Regenerated Marine Gas Turbines, Part II: Regenerator Technology and Heat Exchanger Sizing

Analytical studies are currently being conducted by the David Taylor Naval Ship R&D Center to assess the suitability of regenerative-cycle and intercooled, regenerative-cycle gas turbines for naval applications. This paper is the second part of a two-part paper which discusses results of initial investigations to identify attractive engine concepts based on existing turbomachinery and to consider the regenerator technology required to develop these engine concepts. Part I of the paper analyzed existing and next generation engines for performance improvement. Part II includes: definitions of performance parameters such as effectiveness and pressure drop, a discussion of regenerator types, and comments on regenerator materials, life, maintenance, and fouling. Tradeoffs between size, weight, and performance of plate-fin recuperators are examined using two of the hypothetical engines from Part I as examples. Results are compared for several different recuperator matrices to illustrate the effects of air-side and gas-side fin density and plate spacing on size, weight, and performance.

INTRODUCTION

The performance of existing and next generation marine gas turbines can be improved significantly as shown in Part I of this paper by adding a regenerator, and possibly a compressor intercooler. Hypothetical engine concepts including a low-compression, 8:1, regenerative cycle with and without intercooling and a high-compression, 16:1, intercooled-regenerative cycle were found to have thermal efficiencies as high as 45%. Obviously, part of the price for better efficiency is an increase in engine size and weight due to the addition of a regenerator and in some cases an intercooler as well. In order to assess the suitability of regenerative-cycle and intercooled, regenerative-cycle gas turbines for naval applications, estimates of heat exchanger weight and size as a function of performance are needed. Heat exchanger performance is characterized by effectiveness and pressure drop parameters which vary in value, depending upon the type, size, and geometry. Five different types of recuperators are discussed here to illustrate that the selection of a regenerator type can be heavily influenced by the specific constraints of a particular marine application. Materials selection must take into account the high temperature and corrosive environment in which a marine regenerator will operate. These areas plus regenerator fouling, maintenance, and life are discussed briefly in an overview of regenerator technology which is intended only to highlight areas deserving special attention and to note some authoritative references for further reading. Plate-fin counterflow heat exchangers have many advantages which make them attractive candidates for marine recuperators. Tradeoffs between size, weight, and performance of plate-fin counterflow recuperators are examined using two hypothetical engine concepts defined in Part I of this paper as examples. These engine concepts include a low-compression, regenerative cycle (engine type I) and a high-compression, intercooled-regenerative cycle (engine type II). Various recuperator matrices are...
examined to show the effects of air-side and gas-side fin density and plate spacing on size, weight, and performance.

PERFORMANCE CHARACTERISTICS

There are two design parameters which impact on the size and shape of heat exchangers used for regeneration of gas turbine engines. These are effectiveness (ε) and pressure drop (ΔP). Fuel efficiency is a parameter which includes the effects of both ε and ΔP in one number, and thus is a convenient optimization parameter. Superimposed on this simplified regenerator design picture are life-cycle cost considerations which include: the weight and space penalty on board a ship, operating cost based on a mission profile, and development/acquisition/fleet introduction costs of a regenerated gas turbine.

Effectiveness is improved by increasing either the overall heat transfer coefficient (U) or the heat transfer surface area (As) according to the following relationships (1):

$$\text{NTU} = \frac{A_s}{\text{U}_{\text{air}}}$$  (1)

$$\varepsilon = \frac{\text{NTU}}{1 + \text{NTU}}$$  (2)

Equation (2) assumes equal capacity rates on the air and gas sides. Increasing U usually results in an undesirable increase in ΔP by the Reynolds analogy, the relationship between heat transfer and skin friction coefficient. Thus, there is limited improvement in SFC by trying to improve U. The area A_s can be increased without increasing ΔP by making the heat exchanger "pancake" shaped, but this, too, has its limitation in the weight of header required and in flow distribution problems. By writing equation (2) as:

$$\text{Volume} \propto A_s \propto \text{NTU} = \varepsilon \frac{1}{1 - \varepsilon}$$  (3)

it can be observed that for any given heat exchanger type, approximately a doubling of volume is required for improving ε half way closer to 100%, i.e., from 80% to 90% or from 90% to 95%.

