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INNOVATIVE OPEN AIR BRAYTON COMBINED CYCLE SYSTEMS FOR THE NEXT GENERATION NUCLEAR POWER PLANTS

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DISSERTATION

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Abstract

The purpose of this research was to model and analyze a nuclear heated multi-turbine power conversion system operating with atmospheric air as the working fluid. The air is heated by a molten salt, or liquid metal, to gas heat exchanger reaching a peak temperature of 660 ^{0}C. The effects of adding a recuperator or a bottoming steam cycle have been addressed. The calculated results are intended to identify paths for future work on the next generation nuclear power plant (GEN-IV). This document describes the proposed system in sufficient detail to communicate a good understanding of the overall system, its components, and intended uses. The architecture is described at the conceptual level, and does not replace a detailed design document. The main part of the study focused on a Brayton -- Rankine Combined Cycle system and a Recuperated Brayton Cycle since they offer the highest overall efficiencies. Open Air Brayton power cycles also require low cooling water flows relative to other power cycles. Although the Recuperated Brayton Cycle achieves an overall efficiency slightly less that the Brayton -- Rankine Combined Cycle, it is completely free of a circulating water system and can be used in a desert climate.

Detailed results of modeling a combined cycle Brayton-Rankine power conversion system are presented. The Rankine bottoming cycle appears to offer a slight efficiency advantage over the recuperated Brayton cycle. Both offer very significant advantages over current generation Light Water Reactor steam cycles. The combined cycle was optimized as a unit and lower pressure Rankine systems seem to be more efficient. The combined cycle requires a lot less circulating water than current power plants. The open-air Brayton systems appear to be worth investigating, if the higher temperatures predicted for the Next Generation Nuclear Plant do materialize.

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CHAPTER 1 INTRODUCTION

The purpose of this dissertation is to present the results of a analysis of a nuclear multi-turbine power conversion system operating with atmospheric air as the working fluid. The air is heated by a liquid metal, or molten salt, to gas heat exchanger reaching a peak temperature of 660 ^{0}C. The effects of adding a recuperator or a bottoming steam cycle have been addressed. The calculated results are intended to identify paths for future work on new generation nuclear power plants (GEN-IV). This document describes the proposed systems in sufficient detail to communicate a good understanding of the overall solution. The architecture is described in this document at the conceptual level, and does not replace a detailed design document. The Brayton-Rankine Combined Cycle and Recuperated Brayton Cycle have been analyzed since they offer the overall highest efficiencies. Open Air Brayton power cycles require significantly lower cooling water flows relative to other power cycles. The Recuperated Brayton Cycle is slightly less efficient than the Brayton-Rankine Combined Cycle. But it is completely free of a circulating water requirement and can be used in an environment without a water source. These results demonstrate a very appealing and attractive case for the technology.

The background for this work is based on two main development trends. The first is the increased consumption of electrical energy in the US and the world with its associated environmental effects, and the second is the development of technologies in the electrical utility industry and research communities to meet this trend. The first trend is discussed in Chapter 3 and the utility and power production industry's attempt to meet these challenges, with an emphasis on nuclear systems is discussed in Chapter 4. At this time the most efficient and environmentally friendly nuclear system appears to be an open air nuclear Brayton combined cycle system. The analytic method used to justify this claim is laid out in Chapter 5, and the results of applying this method to a 25 MW(e) power conversion system that is either a Brayton-

Rankine Combined Cycle or a Recuperated Brayton Cycle are presented in Chapters 4 trough 6. Conclusions are discussed in Chapter 7.

CHAPTER 2 ENERGY RESOURCES AND THE ROLE OF NUCLEAR ENERGY

Energy is broadly defined as the ability to produce a change from the existing conditions. Thus, the term *energy* implies that a capacity for action is present. The evaluation of energy is done by measuring certain effects that are classified by descriptive names, and these effects can be produced under controlled conditions. For example, mass that is located at a certain position may have a *potential energy*. If the same mass is in motion then it may possess *kinetic energy*. If its temperature is above absolute zero, it will possess thermal energy.

