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# CYCLE EFFICIENCY OF AIR-COOLED STEAM POWERPLANTS

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16. Abstract

Steam powerplants for mobile applications such as automobiles must use air-cooled condensers to reject waste heat. A preliminary study is made of the effect of major design parameters on the efficiency of various steam cycles of this type. The efficiency of a typical system of the type usually considered for steam cars can be improved by a factor of 1.3 through use of a more complex system that employs an air turbine.

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# CYCLE EFFICIENCY OF AIR-COOLED STEAM POWERPLANTS

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## SUMMARY

Steam powerplants for mobile applications such as automobiles must use air-cooled condensers to reject waste heat. A preliminary study is made of the effect of major design parameters on the full-power efficiency of various steam cycles of this type. A typical overall efficiency for the system usually considered for steam cars is 0.155. An alternative cycle is derived that reduces the penalty of driving the forced-convection boiler and condenser fans. This cycle has an efficiency of 0.20, but requires a more complex system including an air turbine. Other important factors that enter into selection of the best cycle, such as component weights and volumes, off-design characteristics, cost, etc., are not considered in this study.

## INTRODUCTION

Rankine-cycle powerplants employing water as the working fluid have been in commercial use for hundreds of years. Recently interest has renewed in automotive applications due to the low air-pollution characteristics of such external-combustion engines.

Steam cycles achieve overall efficiencies in excess of 40 percent in large central powerstations, where waste heat can be rejected to lakes or streams at low temperatures (and subatmospheric pressures) in a condenser. For mobile applications, in the absence of a low-temperature heat sink, the engine can be operated in a noncondensing mode, that is, exhaust steam is discharged at atmospheric pressure. This technique, commonly used in steam locomotives, leads to lower overall efficiency and a continual need for large quantities of water to refill the boiler. For automobiles, where the weight and volume of extra boiler-feed water cannot be readily accommodated, the noncondensing mode is undesirable for normal operation. In order to avoid water loss, the engine exhaust steam must be condensed and recirculated through the cycle.

The only available sink for the waste heat removed in the automobile condenser is the atmosphere. Because ambient air is usually higher in temperature than lake or river water, the efficiency of the mobile system tends to be lower than that of stationary powerplants. Moreover, in order to reduce both boiler and condenser size, fans must be provided to improve the air-side heat transfer. The power required to drive the fans further reduces overall cycle efficiency.

The purpose of this study is to consider various steam cycles in order to see how this efficiency problem can be minimized. Only preliminary estimates of design-point performance are made, although the major design variables are perturbed in order to demonstrate their importance. To best reveal cycle differences, full-power operation with complete condensing is assumed. The benefit of not condensing all the steam during full-power operation is thus not considered. Also ignored is the probable improvement in efficiency during part-power operation when the fan work is greatly reduced.

## ANALYSIS

The usual heat and work relations for Rankine cycles as described in standard thermodynamics texts were employed in the study. Steam tables were used for the properties of water. The specific heat of both air and combustion products was taken as 0.24 Btu/(lb)($^{o}$F) (1.05 kJ/(kg)($^{o}$C)).

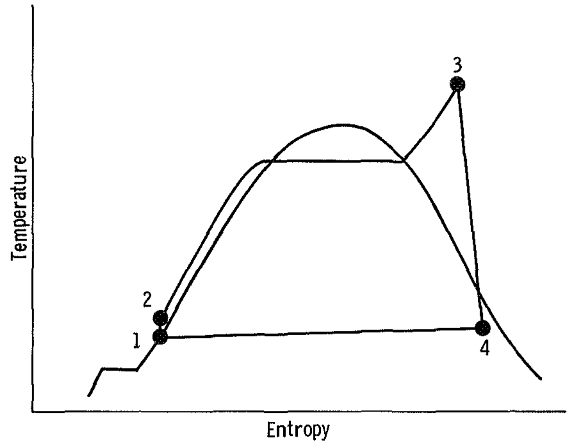
The adiabatic efficiency of the engine, which could be either a turbine or a reciprocating device, was a constant 0.70. Adiabatic efficiency of air fans and turbines was generally 0.87. Feed-water pump work and nonadiabatic thermal losses were assumed to be small and were ignored. Combustion efficiency was 100 percent. Other assumptions are mentioned at appropriate points in the following discussion.

## RESULTS AND DISCUSSION

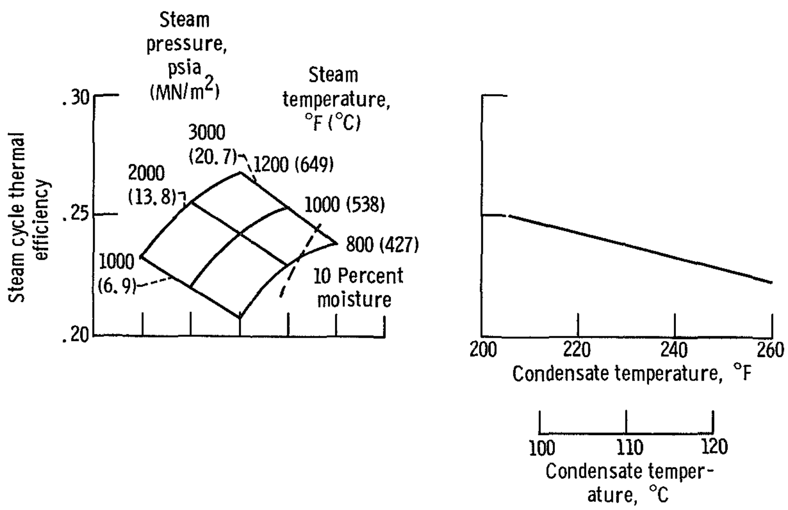
### Basic Steam Cycle

As illustrated in sketch (a), a temperature-entropy diagram, the basic Rankine cycle consists of four processes: liquid water (point 1) is compressed by a pump to high pressure (2); the water is heated, boiled, and possibly superheated (2-3); the steam is expanded through an engine, yielding mechanical work (3-4); and finally the low-pressure, low-temperature steam is condensed (4-1).

