3. BRAYTON CYCLE SYSTEMS

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This discussion is concerned with the Brayton or "gas turbine" powerplant, which does not currently represent a major contribution to the total world power output but is starting to find use in certain areas of applications and in the future could become a major power source. This type of engine has been developed in the last 30 years to a high degree of sophistication through its use in the aviation industry, and more recently it is being used for a myriad of applications covering a wide range of power levels.

The use of the "airbreathing" gas turbine in the utility industry began several years ago when such units were put on the line in a "peaking" capacity. Their features of low first cost and low maintenance made them ideal for this application, where plant efficiency and fuel cost were not dominant factors. Another use involves the combining of the gas turbine with a base-load steam system in order to increase the potential for achieving high cycle efficiency. A third use is that of the gas turbine in a more complicated fashion to increase its cycle performance and apply it as a straight base-load power source.

Still another area where the gas turbine is emerging as a power source is the closed loop configuration, where a gas such as helium is used as the working fluid. Such systems are in operation in Europe using a coal-fired heat source and are currently in the initial phases of being mated to a reactor heat source.

It is the intention of this discussion to review the principles of the gas turbine engine, with particular emphasis on the performance improvements made in the system through use of the advanced technology generated in the propulsion field. Special features of the cycle as applied to airbreathing power systems as well as closed loops, where gases other than air can be used, will be described.

OPEN CYCLES

SIMPLE CYCLE DESCRIPTION AND PERFORMANCE

A schematic representation of a simple open-cycle gas turbine system is shown in figure 3-1. Air entering the system at atmospheric pressure is compressed, heated by means of combustion of a fuel, expanded through a turbine, and exhausted back to the atmosphere. The power produced by the turbine is used to drive the compressor and the generator. This open cycle is shown thermodynamically in figure 3-2 on a temperature-entropy diagram. The dashed line represents atmospheric pressure. The three steps in the process, compression, heat addition, and expansion back to atmospheric pressure, are indicated. Gas turbine peaking units, in general, use this simple process. Often, the turbine in a peaking system will be divided into two sections, one driving the compressor and one driving the generator.

Two major factors affecting the performance of the gas turbine system are compressor pressure ratio and turbine inlet temperature. The influence of these factors on open-cycle plant performance is shown in figure 3-3. This figure is based on the assumption of a compressor efficiency of 88 percent and a turbine efficiency of 90 percent. These are polytropic or, as sometimes called, small-stage efficiencies and would correspond to stage efficiency in machines having a very large number of stages. All turbomachinery efficiencies subsequently referred to in this discussion will be polytropic efficiencies. Plant cycle efficiency is plotted against pressure ratio for turbine inlet temperatures of 1500°, 2000°, and 2500° F. Plant cycle efficiency is defined as the net plant power output divided by the heat added by fuel combustion. Plant efficiency can also be expressed as a net plant heat rate. As a point of reference, a plant cycle efficiency of 40 percent corresponds to a heat rate of 8530 Btu per kilowatt-hour. This figure shows, first of all, the significant improvements in efficiency that can be achieved by increasing turbine inlet temperature. Maximum cycle efficiencies are seen to be about 31 percent at 1500° F, 39 percent at 2000° F, and 44 percent at 2500° F. Current peaking units of advanced design operate at turbine inlet temperatures of about 1500° F and yield cycle efficiencies of about 30 percent. Another point to note from this figure is the importance of selecting the proper pressure ratio that is required for obtaining maximum cycle efficiency. For the simple open-cycle system considered herein, the optimum pressure ratio is about 15 for a turbine inlet temperature of 1500° F and increases to about 50 at 2500° F.

Another important performance consideration is the required flow rate, as this reflects the sizes required for the components. The influence of turbine inlet temperature on specific flow, which is the flow rate in pounds per second per megawatt of electric power, is shown in figure 3-4. These flows are for optimum pressure ratio. At 1500° F, the required flow rate is about 15 pounds per second per megawatt. At 2500° F, the flow is less than half of that at 1500° F. Figures 3-3 and 3-4 indicate the reasons why there is a great desire for increasing turbine inlet temperature.

Another important factor affecting plant performance is the efficiency of the turbomachinery. This is illustrated in figure 3-5, where plant cycle efficiency is plotted against turbine efficiency for a constant compressor efficiency of 88 percent and a turbine inlet temperature of 2000° F. As turbine efficiency increases from 80 to 90 percent, there is nearly a ten-point increase in cycle efficiency. Thus, a one-point change in turbine efficiency results in about a one-point or about a 3 percent relative change in cycle efficiency. Variations in compressor efficiency have a slightly smaller effect on cycle efficiency.

These two aspects, high turbomachinery efficiency and high turbine inlet temperature, were actually the principal reasons why the gas turbine engine did not enter into the realm of practicality until the late 1930's when aerodynamic and mechanical know-how developed to a point where the gap between performance requirements and achievability was closed. Some of the results of the technology advancements made since that time will now be discussed.

TURBOMACHINERY TECHNOLOGY

Extensive research and development on compressors and turbines for aircraft propulsion have been carried on for the past 30 years. This extensive study has provided a large background of experience and data which form the basis of current design procedures. These procedures permit design of compressors and turbines which produce high levels of performance and require a minimum amount of development.

Efficiency

An example of a compressor based on current aircraft technology is shown in figure 3-6. This is an eight-stage research compressor. Because it is a research unit, it is not of flight weight construction; this is evident from the boiler-plate-type casing and heavy flanges. The aerodynamics, however, are typical of the current state of the art for jet engine compressors. The peak polytropic efficiency of this unit is 90.5 percent. This efficiency was obtained at a pressure ratio of 7.5, which corresponds to an average stage pressure ratio of 1.29.

Weight and operating range are extremely important in aircraft propulsion systems, and for an aircraft engine, this compressor would be operated at a higher speed and pressure ratio condition than discussed and would, therefore, operate at a somewhat lower efficiency.

In addition to aircraft-type compressors, there has been some experience here at Lewis with large compressors which are more representative of industrial units. Figure 3-7 is a picture of one such unit. This is the number one compressor from the 10- by 10-foot supersonic wind tunnel flow system. This unit is 20 feet in diameter. The man working on the blade fairing indicates the size of this machine.

This compressor also has eight stages. The peak efficiency of this machine is 91 percent and the pressure ratio at this operating point is 2.4. This is an average stage pressure ratio of about 1.12, as compared with a value of 1.29 for aircraft-type compressors. This 20-foot-diameter machine was designed with moderate blade loadings and low rotative speed, which results in this low level of stage pressure ratio. The high level of efficiency may be partly due to the large size of the unit and partly due to the conservative design. In general, big compressors are somewhat more efficient than small ones. The efficiency of a compressor of this size, 20 feet in diameter, will probably be 1 percent higher than a similar unit of conventional aircraft engine size, about 3 feet in diameter. The examples discussed indicate that compressor efficiencies on the order of 90 percent or higher are attainable with current state-of-the-art designs.

The turbine situation is parallel to that for the compressor, and the state of the art for turbines is just as advanced. An example turbine of the type encountered in modern aircraft engines is shown in figure 3-8. This is a $2\frac{1}{2}$-foot-diameter research turbine currently being studied at Lewis. The efficiency for this unit is 92 percent, which is slightly higher than the values quoted for compressors. Efficiencies for larger turbines might be slightly higher, perhaps 93 percent. The aforementioned compressor and turbine efficiencies indicate that the values assumed for the cycle analysis are reasonable.

Stage Loading

It is evident that rather high overall pressure ratios will be required for high plant efficiencies. The optimum pressure ratio for a peak cycle temperature of 2500° F is 50. Because of stage matching problems it is impractical to try to achieve this in a single compressor, and more than one compressor in series will be required. The first compressor, which would be the largest in diameter, is assumed to have a design pressure ratio of 5. Conservative compressors, such as the 20-foot-diameter compressor in the 10- by 10-foot wind tunnel, would require about 15 stages for a pressure ratio of 5. This would result in an extremely long unit and could result in mechanical problems as well as high cost of fabrication. The aircraft-type compressor discussed could achieve a pressure ratio of 5 in seven stages.

The levels of efficiency currently attained are very high, so compressor research efforts at Lewis are directed primarily at increasing stage pressure ratio without sacrificing efficiency. The principal factors affecting stage pressure ratio are shown in figure 3-9. Stage pressure ratio is plotted against rotor tip speed with loading as a parameter. This loading parameter is the ratio of work input per stage to a reference work expressed as a function of tip speed squared. The constants J and g are dimensional constants which relate the tip speed squared to units of work. Stage pressure ratio increases as the tip speed is increased at a constant value of loading parameter. Stage pressure ratio also increases as loading parameter is increased at a fixed value of top speed. The 10- by 10-foot wind tunnel compressor previously discussed is represented by the circle. The cross-hatched area represents the general range of present-day jet engine compressors.

There are research projects at Lewis directed toward achieving higher loadings at low tip speeds as well as high tip speeds. As tip speed is increased, the flow velocities relative to the blades increase to values above the sonic level, and shock losses are encountered. For example, at a tip speed of 1400 feet per second the velocity relative to the blade may be as high as 1.4 times the speed of sound. This is a relative Mach number of 1.4. Efficient compressors with relative Mach numbers of 1.1 to 1.2 have been operational for many years. Improved blade shapes have recently been developed which permit efficient operation at relative Mach numbers of 1.4.

Figure 3-10 shows a 20-inch-diameter rotor from the high tip speed program. This type of rotor is referred to as a high speed transonic rotor. It is designed for a tip speed of 1400 feet per second and a pressure ratio of 1.6 and is represented by the triangle on the loading curve in figure 3-9. As can be seen, the blades of this rotor are very thin and have very little curvature at the tip. The band at the midspan is for the purpose of damping blade vibrations.

The performance of one of these high speed transonic rotors is shown in figure 3-11. This is a typical compressor performance map and is a plot of pressure ratio against weight flow expressed as percent of design flow. The curves through the data points represent performance along lines of constant percent of design rotational speed. The heavy line is the stall limit. Operation to the left of this limit results in severe flow instabilities and deterioration in efficiency and pressure ratio. The dashed lines are contours of constant efficiency. At design flow, this rotor produced a pressure ratio of about 1.7 with an efficiency between 90 and 91 percent.

These data are for the rotor only. If a stator were to be added to make a complete stage, the efficiency would be decreased about $1\frac{1}{2}$ or 2 percentage points, but the effect on pressure ratio would be moderate. Tip speeds of 1400 feet per second and greater are within current mechanical technology and are being used in fan stages of fan-jet engines and in the latter stages of high expansion ratio steam turbines. A multistage compressor utilizing stages of this type could achieve a pressure ratio of 5 in four stages. Current aircraft-type compressors employ about seven stages to achieve this pressure ratio. For open-cycle power generation, such stages would probably be designed for a somewhat lower speed in order to operate within the region of maximum efficiency. Under these conditions, an additional stage might be required.

To summarize this discussion, the techniques for design of the compressor and turbine components with high efficiency are in hand. Current research is directed to provide higher compressor stage pressure ratios at no sacrifice in efficiency.

Although, in general, the incentive to reduce the number of turbine stages has traditionally not been correspondingly great because of the relatively small number of stages involved, such is not the case now with high bypass ratio fan engines and would also not be the case for the power systems to be discussed. Therefore, there is other research, paralleling that of the compressor, being done to reduce the number of turbine stages while maintaining high efficiency. However, the major problem, by far, that has always confronted the turbine has been that of high inlet temperatures, and this will now be discussed.

HIGH TEMPERATURE TECHNOLOGY

The incentive to increase the turbine inlet temperature of aircraft gas turbine engines has existed for the same general reasons that were mentioned for the simple open-cycle gas turbine engine for electric power generation. Improvement in the overall performance of aircraft gas turbine engines results when higher turbine inlet temperatures are employed; as a consequence, in the development and application of gas turbine powerplants to airplanes, there has been a steady increase in turbine inlet temperature with time. An example of the increase in turbine inlet temperature for aircraft engines for the past 10 years is shown in figure 3-12. Turbine inlet temperature is plotted against time for both commercial and military engines. The band of temperatures represented by the double crosshatching indicates the range of cruise and takeoff turbine inlet temperatures for commercial engines. The double crosshatched band plus the single crosshatched band represents the range of cruise and takeoff turbine inlet temperatures for military engines. It can be seen that the military operate their engines at maximum temperatures that are about 100° F higher than that for commercial aircraft. This is done by the military to achieve higher engine performance during takeoff and certain maneuvers;

such high temperature operation, however, frequently results in reduced military engine life relative to that of a commercial engine. In this 10-year period, the turbine inlet temperature increased about 350° F for an average increase per year of about 35° F.

