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APPENDIX I

The numerical values used in the examples presented are listed below.

Variables	Value used
Machine variables:	
compressor polytropic efficiency	0.9
turbine polytropic efficiency	0.85
intercooler pressure drop	5%
main combustor pressure drop	4%
reheat combustor pressure drop	2%
Cooling parameters:	
mean heat-flux to work-flux scaling factor, σ	0.15
mainstream stagnation pressure-loss factor, Y	0.2
blade heat exchanger effectiveness, ϵ	0.5
factor for reduction of Stanton number, n	0.25
Other:	
ambient temperature for bottoming cycle	290K
gas constant	8314 J/mole-K
specific heat ratio, γ	1.4

DISCUSSION

I. G. Rice, P. E.[1]

The author is to be commended on the three-part paper presenting gas turbine cycle analysis based on the Second Law of Thermodynamics. This discussor also wishes to thank the author for including some of the discussor's pertinent papers as references. There are several points of discussion following which are presented for consideration by both the author and readers of the papers. All three parts are covered together in the discussion.

The Second Law can be usefully used to evaluate the availability of heat to do work and is applied extensively to measure both compression and expansion efficiency wherein the work required to compress a gaseous fluid or the work obtained in expanding a gaseous fluid is referenced to the optimum values at constant entropy. The author goes a step further and effectively relates all losses through the Second Law. There is, however, no substitute for the First Law and the conventional heat balance method.

Gas turbine component efficiencies have been improving over the years and firing temperatures have been increasing. Any computer program used in this regard should provide results that agree with actual industrial gas turbines now on the market. Projections can then be made, using this benchmark, for both higher turbine inlet temperatures and cycle pressure ratios (CPRs). A tabulation of published base-load cycle data of five current industrial gas turbines to compare with the data presented by the author is given in Table 1. It can be noted that the LM2500 has the highest base-load firing temperature of any industrial gas turbine on the market – 2214°F (1212°C) – and the highest specific work – 141.4 BTU/lb of air (328.9 kJ/kg). This particular aeroderivative gas turbine also has the lowest excess air in the exhaust gas, as can be noted, and the best state-of-the-art cycle parameters available today for combined cycles. This machine represents a new landmark for combined cycle projections.

[1]Consultant, Spring, TX; Fellow ASME

The data presented in the papers, particularly in Parts 2 and 3, do not correlate too well with the engine data in Table 1. The simple cycle gas turbine combined cycle efficiency at turbine inlet temperatures of around 2000°F (1093°C) would peak at about a 16 CPR and not at a CPR of about 8 as given in Fig. 9 of Part 2. Likewise, the reheat gas turbine combined cycles peak at much higher CPRs of 50 to 55 and not 16 as shown in Figs. 1 and 3 of Part 3. The Japanese studies and tests of their research reheat gas turbine indicate a peak combined cycle efficiency at a 55 CPR.

Turbine innovations such as already introduced by Rolls-Royce on their RB 211-535 E4 engine whereby the amount of first-stage cooling air required is reduced by placing a ceramic thermal barrier on the pressure side of the rotating blades should be considered. New innovations such as this must be recognized when projecting future cycles. Likewise, as a result of the NASA E^3 program, gas generators now being designed for both the unducted fan jets and the ducted prop jets by General Electric, Pratt & Whitney Aircraft, and Rolls-Royce will have CPRs of 38 to 42 for a base-load turbine inlet temperature of about 2250°F (1232°C) to obtain maximum simple cycle efficiency. Such gas generators will be prime candidates for the industrial reheat gas turbines of the future.

Turbine Inlet Temperatures

When large quantities of cooling air (or steam) are used to cool the first stage stationary nozzle vanes, the coolant acts thermodynamically the same as tertiary combustor liner cooling air – particularly when the cooling air (or steam) is introduced at the vane leading edge where the approach velocity is rather low – being about Mach 0.1. The nozzle exit velocity achieved is dependent upon inlet static pressure, inlet temperature, and static exit pressure wherein inlet static pressure is more dominant than temperature.

