ERICSSON CYCLE GAS TURBINE POWERPLANTS

PREPARED FOR THE U.S. DEPARTMENT OF ENERGY

W. H. KRASE

R-2327-DOE
MARCH 1979
ERICSSON CYCLE GAS TURBINE POWERPLANTS

PREPARED FOR THE U.S. DEPARTMENT OF ENERGY

W. H. KRASE

R-2327-DOE
MARCH 1979
The work upon which this publication is based was performed pursuant to Contract No. EX-76-C-01-2337 with the U. S. Department of Energy. Reports of The Rand Corporation do not necessarily reflect the opinions or policies of the sponsors of Rand research.
This preliminary study of an improved thermal-to-electrical energy conversion cycle arose out of Rand research on two-phase (gas and liquid metal) magnetohydrodynamic (MHD) powerplants, funded by the U.S. Department of Energy (DOE). Such plants have a thermodynamic feature of fundamental importance: They can very closely approximate a constant temperature expansion. The theoretical efficiency advantage of plants of this kind has not yet been demonstrated to be practicable, owing to problems associated with two-phase flow and the MHD conversion device.

The theoretical thermodynamic advantage of these plants, however, can be realized in elaborated gas turbine powerplants that involve neither two-phase flow nor MHD conversion. This type of gas turbine plant, referred to as an "Ericsson cycle approximation," can be adapted to either open- or closed-cycle versions, and to both fossil and renewable heat sources. It can be built in a very wide range of output sizes.

The components and materials required for an Ericsson cycle approximation plant have been extensively demonstrated, but no plant has yet been built that fully exploits the potential of the Ericsson cycle for high thermal efficiency and low capital cost. The purpose of this report is to indicate, in a necessarily preliminary way, what the potential of this cycle is.

This report should be of interest to powerplant researchers, to gas turbine manufacturers, and to DOE agencies dealing with fossil, advanced nuclear, and solar energy conversion.
A preliminary exploration is made of a potentially low-cost gas turbine thermodynamic cycle that appears capable of unprecedented efficiency. The cycle is an approximation to an Ericsson cycle and uses stepwise expansions in turbines with intervening reheat and stepwise compression with intervening intercooling. The cycle also uses a high-effectiveness recuperator. At a peak cycle temperature of 1500°F, and using five stages of compression and expansion, a 50 percent thermal efficiency is attainable with component performance that has already been demonstrated. (Present utility plants have a thermal efficiency in the range of 35 to 40 percent.) At 1800°F, the thermal efficiency reaches 56 percent. This performance is achievable without going to extremes of temperature or pressure, without introducing new materials, and without introducing fundamentally new techniques.

Technically successful plants of this type, using a rudimentary (single reheat stage) approach to the Ericsson cycle, were built in the late 1940s. Since that time, substantial advances have been made in component efficiency and materials properties, so that much higher efficiency can now be realized. Yet the Ericsson cycle has not been considered even in such inclusive studies as ECAS.***

Although the cycle is clearly more complicated than simple gas turbines, it is not complicated in comparison with the advanced combined gas turbine/steam turbine cycles that are being considered for high-efficiency advanced fossil-fuel-fired baseload plants. A representative Ericsson cycle approximation plant requires twice as many rotating shafts as do some already developed advanced industrial gas turbines, about 20 percent more blade rows, fewer individual combustors, a more complex fuel control, intercoolers, and a regenerator. But the

---

* Close to the cost of a regenerative Brayton cycle.
** "Stage" is used to denote an individual component, not a pair of blade rows.
*** The Energy Conversion Alternatives Study, administered by the National Aeronautics and Space Administration.
major improvement in efficiency can justify the additional complication, and the parts that are required in greater numbers have not generally had reliability problems. The cycle has a unique capability to produce high efficiency at relatively low peak temperatures. Sulfidation corrosion and hot-parts distortion and cracking have been the largest contributors to gas turbine unreliability; the low temperature required for the Ericsson cycle should eliminate those problems.

The capital cost of the major components (turbomachinery and recuperator) for an Ericsson cycle approximation powerplant will be of the same order as that of a regenerative Brayton cycle plant because of the high net work per unit flow and the low amount of regeneration required in the Ericsson cycle.

This preliminary assessment of the Ericsson cycle shows that it has excellent potential for achieving high thermal efficiency at moderate capital cost. But this work has not included any investigation of fuels and applications, or of complete design and plant layout, or of detailed cost estimates, or of plant parasitic, electrical, and fuel processing losses. More detailed study in these areas would be desirable.

In addition to its potential use as an open-cycle fossil-fueled central station powerplant using petroleum or coal-derived fuels, the Ericsson cycle, because of its potentially high efficiency at low turbine temperatures, could also be a desirable closed-cycle thermal/electrical converter for advanced nuclear, thermonuclear, and solar energy sources. Such applications also require further study.
<table>
<thead>
<tr>
<th>Section</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>THERMODYNAMICS OF GAS TURBINE CYCLES</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td>Ideal Cycles</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td>Realizable Cycles and Comparisons</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Previous Approaches to the Ericsson Cycle</td>
<td>16</td>
</tr>
<tr>
<td>II</td>
<td>COST ASPECTS OF ERICSSON APPROXIMATION POWERPLANTS</td>
<td>17</td>
</tr>
<tr>
<td>III</td>
<td>THE RELIABILITY ISSUE</td>
<td>22</td>
</tr>
<tr>
<td></td>
<td>Inquiry of Users and Manufacturers</td>
<td>22</td>
</tr>
<tr>
<td></td>
<td>Fundamental Factors Affecting Reliability</td>
<td>25</td>
</tr>
<tr>
<td></td>
<td>Gas Turbine Difficulties in Utility Service</td>
<td>26</td>
</tr>
<tr>
<td></td>
<td>Comparisons of Complexity: Ericsson Cycle Versus Advanced Simple Brayton and Combined Cycles</td>
<td>28</td>
</tr>
<tr>
<td>IV</td>
<td>APPLICATION AND FUELS</td>
<td>33</td>
</tr>
<tr>
<td>V</td>
<td>CLOSING REMARKS</td>
<td>35</td>
</tr>
<tr>
<td>REFERENCES</td>
<td></td>
<td>37</td>
</tr>
</tbody>
</table>
I. THERMODYNAMICS OF GAS TURBINE CYCLES

IDEAL CYCLES

All ideal gas thermodynamic cycles that are capable of the theoretical maximum efficiency (Carnot, Stirling, and Ericsson cycles) necessarily use constant temperature expansion and compression processes. That this is so can be seen from the irreversibility that is introduced if heat is added (or removed) during a variable temperature process. These cycles were originally conceived as piston-engine cycles, and in principle at least, the constant temperature expansion process was realizable.