Materials

Part of the increase in turbine inlet temperature, but not all, has been permitted by improved turbine blade materials. An indication of the improvement in the elevated temperature capabilities of turbine blade materials in the past 10 years is shown in figure 3-13. In this figure, the 1000-hour stress to rupture strength has been plotted against metal temperature. It should be realized that for peaking and base powerplant operation of gas turbine engines, turbine blade life well in excess of 1000 hours will be required. As a consequence, actual blade design will be based on a criterion other than the 1000-hour stress-rupture properties of a material. It is common practice, however, to compare high temperature turbine blade materials on the basis of their 100- or 1000-hour stress-rupture properties. The stress-rupture properties of the materials shown in figure 3-13 are being used for comparison purposes only and are not intended for design use. All the materials represented in this figure are nickel-base alloys. Ten years ago, in 1958, one of the best turbine blade alloys was Udimet 600, represented by the lower curve. One of the best turbine blade alloys today is IN-100, represented by the upper curve. In the stress range of about 25 000 psi, which is typical for aircraft gas turbine blades, the 1958 material permitted a metal temperature of about 1600° F. For the same stress condition, the 1968 material permits a metal temperature of about 1700° F. From a metallurgical standpoint it becomes increasingly difficult to achieve ever-increasing high-temperature strength in metal alloys, particularly as the desired operating temperature approaches the melting point of the alloy. There is evidence, however, that further improvement in the high-temperature performance of nickel-base alloys can be achieved as evidenced in some initial results from an experimental alloy being developed and investigated by the Lewis Research Center. This material, known as NASA TRW 6A Alloy, is represented by a single data point. For a stress level of 15 000 psi, this experimental alloy has an allowable metal temperature of 1870° F, which is an improvement of about 80° F over that of the IN-100 material.

Figure 3-13 shows that, even for one of the best materials available today, the metal temperatures associated with turbine inlet temperatures of 2000° F or higher cannot be tolerated if reasonably long turbine blade life is desired. As a result, the engine designer must resort to cooling of the turbine component so that reduced metal temperatures commensurate with the blade stress level and the desired blade life can be achieved.

Before getting into the details of turbine cooling, the effect of temperature on material life should be considered. Figure 3-14 shows the stress to rupture strength of the nickel-base alloy IN-100 for a range of material temperatures. Curves for 1000-, 10 000-, and 100 000-hour life are shown. The 1000-hour-life curve is based on experimental data, while the 10 000- and 100 000-hour curves are extrapolated from short duration test results.

In order to examine the effect of metal temperature on material life, assume a blade stress of 25 000 psi; if the metal temperature is 1700° F, a stress-rupture-type failure could be expected after about 1000 hours. If the metal temperature could be reduced about 270° F from about 1700° to 1430° F, the stress to rupture life would be increased by a factor of 100 to a 100 000-hour life.

Based on the curves shown, it is indicated that to achieve a blade life significantly greater than 100 000 hours, perhaps 175 000 hours, which is equivalent to about 20 years of operation, the blade metal temperature should be of the order of 1400° F for blade stresses on the order of 20 000 psi. For lower levels of stress, the metal temperatures could be somewhat higher.

In general, when high turbine inlet temperatures are desired, the engine designer is faced with the problem of employing sufficient turbine cooling to achieve blade metal temperatures that are capable of long operating life. The precise blade metal temperature requirements depend on the actual life desired and the turbine blade stress levels associated with a given engine design.

Turbine Cooling

Some of our latest aircraft engines for military aircraft already employ turbine cooling. The next generation of American commercial airplanes now in the development stage, such as the airbus and supersonic transport, will also use engines incorporating turbine cooling.

Air cooling. - Turbine cooling can be accomplished with coolants such as air, water, or liquid metals. To date, all production-type aircraft engines that are cooled use air as the cooling medium. An example of how an aircraft engine turbine is cooled with air can be seen in figure 3-15. Shown is a cross section of an aircraft engine with air flowing through the engine from left to right through the compressor, the burner or combustor, the turbine, and finally the exhaust duct. In this example, air for cooling the turbine is bled from the secondary air flowing around the combustor. The stator vanes are cooled with air taken from the outer annulus, while the rotor cooling air is taken from the inner annulus. The details of the cooling air flow within the stator vanes and the rotor blades can be seen in the enlarged section. Here, stator cooling air is introduced at the base of the vanes, flows radially inward through small passages within the vanes, and is discharged at the ends of the vanes where it mixes with the combustion gas. The rotor cooling air is first ducted over the rotor disk before entering the rotor blade bases. It then flows radially outward through cooling passages within the blades and is discharged at the blade tips where it mixes with the combustion gas. In applying air cooling to an engine, care must be taken to make efficient use of the cooling air. Any air that is removed from the main combustion gas system and is not available to do work in the turbine penalizes the overall thermodynamic performance of the engine. It is therefore necessary to design cooled turbine vanes and blades that incorporate effective heat exchangers in the airfoil sections so that the cooling air requirements are kept at a reasonable minimum.

An example of an advanced air-cooled turbine blade for an aircraft engine is shown in figure 3-16. The full cross section shown in the center of the figure is discussed first. This is a cross section of a blade in which the cooling air enters a central cavity within the airfoil. The blade tip would be capped so that all the cooling air would flow out of the central cavity in a forward direction, as shown by the arrow. The air in the central cavity would be forced to flow through a series of orifices and cause high-velocity air jets to impinge upon the inside surface of the leading edge. These impingement jets result in high local coolant-side heat-transfer coefficients and provide excellent cooling of the leading edge. After cooling the leading-edge region, the cooling air enters a series of small finned passages which extend chordwise in the midchord region of the airfoil. A cross section of the finning and cooling air passages is shown in figure 3-16 (Section A-A). The cooling air flows through these passages at high velocity and cools the midchord region of the airfoil by forced convection. Near the trailing edge, the cooling-air streams from the finned passages adjacent to the suction and pressure surfaces join and flow through passages in the trailing edge region and is finally ejected from the extreme trailing edge into the gas stream.

When the combustion gas temperatures and pressures are high, the heat flux into the leading edge can cause high temperature rises in the cooling air during the impingement cooling process. In certain instances, the cooling air temperature after impingement may be so high that the air is no longer a suitable heat sink for the convection cooling process required in the midchord region of the airfoil. When this occurs, it may be necessary to remove the heated air from the blade. An example of how this might be done is shown in the lower cross section in figure 3-16, where only the forward portion of the blade has been shown. The balance of the blade is the same as described previously. In this example, a portion of the cooling air in the central chamber is permitted to enter the chordwise passages in the midchord region of the blade before passing through the impingement orifices. The balance of the cooling air enters the impingement orifices and impingement cools the leading edge as in the previous blade. However, after the impingement process, the air is permitted to flow out into the gas stream through a series of slots or holes in the blade wall. With proper direction and velocity of this coolant, it can be placed on the gas-side surface of the blade so that a thin film of air, lower in temperature than that of the combustion gas, forms an insulating layer between the combustion gases and the blade metal. This is called film cooling and results in effective local cooling of the blade immediately aft of the film cooling slot.

Currently being fabricated at the Lewis Research Center are research types of vanes and blades incorporating the cooling systems shown in figure 3-16. Figure 3-17 illustrates a partly assembled vane incorporating impingement cooling, film cooling, and chordwise fins. The vane is made from several subassemblies (figs. 3-17(a) and (b)) that are brazed and electron-beam welded into a final airfoil assembly as seen in figure 3-17(c). The subassembly in figure 3-17(a) shows the chordwise fins adjacent to the pressure surface of the airfoil as well as the leading edge portion of the vane. The impingement orifices and the film cooling holes on the surface near the leading edge are also visible. The subassembly in figure 3-17(b) shows the finning adjacent to the suction surface of the airfoil. Before welding the two major subassemblies together, a thin sheet metal wall which forms the central cavity and confines the cooling air to the finned passages (see fig. 3-16) must be brazed to the fins. In the assembled view of the airfoil in figure 3-17(c), the film cooling holes on the pressure surface and the cooling air slots in the trailing edge are clearly visible. The performance of this vane will be experimentally evaluated at Lewis in a research-type engine capable of operating at turbine inlet temperatures up to 2500° F.

Air-cooled vanes and blades for either peaking or base-load gas turbine powerplants will have to be efficiently cooled so that long turbine life and minimum cooling penalties are realized. For a landbased power system, water-cooled heat exchangers could be used to reduce the temperature of the cooling air to 100° or 150° F as compared with 800° to 1200° F for aircraft applications. Lower temperature cooling air would permit the use of more simple blade configurations. Figure 3-18 shows a possible air-cooled blade or vane for a stationary gas turbine powerplant. This design is not quite as complex to fabricate as the vane shown previously for an advanced aircraft engine; in fact, it may be possible to cast this blade in one piece, except for the tip cap that would be required. In this design, a portion of the cooling air entering the blade base flows into a radial chamber near the leading edge; this air then passes through orifices to impingement-cool the leading edge. After the impingement process, the air is ducted rearward to enter the central cavity. The remaining portion of cooling air entering the blade base flows radially outward through the finned passages in the midchord region of the airfoil. Near the blade tip, this air is also ducted into this central chamber and mixes with the air used for impingement cooling. The air in the central chamber then leaves the blade through a series of slots in the trailing edge.

The calculated blade wall temperature distribution for this design is shown in figure 3-19, where the wall temperature is plotted against the percent chord from the leading edge. The vertical dashed line represents the leading edge of the blade. The portion of the curve designated as pressure surface is for the concave side of the blade section, and the portion designated as suction surface is for the convex side of the blade. The two extremes of the curve represent the trailing edge of the blade. The wall temperatures shown were calculated for a turbine inlet temperature of 2200° F and a cooling air temperature of 130° F. The ratio of cooling air flow to gas flow was 0.013 or 1.3 percent of the main flow and is considered a very acceptable value for this use. As shown here, the metal temperatures are about 700° to 800° F below the turbine inlet temperature. The minimum metal temperature is 1375° F and the maximum is 1500° F.

Based on the stress-rupture properties of available high-temperature materials, these blade temperatures should give acceptable blade life for a base power system. It should be realized that material properties other than stress-rupture ones must also be considered when designing turbine vanes or blades. Some of these factors are creep, low cycle fatigue, long term fatigue, material oxidation and sulfidation characteristics, and the change in material properties under long time exposure to elevated temperatures. It is beyond the scope of this discussion to consider each of these factors in detail. Some of them are discussed in the paper by G. Mervin Ault.

Liquid cooling. - It was indicated earlier that liquid-cooled turbines might also be used. To date, liquid cooling has not been used in production aircraft engines because it has not yet been necessary to take advantage of the higher coolant-side heat-transfer coefficients that result from the use of liquids. Liquid cooling schemes for turbines are more complex and difficult to employ than air cooling and probably will not be used until the demands of ever-increasing turbine inlet temperatures make it absolutely necessary.

When a liquid-cooled turbine system is considered, liquid water may intuitively appear to be a suitable coolant. Actually, liquid water results in overcooling of the turbine and causes an excessive amount of heat to be removed from the blades; this in turn results in undesirably large thermodynamic penalties for liquid-water cooling systems. As a consequence, the use of liquid metal or superheated steam appears to be a more suitable approach to highly effective turbine blade cooling with a minimum of thermodynamic penalties.

An example of a liquid-cooled turbine scheme is shown in figure 3-20. This concept is being developed for a small gas turbine engine by the Continental Aviation and Engineering Company under an Army contract. The system actually employs two coolants, a primary coolant, which is permanently sealed within each individual blade, and a secondary coolant, which removes heat from the primary coolant in a heat exchanger built into the base of each blade. In this system, the primary coolant is a very small quantity of water which is placed in each blade during the fabrication process. The secondary coolant is kerosene-type fuel which is circulated through the heat exchanger built into the base of each blade.

When the engine is operating, the hot combustion gases passing over the blade airfoil section convert the liquid water to superheated steam. The steam is circulated within the blade by natural convection effects caused by the difference in density between the hot steam in the airfoil section and the relatively cool steam in the heat exchanger; the high rotative speed of the turbine rotor results in very high natural convection pumping effects. The fuel in the rotor is also pumped by natural convection effects caused by the combination of high rotative speeds and the difference in density between the fuel entering and leaving the rotor. In this concept, the heat removed from the blades is returned to the cycle when the fuel is burned in the combustor. Therefore, there are no significant thermodynamic losses chargeable to the cycle as a result of the turbine cooling process.

It is evident that high turbine inlet temperatures, in excess of 2000° F, are currently finding their way into advanced gas turbine engines, and high efficiencies are being achieved for the major rotating components. In the future, continued progress will be made in these areas with further improvements in the engine performance. In view of this, it could be expected that this type of engine could in the future have a rather dramatic impact on the utility field; that is, their role in this field could very well be extended from that of peaking service to that of a source of base-load power.

USE OF INTERCOOLING AND REHEATING

Since base-load units would have high plant cycle efficiency as their primary objective, not only would they utilize high temperatures with high efficiency components, but they would also include some additional components to achieve this goal. A system using additional components, an intercooler and a reheater in this case, is illustrated schematically in figure 3-21. Air enters the low pressure compressor and is partially compressed. The air is then cooled within the intercooler. Since heat is generated during compression and compressor work is proportional to temperature, the use of intercooling serves to reduce the total compression work required for a given pressure ratio. The cooled air is then further compressed in the high pressure compressor and heated in the combustor by the burning of fuel. Partial expansion occurs in the high pressure turbine, which drives the high pressure compressor, and heating again occurs, this time in the reheater. Since turbine work is proportional to temperature, the use of reheating serves to increase the total work obtained for a given expansion. Final expansion to atmospheric pressure then occurs in the low pressure turbine, which drives both the low pressure compressor and the generator.