Cycle performance. - The thermal efficiency (shaft work output ÷ heat energy input) for the basic steam cycle is shown in figure 1. Figure 1(a) shows that efficiency im-




(a)




(a) Effect of steam pressure and temperature; condensate temperature, 220° F (104° C).

(b) Effect of condensate temperature; steam pressure, 2000 psia; steam temperature, 1000° F (538° C).

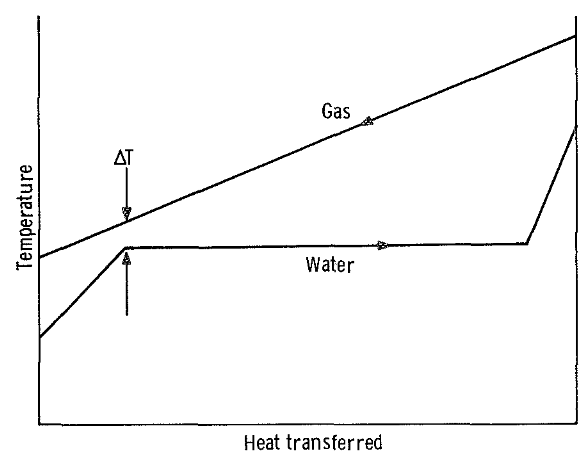
Figure 1. - Efficiency of basic steam cycle; engine efficiency, 0.70; no fan work.

proves with increasing pressure and temperature of the steam produced in the steam generator or boiler. On the other hand, increases in these parameters tend to cause heavier structural weights and/or higher cost and may increase the difficulty of designing an efficient engine to extract mechanical work. Another factor arguing against excessively high temperatures is the problem of finding thermally stable cylinder lubricants if reciprocating engines are used. However, low temperatures, if combined with high pressures, tend to result in wet steam (i. e., the presence of liquid water) after the expansion through the engine. A line of constant 10 percent moisture is indicated in the figure. Points to the right of that line result in more than 10 percent moisture. This is undesirable as it may cause blade erosion in turbines or efficiency losses in both turbines and reciprocating engines.

In view of these considerations, a nominal or reference steam condition of 2000 psia ($13.8\ MN/m^2$) and $1000^\circ$ F ($538^\circ$ C) has generally been assumed throughout the analysis. Whether or not this represents a proper design condition for low-cost mobile powerplants will require further study.

Figure 1(b) shows the effect of condenser temperature on efficiency. The lower the temperature, the better. However, low temperatures make it more difficult to reject heat to the atmosphere, which then requires bulkier and heavier heat exchangers. Furthermore, if the temperature is reduced below $212^\circ$ F ($100^\circ$ C), the pressure within the condenser is subatmospheric. Unless the system is hermetically sealed, air will then tend to leak in, with a consequent loss in condensing effectiveness. Special means must then be provided to remove the noncondensable gases. To avoid this complication, it is desirable to maintain atmospheric or higher pressure in the steam system. For future reference, at a condensate temperature of $220^\circ$ F ($104^\circ$ C) the thermal efficiency of the basic steam cycle is 0.243, using a nominal engine efficiency of 0.70. The overall efficiencies of the following complete cycles (at the same steam conditions) will be lower than this value by amounts depending on how the fan power is extracted from the basic steam cycle and the boiler efficiency.

Heat-exchanger assumptions. - Heat exchanger design as such will not be studied in this report. However, in the present thermodynamic study it was necessary to assume representative characteristics for the boiler and condenser. Specifically it was usually assumed that the steam-side pressure drop (for each heat exchanger) was 10 percent (this had trivial effect on net performance since pump work was ignored) and the air-side pressure drop was 5 percent. In addition it was usually assumed that the driving temperature difference between the outside air (or combustion gas) and the



(b)

inside steam (or water) was always at least 50° F (28° C).

It is not adequate to examine temperature differences at only the inlet or exit of the heat exchangers. As a result of the two-phase flow, the minimum temperature difference can occur at some point within the heat exchanger. This is pictured in sketch (b), which represents the temperature changes occurring in the boiler. It is desired to have the gas temperature leaving the combustor as low as possible for structural reasons and to have the gas temperature leaving the heat exchanger as low as possible for minimum exhaust-gas heat loss. This dictates the counter-flow arrangement pictured. Note, however, that the minimum temperature difference between the gas and the water occurs at the point where the water has been heated to its boiling point. The temperature difference $\Delta T$ at this station, usually called the pinch point, must be positive for heat to flow in the proper direction and sufficiently large to avoid excessive heat-transfer surface area.

The constraint thus placed on gas entrance and exit temperatures and the gas-to-steam flow ratio is illustrated in figure 2. As an example of the importance of this factor, suppose a gas entrance temperature of 1100° F (593° C) and an exit temperature of 300° F (149° C) were desired. Both values are 100° F (56° C) higher than the corresponding final and initial temperatures of the water. Nevertheless, the figure shows that this results in a negative value of pinch $\Delta T$. If the pinch $\Delta T$ is required to be 50° F (28° C), it would be necessary to raise the gas entrance temperature to 1300° F

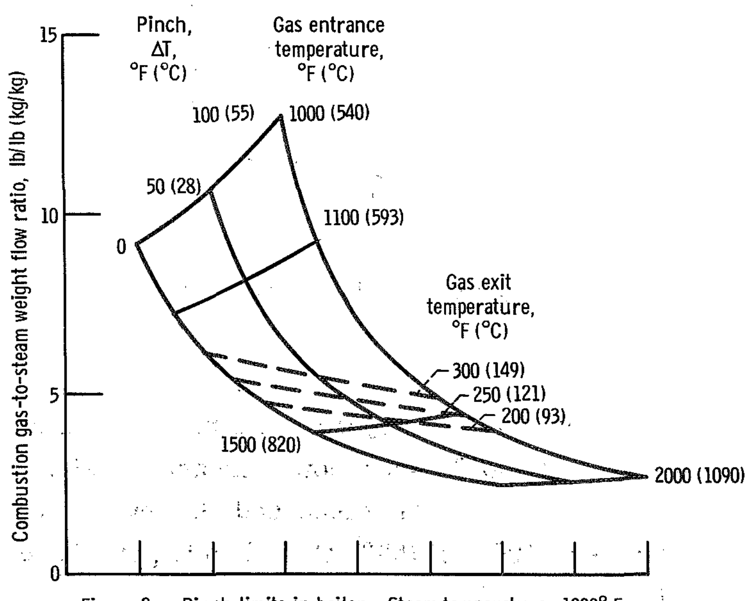



Figure 2. - Pinch limits in boiler. Steam temperature, 1000° F (538° C); steam pressure, 2000 psia (13.8 MN/m$^2$); feed-water temperature, 200° F (93° C).