Fractional pressure drop (ΔP = ΔP/ΔP_j) directly relates to a power loss regardless of whether it occurs on the gas or air side of the heat exchanger. Since the air side is at a higher absolute pressure, pressure losses on the air side are not as penalizing as pressure losses on the gas side by a factor approximately equal to the compression ratio. The fractional pressure drops on air and gas sides can be added to give a number which indicates power loss due to flow of gas through the regenerator. This total pressure drop number should be kept low because it penalizes thermal efficiency and also lowers maximum obtainable power.

REGENERATOR TYPES

Five different regenerator types are shown in Figure 1 and their advantages and disadvantages are compared in Table 1. The selection of regenerator type for any given marine application will depend heavily on the gas turbine cycle parameters.

Examples of where each of the five types might be used in a marine application are given below.

1. Plate-fin Counterflow.
   This heat exchanger type would be a good choice for a gas turbine cycle which has a moderate to high R_c with a large θ. Compressor intercooling may be present. There is a high cost of fuel, encouraging a high effectiveness at the expense of a heavy exchanger.

2. Tube Shell.
   This is a good choice for a turbine cycle having a high R_c with a relatively low θ. The low θ makes a high ε have less significance. The relatively low weight of tube-shell heat exchangers makes them attractive for weight critical applications, especially with the above cycle conditions.

3. Rotary.
   This is a good choice for a gas turbine cycle having a low R_c (below 5:1) and a high θ. The light weight of the rotary (lightest of all recuperators/regenerators) looks especially attractive for weight critical applications. At current technology, leakage and carryover are problems above a pressure ratio of 5:1.

4. Liquid Coupled Indirect Transfer Heat Exchanger (LCITHX).
   Improvements in the gas and air ducting arrangements are possible with a liquid coupled heat exchanger but with a weight penalty. It is possible that a saving in ducting and header size and in pressure losses, for certain engines and ducting arrangements, would justify the use of this concept.
MATERIALS CONSIDERATIONS

Resistance due to the higher chromium content (6).

Resistance. More recently, a change to certain operating at rated power varies from 800 to 1100 F.

The hottest temperature in a gas turbine regenerator performance, done below, is for plate-fin type.

WAVEPLATE EXCHANGERS IS CONSIDERED MORE IMPORTANT IN HIGH-COMPRESSION RATIO GAS TURBINES, THE MOST PROPELLION SYSTEMS IS HIGH EFFECTIVENESS, PLATE-FIN COUNTERFLOW EXCHANGERS USE ALL THE METAL IN ITS APPLICATION THAN THE ADDITIONAL WEIGHT OF THE WAVEPLATE EXCHANGER.

In summary, there are a variety of exchangers available for recuperation of marine gas turbines. Recent improvements in manufacturing make plate-fin exchangers very competitive with tube shell, even in low effectiveness applications, because tube-shell exchangers require much hand labor in their manufacture. Waveplate exchangers use all the metal in primary exchanger surface, but, because of headering problems, they, too, are not as weight and volume efficient as plate-fin exchangers. Given today's high-compression ratio gas turbines, the most attractive regenerator design for naval ship main propulsion systems is high effectiveness, plate-fin counterflow exchangers, especially in the light of increased fuel costs. Thus, the exchanger sizing and performance, done below, is for plate-fin type.

