisiana, manufactures self-contained equipment which does not require an external heat source. Sizes range from 200 to 600 GPH. Diesel operated production of 180-190 gallons of water per gallon of diesel oil is obtainable. The MECO unit is essentially a vapor compression type operating at high vacuum, with forced circulation and using a centrifugal blower for steam compression. The low operating temperature eliminates scaling. It has achieved an economy in tests unmatched by other processes.[13]

The U. S. Naval Civil Engineering Laboratory, Port Hueneme, California, is conducting tests of another type of equipment which is of interest in this problem. The principle of operation is the recovery of the rejected exhaust heat of a diesel engine used to drive a 60 Kw generator. The distillation equipment is manufactured by Aqua-Chem and is designed to produce 200 gph of fresh water from salt water. No results are available at this time.

E. Power

The required power both AC and DC can be obtained using standard generators which are available from a large number of manufacturers.

The possibility of use of high frequency generators and converters was not examined in the study.

F. Heating

Use of central heating plant steam produced by a waste heat boiler on an advanced base imposes requirements for piping, fittings, insulation and heat exchangers. The extent depends on the mission, size, arrangement and the climate of the base. Installation would require time and manpower, and would increase the permanence of the facility.

The present method of use of space heaters offers almost immediate availability of heating where required, flexibility and transportability. The initial logistic problem is simple in that each unit is packaged complete with heater, fuel regulator and flue piping. Units of 45,000 $\frac{BTU}{hr}$ output are available at \$60. The space heater method has the disadvantage of a continuing requirement for fuel distribution, although practical experience with the method has shown that this is not serious.

G. Calculations For A Typical Gas Turbine

The following calculations are based on information on performance of the Boeing 502-21, obtained from the Boeing Company.

Continuous rating: 200 HP at $80^{o}F$ 1000 ft.

Air flow: 3.69 lbs/sec

T exhaust: $1075^{o}F$

Fuel flow: 226.4 lbs/hr

Assuming diesel fuel at HHV 18,500 BTU/lb:

$$\text{Heat input} = \text{Fuel Flow} \times \text{HHV}$$
$$= 226.4 \times 18{,}500$$
$$= 4{,}190{,}000 \text{ BTU/hr}$$

$$\text{Shaft output} = \text{bhp} \times 2545$$
$$= 200 \times 2545$$
$$= 509{,}000 \text{ BTU/hr}$$

$$\text{Cycle Efficiency} = \frac{509{,}000}{4{,}190{,}000} = 12.15\%$$

The waste heat boiler capacity data furnished is 150 psig at 2400 lbs/hr. Assuming a feed water entering at $212^{o}F$:

$$h_g = 1194.2 \text{ BTU/lb}$$

$$h_f = 180.1 \text{ BTU/lb}$$

$$\text{Heat recovered} = \text{steam rate x enthalpy change}$$

$$= 2400\ (1194.2 - 180.1)$$

$$= 2{,}440{,}000 \text{ BTU/hr}$$

Assuming that the full amount of heat could be continuously utilized, the over-all cycle efficiency or % utilization may be calculated.

$$\%\ \text{Utilization} = \frac{509{,}000 + 2{,}440{,}000}{4{,}190{,}000}$$

$$= 69.5\%$$

If a two-stage flash evaporator were used to produce 4800 gal of water per day, the BTU/hr requirement would be:

$$\text{Heat required} = \frac{4800\ \frac{\text{gal}}{\text{day}} \times 8.33\ \frac{\text{lb}}{\text{gal}}}{\frac{1.58\ \text{lb}}{1000\ \text{BTU}} \times 24\ \frac{\text{hrs}}{\text{day}}}$$

$$= 1{,}053{,}000\ \frac{\text{BTU}}{\text{hr}}$$

If a vapor compression evaporator were used to produce 4800 gal of water per day:

$$\text{Heat required} = \frac{4800\ \frac{\text{gal}}{\text{day}} \times 8.33\ \frac{\text{lb}}{\text{gal}}}{\frac{16\ \text{lb}}{1000\ \text{BTU}} \times 24\ \frac{\text{hrs}}{\text{day}}}$$

