OPTIMIZATION OF VARIOUS GAS TURBINE CYCLES

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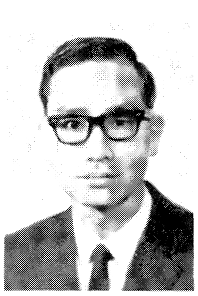

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Nomenclature listed at the end of the paper.

ABSTRACT

The gas turbine, as is used in most applications, does not utilize its maximum potential. There are many reasons that this state of affairs remains, most significant being cycle and metallurgical limitations. In this paper, the attention is paid to the cycle itself and to the thermodynamic parameters which affect it.

The present paper analyzes numerous variations of the gas turbine cycles, which include various types of open and closed cycles, gas and water injection cycles, and the Rankine-Brayton combination cycles. An attempt has been made in each case to optimize the cycle in terms of pressure ratio, turbine inlet temperature, regenerator size, and other pertinent parameters. The goal of this paper is to enable the reader to evaluate various types of cycles that are being offered to him, and to choose a cycle which would best fit his needs. The paper will also aid the user in making changes to existing turbines for higher efficiency and lower NO_x emissions.

INTRODUCTION

The energy shortage throughout the world has made the engineers face up to some of the inefficiencies of their prime movers. Gas turbines are being utilized at an increasing rate by the utility pipeline, and the process industries of this nation. Their efficiencies run as low as 12 to 17 percent at design conditions. This low efficiency is unacceptable in these days of the energy crunch. Low efficiency is also not a necessary phenomenon since prudent use of the exhaust gases in most cases would sharply increase the overall thermal efficiency.

This paper deals with many variations of the gas turbine cycle. Effect of pressure ratio, inlet turbine temperature, regenerator effectiveness, and other pertinent parameters are explored so as to evaluate the cycle completely and obtain the optimum parameters which are necessary for the high efficiencies. In many cases higher efficiency does not necessarily mean a very complicated cycle but could signify a better choice of thermodynamic parameters. On the other hand it may be necessary to pay higher initial costs for a more complex and higher efficiency machine if fuel becomes unavailable.

Throughout this paper the authors have attempted to not only present theoretical cycles but to try and incorporate them with existing technology in the area of blade design (aerodynamic and metallurgical), heat exchangers, combustion chambers, and overall corrosion problems.

From the many cycles that are discussed in this paper, it is obvious that higher overall thermal efficiencies can be obtained if both the user and manufacturer cooperate to achieve these goals.

GENERAL ANALYSIS

Application of the first law to the air-standard Brayton cycle shown in Figure 1 gives us the following relationships:

Work of compressor

$$W_c = \dot{m}_a (h_{t_2} - h_{t_1}) \tag{1}$$

Work of turbine

$$W_t = (\dot{m}_a + \dot{m}_f)(h_{t_3} - h_{t_4}) \tag{2}$$

Total work output

$$W_{cycle} = W_t - W_c \tag{3}$$

The heat added to the system

$$_2Q_3 = \dot{m}_f \times \text{LHV (fuel)} = (\dot{m}_a + \dot{m}_f)(h_{t_3}) - \dot{m}_a h_{t_2} \tag{4}$$

Thus the overall cycle efficiency

$$\eta = \frac{W_{cycle}}{_2Q_3} \tag{5}$$

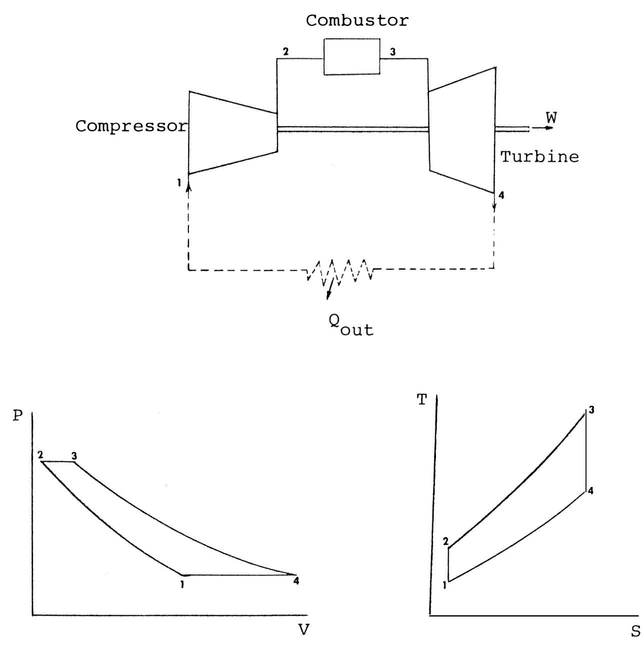

Figure 1. The air-standard gas turbine cycle.

A simplified approach to calculating the overall cycle efficiency can be obtained from the above relationships if one were to make certain assumptions such as (1) $\dot{m}_f << \dot{m}_a$, (2) specific heat (c_p) and the specific heat ratio (γ) remain constant throughout the cycle, (3) the pressure ratio in both the compressor and the turbine is the same, (4) all components operate at 100 percent efficiency. With these assumptions, the following relationships for efficiency can be obtained.

$$\eta_{cycle} = 1 - \frac{1}{r_p^{(\gamma - 1)/\gamma}} \tag{6}$$

$$\eta_{cycle} = 1 - \frac{T_1}{T_2} \tag{7}$$

$$\eta_{cycle} = 1 - \frac{T_4}{T_3} \tag{8}$$

Thus, a cursory inspection of equations 6, 7, and 8 indicate that overall efficiency of a cycle can be improved by increasing the pressure ratio, decreasing inlet temperature and increasing the turbine inlet temperature. However, these relationships do not indicate the effect on the work output of the unit nor the inaccuracies which are inherent in them at high pressure ratios, high turbine inlet temperatures, and the inefficiencies in the components.

