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An Indirectly Heated Gas-Turbine Cycle for
Minimizing Sulfidation Corrosion

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ABSTRACT

A method for minimizing sulfidation corrosion by indirectly heating the air for the high-temperature portion of a gas-turbine cycle is proposed and discussed. The exhaust air from the high-pressure turbine passes through the combustion chamber and air heater and then through the low-pressure turbine.

Calculations of thermal efficiency and weight are made for both simple and regenerated versions of this cycle. For comparison, similar weight and efficiency calculations are made for conventional simple and regenerated cycles. These calculations show that, in comparison with the conventional simple cycle marinized aircraft engine, weight of the proposed engine is about 100 percent greater; thermal efficiency is about the same; but on the favorable side, air flow per horsepower is less by about 40 percent. In comparison with a conventional regenerated engine, the thermal efficiency of a regenerated indirect air cycle engine is about the same, but the weight is about 50 percent less and the air flow is reduced by about 40 percent. With the proposed cycle, high compression-ratio engines are shown to provide high efficiency in the regenerated version, in contrast to conventional regenerated engines where low compression ratios must be used.

TECHNICAL REFERENCES

1 - Danek, G. J., Jr., "State-of-the-Art Survey on Hot Corrosion in Marine Gas Turbines," MEL R&D Rept 32/65, Mar 1965

2 - "Final Report of the Hot Corrosion Mechanism Studies Conducted for the U. S. Navy Marine Engineering Laboratory, Annapolis, Maryland," Research and Development Center, General Electric Co., 15 Feb 1966

3 - "Development of Hot-Corrosion Resistant Alloys for Marine Gas Turbine Service," Materials and Processes Laboratory, General Electric Co., 21 Jan 1966

4 - Garrett, John H., "Relation Between Machinery Weight Thermal Efficiency and Endurance of Vessels," MEL tech memo (in preparation)

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AN INDIRECTLY HEATED GAS-TURBINE CYCLE FOR MINIMIZING SULFIDATION CORROSION

1.0 INTRODUCTION

A major concern relative to prospective use of high-performance gas-turbine machinery for ship propulsion is the tendency for sulfur in the fuel to combine with salt from the combustion air or from saltwater contamination of the fuel to form a highly corrosive molten slag on the hot parts of the turbine. In order to extend the life of gas-turbine machinery in shipboard applications, it has become the practice to keep turbine inlet temperature below 1550 F,* and in some cases as low as 1400 F, depending upon the nature of the fuel and the service of the vessel.[1] Temperature limitations adversely affect both the power output and the thermal efficiency of turbine machinery.

This laboratory is carrying on an extensive research effort (partly in-house and partly under contract) to determine the causes of, and means of avoiding, sulfidation corrosion.[2,3] One means of avoiding the problem, which has not been extensively explored, would be to keep the high-temperature combustion gases out of the turbine by indirectly heating air for the high-temperature turbine.

2.0 DESCRIPTION OF CYCLE

In the indirect-heating cycle, shown in Figure 1, the air from the compressor passes through a heat exchanger, where its temperature is raised to a maximum of, say, 1750 F. It then passes through a high-temperature turbine that drives the air compressor. The exhaust from the high-temperature turbine passes through the combustion chamber; then the combustion gases pass through the other side of the heat exchanger and are exhausted to the atmosphere through the power turbine, at an inlet temperature not exceeding 1500 F.

In the usual indirectly heated gas-turbine cycle, the combustion gases pass directly from the heat exchanger up the stack so that, unless the waste heat can be used for process steam or in some other waste-heat recovery method, a large, cumbersome high-effectiveness heat exchanger must be used to achieve even moderate thermal efficiency. In the method described here, the heat remaining in the combustion products emanating from the heat exchanger is largely recovered in passing through the power turbine. Low effectiveness in the heat exchanger has little bearing on the thermal efficiency of the cycle. The pressure in the combustion chamber is about three atmospheres; thus both the burning rate and the heat transfer rate are higher than in a combustion chamber operated near atmospheric pressure. The heat exchanger can be relatively small and light in weight, since a large temperature difference can be maintained between the gases on the two sides of the exchanger.

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*Abbreviations used in this text are from the GPO Style Manual, 1959, unless otherwise noted.

[1]Superscripts refer to similarly numbered entries in the Technical References at the beginning of this memorandum.

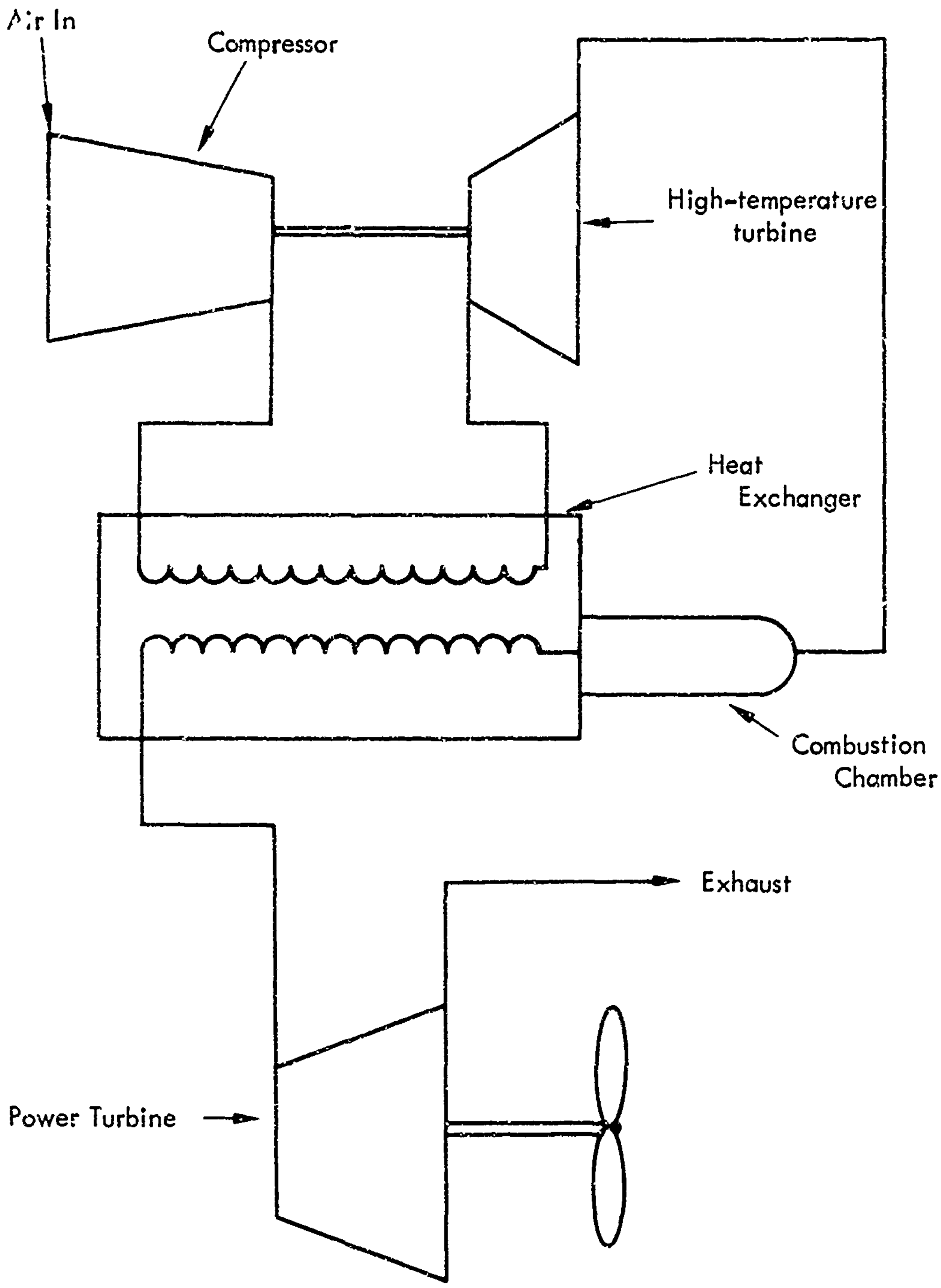

Figure 1

Indirect Air Cycle

3.0 COMPARATIVE ANALYSIS

A calculation of the temperatures at various points in the cycle and thermal efficiencies at three power levels is shown in Appendix A, Cycle 1. For comparison, a similar power and efficiency calculation is shown in Appendix A, Cycle 2, for a simple open-cycle turbine of the usual aircraft type, with addition of a power turbine to replace the aircraft engine jet nozzle, and with turbine inlet temperature limited to 1500 F to minimize sulfidation corrosion. The power turbine inlet temperature in Cycle 1 was limited to 1400 F to avoid any possibility of sulfidation corrosion in the turbine. However, this does not provide a strictly comparable cycle to the base cycle calculated in Cycle 2, since in the latter example a more usual 1500 F was used for the turbine inlet temperature. In order to show the effect of raising the power turbine inlet temperature in the indirect cycle to 1500 F, an additional set of calculations on this basis was made for Cycle 3 in Appendix A. The main points brought out by these calculations are shown in Table 1.

Table 1

Comparison of Two Indirect Air Cycles With Conventional Cycle

	Conventional Cycle (Cycle 2)	Indirect Air Cycle	
		1400 F Turbine Inlet (Cycle 1)	1500 F Turbine Inlet (Cycle 3)
Overall compression ratio	12	12	12
High-temperature turbine inlet temperature, F	1500	1750	1750
Power turbine inlet temperature, F	-	1400	1500
Power turbine exhaust temperature, F	683	1045	1101
Pressure loss in heat exchanger, %[1]	0	6	6
Horsepower per pound per second air flow	81.3	135	143
Thermal efficiency (at optimum power), %	27.7	28.2	27.5
Specific fuel consumption, lb/hp-hr	0.496	0.488	0.500

[1]This was arbitrarily assumed as a conservative estimate. A trial calculation shows that actual losses are likely to be substantially less.