(704° C), provided the exit temperature is held fixed. Higher gas exit temperatures would also relieve the pinch constraint but would increase the heat losses in the exhaust gas.

## Cycle A

The first complete cycle to be considered is the one most commonly incorporated in current steam-car designs. This reference case will accordingly be examined somewhat more completely than the other systems. As pictured in figure 3, this cycle consists of the basic steam cycle coupled with two separate gas streams, one to heat the boiler, and the other to cool the condenser. Two separate fans are provided for forced convection, whose power is presumed to be extracted directly from the engine.

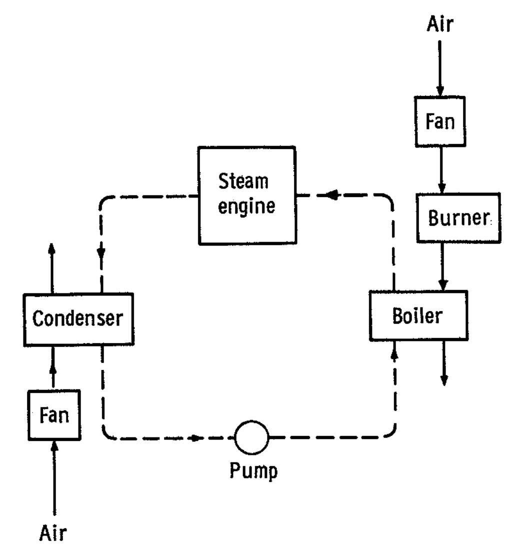



Figure 3. - Schematic diagram of cycle A.

Figure 4 presents the effect on overall cycle efficiency of variations in the major design parameters. Figures 4(a) and (b) show how increases in steam temperature and pressure improve efficiency. As previously discussed, however, these parameters will be nominally limited to values of 1000° F (538° C) and 2000 psia (13.8 $MN/m^2$), as indicated by the circled points.

The effect of condenser temperature is shown in figure 4(c). Note that the beneficial effect on the basic steam cycle of lower temperature is finally outweighed by the increased difficulty of rejecting heat from the condenser to the air. Greater and greater amounts of air (relative to the steam flow) are required in this region, with consequent