MATERIALS CONSIDERATIONS

Several factors which must be considered when selecting a material for a regenerator include temperature capability, oxidation and corrosion resistance, material cost, and ease of fabrication. The hottest temperature in a gas turbine regenerator is the inlet gas temperature which is also the turbine exhaust temperature. The exhaust gas temperature of most marine or industrial gas turbines operating at rated power varies from 800 to 1100 F (430 to 590 C). Early industrial regenerators, made from low alloy steel containing 1% chromium, would resist oxidation up to a maximum temperature of about 950 F (510 C) but had relatively poor corrosion resistance. More recently, a change to certain stainless steels has resulted in low oxidation rates to well over 1200 F (650 C) and improved corrosion resistance due to the higher chromium content (6). A problem observed in many metals having a high chromium content is caused by a phenomena called "sensitization" when temperature reaches about 700-800 F (370-430 C). During sensitization the chromium in the metal migrates to the surface of the metal via grain boundaries, thereby lowering the metal's ability to resist corrosion. On the other hand, some high chromium alloys also containing molybdenum do not undergo sensitization at these temperatures. A ferritic stainless steel, type 409 (11% chromium), was selected by one manufacturer of industrial regenerators over austenitic stainless steels such as type 304 (19% chromium) or 316 (18% chromium, 25% molybdenum) due to its better corrosion resistance at temperatures around 1100 F (590 C). However, a material which has sufficient corrosion resistance for coastal industrial application may not have sufficient corrosion resistance in a shipboard application. Materials which have better corrosion resistance than the stainless steels include such nickel alloys as Inconel 625, Incoloy 800, and Incoloy 825. These materials may be required for marine regenerators to have acceptable life under high-temperature, cyclic, and corrosive conditions.

Resistance to such corrosion improves with increasing chromium content and decreasing molybdenum content, especially in the light of increased fuel costs. Thus, the exchanger sizing and performance, done below, is for plate-fin type.

TABLE 1. COMPARISON OF VARIOUS HEAT EXCHANGER TYPES

<table>
<thead>
<tr>
<th>Type</th>
<th>Advantages</th>
<th>Disadvantages</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. 3 - Rotary Counterflow</td>
<td>1. Minimize flow resistance to achieve large frontal areas (5).</td>
<td>1. Large weight and volume, 70% increase in total weight of transfer area, 2. Additional maintenance problem, 3. Additional losses and carryover (5).</td>
</tr>
<tr>
<td>No. 4 - Liquid-Coupled Indirect Transfer Heat Exchanger (LCITHX)</td>
<td>1. Flexible geometry, 2. Ease of manufacture.</td>
<td>1. Flow and distribution problems (watering), 2. Weight versus c values are higher than tube shell (6).</td>
</tr>
<tr>
<td>No. 5 - Waveplate</td>
<td>1. Flexible geometry, 2. Ease of manufacture.</td>
<td>1. Flow and distribution problems (watering), 2. Weight versus c values are higher than tube shell (6).</td>
</tr>
</tbody>
</table>

5. Waveplate.

The gas turbine cycle conditions are the same as No. 2, i.e., high R and low 0. In addition, the geometry of inlet and outlet streams obtainable with waveplate exchangers is considered more important in this application than the additional weight of the waveplate exchanger.

On the lower temperature end of the heat exchanger, the gas-side exit temperature should not be allowed to go below 275 F (135 C). Near this temperature sulfuric acid will condense from the exhaust gas when operating with a fuel containing 1% sulfur by weight (8). The lowest gas-side exit temperature shown for the hypothetical engine concepts discussed in Part I of this paper was 500 F (260 C) in the low-compression, intercooled, regenerative-cycle engine. At low part-power conditions, regenerator gas-side exit temperature will decrease significantly below 500 F (260 C) unless VATNs are incorporated in the design of the power turbine.

Ceramic materials, aluminum silicate and magnesium silicate, have found application in rotary regenerators and have good life up to 1800 F (980 C) (9). Ceramics are also beginning to appear in stationary exchangers. However, if material selection were made today for marine gas turbine regenerators, it would probably be a nickel alloy based on high-temperature strength and resistance to corrosion. Material cost and the ease of fabrication are also important factors to be considered in early stages of regenerated engine design.
HEAT EXCHANGER SIZING

Two of the three engine cycles presented in Part I will be used to size heat exchangers for regeneration using a plate-fin heat exchanger computer model. These cycles include a low-compression, 8:1, regenerative cycle and a high-compression, 16:1, intercooled-regenerative cycle.