More recently, two-phase steady flow cycles have been proposed\(^{(1,2)}\) that can produce an essentially constant temperature expansion, either in a turbine or in a magnetohydrodynamic duct, but these have not as yet been fully demonstrated. Aside from this type of two-fluid scheme, no other means of realizing constant temperature expansion and compression in a steady-flow machine is known, despite the prominent position of such processes in theoretical thermodynamics. However, conventional steady-flow machinery can be arranged to approximate constant temperature expansion and compression, as is discussed in the remainder of this report.

Figure 1 shows generic T-S and p-v diagrams for ideal gas turbine cycles, assuming that constant temperature expansion and compression processes can be closely approximated for the Ericsson and Carnot cycles. (The Stirling cycle is omitted, as it is not adaptable to a steady-flow machine.) Also shown are expressions for the thermal efficiency, the net work per unit flow (here made dimensionless in the form \(\frac{W_N}{RT_2}\), where \(W_N\) is the net work, \(R\) the gas constant, and \(T_2\) the minimum temperature in the cycle), and the relative regeneration \(\frac{Q_R}{W_N}\), where \(Q_R\) is the amount of regenerated heat. The net work per unit flow is an inverse measure of the size of turbomachinery required, and

\*That is, having 100 percent efficient components and 100 percent effective recuperators.
<table>
<thead>
<tr>
<th>Simple Brayton cycle</th>
<th>Regenerative Brayton cycle</th>
<th>Isothermal expansion, isentropic compression</th>
<th>Ericsson cycle</th>
<th>Carnot cycle</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image" alt="T-S diagram" /></td>
<td><img src="image" alt="T-S diagram" /></td>
<td><img src="image" alt="T-S diagram" /></td>
<td><img src="image" alt="T-S diagram" /></td>
<td><img src="image" alt="T-S diagram" /></td>
</tr>
<tr>
<td><img src="image" alt="p-v diagram" /></td>
<td><img src="image" alt="p-v diagram" /></td>
<td><img src="image" alt="p-v diagram" /></td>
<td><img src="image" alt="p-v diagram" /></td>
<td><img src="image" alt="p-v diagram" /></td>
</tr>
<tr>
<td>Ideal thermal efficiency</td>
<td>$\eta_{th} = 1 - \frac{1}{\frac{1}{\gamma} - 1}$</td>
<td>$\eta_{th} = 1 - \frac{1}{\frac{1}{\gamma} - 1}$</td>
<td>$\eta_{th} = 1 - \frac{1}{\frac{1}{\gamma} - 1}$</td>
<td>$\eta_{th} = 1 - \frac{1}{\frac{1}{\gamma} - 1}$</td>
</tr>
<tr>
<td>Ideal dimensionless net work per unit flow</td>
<td>$Q_{th}/P_{1}$</td>
<td>$Q_{th}/P_{1}$</td>
<td>$Q_{th}/P_{1}$</td>
<td>$Q_{th}/P_{1}$</td>
</tr>
<tr>
<td>Relative regeneration</td>
<td>$Q_{th}/Q_{in}$</td>
<td>$Q_{th}/Q_{in}$</td>
<td>$Q_{th}/Q_{in}$</td>
<td>$Q_{th}/Q_{in}$</td>
</tr>
</tbody>
</table>

**Fig. 1**—Ideal steady-flow perfect gas thermodynamic cycles
the relative regeneration is a measure of the size of recuperator required per unit output.

Figures 2, 3, and 4 show the cycle thermal efficiency, net work, and relative regeneration as a function of cycle pressure ratio (for monatomic gases, $\gamma = 5/3$) for a temperature ratio

$$\frac{T_4}{T_2} = 3.698 \quad (T_4 = 1500^\circ F, \ T_2 = 70^\circ F).$$

The various cycles are seen to require operation in different pressure-ratio regimes. Two specific pressure ratios are of special interest: (1) the pressure ratio at which the isentropic temperature ratio is equal to the assumed maximum cycle temperature ratio (this is the maximum pressure ratio allowed for a simple Brayton cycle, and the minimum pressure ratio allowed for a Carnot cycle), and (2) a pressure ratio that is the square root of the first, at which the turbine exhaust temperature of a Brayton cycle equals the compressor

---

![Diagram](image_url)

**Fig. 2**—Thermal efficiency versus pressure ratio: Ideal cycles, monatomic gas
Fig. 3—Net work per unit flow versus pressure ratio: ideal cycles, monatomic gas

Fig. 4—Relative regeneration versus pressure ratio: ideal cycles, monatomic gas
discharge temperature. This second pressure ratio is the maximum possible pressure ratio for a regenerative Brayton cycle and is the ratio at which the net work is a maximum for Brayton cycles (both simple and regenerative). Both simple and regenerative Brayton cycles can be made to approach the Carnot cycle efficiency (73 percent for the temperatures considered in Figs. 2, 3, and 4), but the net work of the cycle approaches zero at the same time.

The Ericsson cycle can operate at any pressure ratio at the ideal efficiency. Like all gas turbine cycles, its net work per unit flow is zero at a pressure ratio of unity, but it increases continuously as pressure ratio increases. At any pressure ratio, it has the highest net work of any cycle shown.

The ideal isothermal expansion, isentropic compression cycle, which approximates the two-phase-flow liquid-metal MHD cycles proposed in Ref. 2, lies as expected between the regenerative Brayton cycle and the Ericsson cycle, in efficiency, net work, and relative regeneration.

Finally, as a matter of completeness, the Carnot cycle is shown. It must have a pressure ratio at least equal to

\[
\left( \frac{T_4}{T_2} \right)^{\frac{\gamma}{\gamma-1}}
\]

(at which the net work per unit flow is zero), and as the pressure ratio is further increased, the net work increases continuously. It is the only cycle capable of the maximum thermal efficiency that does not require regeneration by heat exchange (in effect, regeneration is done by the isentropic compressor and turbine, and the net external work is due to the isothermal expansion and compression).

The fact that the Ericsson cycle has high work per unit flow and low relative regeneration at a high pressure ratio is of great importance, as will be seen in the following.

REALIZABLE CYCLES AND COMPARISONS

The sensitivity of gas turbine cycles to component efficiencies is well known, and before drawing any firm conclusions about the various cycles considered, we must introduce realistic assumptions as to
the efficiency of their components. For the two-phase MHD isothermal/isentropic cycle, the expander efficiency is treated parametrically.