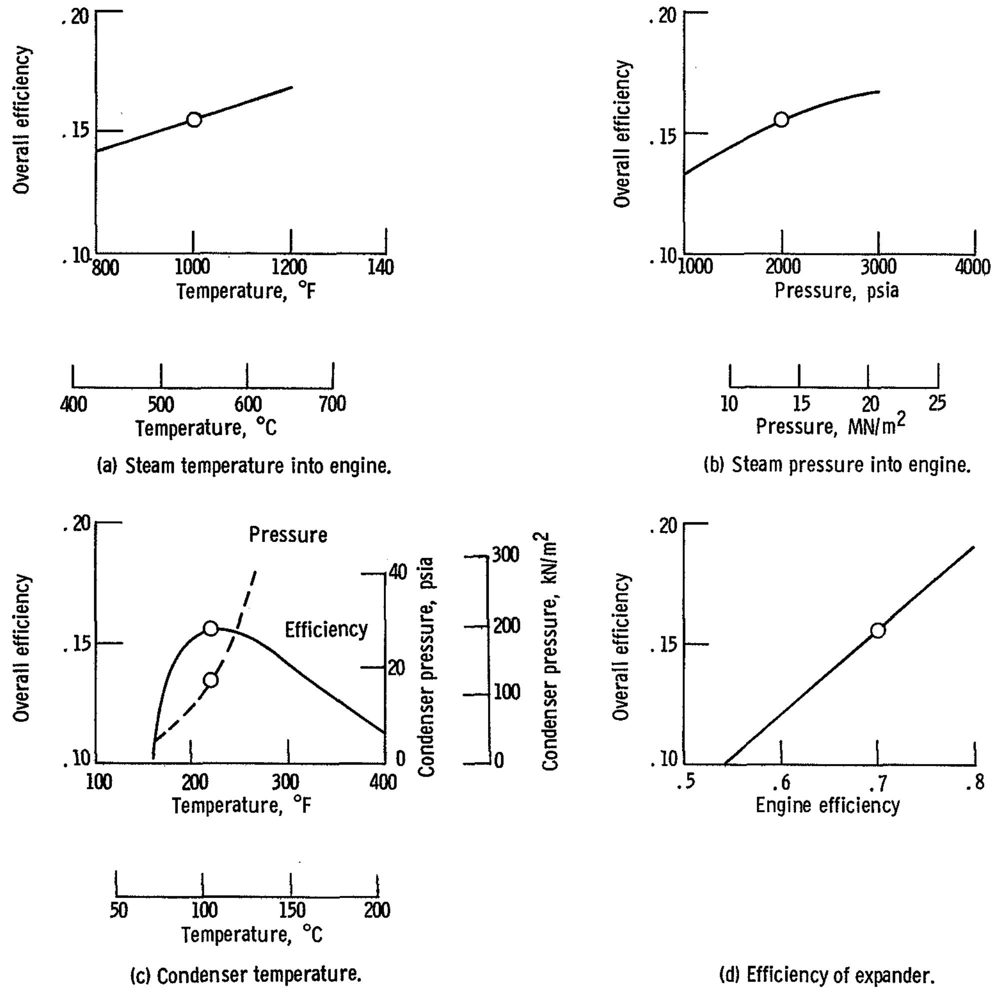
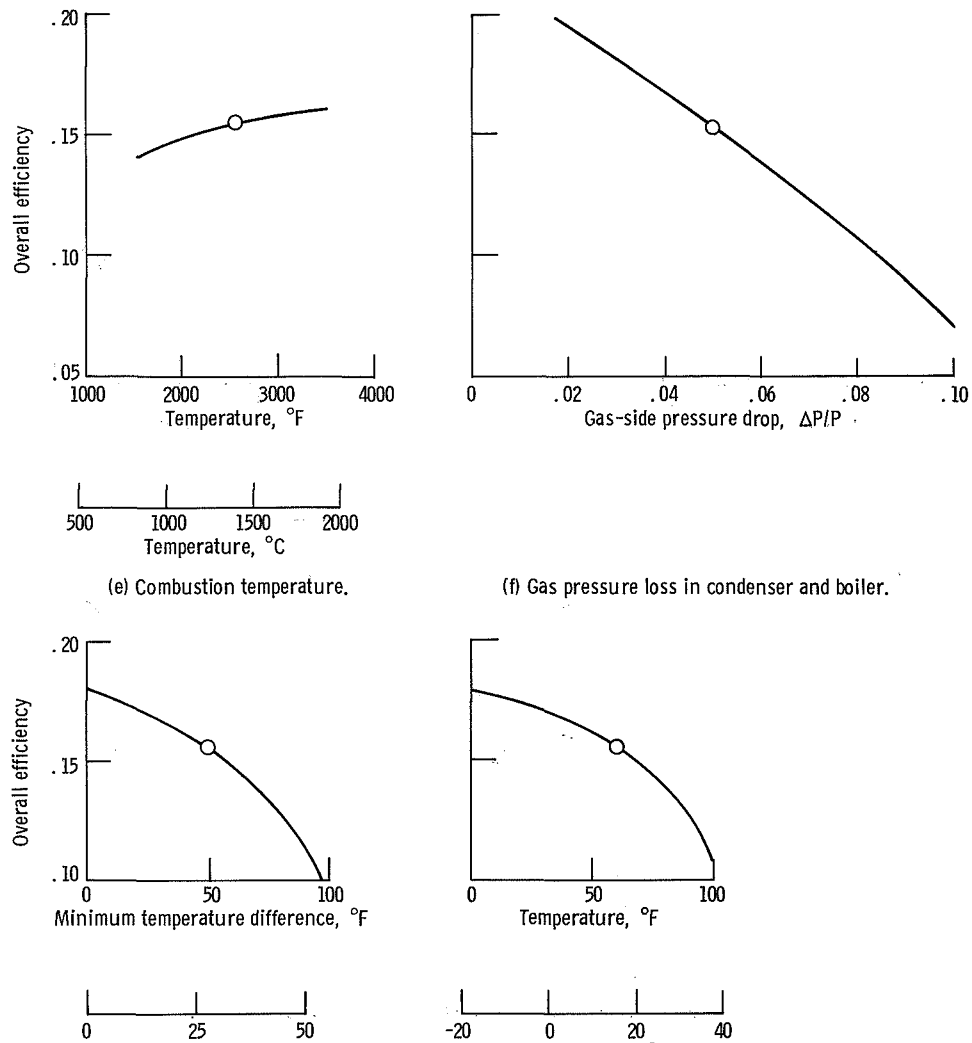



Figure 4. - Effect of design parameters on performance of cycle A.

demands for more power to drive the condenser fan. The optimum condenser temperature is about 220° F (104° C), which is used as the reference value in the other parts of figure 4. As previously discussed, this yields a desirably greater-than-atmospheric pressure within the condenser.

The effect of engine efficiency is displayed in figure 4(d). Overall efficiency increases with engine efficiency with a slope somewhat greater than just a direct proportionality. This is because higher engine efficiencies both increase the work output and reduce the amount of condenser waste heat with a corresponding saving in condenser fan power. The reference engine efficiency of 0.70 is representative of reciprocating engines at the small power levels typical of automobiles or of steam turbines at larger power levels. Other factors are also important, of course, in selecting an engine type, such as weight, volume, torque characteristics, durability, and cost.

The effect of combustion temperature is shown in figure 4(e). The previously de-




(e) Combustion temperature.

(f) Gas pressure loss in condenser and boiler.

(g) Steam-air temperature difference in condenser.

(h) Ambient temperature.

Figure 4. - Concluded.

scribed pinch effect requires that the combustion temperature be at least about 1300° F (704° C). Still higher temperatures are beneficial, however, since this reduces the amount of combustion gas required to heat the steam. Less heat is then lost in the exhaust gases leaving the boiler. (It is assumed here that the exhaust temperature is held fixed at 50° F (28° C) higher than the boiler feed-water (condensate temperature).) The overall efficiency increases only slightly for temperatures over 2540° F (1394° C), which was selected as the reference value for structural reasons. Even lower temperatures might be more prudent to lessen the possibility of boiler burnout if the water flow were interrupted for any reason.

One of the main energy losses in this cycle is the work required to drive the condenser fan and, to a lesser degree, the boiler fan. The fan work is primarily a function of the pressure ratio required to overcome the air friction losses in the heat exchangers. The importance of this factor is shown in figure 4(f). Doubling the pressure drop (for each heat exchanger) from the reference value of 0.05 to 0.10 results in a 53 percent loss in cycle efficiency. Selection of the best value of pressure drop will involve a complicated tradeoff between fan complexity, overall efficiency, and heat-exchanger size and weight. It should be recognized that the justification for the present study rests heavily on the assumption that such tradeoffs will lead to appreciable pressure drops, in the order of 0.05, for example. If the pressure drop were negligible, the fan power would be trivial, the overall efficiency of this reference cycle would approach that of the basic steam cycle, and there would be little incentive to examine alternative cycles.

An important factor in heat-exchanger design is the allowable difference between the bulk air temperature on one side of the tubes and the bulk steam temperature on the other. For two-phase flow this is a more useful criterion than the more-common quantity of "effectiveness." As shown in figure 4(g), small temperature differences are beneficial in terms of overall efficiency. On the other hand, heat exchanger weight increases very rapidly as the temperature difference approaches zero.