$$= 104{,}000\ \frac{\text{BTU}}{\text{hr}}$$

Since the requirement for air conditioning/ refrigeration cannot feasibly be provided by absorption, or economically by steam turbine, this use of steam is eliminated. Heating of the water obtained from the evaporators would require only a small heat addition. For example, if half the water produced was delivered to hot water storage, and the temperature out of the evaporator was 100°F, using a hot water temperature of 140°F:

$$\text{Heat required} = \frac{2400\ \frac{\text{gal}}{\text{day}} \times 8.33\ \frac{\text{lb}}{\text{gal}} \times (140-100)\frac{\text{BTU}}{\text{lb}}}{24\ \frac{\text{hrs}}{\text{day}}}$$

$$= 33{,}300\ \text{BTU/hr}$$

The operation of the gas turbine with waste heat boiler has obtained the following at the expense of 226.4 lbs of diesel fuel per hour:

Electrical:	140 Kw
Water:	4800 gal per day
Hot Water:	2400 gal per day

Using a diesel engine and electric generator, the MECO evaporator, and using a diesel fired water heater:

$$\text{Generator HP} = \frac{140\ \text{Kw}}{.7}$$

$$\text{Fuel rate} = 200\ \text{HP} \times 0.55\ \frac{\text{lb}}{\text{hp-hr}}$$

$$= 110\ \text{lbs/hr}$$

Water production:

$$\text{MECO fuel rate} = \frac{4800\ \frac{\text{gal}}{\text{day}}}{185\ \frac{\text{gal.}}{\text{gal fuel}} \times 24\ \frac{\text{hrs}}{\text{day}}}$$

$$= 1.08\ \frac{\text{gal}}{\text{hr}}$$

$$= 7.56\ \frac{\text{lbs}}{\text{hr}} \quad \left(\text{diesel oil @ } 7\ \frac{\text{lbs}}{\text{gal}}\right)$$

Hot Water production:

$$\text{Fuel rate} = \frac{2400\ \frac{\text{gal}}{\text{day}} \times 8.33\ \frac{\text{lbs}}{\text{gal}} \times (140-100)\frac{\text{BTU}}{\text{lb}}}{24\ \frac{\text{hrs}}{\text{day}} \times 18{,}500\ \frac{\text{BTU}}{\text{lb}} \times 0.65}$$

$$= 2.77\ \frac{\text{lbs}}{\text{hr}}$$

assuming heater 65% efficient

The operation of the diesel engine generator, the MECO evaporator, and a diesel fired hot water heater, using the higher fuel rate for the diesel engine of 0.55 lbs/hp-hr can deliver the equivalent output for 120.3 lbs of diesel fuel per hour.

Table 7 is a comparison of data on weights and costs of the two schemes.

Table 7. Comparison of Weights and Costs, Gas Turbine and Diesel Engine Schemes

Units	Weight lbs	Cost dollars
Gas turbine w/o generator	335	\$14,000
Vapor Compression Evaporator	9,100	12,000 (1)
Waste Heat Boiler	5,000	5,000 (1)
Total	14,435	\$31,000
Diesel engine w/o generator	2,000 (2)	\$ 6,000 (2)
MECO unit	8,400	16,000
Total	10,400	\$22,000

(1) Estimated

(2) Based on data in Table 2

H. Requirements

There has been an obvious departure in the study from the gas turbine size of the statement of the problem. The examination of a unit larger than 150 HP and consideration of electrical output in excess of the 60-75 Kw range initially contemplated arise from the equipment availability as mentioned in PART III. A, from the better performance of the gas turbine in the larger sizes, from operation close to the full rated load, and from personal experience with the smaller size generator units. An additional factor which

must be considered regardless of size is that of the balance of the electrical loads with those utilizing the recovered energy as the number of generator units in service increase.