Aircraft manufacturers, and also industrial gas turbine makers, for cycle analysis purposes have gone to the term "Turbine Rotor Inlet Temperature" (TRIT) which is the average total temperature with the first-stage nozzle vane cooling air included. Tabla 1 reflects TRIT. There can be a variation of some 150°F (83°C) with this value and the average combustor outlet temperature. When considering water-cooled first-stage nozzle vanes, the gas stream total temperature is not lowered to as great a degree, and therefore there arises a discrepancy in comparing cycles. In any regard, blade metal temperature is the design criterion and not the inlet temperature definition.

Another factor to consider is the type of combustor employed. An annular combustor provides a far more uniform circumferential exit temperature profile and avoids the temperature spikes of a multiple can-type combustor. Therefore, in the opinion of this discussor, the type of combustor assumed should be stipulated for the analysis.

Intercooling

Intercooling can be accomplished without too much difficulty and one stage of intercooling, and possibly two stages, are within reason for the reheat gas turbine combined cycle. Intercooling does, however, degrade combined cycle efficiency. This degradation is shown in Fig. 3 of Part 3. If one stage is used, then the cooling should take place very early in the compression at about a pressure ratio of 2. See this discussion's reference [1]. If a second stage is used, then the second stage should likewise follow the first stage after another compression ratio of about 2. The author does not indicate at what compression ratio intercooling should take place. One stage only of intercooling should not be at the square root of the total pressure ratio for a maximum combined cycle efficiency. As a note of interest, when considering regeneration instead of the combined cycle, the intercooling pressure ratios are indeed higher.

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Table 1 Base-load gas turbine data: iso conditions*

		A General Electric LM2500PE	*B* Rolls-Royce R-B211C	*C* General Electric LM5000A	*D* Westinghouse 501-D	*E* General Electric PG7111E
1	Shaft output**					
	HP	29,500	34,000	43,525	138,699	103,682
	kW	22,007	25,364	32,469	103,469	77,347
2	Cycle efficiency					
	percent LHV	37.0	36.4	37.4	33.4	32.3
3	Cycle pressure ratio	18.7	20.0	30.0	14.0	11.7
4	Air flow					
	lb/s	147.5	196.2	265.0	801.0	605.0
	kg/s	66.9	89.0	120.2	323.3	274.4
5	Specific work output					
	BTU/lb air	141.4	122.6	116.2	122.4	121.2
	kJ/kg air	328.9	285.2	270.3	284.7	281.9
6	Turbine rotor inlet temperature					
	°F	2214	2133	2113	2025	2019
	°C	1212	1162	1156	1107	1104
7	Exhaust temperature					
	°F	955	887	797	970	1000
	°C	513	475	425	521	538
8	Exhaust excess air					
	percent	226	270	302	241	232

*Data published by *Gas Turbine International Catalog*, 1984
**Assumed generator loss of 2 percent added as required

Reheating

Reheating, when considering high turbine inlet temperatures, is far more difficult to accomplish than intercooling. The gas exiting the gas generator has a rather high temperature – 1200 to 1400°F (650 to 760°C) – and high velocity – about Mach 0.4 to 0.5 – which accounts for 10 to 15 percent of the total pressure. This high velocity must be fully diffused without separation to a low velocity of about 100 ft/s (30 m/s) combustor reference velocity to avoid an excessive pressure loss and to control the flow to the reheat combustor. Even with the best diffuser, a 1 percent pressure loss will take place. Another 2 to 3 percent is needed for the combustor liner. Another design challenge relates to the rather high temperature encountered to the reheat combustor and the cooling of the reheat combustor liner. Adding a second reheat combustor only compounds these two difficulties. The assumed total value of 2 percent reheat combustor pressure loss given in Part 3 is unrealistic. A more realistic value would be 4 percent.

Reheating cannot readily be done at any particular and desired expansion ratio. A reheat combustor must be placed between turbine stages. This fact can complicate the physical configuration. Reheating, similar to intercooling, should be done early in the expansion for maximum combined cycle efficiency, but this ideal arrangement is impossible to accomplish. There are two basic machine arrangements: the configuration used by the Japanese and the one proposed by the discussor per the references given by the author.