Because we wish to compare different powerplant types over a wide range of pressure ratios, we have adopted a "small stage" approach to the efficiency of turbomachinery components, thereby at least approximately accounting for the expected drop in compression efficiency (and rise in turbine efficiency) as the pressure ratio becomes large. The resulting compressor and turbine efficiencies are shown in Fig. 5. It is more difficult to specify reasonable (and ultimately economic) heat-exchanger characteristics. For regenerators, we have been guided by experience with experimental units, but it is likely that some further improvement could be made. For regenerators, we assume an effectiveness $\varepsilon = 0.85$ and a pressure ratio of the gas on each side of 0.98. For heaters, we assume a pressure ratio of 0.98 on the gas side (for each one, if the cycle involves more than one), and for coolers, a

![Graph](image)

**Fig. 5**—Assumed variation of compressor and turbine efficiency with pressure ratio
pressure ratio of 0.99 on the gas side. No allowance has been made for leakage, for fuel addition, or for bearing or windage losses. For the Ericsson cycle approximation, it is assumed that the inlet temperature to each stage of compression is the same as the initial gas temperature. In effect, this assumes, for open cycles, a cooling water temperature somewhat below the ambient air temperature—a common situation, especially for coastal plants. These values of pressure drop and effectiveness have been realized in a small experimental closed Brayton cycle powerplant.(4) Other values clearly are possible. A uniformly poorer level of component efficiency and pressure drop will reduce the thermodynamic efficiency of all cycles, but the advantages of the Ericsson cycle will still be present.

Figures 6 through 8 show the resulting performance for the various cycles, still using a peak cycle temperature of 1500°F and a minimum of 70°F, and still using a monatomic gas ($\gamma = 5/3$).

The best regenerative Brayton cycle has about six points better efficiency than the best simple Brayton cycle. At best efficiency,

![Diagram](image)

**Fig. 6** — Thermal efficiency versus pressure ratio; cycles with component losses, monatomic gas.
Fig. 7—Net work per unit flow versus pressure ratio: cycles with component losses, monatomic gas.

Fig. 8—Relative regeneration versus pressure ratio: cycles with component losses, monatomic gas.
both Brayton cycles have a net work \((\dot{W}_N/RT_2) \approx 1.2\). The regenerative Brayton cycle at best efficiency has a relative regeneration \((Q_R/\dot{W}_N) = 2\).

The use of a two-phase flow during the expansion process is one way in which the ideal Ericsson cycle can be approximated, but it can also be approximated with ordinary turbomachinery. By using a series of turbines and reheaters to approximate a constant temperature expansion, and a series of compressors and intercoolers to approximate a constant temperature compression, the Ericsson cycle can be approached quite closely. It is a nice problem of design and engineering judgment to determine an optimal arrangement, but in view of the other complications that have been introduced in order to increase the thermal efficiency of a powerplant, it seems fully justifiable to consider from three to five stages of expansion (and compression) as reasonable numbers. The cycle then is as sketched in Fig. 9. In the calculations made here, we have assumed that the compressors all have the same pressure ratio \(n \sqrt{P_3/P_2}\), where \(n\) is the number of compressors in series; we have made the same assumption for the turbines as well, with allowance for the heat-exchanger pressure losses. This is not necessarily a realistic arrangement, but it has the virtue of being simple to compute, and it is probably near the ideal arrangement. Figures 6 through 8 show the resulting performance for three-stage and five-stage approximations to an Ericsson cycle. For a five-stage system, the efficiency reaches 50 percent and is relatively insensitive to pressure ratio. At a pressure ratio of 10, the dimensionless net work per unit flow \((\dot{W}_N/RT_2)\) is 3.5 (almost three times that of a Brayton cycle) and the relative regeneration \((Q_R/\dot{W}_N)\) is 1.4, substantially less than for the regenerative Brayton cycle.

The improvements in cycle efficiency and size parameters that are made possible purely by elaborations of the cycle in order to approximate an ideal Ericsson cycle are very significant. It is notable that improvements in efficiency in Ericsson cycles (unlike regenerative Brayton cycles) do not result in a decrease in specific output, nor do they require more regeneration in relation to the output. These characteristics make the Ericsson cycle a "robust" candidate for future
powerplants; i.e., the potential gains in performance are not likely to be eaten up by miscellaneous losses.

As a practical matter, if such a powerplant is to use fossil fuel, it should be an open cycle using air as the working fluid rather than an inert gas. No heat exchanger is then required to operate at the peak cycle temperature, and the "stack loss" of heat that never gets into the cycle is avoided. Such an arrangement requires only one gas-to-gas heat exchanger—the recuperator. The intercoolers reject heat to water and can be relatively compact. A possible cycle arrangement is sketched in Fig. 10. A coaxial arrangement is also possible, having one or multiple rotors, with annular coolers and combustors in line with the turbomachinery. The performance of such systems for ideal
diatomic gas ($\gamma = \frac{7}{5}$) with constant specific heat* at peak cycle temperatures of 1500°F and 1800°F is shown in Figs. 11 through 13. In principle, the number of stages can be increased until all of the oxygen in the air is used up, with further improvements in cycle efficiency. The cases shown are well short of that situation. Specific fuels and their detailed characteristics in terms of fouling as well as heating value need to be considered in detail.

The Ericsson cycle approximation can be achieved either from the starting point of an ideal Ericsson cycle, as above, or by an

---

*The effect of the real variation of specific heat is expected to be small and favorable.
Fig. 11—Thermal efficiency versus pressure ratio: Ericsson cycles with component losses, diatomic gas

Fig. 12—Net work per unit flow versus pressure ratio: Ericsson cycles with component losses, diatomic gas
alternative path starting from a regenerative Brayton cycle and elaborating it. This second approach is particularly valuable in achieving insight into the effect of the number of stages * on efficiency, into the change in recuperator pressure loading, and into the effect of adding stages on the cost.

Consider first a low-pressure-ratio regenerative Brayton cycle engine, for which the temperature-entropy diagram is sketched in Fig. 14. This type of engine has been and is being used for pipeline pumping and for load peaking, so that we are assured that it is viable and relatively inexpensive, compared to base-load plants. The modified Ericsson cycle can be regarded as an elaboration of a regenerative Brayton cycle in order to improve its efficiency. Adding a stage to the Brayton cycle of Fig. 14 with the same pressure ratio as the original one, to get the cycle of Fig. 15, results in an increment of net work per unit flow equal to the net work of the original cycle.

* "Stage" is used here to denote an entire compressor between other components, rather than in the sense of a pair of blade rows.
From Fig. 15, it can be seen that the heat added is less than doubled for this new cycle, so the efficiency is improved. And the amount of heat transferred in the recuperator is unchanged, although the output is doubled. The pressure differential across the recuperator is increased, so the heat-transfer surface must support higher pressure loads. But because most of the resistance to heat transfer is in the film coefficients on each side, and because the increased pressure on the high-temperature side will tend to decrease the film resistance, the actual heat transfer area required will decrease slightly. Because
the gas density is higher in the topping stage, the turbomachinery is smaller, and the cost per unit output is less.