Small size gas turbines are available which are capable of delivering the power needed for a 60-75 Kw generator and operating at the full load end of the load-efficiency curve. However, the units are inefficient. A 125 Kw generator has been described with an anticipated efficiency of 9.2% and an SFC of 2.3 lbs/Kwh (1.61 lbs/hp).[30] It is not until the 200-250 HP gas turbine size is reached that efficiencies begin to rise to the point where they become competitive.[38] The disadvantage of part-load operation as the maximum that can be expected of the designed arrangement has already been discussed. If the proposed use of a 150 HP or 300 HP turbine was based on an anticipated insufficiency of exhaust heat, this will not be the case. In the calculation of PART IV. G, a total heat of 2,440,000 BTU/hr was available for a 200 HP gas turbine. This is more than twice the 1,053,000 BTU/hr required to produce 4800 gpd of water, if the flash equipment were used. Assuming the more likely use of the vapor compression unit because of its better production rate and mobility, the heat recovered from the 200 HP

arrangement is over twenty times the required 104,000 BTU/hr. For a 100 HP unit, such as the Vector:

$$\text{Heat input} = \text{Fuel flow} \times \text{HHV}$$
$$= 150 \times 18{,}500$$
$$= 2{,}780{,}000 \text{ BTU/hr}$$

$$\text{Shaft output} = \text{bhp} \times 2545$$
$$= 100 \times 2545$$
$$= 254{,}500 \text{ BTU/hr}$$

$$\text{Cycle Efficiency} = \frac{254{,}500}{2{,}780{,}000}$$
$$= 9.17\%$$

This efficiency is not surprising in view of the previous finding for a 125 Kw generator. Assuming that a waste heat boiler used with the Vector turbine was capable of reducing the boiler exhaust temperature to 400°F:

$$\text{Heat recovered} = m\, c_p\, (T_{gte} - T_{whbe})$$
$$= (2.4)(3600)(0.256)(1200\text{-}400)$$
$$= 1{,}770{,}000 \text{ BTU/hr}$$

Of this heat recovered, only 104,000 BTU/hr, that required to operate the 4800 gpd vapor compression still, can be effectively used. There is no apparent justification then for the selection of the gas turbine in excess of the shaft power requirements for the generator.

A careful review of the electric power requirements of advanced bases should be made with respect to the continued retention of the 60-75 Kw size. Best information available indicates that this size was selected in the World War II period on the basis of equipment "on the shelf" at the time. In Korea at a Marine Air Base, experience with diesel 100 Kw units obtained from the U. S. Army was far superior to that of the Navy furnished 75 Kw generators. During seven months of operation of the air base power plant, the peak load growth was from 5-700 Kw to over 1000 Kw. During the early part of the growth period the least number of units in continuous service was three and in the latter stage, the least number was six. The generator equipment at the high point of the period consisted of two 75 Kw units which were operated on an isolated section of the distribution system, and six 100 Kw BUDA field generator-sets operating in parallel with a permanently installed 300 Kw Caterpillar diesel-generator supplying the main distribution system.

Since the gas turbine engine offers a weight and size saving over the diesel and since the greater efficiencies are attained in the larger units, it is considered that the gas turbine engine driven generator is more likely to be an

[illegible]

asset in an advanced base application in a size such as the 100-300 Kw range. Figure XXVIII is an example of a 300 Kw generator set which is available. This unit is approximately 15 feet long and weighs eight tons. The BUDA (diesel) 100 Kw set referred to is 10-11 feet long and weighs 5.5 tons.