3.1 Thermal Efficiency and Weight. It will be noted that, in spite of pressure losses in the heat exchanger, the thermal efficiency of the indirect air cycle is slightly higher than that of the simple cycle, due to the higher peak cycle temperature which may be employed. More significant is the fact that the net power per pound per second air flow of the indirect cycles is about 166 and 176 percent respectively, of that of the direct cycle. This is a consequence of the higher turbine inlet temperature employed, as well as the reheat of the air between the high- and low-temperature turbines. As the size and weight of gas-turbine machinery, as well as the associated intake and exhaust ducts, are roughly proportional to air flow, this provides a substantial weight and space advantage for parts of the indirect cycle plant, which partially offset the added weight of the heat exchanger.

The estimated weight comparisons for the three cycles are shown in Table 2. The weights shown for the conventional cycle, at 2.3 lb per hp, may seem high. However, this includes the entire gas-turbine plant, including intake and exhaust ducts and all accessory equipment, and is believed to be a reasonable figure. In any event, the absolute accuracy of the values assumed is not particularly pertinent to this study, as the relative weights of other systems in comparison to the base system are obtained by multiplying the base system weights by a factor determined by the relative air flow, the primary determining factor in sizing a gas-turbine plant.

3.2 Heat Exchanger. The heat exchanger weights shown are based on a conventional counter-cross-flow design. All design parameters are conservative, so that the resulting weights should not necessarily be considered typical. For example, the effective core area is only about 40 sq ft per cu ft, whereas core designs providing up to ten times this amount of surface area per unit volume have been tested. Air and gas velocities have, for the most part, been kept below 100 fps, in order to minimize pressure drops. It is probable that values three or four times those used would provide more nearly optimum designs. The calculated weights should be adequate to provide a rugged, long-life, heat exchanger for possible use in a base-load plant of a major ship. For a high-speed special purpose vessel, it would be possible to reduce weights to 25 percent or less in comparison to the weights shown in these calculations.

Table 2

Weight Comparisons, Unregenerated Cycles
(Pounds Per Horsepower)

	Conventional Cycle (Cycle 2)	Indirect Air Cycle	
		1400 F Inlet (Cycle 1)	1500 F Inlet (Cycle 3)
Gas generator	1.0	0.6	0.55
Power turbine	1.0	0.6	0.55
Air and exhaust duct	0.3	0.2	0.2
Heat exchanger	-	3.7	2.9
Total	2.3	5.1	4.2

3.3 Heat Transfer. The detailed calculation of heat transfer rates and heat exchanger weights for the comparisons of Table 2 are shown in Appendixes B and C. It may be seen that when the power turbine inlet temperature of the indirect air cycle is equal to the turbine inlet temperature of the simple cycle, comparisons of weight and efficiency are close enough to be of possible interest, depending upon how critical engine weight is in a particular application.

3.4 Materials Problems. To achieve the benefits of the indirect cycle, a high-temperature heat exchanger is required, in which the wall temperature will vary from about 950 F in the coolest part to a maximum of a round 1950 F at the gas inlet. This range of temperature will present a number of materials problems, principally sulfidation corrosion in those parts of the heat exchanger operating between 1550 and 1750 F, and oxidation above 1750 F. One might question if any advantage is to be gained by simply transferring the materials problem from the turbine hot gas passages to the heat exchanger. This question cannot be answered without undertaking an extensive heat exchanger development and testing program. However, the problem of sulfidation corrosion in turbine blades has so far proved to be singularly intractable, and it is quite possible the solution will prove easier in the different mechanical environment of a heat exchanger. Experience with conventional gas turbines lends credence to the belief that corrosion and oxidation may be easier to prevent in the differently stressed conditions of heat exchangers. There has never been any appreciable success in efforts to maintain a coating on gas-turbine blades over a long operating period. On burner cans and hot gas ducts, on the other hand, both ceramic and diffusion coatings are regularly used in production engines.

3.5 Temperatures. From the calculations in Appendix A, as summarized in Table 1, it appears that the exhaust temperature of the indirect air cycle turbines is much higher than that of the conventional cycle. This is due, of course, to the fact that the temperature of the gas entering the power turbine may be several hundred degrees higher than in the case of the standard cycle. In the latter, the temperature of gas entering the power turbine is the residual temperature after the gas has been partially expanded through the gas generator turbine. In the proposed cycle, the temperature of the exhaust from the gas generator turbine is brought back up in the combustion chamber to any desired level for the power turbine inlet. The indirect air cycle is, in effect, a reheat cycle.

3.6 Heat Recovery. The high exhaust temperature makes it profitable to consider means of further improving thermal efficiency by recovering a portion of the heat from the exhaust. The two usual means of recovering waste heat from the exhaust are through a regenerator or by means of an exhaust-heated steam boiler. As a general rule, the efficiency of any heat transfer process is higher in proportion to the temperature at which the transfer takes place. Thus, because of the low temperature of the exhaust and high temperature of the air leaving the compressor in the standard aircraft turbine cycle, a regnerator is wholly ineffective and a waste-heat recovery boiler is the only practical method of recovering heat from the exhaust. With the indirect air cycle, either a regenerator or a waste-heat recovery boiler would be of interest. Figure 2 is a schematic diagram of the regenerated indirect air cycle.

In Appendix A, Cycles 4 and 5, the temperatures, power output, and thermal efficiencies are calculated for an indirect air cycle, 4:1 compression ratio, with a recuperator of 75-percent effectiveness and one stage of intercooling on the compressor. Cycle 4 is for 1400 F power turbine inlet temperature, while in Cycle 5, the

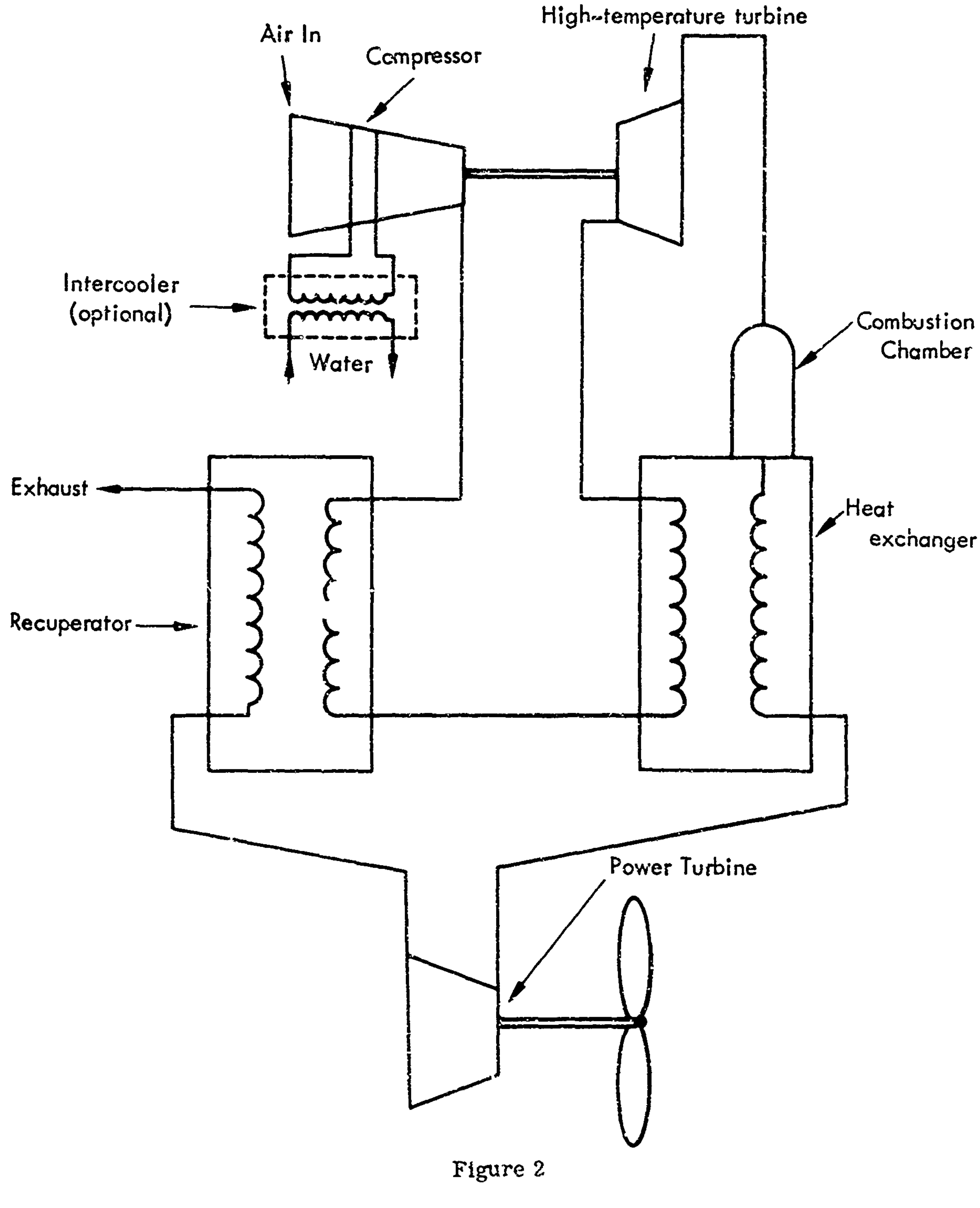

Figure 2

Regenerated Indirect Air Cycle

power turbine inlet temperature is 1500 F. Similar calculations are made in Cycle 6 for a regenerated engine with a compression ratio of 12:1, more typical of available aircraft engines. For comparison, similar calcualtions are made in Cycle 7 for a conventional regenerated direct cycle.

3.7 Comparative Characteristics. In Table 3, the chief characteristics of the four cycles are listed; weights are summarized in Table 4. Table 5 compares the specific fuel consumption of all seven cycles, at various loads, and this comparison is shown graphically in Figure 3.