2.1 The World's Energy Resources

For the past half century fossil fuels, namely, coal, oil, and natural gas, have supplied the major portion of the world's energy requirements. It has long been realized, however, that in the not too distance future these sources of energy will be largely exhausted. At the present time the total energy consumption, for all countries, is about 1×10^{17} Btu per year. Since the world's population is steadily growing and the energy use per capita is increasing as well, the rate of energy utilization by the year 2020 could well be five to ten times the current value. According to one estimate, the known coal, oil, gas and oil shale which can be extracted at no more than twice the present cost would be equivalent to roughly 4×10^{19} Btu [10]. This means that in about 100 years the world's economically useful reserves of fossil fuels may approach exhaustion.

2.2 Today's Global Energy Market

Today's global energy market places many demands on power generation technology including high thermal efficiency, low cost, rapid installation, reliability, environmental compliance and operation flexibility. The conclusion from the above estimates, even considering some margin for error, is that it is inevitable that new sources of energy must be found during the next 50 years or so if the earth is to support the growing population with some increase in living standards. Some consideration has been given to a few such source, for example, solar and wind energies as well as nuclear energy. Although solar and wind energies are very attractive, developing large scale production processes along with large farms of such systems are still some years away. On the other hand nuclear energy systems are available using fission of the heaviest elements or at the stage of research using fusion of very light nuclei. The technology of the fusion process for commercial use with controlled release of such energies using either magnetic confinement or laser driven pellets of deuterium and tritium is still far in the future. Nuclear fission, on the other hand, has already been established as a practical means for production of energy. And it may be very competitive with energy produced from fossil fuels in the very near future.

The total amount of amount of basic raw materials as sources of fuel for fission power plants such as Uranium and Thorium, in the earth's crust, to a depth of three miles, is very large, possibly something like 10^{12} tons. However, much of this is present in minerals containing such a small proportion of the desired element that extraction would be very expensive and not very cost effective. In particular high-grade ore reserves are believed to be on the order of 2×10^6 tons. We need to reduce the cost of recovery from moderately low-grade ores to at least $100 or less per pound of metal with advanced technology to fully use this resource.

Development of plant layout and modularization concepts requires an understanding of both primary and secondary systems.

General Electric's **ST**eam **A**nd **G**as (STAG) combined cycle power generation equipment has met these demands and surpassed them, taking power plant performance to unprecedented levels.

The development of steam and gas turbine combined cycles has paralleled gas turbine development, resulting in reliable combined cycle plants. Those incorporating GE's advanced gas turbine technology have achieved efficiency levels approaching 58 percent, due primarily to the higher firing temperatures of advanced technology gas turbines. The MS9001H gas turbine will achieve 60 percent efficiency in combined cycle application when it goes into full operation.

In addition to advances in gas turbine technology, steam turbine performance also has evolved. GE's STAG combined-cycle power generation product line includes steam cycle options that satisfy a wide range of economic considerations including fuel flexibility, fuel cost, duty cycle and space limitations.

Heat-exchangers, filters, turbines, and other components in integrated coal gasification combined cycle system must withstand demanding conditions of high temperatures and pressure differentials. Under the highly sulfiding conditions of the high temperature coal gas, the performance of components degrades significantly with time unless expensive high alloy materials are used. Deposition of a suitable coating on a low cost alloy may improve is resistance to such sulfidation attack and decrease capital and operating costs. A review of the literature indicates that the corrosion reaction is the competition between oxidation and sulfidation reactions. The Fe- and Ni-based high-temperature alloys are susceptible to sulfidation attack unless they are fortified with high levels of Cr, Al, and Si. To impart corrosion resistance, these elements need not be in the bulk of the alloy and need only be present at the surface layers.

2.3 End of Cheap Oil and the Future of Energy

Global production of conventional oil will begin to decline sooner than most people think, probably within 10 years. As we recall, two sudden price increases took place in 1973 and 1979 and rudely impacted the industrial world and made it to recognize its dependency on cheap crude oil. The first event in 1973 that caused an oil price increase took place in response to an Arab embargo during the Arab and Israeli war. The price tripled and then nearly doubled again when Iran's Shah was dethroned, sending the major economies into a spin. Just a few years earlier oil explorers had discovered enormous new oil reservoirs on the North Slope of Alaska and below the North Sea off the cost of Europe. The emotional and political reactions of most analysts predict a shortage of crude oil in the world due to these types of crises. Not having enough underground reservoirs for exploration of oil will put the future survival of the world economy on a critical path.