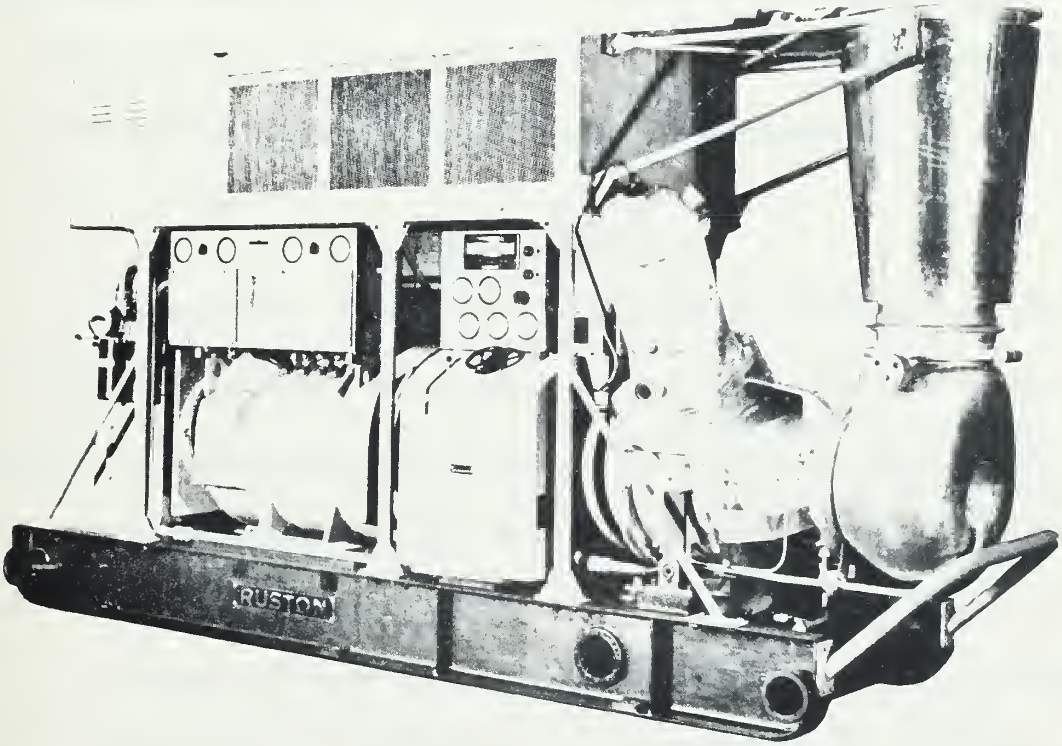

Figure XXVIII. Ruston & Hornsby 300 Kw "TE" Skid Mounted Alternator Set

In regard to the factor of balance of electrical loads and those which utilize the recovered energy of the exhaust, the number of units operated is important. It is

clear that the best utilization is obtained, considering a single unit in operation, when the heat recovered can be used to its fullest. It is expected that total electrical requirements will make the operation of multiple units in parallel necessary. Balance of electrical and heat loads must occur as the number of generators in service increases in order that continued over-all utilization remain favorable. This is true of any application of gas turbine engine with waste heat boiler, advanced base or otherwise.

I. Developments

There are a number of developmental aspects of the gas turbine field which will have an influence on the future prospects of application problems. There are, as well, the developmental programs in the water distillation field which are of interest.

In gas turbine driven generator sets, the U. S. Army Engineer Research and Development Laboratories, Fort Belvoir, Virginia, are currently evaluating proposals for simple cycle engine suitable for application to a 100 or 150 Kw generator set. Their specification calls for an engine weight not to exceed 250 lbs. and an SFC not to exceed 1.0 lb/bhp-hr for the 100 Kw and 0.9 lb/bhp-hr for the 150 Kw size engine on

$60^{o}F$, sea level conditions. The objectives of this program are such that it would be considered advantageous to the Bureau of Yards and Docks to maintain close liaison with ERDL in their findings.

Another gas turbine engine developmental program which has a potential interest to the Bureau of Yards and Docks is the Army-Navy 600 HP Gas Turbine Development Program. The objectives of this program are low SFC (comparable with the diesel engine, less than the gasoline engine), lightweight, small volume, low maintenance, good reliability, and ease of starting. The development specifications call for 600 HP maximum at $80^{o}F$, SFC 0.4 lb/hp-hr from 40 to 100% power, weight of less than 2.5 lbs/hp, volume less than 50 cubic feet, overhaul interval of 1000 hours at 600 HP and 5000 hours at 500 HP, and purchase cost of production units of $20/hp.[55]

The design concepts of the three engines which were presented by Ford, Solar and Orenda show an encouraging future for the small gas turbine engine.[47,58] The results of the AN 600 HP program will have a very positive influence on the problem of this study.

J. Areas Requiring Further Examination

In the examination of the feasibility of use of the gas turbine for the advanced base application, other questions occur which require further study. Among those which are in this category are diesel engine progress in the increase of power/weight ratio, free piston engine application comparison, high frequency generation analysis, and advanced base generator-set size requirements.

Since one of the most advantageous features of the gas turbine engine is in the weight and size saving, it is possible that this advantage might also be available to some extent in new designs in diesel engines. A review of the efforts would be worthwhile. In particular a similar study of rejected heat utilization would show over-all cycle efficiencies and the balance obtainable between electrical and heat loads which could be compared with the results of present equipment performance without heat recovery, and that of the gas turbine and boiler combination.