Computer Model

A plate-fin counterflow heat exchanger computer model based on the method and data from Kays and London (1) was used to size the regenerators. A flow diagram for this heat exchanger model is given in Figure 2. The model has the following features and limitations:

1. The model assumes even distribution of flow across the flow cross section; i.e., the mass flow rate through any channel is simply the mass flow for the entire cross section divided by the cross-sectional area of the channel. If this assumption is not true, effectiveness drops rapidly. Even flow distribution into the heat exchanger is not easy to achieve, especially in a short flow length, low pressure drop exchanger (10).

2. Effects of fouling and fin deformation are not included in the model.

3. An additional heat transfer (≈10%) in a counterflow exchanger is obtained from the cross-counterflow in the header. This was not considered in the model. In a pancake-shaped counterflow exchanger, the error in this assumption becomes appreciable.

4. All plate-fin surfaces given in Figures 9-3 to 9-7 of Kays and London (1) are included in the program.

Delta SFC Versus Size Curves

Using the above described computer model for plate-fin heat exchanger hardware, curves have been produced which aid in sizing a regenerator for the typical gas turbine cycles under investigation. (See Figure 3 for example.) These curves plot SFC improvement (over that of an equivalent simple cycle) as a function of regenerator weight at constant lines of regenerator cross-sectional or frontal area. The rationale for using this format for the plots includes the following points:

1. Heat exchanger weight relates to heat exchanger cost. Delivered cost is often just a multiple of weight.

2. SFC improvement (i.e., decrease from simple-cycle fuel consumption obtained by adding a heat exchanger) combines both effectiveness and pressure drop into one number. The data presented below shows SFC improvement only for rated power. To obtain fuel savings over a mission profile requires SFC improvement calculations at off-design conditions. However, the heat exchanger that gives the best SFC improvement at full power will likely also give the best fuel savings over an entire mission profile.

3. Lines of constant cross section are important when the heat exchanger is counterflow. Here the gas side (turbine exhaust) should have the simplest flow channel, which requires putting the heat exchanger in the flow path of the exhaust. The most likely place for the exchanger on a ship is in the bottom of the exhaust uptake. Thus, the cross section or frontal area of the heat exchanger is analogous to deck area which the regenerator will require aboard ship.

4. The simple-cycle point on this graph is at the origin, i.e., at zero heat exchanger weight and zero SFC improvement. Thus, the two curves are easily compared.

Fig. 2 Heat Exchanger Computer Model Flow Chart
5. An ideal heat exchanger would have an effectiveness equal to 100% and zero pressure drop. The SFC improvement resulting from the "perfect" heat exchanger is shown on each plot as a straight line at the top. As this line is approached, additional heat exchanger volume will show little SFC improvement. Thus, this ideal limit line, like the simple cycle baseline, is another convenient reference on the plots.

The plots are presented in English units. To prevent cluttering of the graphs, the conversion to metric is given in Table 2 rather than dual labeling of axes and points. Also, heat exchanger weight will be presented to two significant figures in long tons which is the same number in metric tons since the conversion is 1.016 long tons equals 1.0 metric ton.

