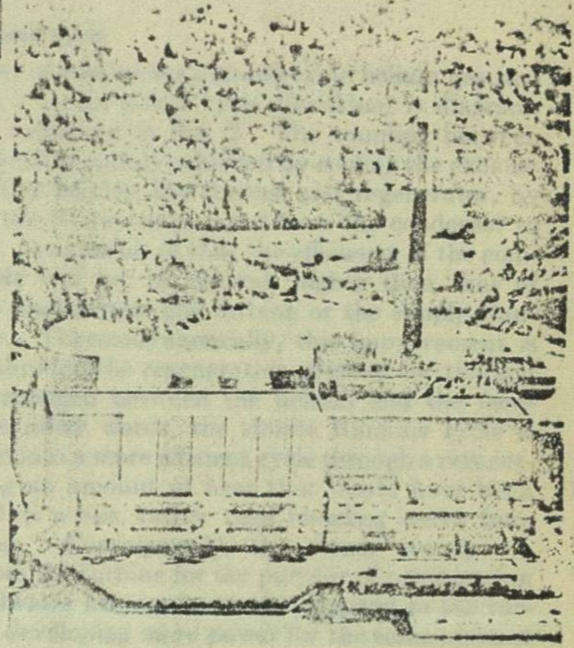
combined helium and steam cycle for NUCLEAR POWER PLANTS



Here's how a closed-cycle helium gas turbine can be combined with a Rankine steam cycle to achieve an appreciable improvement in thermal efficiency. Thermal energy in the hot gases from the regenerator of the helium cycle heats the feedwater in the Rankine cycle. Although the study includes different arrangements of the gas cycle using combinations of intercooling and reheating, the most favorable results are obtained with a simple gas cycle.

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THE IDEA of using closed-cycle gas turbines in connection with gas-cooled nuclear reactors was introduced as early as 1946 [1].[3] Rapid development in the gas turbine during the late forties and early fifties, using both the open- and closed-cycle concepts, brought it up to a competitive level with the steam turbine. During the last ten years or more, closed-cycle gas turbines using air as a working medium have been operating successfully. With a parallel and equally successful effort in the development of high-temperature gas-cooled nuclear reactors, direct coupling of helium-cooled reactors to closed-cycle gas turbines proved to be of great advantage. Gas-turbine cycles with one or two intercooling stages together with one reheating stage have been studied in detail for plant sizes up to 1000 MWe. For very large sizes of power plants (2000 MWe and more), the idea of combining the helium-gas-turbine cycle with a steam power cycle has now become more realistic.

Such a combination is thermodynamically advantageous and yields an overall thermal efficiency higher than that of either the steam or gas cycle operating separately. The improvement in the combined thermal efficiency is mainly achieved through a total or partial utilization of the waste heat from the gas cycle into a steam cycle.

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This article is based on a paper contributed by the ASME Nuclear Engineering Division.

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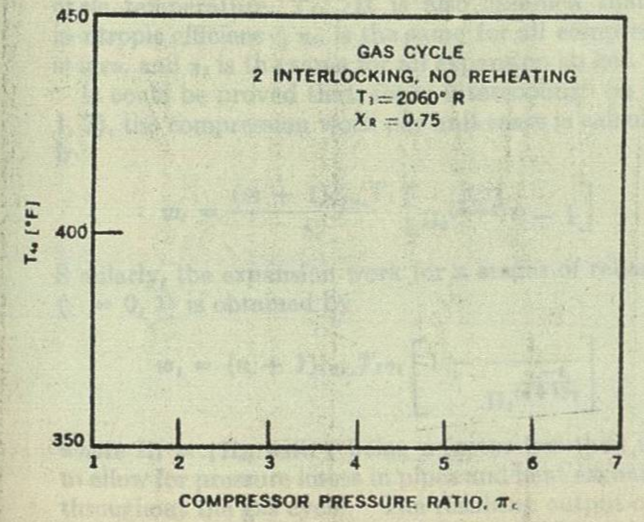
There are different possibilities for such a combination:

- Non-regenerative simple gas cycle where the waste heat in the exhaust gases is used to generate the steam necessary for the Rankine cycle. In this case, only the high-temperature part of the exhaust-gas energy could be used, while the low-temperature part is considered as a loss.
- The same cycle combination as suggested above but where the low-temperature heat is led back to the gas cycle to be partially utilized in regeneration of compressed gases [2]. An amount of low-temperature thermal energy still will be lost because the gas temperature after the compressor is relatively high.
- A combined gas-steam cycle with or without intercooling and reheating in the gas cycle, where the high-temperature exhaust energy is used for regeneration in the gas cycle. The remaining energy in the gases could be used for heating the feedwater in the steam cycle.