In the other parts of this study, the ambient temperature has been taken as $60^{\circ}$ F ($16^{\circ}$ C). Figure 4(h) shows that operation at higher temperatures severely penalizes overall efficiency. This is due to an increase in the amount of air needed to cool the condenser with a consequent increase in fan power. This penalty could be minimized by designing the condenser for a lower air-pressure drop, which would cost additional condenser weight. Or, the system could be operated in a partially noncondensing mode during full-power operation on hot days, which would require periodic recharging with more water.

For the reference cycle described by the circled point, the peak value of overall efficiency is 0.155. This compares with a value of 0.243 for the basic steam cycle. The difference between the two is primarily due to the large quantity of air required to cool the condenser and the associated fan power. Of the total mechanical power generated by the engine, 27 percent is needed to drive the condenser fan. Much less air is required to heat the steam in the boiler because of the fairly high combustion temperature. As a result, only 3 percent of the engine output is needed to drive the boiler fan. And not all of this 3 percent is a loss, since it adds thermal energy to the air, reducing the amount of fuel that must be burned to achieve the desired combustion temperature.

A more direct loss in the boiler process is that, of the total amount of heat energy added in the burner, about 9 percent still remains in the exhaust gas that emerges from the boiler.

A number of alternative cycles will be examined next, in order to see if the performance of this reference cycle can be improved upon. Unless otherwise stated the various design variables will be set at the values indicated by the circled points in figure 4.

## Cycle B

An alternative cycle that is based on turbojet practice is sketched in figure 5. A gas turbine is placed between the burner and the boiler to generate shaft work to drive the

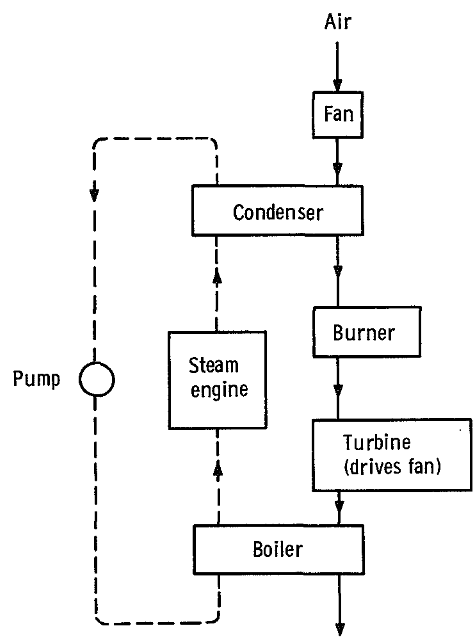



Figure 5. - Schematic diagram of cycle B.

fan. A single airstream is utilized to cool the condenser and later heat the boiler. By this arrangement it is hoped to reduce the large amount of energy lost to the condenser-cooling air in the previous cycle.

The effect of fan pressure ratio on the overall efficiency of the cycle is shown in figure 6(a). High pressure ratios are beneficial but require more turbine work to drive the fan, which lowers the gas temperature entering the boiler. The largest usable pressure ratio is the one that first encounters the minimum allowable temperature difference in the boiler. For the case illustrated the boiler pinch point limits the pressure ratio to a value of 1.32. Combustion temperature is a dependent variable in this cycle, with a value of about $1200^\circ$ F ($649^\circ$ C).

The effect of condensate temperature is shown in figure 6(b). Overall efficiency increases with temperature up to $710^\circ$ F ($377^\circ$ C). Thermal efficiency of the basic steam

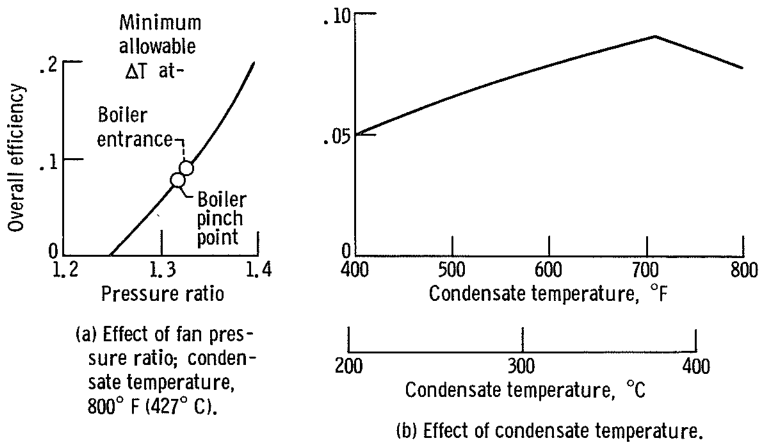



Figure 6. - Effect of design parameters on efficiency of cycle B.

cycle falls with higher temperature as previously discussed (fig. 1(b)). However, this is more than compensated for by the accompanying reduction in air flow needed to cool the condenser. The waste heat loss in the gas leaving the boiler is thus lowered, which improves the overall efficiency. Below $710^{\circ}$ F ($377^{\circ}$ C) the pressure ratio is limited by the temperature difference at the boiler entrance. Above $710^{\circ}$ F ($377^{\circ}$ C) the temperature difference at the boiler pinch point becomes constraining, and the pressure ratio and overall efficiency both drop.

The maximum efficiency is only 0.09. It is low because, even with a rather high condensate temperature, a large amount of air is required to cool the condenser. The large airflow then carries with it an excessive amount of thermal energy after leaving the boiler.

## Cycle C

The next cycle considered (fig. 7) strikes a compromise between cycles A and B. Separate airstreams for condenser and boiler are retained as in A, since the use of a single stream resulted in severe cycle constraints. However, a gas turbine is used to drive both fans as in B, rather than extracting the work from the steam engine.