Table 2. Conversions, English to Metric Units for Figures 3 thru 10

<table>
<thead>
<tr>
<th>English Units</th>
<th>Metric Units</th>
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<tbody>
<tr>
<td>fins/inch</td>
<td>m/m</td>
</tr>
<tr>
<td>plate spacing (inches)</td>
<td>m (mm)</td>
</tr>
<tr>
<td>area (ft²)</td>
<td>m²</td>
</tr>
<tr>
<td>length (ft)</td>
<td>m</td>
</tr>
<tr>
<td>SFC (lb/hp)</td>
<td>g/kWh</td>
</tr>
<tr>
<td>specific weight (lb/hp)</td>
<td>kg/kW</td>
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<tr>
<td>weight (long tons)</td>
<td>weight (metric tons)</td>
</tr>
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Both gas turbine cycles to be examined have inlet airflow of 100 lb/s (45 kg/s), turbine inlet temperature of 2100°F (1149°C), and operate on an 80°F (27°C) day, with stack pressure losses of 4 and 6 inches (1.0 and 1.5 kPa) of water. Engine type I has an 8:1 compression ratio, and engine type II has a 16:1 compression ratio but with compressor intercooling, which assumes 75°F (24°C) seawater and 90% intercooler effectiveness. The nominal power output from engine Types I and II is 16,000 and 20,000 hp (12 and 15 MW), respectively.

**Engine Type I Analysis**

Figures 3 thru 7 show delta SFC curves (sizing plots) for the heat exchanger of engine type I at rated power at various fin densities and plate spacings described by number in Table 3. All plots are for counterflow exchangers. At any given plate-fin type, one can approximate the optimal heat exchanger size using these delta SFC curves. For example, using fin geometry 1 shown in Figure 3, going from a cross-sectional area of 25 to 50 ft² (2.3 to 4.6 m²) results in a considerable SFC improvement at any given exchanger weight. However, going from 50 to 100 ft² (4.6 to 9.2 m²) shows marginal improvement, which together with the restriction of space available aboard ship, would point toward an exchanger of about 50 ft² (4.6 m²) cross section. The length would be 4 to 6 ft (1.2 to 1.8 m) since the SFC improvement levels out at this point. A 6-ft (1.8-m) length would perhaps give a better flow distribution through the exchanger than 4 ft (1.2 m), and might still fit in the available ship space; however, assuming that 4-ft (1.2-m) length gives a reasonable flow distribution, then a logical choice for this exchanger would be a 50-ft² (4.6-m²) cross section and 4-ft (1.2-m) length with an SFC improvement (clean surface) of 0.12 lb/hph (73 g/kWh) and regenerator weight of 9 tons.

This may not be the best fin geometry for this application. For example, salt deposits on the air side may cause high pressure drop using fin geometry 1 (20 fins/inch air side). If one uses 11 fins/inch there will be less chance of high air-side pressure drop due to fouling. The heat transfer coefficient can be improved (at the expense of some pressure drop) by choosing louvered fins. Pressure drop is not as critical on the air side as on the gas side. On the gas side, one can improve the gas- to airflow area by choosing a wider plate spacing, which will also be less subject to gas-side fouling and pressure drop. This leads to the fin geometries 2 and 3 whose delta SFC curves are given in Figures 4 and 5, respectively. Figure 4 shows that for matrix number 2, an exchanger of 50 ft² (4.6 m²) in cross section and 6 ft (1.8 m) in length would provide an SFC improvement of 0.13 lb/hph.
REGENERATOR WEIGHT (long tons)

Fig. 5 Delta SFC Curves for Engine Type I and Fin Geometry 3

(79 g/kWh) with clean surface at an exchanger weight of 10.5 tons. The slightly improved SFC over that of Figure 3 together with less chance of high-pressure drop due to fouling would favor fin geometry 2 over fin geometry 1. The wider gas-side spacing of fin geometry 3 over that of fin geometry 2 would probably make it a preferred choice between the two geometries due to the reduced chance of gas-side pressure drop problems.

Other fin geometries for engine type I are shown in Figures 6 and 7. Figure 6 is presented because it has the manufacturing simplicity of having the same fin geometry on both air and gas sides. An SFC penalty of about 0.015 lb/hp (9 g/kWh) is the price paid for this manufacturing ease. Figure 7 shows the excellent performance of 46 fins/inch air side with 0.1-inch plate spacing (for geometry 5), but this matrix would be subject to pressure drop degradation and possible flow distribution problems. It is shown here as an example to broaden the sizing picture and not as a necessarily feasible piece of hardware for marine application.