This last possibility provides a nearly complete utilization of the waste heat from the gas cycle and, therefore, is an interesting case for study. The purpose of this article is to analyze this possibility and explore different combination potentials.

The curve in Fig. 1 shows an example of the exit temperature from a closed-cycle helium gas turbine with two intercoolings and one reheating and using a regenerator of 0.75 effectiveness [3]. The temperature of the helium gases leaving the turbine varies between about 440 F at a compressor pressure ratio of 1.5 to nearly 370 F at a pressure ratio of 6; even higher temperatures are expected if no intercooling is used. These gases have to be cooled down to 60 to 80 F before entering the compressor again. The amount of thermal energy rejected in this case represents a waste that could be utilized in combined cycles. This thermal energy, being at a relatively low temperature, cannot be used for steam generation with the quality required to operate a power plant. However, it could be used for feedwater heating. Another advantage in the case of using helium is its relatively high specific heat and thermal conductivity, which makes it a very favorable fluid for heat-exchanger design.

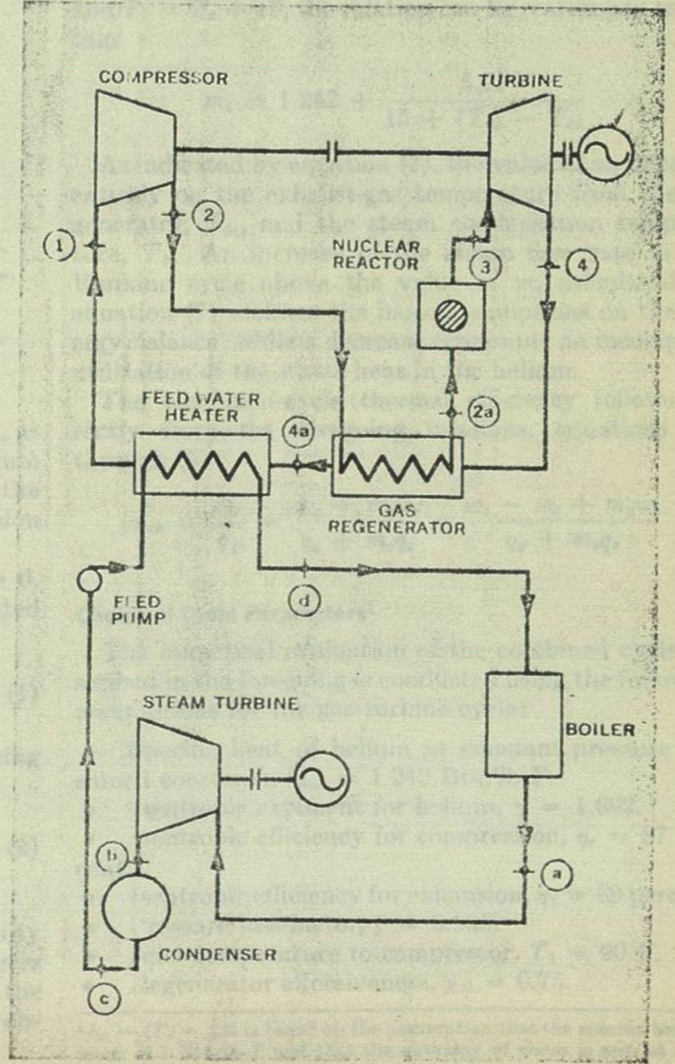
Fig. 1 Helium gas temperature at exit from the regenerator.