Table 3

Salient Characteristics of Three Regenerated Indirect Air Cycle

	Cycle Number			
	4	5	6	7
Overall compression ratio	4	4	12	4
High-temperature turbine inlet temperature, F	1750	[illegible]	1750	1500
Power turbine inlet temperature, F	1400	1500	1500	
Overall pressure losses in heat exchanger (assumed), %	15	15	15	9
Horsepower per pound per second air flow	108.5	114.5	170	100.0
Thermal efficiency (at optimum power), %	38.6	38.7	40.5	38.4
Specific fuel consumption (full load), lb/hp-hr	0.356	0.356	0.339	0.358

Cycle Description:

Cycle 4 - Regenerated and intercooled indirect air, 4:1 compression ratio, 1400 F turbine inlet.
Cycle 5 - Regenerated and intercooled indirect air, 4:1 compression ratio, 1500 F turbine inlet.
Cycle 6 - Regenerated and intercooled indirect air, 12:1 compression ratio, 1500 F turbine inlet.
Cycle 7 - Regenerated and intercooled conventional cycle, 4:1 compression ratio, 1500 F turbine inlet.

Table 4

Weight Comparisons, Regenerated Cycles
(Pounds per Horsepower)

	Cycle Number			
	4	5	6	7
Gas generator	0.75	0.75	0.5	0.8
Power turbine	0.75	0.75	0.5	0.8
Air and exhaust ducts	0.2	0.2	0.1	0.2
Air heater	8.2	6.8	2.3	—
Recuperator	12.3	11.2	4.0	11.9
Intercooler	1.7	1.7	0.7	1.8
Total	23.9	21.4	8.1	15.5

Table 5

Comparison of Specific Fuel Consumption
(Pounds per Horsepower Hour)

	Air Flow hp/lb-sec	Full Load	Two-thirds Load	One-third Load
Conventional aircraft cycle (Cycle 2)	81.5	0.496	0.640	0.940
Indirect air cycle, 1400 F turbine inlet (Cycle 1)	135	0.488	0.570	0.740
Indirect air cycle, 1500 F turbine inlet (Cycle 3)	143	0.500	0.610	0.860
Regenerated and intercooled indirect air cycle, 4:1 compression ratio, 1400 F turbine inlet (Cycle 4)	108.5	0.356	0.366	0.310
Regenerated and intercooled indirect air cycle, 4:1 compression ratio, 1500 F turbine inlet (Cycle 5)	114.5	0.356	0.356	0.340

Table 5 (Continued)

	Air Flow hp/lb-sec	Full Load	Two-thirds Load	One-third Load
Regenerated and intercooled indirect air cycle, 12:1 cor pression ratio, 1500 F turbine inlet (Cycle 6)	170	0.339	0.382	0.475
Regenerated and intercooled conventional cycle, 4:1 compression ratio, 1500 F turbine inlet (Cycle 7)	102	0.358	0.370	0.344

3.7.1 These calculations show that even with the high compression ratio typical of aircraft engines, a large benefit is gained from the recuperator. The thermal efficiency is in the same range as with the low compression ratio turbines (although it falls off somewhat at part load), while the power output per pound per second of air is considerably higher.

3.7.2 In order to estimate the specific weight of these systems, three heat exchangers must be calculated - the air heater, the recuperator, and the intercooler. The weight calculation for the heat exchangers is summarized in Appendix B for each of the cycles. For purposes of these calculations, the recuperator is assumed to be made up of core elements identical to those of the hot air heater.

3.7.3 The intercooler is made up of bundles of 0.250-inch inside diameter tubes, with 0.030-inch wall thickness, cooled by water at an average temperature of 120 F, with intake at 60 F. The inside surface of 1 foot of tubing is π d (where d = 0.250/12 or 0.0655 sq ft.

Length of tubing for 1 sq ft of surface 1/0.0655 = 15.5 feet.

Weight of 15.5 feet (that is, weight of 1 sq ft of surface) at 0.29 lb/cu in.:

$$\pi \, (d + 0.030) \times 15.5 \times 12 \times 0.030 \times 0.3 = 1.43 \text{ lb per sq ft of surface.}$$

Assuming that the case, ducts, and piping are equal to the core weight, the intercooler would weigh approximately 2.9 lb per sq ft of surface. (This neglects the weight of water in the intercooler which, properly speaking, should be included.)

4.0 DISCUSSION

4.1 Weight Considerations. The importance of light weight in propulsion machinery varies widely with the class of vessel. The maximum machinery specific weight acceptable for any class of vessel is closely related to the ratio of vessel weight to horsepower. For a merchant ship, for example, 15,000 hp might be adequate to power a 20,000-ton vessel, giving a weight-to-horsepower ratio of nearly 3,000 lb per hp.

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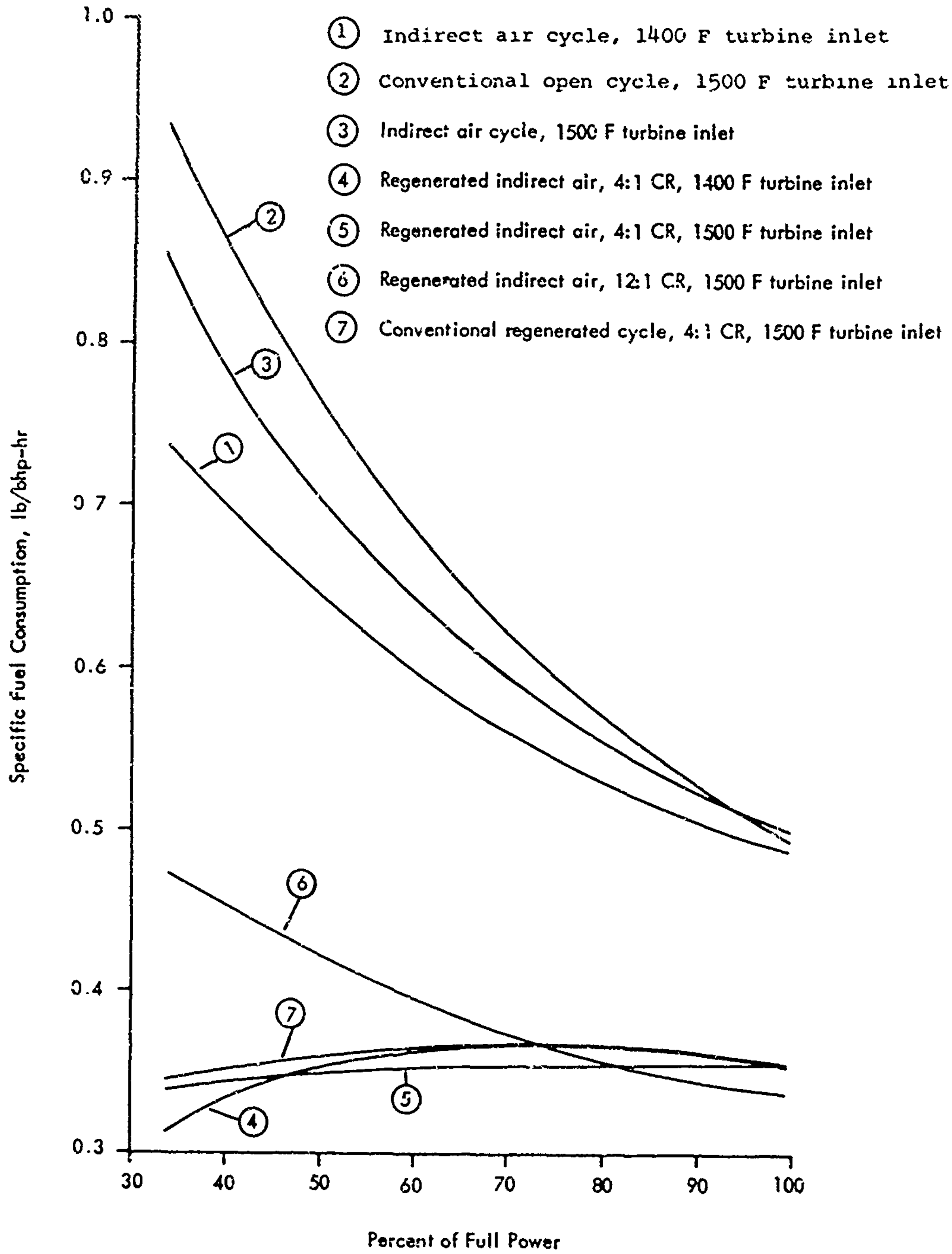

Figure 3

Comparison of Specific Full Consumption of Seven Cycle

At the other extreme are hydrofoils which may have a vessel weight to horsepower ratio of about 12. Conventional naval vessels, using moderately lightweight steam turbines or diesel propulsion machinery, fall between these extremes, a ratio of 200:300 being typical of destroyers, for example. The proposed captured-air-bubble type of air-cushion vessel, with a weight-to-power ratio of 60, will be nearer the lightweight end of the range but will allow considerably more leeway for trade-off between engine weight and fuel than in the case of the hydrofoil.

In a parallel study by the writer,[4] the following equation is developed, relating vessel weight-horsepower ratio, machinery specific weight, specific fuel consumption, and endurance:

$$E = \frac{\frac{KW_s}{P} - W_e}{f}$$

where

E = Endurance, hours

W_s = Gross vessel weight, pounds

K = Fraction of gross weight allocated to machinery plus fuel

P = Horsepower

W_e = Machinery specific weight, pounds per horsepower

f = Specific fuel consumption, pounds per horsepower-hour

Range = $E V_k$ ——, where V_k is the speed, knots.

4.2 Destroyer Application. Using these relationships, it is a simple matter to consider which among the propulsion plants discussed in this memorandum could be considered for a particular class of vessel. Consider, for example, a fast 4000-ton displacement vessel (such as a destroyer) operating at full power at 30 knots, at two-thirds power at 25 knots, or at one-third power at 20 knots. Assume the following design factors as being applicable:

K = 0.45

Weight of power transmission (reduction gears, shafts, bearings, and propellers)	= 6 lb per hp
Weight of auxiliary machinery	= 4 lb per hp
Installed horsepower	= 30,000
Gross weight of vessel	= 8,960,000 lb.