To obtain more accurate relationship between the overall thermal efficiency and the inlet turbine temperatures, overall pressure ratios, and output work, the following relationships were obtained. Thus, for maximum overall thermal cycle efficiency, the following relationship gives us the optimum pressure ratio at a given inlet temperature, turbine inlet temperature, and component efficiencies.

$$(r_p)_{opt} = \left[\frac{T_3\eta_t}{T_1 + \eta_t T_3 - T_3} - \sqrt{\left(\frac{T_3\eta_t}{T_1 + \eta_t T_3 - T_3}\right)^2 - \left(\frac{\eta_c\eta_t T_3^2 - \eta_t\eta_c T_1 T_3 + \eta_t T_1 T_3}{T_1^2 + \eta_t T_1 T_3 - T_1 T_3}\right)} \right]^{\frac{\gamma}{\gamma - 1}} \tag{9}$$

This equation reduces to

$$(r_p)_{opt} = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{\gamma - 1}} \tag{10}$$

if one assumes $\eta_c = \eta_t = 1.0$. However, this simplified equation is not very accurate. The optimum pressure ratio for maximum output work is given by the following relationship:

$$(r_p)_{opt} = \left[\left(\frac{T_1}{T_3}\right) \left(\frac{1}{\eta_c\eta_t}\right) \right]^{\frac{\gamma}{2 - 2\gamma}} \tag{11}$$

Equations 9 and 11 give a relatively accurate analysis, but the two values do not necessarily equal each other. The discrepancy between these equations and the more complete analysis is due to the fact that in the above equations the following assumptions were made: (1) $\dot{m}_f << \dot{m}_a$, (2) that c_p and γ are constant throughout the system, and (3) the pressure ratio is constant throughout the system. The advantage of equations 9 and 11 is that a fast, quick analysis can be obtained. To obtain the overall efficiency in a regenerative cycle is more complex, and to obtain the optimum pressure ratio formulae is long and tedious. The overall efficiency for this cycle may be written as

$$\eta_{cycle} = \frac{\eta_t (h_3 - h_4) - (h_2 - h_1)/\eta_c}{h_3 - (h_4 - h_2')\,\eta_{reg} + h_2'} \qquad (12)$$

where h_2' is the actual enthalpy of the air at the compressor exit.

The values of the optimum pressure ratio for a regenerative cycle are much lower for the same inlet temperature, turbine inlet temperature, and compressor and turbine efficiencies than the corresponding value for a simple cycle. Thus, careful inspection of a turbine system is required before a regenerator is added. The interesting thing to note is that the optimum pressure ratio for maximum work is the same in the two cycles. Once again, it is important to note here that the values from these equations are approximate, but will be close to the values presented at the latter half of this paper, which are rigorous solutions.

The other cycles presented in the paper are various variations of the thermodynamic principles presented in the two cycles above. Thus, the detail analysis of these systems have been left out of this paper.

REPORT ON VARIOUS CYCLES

In all the cycles compared in this section, the inlet temperature, inlet pressure (open cycles only), and the various efficiencies of the components are kept constant. In this manner, an accurate comparison between the cycles can be obtained.

The efficiency of the compressors has been assumed to be 85 percent and the turbine efficiency to be about 87 percent. These efficiencies are high but properly designed compressors, and turbines have obtained these efficiencies. Combustor efficiency is about 97 percent, with a pressure loss of about 5% from exit of compressor to turbine. These efficiencies have been kept constant; thus, even though particular designs may not meet these efficiency values, the trends obtained here between various cycles are still very valid.

THE SIMPLE CYCLE

This is the most common type of cycle being used in gas turbines in the field today. Analysis of this cycle indicates that an increase in inlet temperature to the turbine causes an increase in the cycle efficiency, and the optimum pressure ratio varies with the turbine inlet temperature from an optimum of about 7.5:1 at a temperature of 1660°R to a pressure ratio of about 18.5:1 at a temperature of 2660°R. The pressure ratio for maximum work, however, varies from about 5:1 to about 12.5:1 for the same respective temperatures. Thus, from figure (2), it is obvious that, for maximum performance, a pressure ratio of 9.5:1 for a temperature of 2260°R would be about the most optimum. If we are to assume that the type of compressor to be used is an axial flow, we are talking of a sixteen-stage compressor with a pressure ratio of 1.151:1 per stage. This is a relatively conservative design. If the pressure ratio were increased to 1.252:1 per stage, the number of stages would be ten. The latter pressure ratio has been achieved with high efficiencies. This reduction in number of stages could mean a great reduction in the overall cost. Turbine temperature increases give a great rise in efficiency, and power temperatures in the 2000°F range at the turbine inlet should be the state of the art in the next few years.

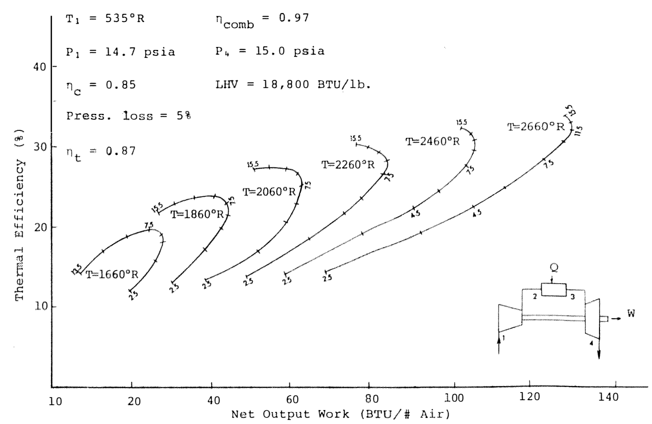

Figure 2. Performance map showing the effect of pressure ratio and turbine inlet temperature on a simple cycle.

The above discussion was a simple cycle on a single shaft; the split shaft or free power turbine is another important cycle to be considered. The advantage of the free turbine or split shaft gas turbine is mainly in the area of high torque. A free power turbine gives a very high torque at low rpm. This is of great advantage for automotive use, but for the case of constant full power operation, it is of little or no value. From figure 3, it is obvious that the split shaft turbine has slightly higher efficiences, but the higher efficiency and work is not sufficient to offset the extra complexity and cost which would result in such a design. Its use should be limited to the variable mechanical drive applications.

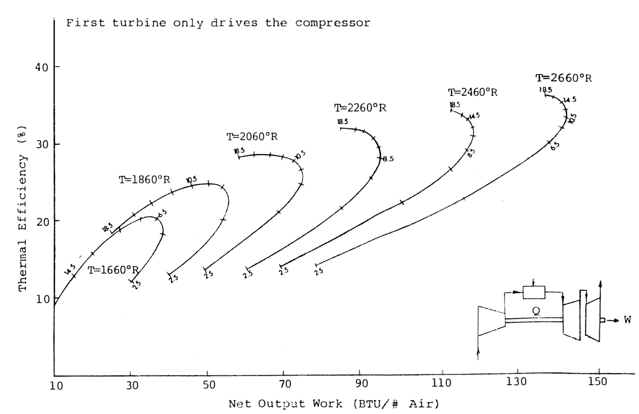

Figure 3. Performance map showing the effect of pressure ratio and turbine inlet temperature on a simple cycle split shaft system.