Adding stages with intercooling and reheat to a regenerative Brayton cycle in order to approximate an Ericsson cycle results in increasing the output in proportion to the number of stages. It is possible to view each stage as having a thermodynamic efficiency. The efficiency of one of the stages is the same as that of the original regenerative Brayton cycle, but the efficiency of each of the added stages is equal to that of a Brayton cycle with a 100 percent effective regenerator, although there is no regenerator present in the added stages. The overall efficiency is (for the equal stage pressure ratio case discussed) equal to the arithmetic average of the efficiency of all the stages.

From this viewpoint, the high efficiency of the Ericsson cycle approximation is seen to result from averaging one stage of relatively low efficiency with several more stages of higher efficiency. The first added stage will result in the largest increment in overall efficiency, and adding further stages results in approaching a limiting efficiency more and more closely. There is no thermodynamic limitation to the number of stages or overall pressure ratio that can profitably be used. Practical limitations will, of course, be encountered due, for instance, to the pressure differential that can be sustained by the recuperator, or simply because the increment in efficiency (but not in output) for an additional stage will ultimately become very small. For open cycles, a limit due to oxygen exhaustion will be reached. The examples shown are short of that situation. More detailed study is required to define such practical limitations. The turbomachinery cost of additional stages (as discussed in the next section) does not appear to be limiting, nor does the recuperator pressure differential, in the region of overall pressure ratio explored here. But the recuperator pressure differential will require that the recuperator be specifically designed for use in an Ericsson cycle.
PREVIOUS APPROACHES TO THE ERICSSON CYCLE

Previous approaches to an Ericsson cycle—notably that by Brown-Boveri in 1948—have been made with gas turbine plants having two intercoolers, a single reheater, and a recuperator. But those plants operated at a very low peak temperature (600°C, 1112°F) and low pressure ratio and were only marginally competitive with alternatives, even though they had thermal efficiencies over 30 percent, a remarkable achievement for the time. Plants of this type were built in Great Britain in the 1950s.

In addition to the high thermal efficiency potential that has been known (if not exploited) for some time, the Ericsson cycle approximation will reduce (relative to a Brayton cycle) the size and cost per unit output of the principal components: turbomachinery and regenerator. Developments since the early plants that further improve the prospects for Ericsson cycles are extended-surface heat exchangers, high-temperature materials, and many improvements in the performance of combustors, turbomachinery, and seals. High reliability of multiple-rotor gas turbine plants has also been demonstrated.

In addition to the early Brown-Boveri and British work on Ericsson cycle approximations, which was limited by the low temperatures and pressures used, a more recent Russian-built powerplant installed in Hungary includes some but not all of the technical features proposed here.* This plant uses a nonregenerative cycle but has two separate rotors with a single intercooler and reheater burner. Its thermal efficiency (28 percent) is accordingly not as high as that of the earlier Brown-Boveri plants despite a higher peak temperature, but it does exploit the potential of staged, intercooled systems to make high-pressure-ratio machines at low cost. It is also a demonstration of acceptance of a more complicated cycle in a utility environment.

*We are indebted to Dr. Arthur Cohen of the Electric Power Research Institute for bringing this plant to our attention.
II. COST ASPECTS OF ERICSSON APPROXIMATION POWERPLANTS

The capital cost of Ericsson cycle powerplants is naturally a major concern. The approach we have taken to assessing that cost is to compare the cost of the major components of such a plant with those of a regenerative Brayton cycle plant, known already to be viable in some applications. We adopted this approach because of the preliminary nature of this investigation and the limited resources available. No complete cycle arrangement has been determined as yet, and no total plant layout has been attempted. We are concerned here purely with the conversion cycle, not with fuel type and processing costs, electrical equipment, etc. Therefore we have made no attempt to arrive at a total capital cost of a plant. But this limited approach does have the advantage of "transparency," that is, the costs are related explicitly to those of an existing type of plant. This is a virtue often lost in the elaboration of a more complete cost analysis. In addition, the present cost comparisons are independent of powerplant size.

The modified Ericsson cycle consists of components that (with a few exceptions) have been thoroughly demonstrated and built in a wide range of sizes and for many applications. The principal exceptions are the intercoolers, the combustors (which must operate without the availability of low-temperature cooling air), and the transfer casings. The fuel supply system, if based on coal gasification, has not been fully commercialized, but that shortcoming applies to combined cycles as well. Only the conversion cycle proper is considered here; fuel treatment and pollution control are outside the scope of this discussion. The lack of experience with and cost data on intercoolers, reheat combustors, and transfer casings precludes all-inclusive cost estimates. But it is possible to show that the major components (turbomachinery, recuperator) should be no more expensive per unit output than those of a regenerative Brayton cycle.

The cost of the turbomachinery for an Ericsson cycle is most readily compared to that of a regenerative Brayton cycle by referring
back to Figs. 14 and 15, which show that adding an upper stage of the same pressure ratio as the original one will double the output. The added stage will be smaller than the original one because of the higher density of the airflow entering its compressor. Suppose that each stage has a pressure ratio of 2.0. * Because the intercooler reduces the inlet temperature of the second stage to the same value as that of the first stage, the air density at the second stage inlet is exactly twice that at the first stage, and, for aerodynamically similar compressors, the second stage will have one-half the frontal area (or one-half the corrected airflow) of the first stage. The result is that an additional stage produces the same output as the original stage, with turbomachinery one-half the size. And a third stage will be one-half the size of the second or one-fourth the size of the first but will put out the same net work as each of the previous stages. The turbomachinery cost per unit output of successive stages is therefore expected to improve, as is the cost per unit output of the assembly.

It is known (on the basis of analysis of historical data) that the cost of turbomachinery is not directly proportional to the airflow, but rather increases less rapidly than airflow. Statistical data are usually reduced to the form

\[ \text{cost} \approx (\text{airflow})^n, \]

where \( n \) is a statistically determined exponent. Cost studies of aircraft turbine engines, ** which cover all the engine elements, including accessories and controls, have variously estimated \( n \) to be from 0.6 to 0.8. Other cost studies for commercial gas turbines, (8) although not in precisely equivalent form, suggest an exponent \( n \) of roughly 0.6 to 0.7. The cost implications of exponents ranging from

---

* No present open regenerative cycle has this low a pressure ratio. We have used this value because the Ericsson cycle optimizes with a large number of stages. The cost of such a low-pressure-ratio cycle is not expected to differ drastically from that of present engines.