The free piston engine[66] is in use as the prime mover of a 200 Kw generator set.[34] It is possible that the developments in this field may result in some suitable application for military use. The 200 Kw unit operates at an SFC

of 0.408 lbs/shp-hr, but it weighs 14,000 lbs. dry. Further examination would be needed to determine the capabilities and limitations of free piston engines.

The subject of high frequency generation was mentioned in connection with the Glen Ellyn and Aurora, Illionis, gas turbine installations. An investigation of the equipment availability, characteristics, costs, performance, experience, factors to be considered in application, etc., should be developed to permit evaluation of this method of producing and using electrical energy.

The subject of generator sizes should be reviewed with respect to advanced base electrical loads at the different types of bases that may be planned under varying military situations. There are advantages to be gained by use of a smaller number of larger size units where the electrical load warrants it. From the viewpoint of base power generation, the largest possible generator-set is desirable, consistent with transportability. This does not propose that all advanced-base generator sets be as large as possible. However, it is the intention that the requirements be reviewed with reference to loads and that the planning, allowance lists and procurement programs be

adjusted accordingly. The very recent reference to the 60 Kw generator as a "popular" size[13] leads to the question "why" in the face of increasing magnitude and complexity of requirements of even advanced bases. Concurrently with review of the electrical load requirements and generator sizes, the question of standardization with Army equipment and review of ERDL generator-set development could be reviewed. During the Korean War, construction battalion units obtained replacement equipment through the Army supply system. At the time, the Army and Navy generator sets, as described in PART IV. H, were not the same size. The obvious advantages of interchangeability of parts and electrical compatability through standardization are well known. It may well be that the Bureau of Yards and Docks has already taken the necessary steps to accomplish this examination. But the reference to the popularity of the small size unit creates an element of doubt.

PART V.

CONCLUSIONS AND RECOMMENDATIONS

A. Conclusions

The conclusions to be reached on the basis of this study are those on the specific proposal of the Statement of the Problem, and the generalized conclusions with respect to gas turbine applications in a broader sense. The recommendations may be divided in a like manner.

The generation of 60-75 Kw of electric power using a 150 HP or a 300 HP gas turbine engine, and using a waste heat boiler for recovery of the rejected heat is feasible, but such an arrangement fails to meet the objectives because:

1. There is no weight and space saving in the initial shipment of equipment.
2. There is no apparent initial cost saving.
3. There is insufficient requirement for the rejected heat recovered, compared with the capacity of the gas turbine to provide this energy, to give any economy of operation by this means.
4. The SFC of the available gas turbine engines is such that in comparison with the same output using available diesel engine and fired components, the gas turbine arrangement creates a fuel supply problem.

5. The developmental engines which, from reports, have an improved economy of fuel are unknown quantities with respect to initial cost.
6. The developmental engines are two-shaft designs and present the control problems and cost, to be added to initial cost, in order to permit operation as prime movers for electric power generation.

In general, the conclusions on gas turbine engine application are as follows:

1. The selection for application of gas turbines must be predicated on a specific analysis of the requirements of the individual problem, and the equipment performance specifications must be established by these requirements. There can be no "universal solution" of an application problem using this equipment without such an analysis.
2. The application of a gas turbine engine for continuous operation when the maximum loading will only result in part-load performance is to be avoided.

3. Small gas turbine engines available on the market, 100 HP and below, are in general too inefficient to be used for continuous service.
4. The application of a gas turbine engine with rejected heat recovery is not warranted unless the gas turbine engine can be operated the majority of the time at or near rated load and the requirement for the recovered heat is steady and at or near the quantity produced by the gas turbine engine.

B. Recommendations

It is recommended that for the advanced base power supply requirement:

1. The developmental trends in gas turbine engines be watched closely for improvements which overcome some of the undesirable characteristics of current equipment.
2. A further examination be made of the loads which might be economically satisfied through utilization of the recovered heat.