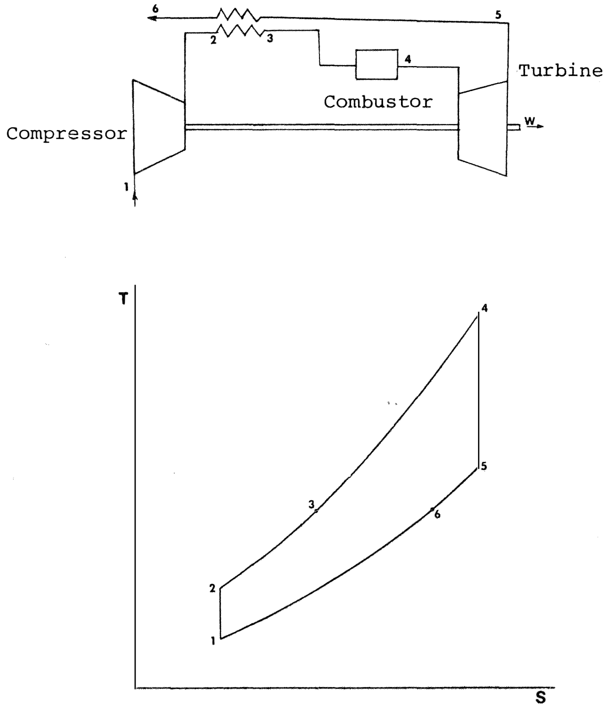

Figure 4. The regenerative gas turbine cycle.

THE REGENERATIVE CYCLE

The regenerative cycle is coming to great prominence in these days of tight fuel reserves and high fuel costs. It is a simple concept; the utilization of high temperature exhaust gases in heating of the air as it leaves the compressor. In the study here the regenerator was assumed to have an effectiveness of 80 percent. Figure 4 is the T-S diagram of such a cycle. Regenerator effectiveness is given by the following relationship:

$$\eta_{reg} = \frac{T_3 - T_2}{T_5 - T_2} \tag{13}$$

As can be seen from figure 5, the efficiency of a regenerative cycle is about 40 percent higher than its counterpart in the simple cycle case. The work output per pound of air flow is about the same or slightly less than that experienced in the case of the simple cycle. Another very important point to note is that the point of maximum efficiency in the case of the regenerative cycle occurs at a lower pressure ratio than that of the simple cycle. Thus, when companies are designing gas turbines, the choice of pressure ratio should be such that maximum benefit from both cycles can be obtained. Since most offer a regenerator option. This is not to say that a regenerator at off optimum would not be effective, but a proper analysis should be made before such a large expense is incurred.

The case of the split shaft regenerative turbine is very similar to the split shaft simple cycle. The advantages of this turbine are similar to the ones mentioned before; namely, high torque at low rpm. The cycle efficiencies are about the same. Figure 6 indicates the performance that may be expected from such a cycle. Unless the operation is over a large range of values, the advantage of the split shaft regenerative turbine over the single shaft regenerative turbine is negligible. The split shaft is ideal for mechanical drive applications where it can utilize its high torque at low rpm and its wide range of power settings.

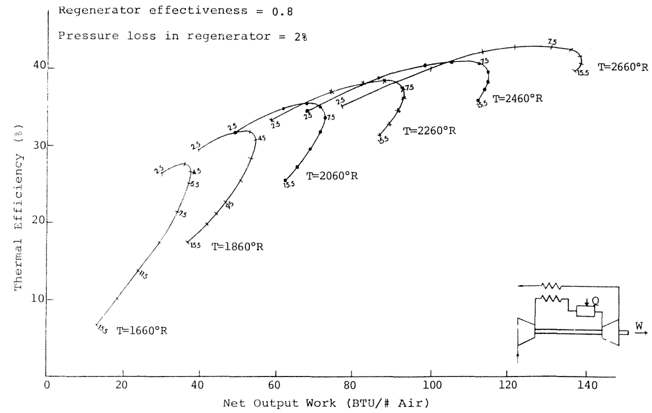

Figure 5. Performance map showing the effect of pressure ratio and turbine inlet temperature on a regenerative cycle.

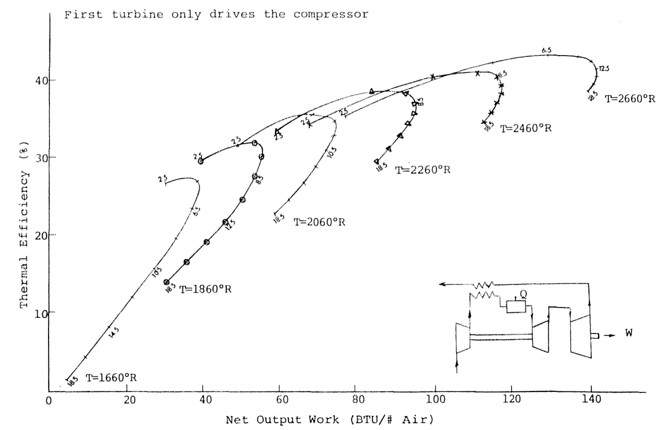

Figure 6. Performance map showing the effect of pressure ratio and turbine inlet temperature on a regenerative cycle split shaft system.

THE REHEAT CYCLES

The regenerative cycles improved the efficiency of the cycle, but did not provide any added work per pound of air flow. To achieve this latter goal, the concept of the reheat cycle must be utilized. The reheat cycle produces more work per pound of air than the other cycles. The reheat cycle, as shown in figure 7, consists of a two-stage turbine with a combustion chamber before each stage. The assumptions made in this study are that the high pressure turbine's only job is to drive the compressor and, second, that the gas leaving this turbine is then reheated to the same temperature, as in combustor one before entering the low pressure or power turbine. This reheat cycle has an efficiency which is slightly less than that encountered in a simple cycle, but produces about a 35 percent increase in the shaft output power, as shown in figure 8.