Figure 8 is a comparative summary for engine type I showing various fin geometries at a cross-sectional area of 50 ft² (4.6 m²). The best performance is from the louvered air-side geometries with wider gas-side fin and/or plate spacing (geometries 2 and 3). These fin geometries should give better performance with gas-side fouling than any of the other configurations shown in Figure 8. An economic study of plate-fin counterflow exchangers for engine type I should look at the following fin geometries:

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</table>

Table 3. Fin Geometries of the Plate-Fin Exchangers Examined

<table>
<thead>
<tr>
<th>No.</th>
<th>Fins/In. (fins/mm)</th>
<th>Plate Spacing in. (mm)</th>
<th>Fins/In. (fins/mm)</th>
<th>Plate Spacing in. (mm)</th>
</tr>
</thead>
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<tr>
<td>1</td>
<td>20 (0.79)</td>
<td>0.25 (6.4)</td>
<td>11 (0.43)</td>
<td>0.25 (6.4)</td>
</tr>
<tr>
<td>2</td>
<td>11 (0.43)</td>
<td>0.25 (6.4)</td>
<td>11 (0.43)</td>
<td>0.48 (12.2)</td>
</tr>
<tr>
<td>3</td>
<td>11 (0.43)</td>
<td>0.25 (6.4)</td>
<td>6.2 (0.24)</td>
<td>0.4 (10.2)</td>
</tr>
<tr>
<td>4</td>
<td>11 (0.43)</td>
<td>0.25 (6.4)</td>
<td>11 (0.43)</td>
<td>0.25 (6.4)</td>
</tr>
<tr>
<td>5</td>
<td>46 (1.81)</td>
<td>0.10 (2.5)</td>
<td>12 (0.43)</td>
<td>0.25 (6.4)</td>
</tr>
<tr>
<td>6</td>
<td>20 (0.79)</td>
<td>0.25 (6.4)</td>
<td>11 (0.43)</td>
<td>0.48 (12.2)</td>
</tr>
</tbody>
</table>

Note: (L) - louvered
Engine Type II Analysis

Adding compressor intercooling is attractive only if regeneration is present. Conversely, for this 16:1 compression ratio engine, regeneration now looks economically feasible because of the intercooling. The delta SFC curves presented for the type II engine (Figures 9 and 10) show the intercooler weight as an offset interval from the simple-cycle engine weight. The leftmost line of this interval represents the upper limit of regenerator weight if the total heat exchanger weight (intercooler and regenerator) is to be less than the base engine weight. Figures 9 and 10 show the air-side louvered fin configuration (geometries 2 and 3) that looked promising for engine type I. Again, sizing of the heat exchanger can be approximated from the curves. Looking at Figure 9 (fin geometry 2), an exchanger of 50 ft$^2$ (4.6 m$^2$) and 6 ft (1.8 m) in length appears to be about optimum at an SFC improvement of 0.11 lb/hph (67 g/kWh). Picking the optimal size for fin geometry 3 (Figure 10) is a little tougher since the SFC improvement shows a relatively small increase from 25 ft$^2$ (2.3 m$^2$) and 6 ft (1.8 m) in length to 50 ft (1.8 m) length with almost a doubling of regenerator weight. So a mission profile life-cycle cost study should perhaps include two or more points from this plot.

In general, the sizing calculations resulted in a plate-fin counterflow exchanger of about 50 ft$^2$ (4.6 m$^2$) of cross-sectional area for engines of 100 lb/s (45 kg/s) airflow. The exchangers should be about 4 to 6 ft (1.2 to 1.8 m) long at a weight of 8 to 10 long tons. There are many economic tradeoffs to be examined, one of which is improvement in performance by having different fin and plate spacing on the air and gas sides. Louvered air-side and wide gas-side spacing gives excellent performance with little chance of high gas-side pressure drops due to fouling. By making both air- and gas-side fin geometries the same (for example, 11 fins/inch and 0.25-inch plate spacing), there results a manufacturing ease, but an SFC improvement penalty of about 0.015 lb/hph (9.1 g/kWh). This is one of the economic tradeoffs to be examined.