The Combined Cycle

The idea of combining a closed-cycle helium gas turbine with a steam power cycle (Rankine) is diagrammatically explained in Fig. 2. The coupling between the two cycles is mainly achieved by cooling the exhaust helium, after leaving the turbine and regenerator, by means of the feedwater pumped from the condenser to the boiler. As a result of this, the efficiency of the combined cycle will be, in general, higher than that of either the closed-cycle gas turbine or the simple Rankine cycle. Thermodynamically, this improvement is achieved through the regenerative effect of the thermal energy exchanged between the helium and the feedwater. In other words, the simple Rankine cycle is transferred into a more efficient cycle through a regeneration using an amount of heat that would have been considered as a loss, rather than bleeding steam from the turbine. Consequently, the steam usually extracted from the turbine for the purpose of regeneration in the feedwater heaters is saved to expand in the turbine, thus developing more power for the same amount of heat added in the boiler.

Fig. 2 Combined-cycle arrangement.



In this analysis, a simple non-reheating Rankine cycle is combined with different arrangements of the gas cycle. The general layout of the cycle is shown in Fig. 2, while Fig. 3 represents a combined temperature-entropy diagram. Although the representation in both Figs. 2 and 3 is made for a simple gas cycle, it should be kept in mind that:

- The compression in the gas cycle could be performed with none, one, or two intercooling processes.
- The expansion in the gas cycle could be either straight or with one reheating.
- The feedwater heater in the steam cycle is only diagrammatically symbolized in Fig. 2; in fact, there could be more than a single heater in series.

The thermodynamic analysis of the gas turbine is further simplified by assuming complete and ideal intercooling to the compressor inlet temperature, T_1, as well as complete and ideal reheating to the maximum cycle temperature, T_3. It is also assumed that the isentropic efficiency, η_c, is the same for all compression stages, and η_t is the same for all expansion stages.

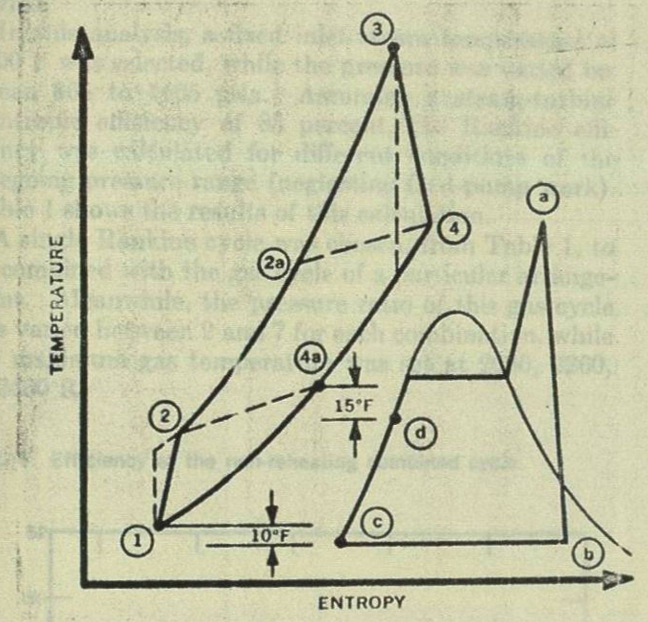


Fig. 3 Temperature-entropy diagram of the combined cycles.

It could be proved that, for m intercoolings ($m = 0, 1, 2$), the compression work per unit mass is calculated by

$$w_c = \frac{(m+1)c_{p_{He}}T_1}{\eta_c}\left[\Pi_c^{\frac{\gamma-1}{(m+1)\gamma}} - 1\right] \qquad (1)$$

Similarly, the expansion work for n stages of reheating ($n = 0, 1$) is obtained by

$$w_t = (n+1)c_{p_{He}}T_3\eta_t\left[1 - \frac{1}{\Pi_t^{\frac{\gamma-1}{(n+1)\gamma}}}\right] \qquad (2)$$

where $\Pi_t = \zeta\Pi_c$, with ζ being a factor less than unity to allow for pressure losses in pipes and heat exchangers throughout the gas cycle. The resulting output of the gas cycle per unit mass flow of helium could thus be obtained by