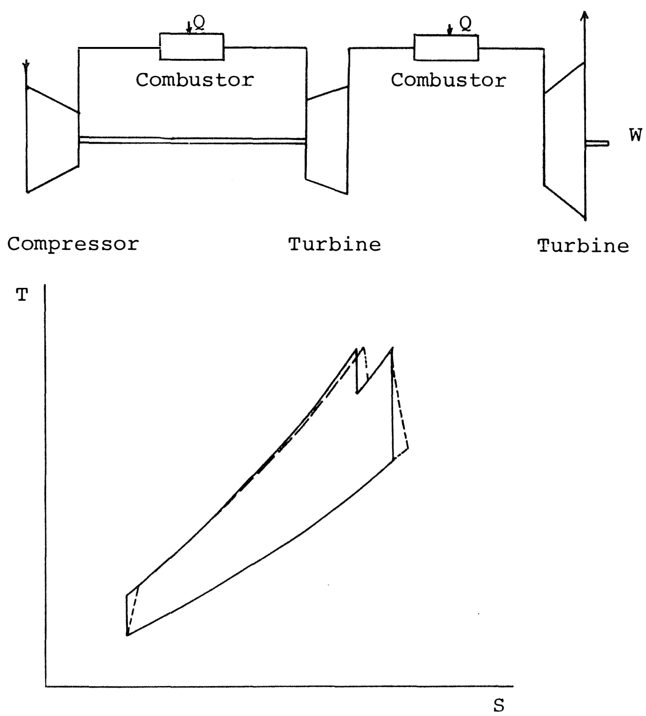

Figure 7. The split shaft reheating gas turbine cycle.

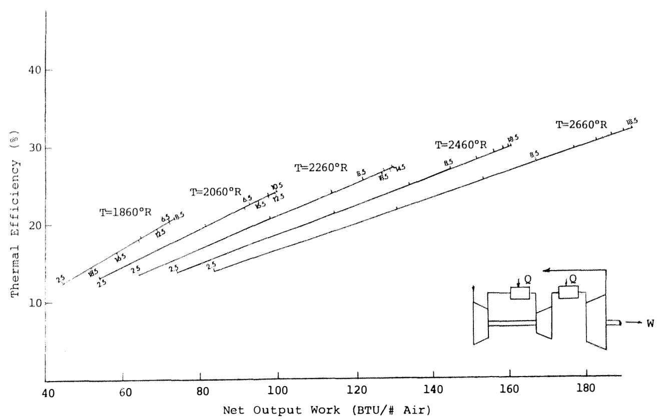

Figure 8. Performance map showing the effect of pressure ratio and turbine inlet temperature on a split shaft reheating cycle.

To obtain both an increase in the power output and the thermal efficiency, the natural conclusion would lead us to a reheat, regenerative cycle. In this cycle, the hot gases leaving the power turbine are used to heat the cooler air leaving the compressor. This combination gives us an increase in efficiency of about 40 percent and an increase in power output of about 35 percent, as indicated in figure 9. (Maximum efficiency occurs at lower pressure ratios, as compared with the simple or reheat cycles. It matches closely the pressure ratio of the regenerative cycle.) It would then seem obvious that the answer to many of our problems would be this regenerative reheat cycle. Its disadvantages, however, should also be considered. It is a more complex machine, having two shafts and two combustion chambers and two hot turbines. This means that the hot section is prone now to two failures. This could decrease the reliability of the engine and increase down time. There would be a necessity of having two rows of air

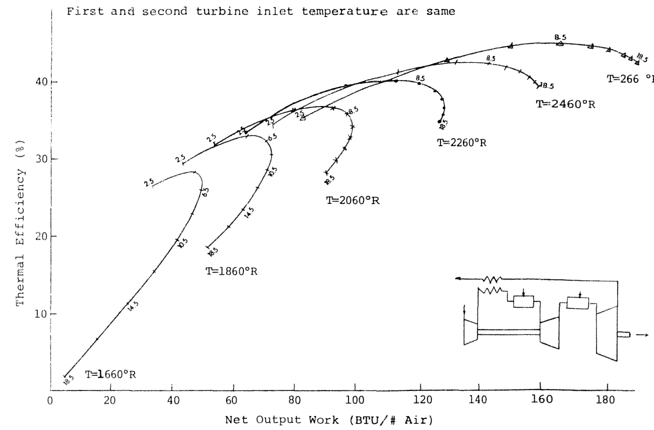

Figure 9. Performance map showing the effect of pressure ratio and turbine inlet temperature on a split shaft regenerative and reheating cycle.

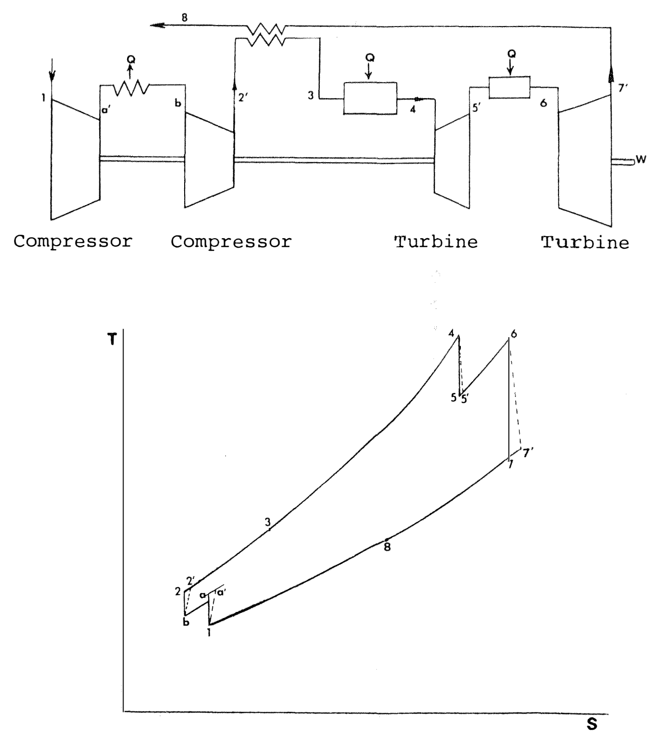

Figure 10. A gas turbine cycle incorporating intercooling, regenerative and reheating modes.

cooled blades. This factor alone increases the complexity of the overall engine.