The gas turbine engine should be considered on a comparative basis with other prime movers for applications in general as follows:

1. For emergency power generation where reliability and minimum installation requirements can be shown to be advantageous, and where fuel consumption over a short period of time is secondary.
2. For peaking and/or standby electric power generation at stations presently generating power, or generating and purchasing, where analysis shows such an installation to give an economical advantage.
3. For stations now generating power where combined operation with existing steam equipment on analysis is found to be profitable.
4. For base load power generation at remote sites where cheap fuel supply is readily available or where steam requirements for heating, cooling or process show that power generation with recovery of rejected heat is economical.

PART VI.

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APPENDIX A

Chronology of Gas Turbine Development[6,18,20,21,22,24,32,40]

130 B.C. (c)	Aeolipile of Hero	reaction-steam
1629	Giovanni Branca	impulse-steam
1791	John Barber of Nuneaton	"original gas turbine scheme"
1884	Sir Charles Parson	patented essential features of modern gas turbine
1894	Armengaud & Lemale (Fr.)	500 HP turbine
1900-04	Stolze	Axial flow multi-blade compressor and turbine with heat exchanger. Low component efficiencies. Engine unsuccessful.
1902	Dr. S. A. Moss (U.S.)	Exhaust driven turbines and aircraft turbochargers
1905-on	H. H. Holzwarth	Constant volume gas turbine manufactured by Brown Boveri from 1911
	Brown Boveri	Constant pressure gas turbines for power, locomotive and other uses.
1908	Dr. Alfred Buchi (Swiss)	Investigated use of diesel engine exhaust to operate air compressor leading to turbochargers

1917	Rateau (Fr.)	First aircraft piston engine turbochargers
1930	Whittle	patents on basic gas turbine and propulsive duct
1936-37	Whittle	Demonstration of first successful jet engine
1937	German Industry	Aircraft gas turbine development
1937	Houndry Catalytic Cracking Process	Sun Oil Company, Marcus Hook, Pa.
1939	Jendrassik	Constant pressure unit. Compressor efficiency 85%, turbine efficiency 84-85%
1939	Aircraft engine	Heinkel Turbo-jet
1939	Electric Power Generation	4000 Kw Brown Boveri unit, Neuchatel, Switzerland
1940	Secondo Campini	Aircraft jet propulsion development
1941	Locomotive	Gas turbine electric 2200 hp Brown Boveri for Swiss Federal Railways
1941	Aircraft propulsion	First aircraft flight in England using Whittle engine
1947	Marine propulsion	High speed naval craft, British "Gatric" project

1949	Gas line pumping	Westinghouse 1800 hp unit at Wilmar, Arkansas, for Mississippi River Fuel Corp.
1949	Electric power generation (U. S.)	3500 Kw General Electric Oklahoma City. The first in the U.S. for power generation for a utility company
1950	Automobile	Rover Turbocar
1953	Bus - Turbocruiser	GM engine 325 HP GT-300
1953	Marine propulsion	Auris. Merchant ship with 25% power by gas turbine
1956	Chrysler turbine car	Cross continent trip at 13.5 mpg
1958	Product pumping	The first installation of a unit to pump products. 1150 HP at Great Lakes Pipe Line Co., Tulsa, Oklahoma

APPENDIX B

Gas Turbine Engine Manufacturers Contacted

The following manufacturers produce equipment in the size range required:

Vector Corporation, 1490 Coolidge Road, Troy, Michigan

Solar Aircraft, 2200 Pacific Highway, San Diego, California

Ford Motor Company, American Road, Dearborne, Michigan

Boeing Aircraft Company, 7755 South Marginal Way, Seattle, Washington

Allison Division of General Motors Corporation, Indianapolis, Indiana

AiResearch Manufacturing Company, Division The Garrett Corporation, Phoenix, Arizona

Equipment of the following manufacturers is larger than the size range required:

Clark Brothers Company, Olean, New York

General Electric Company, Apparatus Division, 1 River Road, Schenectady, New York

Brown-Boveri, 17-19 Rector Street, New York 6, New York

Westinghouse Electric Corporation, P. O. Box 868, Pittsburgh, Pennsylvania

APPENDIX C

Refrigeration and Air Conditioning Equipment Manufacturers Contacted

The following are manufacturers of absorption equipment:

Carrier Air Conditioning, Carrier Building, Syracuse, New York

York Division Borg Warner Corporation, 1960 Grantley Road, York, Pennsylvania

Arkla Air Conditioning Corporation, Shannon Building, Little Rock, Arkansas

Trane Company, 206 Cameron Avenue, LaCrosse, Wisconsin

The following manufacturer does not make absorption equipment:

Frigidaire Division General Motors Corporation, Dept 2361, Dayton, Ohio

APPENDIX D

Evaporator Equipment Manufacturers Contacted

The following manufacturers produce equipment in the size range of the problem:

Aqua-Chem, Inc., 225 North Grand Avenue, Waukesha, Wisconsin (Affiliated with Cleaver-Brooks)

Maxim Division of Emhart Manufacturing Company, 100 Homestead Avenue, Hartford, Connecticut

Mechanical Equipment Company, Inc., 861 Carondelet, New Orleans, Louisiana

The following manufacturer's smallest unit is 2000 gph ($75,000):

Chicago Bridge & Iron Company, 332 South Michigan Avenue, Chicago 4, Illionis

The following manufacturers fabricate on a custom basis, provided no information, fabricate only components, or are not in the evaporator manufacturing business:

Whitlock Manufacturing Company, 77 South, West Hartford, Connecticut

Submerged Combustion, Inc., Logan & Calumet, Hammond, Indiana

Process Engineering, Inc., Foot of Chase, Methuen, Massachusetts

Continental Boiler & Sheet Iron Works, 5603 West Park Avenue, St. Louis, Missouri

Missouri Boiler & Tank Company, 2222 Papin, St. Louis, Missouri

Nooter Corporation, 1414 South Third, St. Louis, Missouri

Buflovak Equipment Division of Blaw Knox, 43 Winchester Avenue, Buffalo, New York

Foster Wheeler Corporation, Dept. T-60, 666 Fifth Avenue, New York, New York

Alco Products, Inc., Dept. 901, Schenectady, New York

APPENDIX E

Development of the Equation for Ideal Thermal Efficiency

From Figure IV:

Compressor Work $= c_p\ (T_2 - T_1)$

Turbine Work $= c_p\ (T_3 - T_4)$

Net Work $= c_p\ (T_3 - T_4) - c_p\ (T_2 - T_1)$

Heat Added $= c_p\ (T_3 - T_2)$

$$\text{T.E.} = \frac{\text{Net Work}}{\text{Heat Added}}$$

$$= \frac{c_p(T_3 - T_4) - c_p\ (T_2 - T_1)}{c_p\ (T_3 - T_2)}$$

$$= \frac{T_3 - T_2 \ - T_4 + T_1}{T_3 - T_2}$$

$$= 1 - \left(\frac{T_4 - T_1}{T_3 - T_2}\right) \qquad (1)$$

Assuming a reversible adiabatic process, for a perfect gas, i.e., compressor efficiency equal to turbine efficiency equal to 100%:

$$\frac{T_1}{T_2} = \left(\frac{P_2}{P_1}\right)^{\frac{1-k}{k}} \qquad \text{Where } k = \frac{c_p}{c_v}$$

$$\text{or } T_1 = T_2\left(\frac{1}{R}\right)^{\frac{k}{k-1}} \qquad \text{where } R = \frac{p_2}{p_1}$$

$$\frac{T_4}{T_3} = \left(\frac{p_2}{p_1}\right)^{\frac{1-k}{k}}$$

$$\text{or } T_4 = T_3\left(\frac{1}{R}\right)^{\frac{k}{k-1}}$$

Substituting in (1):

$$\text{T.E.} = 1 - \frac{T_3\left(\frac{1}{R}\right)^{\frac{k}{k-1}} - T_2\left(\frac{1}{R}\right)^{\frac{k}{k-1}}}{T_3 - T_2}$$

$$= 1 - \left(\frac{1}{R}\right)^{\frac{k}{k-1}}$$

APPENDIX F

Typical Calculation of Theoretical Air and Percent Excess Air[26]

Diesel Fuel: $C_{16}H_{30}$

$C = 12$

$H = 1$

$O = 16$

$N = 14$

$$C_{16}H_{30} + 23.5\ O_2 + 23.5\left(\frac{79}{21}\right) N_2 \longrightarrow$$

$$16\ CO_2 + 15\ H_2O + 23.5\left(\frac{79}{21}\right) N_2$$

$$222 + 752 + 2475 \longrightarrow 704 + 270 + 2475$$

$$1 + 3.39 + 11.15 \longrightarrow 3.17 + 1.22 + 11.15$$

Theoretical Air Required/lb of fuel = 14.54 lbs

Boeing 502-21 $80^{o}F$ 1000 ft.