The effect of some important cycle parameters is shown in figure 8. As shown in figure 8(a), high combustion temperatures improve the overall efficiency. However, the structural problems of exposing the turbine blades to these temperatures force the use of more moderate temperatures. A rather optimistically high reference temperature of $1730^{\circ}$ F ($943^{\circ}$ C) is encircled in the figure. This is about as high a turbine temperature as can be presently achieved in modern aircraft jet engines without the complication of blade cooling. Even so, expensive materials with limited lifetimes must

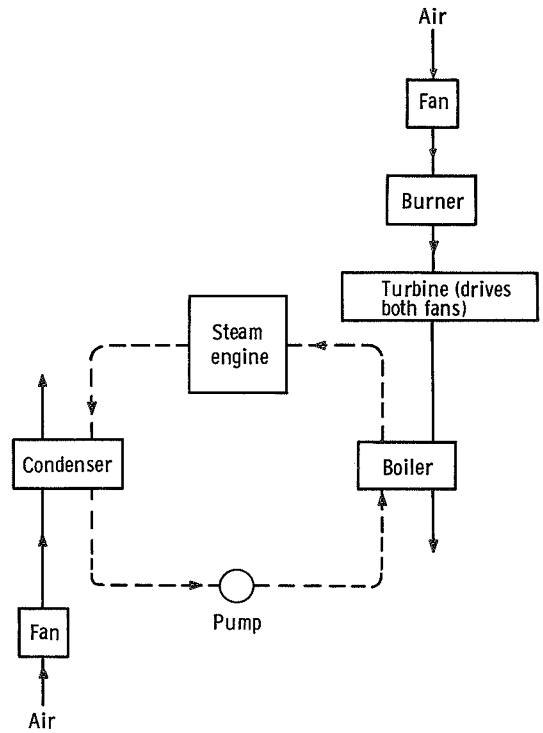


Figure 7. - Schematic diagram of cycle C.

then be used. Still lower combustion temperatures are thus probably more practical and do not cause much loss in overall efficiency. A value of about $1500^{\circ}$ F ($815^{\circ}$ C) is as low as can be accepted for the reference steam cycle, however, due to the pinch effect in the boiler.

The effect of condensate temperature is shown in figure 8(b). As in cycle A, a temperature of $220^{\circ}$ F ($104^{\circ}$ C) is selected to avoid subatmospheric steam pressures. Lower condensate temperatures are also undesirable as they cause higher turbine-inlet temperatures for any given level of gas temperature entering the boiler.

The resulting overall efficiency is just under 0.20, which is significantly better than the 0.155 achieved by cycle A. This improvement is obtained at the cost of adding a high-temperature gas turbine to the system and a more complex burner fan that delivers a higher pressure ratio. However, it is interesting to note that the efficiency of this cycle is entirely independent of the efficiencies of the burner fan and the turbine, which may lower their cost. Any inefficiencies in these components merely add to the thermal energy of the boiler gas and correspondingly reduce the need for further heating by combustion.

Lower efficiencies do have some adverse effects, however. As indicated in figure 8(c), a lower fan efficiency tends to raise the required fan pressure ratio and the combustion temperature. Increases in pressure losses through the burner and boiler have the same effect (fig. 8(d)).

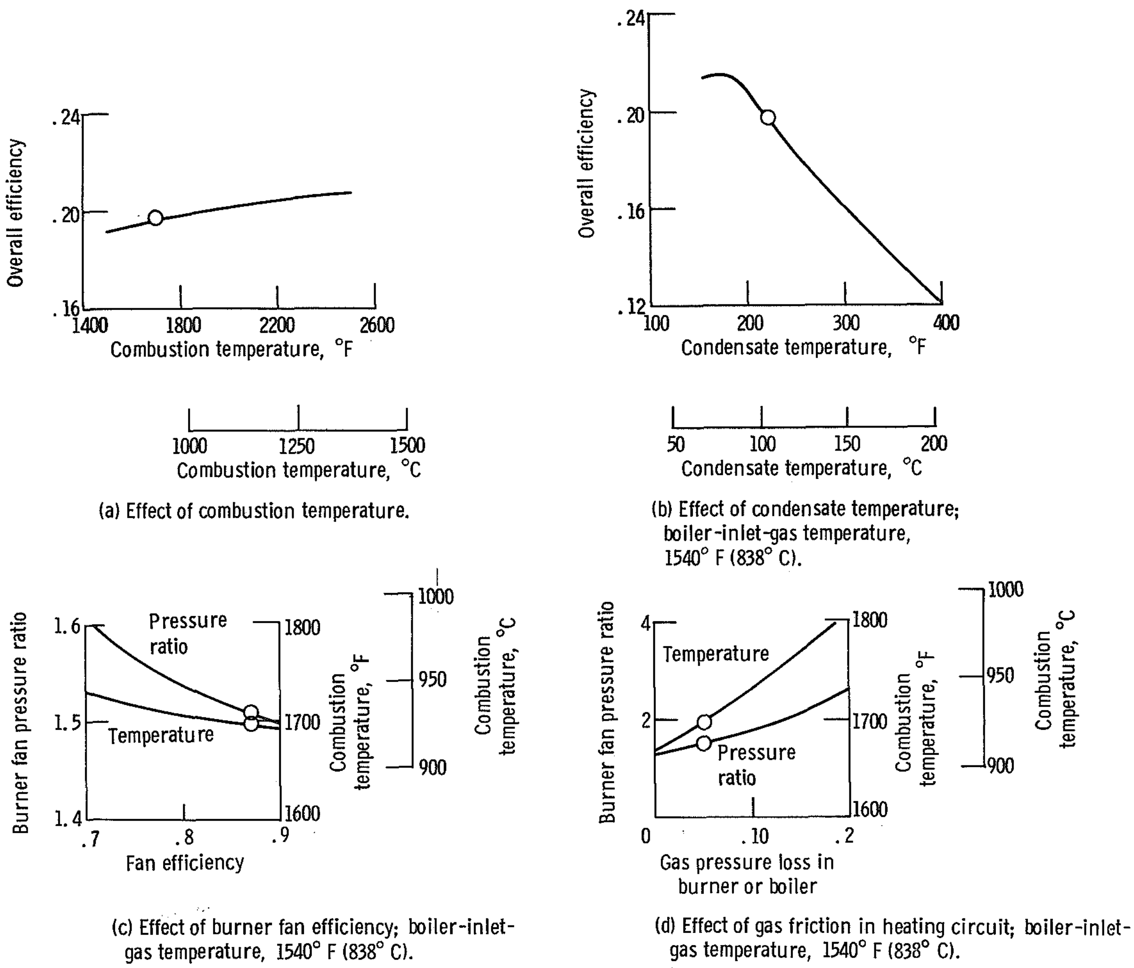



Figure 8. - Effect of design parameters on cycle C.