** Published in Ref. 7 and in classified Rand Corporation reports.
1.0 to 0.6 are shown in Table 1, as they apply to the machinery and burner elements of an Ericsson cycle approximation powerplant. This table is based on individual stage pressure ratios of 2.0 and does not reflect the fact that the first Ericsson cycle stage is less expensive than the higher-pressure-ratio stage of a regenerative Brayton cycle. Because adding stages to the Ericsson cycle increases output, it is compared to a Brayton cycle of the same output.

### Table 1

EFFECT OF COST SCALING EXPONENT (n) AND NUMBER OF STAGES IN ERICSSON CYCLE APPROXIMATION ON THE RELATIVE COST OF TURBOMACHINERY

<table>
<thead>
<tr>
<th>Cost Item</th>
<th>Number of Stages</th>
<th>Relative Output/Unit Flow</th>
<th>Relative Turbomachinery Size</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1</td>
<td>2</td>
<td>3</td>
</tr>
<tr>
<td>Relative turbomachinery cost</td>
<td>1</td>
<td>1.5</td>
<td>1.75</td>
</tr>
<tr>
<td>Relative cost/unit output</td>
<td>1</td>
<td>0.75</td>
<td>0.58</td>
</tr>
<tr>
<td>Relative cost/unit output of Brayton cycle of same output</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>

Cost Scaling Exponent n = 1.0

<table>
<thead>
<tr>
<th>Cost Scaling Exponent n = 0.8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Relative turbomachinery cost</td>
</tr>
<tr>
<td>Relative cost/unit output</td>
</tr>
<tr>
<td>Relative cost/unit output of Brayton cycle of same output</td>
</tr>
</tbody>
</table>

Cost Scaling Exponent n = 0.6

<table>
<thead>
<tr>
<th>Cost Scaling Exponent n = 0.6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Relative turbomachinery cost</td>
</tr>
<tr>
<td>Relative cost/unit output</td>
</tr>
<tr>
<td>Relative cost/unit output of Brayton cycle of same output</td>
</tr>
</tbody>
</table>
It can be seen from Table 1 that the turbomachinery costs for the Ericsson cycle powerplant can be expected to be lower than those for the less efficient regenerative Brayton cycles in all cases, except those with both the lowest scale exponent \( n = 0.6 \) and fewer than five stages. Even in these cases, the turbomachinery costs are fairly close.

The cost of the recuperator—the other major cost contributor—is expected to be much lower than that for a regenerative Brayton cycle, because the heat transferred per unit flow is essentially the same, the effectiveness is the same, and the output per unit flow is much larger. It is true that the regenerator must withstand a higher pressure differential than that of the regenerative Brayton cycle, but the surface area is somewhat smaller and the loading is not extreme.

We have not taken into detailed account the additional costs in an Ericsson cycle of such components as the transfer casings and ducts or the intercoolers. Presently, we have no statistical body of cost information for such components. However, they are unlikely to be major cost elements in an Ericsson cycle approximating a powerplant. Certainly, they are fundamentally more compact and/or less heavily loaded than the corresponding components (condensers, exhaust hoods) in a steam plant, so their costs cannot be prohibitive.

These arguments are intended to show that the Ericsson cycle modification proposed here is, despite its complexity, potentially cost-competitive with a regenerative Brayton cycle. The regenerative Brayton cycle in turn has lower capital cost than a modern steam plant, as is demonstrated by the procurement of Brayton cycles for peak-load duty despite their lower efficiency and higher fuel cost. We have already seen that the modified Ericsson system is potentially more efficient than present steam plants or combined cycle (gas turbine/steam turbine) plants. It therefore can be expected to find many applications. Its use as a coal-fired base-load powerplant requires full commercialization of integrated coal gasification systems, as does the use of combined cycles. For load-peak applications, where coal gasification systems may be impractical for short period use, or their cost may be prohibitive, the modified Ericsson system plant using liquid fuel will give both a low capital cost and outstanding efficiency.
This discussion of capital cost relates to procurement costs of developed systems. The development cost of an Ericsson approximation cycle is a substantial consideration as well. For the illustrative five-stage plant, five different-sized compressors would be required. These compressors would be simpler than the usual gas turbine compressor, and the total number of blade rows in all compressors would only be about 20 percent more than in a Brayton cycle engine, but nonetheless they would require more tooling costs. The development cost, particularly for a demonstration plant, could be made relatively small by the modular use of identical compressors in a series and parallel arrangement. In such an arrangement with a stage pressure ratio of 2, two compressors in the first stage would feed a single compressor in the second stage, etc., and all compressors could be identical. The bending loads on the blades of the highest stage are, of course, higher than those on the lower stages, so that for identical components, the blade bending strength (as well as flutter resistance and casing strength) is dictated by the highest stage. Individual compressors in lower stages will be overdesigned in these respects, but that represents only a small cost or performance penalty. Many arrangements are possible. Very large numbers of compressors can be avoided by using, for example, two sizes of compressor in a modular arrangement. There are also possibilities for building a wide range of plant sizes with a few building-block components. All these possibilities need further detailed study.
III. THE RELIABILITY ISSUE

The Ericsson cycle approximation is fundamentally more complicated than a Brayton cycle. Complication is an important issue because it may result in poorer reliability or availability. The question addressed in this section is whether the reliability of an Ericsson cycle is likely to be poor enough to preclude its use in utility applications. The problem is pressing because utility experience with simple-cycle gas turbines has not been uniformly good.

To address this question, we summarize below a brief inquiry of users and manufacturers, discuss fundamental reasons for poor reliability, discuss the major types of trouble met in operation, and make a comparison of machinery complication for an Ericsson cycle and some existing gas turbines and proposed combined cycles for industrial use. Finally, we attempt to draw inferences about the probable reliability of an Ericsson cycle plant.

INQUIRY OF USERS AND MANUFACTURERS

We found early in this effort that the Electric Power Research Institute has under way a major survey of gas turbine reliability in utility applications, and we have made no attempt to anticipate their results. The existence of that major study effort is evidence that detailed and comprehensive data on gas turbine reliability are not now available. Our limited inquiry confirmed this. No single number should be quoted for reliability, availability, or maintenance cost. Experience varies widely as a function of type of service, air quality and treatment, fuel type and treatment, amount and type of inspection and preventive maintenance, and engine type. There are also major differences among values quoted by users and manufacturers. (This is not due to deception but rather to the natural tendency of users to quote numbers for the entire gas turbine plant, including, for example, fuel treatment and supply provisions and diverter valves [dampers] for the exhaust flow, whereas the gas turbine manufacturers tend to consider only failures associated with their equipment, and costs for
replacement parts which they supply.) Inspection and preventive maintenance measures vary widely. Definitions of reliability, availability, and maintenance cost, and the completeness of reporting were also found to vary, even among the limited contacts that we made. Reliability is sometimes used in the sense of probability that a machine will start when required; or it may refer to the probability that a machine will function within specified limits for a specified time. The time specified is seldom quoted in connection with reliability numbers, and there appear to be no standards. Neither is reliability often associated with a specified preventive maintenance program. Some reliability analyses assume a mean-time-between-failures (MTBF) and a specified operating time, leading to an equation of the form

\[ R = e^{-\frac{t}{\text{MTBF}}}. \]

For a given MTBF, the reliability can vary widely, depending on the operating time specified. But this type of analysis makes the tacit assumption that failures occur randomly in time, whereas it is known that the most important failure modes are dependent on operating time. The relevance of this type of analysis is therefore questionable.