The Carnot cycle is the optimum cycle, and all cycles tend towards this. This is achieved by approaching the isothermal compression and expansion of the Carnot cycle. This is achieved somewhat by intercooling in compression and reheating in the expansion process. Figure 10 shows the cycle which approaches in a practical fashion this optimum cycle. This cycle achieves the maximum efficiency and work output of any of the cycles described to this point. It is interesting to note that, with the insertion of an intercooler in the compressor, the pressure ratio for maximum efficiency moves to a much higher ratio, as indicated in figure 11. This cycle was made into an

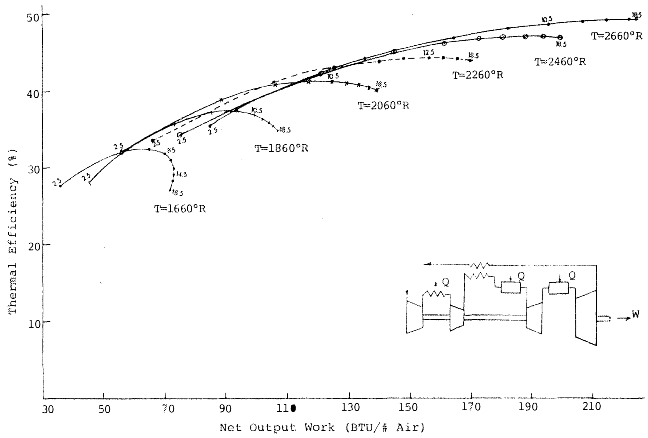

Figure 11. Performance map showing the effect of pressure ratio and turbine inlet temperature on an intercooling, regenerative, reheating cycle.

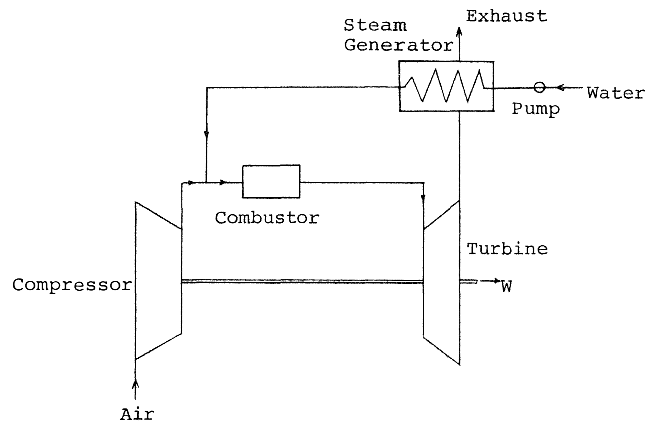

Figure 12. The steam injection cycle.

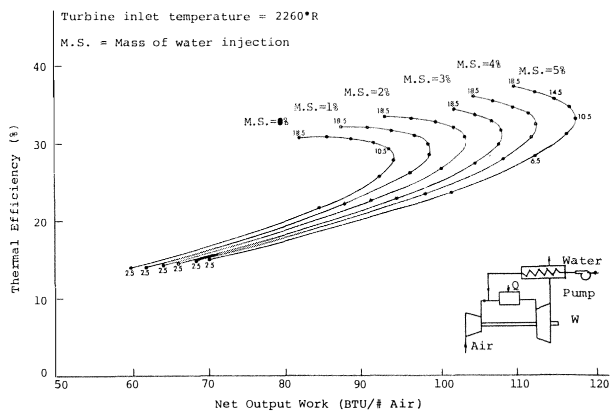

Figure 13. Performance map showing the effect of pressure ratio and steam flow rate on a steam injection cycle.

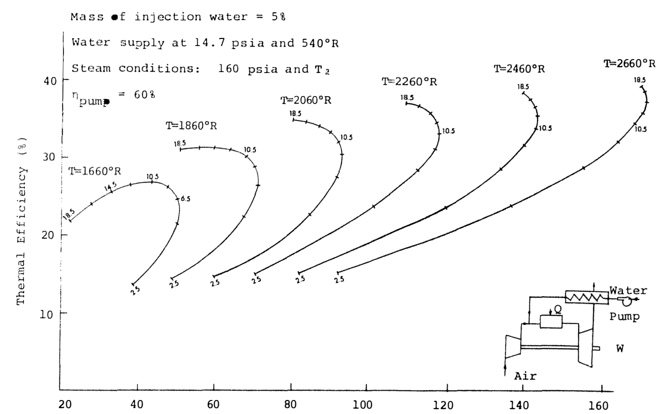

Figure 14. Performance map showing the effect of pressure ratio and turbine inlet temperature on a fixed steam rate in a steam injection cycle.

engine by Ford Motor Company. The engine was very efficient; but its costs are high, due to the extreme complexity of the cycle. With the advent of high fuel costs, this cycle may be the cycle of the future.

To sum up the open air cycles examined, it is obvious that high temperatures at the turbine inlet increases power output and cycle efficiency. With regenerative cycles, the point of maximum efficiency occurs at a lower pressure than that observed in the other cycles.

THE STEAM INJECTION CYCLE

Steam injection has been used in reciprocating, as well as gas turbines, for a number of years. With the emphasis on pollution control and higher efficiency, this cycle may be an answer. Corrosion problems are the major concern in such a system. The concept is simple and straightforward; water is injected into the compressor discharge air, and this increases the mass flow rate through the turbine, as shown in the schematic in figure 12. The steam being injected downstream from the compressor does not increase the work required to drive the compressor. Figures 13 and 14 show the effect of various rates of steam injection and the turbine inlet temperatures on the system. With about 5 percent injection at 2260°R and a pressure ratio of 8.5, about a 20 percent increase in work is noted with an increase of about 12 percent in cycle efficiency over that experienced in the case of the simple cycle. The assumption here was that steam was injected at a pressure of about 50 psia above the air from the compressor discharge, and that all the steam was created by the waste heat from the turbine. The calculations indicated that there was more than enough waste heat to achieve these goals.

This cycle's great advantage is in the low level of the oxides of nitrogen. This is accomplished by the steam being injected in the compressor discharge diffuser wall, well upstream from the combustor, so that a uniform mixture of steam and air throughout the region is formed. This reduces the oxygen content of the fuel-air mixture, thus increasing its heat capacity, which in turn reduces the temperature of the combustion zone, thus reducing the NO_x formed. Field tests have found that about 5 percent by weight of steam will reduce the amount of NO_x emissions to acceptable levels. The major problems encountered are corrosion. These problems are being investigated, and progress is being made in this area. The attractiveness of this system is that it does not need major changes to add this feature on to an existing system. The location of the water injectors is crucial for the proper operation for this system.