$$w_g = w_t - w_c \qquad (3)$$

while the corresponding heat added is calculated by

$$q_g = c_{p_{He}}[(n+1)T_3 - T_{2a} - nT_4] \qquad (4)$$

The calculation of the power developed by the Rankine cycle is obtained with the help of a Mollier chart. Neglecting the feed-pump work and referring to Fig. 3, the net output of the Rankine cycle per unit mass flow of steam is given by

$$w_s = (h_a - h_b) \qquad (5)$$

whereas the heat added in the boiler is calculated by

$$q_s = (h_a - h_d) = [h_a - (T_d - 32)] \qquad (6)^4$$

In examining Fig. 2, it can be seen that the hot helium leaving the regenerator is used in a counterflow heat exchanger to heat the feedwater. It is assumed here that the temperature difference between helium and water is about 15 F at the hot end and 10 F at the cold end of the heat exchanger. An energy balance shows that in order to satisfy this condition, m_s pounds of steam (feedwater) have to be used for each pound of helium such that

$$c_{p_{He}}(T_{4a} - T_1) = m_s c_{p_{H_2O}}(T_d - T_c)$$

Given $c_{p_{He}} = 1.242$ Btu/lb-F and considering that according to the foregoing assumption, $T_{4a} = T_d + 15$ and $T_1 = T_c + 10$, the relation can be rearranged to obtain

$$m_s = 1.242 + \frac{5.21}{15 + (T_{4a} - T_c)} \qquad (7)$$

As indicated by equation (7), the value of m_s depends entirely on the exhaust-gas temperature from the regenerator, T_{4a}, and the steam condensation temperature, T_c. An increase of the steam flow rate in the Rankine cycle above the value of m_s calculated by equation (7) violates the basic assumptions on the energy balance, while a decrease represents an incomplete utilization of the waste heat in the helium.

The combined-cycle thermal efficiency follows directly from the foregoing relations, equations (1) through (7)

$$\eta_{th} = \frac{w_T}{q_T} = \frac{w_g + m_s w_s}{q_g + m_s q_s} = \frac{w_t - w_c + m_g w_s}{q_g + m_s q_s} \qquad (8)$$

Choice of Cycle Parameters

The numerical evaluation of the combined cycle described in the foregoing is conducted using the following assumptions for the gas-turbine cycle:

- Specific heat of helium at constant pressure (assumed constant), $c_{p_{He}} = 1.242$ Btu/lb-F.
- Isentropic exponent for helium, $\gamma = 1.662$.
- Isentropic efficiency for compression, $\eta_c = 87$ percent.
- Isentropic efficiency for expansion, $\eta_t = 89$ percent.
- Pressure loss factor, $\zeta = 0.935$.
- Inlet temperature to compressor, $T_1 = 90$ F.
- Regenerator effectiveness, $\chi_R = 0.75$.

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[4] $h_d = (T_d - 32)$ is based on the assumption that the specific heat for water is 1 Btu/lb-F and that the enthalpy of water is zero at 32 F.

In choosing the appropriate steam cycle for use in combination with the helium gas cycle, the following should be considered: The condenser pressure of the Rankine cycle should be chosen as low as possible for favorable operation of the Rankine cycle while at the same time matching the inlet temperature to the compressor of the gas cycle. The 10 F temperature difference previously assumed between the cold helium and water condensate would result in a condenser temperature of 80 F. The corresponding vapor pressure of water is about 1 in. Hg, which determines the condenser pressure. On the other hand, the inlet pressure and temperature to the steam turbine may be changed. These changes are basically intended to examine the effect of varying the Rankine-cycle efficiency on the performance of the combined cycle. In an attempt to reduce the amount of calculation without endangering the generality of the results, only one parameter is varied.

In this analysis, a fixed inlet steam temperature of 1000 F was selected, while the pressure was varied between 865 to 1465 psia. Assuming a steam-turbine isentropic efficiency of 85 percent, the Rankine efficiency was calculated for different conditions of the foregoing pressure range (neglecting feed-pump work). Table 1 shows the results of this calculation.

A single Rankine cycle was chosen, from Table 1, to be combined with the gas cycle of a particular arrangement. Meanwhile, the pressure ratio of this gas cycle was varied between 2 and 7 for each combination, while the maximum gas temperature was set at 2060, 2260, or 2460 R.