## Cycle D

The preceding cycle produced a high efficiency but exposed the gas turbine to possibly excessive temperatures. In order to alleviate this situation, the positions of the boiler and turbine were interchanged (fig. 9). The turbine is thus not exposed to the combustion gases until after they have been cooled during passage through the boiler.

The effect of combustion temperature is shown in figure 10(a). The benefit of temperatures above 2000° F (1093° C) is small. For the full range of combustion temperatures considered, the turbine-inlet temperature, as hoped, is very moderate and should pose no difficulties.

As seen in figure 10(b), the required burner fan pressure ratio is rather high - in the order of 4 to achieve near-maximum overall efficiency. Furthermore, this cycle

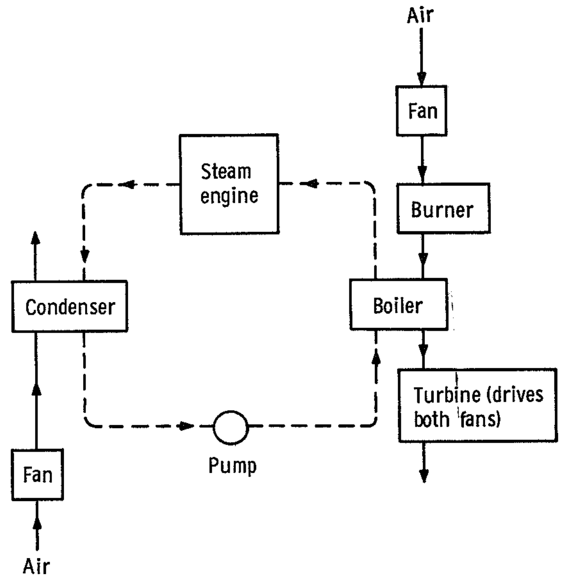
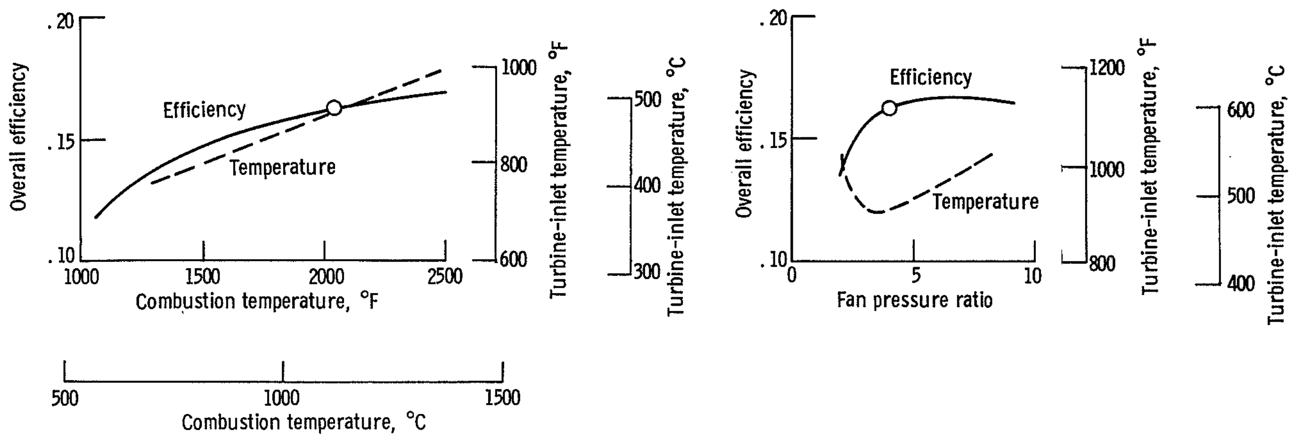



Figure 9. - Schematic diagram of cycle D.




(a) Effect of combustion temperature; fan pressure ratio, 4.

(b) Effect of burner fan pressure ratio; combustion temperature, 2040° F (1115° C).

Figure 10. - Effect of design parameters on cycle D.

does not enjoy the independence of cycle efficiency to component efficiency that was the case in cycle C. So the high-pressure-ratio fan and turbine cannot be carelessly designed or constructed, which may imply high cost. This situation comes about because the thermal energy of the gases leaving the turbine is not used in the boiler as in cycle C but is wasted. As a result the cycle efficiency is lower than that of cycle C. It is, in fact, insignificantly improved over the much simpler reference cycle A. The higher pressure of the gas in the burner and boiler would reduce the size and weight of these components, however, which is of definite advantage.

## Cycle E

The efficiency of the preceding cycle suffered because of the loss of thermal energy contained in the turbine-exhaust gas. A modified version of that cycle is now considered in which a heat exchanger (economizer) is used to recover much of this energy, by transferring it into the boiler-feed water (fig. 11).

The performance of this cycle as a function of burner fan pressure ratio is shown in figure 12. The overall efficiency is back to the 0.2 level achieved by cycle C. The re-

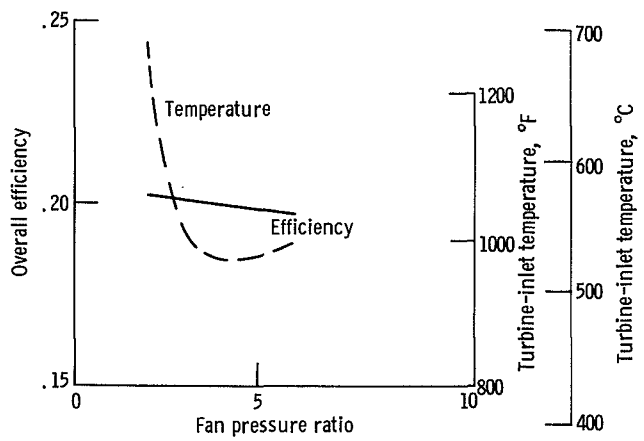



Figure 12. - Effect of burner fan pressure ratio on cycle E. Combustion temperature, 2040° F (1115° C).