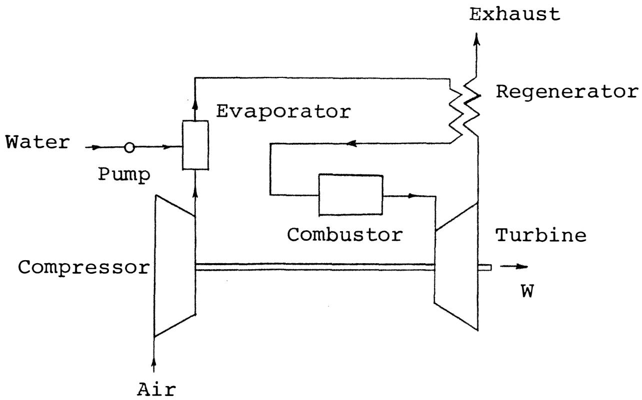

Figure 15. The evaporative regenerative cycle.

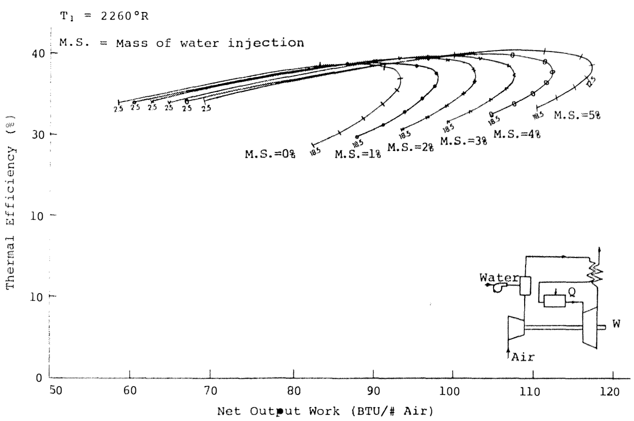

Figure 16. Performance map showing the effect of pressure ratio and steam flow rate on an evaporative regenerative cycle.

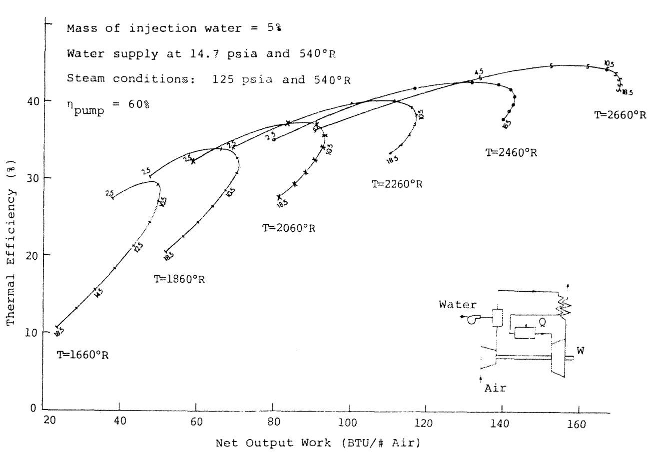

Figure 17. Performance map showing the effect of pressure ratio and turbine inlet temperature on a fixed steam rate evaporative regenerative cycle.

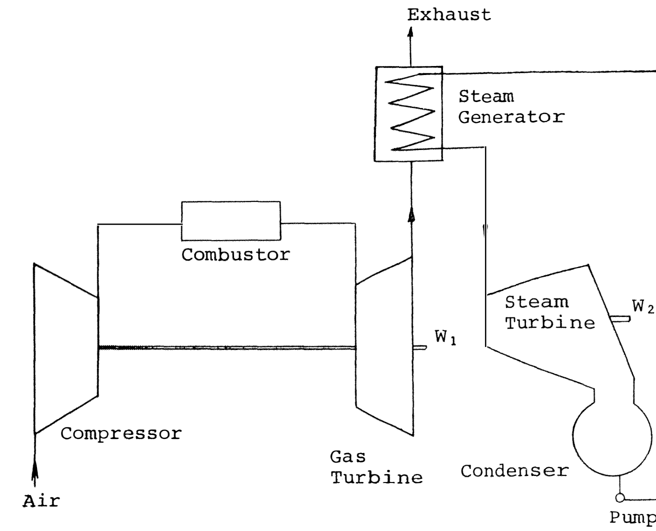

Figure 18. The Rankine-Brayton cycle.

EVAPORATIVE REGENERATIVE CYCLE

This cycle, as shown in figure 15, is basically a regenerative cycle with water injection. Theoretically, it has the advantages of both systems—reduction of NO_x emissions and higher efficiency. The work output of this system is about the same as that achieved in steam injection cycle, but the thermal efficiency of the system is much higher. Similar to the regenerative cycle, it has higher efficiencies at lower pressure ratios. Figures 16 and 17 show the performance of the system at various rates of steam injection and turbine inlet temperatures. Similar to the steam injection cycle, the steam is injected at 50 psia higher than the air leaving the compressor.

Corrosion in the regenerator is one major problem of this system. Regenerators, when not completely clean, tend to develop hot spots, which could lead to fires. With proper regenerator designs, this problem can be overcome. The NO_x emission level would be low and should meet EPA requirements.

THE RANKINE-BRAYTON CYCLE

The combination of the gas turbine with the steam turbine is a very attractive proposal, especially for the electric utilities and process industry where steam is being used. In this cycle, as shown in figure 18, the hot gases from the turbine exhaust are used in a supplementary fired boiler to produce superheated steam at high temperatures and pressures for a steam turbine.

The cycles' calculations in this paper are based on the assumption that the efficiency and net output work is based only on the parts contributed by the gas turbine itself. This system, as can be seen from figure 19, indicates that the network is about the same as one would expect in a steam injection cycle, but the efficiencies are much higher. The disadvantages of this system are its high initial cost, and it also does not reduce the NO_x content. This system has been used here at Texas A&M University for the past five years and has been very successful.*

*Detailed description of Texas A&M University system is given by Boyce in September-October 1973 issue of Gas Turbine International.