The flow process is shown thermodynamically on the temperature-entropy diagram in figure 3-22. This is similar to the one shown previously for the simple cycle except that two compression steps with intermediate intercooling are indicated at the bottom, and the two expansion steps with intermediate reheating are indicated at the top.

The improvements in plant efficiency resulting from intercooling and reheating are indicated on figure 3-23, where plant cycle efficiency is plotted against turbine inlet temperature for the simple cycle, a cycle with one intercooling, and a cycle with one intercooling ane one reheating. The values shown in this figure are based on the same turbomachinery efficiencies indicated previously, 88 percent for the compressor and 90 percent for the turbine. At a temperature of 2000° F, as an example, the use of one intercooling alone increases the efficiency from 39 to 43 percent and the addition of one reheating along with the intercooling increases efficiency up to 45 percent. For a plant of fixed power output, this corresponds to about a 10-percent reduction in fuel consumption with intercooling and about 5-percent additional reduction in fuel consumption with the reheating added. At other temperatures, the benefits are about the same magnitude with efficiency reaching about 50 percent at 2500° F. The reductions in fuel cost associated with such improvements in efficiency should more than pay for the additional capital cost of the plant. Additional intercoolings and reheatings can be used to yield further improvements in cycle efficiency. The incremental improvements, however, become smaller and smaller as additional intercool and reheat steps are added. The optimum number of intercoolers and reheaters to be used will depend on a detailed economic analysis for a given plant.

As mentioned previously, the airflow through the system is a prime determinant of the size required for the components. The use of one intercooling and one reheating results in about 40 percent less flow than for the simple cycle. Although the number of components is increased by using intercooling and reheating, the sizes of the components are reduced as compared with the simple cycle.

In order to realize the benefits associated with the use of intercooling and reheating, the system must operate at increased pressure ratios as compared with the simple cycle. This is indicated in figure 3-24, where plant cycle efficiencies for the simple system and for a system with one intercooling and one reheating are plotted against pressure ratio at a turbine inlet temperature of 2000° F as an example. For the simple cycle, the optimum pressure ratio is 30, and efficiency decreases markedly if the pressure ratio is less than 20 or greater than 40. For the cycle with intercooling and reheating, the optimum pressure ratio is about 120. However, the curve is very flat in the region of maximum efficiency, and it is possible to operate at a pressure ratio of about 75 without a significant decrease in efficiency. This pressure ratio of 75 is still considerably higher than that for the simple cycle. Thus, with intercooling and reheating, nor only will additional heat exchangers be required, but also additional stages will be required for the turbomachinery.

EXAMPLE BASE-LOAD SYSTEM

In order to illustrate what the features of a base-load gas-turbine powerplant might be like, one example plant will be described. Some of the principal requirements selected for this plant are as follows:

Power level, mW_e	1000
Generator speed, rpm	1800
Peak temperature, °F	2200
Compressor pressure ratio	75
Intercooler	One
Reheater	One
Fuel	Natural gas
Plant cycle efficiency, percent	~42

Power level was selected at 1000 megawatts, with a single generator operating at 1800 rpm. The reason for this speed being selected will be indicated later. The turbine inlet temperature was selected at 2200° F, with an overall compressor pressure ratio of 75. The plant utilized one intercooler as well as one reheat back up to 2200° F. Natural gas was selected as the fuel. A plant efficiency on the order of 42 percent was estimated. This efficiency is somewhat lower than indicated on figure 3-23. The reduction was made (1) to reflect consideration of work necessary to pump the natural gas to the high required pressures, and (2) to provide some allowance for the thermodynamic penalties involved in cooling the high-temperature section of the engine.

Figure 3-25 shows a schematic of the major turbomachinery and heat exchanger components, with an indication of their size. This plant which uses two rotating spools, will be described by tracing through the airflow path. Approximately 4000 pounds of air per second are pulled into the low pressure compressor which is about 14 feet in diameter and develops a pressure ratio of 5. The flow then goes into the intercooler, which is of the crossflow finned tube type with the cooling water in the tubes. This heat exchanger is only moderate in size being about 26 feet in diameter and 4 feet long exclusive of the inlet and exit headers. The air then goes into the high pressure compressor, which is approximately 7 feet in diameter and develops a pressure ratio of 15. From the compressor, the air goes through the primary combustor and from there into the high pressure turbine. This turbine is split into two segments, with the single reheat located in between. After leaving the high pressure turbine, the flow is ducted into the low pressure turbine, which powers both the low compressor and the generator. This turbine is a multistage double-exit unit with a maximum diameter of about 14 feet. The use of a double-exit turbine and a rotative speed of 1800 rpm were required as a result of the very high volume flow exiting from the turbine. This very high volume flow is due to the high power level selected and the atmospheric exhaust pressure, which is obviously a requirement for an open-cycle plant.

This gives an idea what a high power, high performance base-load powerplant might look like. The system described is intended only to serve as an example. There are many configurations that might be evolved, including perhaps an additional spool with more intercooling and reheating. Also, the use of recuperation (transferring heat from the hot turbine exhaust to the cooler compressor exhaust) has not been examined for the open cycle. Recuperation was examined for the closed-cycle system to be subsequently discussed, and its effects are illustrated in that discussion. Such additions with possibly higher turbine temperature levels offer the possibility of achieving plant cycle efficiencies approaching 50 percent. Evolving such a plant would, however, have to be tempered by the additional complexity and associated plant cost.

FUELS

Before leaving the discussion of open-cycle gas turbine systems, a few comments relative to fuels are in order. Generally, three types of fuel can be considered for gas turbine peaking and base-load systems; these are natural gas, petroleum, and coal.

Natural Gas

Natural gas is a convenient, clean, easy-to-handle fuel for gas turbine powerplants and has been used successfully in peaking gas turbine applications for a number of years. In certain regions of the United States, which are close to natural gas sources, this fuel can be employed economically in either peaking or base-load powerplants. Use of gaseous fuels in a base powerplant having a pressure ratio of 75, such as the one just discussed, would, of course, necessitate pressurizing the gas to a high level so that it could be introduced into the combustor. Suitable methods for pressurizing the gas would have to be evolved. One approach would be to use compressors with intercooling. It is possible that liquefying the gas and then pumping it to high pressure in a cryogenic pump might be the best approach. In any event, methods for pressurizing natural gas to levels of the order of 1200 psi would have to be evolved, and the economics of the pumping systems would have to be considered in evaluating overall powerplant performance.

Combustors for natural gas should present no severe problems. In recent years the possibility of using liquefied natural gas in some aircraft applications has stimulated the research and development of natural gas combustors for advanced aircraft engines. Figure 3-26 shows a combustor that is currently under investigation at the Lewis Research Center. This is called a "swirl" can combustor, and it burns natural gas very efficiently with low pressure loss. The combustor shown is research-type hardware and incorporates a mounting flange which is excessively heavy for flight application. The horizontal tubes are fuel manifolds, and the daisy-shaped pieces are the swirl cans. Additional fuel manifolds are located at the top and bottom of the combustor for local fuel distribution control, and are strictly for research purposes. An enlarged view of one of the "swirl" cans is shown in figure 3-27. Natural gas enters the swirl can from two small orifices located in the manifold passing through the horizontal centerline of the can. The fuel is injected near each side of the swirl can at right angles to the main airflow and in a manner such that the flow is tangential to the inner wall of the swirl can. The fuel thus "swirls" and mixes with the combustion air. Efficient burning is obtained as the mixture moves downstream from the fuel manifold. For an actual engine application, the combustor would be curved into an annular shape rather than the rectangular cross section shown. This combustor is $2\frac{1}{2}$ feet wide and 1 foot high and represents the equivalent of a 90° sector of a combustor for an engine of the size required for the supersonic transport.

Petroleum

The second general category of fuel mentioned previously was petroleum. Generally, petroleum fuels are relatively expensive per unit of heat supplied and are therefore too costly for use in base power systems. For peaking systems, however, the petroleum-type fuels may be practical from the overall economic standpoint.

Coal

Coal has been considered for many years as a fuel for gas turbine engines that might be used in railroad locomotives or for stationary powerplants. Its major drawback, of course, is the problem of ash erosion of the turbine component. In the United States, the Locomotive Development Committee of Bituminous Coal Research, Inc. and the Bureau of Mines have investigated the use of coal in gas turbines quite extensively. Figure 3-28 shows the results of severe ash erosion on turbine blades that was observed in an investigation conducted by Bituminous Coal Research, Inc. In this investigation, the erosion was particularly severe along the blade trailing edge between the blade base and about the midspan region. The erosion is evidenced by the severe curve in the blade trailing edge; the trailing edge was originally a straight radial line. The damage shown in figure 3-28 resulted from about 750 hours of operation in a gas stream that had about 85 percent of the ash removed through the use of centrifugal separators. The Bureau of Mines has shown that, by careful aerodynamic redesign of the turbine and the use of titanium carbide protective devices near the root of the blade leading edge, the erosion problem on blades can be greatly reduced. From tests made on an improved turbine that has been operated by the Bureau of Mines for about 200 hours, it has been estimated that the turbine blade life in the modified turbine will be of the order of 30 000 hours before erosion becomes a degrading factor in turbine performance. Stator blade erosion for this turbine, however, remains a problem with stator life currently estimated at about 5000 hours. At this time it appears that considerable attention must still be given to the ash erosion problem before suitable life can be obtained in base-load coal-fired gas turbines.

An interesting powerplant concept in which coal is the primary fuel and in which both gas and steam powered turbines are used is shown schematically in figure 3-29. On the left side of the figure, coal is shown being heated. The volatiles are driven from the original coal and collected as a gas, which is used to drive a gas turbine shown in the right side of the figure. The nonvolatile portion of the coal is removed as a slurry of solids and liquid, which is called "char." The char is used to reheat the gas turbine exhaust, which in turn fires a steam powerplant, as shown in the center portion of the figure. With such a system the gas may produce about 20 to 40 percent of the electrical energy while the char would produce about 60 to 80 percent of the electrical energy. Although the system is feasible from a technical standpoint, the overall economics of such a system are unknown at this time and require considerable study.

CLOSED CYCLES

Now the discussion will turn to the other type of gas turbine power system, that is, the closed cycle. This cycle is similar to the steam cycle, except that the working fluid remains a gas throughout the entire loop. Closed-loop gas systems have enjoyed considerable attention within NASA as a strong candidate as a source of power in such applications as space stations or lunar outposts. Also, as indicated earlier, such plants have been in operation in Europe with coal as the heat source and offer excellent prospects as part of a nuclear powerplant.

INERT GAS CYCLE

A closed-loop gas cycle power system is shown schematically in figure 3-30. The corresponding thermodynamic diagram is shown in figure 3-31. The closed cycle, in general, is similar to the open cycle except that the turbine exhaust gas is cooled and returned to the compressor inlet. The closed-cycle system illustrated has an intercooler but no reheater. A reheater was omitted on the basis that if the heater were a gas-cooled reactor, it might not be desirable to go through the reactor shield with two more gas ducts. If the gas heater shown here were only a heat exchanger forming part of a reactor coolant loop, then reheating could reasonably be included. The closed-cycle system includes a recuperator, which, as indicated by the arrow on the thermodynamic diagram (fig. 3-31), serves to transfer heat from the hot turbine exhaust gas to the cooler compressor exhaust gas.

The gas entering the low pressure compressor is partially compressed, intercooled, and then compressed further in the high pressure compressor. The compressor exhaust gas passes through the recuperator, where it is heated by the hot turbine exhaust gas. Final heating to turbine inlet temperature takes place in the gas heater. The hot gas is then expanded through the turbine, cooled in the recuperator, and further cooled back to compressor inlet temperature in the waste heat exchanger.

Performance

Plant performance for this system is presented in figure 3-32, where plant cycle efficiency is plotted against pressure ratio for turbine inlet temperatures of 1000°, 1500°, and 2000° F. The working fluid is assumed to be one of the inert gases such as helium or argon, and the assumed turbomachinery efficiencies are again 88 percent for the compressor and 90 percent for the turbine. The recuperator effectiveness assumed for this figure is 0.9. These curves are similar in trend to those discussed previously for the open cycle, with efficiency increasing with increasing temperature and maximizing at some optimum pressure ratio at each temperature. The maximum efficiencies are 25 percent for 1000° F, 39 percent for 1500° F, and 48 percent for 2000° F. One major point to note is the considerably lower values of pressure ratio for maximum efficiency for this closed-cycle system as compared with the previously shown open-cycle system; these pressure ratios are in the range of 2 to 4 for this system. This reduction in pressure ratio is principally due to the effect of recuperation, as will be shown subsequently. There is also an effect of the different working fluid. For given changes in temperature, pressure ratios are lower for monatomic gases such as helium than for diatomic gases such as air.