Combined cycle part-load efficiency has not been addressed by the author, which is a decided plus for the reheat gas turbine. The Japanese have now proven that the reheat cycle offers exceptionally good part-load efficiency, even down to 40 percent cycle output. Such part-load efficiency is not possible with the simple cycle gas turbine combined cycle.

Steam Cooling/Injection

Steam cooling/injection falls in an entirely different category from either air cooling or water cooling, and the author does not make any clear distinction. Such steam cooling/injection, in this discussor's judgment, creates not a combined cycle but rather, an integrated gas/steam cycle. The injected steam becomes part of the working fluid of the gas turbine. Steam cooling/injection degradation of combined cycle efficiency does not follow any of the cycles (*a*), (*b*), or (*c*) of Fig. 9, Part 2; and, likewise, Part 3 does not cover the concept of the integrated cycle.

The application of steam cooling/injection makes it possible to drive a high-pressure core compressor (pressure ratio 20) with a single-stage turbine and one disk without excessive expansion losses associated with otherwise transonic blading. Subsonic surface Mach numbers can be realized. Cooling requirements are then reduced by eliminating one complete stage. The additional work for the one stage is accomplished because of the following:

(*a*) All the cooling air is heated in the combustor to do work. Incremental work is obtained.

(*b*) The cooling steam injected at the leading edge of the nozzle vanes provides additional mass flow and more work is realized therefrom.

(*c*) Steam injection in the combustor can augment the cooling steam to provide the required work to drive the high-pressure compressor with the single stage.

Only one turbine stage is required to drive the low-pressure compressor which is normal practice for such gas generators as the LM5000.

In addition to the advantage of one less gas generator turbine stage requiring cooling, the gas generator expansion ratio required is substantially reduced and reheat can be introduced earlier in the overall expansion which increases the power turbine expansion ratio and thus increases output.

If compression intercooling is incorporated, the inlet temperature is reduced to the high-pressure compressor and a reduction of core compression work is realized to further enhance the single stage turbine design.

The high temperature reheat gas turbine is a special machine and there are only a few windows of design opportunities and options open. Generalized studies incorporating 3, 4, 5, etc. stages of intercooling and 2, 3, 4, etc. stages of reheat are of academic interest, but design criteria and practicality must also be considered.

It appears to this discussor that there is no thermodynamic justification for water-cooled first-stage vanes; and the only justification would be for rare occasions where high ash content fuel is burned and a water-soluble ash is formed at a reduced temperature which can be removed by water washing.

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The three-part paper does, however, point out the fact that cooling degrades combined cycle efficiency and high cycle pressure ratio gas turbines – particularly reheat gas turbines – are susceptible to excessive cooling losses and these losses should be properly recognized, evaluated and offset by design innovations.

The reheat gas turbine combined cycle efficiency gains are too great to be ignored and Part 3 of the author's paper focuses on and draws deserving attention to this advanced cycle of the future.

References

1 Rice, I. G., "Evaluation of the Compression Intercooled Reheat Gas Turbine Combined Cycle," ASME Paper No. 84-GT-128.

Author's Closure

The author is appreciative of Mr. Rice's thoughtful and insightful comments. However, the author is unable to agree with some of the discussor's statements, which seem to arise from a misinterpretation of the letter or philosophy of the work.

The author sought to provide a general, compact, and broad-ranging method of analysis that can navigate through and assess the myriad possibilities for future combined cycle developments. The numerical results cited are used as an example of the power and versatility of the model presented. The default values of the three dimensionless parameters (σ, β, Y) used are typical examples. They cannot be used to predict the performance of all existing machines. If the performance of a particular machine were to be modeled, those parameters would have to be calculated for its unique design. The stage design and heat transfer data necessary to calculate σ are proprietary to the manufacturers. Based on publically available information, the author has estimated that the value of this parameter for the AGTJ (moonlight) high-pressure turbine is approximately 0.2. For models of certain older, well-proven machines, the author's estimate is 0.35. The blade temperature β has been estimated at 3.8 for the AGTJ first stage and about 3.6 for the same older model. The cycle performance and optimum pressure ratio depend strongly on those parameters. Therefore, any calibration of the model against existing designs has to be done on a case-by-case basis. The author wishes to report that since this work was completed in early 1984, he has initiated a joint study of future cycles with the General Electric Company and the Electric Power Research Institute. Application of the author's model to calibration test cases on the EPRI GATE program, which provides detailed (but cumbersome) stage-by-stage chemical and thermodynamic calculations, has resulted in remarkably close agreement. The results of those studies are expected to be published in the near future.