The five Middle Eastern nations who are member of Organization of Petroleum Exporting Countries (OPEC) were able to hike the price of crude oil not because oil was growing short but because they managed to control 36 percent of the international market. Later, when due to pumped oil from Alaska and North Sea, the demand for crude oil sagged, then prices of oil dropped and the OPEC's control of prices collapsed. The next oil crunch will not be so temporary. The exploration and discovery of oil fields, as well as production of it, around the world suggests that within the next decade, the supply of conventional oil will not support and cannot keep up with demand. Whether this conclusion is in contradiction with what oil companies are reporting is an open question. Today's oil production rate of about 23.6 GBO (Giga Barrel Oil) per year may suggest cheap crude oil for the next 43 years, or more, based on the official charts that show the reserves are growing. But there are three critical errors.

- **First,** it relies on distorted estimates of reserves.
- **A second mistake** is to pretend that production will remain constant.
- **Third** and most important, conventional wisdom erroneously assumes that the last bucket of oil can be pumped from the ground just as quickly as the barrels of oil gushing from wells today.

In fact, the rate at which any well—or any country—can produce oil always rises to a maximum and then, when about half the oil is gone, begins falling gradually back to zero.

From an economic perspective then, when the world runs completely out of oil is thus not directly relevant:

What matters is when production begins to taper off. Beyond that point, prices will rise unless demand declines commensurately.

Using several different techniques to estimate the current reserves of conventional oil and the amount still left to be discovered, many experts in the field concluded that the decline would begin before 2010.

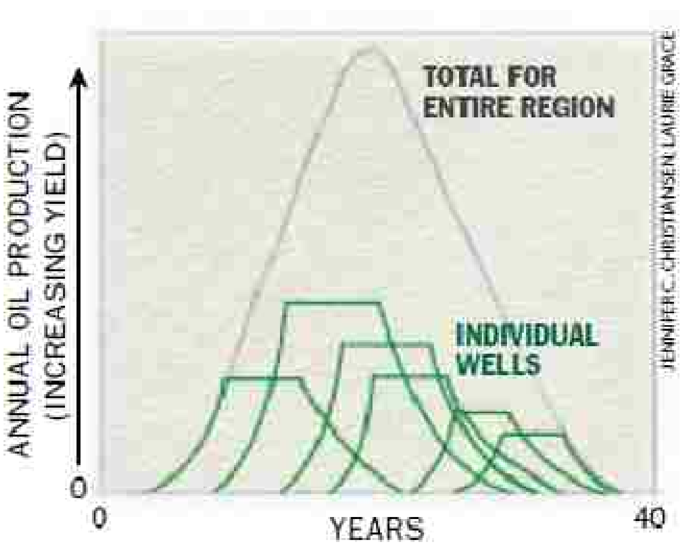


Figure 2-1: FLOW OF OIL starts to fall from any large region when about half the crude is gone. Adding the output of fields of various sizes and ages (*green curves above*) usually yields a bell-shaped production curve for the region as a whole. M. King Hubbert, a geologist with Shell Oil, exploited this fact in 1956 to predict correctly that oil from the lower 48 American states would peak around 1969.

In practice, companies and countries are often deliberately vague about the likelihood of the reserves they report, preferring instead to publicize whichever figure, within a P10 to P90 range, best suits them. Exaggerated estimates can, for instance, raise the price of an oil company's stock.

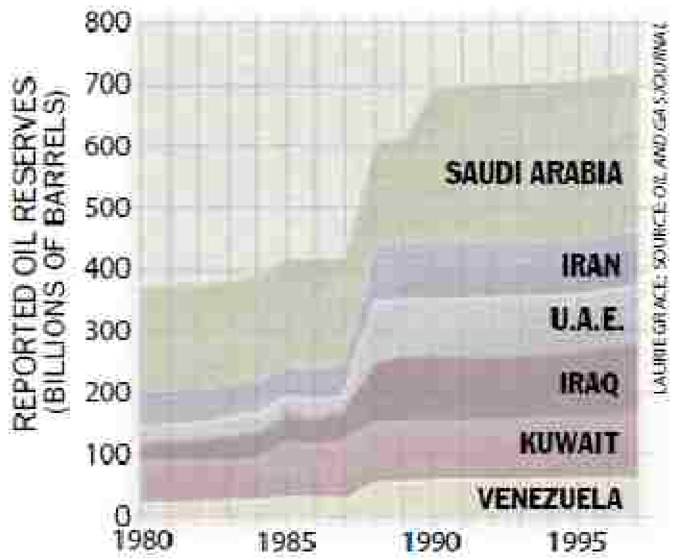


Figure 2-2: SUSPICIOUS JUMP in reserves reported by six OPEC members added 300 billion barrels of oil to official reserve tallies yet followed no major discovery of new fields.