Analysis of Results

Selecting a representative sample of the results obtained, it is possible to demonstrate the improvement in the thermal efficiency of the combined cycle. For this purpose, one can consider the different arrangements obtained by combining one of the Rankine cycles in Table 1 with a gas cycle operating with different parameters. By choosing cycle number 2 in the table (operating with superheated steam at 1000 F and 1065 psia at a condenser pressure of 1 in. Hg), the resulting Rankine thermal efficiency is 36.57 percent. This cycle is combined with two main gas-cycle arrangements:

Non-reheating helium gas cycle
2 Single reheating helium gas cycle.

In both arrangements, none, one, or two intercoolings were used as an additional parameter. The results obtained are plotted in Figs. 4 and 5 for the whole range of

TABLE 1

Cycle	Inlet Pressure	Rankine Efficiency %
1	865	35.94
2	1065	36.57
3	1265	37.1
4	1465	37.58

Fig. 4 Efficiency of the non-reheating combined cycle.

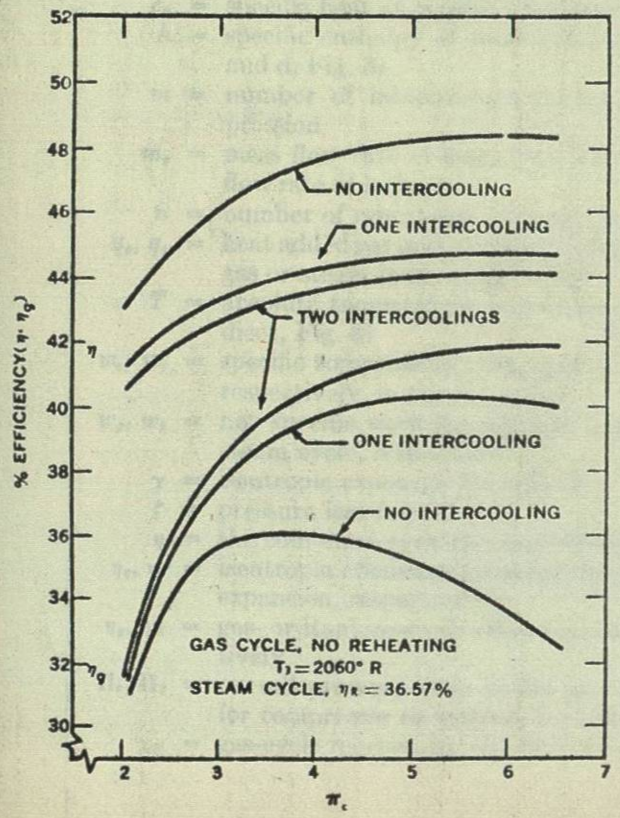
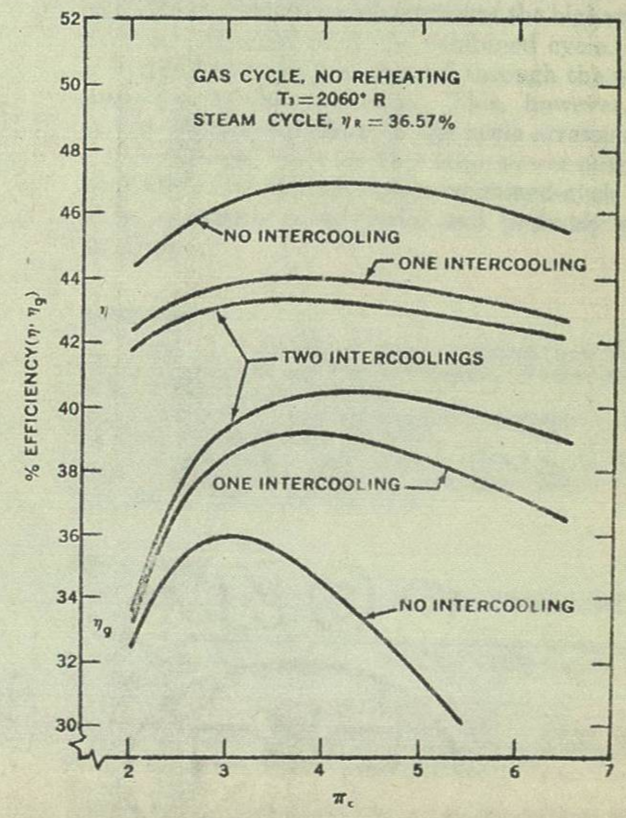