Returning to some of Mr. Rice's specific comments, the author would like to elucidate several points:

The author disagrees with the discussor's comment that the optimum pressure ratio for the nonreheat combined cycle at about 1100°C TIT is approximately 16. Such a conclusion might be obtained if one were to do the calculation with a conservative steam cycle without allowing for improvement of steam cycle efficiency with increasing gas turbine exit temperature. Such a calculation was presented by the discussor (ref. [1], Part 3). When one continuously optimizes the steam cycle as turbine exit temperature increases, the optimum pressure ratio obtained is in the 8–12 range. This conclusion has been corroborated by runs on the EPRI GATE program with detailed simulation based upon existing engines and steam cycle technology. In general, the following factors tend to drive the optimization toward higher pressure ratios: (1) lower values of σ; (2) higher values of β; (3) smaller slopes of the curve of steam cycle effectiveness versus turbine exit temperature (Fig. 8, Part 1) which would arise with multipressure steam evaporators.

The discussor's comment on the optimum CPR for the reheat cycle is affected by the same factors. The optimum pressure ratios are in the range 16–25 without intercooling (Figs. 1 and 3, Part 3). The discussor incorrectly compares those to the intercooled Japanese cycle. The latter should be compared with Fig. 6 of Part 3, which shows optima in the range of 30–50 depending on cooling method. With the high value of β (3.8) and low σ (about 0.2) used in the AGTJ, the optimum pressure ratios should be higher (about 50).

On the subject of intercooling, the discussor's own work (ref. [1] of his discussion) is in agreement with the author's conclusion that for optimum efficiency the intercooler pressure ratio should be unity (no intercooler at all). If one assumes its existence, with a finite pressure drop, then the optimum pressure ratio to which the discussor refers is obtained at some low value. Intercooling provides additional specific power output at the expense of efficiency. If one were to optimize for efficiency, intercooling should not be employed at all. For maximum specific output, the intercooler pressure ratio should be at about the square root of the overall value. In general, the intercooler, if used, should be intermediate between those limits, the efficiency falling and specific output increasing as the intercooler pressure ratio increases from unity to the square root of the overall CPR. This tradeoff has to be recognized in deciding whether or not to intercool, and if so at what pressure ratio.

The discussor's comment on the reheat combustor pressure loss is well taken. The author agrees that a total pressure loss of 4 percent is more realistic than the 2 percent used in the examples. Figure 8 of Part 3 shows that the effect of such a change should be a penalty of about ⅓ of a percentage point to the reheat cycle. Notwithstanding, the author wishes to point out some silo-type primary combustors (such as Brown-Boveri's and Kraftwerke Union's) do achieve pressure losses less than 2 percent. Also, the reheat combustor has a smaller fundamental loss than the primary combustor due to the smaller heat addition per unit mass [1].

On the subject of steam cooling, the discussor mentions many advantages but none of the disadvantages. Even though steam cooling requires less coolant massflow, it can be shown that the mole flow rates of coolant are roughly equal for air and steam [2]. The thermodynamic penalties in the gas flow path are essentially a function of the mole fractions of coolant to main gas (equations 3–9 of Part 3). Steam cooling is found to provide modest gains, which derive more from the cycle compounding effect (ref. [10] of Part 3) than from the reduction of the cooling penalty.

References

1 LeFebvre, A. H., "Aerodynamics," *Gas Turbine Combustion*, McGraw-Hill, New York, 1983, Chap. 4.

2 Pourkey, F., and El-Masri, M. A., "Thermal Design Scaling for Turbine Cooling Systems," submitted to ASME 1986 Gas Turbine Conference, Düsseldorf, Germany.

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