Table 6 then shows the comparative ranges, using each of the seven cycles. For these calculations, all theoretical thermal efficiencies, as shown in Tables 1 and 3, were reduced 10 percent to allow for uncalculated losses, such as loss of heat through engine casings, and internal friction. For the unregenerated cycles, power was assumed to be supplied by several individual units, so that power could be reduced by shutting down units, the remaining units continuing to operate near top efficiency. This assumption tends to weigh the results of range calculations in favor of the nonregenerated cycles. In all cases, ship's hotel load fuel requirement was taken to be 600 lb per hr.

Further inspection of Table 6 shows that the practical differences among the three nonregenerated cycles, and among the four regenerated cycles, are small. As a group, the four regenerated cycles show a substantial range advantage in comparison with the unregenerated cycles, but the differences between the conventional cycles and the indirectly heated air cycles within the regenerated group, and nonregenerated group, respectively, are not significant.

All of the gas-turbine installations show longer ranges than a conventional steam turbine ship of equivalent characteristics. In the case of the regenerated gas-turbine plants, which may be about the same weight as steam machinery, the extended range is due to higher thermal efficiency at all loads. In the case of the nonregenerated gas-turbine plants, the extended range is due to the substantially lighter weight of machinery, allowing more fuel to be carried.

4.3 Surface Effect Ship. If a captured-air-bubble (CAB) type of fast surface-effect ship is considered, the relative ratings of the various cycles turn out to be somewhat different, but not as much as one might suppose. The following parameters may be assumed:

Gross weight	8,960,000 lb
Installed horsepower (W/P = 60)	150,000 hp
Weight of power transmission and auxiliary machinery	2 lb/hp
K	0.40
Fuel requirement for hotel load	300 lb/hr

Power-speed relationship (1-ft wave height):

Power	Full	Two-thirds	One-third
Speed, knots	118	74	28

With these characteristics as parameters, the ranges achievable with the several power plant arrangements are shown in Table 7. In this type of vessel, the two low-compression-ratio regenerated indirect cycles, as well as the conventional regenerated cycle (4, 5, and 7) would be impractical. This is because the specific weight of these cycles is so high as to preclude carrying a useful fuel load. However, the

Table 6

Comparative Ranges for Fast 4000-ton-Displacement Vessel
(Nautical Miles)

Power	Cycle Number						
	1	2	3	4	5	6	7
Full (30 Knots)	6,480	6,420	6,340	7,320	7,5[illegible]0	8,850	8,020
Two-thirds (25 Knots)	7,800	7,750	7,630	8,500	8,920	9,470	9,620
One-third (20 Knots)	12,100	11,760	11,800	15,000	14,240	11,980	14,300

Cycle Description:

Cycle 1 - Conventional open cycle, 1500 F turbine inlet
Cycle 2 - Indirect air cycle, 1400 F turbine inlet
Cycle 3 - Indirect air cycle, 1500 F turbine inlet
Cycle 4 - Regenerated indirect air, 4:1 compression ratio, 1400 F turbine inlet
Cycle 5 - Regenerated indirect air, 4:1 compression ratio, 1500 F turbine inlet
Cycle 6 - Regenerated indirect air, 12:1 compression ratio, 1500 F turbine inlet
Cycle 7 - Conventional regenerated cycle, 4:1 compression ratio, 1500 F turbine inlet

Table 7

Range Comparison for 4000-Ton CAB Vessel
(Range in Nautical Miles, 1-Foot Waves)

Power	Cycle Number						
	1	2	3	4	5	6	7
Full (118 Knots)	4,230	3,710	3,800	-0-	177	4,380	1,910
Two-thirds (74 Knots)	4,000	3,480	3,580	-0-	150	3,330	1,820
One-third (28 Knots)	3,000	3,000	2,690	-0-	132	2,210	1,315

Cycle Description:

Cycle 1 - Conventional open cycle, 1500 F turbine inlet
Cycle 2 - Indirect air cycle, 1400 F turbine inlet
Cycle 3 - Indirect air cycle, 1500 F turbine inlet
Cycle 4 - Regenerated indirect air, 4:1 compression ratio, 1400 F turbine inlet
Cycle 5 - Regenerated indirect air, 4:1 compression ratio, 1500 F turbine inlet
Cycle 6 - Regenerated indirect air, 12:1 compression ratio, 1500 F turbine inlet
Cycle 7 - Conventional regenerated cycle, 4:1 compression ratio, 1500 F turbine inlet

high-compression-ratio regenerated indirect air cycle provides the longest range of any, while the lightest weight plant (simple open cycle) is a close second. It is interesting to note that, with this type of craft, throttling the engines will not result in extending range, as in displacement craft, but will have the opposite effect. This effect occurs in varying degrees with all of the cycles studied. In the particular vessel taken as the example, two-thirds power provides a speed just above the hump in the power-speed curve and one-third power results in operation just below the hump.

The hump is related to the wave-making drag of this class of vessel. The primary wave-making drag, so-called, is due to the depression in the water under the vessel, caused by the pressure of the air bubble. This depression causes the vessel to assume a bow-up attitude which moves the resultant of the lift and drag vectors aft, thus increasing drag. The extent to which this happens depends upon the position of the wave relative to the center of gravity of the vessel. At low speeds, the wave is well forward and maximum drag is encountered. As speed increases, the wave moves aft, the vessel levels out, and primary wave-making drag is reduced. A trace of primary wave-making drag versus speed thus passes through a maximum at some relatively low speed, forming the characteristic hump in the total drag curve.

4.4 Thermal Efficiency. The two conventional means of improving the thermal efficiency of a turbine cycle are (a) use of a regenerator, or (b) use of an exhaust heat-recovery steam boiler and turbine. Both of these approaches involve, in the present state of technology, very substantial increases in the weight of the propulsion system. In this study, no calculations have been made of an exhaust heat-recovery steam turbine, but three regenerated cycles show system specific weights in the range of 8.1 to 23.9 pounds, compared with 2.3 pounds for the basic cycle. In the usual regenerative cycle, it is necessary to use a low compression ratio, in contrast with the nonregenerative cycle as used in aricraft engines, where a high compression ratio improves thermal efficiency. The equations for ideal thermal efficiency show clearly why this is so.

- For the nonregenerative cycle, the ideal thermal efficiency is expressed by the equation:

$$\eta_{ideal} = 1 - \frac{1}{r^{\left(\frac{k-1}{k}\right)}}$$

where r is the compression ratio and k is the ratio of specific heats at constant pressure and constant volume, respectively. Thus, the larger r is, the more nearly thermal efficiency approaches 1.

- For the regenerative cycle, the corresponding equation is:

$$\eta_{ideal} = 1 - \frac{T_2}{T_4} (r)^{\frac{k-1}{k}}$$

where T_2 is the compressor inlet temperature and T_4 is the turbine inlet temperature. In this case, the smallest practicable r provides the highest thermal efficiency.

4.4.1 In calculating a representative regenerated cycle for this study, a compression ratio of 4 was selected as a reasonable compromise between the conflicting requirements of high specific power and high thermal efficiency. With this compression ratio and use of one stage of intercooling on the compressor, an essentially flat specific fuel consumption of about 0.360 lb per bhp-hr (corresponding to thermal efficiency of about 38%) was obtained at all power settings.

4.4.2 One of the difficulties associated with use of a regenerator is that all existing aircraft engines (which represent almost the only source of highly developed basic gas generators for marine adaptation) are high-compression engines and basically unsuitable for use with regenerators. The reason they are unsuited for regeneration is that the temperature of the air at the compressor exit is usually about the same as the power turbine exhaust temperature (particularly when turbine inlet temperature must be limited to moderate values to prolong life or avoid sulfidation corrosion); so there is little opportunity to transfer heat from the exhaust to the incoming air.

4.4.3 With the indirect air cycle described here, however, even when a high-compression aircraft-type gas generator is used, the turbine exhaust temperature is relatively high-around 1050 F at full power with 12:1 compression ratio. By using a two-stage compressor with intercooling between the stages (which, in contrast with the aircraft application, is feasible in marine engines since an adequate supply of cooling water is available), the compressor exit temperature is below 400 F; so there is an ample temperature difference for efficient use of a regenerator. Thus, this cycle affords the opportunity of developing an efficient regenerated cycle engine, using basic aircraft gas generator technology, and avoiding the cost of developing a completely new engine.

4.4.4 The specific fuel consumption (SFC) of this engine at full power (0.339) is actually lower than that of the low-compression-ratio regenerative cycle. The part-power SFC curve is not as flat as those of the low compression ratio cycles, but there is still a great improvement over the unregenerated engines. Curves of SFC in relation to load for all seven cycles are shown in Figure 3.

4.5 Air Flow. For equivalent power output, the gas turbine requires air flow rates several times that of a diesel engine which, in turn, requires considerably more air than a steam plant. Therefore, either air and exhaust ducts must be very large, or the flow velocity must be undesirably high. In either case, the problem of salt spray ingestion is more critical and more difficult to control than with a steam or diesel installation. The space required within the ship for large ducts, as well as both the amount of salt ingested and the size of ducts, can be reduced considerably if the horsepower produced per pound of air can be increased. Accordingly, the specific power output in terms of horsepower per pound, per second of air flow, is a figure of merit for comparing different gas-turbine cycles. The seven cycles included in this study cover the range from 81.6 to 170 hp per lb-sec. At the lower end of the scale is the standard aircraft cycle, while the regenerated indirect air cycle turbine with 12:1 compression ratio is at the other end of the scale.

4.6 Problem Areas.

4.6.1 Heat Exchanger Problems. All of the cycles except the simple open cycle, require one or more heat exchangers, which, in fact, are the chief weight-producing elements of the entire system. A substantial amount of work has been sponsored by the Naval Ship Systems Command (formerly Bureau of Ships) on development and testing of lightweight heat exchanger core designs for use in gas-turbine recuperators. Use of some of the developments resulting from these programs would reduce the weight of the recuperators below that derived in the calculations in this study.