Pressure Level

The fact that the loop is closed allows additional design freedoms as compared with the open-loop system. There is the freedom to set system pressure level as desired. High pressure is desirable from the standpoint of reducing the sizes of the heat exchange equipment, but selection of a pressure level must be tempered by structural considerations as well as the achievement of satisfactory turbomachinery configurations.

Working Fluid

There is also the freedom to select any working fluid desired. The choice of a working fluid for a closed-cycle system has a major effect on the various components. For this discussion the working fluid is considered to be one of the inert gases, or a mixture of these gases, so as to permit a continuous variation of molecular weight. Figure 3-33 illustrates the effect of fluid molecular weight on the number of stages required for the turbine and the compressor. The left end of the curve represents helium and the right end represents argon, both of which are available in sufficient quantities for use in large powerplants. The in between values of the molecular weight can be achieved with mixtures of these two gases. There are a relatively large number of stages required for use with helium, the example number shown being 22 for the compressor and 8 for the turbine. These numbers can be reduced significantly by increasing molecular weight, since the number of stages is inversely proportional to molecular weight.

The choice of fluid molecular weight also has a significant effect on the size of the heat-transfer components. This effect is illustrated in figure 3-34. The relative size of the recuperator, which is the largest heat exchanger in the system, is plotted against molecular weight. As molecular weight increases from that of helium to that of argon, the size of the recuperator increases by a factor of about $3\frac{1}{2}$. Since a recuperator for helium, based on an effectiveness of 0.9, would have a volume of about 15 000 cubic feet for a 1000-megawatt powerplant, any increases over this would be most significant in terms of size and cost. Figures 3-33 and 3-34 illustrate two major factors that must be considered when making a selection of working fluid.

Use of Recuperation

As previously indicated, a recuperator was included in the closed-cycle system. The use of recuperation serves to improve cycle efficiency by using some of the heat available in the turbine exhaust gas to heat the compressor discharge gas. In this way, the amount of heat that must be supplied by the heat source is reduced. The improvement in plant performance due to recuperation is shown in figure 3-35, where plant cycle efficiency is plotted against recuperator effectiveness. In discussing plant cycle efficiency, reference should be made to the curve with the arrow pointing to the scale on the left. Significant increases in plant efficiency are seen to result from increasing effectiveness, with these increases becoming more rapid as effectiveness approaches unity. At an effectiveness of 0.5, the curve indicates an efficiency of about 31 percent. At 0.8, the efficiency increases to about 36 percent, while at 1.0 there is an additional nine-point increase to about 45 percent.

Also plotted on figure 3-35 is optimum pressure ratio as a function of effectiveness. This is the curve with the arrow pointing to the scale on the right. Increasing effectiveness also serves to decrease pressure ratio and thus reduce the number of turbomachinery stages. For this system, pressure ratio decreases from 5 to 2 as effectiveness increases from 0.5 to 1. Therefore, from the cycle efficiency and pressure ratio standpoint, the effectiveness should be near 1.0.

Unfortunately, the attainable recuperator effectiveness is limited by the size of the recuperator. Figure 3-36 shows how this size changes with effectiveness. This is a plot of relative recuperator size against effectiveness. The size increases rapidly with increasing effectiveness and approaches infinity as effectiveness approaches 1. As effectiveness increases from 0.8 to 0.9, the size more than doubles. Going from 0.9 to 0.95 again more than doubles the size. As mentioned before, a recuperator designed for an effectiveness of 0.9 for a 1000-megawatt system would have a volume of about 15 000 cubic feet. Because of the rapid increase of recuperator size with effectiveness, recuperator effectiveness must be limited, probably to some value between 0.9 and 0.95.

Pressure Loss

Another important variable associated with both cycle efficiency and heat exchanger size is the pressure drop across the heat exchanger. This effect is illustrated with respect to the recuperator in figure 3-37. This plot of plant cycle efficiency against recuperator pressure loss shows a decrease in efficiency from 40 to 38 percent as recuperator pressure loss increases from 1 to 5 percent. This corresponds to a one-half-point decrease in cycle efficiency for each 1 percent increase in pressure loss. Figure 3-38 shows how the relative size of the recuperator varies with pressure loss. Since the heat transfer increases with increased pressure drop, increased pressure loss will result in a smaller heat exchanger. For the pressure loss range shown in figure 3-38, an increase of 1 percent in pressure loss can result in a 10 to 20 percent reduction in recuperator size. In view of the strong influence of recuperator effectiveness on the system both with regard to performance and hardware, it is obvious that extreme care must be exercised in the design of this component so that high levels of system performance can be obtained with a practical recuperator size.

Example Base-Load System

Now that some of the principal features of the closed-gas cycle have been discussed, it would be appropriate to describe one such example system in much the same way as was done for the open cycle. The following are the principal design requirements and features of one such unit:

Power level, mW_e	1000
Generator speed, rpm	3600
Fluid	Helium
Turbine inlet temperature, ^{o}F	1350
Compressor pressure ratio	2.5
Peak system pressure, psi	2500
Intercooler	One
Recuperator effectiveness	0.925
Plant cycle efficiency, percent	~37

Consider it to be mated to a reactor, possibly a gas-cooled reactor. A 1000-megawatt system and a rotational speed of 3600 rpm are assumed. Helium was assumed to be the working fluid on the basis of its favorable heat transfer properties, as discussed previously.

A turbine inlet temperature of 1350^{o} F is selected as a compromise between (1) a reasonable plant cycle efficiency, (2) reactor restrictions such as were described in the paper on nuclear reactors, and (3) the desire to use conventional superalloys in the turbine without cooling. A compressor pressure ratio of 2.5 is selected with a peak pressure level of 2500 psi. One stage of intercooling was assumed with no reheating, and the recuperator effectiveness was assumed to be 0.925. With these conditions, a plant cycle efficiency of approximately 37 percent was estimated.

Figure 3-39 shows a possible arrangement for this system. A single-shaft arrangement is indicated, with the generator located at one end and the reactor heat source at the other end. The turbine inlet is located at the reactor end in order to minimize high temperature ducting. The heat exchangers are split into two parts and are symmetrically located on each side of the turbomachinery shaft.

Helium at 4200 pounds per second and 1000 psi is drawn into the low pressure compressor, which is approximately $5\frac{1}{2}$ feet in diameter and develops a pressure ratio of 1.6 with 11 stages. From this compressor, the flow goes into the intercooler and then back into the high pressure compressor, where 11 stages are used to bring the pressure up to the peak value of 2500 psi. From there, the high pressure gas passes through the recuperator, where it is preheated by heat transfer from the turbine exhaust. The recuperator is the largest of the heat exchangers, and consists of two sets of four units in series, each unit being $9\frac{1}{2}$ feet in diameter and 63 feet long. The units would be of the multipass crossflow tube and shell type with the high density compressor exit gas going through the tubes.

From the recuperator, the gas goes into the reactor area where its temperature is raised to the peak value of 1350^{o} F. It then passes through the eight-stage

$7\frac{1}{2}$-foot-diameter turbine, where a total of about 2500 megawatts of power is extracted, with approximately 40 percent of this power representing useful energy to power the generator. After the flow leaves the turbine, it passes through the recuperator and then through the heat rejection cooler where it returns to the compressor inlet. As in the case of the open cycle, this configuration serves only to illustrate how a high power closed-loop helium plant might be configured. Final determination of powerplant requirements and configuration would require considerable in-depth study involving both technical and economic aspects.

As indicated before, the selection of 1350° F turbine inlet temperature represents a compromise between desired cycle performance and restrictions from a reactor and materials standpoint. Boosting the turbine inlet temperature to 1500° F would result in a three-point improvement in plant cycle efficiency but at considerable expense of mechanical integrity. Since this plant assumes the use of a reactor heat source, possibly of the breeder type, the incentive to go to the high turbine inlet temperatures normally considered desirable for gas turbines is not nearly so great as it would be with a fossil-fueled plant.

CARBON DIOXIDE CYCLE

An example of a closed-cycle gas system using helium as the working fluid has just been discussed. Helium was selected as the working fluid because of its good heat-transfer properties and because it is chemically inert. As discussed previously, however, its low molecular weight results in a large number of turbomachinery stages. Another fluid of higher molecular weight that has been receiving considerable attention recently for application to closed-cycle gas systems is supercritical carbon dioxide. This fluid not only has a higher molecular weight, but also offers an advantage of reduced compressor work. This results because the compression process occurs in a region of the temperature-entropy diagram where the supercritical fluid properties are more like a liquid than a gas.

In figure 3-40 the temperature-entropy diagram for a simple form of this system is presented showing the location of the cycle relative to the saturation curve of the carbon dioxide working fluid. This cycle involves the same processes and equipment as the closed-cycle helium system. However, the compression process takes place in a region near the critical point of the carbon dioxide working fluid, which occurs at a temperature of about 88° F and a pressure of about 1070 psi. As in the closed-cycle helium plant, there are adiabatic compression and expansion processes and a recuperation process to transfer heat from the turbine exhaust to the compressor exhaust. Because of the 1070-psi critical pressure, the supercritical carbon dioxide cycle is a high-pressure cycle operating with a compressor inlet pressure between 1500 and 2000 psi and a compressor discharge pressure of the order of 3000 to 3500 psi. By operating near the saturation curve, the compressor work, as shown by the compression line, becomes a much smaller fraction of the turbine work, represented by the expansion line, than in the closed-cycle helium system. As a result, the efficiency of the supercritical carbon dioxide plant is somewhat higher than the 37 percent plant efficiency mentioned for the example closed-cycle helium plant.

The primary advantage of the supercritical carbon dioxide cycle lies in the compactness of the turbomachinery. This is a result of the use of a high molecular weight gas as a working fluid at high pressure. The turbomachinery for the example 1000-megawatt closed-cycle helium plant utilized two 11-stage axial flow compressors and an 8-stage axial flow turbine. For a similar size plant using supercritical carbon dioxide, this multistage axial flow machinery could be replaced by a single-stage centrifugal compressor, or pump, about $2\frac{1}{2}$ feet in diameter and a single-stage radial inflow turbine about $5\frac{1}{2}$ feet in diameter.

The compactness of the turbomachinery for the supercritical carbon dioxide system, however, is accompanied by an increase in the required heat exchanger size. The supercritical carbon dioxide cycle is a highly recuperated cycle, and this coupled with the poorer heat transfer properties of carbon dioxide as compared with helium, result in a recuperator that may be of the order of two to three times the size of that required for the closed-cycle helium plant.

While on the subject of the carbon dioxide cycle, it should be mentioned that there are variations of this cycle that employ condensation. A temperature-entropy diagram for the condensing carbon dioxide cycle is shown in figure 3-41. In these cycles, the turbine expansion is continued to a pressure lower than the critical pressure, and the exhaust gases are cooled to a temperature lower than the critical temperature, so that condensation takes place. Some of these cycles result in higher efficiencies than the supercritical cycle; however, a heat sink is required with a temperature lower than the carbon dioxide critical temperature of 88° F.

Although the carbon dioxide cycle is of interest because of its good cycle efficiency and compact turbomachinery, there are some considerations that arise because of the properties of the working fluid, and these will require investigation. The chemically active nature of carbon dioxide, as compared with a gas such as helium, may present material compatibility problems, particularly at temperatures above 1300° to 1400° F. System dynamic studies will also be required because the rapidly changing properties of the carbon dioxide near the critical point raise a question concerning system stability.

CONCLUDING REMARKS

Some of the major thoughts brought out during the course of this discussion are as follows.

First, with regard to the open-cycle gas turbine, the technical advancements made in such areas as high component efficiencies and high turbine inlet temperatures have in the last three decades brought this type engine to a very high level of performance, and it would be expected that even further gains will be made in the future. As a result, not only will its use as a peaking device continue to expand in the utility industry, but also it could foreseeably become a major contender for base load. Fuel-type restrictions do currently represent an important factor such that potential use of this power source for base load will probably be limited to areas where natural gas is economically suitable.

Secondly, the gas turbine in the closed-loop form does represent an attractive candidate for base load in conjunction with a nuclear reactor. Helium is generally accepted as the desired working gas although others are being considered. It would also appear that moderate turbine inlet temperatures can be utilized with the small sacrifices in plant efficiency offset by the considerable reduction in the reactor and materials problem at the hot end.

In conclusion, although these systems appear technically attractive, the ultimate selection of a system for utility use must be based upon economic considerations. In-depth studies of these systems, therefore, must be made from both technical and economic standpoints in order that their competitive position be more firmly established.

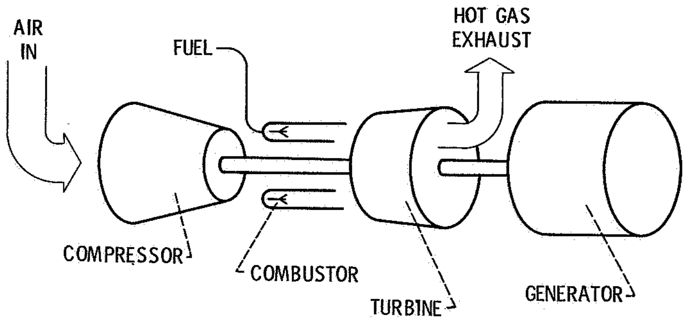


Figure 3-1. - Schematic of simple open-cycle gas turbine system.