Availability of powerplants is calculated on a more nearly uniform basis. It is defined as

\[ \frac{\text{time operating} + \text{time available for operation}}{\text{calendar time interval}}. \]

For a given engine, availability is known to depend on maintenance program and on fuel type.

A survey by the Edison Electric Institute\(^{(9)}\) covering all types of utility plants for the ten-year period 1967-1976, summarizes the operating availability of the various plant types as follows:

---

\(^*(\text{Time available for operation excludes the time devoted to unscheduled maintenance and repair and, usually, scheduled maintenance as well.})\)
<table>
<thead>
<tr>
<th>Plant Type</th>
<th>Operating Availability (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fossil (i.e., steam)</td>
<td>84.57</td>
</tr>
<tr>
<td>Nuclear</td>
<td>72.24</td>
</tr>
<tr>
<td>Gas turbine</td>
<td>86.31</td>
</tr>
<tr>
<td>Jet engine</td>
<td>86.45</td>
</tr>
<tr>
<td>Diesel</td>
<td>94.96</td>
</tr>
<tr>
<td>Hydro</td>
<td>95.48</td>
</tr>
<tr>
<td>Pumped storage</td>
<td>86.54</td>
</tr>
</tbody>
</table>

This tabulation shows that the availability of gas turbine powerplants has been slightly higher than that of fossil (steam) plants. The data of the Edison Electric study do not include maintenance cost, nor do they give significant detail (other than identifying the particular component that gave difficulty) on the nature of the problems. The data do show that combustor and turbine hot parts experience the most serious difficulties, with control and instrumentation problems being more numerous but less time-consuming.

Maintenance operations are handled in a variety of ways: Some utilities have much of their maintenance done by suppliers, others do it in-house. Replacement parts have varied widely in cost, and different accounting procedures further obscure maintenance cost issues.

The situation described can be considered representative of an "immature" application of a technology. The application of gas turbines to utilities can be considered immature in other respects as well, as we shall see. An independent study\(^{10}\) concluded that gas turbines in industrial plants (with and without heat recovery) are quite reliable, but that combined cycles for utilities have not performed as well. This was attributed in part to utility plants being designed in a hurry, using gas turbines rated in excess of their capacity and marketed before being fully developed. In some cases, gas turbines designed for load-peaker service have been used for mid-range and base-load applications. Many of the difficulties with combined cycle plants were attributed to poor design of the steam side components. That same study\(^{10}\) assumed a forced-outage rate of 10 percent for a combined cycle plant and 6.7 percent for a steam plant. Nonetheless, the combined cycle, because of its better efficiency, was
preferred. Representative availabilities for modern gas turbine engines, as quoted by two manufacturers on the basis of operating experience, are in excess of 99 percent.

**FUNDAMENTAL FACTORS AFFECTING RELIABILITY**

Differences between the reliabilities of gas turbine plants and those of steam plants are due, in a fundamental sense, to exposure of the hot, highly stressed, rotating parts of a gas turbine to a partially uncontrolled environment. Some early gas turbine plants had little or no inlet air filtration and little control over the sulfur and trace metal content of the fuel. Sulfidation corrosion on hot parts became troublesome when turbines with hot-parts temperatures over about 1500°F were introduced. Inlet air filtration, close control of fuel specifications, fuel treatment, and hot-parts cooling and coating have all been introduced as measures against sulfidation, but it continues to be a problem for high-temperature gas turbines using liquid fuels.

In addition to hot-parts corrosion, erosion and dirt deposits in cold as well as hot engine parts are also troublesome. Air filtration and periodic cleaning (without disassembly) have been introduced to control these problems.

These fundamental problems with gas turbines may be likened to the difficulties experienced by steam plants before feedwater treatment and control were developed to their present state.

Closed-cycle gas turbines have long been of interest because of their potential for avoiding this class of difficulties. But for fossil-fuel applications, at least, the problems of the high-temperature heat source and heat-source heat exchanger have generally appeared to be even more daunting.

Difficulty with hot-parts corrosion is associated with high gas and metal temperatures, which have been considered necessary in order to improve gas turbine efficiency. The Ericsson cycle approximation offers a way to improve efficiency without increasing temperature, thus bypassing that fundamental problem.
GAS TURBINE DIFFICULTIES IN UTILITY SERVICE

A wide variety of difficulties have been experienced with gas turbines in utility service. Most prominent is the sulfidation corrosion of hot parts (discussed above), which at one time, at least, accounted for over 50 percent of the maintenance costs. Other types of difficulties are discussed briefly below.

Bearings

Most difficulties with bearings have been experienced on aircraft-derivative engines using rolling element bearings. In one case, two ball thrust bearings were supposed to share the load, and an out-of-tolerance condition permitted one bearing to carry all the load. The bearing life was limited. Close control of installation tolerances cured the problem. This is perhaps typical of bearing problems—once diagnosed, they are susceptible to permanent cure. The hydrodynamic bearings typical of heavy industrial gas turbines have given very little difficulty.

Hot Parts

Failures of hot parts, such as combustion liner cracks or stator blade warping or cracks, are probably next in importance to sulfidation corrosion as a source of difficulty. The detailed cause of such troubles has to be analyzed for each case. Often the difficulty is associated with "low-cycle fatigue," i.e., with high stresses induced by thermal gradients especially during startup and shutdown. These difficulties tend to be most serious in high-temperature machines, and to be relieved by slow transients. In other cases, cracking can be due to an unanticipated vibration mode—for instance, in turbine stator vanes—and can be eliminated by adjustments to mass or stiffness. Combustion liner cracks may result from any of several causes, but they are especially sensitive to minor changes in fuel spray or secondary air distribution. They are difficult to anticipate analytically but relatively easy to repair and to forestall, given the opportunity for redesign.
All these hot-parts difficulties become more pressing as the firing temperature is increased. There is no guaranteed method to avoid them other than a substantial development test program. Aircraft-engine-derivative gas turbines have had the benefit of much more of this type of development than have heavy industrial gas turbines. Despite their higher firing temperature and very low weight, the aircraft-derivative engines are essentially on a par with industrial turbines as far as maintenance costs are concerned. (11)

**Fuel System and Combustors**

Nearly all present gas turbines use multiple (6 to 10) fuel injectors and burner cans. Industrial gas turbines sometimes use separate retractable igniters in each burner can to avoid temperature nonuniformities in the wake of spark plugs or cross-over-ignition tubes. In addition to the multiplicity of components that can affect reliability, very precise tolerances on the flow rate and spray characteristics of the fuel nozzles are required in order to avoid temperature maldistributions. Some difficulty has been experienced due to nonuniform fuel-nozzle flow developing during operation, and recalibration of fuel nozzles is required periodically. In this respect, the Ericsson cycle will have an advantage, because a single burner can per stage (for a total of 3 to 5) can be incorporated in the transfer ducting.