The members of OPEC have faced an even greater temptation to inflate their reports because the higher their reserves, the more oil they are allowed to export. National companies, which have exclusive oil rights in the main OPEC countries, need not (and do not) release detailed statistics on each field that could be used to verify the country's total reserves. There is thus good reason to suspect that when, during the late 1980s, six of the eleven OPEC nations increased their reserve figures by colossal amounts, ranging from 42 to 197 percent; they did so only to boost their export quotas.

Meanwhile global demand for oil is currently rising at more than 2 percent a year. Since 1985, energy use is up about 30 percent in Latin America, 40 percent in Africa and 50 percent in Asia.

The Energy Information Administration forecasts that worldwide demand for oil will increase 60 percent (to about 40 GBO a year) by 2020.

The switch from growth to decline in oil production will thus almost certainly create economic and political tension. Unless alternatives to crude oil quickly prove themselves, the market share of the OPEC states in the Middle East will rise rapidly. Within two years, these nations' share of the global oil business will pass 30 percent, nearing the level reached during the oil-price shocks of the 1970s.

The world could thus see radical increases in oil prices. That alone might be sufficient to curb demand, flattening production for perhaps 10 years (Demand fell more than 10 percent after the 1979 shock and took 17 years to recover). Many Middle Eastern nations will soon themselves be past the midpoint. World production will then have to fall.

With sufficient preparation, however, the transition to the post-oil economy need not be traumatic. If advanced methods of producing liquid fuels from natural gas can be made profitable and scaled up quickly, gas could become the next source of transportation fuel [see "Liquid Fuels from Natural Gas," by Safaa A. Fouda, on page 92] [11]. Safer nuclear power, cheaper renewable energy, and oil conservation programs could all help postpone the inevitable decline of conventional oil.

Countries should begin planning and investing now. In November 2009 a panel of energy experts appointed by President Bill Clinton strongly urged the administration to increase funding for energy research by $1 billion over the next five years. That was a small step in the right direction, one that must be followed by giant leaps from the private sector.

The world is not running out of oil—at least not yet. What our society does face, and soon, is the end of the abundant and cheap oil on which all industrial nations depend.

2.4 What to Do About Coal

Cheap plentiful coal is expected to fuel power plants for the foreseeable future, but can we keep it from devastating the environment?

Figure 2-3: Iron hydroxide precipitate (orange) in a Missouri stream receiving acid drainage from surface coal mining Image: U.S. Geological Survey

To deal with climate change means addressing the problems posed by emissions from coal-fired power plants. Unless we take prompt action to strictly limit the amount of carbon dioxide (CO_2) released into the atmosphere when consuming coal to make electricity, we have little chance of gaining control over global warming. The overview of burning coals side effects are;

- Coal is widely burned for power plants to produce electricity, but it also produces large quantities of climate changing carbon dioxide.
- Compared with conventional power plants, new gasification facilities can more effectively and affordably extract CO_2 so it can be safely stored underground.
- The world must begin implementing carbon sequestration to stave off global warming.

Coal—the fuel that powered the Industrial Revolution—is a particularly worrisome source of energy, in part because burning it produces considerably more carbon dioxide per unit of electricity generated than burning either oil or natural gas does. In addition, coal is cheap and will remain abundant long after oil and natural gas have become very scarce. With coal plentiful and inexpensive, its use is expanding in the U.S. and elsewhere and is expected to continue rising in areas with abundant coal resources [11].

Indeed, U.S. power providers are expected to build the equivalent of nearly 280 500-megawatt, coal-fired electricity plants between 2003 and 2030. Meanwhile China is already constructing the equivalent of one large coal-fueled power station a week. Over their roughly 60-year life spans, the new generating facilities in operation by 2030 could collectively introduce into the atmosphere about as much carbon dioxide as was released by all the coal burned since the dawn of the Industrial Revolution [11].

Coal use can lead to a range of harmful consequences, including decapitated mountains, air pollution from acidic and toxic emissions, and water fouled with coal wastes. Extraction also endangers and can kill miners. Together such effects make coal production and conversion to useful energy one of the more destructive activities on the planet [11].