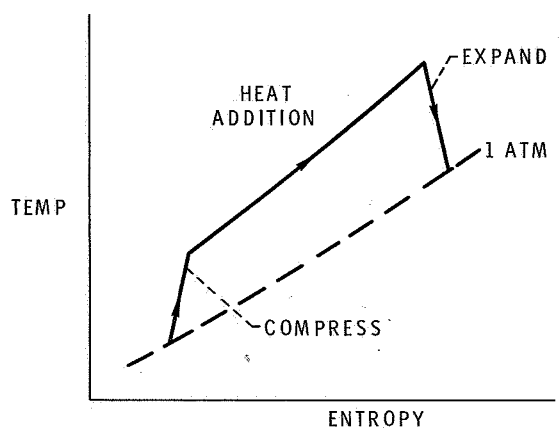


Figure 3-2. - Thermodynamic diagram for simple open-cycle system.

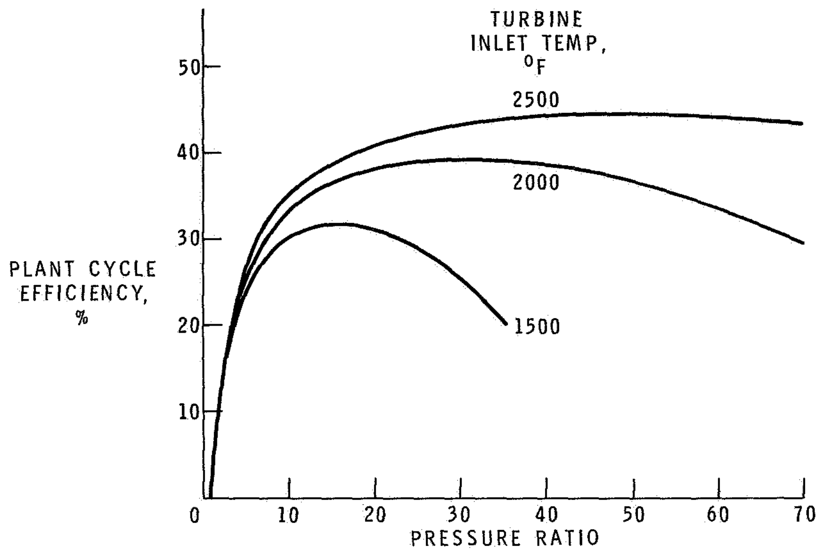


Figure 3-3. - Simple open-cycle plant performance. Compressor polytropic efficiency, 88 percent; turbine polytropic efficiency, 90 percent.

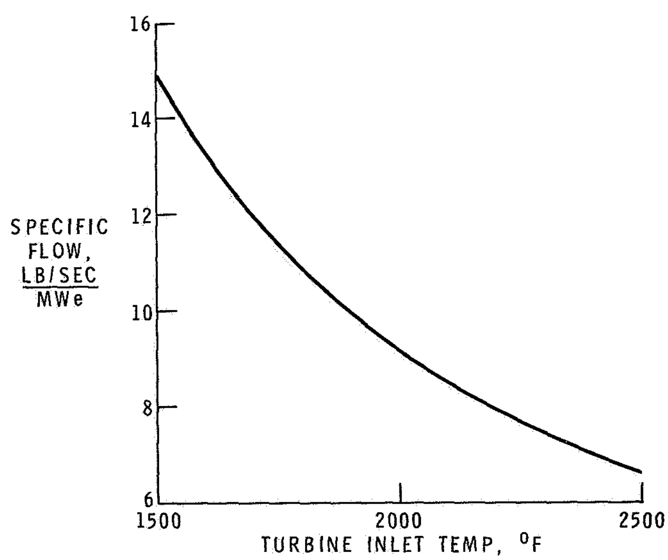


Figure 3-4. - Simple open-cycle airflow at optimum pressure ratio.

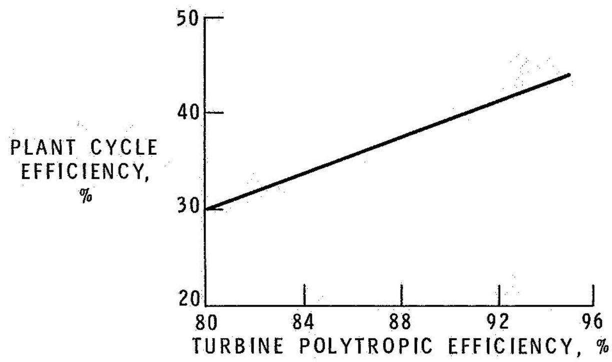


Figure 3-5. - Effect of turbine efficiency on plant cycle efficiency. Turbine inlet temperature, 2000° F; compressor polytropic efficiency, 88 percent.

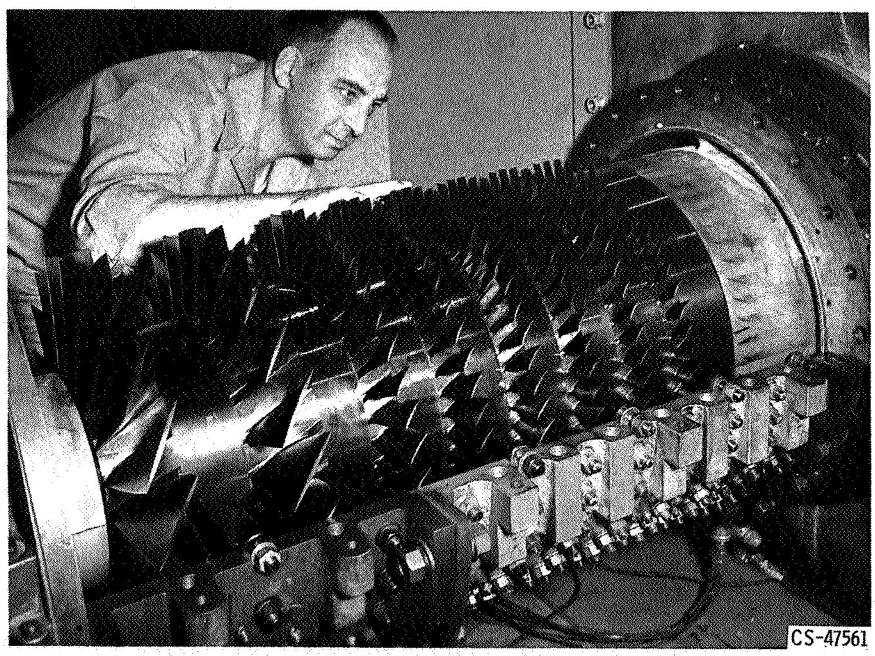


Figure 3-6. - Research compressor. Peak polytropic efficiency, 90.5 percent.

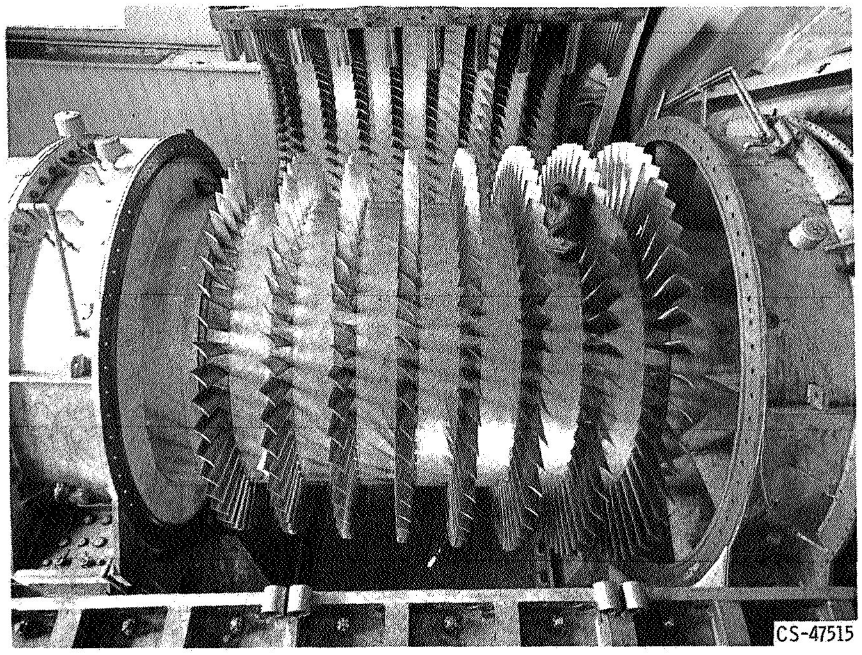


Figure 3-7. - 20-Foot-diameter compressor. Peak efficiency, 91 percent.

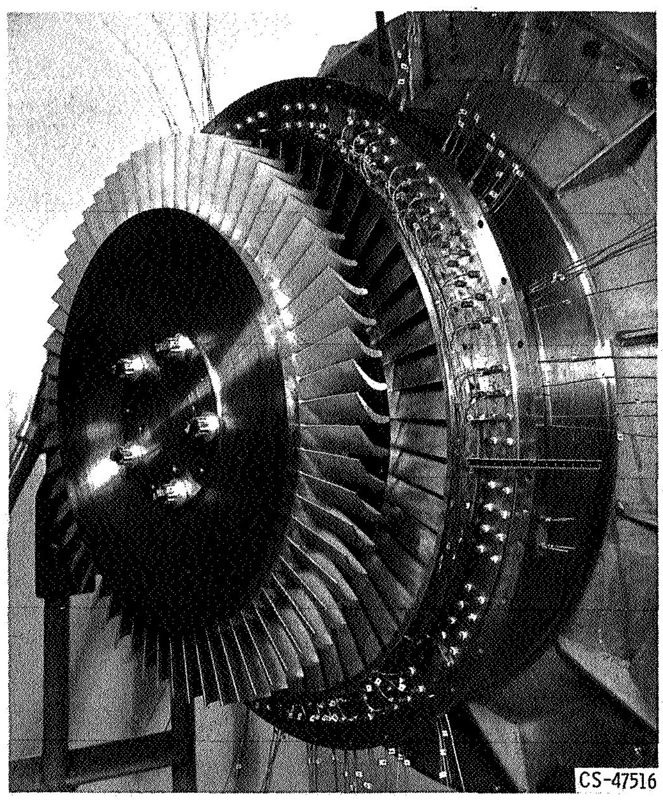


Figure 3-8. - $2\frac{1}{2}$-Foot-diameter research turbine. Efficiency, 92 percent.

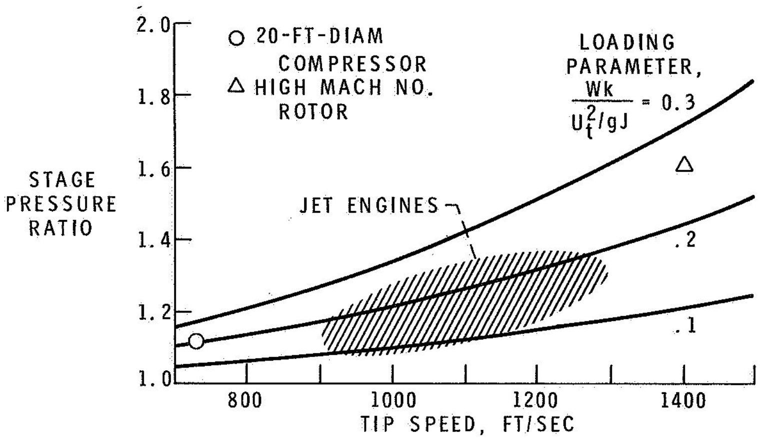


Figure 3-9. - Compressor loading.

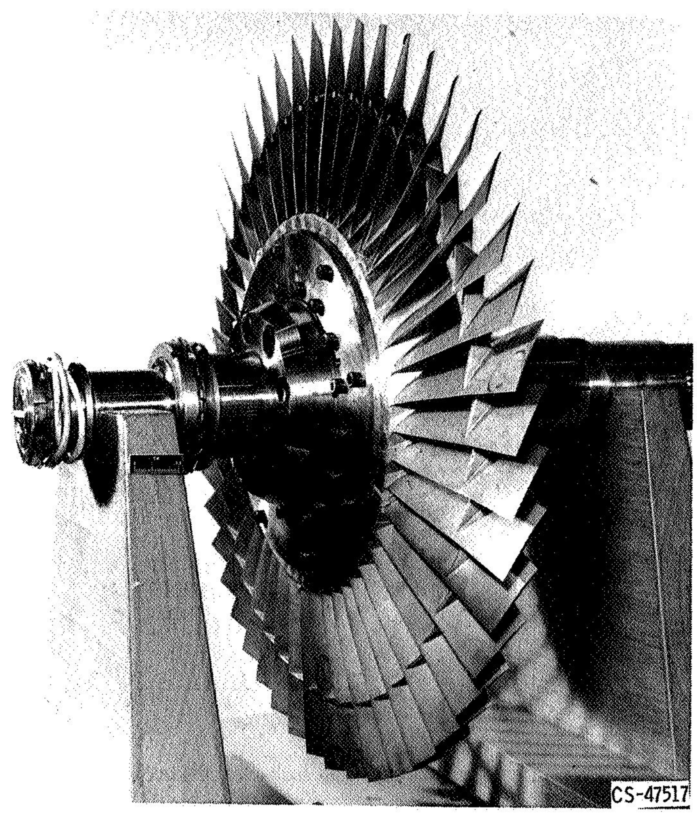


Figure 3-10. - High Mach number compressor rotor.