On the other hand, the fuel flow to each Ericsson cycle combustor must be individually controlled, so the fuel control itself will be substantially more complicated.

Hydromechanical fuel controls have been highly developed and are a relatively minor source of operating difficulty. The new electronic controls for utility gas turbines, however, have caused some difficulty because of unreliability of electronic components.

An Ericsson cycle will require fewer burners, fuel nozzles, and igniters than are commonly used in industrial gas turbines. The fuel control will be more complex than for a simple cycle, however, and remains to be worked out.
Many additional problems have surfaced in the mating of gas and steam turbines in a combined cycle. In many cases a diverter valve, or damper, is provided for the gas turbine exhaust, so that the gas turbine can be run without heat recovery in the steam cycle. Many of the early diverter valves encountered mechanical difficulty. In some cases, supplementary firing is used either to increase the heat available to the steam plant or to make it possible for the steam plant to operate independent of the gas turbine. Provisions for supplementary firing have also caused serious difficulties. But these problems have proved susceptible to cures, and in any case they do not reflect on the reliability of the gas turbine plant proper, which is our present concern. They are symptomatic of the then-relatively-immature state of combined cycle design, and of the complexity of such designs.

This discussion of gas turbine difficulties should not be interpreted to imply that these plants have been plagued with lots of failures. Most of the difficulties have found solutions in the natural course of events, which is to be expected when new technology is introduced into a demanding and different application. It is reasonable to expect future performance to improve, as has been shown possible in some plants. The capability of the Ericsson cycle to achieve very high efficiency at relatively low firing temperatures will tend fundamentally to improve reliability.

COMPARISONS OF COMPLEXITY: ERICSSON CYCLE VERSUS ADVANCED SIMPLE BRAYTON AND COMBINED CYCLES

A comparison of the complexity of an Ericsson cycle with that of advanced heavy industrial gas turbines suggests that the Ericsson cycle complexity is not unreasonable. Some industrial engines, in order to improve efficiency, have quite high pressure ratios. The Stal-Laval (Swedish) GT35 and the developmental Turbo Power & Marine (UTC) FT-50 are heavy industrial machines with great emphasis on reliability. Nonetheless, they have three independent, separately supported rotors in a coaxial arrangement with a dual rotor compressor driven by concentric shafts, and a separate power turbine. The FT-50 has 17 compressor stages (pairs of blade rows), four turbine stages, eight journal bearings, three thrust bearings, and eight combustor cans with individual ignition provisions.
Although the scope of this study has not permitted selection of a powerplant application and layout for an Ericsson cycle plant, a representative plant can be taken, for purposes of discussion, to involve five compressors in series, each having a 2:1 pressure ratio, five compressor drive turbines, a separate power turbine, four intercoolers, five combustion chambers, and a recuperator. This plant, then, would have six independent shafts with (probably) twelve journal bearings and six thrust bearings. The number of compressor stages in each compressor would be (depending on the design margins sought) 4 to 5, for a total of from 20 to 25 stages. This Ericsson plant has twice the number of rotors and 20 to 40 percent more compressor blade rows than the FT-50. (That the number of blade rows can be kept relatively small is a consequence of the intercoolers and the multiple shafts of the Ericsson cycle, features which also make possible higher compressor efficiency.) The Ericsson cycle also requires intercoolers and a regenerator not present in the FT-50, and a more complex fuel control. The gain in thermal efficiency from about 35 to 50 percent seems fully to justify the additional complication.

The noncoaxial shaft arrangement suggested in Fig. 10 allows some significant simplification of bearing and seal arrangements and of provisions for maintenance access. The turbomachinery rotors can be short enough to be readily supported on two bearings, and the number of oil seals can be greatly reduced. Shaft shortness and stiffness will aid in reducing air-seal clearances and leakage. The separate compressors, with their independent speeds, can be designed to optimize efficiency, rather than having the high-pressure-ratio stages constrained to inefficient proportions.

The complexity of the Ericsson cycle plant, as measured by a count of the major components, is not as great as that of a combined cycle, taken from ECAS* reports (Fig. 16). Table 2 shows the count of major components: 29 for the combined cycle, and 26 for the Ericsson cycle of Fig. 10. The combined cycle has 7 major rotating components, and a total of 12 rotating components including pumps (but neglecting

---

*The Energy Conversion Alternatives Study, administered by the National Aeronautics and Space Administration.
Table 2

COMPONENT COUNTS OF ERICSSON AND COMBINED CYCLES

<table>
<thead>
<tr>
<th>Combined Cycle</th>
<th>Ericsson Approximation Cycle (Fig. 10)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 Brayton cycle turbine</td>
<td>5 Compressors</td>
</tr>
<tr>
<td>1 Brayton cycle compressor</td>
<td>8 Turbines</td>
</tr>
<tr>
<td>1 Brayton cycle generator</td>
<td>7 Combustors</td>
</tr>
<tr>
<td>1 Brayton cycle cooling air cooler</td>
<td>4 Intercoolers</td>
</tr>
<tr>
<td>1 Brayton cycle combustor</td>
<td>1 Regenerator</td>
</tr>
<tr>
<td>1 HP superheater</td>
<td>1 Generator</td>
</tr>
<tr>
<td>1 Reheater</td>
<td>TOTAL = 26</td>
</tr>
<tr>
<td>1 HP evaporator</td>
<td></td>
</tr>
<tr>
<td>1 IP superheater</td>
<td></td>
</tr>
<tr>
<td>1 HT economizer</td>
<td></td>
</tr>
<tr>
<td>1 IP evaporator</td>
<td></td>
</tr>
<tr>
<td>1 LT economizer</td>
<td></td>
</tr>
<tr>
<td>1 Deaerator evaporator</td>
<td></td>
</tr>
<tr>
<td>1 HP circulating pump</td>
<td></td>
</tr>
<tr>
<td>1 IP circulating pump</td>
<td></td>
</tr>
<tr>
<td>1 HP boiler feed pump</td>
<td></td>
</tr>
<tr>
<td>1 LP circulating pump</td>
<td></td>
</tr>
<tr>
<td>1 Deaerator</td>
<td></td>
</tr>
<tr>
<td>1 Deaerator storage tank</td>
<td></td>
</tr>
<tr>
<td>1 HP steam turbine</td>
<td></td>
</tr>
<tr>
<td>1 IP steam turbine</td>
<td></td>
</tr>
<tr>
<td>1 LP steam turbine</td>
<td></td>
</tr>
<tr>
<td>1 Steam turbine generator</td>
<td></td>
</tr>
<tr>
<td>1 Condensor</td>
<td></td>
</tr>
<tr>
<td>1 Feedwater heater</td>
<td></td>
</tr>
<tr>
<td>1 Air ejector</td>
<td></td>
</tr>
<tr>
<td>1 Condensate pump</td>
<td></td>
</tr>
<tr>
<td>1 Gland steam condenser</td>
<td></td>
</tr>
<tr>
<td>1 Feedwater treatment and storage</td>
<td></td>
</tr>
<tr>
<td>TOTAL = 29</td>
<td></td>
</tr>
</tbody>
</table>