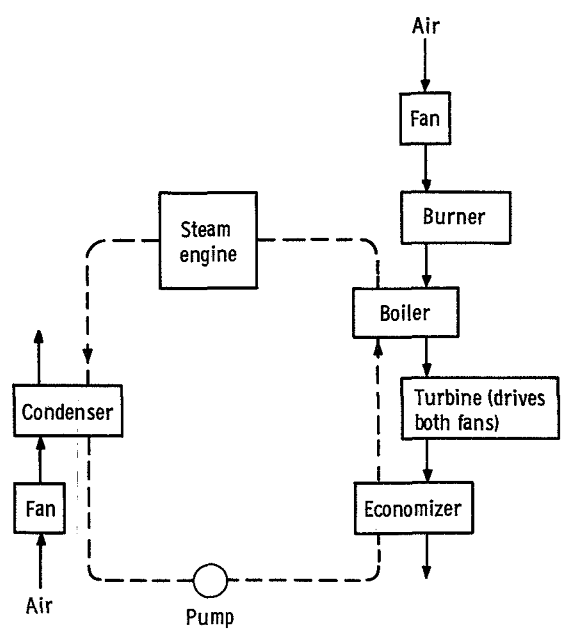



Figure 11. - Schematic diagram of cycle E.

quired turbine-inlet temperature is not excessive, especially if fan pressure ratios greater than 2 are accepted. This cycle, too, is insensitive to fan and turbine efficiencies.

## Comparison with Gas Turbine Cycle

Most of the preceding steam cycles incorporated a gas turbine to drive the forced-convection fans. It may therefore be of some interest to contrast these cycles with one where the gas turbine also generates the primary shaft power. In its simplest form this system consists of an air compressor followed by a burner and a turbine. The output of the turbine drives the compressor, and any surplus power is available for external work.

Efficiency of the simple gas turbine cycle is shown in figure 13. Performance is heavily dependent on the allowable turbine-inlet temperature. However, efficiencies

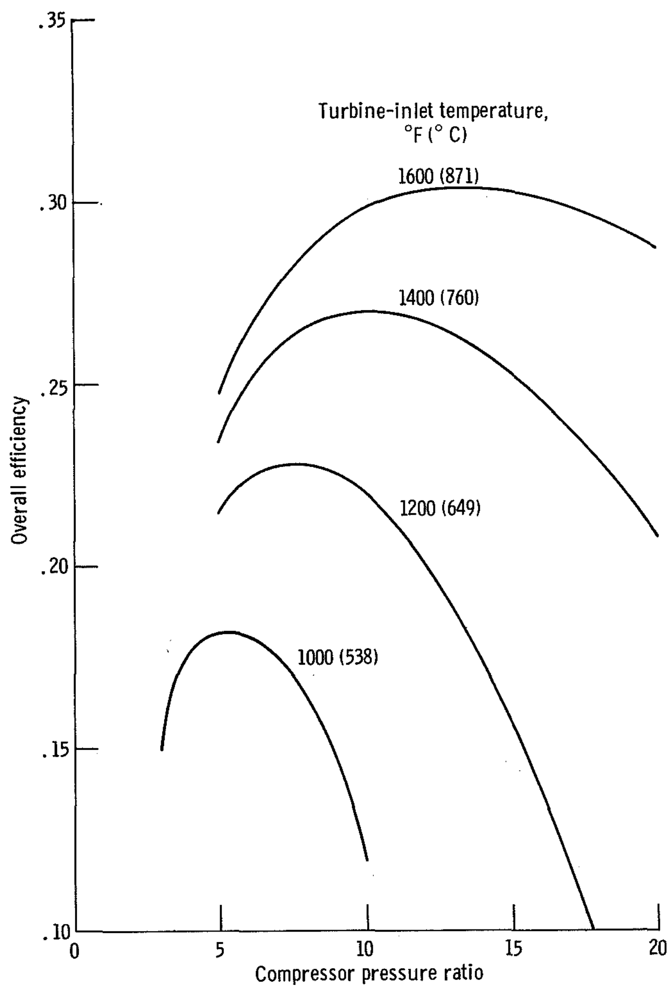



Figure 13. - Efficiency of gas-turbine cycle. Compressor and turbine efficiency, 0.87; combustor pressure drop, 0.05.

are achievable well above the 0.20 level that was obtained for the steam system. Even higher values can be achieved with the use of regeneration.

## CONCLUDING REMARKS

The efficiencies of a variety of steam cycles suitable for mobile applications were calculated. Full-power full-condensing operation was considered, which places the maximum demand on fan power. The fan power needs plus the heat loss in the combustion gas leaving the boiler reduce the overall efficiency of the basic Rankine steam cycle from 0.243 to 0.155 for a representative system with fans driven by the primary steam engine.

A more complex system in which the fans are driven by a gas turbine was derived that yields an overall efficiency of 0.20. An additional heat exchanger to transfer heat from the turbine-exhaust gas to the boiler-feed water was necessary in this system. The general efficiency level of all the cycles studied could have been increased by the use of more sophisticated Rankine cycles. For example, a regenerative steam cycle with interstage bleed could improve each cycle by perhaps 0.03.

Still better efficiency is available with a simple gas turbine cycle. However, selection of the best cycle requires consideration of many more factors than just the thermal efficiency. Component weights and volumes, off-design characteristics, transmission requirements, manufacturing cost, and maintenance must all ultimately be examined. In particular it should be stressed that the differences between the various steam cycles studied arose principally from the differences in how fan power was supplied. Fan power, in turn, is directly affected by the assumed heat-exchanger gas-side pressure drop. If smaller pressure drops were selected (which is a design decision based on consideration of the sort of factors listed above), then there would be less difference in the calculated overall efficiencies of the cycles.

Lewis Research Center,
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