In all of the cycles (except the direct air cycles), a high-temperature heat exchanger is required to transfer heat from the combustion products to the high-pressure air before entering the high-temperature turbine. The temperature of the metal wall in the hottest part of this heat exchanger may reach slightly over 1900 F. While development of a lightweight heat exchanger to sustain wall temperatures of this magnitude is by no means insurmountable, it does represent an area where there is little direct experience, and therefore a development and experimental program in heat exchanger core design would be a necessary preliminary to design of a complete system.

4.6.2 Trade-off between Pressure Loss and Heat Transfer Rate. The weight of the heat exchangers is directly related to the heat transfer rate which, in turn, is a function of the velocity of the gases and the geometrical form of the gas passages. High gas velocity and small passages result in large pressure loss due to friction in the heat exchanger passages. Pressure loss adversely affects both efficiency and power output. Accordingly, for each system, an optimum gas velocity and heat exchanger configuration must be found that will represent a compromise between weight and efficiency. It would be helpful to program the calculations in order to obtain computer assistance in optimizing the various heat exchanger design factors.

4.6.3 Control Problems. In complicating the gas-turbine cycle by, in effect, introducing reheat into the system between the compressor turbine and power turbine, problems of controlling and balancing the various parts of the system at various loads may be introduced. These have not been investigated.

4.7 Extent of Required Modification of Existing Engines. In view of the advanced state of development of aircraft gas turbines and the high cost of developing a completely new engine, it currently appears desirable to base a marine gas turbine as far as possible on some existing aircraft engine. Since all aircraft engines are high compression-ratio engines, unsuited for application of regenerators in the usual way, the two regenerated cycles included in this study which use low compression-ratio engines may not be practical for development at this time. On the other hand, Cycle 6, which is a regenerated cycle but uses 12:1 compression ratio, does not carry this burden and could effi iently use a standard aircraft gas generator.

5.0 CONCLUSIONS

Two main conclusions may be drawn from this study.

- First, it may be possible to minimize some sulfidation corrosion problems by use of indirectly heated air for the high-temperature turbine without unduly increasing weight or reducing thermal efficiency.

- Second, the added flexibility in cycle arrangements thus provided may permit adaptation of high compression aircraft gas generators to obtain an efficiently regenerated marine gas turbine.

Appendix A

Calculations of Power Outputs and Thermal Efficiencies for Seven Gas-Turbine Cycles

Stations

2	-	Compressor inlet
2A	-	First compressor outlet, intercooler inlet
2B	-	Second compressor inlet, intercooler outlet
3	-	Compressor outlet, regenerator inlet, air side
3A	-	Regenerator outlet, air side
4	-	High-temperaturre turbine inlet
5	-	High-temperature turbine exhaust
6	-	Low-temperature turbine inlet
7	-	Low-temperature turbine exhaust Regenerator inlet, gas side
8	-	Regenerator outlet, gas side
T	-	Temperature, R
P	-	Pressure, psia
h	-	Enthalpy, Btu/lb

Note: Thermodynamic properties of air were taken from gas turbine charts in Research Memorandum 6-44, BUSHIPS, Navy Dept., NAVSHIPS 330, Dec 1944.

Description of Cycles

Cycle No.	Description
1	Indirect air cycle, 1400 F low-temperature turbine inlet. High-temperature turbine drives air compressor. Low-temperature turbine is free power turbine.
2	Simple cycle, 1500 F high-temperature turbine inlet. Low-temperature turbine is free power turbine.
3	Same as Cycle 1, except 1500 F low-temperature turbine inlet
4	Indirect air cycle, regenerated and intercooled, 1400 F low-temperature turbine inlet, 4:1 compression ratio.
5	Same as Cycle 4, except 1500 F low-temperature turbine inlet.
6	Same as Cycle 5, except 12:1 compression ratio.
7	Conventional regenerated an intercooled cycle, compression ratio, 1500 F turbine inlet.

Operating Conditions

For all cycles, inlet conditions were taken as 70 F and 14.5 psia, and exhaust pressure as 15 psia. Regenerators were calculated on the basis of 75-percent effectiveness. Where intercoolers are used, inlet temperature of water in intercooler was taken as 60 F. Pressure loss assumed 3 percent of overall cycle pressure ratio for each passage through a heat exchanger.

Cycle 1

High-temperature turbine inlet, F	1750	1600	1450
Compression ratio	12	10	8
Component efficiencies assumed:			
Compressor	0.82	0.80	0.75
Turbines	0.88	0.85	0.80

Cycle 2

High-temperature turbine inlet, F	1500	1400	1340
Compression ratio	12	10.5	9.5
Component efficiencies assumed:			
Compressor	0.82	0.80	0.78
Turbines	0.88	0.85	0.83

Cycle 3

High-temperature turbine inlet, F	1750	1600	1450
Low-temperature turbine inlet, F	1500	1400	1300
Compression ratio	12	10	8
Component efficiencies assumed:			
Compressor	0.82	0.80	0.75
Turbines	0.88	0.85	0.80

Cycle 4

High-temperature turbine inlet, F	1750	1600	1450
Low-temperature turbine inlet, F	1400	1200	1000
Compression ratio	2x2	1.72x1.72	1.42x1.42
= overall	4	3	2
Component efficiencies assumed:			
Compressor	0.90	0.90	0.90
Turbine	0.92	0.90	0.90

Cycle 5

High-temperature turbine inlet, F	1750	1600	1450
Low-temperature turbine inlet, F	1500	1300	1100
Compression ratio	2x2	1.72x1.72	1.42x1.42
= overall	4	3	2
Component efficiencies assumed:			
Compressor	0.90	0.90	0.90
Turbines	0.92	0.90	0.90

Cycle 6

High-temperature turbine inlet, F	1750	1600	1450
Low-temperature turbine inlet, F	1500	1300	1100
Compression ratio	3.46x3.46	3.16x3.16	2.82x2.82
= overall	12	10	8
Component efficiencies assumed:			
Compressor	0.85	0.83	0.81
Turbines	0.88	0.86	0.84

Cycle 7

Turbine inlet, F	1500	1360	1220
Compression ratio	4	3.44	2.84
Component efficiencies assumed:			
Compressor	0.90	0.90	0.90
Turbines	0.92	0.90	0.90

Cycle Calculation Procedure

Step	Quantity	Source
1	T_2	Assumed
2	h_2	Chart of thermodynamic properties
3	h_{2A} (theor)	Chart
4	Δh_A	$\frac{h_{2A} - h_2}{\text{Comp Eff}}$
5	h_{2A}	$h_2 + \Delta h_A$
6	T_{2A}	Chart
7	T_{2B}	560 R - function of intercooler design and water temperature
8	h_{2B}	Chart
9	h_3 (theor)	Chart
10	Δh_B	$\frac{h_{3\,(\text{theor})} - h_{2A}}{\text{Comp Eff}}$
11	Δh_{comp}	$\Delta h_A + \Delta h_B$
12	h_3	$h_{2B} + \Delta h_B$
13	T_3	Chart
14	ΔT_{recup}	$0.75\,(T_7 - T_3)$
15	T_{3A}	$T_3 + \Delta T_{recup}$
16	h_{3A}	Chart
17	T_4	Assumed
18	h_4	Chart
19	h_5 (theor)	$h_4 - \frac{\Delta h_{comp}}{\text{Turbine Eff}}$

Cycle Calculation Procedure

Step	Quantity	Source
20	h_5	$h_4 - \Delta h_{comp}$
21	T_5	Chart
22	$r_{HT\ turbine}$	Chart
23	T_6	Assumed
24	h_6	Chart
25	Overall r	$\dfrac{P_7}{P_2 \times r \times (0.97)^n}$ n = number of passages through a heat exchanger
26	$r_{LT\ turbine}$	$\dfrac{r}{r_1}$
27	$h_{7(theor)}$	Chart
28	$\Delta h_{turbine}$	Turb Eff $(h_6 - h_{7(theor)})$
29	Net power	$\Delta h_{turbine} - \Delta h_{comp}$
30	h_7	$h_6 - \Delta h_{turbine}$
31	T_7	Chart
32	ΔT_{recup}	Same as 14
33	T_8	$T_7 - \Delta T_{recup}$
34	Thermal Eff	$\dfrac{\text{Net Power}}{(h_4 - h_{3A}) + (h_6 - h_5)}$
35	hp/lb-sec	air flow $\dfrac{778}{550}$ (Net Power)
36	SFC	$\dfrac{3600}{18500} \times \dfrac{550}{778} \times \dfrac{100}{\text{Eff}}$
	(18,500 Btu fuel)	$= \dfrac{13.76}{\text{Eff}}$

Cycle 1

	% of Full Power		
	100	56.5	25
T_2	530	530	530
h_2	31.2	31.2	31.2
$h_{2A(theor)}$			
Δ_{hA}			
h_{2A}			
T_{2A}			
T_{2B}			
h_{2B}			
$h_{3(theor)}$	162.2	149.4	134.2
Δh_B			
Δh_{Comp}	159.8	147.8	137.3
h_3	191	179	168.5
T_3	1180	1133	1091
ΔT_{recup}			
T_{3A}			
h_{3A}			
T_4	2210	2060	1910
h_4	468.3	426.2	384.3
$h_{5(theor)}$	286.6	252.5	213.0
h_5	308.5	278.4	247.3
T_5	1548		
$r_{HT\ turb}$	0.24	0.23	0.20

Cycle 1 (Cont)

	% of Full Power		
T_6	1860	1660	1460
h_6	370.9	316.6	263.3
overall r	0.09175	0.10995	0.13743
$r_{LT\ turbine}$	0.3823	0.475	0.687
h_7(theor)	262.1	240.1	227.5
$\Delta h_{turbine}$	95.7	65.0	35.8
Net Power	95.7	65.0	35.8
h_7	275.2	251.6	227.8
T_7	1505	1415	1323
ΔT_{recup}			
T_8			
Thermal Eff	28.2	22.8	15.4
hp/lb-sec air	135		
SFC	0.49	0.60	0.89