The pump drive provisions); the Ericsson cycle of Fig. 10 has 14 rotating components. The combined cycle can function at reduced power, with certain types of component failure. A detailed study of failure modes of the Ericsson cycle has not been made but would be desirable. It is likely, for instance, that a combustor failure (nozzle plugging, liner crack) in an Ericsson cycle could be accommodated at reduced power and efficiency, as could regenerator fouling.
or intercooler coolant interruption. But the major concern will be for failures in the multiple rotating equipment. Even here, it may be possible to accommodate certain types of failures.

The point is that the complexity is not much different from that of the combined cycle, and the Ericsson cycle should not be rejected because it has more compressors and turbines than we are used to. The combined cycle is itself substantially more complicated than a steam plant, and a modern steam plant is much more complicated than early ones. The same trend, for similar reasons, must be anticipated for gas turbines.

From the above discussions, we conclude that the complexity of the Ericsson cycle is comparable to that of an advanced combined cycle and is very roughly twice that of advanced Brayton cycle plants. The most serious difficulties with gas turbine plants have resulted from high-temperature corrosion, distortion, and cracking. The relatively low firing temperature of an Ericsson cycle should eliminate these problems.
IV. APPLICATIONS AND FUELS

The very high thermal efficiency capability of the modified Ericsson cycle suggests that it is a prime candidate for base-load central station powerplants. Where gaseous or liquid fuels are available at low cost for such use, that is indeed the case. But in the more important instances where such fuel is not readily available, the widespread use of the modified Ericsson cycle for base-load plants will depend on the full commercialization of coal gasification systems.

Many load-peak electrical powerplants now use liquid or gaseous fuel and are expected to do so in the future. The modified Ericsson cycle is a strong candidate for these applications as well. The powerplant and particularly the regenerator for such applications need to be designed for cyclic operation without fatigue failure.

There may also be advanced nuclear applications for the modified Ericsson cycle, but these are necessarily speculative. The application to high-temperature fission reactors is one possibility in which reactor coolant (or secondary fluid) would be used to heat the cycle working fluid in heat exchangers. In this case, closed cycles with better heat-transfer fluids than air will be of most interest. The very high thermal efficiency that is possible with relatively low peak cycle temperature (although it is high for a power reactor) makes the Ericsson cycle of particular interest.

For fusion reactors in which the output is manifested as heat generated by neutron slowing outside the reactor proper, the same characteristic may make the modified Ericsson cycle of great utility, especially if the plant is only marginally self-sustaining. Similar comments apply to various other schemes such as nuclear breeding in accelerators, where the efficiency with which heat can be recovered and converted may be critical. The temperature at which heat can be made available seems to match very well the temperature capability of the Ericsson cycle.
Solar thermal powerplants also may benefit significantly from the Ericsson cycle, again most likely as a closed-cycle plant. The thermal energy source for all these renewable-resource plants will tend to be the dominant element of cost. The higher efficiency possible with the Ericsson cycle at temperatures compatible with the heat source will directly reduce the cost of electricity from such plants.
V. CLOSING REMARKS

Because the Ericsson cycle approximation is a relatively novel concept, detailed design studies of the plant components and layout need to be carried out in order to investigate component problems in detail, to suggest component demonstrations where appropriate, and to permit detailed estimates of auxiliary plant losses. These studies should include starting and part-load operations, control logic and mechanization, and operation with partial failures. Then, detailed cost estimates for capital and operating costs should be made.

Another major issue that should be addressed is air pollution control. The low-temperature, staged combustion that is characteristic of an Ericsson cycle may lend itself very well to suppression of nitrogen oxides in the exhaust, but this needs verification.

Relatively minor, but still significant, issues are the design constraints imposed by transportation and plant erection costs. In general, the rotating components of an Ericsson cycle are separable and will be smaller than those of a Brayton cycle of the same net output, so that they will not likely be limiting items. The recuperator can without significant penalty be made in modular form for large sizes. The limiting item for transportation and erection will probably be the alternator, rather than cycle components proper.

The modified Ericsson cycle proposed here can be regarded either as an approximation to the ideal Ericsson cycle, or as an elaboration of the Brayton cycle. The latter view has been more common in the past and may account for the somewhat tentative approaches that have been taken. Brown-Boveri, however, did build two gas turbine powerplants in 1948 using two intercoolers, a single reheat stage, and a regenerator. The thermal efficiency of these plants was 30 percent, despite the low peak temperature of 600°C (1112°F) and a very low pressure ratio. The scheme of Ericsson cycle approximation, however, does not seem to have been further exploited.
The present situation regarding the Ericsson cycle is in some ways analogous to that of the aircraft gas turbine before the introduction of the turbofan. Once a certain level of component efficiencies and turbine temperature was attained in gas turbines, there was no real obstacle to turbofans. No material or aerodynamic advances were required, and the idea had been proposed at a very early stage. A turbofan (the Metropolitan-Vickers F. 2/3) was, in fact, under development as early as 1945, although it was not brought fully to fruition. Nonetheless, many arguments were brought forward\(^{(12)}\) claiming that the weight and/or volume of the turbofan was too big, that such engines were not suitable for fighters, or that if one really wanted to save fuel, one should use a turboprop. But once the first turbofans were introduced, all those arguments had to be forgotten. They represented a failure of imagination and the all too common satisfaction with the present product.

The advantages in efficiency and the potential for low cost of the modified Ericsson cycle have been outlined here. It is time to exploit them.
REFERENCES