We need to find alternative answers to the above issues in order to deal with future demand. The only answer is to move as quickly as possible to alternative fuels—including natural gas and nuclear power, as well as solar, wind and geothermal energy. “Running out of energy in the long run is not the problem, but the bind comes during the next 10 years and we need to get over our dependency on crude oil” [11].

2.5 The Future of Energy

The energy future will be very different. For all the uncertainties highlighted in various reports by expert in the field, we can be certain that the energy world will look a lot different in 2030 than it does today. The world energy system will be transformed, but not necessarily in the way we would like to see. We can be confident of some of the trends highlighted in reports on current global trends in energy supply and consumption, environmentally, economically, and socially. But that can — and must — be altered when there is still time to change the road we are on. The growing weight of China, India, the Middle East and other non-OECD regions in energy markets and in CO_2 emissions is something we need to take under consideration in order to deal with global warming. The rapidly increasing dominance of national oil companies and the emergence of low-carbon energy technologies seems one necessary solution to the problem in hand, but not sufficient enough. And while market imbalances could temporarily cause prices to fall back, it is becoming increasingly apparent that the era of cheap oil is over. But many of the key policy drivers (not to mention other, external factors) remain in doubt. It is within the power of all governments, of producing and consuming countries alike, acting alone or together, to steer the world towards a cleaner, cleverer and more competitive energy system. Time is running out and the time to act is now. So what we need to ask is that "*Can Nuclear Power Compete*?"

A variety of companies that are in the energy production business, say the answer may be yes. Manufacturers have submitted new designs to the Nuclear Regulatory Commission’s safety engineers, and that agency has already approved some as ready for construction, if they are built on a previously approved site. Utilities, reactor manufacturers and architecture/engineering firms have formed partnerships to build plants, pending final approvals. Swarms of students are enrolling in college-level nuclear engineering programs. And rosy projections from industry and government predict a surge in construction.

Figure 2-4: Typical Nuclear Plant in Our Backyard

Like another moon shot, the launch of new reactors after a 35-year hiatus in orders is certainly possible, though not a sure bet. It would be easier this time, the experts say, because of technological progress over the intervening decades. But as with a project as large as a moon landing, there is another question: Would it be worthwhile?

In order to answer this question we need to at least satisfy the four unresolved problems associated with nuclear power that were brought up by an MIT report and they were mentioned at the beginning of this write up. In order to argue the first point which is the cost of producing a nuclear power plant with its modern and today's technologies from total ownership and return on investment, we need to understand the nature of the beast from the day it was born in the basement of University of Chicago.

2.6 Nuclear Reactors for Power Production

In the United States, most reactor design and development for the generation of electrical power branched from early nuclear navy research, when it was realized that a compact nuclear power plant would have great advantages for submarine propulsion. Such a power plant would make possible long voyages across the oceans at high speeds without the necessity for resurfacing at frequent intervals. Argonne National Laboratory was assigned the task of designing such reactor. So the first generation of Pressurized Water Reactors (PWR) was born. It used highly enriched uranium as the fuel, and water under pressure as the moderator as well as coolant. The first prototype of this reactor named STR Mark 1 started operation at Arco, Idaho, in March 1953 and a production version of it was installed in the U.S.S Nautilus, the first nuclear powered submarine in May, 1953.

As a result of the experience gained in successful operation of the submarine reactors, the first commercial version of a PWR, was designed and installed at Shipping port, Pennsylvania and went into operation in December 2, 1957 with a water pressure of 13.8 MPa (2000 psi). The steam produced in the heat exchanger was at a temperature of about 254 ^{0}C (490 ^{0}F) and a pressure of close to 4.14 MPa (600 psi). In order to make the reactor cost effective and reduce the cost of the power produced, only a small number of the fuel elements were highly enriched in Uranium-235 (U^{235}) as an alloy with zirconium. The remainder was of normal Uranium Dioxide. The change in core design required more real estate for the foot print of a commercialized PWR. This was not an issue for a land-based facility. The output power of this reactor was about 60 MW(e) and 230 MW(t) Further enhancement in core design increased the power to 150 MW(e) and 505MW(t). Pressurized Water Reactors, using slightly (2 to 6 percent) enriched Uranium Dioxide as the fuel, are now commonly used in United States and other countries around the Globe for commercial power generation. The most recent plants have electrical output in the neighborhood of 1000 MW(e) (3000 MW(t)).