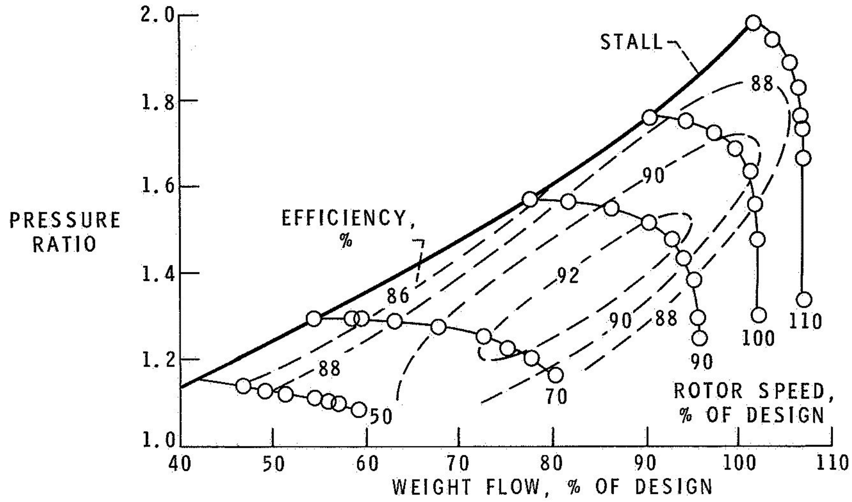


Figure 3-11. - High Mach number rotor performance.

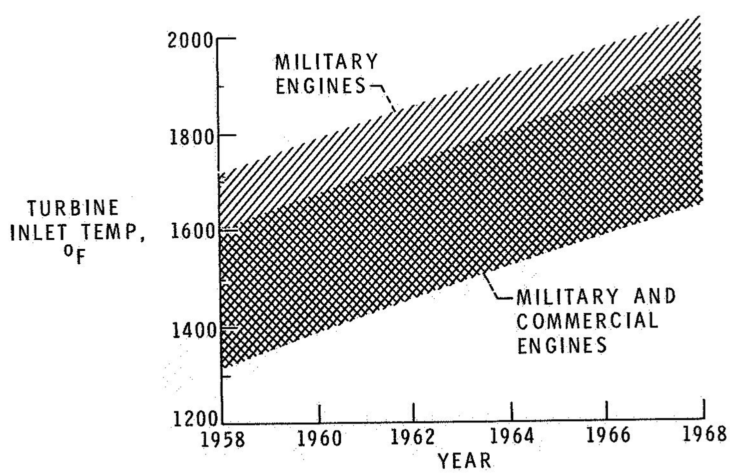


Figure 3-12. - Chronology of aircraft engine temperatures.

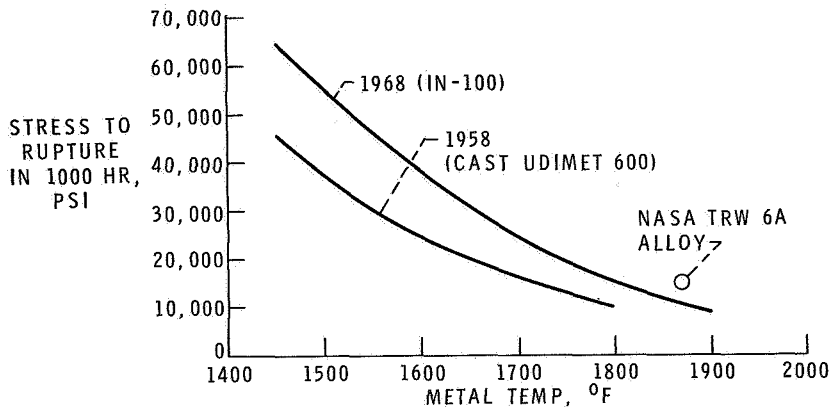


Figure 3-13. - Chronology of material strength.

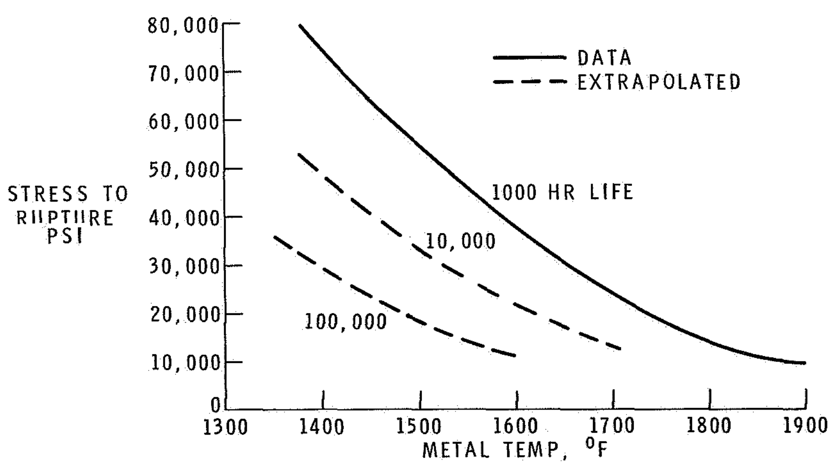


Figure 3-14. - Effect of temperature on life of IN-100.

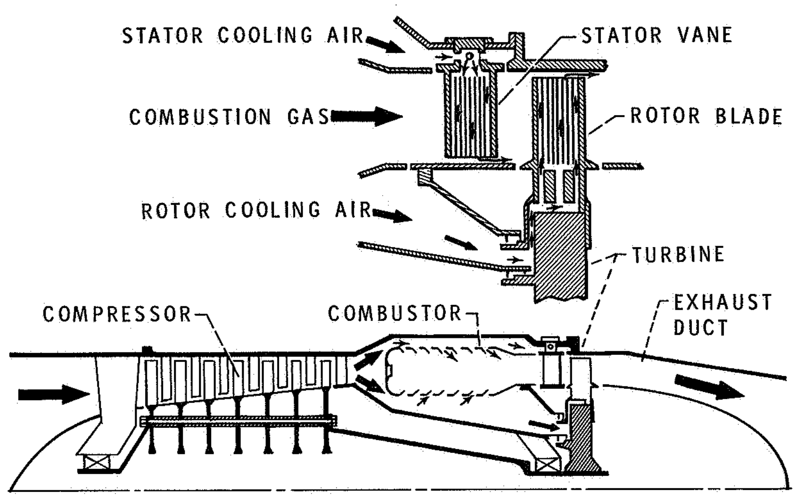


Figure 3-15. - Air-cooled aircraft engine.

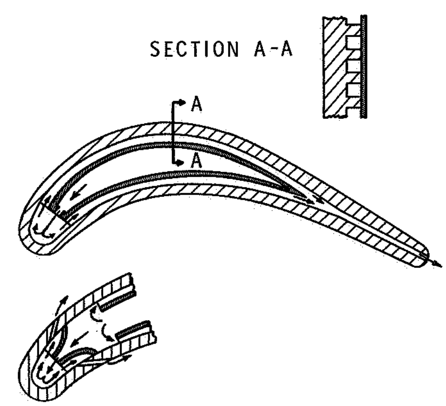
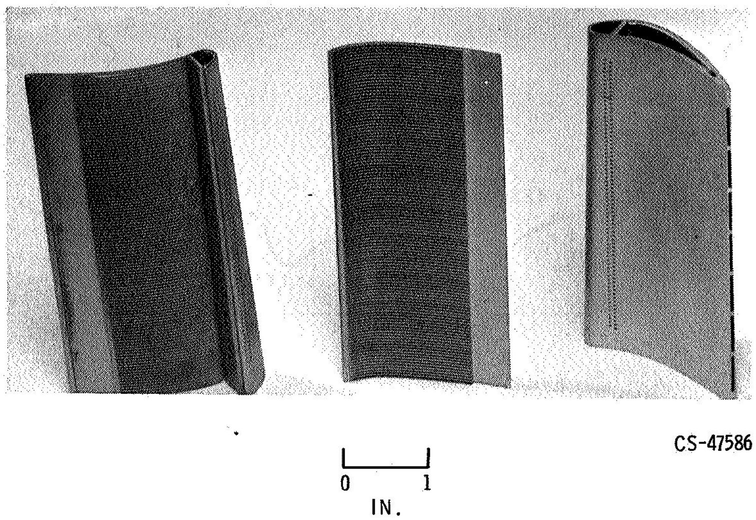


Figure 3-16. - Advanced air-cooled blade.



(a) Leading edge and pressure surface. (b) Suction surface. (c) Assembled airfoil.

Figure 3-17. - Advanced air-cooled vane components.

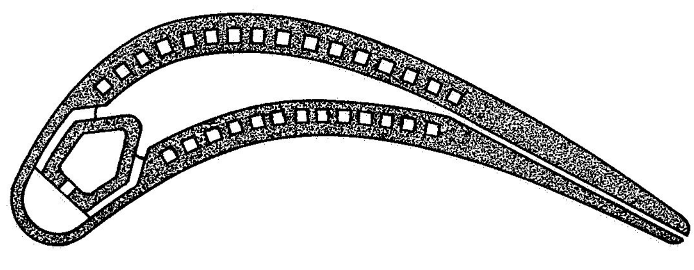


Figure 3-18. - Air-cooled blade for base-power turbine.

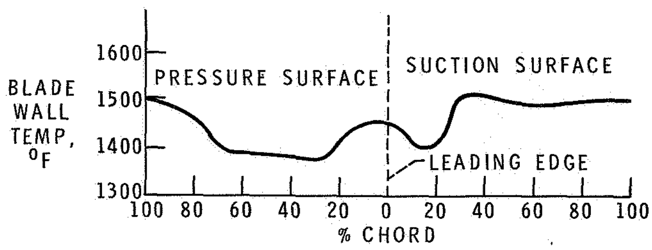


Figure 3-19. - Calculated blade metal temperature distribution. Turbine inlet temperature, 2200° F; cooling air temperature, 130° F; coolant flow/combustion gas flow, 0.013.

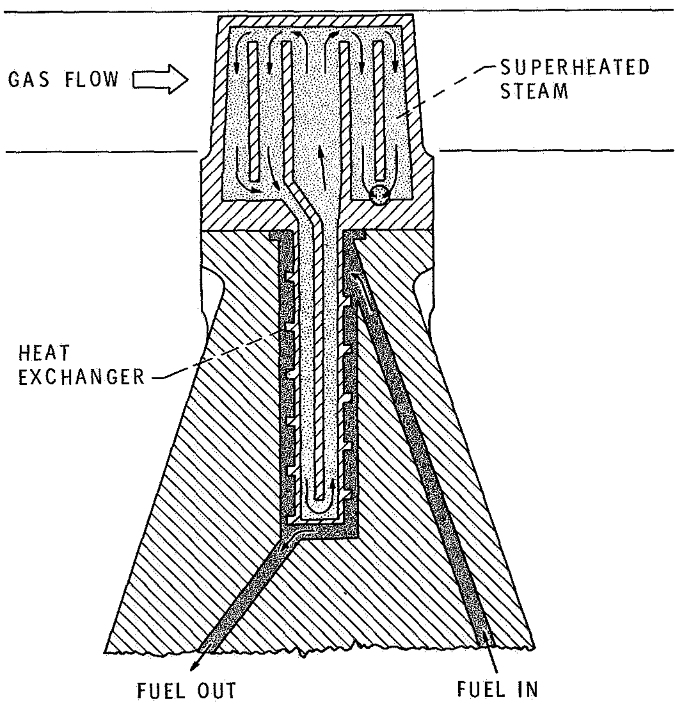


Figure 3-20. - Liquid-cooled turbine.

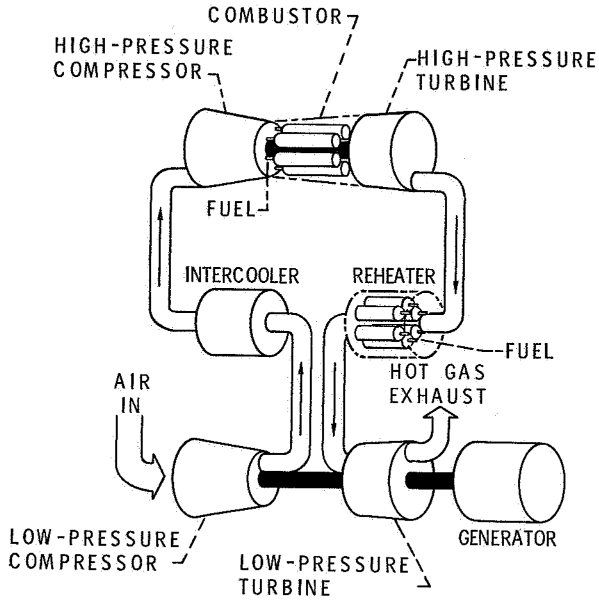


Figure 3-21. - Open-cycle system with intercooling and reheating.