Cycle 2

	% of Full Power		
	100	60	30
T_2	530	530	530
h_2	31.2	31.2	31.2
$h_{2A(theor)}$			
Δh_A			
h_{2A}			
T_{2A}			
T_{2B}			
h_{2B}			
$h_{3(theor)}$	162.2	152.6	145.7
Δh_B			
Δh_{comp}	159.5	151.8	146.2
h_3	190.7	183.0	177.4
T_3	1179	1151	1127
ΔT_{recup}			
T_{3A}			
h_{3A}			
T_4	1960	1860	1800
h_4	398.5	370.9	354.5

Cycle 2 (cont)

	% of Full Power		
$h_{5(theor)}$			
h_5			
T_5			
r_{HT}			
T_6			
h_6			
Overall r	0.0863	0.0986	0.1090
r_{LT}			
$h_{7(theor)}$	151.7	146.3	148.7
Δh_{turb}	217	191	170.8
Net power	57.5	39.4	24.6
h_7	181.5	179.9	183.7
T_7	1143	1137	1152
ΔT_{recup}			
T_8			
Thermal Eff	27.7	20.6	13.8
hp/lb-sec air	81.3		
SFC	0.50	0.67	1.00

Cycle 3

	% of Full Power		
	100	60	22
T_2	530	530	530
h_2	31.2	31.2	31.2
$h_{2A(theor)}$			
Δ_{hA}			
h_{2A}			
T_{2A}			
T_{2B}			
h_{2B}			
$h_{3(theor)}$	162.2	149.4	134.2
Δ_{hB}			
$\Delta_{h\ comp}$	159.8	147.8	137.4
h_3	191	179	168.6
T_3	1180	1133	1092
ΔT_{recup}			
T_{3A}			
h_{3A}			
T_4	2210	2060	1910
h_4	468.3	426.2	384.6
$h_{5(theor)}$	308.5	278.4	247.3
h_5	286.6	252.5	213.0
T_5	1548		

Cycle 3 (cont)

	% of Full Power		
	100	60	22
$r_{HT\ turbine}$	0.24	0.23	0.20
T_6	1960	1860	1760
h_6	398.5	370.9	343.6
Overall r	0.09175	0.10995	0.13743
$r_{LT\ turbine}$	0.3823	0.478	0.687
$h_{7(theor)}$	263.6	285.	300.4
Δ_h turb	101	73	34.6
Net power	101	73	34.6
h_7	290.1		
T_7	1561		
Δ_T recup			
T_8			
Thermal Eff	27.5	21.5	11.1
hp/lb-sec air	143	103.3	47.5
SFC	0.500	0.640	1.24

Cycle 4

	% of Full Power		
	100	69	45
T_2	530	530	530
h_2	31.2	31.2	31.2
h_{2A} (theor)	59.0	52.5	44.7
Δ_{hA}	30.9	23.7	15.0
h_{2A}	62.1	54.9	46.2
T_{2A}	656	629	592
T_{2B}	560	550	540
h_{2B}	38.4	36.0	33.6
h_3 (theor)	67.8	58.1	47.2
Δ_{hB}	32.7	24.6	16.0
Δ_{h} comp	63.6	48.3	30
h_3	71.1	60.6	48.6
T_3	696	652	602
Δ_T recup	660	561	513
T_{3A}	1356	1213	1115
h_{3A}	236.1	199.3	174.4
T_4	2210	2060	1910
h_4	468.3	426.2	384.6
h_5 (theor)	399.1	372.6	351.3
h_5	404.7	377.9	354.6
T_5	1983	1886	1800

Cycle 4 (cont)

	% of Full Power		
	100	69	45
$r_{HT\ turb}$	0.616	0.673	0.762
T_6	1860	1660	1460
h_6	370.9	316.6	263.3
Overall r	0.301	0.350	0.423
$r_{LT\ turb}$	0.489	0.520	0.556
h_7 (theor)	287.6	248.1	209.0
Δ_h turb	76.7	61.7	48.9
Net Power	76.7	61.7	48.9
h_7	294.2	247.5	218.1
T_7	1577	1400	1286
ΔR_{recup}	660	561	513
T_8	917	839	773
Thermal Eff	38.6	37.3	41.1
hp/lb-sec air	108.5		
SFC	0.356	0.368	0.334

Cycle 5

	% of Full Power		
	100	69.5	45
T_2	530	530	530
h_2	31.2	31.2	31.2
h_{2A} (theor)	59.0	52.5	44.7
Δ_{hA}	30.9	23.7	15.0
h_{2A}	62.1	54.9	46.2
T_{2A}	658	629	592
T_{2B}	560	550	540
h_{2B}	38.4	36.0	33.6
h_3 (theor)	67.8	58.1	47.2
Δ_{hB}	32.7	24.6	15.0
Δ_h comp			
h_3	71.1	60.6	48.6
T_3	696	652	602
Δ_T recup	725	648	569
T_{3A}	1421	1300	1171
h_{3A}	253.1	221.7	188.6
T_4	2210	2060	1910
h_4	468.3	426.2	384.6
h_5 (theor)	399.1	372.5	351.3
h_5	404.7	377.9	354.6
T_5	1983	1886	1800
r_{HT} turbine	0.616	0.673	0.762

Cycle 5 (Cont)

	% of Full Power		
	100	69.5	45
T_6	1960	1760	1560
h_6	398.5	343.6	289.8
Overall	0.301	0.350	0.423
$r_{LT\ turb}$	0.489	0.520	0.556
h_7 (theor)	310.4	270.9	231.7
$\Delta_{h\ turbine}$	81.0	65.4	52.3
Net power	81.0	65.4	52.3
h_7	317.5	278.2	237.5
T_7	1663	1516	1361
$\Delta_{T\ recup}$	696	652	602
T_8	967	864	759
Thermal Eff	38.7	38.4	39.9
hp/lb-sec air	114.5		
SFC	0.356	0.358	0.345

Cycle 6

	% of Full Power		
	100	61	33
T_2	530	530	530
h_2	31.2	31.2	31.2
h_{2A} (theor)	85.4	80.7	75.1
Δ_{hA}	63.8	59.6	54.2
h_{2A}	95.0	90.8	85.4
T_{2A}	794	777	755
T_{2B}	560	560	560
h_{2B}	38.4	38.4	38.4
h_3 (theor)	95.6	90.6	84.9
Δ_{hB}	67.2	62.9	57.4
$\Delta_{h\ comp}$	131.0	122.5	111.6
h_3	105.6	101.3	95.8
T_3	838	820	797
$\Delta_{T\ recup}$	509	458	404
T_{3A}	1347	1278	1201
h_{3A}	233.8	215.9	196.2
T_4	2210	2060	1910
h_4	468.3	426.2	384.6
h_5 (theor)	319.5	283.7	251.6
h_5	337.3	303.7	273
T_5	1737	1612	1497
$r_{HT\ turb}$	0.324	0.3135	0.3087

Cycle 6 (Cont)

	% of Full Power		
	100	69.5	45
T_6	1960	1760	1560
h_6	398.5	343.6	289.8
Overall r	0.1001	0.1205	0.1505
$r_{LT\ turb}$	0.309	0.385	0.487
h_7 (theor)	262.0	241.4	219.6
$\Delta_{h\ turb}$	120.0	87.9	59.0
New power	120.0	87.9	59.0
h_7	278.5	255.7	230.8
T_7	1517	1431	1335
$\Delta_{T\ recup}$	509	458	404
T_8	1008	973	931
Thermal Eff	40.5	35.1	28.7
hp/lb-sec air	170		
SFC	0.339	0.392	0.479

Cycle 7

	% of Full Power		
	100	68	42.5
T_2	530	530	530
h_2	31.2	31.2	31.2
h_{2A} (theor)	59.0	52.5	44.6
Δ_{hA}	30.7	24.2	15.2
h_{2A}	61.9	55.4	46.4
T_{2A}	658	631	593
T_{2B}	560	560	560
h_{2B}	38.4	38.4	38.4
$h_{3(theor)}$	67.8	60.9	52.6
Δ_{hB}	32.7	25.6	16.2
$\Delta_{h\ comp}$	63.4	49.8	31.4
h_3	71.1	64.0	54.6
T_3	695	666	627
$\Delta_{T\ recup}$	577	523	410
T_{3A}	1272	1189	1037
h_{3A}	214.4	193.2	154.9
T_4	1960	1760	1460
h_4	398.5	343.6	263.3
h_5 (theor)			
h_5			
T_5			

Cycle 7 (Cont)

	% of Full Power		
	100	68	42.5
r_{HT} turbine			
T_6			
h_6			
Overall r	0.2809	0.3266	0.3957
r_{LT} turbine			
h_7 (theor)	252.6	226.3	181
Δ_h turbine	134	105.6	74
Net Power	70.6	55.8	42.6
h_7	264.5	238	189.3
T_7	1464	1363	1174
Δ_T recup	577	523	410
T_8	887	840	764
Thermal Effq	38.4	37.1	39.3
hp/lb-sec air	100		
SFC	0.358	0.371	0.350