SUMMARY

The selection of regenerator type depends on the gas turbine cycle parameters and the particular application in which the regenerator is to be used. The advantages and disadvantages of five types of regenerators have been discussed, and application examples are given. Plate-fin counterflow heat exchangers have many advantages which make them attractive candidates for marine regenerators. Tradeoffs between size, weight, and performance of this type of regenerator have been examined using two hypothetical engine concepts. The engine concepts include a low-compression regenerative cycle (engine type I) and a high-compression, intercooled-regenerative cycle (engine type II). Based on results of the heat exchanger sizing analysis, particular designs which provide significant SFC improvement with reasonable volume and weight increases were defined for each engine type. The characteristics of four such regenerators are summarized in Table 4.
Table 4. Characteristics of Selected Plate-Fin Heat Exchangers

<table>
<thead>
<tr>
<th>Engine Type</th>
<th>Regenerator Matrix No.</th>
<th>Frontal Area (ft²)</th>
<th>RX Length (ft)</th>
<th>RX Weight (Long Tons)</th>
<th>SFC Improvement lb/hph (g/kWh)</th>
</tr>
</thead>
<tbody>
<tr>
<td>I 2</td>
<td>50 (4.6)</td>
<td>6</td>
<td>10.5 R 0 IC</td>
<td>0.128 (277)</td>
<td>0.373 (10)</td>
</tr>
<tr>
<td>I 4</td>
<td>50 (4.6)</td>
<td>5</td>
<td>9.0 R 0 IC</td>
<td>0.113 (236)</td>
<td>0.388 (9.1)</td>
</tr>
<tr>
<td>II 2</td>
<td>50 (4.6)</td>
<td>6</td>
<td>9.7 R 0 IC</td>
<td>0.107 (236)</td>
<td>0.320 (67)</td>
</tr>
<tr>
<td>II 3</td>
<td>25 (2.3)</td>
<td>6</td>
<td>5.2 R 3 IC</td>
<td>0.090 (205)</td>
<td>0.337 (10)</td>
</tr>
</tbody>
</table>

Note: RX - heat exchanger, R - Regenerator, IC - Intercooler

The first and third designs shown in this table use regenerator matrix 2. Both have a core volume of 300 ft³ (8.28 m³) and weigh about 10 long tons, but engine type II also has an intercooler which increases the total weight of heat exchangers to 13 long tons for that concept. An SFC improvement of 0.13 lb/hph (79 g/kWh) can be obtained with a plate-fin exchanger of about two-thirds the weight of a base 8:1 compression engine. For a higher compression engine (that is already more efficient than the 8:1 engine), the SFC improvement is 0.11 lb/hph (67 g/kWh), but this also requires compressor intercooling. Both exchangers are louvered on the air side with wide fin spacing on the gas side.

The second design in Table 4 shows the characteristics of a regenerator which has the same fin geometry on the air and gas sides. Besides a slightly lowered SFC improvement, 0.015 lb/hph (9.1 g/kWh), this design would have closer gas-side spacing than the other designs given in Table 4. An economic study is needed to compare the lower manufacturing cost of this design to the slightly lower performance.

The fourth design in Table 4 illustrates how the regenerator matrix can be varied to cause a change in weight and volume of the regenerator. Using regenerator matrix 3 with engine type II results in a 50% reduction in volume to 150 ft³ (4.14 m³) and a total weight of heat exchangers less than 9 long tons. Yet, the difference in SFC between engine type II with regenerator matrix 3 is only 0.017 lb/hph (10 g/kWh) higher than that with regenerator matrix 2. Clearly, regenerator design can be optimized to meet the constraints imposed by the end user with a particular application in mind.

ACKNOWLEDGMENT

The authors wish to acknowledge the contribution made by Mr. John Purnell of the Engines Branch, DTVRDC, for his development of a computer model for estimating the size and performance of plate-fin heat exchangers based on data from Kays and London (1).

REFERENCES