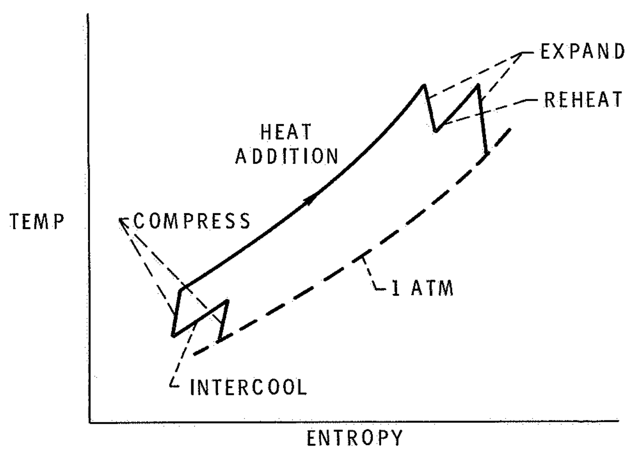


Figure 3-22. - Thermodynamic diagram for open-cycle system with intercooling and reheating.

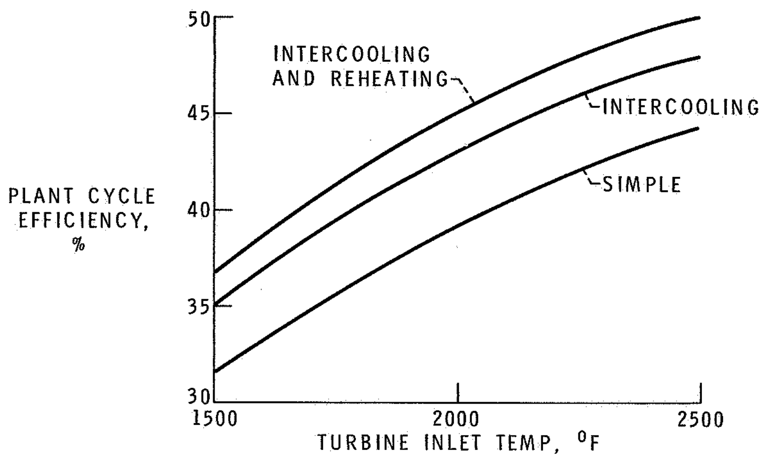


Figure 3-23. - Effect of intercooling and reheating on plant performance at optimum pressure ratio. Compressor polytropic efficiency, 88 percent; turbine polytropic efficiency, 90 percent.

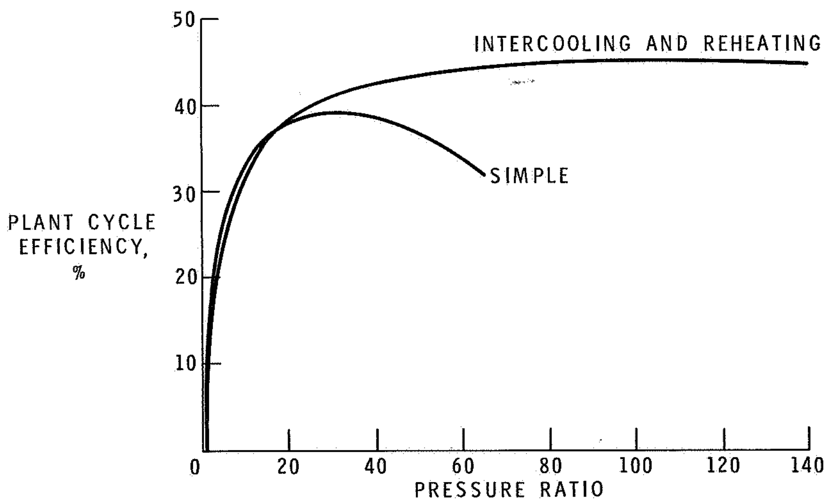


Figure 3-24. - Effect of intercooling and reheating on pressure ratio selection. Turbine inlet temperature, 2000^0 F.

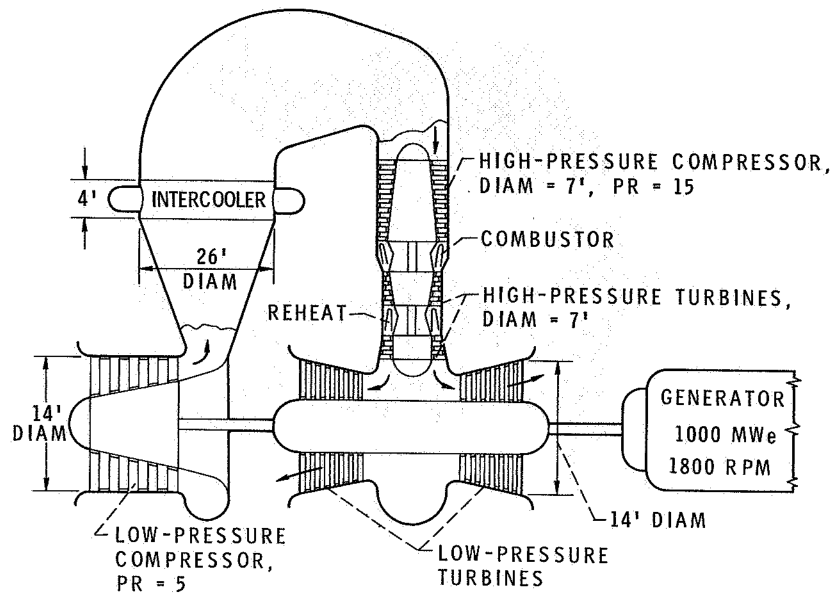


Figure 3-25. - Schematic of base-load open-cycle powerplant.

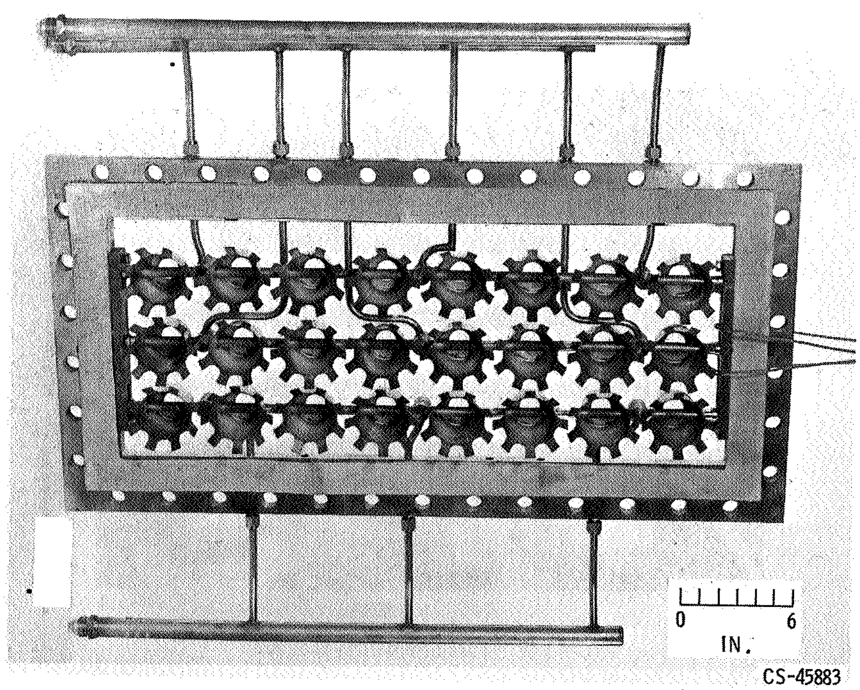


Figure 3-26. - Natural gas combustor.

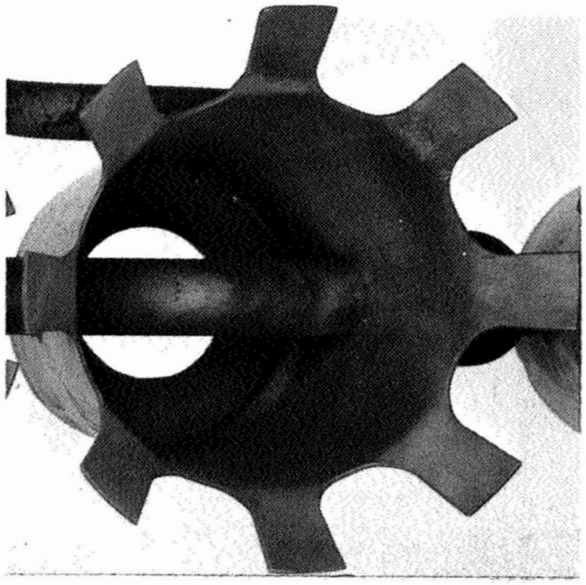


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Figure 3-27. - Combustor swirl can.

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Figure 3-28. - Turbine eroded by coal ash.

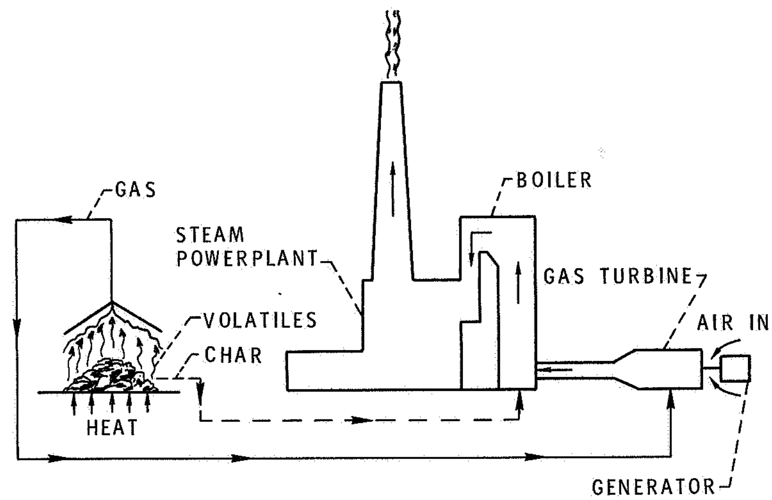

Figure 3-29. - Use of coal in combined cycle. Gas, 20 to 40 percent of energy; char, 60 to 80 percent of energy.

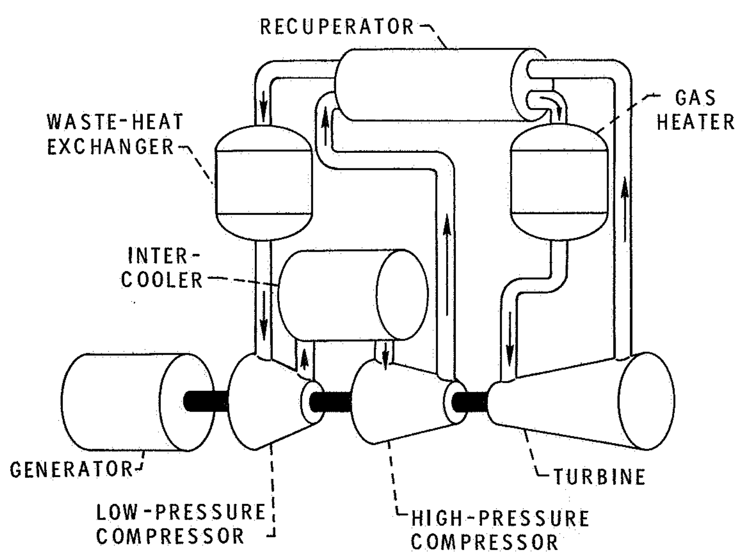

Figure 3-30. - Schematic of closed-cycle system.

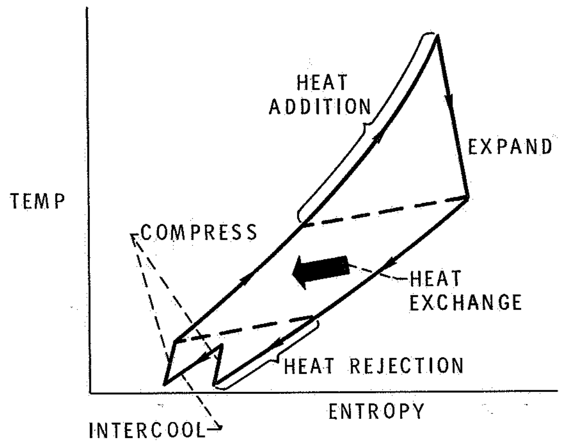

Figure 3-31. - Thermodynamic diagram for closed-cycle system.