Appendix B

Heat Exchanger Weight Calculations Regenerated Indirect Air Cycle

	4:1 Compression Ratio 1400 F Power Turbine (Cycle 4)	4:1 Compression Ratio 1500 F Power Turbine (Cycle 5)	12:1 Compression Ratio 1500 F Power Turbine (Cycle 6)	4:1 Compression Ratio Conventional Cycle (Cycle 7)
Air Heater				
Average temperature of air in heat exchanger $\frac{T_{3A} + T_4}{2}$, R	1782	1816	1779	
ρ_{air} CR x $\frac{530}{T_{avg}}$ x 0.08	0 095	0 093	0 286	
Average temperature of gas in heat exchanger $\frac{T_6 + T_6 + T_4 - T_{3A}}{2}$, R	2287	2355	2392	
Pressure on gas side, atmosphere	2 5	2 5	4	
ρ_{gas} $\frac{P}{P_1}$ x $\frac{530}{T_{avg}}$ x 0 08	0 046	0 045	0 071	
V_{air} (assumed), fps	110	110	80	
V_{gas} $\frac{1}{3}$ x $\frac{\rho_{air}}{\rho_{gas}}$ x V_{air}	75	76	107	
G_{air} (ρV x 3600)	37650	36800	82500	
G_{gas} (ρV x 3600)	12400	12300	27400	
Hydraulic diameter, air	0.0254	0 0254	0 0254	
Hydraulic diamter, gas	0 056	0 056	0 056	
h_{air} 0 0036 $\frac{(G)^{0\ 8}}{(d)^{0\ 2}}$	34 3	33 8	64 3	
h_{gas} 0 0036 $\frac{(G)^{0\ 8}}{(d)^{0\ 2}}$	12 0	11 9	22 7	
Average temperature difference between gas and air, F	504	539	613	
Temperature difference between air and heat exchanger wall, F	.31	140	159	
Temperature difference between gas and heat exchanger wall F	373	399	454	
Temperature of hottest part of heat exchanger wall F	1881	1890	1909	
Area of heat exchanger surface required $\frac{(h_4 - h_{3A})\ 3600}{h \times \Delta T}$, sq ft	158	164	82.5	
Weight per horsepower $\frac{\text{Area} \times 4.75}{\text{HP per 16-sec}}$, lb	8.2	6.8	2.3	

	4:1 Compression Ratio 1400 F Power Turbine (Cycle 4)	4:1 Compression Ratio 1500 F Power Turbine (Cycle 5)	12:1 Compression Ratio 1500 F Power Turbine (Cycle 6)	4:1 Compression Ratio Conventional Cycle (Cycle [illegible])
Recuperator				
Average temperature of air in heat exchanger $\frac{T_3 + T_{3A}}{2}$	1026	1059	1093	[illegible]
Pressure, atmospheres	4	4	12	[illegible]
ρ air $\frac{P}{P_o} \times \frac{530}{Temp} \times 0.08$	0.105	0.1[illegible]0	0.466	0.173
Average temperature of gas in heat exchanger $\frac{T_7 + T_5}{2}$	1247	1315	1263	1176
Pressure, atmospheres	1	1	1	1
ρ_{gas} $\frac{P}{P_o} \times \frac{530}{T} \times 0.08$	0.034	0.032	0.0335	0.036
V_{air} (assumed)	70	70	60	70
V_{gas} $\frac{1}{2} \times \frac{\rho_{air}}{\rho_{gas}} \times V_{air}$	113	117	276	112
G_{air} ($\rho v \times 3600$)	41 550	40,[illegible]60	100 800	43 600
G_{gas}	13,[illegible]30	13 500	33,500	14 500
h_{air} $0.0036 \frac{(G)^{0.8}}{d^{0.2}}$	37.[illegible]	36.3	75.5	43.5
h_{gas} $\frac{0.0036 \times (33500)^{0.8}}{(0.056)^{0.2}}$	13.1	12.9	26.0	13.5
Average temperature difference between gas and air F	221	256	179	192
Temperature difference between air and heat exchanger wall F	57	67	43	45
Area required $\frac{3600 (h_{3A} - h_3)}{h \Delta T}$ sq ft/lb-sec	250	269	142	264
Weight lb/hp $\frac{4.7 \times Area}{hp/lb\text{-}sec}$	12.3	11.2	4.9	11.9

	4:1 Compression Ratio 1400 F Power Turbine (Cycle 4)	4:1 Compression Ratio 1500 F Power Turbine (Cycle 5)	12:1 Compression Ratio 1500 F Power Turbine (Cycle 6)	4:1 Compression Ratio Conventional Cycle (Cycle 7)
Intercooler				
Average temperature [illegible] in intercooler $T_{2A} - \frac{2}{3}(T_{2A} - T_{2B})$ $79[illegible] - \frac{2}{3}(794 - 560)$	592	592	63[illegible]	592
Pressure, atmosphere	2	[illegible]	3.5	2
$\rho = \frac{P}{P_1} \times \frac{530}{T_{avg}} \times 0.06$, lb/cu ft	0.143	0.143	0.23	0.143
V, fps	90	90	90	90
$h = 0.0036 \frac{(3600\,\rho v)^{0.8}}{(d)^{0.2}}$ $0.0036 \frac{(3600 \times 0.25 \times 90)^{0.8}}{(0.0296)^{0.2}}$	42.1	42.1	61.[illegible]	42.1
Average temperature with between air and water ($T_{avg} - 560$), F	32	32	[illegible]	32
Area $\frac{3500\,(h_{2A} - h_{2B})}{h\,\Delta t}$, sq ft	63.2	63.2	2.4	62.8
Weight per hp $\frac{2.9 \times Area}{Hp/lb/sec}$, lb/hp	1.7	1.7	0.7	1.8

Appendix C

Calculations of Heat Transfer Rates and Heat Exchanger Weights

Figure 1-C shows the design of heat exchanger elements selected for purpose of heat transfer and weight calculation. The heat exchanger consists of stacks of thin plates (0.030 inch) separated by corrugations made of the same thickness of sheet stock. The corrugations are set at right angles in alternate layers so as to provide cross flow, and probably about three passes would be used in a countercross-flow arrangement. Because of the higher temperature and lower pressure on the gas side, the spacing of the plates for the gas passages is 0.600 inch, compared with 0.200 inch for the air side.

Heat transfer was based on use of the customary formula:

$$\frac{hd}{k} = 0.023 \left(\frac{Gd}{\mu}\right)^{0.8} \left(\frac{C_p \mu}{k}\right)^{0.4} \quad \ldots\ldots\ldots \text{(C1)}$$

where the symbols have the following meaning:

h = heat transfer coefficient, Btu per (deg F) (sq ft) (hr)

d = hydraulic diameter, ft

k = thermal conductivity, Btu per (deg F) (sq ft) (hr) (ft)

G = mass flow per sq ft cross section = ρv

μ = viscosity, lb/ft-sec

C_p = specific heat at constant pressure, Btu per (deg F) (lb)

$\frac{Gd}{\mu}$ = Reynolds number

$\frac{C_p \mu}{k}$ = Prandtl number

For air at 100 F, this reduces to:

$$h = 0.0036 \frac{G^{0.8}}{d^{0.2}} \quad (1) \quad \ldots\ldots\ldots \text{(C2)}$$

For every 100 F increase in mean air temperature, the values of h should be increased by about 2 percent. For the sake of simplicity and in order to err on the conservative side, this temperature correction was not made in these calculations.

The hydraulic diameter is taken as the perimeter of a unit cell, divided by π.

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(1) Saunders, O. A., and H. Z. Upton, Heat Exchange and Heat Exchangers in Gas Turbine Principles and Practice, Van Nostrand, 1955, pp. 21-22.

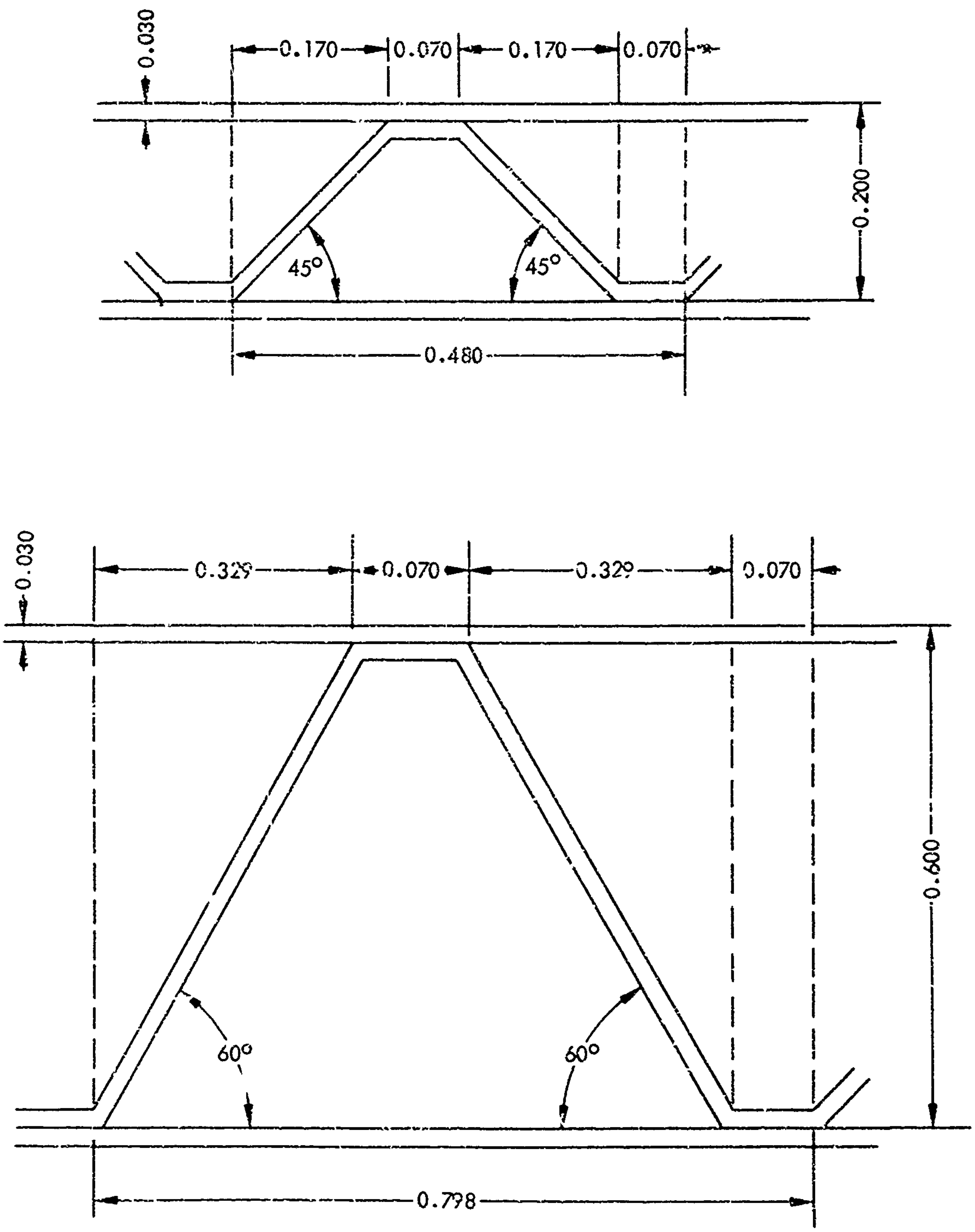

Figure 1-C

Heat Exchanger Core Configuration

For the air side, this is

$$d = \frac{2\,(0.170 + 0.070 + \dfrac{0.170}{\sin 45^\circ})}{\pi} = 0.305 \text{ in.}$$
$$= 0.0254 \text{ ft}$$

For the gas side,

$$d = \frac{2\,(0.329 + 0.070 + \dfrac{0.570}{\sin 60^\circ})}{\pi} = 0.673 \text{ in.}$$
$$= 0.056 \text{ ft}$$

The heat exchanger is sized so that the air and gas velocity will average about 90 fps. This is a conservative value and is selected in order to avoid excessive pressure drops.