Fig. 5 Efficiency of the combined cycle with one reheating.



variation in the compressor pressure ratio. Both the gas-cycle thermal efficiency, η_g, and the combined-cycle thermal efficiency, η, are represented. The maximum gas-cycle temperature in both arrangements was selected to be 2060 R. Fig. 4 shows the results for the non-reheating arrangement, while Fig. 5 includes the results for single reheating. From these plots the following is evident:

- The thermal efficiency, η, is improved appreciably over the individual cycle efficiencies through the utilization of the thermal energy in the exhaust helium leaving the regenerator to heat the feedwater on the Rankine cycle.
- The efficiency improvement is higher in the case of no intercooling than either for one or two intercooling stages because the thermal energy in the exhaust gases decreases as the number of intercoolings increases. For example, the thermal efficiency of a non-reheating combined cycle (Fig. 4) at a compressor pressure ratio of 3 has increased from 36 to 46.5 percent without intercooling, while when using one intercooling the efficiency increases only from 38.5 to 43.8 percent. Even less improvement is obtained when two intercoolings are used.
- The compressor pressure ratio for maximum efficiency changes to a higher value through the use of the combined cycle. The shift is smaller in the case of non-reheating (Fig. 4) than with one reheating (Fig. 5), and it decreases upon increasing the number of intercoolings.

Nomenclature

c_p = specific heat at constant pressure
h = specific enthalpy of steam (index a, b, and d, Fig. 3)
m = number of intercoolings during compression
m_s = mass flow rate of steam per unit mass flow rate of helium
n = number of reheatings during expansion
q_g, q_s = heat added per unit mass flow rate in the gas or steam cycle, respectively
T = absolute temperature (for different indices, Fig. 3)
w_c, w_t = specific compression or expansion work, respectively, in the gas cycle
w_g, w_s = net specific work for the gas cycle or steam cycle, respectively
γ = isentropic exponent for helium
ζ = pressure loss factor
η = the combined-cycle thermal efficiency
η_c, η_t = isentropic efficiency for compression or expansion, respectively
η_g, η_R = gas- or Rankine-cycle efficiency, respectively
Π_c, Π_t = overall pressure ratio in the gas cycle for compressor or turbine, respectively
χ_R = gas-cycle regenerator effectiveness

After examining these results it can be concluded that the highest improvement in the combined-cycle efficiency is obtained with the simplest gas cycle. A combined thermal efficiency of about 47 percent is achieved with a simple gas cycle (no intercooling and no reheating) at a moderate compressor pressure ratio of about 3.5. At the same pressure ratio, when using one reheating, the combined-cycle efficiency is about 49 percent. The latter, however, could be raised to about 50 percent by increasing the compressor pressure ratio to about 6. Although this is a higher thermal efficiency, it is accompanied by two main disadvantages. First, a helium turbomachine designed to operate at a pressure ratio of 6 has a relatively large number of stages, which may cause design difficulties. Second, reheating the helium within a nuclear reactor is difficult and rather complicated.

An investigation into the effect of other operating parameters on the combined-cycle efficiency, limiting the study to the simple gas cycle as being the most favorable, shows that the combined efficiency increases linearly with increasing Rankine-cycle efficiency. On the other hand, an increase of about 100 F in the maximum gas-cycle temperature produces an average improvement of one point in the combined-cycle efficiency.

Summary

An improvement of the combined thermal efficiency is achieved above the thermal efficiency of either the gas or Rankine cycle. The most favorable combination is found to be between a simple gas cycle with no intercooling or reheating and a simple Rankine cycle. The efficiency improvement under these conditions amounts to about 10 points at a compressor pressure ratio of 3.5. This does not represent the highest possible efficiency obtained with the combined cycle. Greater improvement could be achieved through the use of one reheating in the gas cycle. This, however, is connected with complication in the cycle arrangement.

It is believed that, for very large power outputs (i.e., 2000 MWe and above), such a combined-cycle arrangement represents a successful and probably economic solution.

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