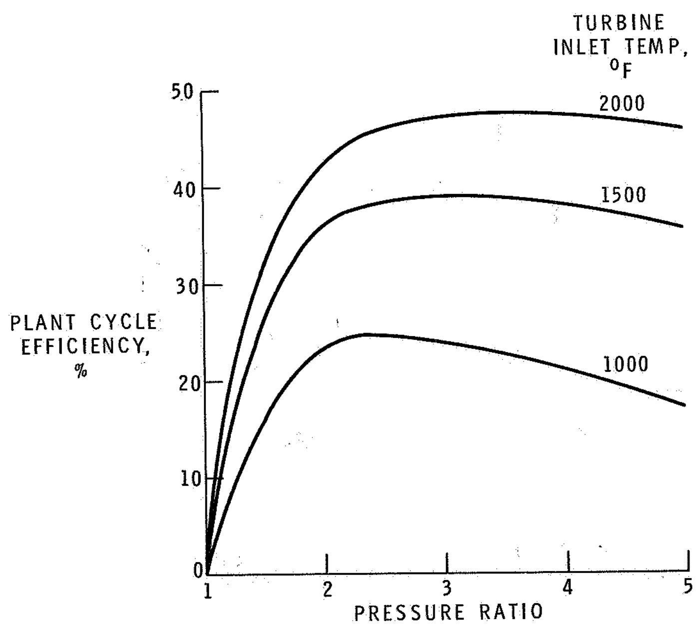

Figure 3-32. - Closed-cycle plant performance. Compressor polytropic efficiency, 88 percent; turbine polytropic efficiency, 90 percent; recuperator effectiveness, 0.9.

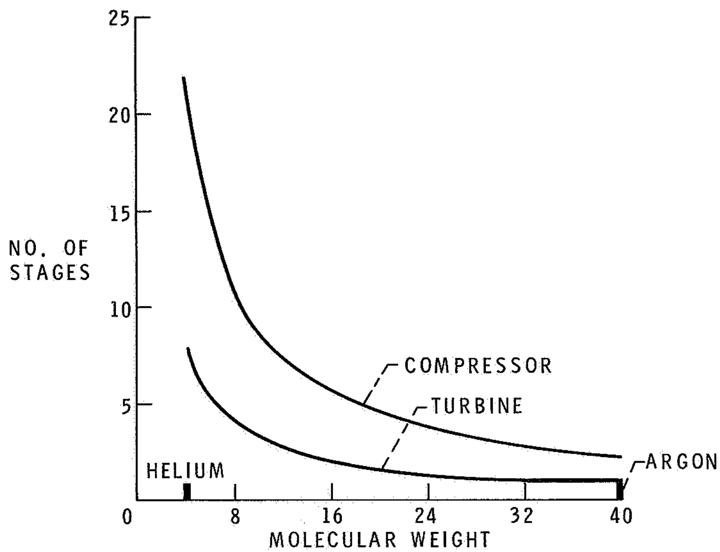

Figure 3-33. - Effect of working fluid on number of turbomachinery stages.

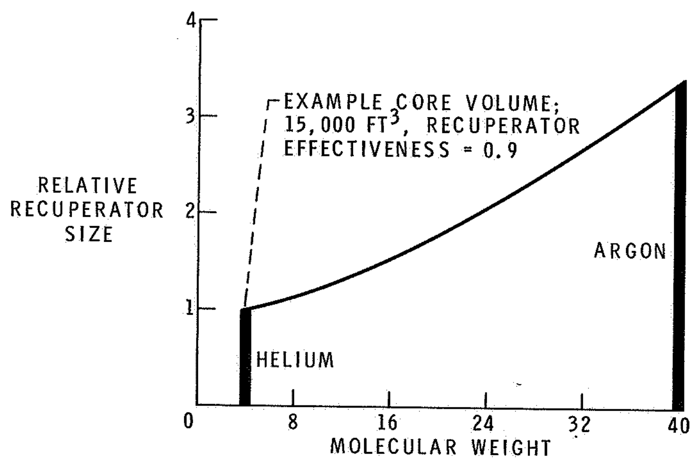

Figure 3-34. - Effect of working fluid on recuperator size.

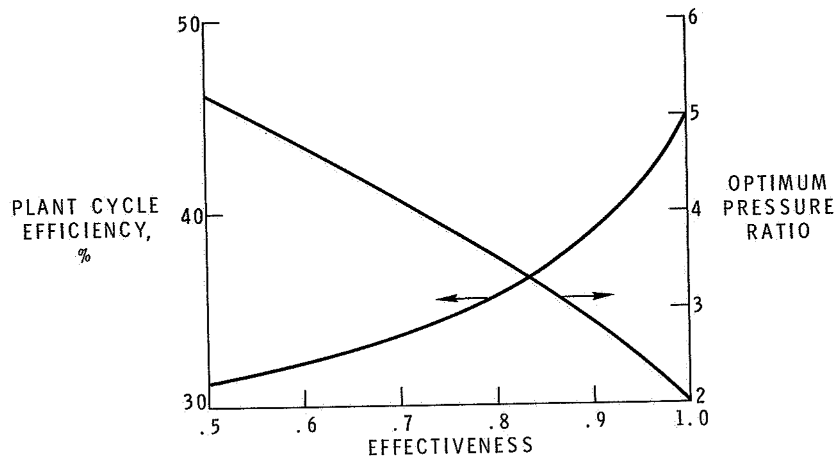

Figure 3-35. - Effect of recuperation on plant performance.

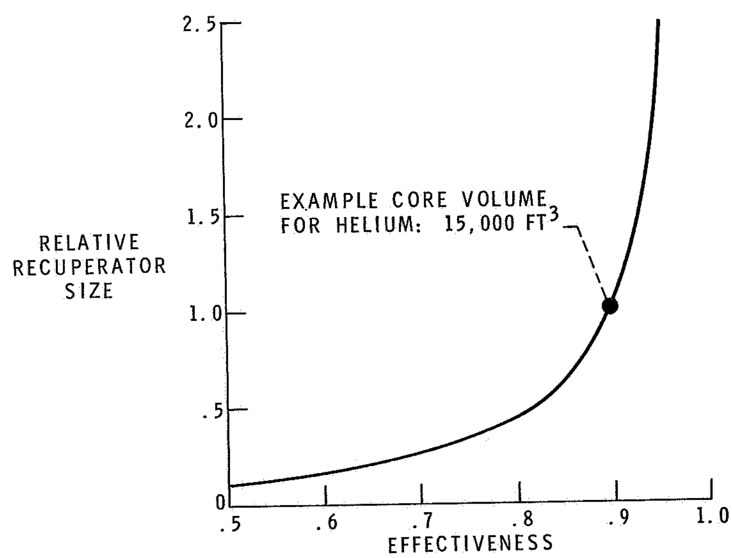

Figure 3-36. - Influence of effectiveness on recuperator size.

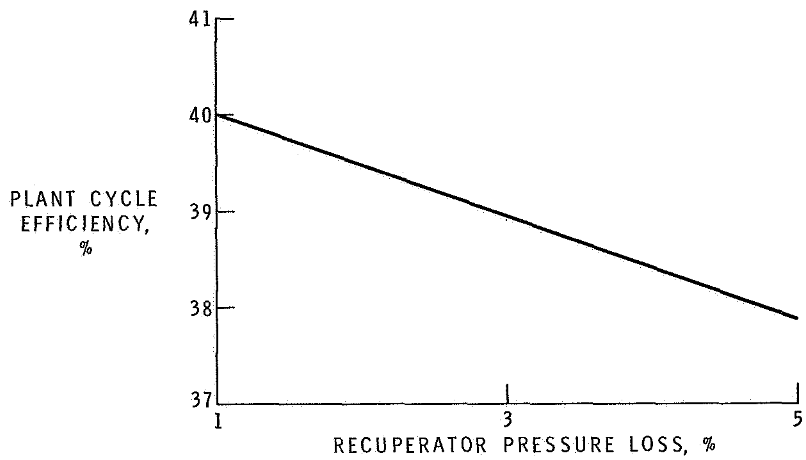

Figure 3-37. - Effect of recuperator pressure loss on plant performance.

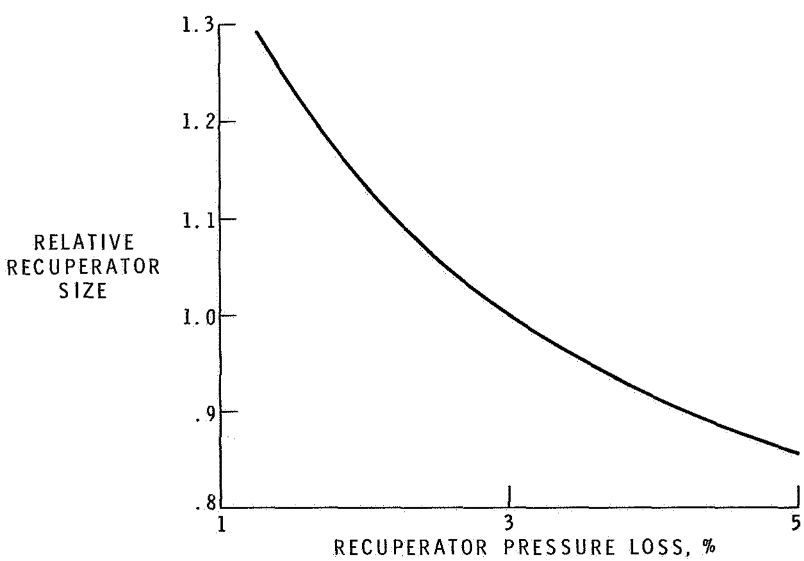

Figure 3-38. - Effect of recuperator pressure loss on recuperator size.

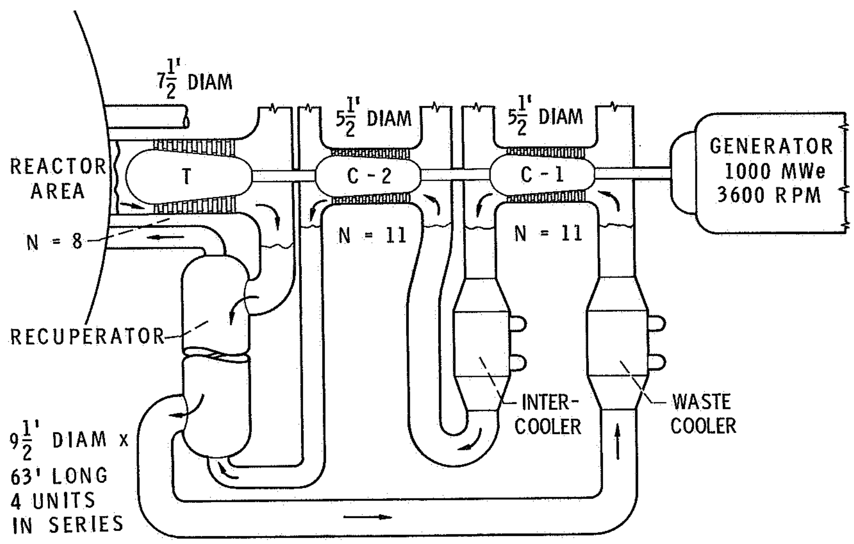

Figure 3-39. - Schematic of helium closed-cycle powerplant.

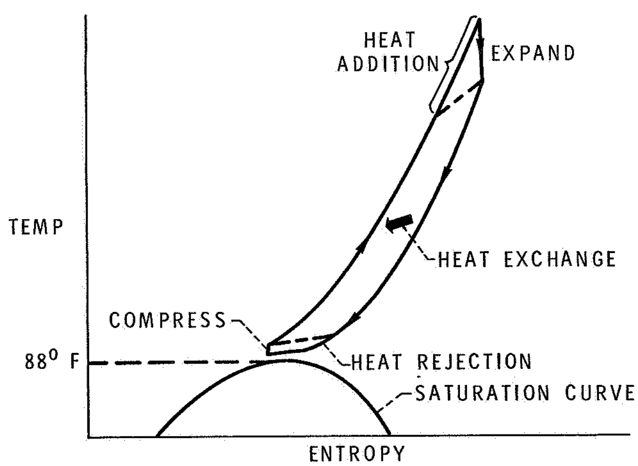

Figure 3-40. - Thermodynamic diagram for supercritical carbon dioxide cycle.

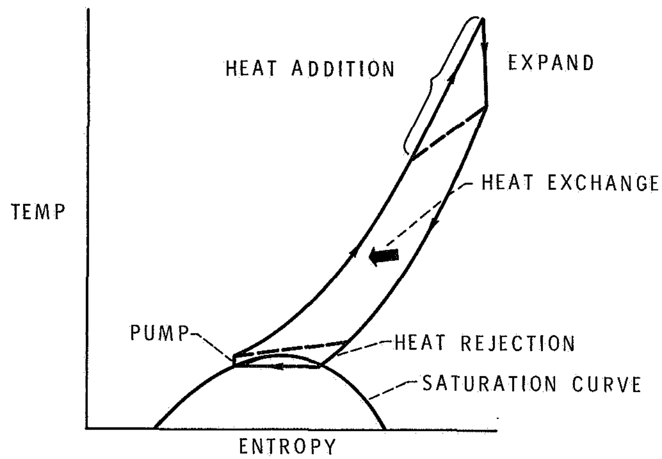

Figure 3-41. - Thermodynamic diagram for condensing carbon dioxide cycle.