Fuel flow $= 226.4$ lbs/hr

Th. Air Reqd. $= (226.4)\dfrac{\text{lbs-fuel}}{\text{hr}}\ (14.54)\ \dfrac{\text{lbs-air}}{\text{lbs-fuel}}$

$= 3295$ lbs-air/hr

Mass air flow (Air Delivered) $= 3.69\ \dfrac{\text{lbs-air}}{\text{sec}}\ (3600)\ \dfrac{\text{sec}}{\text{hr}}$

$= 13{,}300\ \dfrac{\text{lbs-air}}{\text{hr}}$

$$\% \text{ Excess Air} = \frac{\text{Air Delivered - Theoretical Air Required}}{\text{Theoretical Air Required}}$$

$$= \frac{13{,}300 - 3295}{3295}$$

$$= 303\%$$

APPENDIX G

Gas Turbine Engines 600 HP and Under[57]

Manufacturer	Model	Use	HP	SFC lbs/hp-hr	Power RPM	Wt. lbs.	Shafts
AiResearch*	GTP331	SP,H	400	0.66	6,000	200	1
W.H. Allen		M,APU	115Kw	1.72	23,000	2690	1
		M,APU	350Kw	1.42	15,000	3700	1
Allison	T63-A-5	Ap,H	212	0.74	33,150	136	2
	GMT305	G,M,Au	175	0.58	25,000	765	2
Austin	250	Sp	250	1.15	29,200	1300	1
Boeing	502-10-MA	Sp	300	0.92	30,200	335	2
	502-10-MAG	Sp	240	0.98	28,500	350	2
	502-10-V	H,Sp	240	1.06	28,500	334	2
	502-10-VB	H,Sp	300	0.99	26,700	330	2
	502-14	H,Sp	300	0.87	28,600	233	2
	502-18	H,Sp	340	0.81	28,600	215	2
	520-2	H,Sp	375	0.76	24,000	315	2
	520-3	Sp	350	0.78	25,000	360	2
	520-6	H,Sp	500	0.67	26,000	250	2
Pratt Whitney	PT6B-3	Sp	400	0.69	33,000	225	2
	ST6B-3	Sp	350	0.72	33,000	225	2
Centrax	CX600	Sp	600	0.87	22,000	1100	1
Chrysler	CR2A	Au	140	0.51	45,730	450	2
Continental	217-5A	Sp	405	0.70	34,000	210	2
Cooper Bessemer	RT110	GC	300	0.83	33,000	225	2
Daimler Benz		Au	300	0.82	21,000	-	2
B.C. Jet Engines		G,H	350	-	28,000	110	1

Manufacturer	Model	Use	HP	SFC lbs/hp-hr	Power RPM	Wt. lbs.	Shafts
Rover	IS/90	G**	100	1.38	46,000	133	1
Ruston Hornsby	TE	G**	430	1.22	19,250	3698	1
Solar	T350	APU	350	0.98	35,100	195	1
	T520J	G	520	0.92	20,600	900	1
	T522J	M	520	0.92	20,600	1000	2
Turbomeca	Artouste IIC	H,Sp	500	0.87	34,000	253	1
	Artouste IIIB	Sp	550	0.65	34,000	287	1
	Astazou II	Sp	554	0.60	43,500	269	1
Waukesha		Au	500	-	25,270	620	2
Williams		Au	500	-	25,270	620	2

*Combined shaft power/air supply units not considered.

** Multiuse

Key to Symbols Used in Use Column

Ap - Prop Turbine
APU- Auxilliary Power Unit
Au - Automotive
G - Generator Drive
GC - Gas Compressor Drive
H - Helicopter
M - Marine
Sp - Shaft power