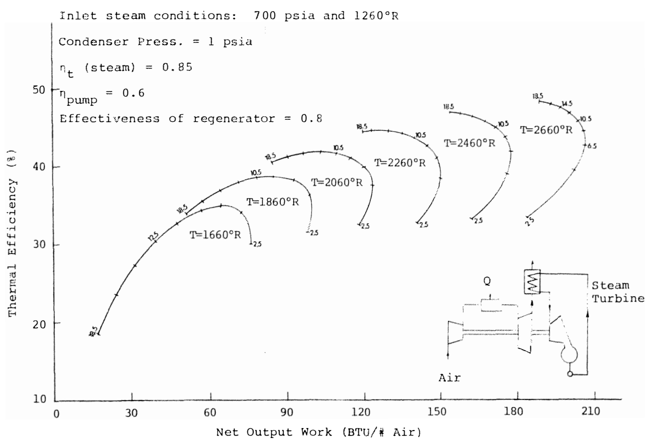

Figure 19. Performance map showing the effect of pressure ratio and turbine inlet temperature on a Rankine-Brayton cycle.

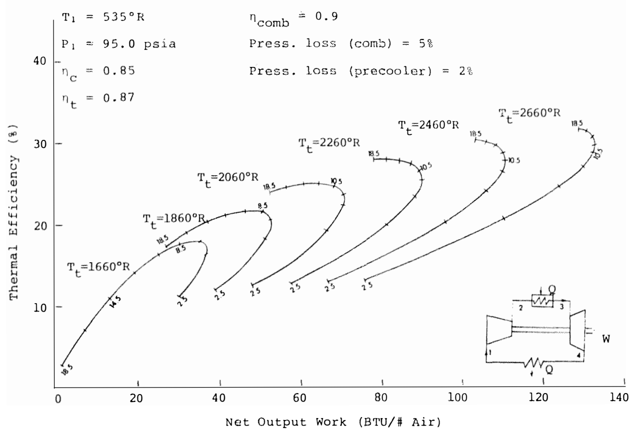

Figure 20. Performance map showing the effect of pressure ratio and turbine inlet temperature on a closed loop simple cycle.

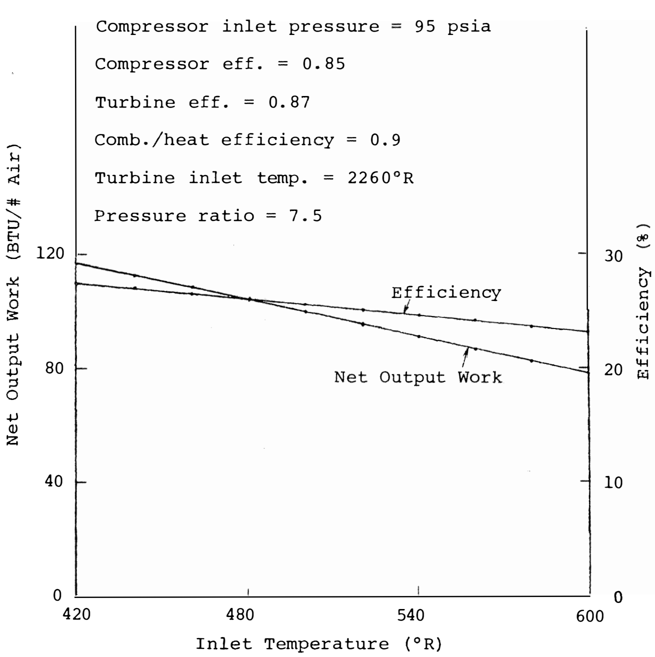

Figure 21. Effect of compressor inlet temperature on a closed loop simple cycle.

CLOSED CYCLES

The previous cycles investigated were all open cycles. No cycle paper would be complete without a look at the closed cycle concept. For this paper, the closed cycle is limited to the air cycles. This was done to place some limit to this investigation; and moreover, since this is aimed at the process and pipeline industry, a closed cycle with some other medium may not be acceptable from a safety and ruggedness viewpoint. The simple closed cycle functions in a similar manner as the open cycle. The effect of base cycle pressure was not very significant; however, the compressor inlet temperature is significant, as it would be in the case of the open cycle. Figure 20 shows the performance of this cycle and figure 21—the effect inlet temperature has in the performance of a closed cycle.

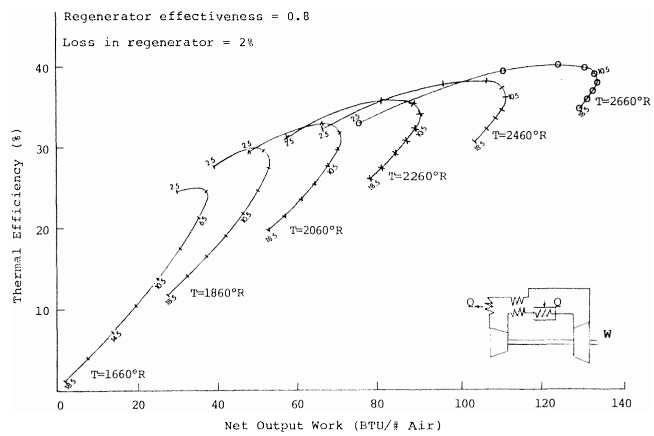

Figure 22. Performance map showing the effect of pressure ratio and turbine inlet temperature on a closed loop regenerative cycle.

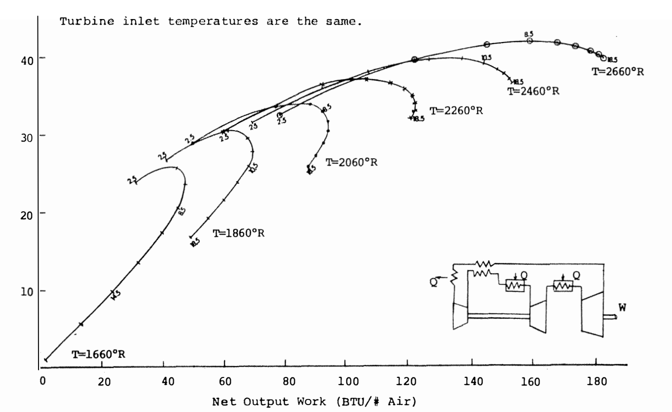

Figure 23. Performance map showing the effect of pressure ratio and turbine inlet temperature on a closed loop reheat, regenerative cycle.