Density was calculated as if the combustion products were air, without allowing for the added density due to combustion products. This leads to small errors on the conservative side.

Following is the heat exchanger calculation for 1400 F power turbine inlet temperature:

The average temperature of the air in the heat exchanger is

$$\frac{T_3 + T_4}{2} = \frac{1180 + 2210}{2} = 1695 \text{ R}$$

$$p_{air} = 12 \times \frac{530}{1695} \times 0.08 = 0.30 \text{ lb/cu ft}$$

If one assumes that all of the heat transfer to the air takes place in the heat exchanger and none from the combustion chamber walls (the most unfavorable assumption), and ignoring the slight difference in specific heat between the combustion products and air, then the temperature drop of the gas in passing through the heat exchanger must equal the temperature rise of the air, which is 2210 - 1180, or 1030 F. As the gas leaves the heat exchanger at 1860 R (the power turbine inlet temperature), it must enter at 1860 + 1030, or 2890 R. The average temperature of the gas is $\frac{1860 + 2890}{2} = 2375$ R. The calculations in Appendix A show that the pressure on the gas side is approximately 2.9 atmospheres. Accordingly,

$$p_{gas} = 2.9 \times \frac{530}{2375} \times 0.08 = 0.051 \text{ lb/cu ft}$$

If $V_{air} = 60$ fps, since the flow area on the gas side is three times as great,

$$V_{gas} = 1/3 \times \frac{p_{air}}{p_{gas}} \times 60 = 1/3 \times \frac{0.30}{0.051} \times 60 = 120 \text{ fps}$$

$$G_{air} = \rho v = 0.3 \times 60 \times 3600 = 64800 \text{ lb/hr-sq ft}$$

$$G_{gas} = 0.051 \times 120 \times 3600 = 22000 \text{ lb/hr-sq ft}$$

Using Equation (C2)

$$h_{air} = 0.0036 \frac{(64800)^{0.8}}{(0.0254)^{0.2}}$$

$$= \frac{0.0036 \times 7060}{0.480}$$

$$= 53.0 \text{ Btu per (hr) (sq ft) (deg F)}$$

$$h_{gas} = 0.0036 \frac{(22000)^{0.8}}{(0.056)^{0.2}}$$

$$= \frac{0.0036 \times 2975}{0.5615}$$

$$= 19.2 \text{ Btu per (hr) (sq ft) (deg F)}$$

The average temperature difference between the air and gas on opposite sides of the heat exchanger wall is nearly equal to the difference in the average temperatures of the air and gas in the heat exchanger or 2375 - 1695 = 680 F.

If one ignores the presumably small temperature difference between the two sides of the heat exchanger wall, the temperature difference between the heat exchanger wall and the air is $\frac{19.2}{53.0 + 19.2} \times 80 = 180$ F.

The temperature difference between the gas and the heat exchanger wall is 680 - 180 = 500 F. The temperature of the hottest part of the heat exchanger wall will then be approximately 500 F cooler than the entering gas, that is, 2890 - 500 = 2390 R, or 1930 F.

The area of heat exchanger surface required

$$= \frac{Q}{h \Delta T}$$

where

Q = heat to be transferred per hour per pound per second air flow (3600 x $[h_4 - h_3]$ from Appendix A)

h = heat transfer rate

ΔT = temperature between air and heat exchanger wall

$$\text{Area} = \frac{277.3 \times 3600}{53.0 \times 180} = 105. \text{ sq ft/lb-sec air flow.}$$

The weight of heat exchanger core per square foot of surface may be calculated as follows:

One unit section includes half of two top and bottom plates, one center plate, one shallow corrugation, and one deep corrugation. The thickness of this unit section is 0.020 + 0.060 = 0.080 inch, and there are thus 15 such sections per foot of heat exchanger core thickness.

Linear length of shallow corrugations per foot of core length:

Unit cell linear length 0.170 + 0.070 = 0.240 in.

Unit cell developed length $\frac{0.170}{0.707} + 0.070 = 0.310$ in.

No. of unit cells per foot $\frac{12}{0.24} = 50$

Developed length of 1 foot = 50 x 0.310 = 15.5 in.

Linear length of deep corrugation per foot of core length:

Unit cell linear length = 0.329 + 0.070 = 0.399 in.

Unit cell developed length $= \frac{0.329}{\sin 30^\circ} + 0.070 = 0.728$ in.

Number of unit cells per foot $\frac{12}{0.399} = 30.1$

Developed length of 1 foot 30.1 x 0.728 = 21.9 in.

Total developed length of one unit section, 1 foot long

2 x 12 + 15.5 + 21.9 = 61.4 in.

Total developed length of 1 cubic foot of core

15 x 61.4 = 921 in.

Weight of 1 cubic foot of core, at 0.3 lb cu in. metal weight

921 x 12 x 0.030 x 0.3 = 99.5 lb.

Effective surface of 1 cu ft, assuming effectiveness of corrugations, per square foot, is 25 percent of that of plates:

Area of plates = 2 x 15 = 30

Area of corrugation

$$\frac{15.5 + 21.9}{12} \times 15 \times 0.25 = \frac{11.7}{\text{Total area} \quad 41.7 \text{ sq ft/cu ft.}}$$

Weight per square foot of effective area:

$$\frac{99.5}{41.7} = 2.38 \text{ lb.}$$

Assuming weight of outer case, supports, and ducts is equal to 100 percent of core weight of heat exchanger = 2 x 2.38 or 4.75 lb/sq ft.

$$\text{Weight per horsepower} = \frac{105 \times 4.75}{135}$$

$$= 3.7 \text{ lb.}$$

A similar weight calculation for the heat exchanger of the indirect air cycle for 1500 F turbine inlet temperature follows:

Average temperature of air in heat exchanger

$$\frac{T_3 + T_4}{2} = \frac{1180 + 2210}{2} = 1695 \text{ R.}$$

$$\text{Pair} = 12 \times \frac{530}{1695} \times 0.08 = 0.30 \text{ lb/cu ft.}$$

Average temperature of gas in heat exchanger $\frac{1960 + 12990}{2} = 2475$ R.

Pressure on gas side 0.26 x 12 = 3.1 atmospheres.

$$\rho_{gas} = 3.1 \times \frac{530}{2475} \times 0.08 = 0.0531 \text{ lb/cu ft}$$

If

$$V_{air} = 65 \text{ fps}$$

$$V_{gas} = 1/3 \times \frac{0.30}{0.0531} = 65 = 122$$

$$G_{air} = pv = 0.30 \times 65 \times 3600 = 70{,}100 \text{ lb/hr-sq ft}$$

$$G_{gas} = 0.0531 \times 122 \times 3600 = 23{,}350 \text{ lb/hr-sq ft}$$

$$h_{air} = 0.0036 \frac{(70.100)^{0.8}}{(0.0254)^{0.2}} = \frac{0.0036 \times 7525}{0.480} = 56.5 \text{ Btu per (sq ft) (hr) (deg F)}$$

$$h_{gas} = \frac{0.0036 \times (23350)^{0.8}}{(0.056)^{0.2}} = \frac{0.0036 \times 3122}{0.5615} = 20.0 \text{ Btu per (sq ft)(hr) (deg F)}$$

Average temperature difference between gas and air

$$2475 - 1695 = 780 \text{ F}$$

Temperature difference between air and heat exchanger wall

$$\frac{20.8}{56.5 + 20.0} \times 780 = 204 \text{ F.}$$

Temperature difference between gas and heat exchanger wall

$$780 - 204 = 576 \text{ F.}$$

Temperature of hottest part of heat exchanger wall, 1954 F.

Area of heat exchanger surface required

$$\frac{(468.3 - 191)\ 3600}{56.5 \times 204} = 86.8 \text{ sq ft/lb-sec air}$$

Weight per horsepower $\frac{4. \quad \times 86.8}{143} = 2.9$ lb

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13 ABSTRACT

A method for minimizing sulfidation corrosion by indirectly heating the air for the high-temperature portion of a gas-turbine cycle is proposed and discussed. The exhaust air from the high pressure turbine passes through the combustion chamber and air heater and then through the low-pressure turbine. Calculations of thermal efficiency and weight are made for both simple and regenerated versions of this cycle. For comparison, similar weight and efficiency calculations are made for conventional simple and regenerated cycles. These calculations show that, in comparison with the conventional simple cycle marinized aircraft engine, weight of the proposed engine is about 100 percent greater thermal efficiency is about the same; but on the favorable side, air fl flow per horsepower is less by about 40 percent. In comparison with a conventional regenerated engine, the thermal efficiency of a regenerated indirect air cycle engine is about the same, but the weight is about 50 percent less and the air flow is reduced by about 40 percent. With the proposed cycle, high compression-ratio engines are shown to provide high efficiency in the regenerated version, in contrast to conventional regenerated engines where low compression ratios must be used.

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