Figure 22 shows the performance of a regenerative closed loop cycle. Once again, the performance is similar to that observed in the open regenerative cycle. Figure 23 shows the performance of a reheat regenerative cycle, and figure 24—the reheat regenerative cycle with intercooler.

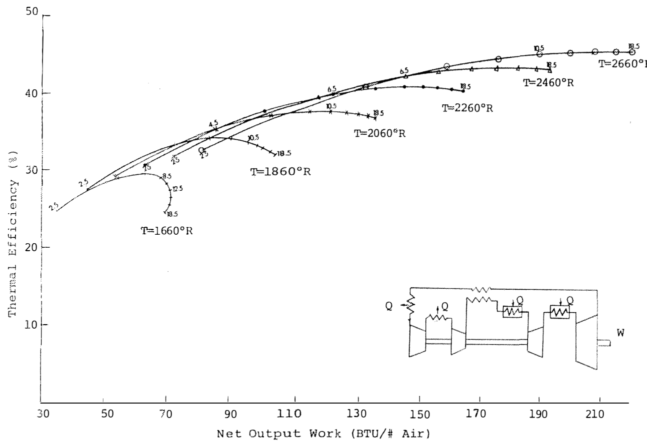

Figure 24. Performance map showing the effect of pressure ratio and turbine inlet temperature on a closed loop reheat, regenerative cycle with intercooling.

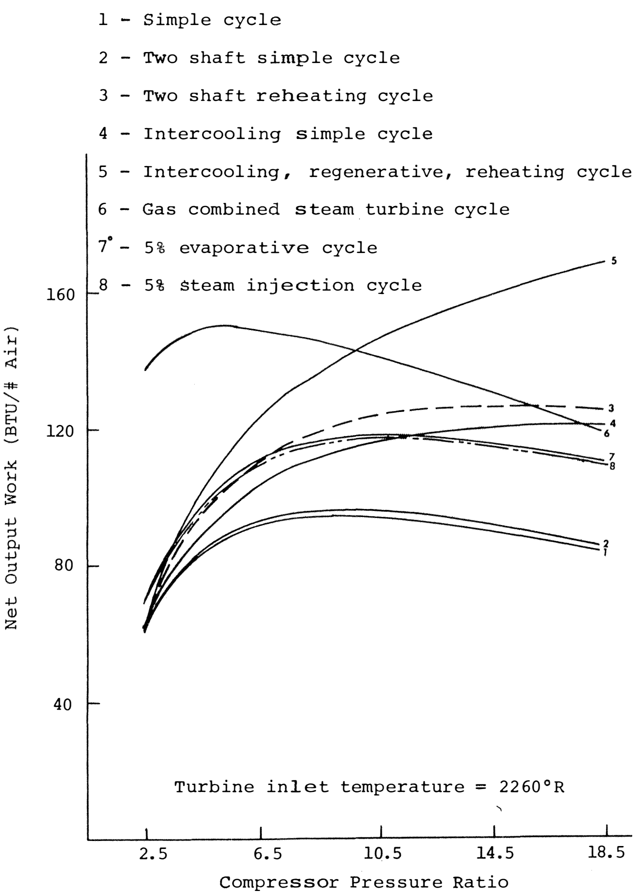

Figure 25. Comparison of net output work of various cycles.

To sum up, the closed cycles behave very similarly to their counterparts in the open cycle. The efficiency overall is slightly lower. Their main advantage would be in the low level of pollutants, since the closed cycles would

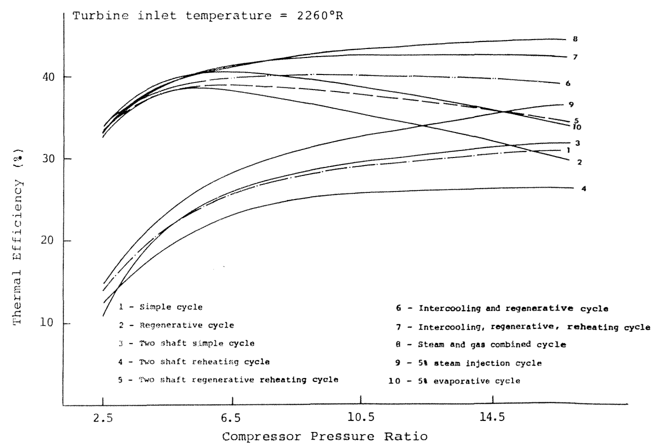

Figure 26. Comparison of thermal efficiency of various cycles.

be external heating, and this external heating could be achieved in a stoichiometric manner. The gas turbine's major pollutant is the high level of nitric oxides. Closed loop air systems are being presently considered as a possible automotive engine.

Figures 25 and 26 give a good comparison of the effect of the various cycles on the output work and thermal efficiency. The curves are drawn for a turbine inlet temperature of 2260°R, which is a temperature presently being used by manufacturers. The most efficient cycle is the combination Rankine-Brayton cycle. The output work of the regenerative cycle is very similar to the output work of simple cycle, and the output work of the regenerative reheating cycle is very similar to the reheat cycle. Thus, these two works have been left out of figure 25. The most work per pound of air could be expected from the intercooling, regenerative reheat cycle. It is, therefore, obvious that each user must make up his mind as to which system he would prefer, depending on his own plant parameters.

CONCLUSION

This paper has gone through a very large number of cycles with various options, and have shown what conditions are needed to obtain maximum performance from each cycle. It must be remembered, however, that local conditions may alter these results somewhat. These results will form a good guide in helping the user to specify equipment to meet his needs. It should also help the user to decide whether installing extra equipment will really save him energy. Every effort has been made to keep the results in this paper as accurate as possible. The efficiencies assumed are ones that can be obtained. In many cases, they are higher than what the user may experience on his own machine. Thus, the results here may be slightly more optimistic than one would expect if he were to modify his present equipment.

NOMENCLATURE

C_p = Specific heat at constant pressure
h = Enthalpy
LVH = Lower heating value of fuel
$\dot{m}_a$ = Mass flow rate of air
$\dot{m}_f$ = Mass flow rate of fuel
Q = Heat input or output
γ = Specific heat ratio
η_c = Efficiency of compressor
η_{comb} = Efficiency of burner
η_{reg} = Efficiency of regenerator
η_t = Efficiency of turbine
r_p = Pressure ratio
T = Temperature
W_c = Work of compressor
W_{cyc} = Net cycle work
W_t = Work of turbine