Later on other reactor designs based on different fuel materials, moderators, and coolants with various electrical and thermal powers output were born, Examples are the following:

- Boiling Water Reactor (BWR) initiated in 1953.
- Water Cooled Graphite Moderated in 1954.
- High Temperature, Gas Cooled Reactor (**HTGR**).
- Liquid Metal Fast Breeder Reactors (**LMFBR**).

Basically, all commercial reactor power plants of present interest are systems for generating steam utilizing the heat of nuclear fission to boil water and produce steam for a turbine. They are often referred to as "**N**uclear **S**team **S**upply **S**ystems" or **NSSS**. The steam is expanded in a turbine which drives a generator to produce electricity in the conventional manner. The exhaust steam from the turbine passes on to a condenser where it is converted into liquid water and this is returned as feed water to the steam generator of the **NSSS**.

The proportion of the heat supplied in a power plant that is actually converted into electrical energy is called the *Thermal Efficiency* of the system; thus, in a nuclear installation,

$$\text{Thermal Efficiency} = \frac{\text{Electrical Energy Generated}}{\text{Heat Produced in the Reactor}}$$

The maximum possible value of the thermal efficiency is the *Ideal Thermodynamic Efficiency*, which is given by following relationship;

$$\text{Ideal Thermodynamic Efficiency} = \frac{T_2 - T_1}{T_2}$$

where

T_1 = is the absolute temperature of the steam entering the turbine (^{0}K Kelvin).

T_2 = is the temperature at which heat is rejected to the condenser (^{0}K Kelvin)

The ideal thermodynamic efficiency can be increased by having T_2 as high as possible and T_1 as low as possible. In practice, T_1 is more or less fixed by the ambient temperature; the thermal efficiency of a steam electric plant is then largely determined by the steam temperature, which should be as high as feasible.

Conditions in PWRs and BWRs are such that the steam temperature is lower than in modern fossil-fuel power plants, in which the heat is produced by burning coal, oil, or gas. The thermal efficiencies of these nuclear reactor plants is only about 33 percent, compared with 40 percent for the best fossil-fuel facilities. With the HTGRs and fast breeder reactors, however, the thermal efficiencies should equal to those of the best fossil-fuel plants, i.e., about 40 percent.

2.7 Future Nuclear Power Plant System

In response to the difficulties in achieving suitability, a sufficiently high degree of safety and a competitive economic basis for nuclear power, the U.S. Department of Energy initiated the Generation IV program in 1999. Generation IV refers to the broad division of nuclear designs into four categories as follows;

1. Early prototype reactor (Generation I).
2. The large central station nuclear power plants of today (Generation II).
3. The advanced light-water reactors and other systems with inherent safety features that have been designed in recent years (Generation III).
4. The next generation system to be designed and built two decades from now (Generation IV).

By 2000 international interest in the Generation IV project had resulted in a nine country coalition that includes:

i. Argentina
ii. Brazil
iii. Canada
iv. France
v. Japan
vi. South Africa
vii. South Korea
viii. United Kingdom
ix. United States of America

Participants are mapping out and collaborating on the research and development of future nuclear energy systems.

Although the Generation IV program is exploring a wide variety of new systems, a few examples serve to illustrate the broad approaches to reactor designs that are developing to meet the objectives.

The next-generation systems are based on three general classes of reactors:

1) Gas-cooled,
2) Water-cooled, and
3) Fast-spectrum.

All these categories and their brief designs are discussed in the following sections.

2.8 Next Generation of Nuclear Power Reactors for Power Production

Experts are projecting worldwide electricity consumption will increase substantially in the coming decades, especially in the developing world. The accompanying economic growth and social progress will have a direct impact on rising electricity prices. This has focused fresh attention on nuclear power plants. New, safer and more economical nuclear reactors could not only satisfy many of our future energy needs but could combat global warming as well. Today's existing nuclear power plants on line in the United States provide a fifth of the nation's total electrical output.

Taking into account the expected increase in energy demand worldwide and the growing awareness about global warming, climate change issues and sustainable development, nuclear energy will be needed to meet future global energy demand.

Nuclear power plant technology has evolved as distinct design generations as mentioned in the previous section and is briefly summarized here again as follows:

- First Generation: prototypes, and first realizations (~1950 - 1970).
- Second Generation: current operating plants (~1970 - 2030).
- Third generation: deployable improvements to current reactors (~2000 and on).
- Fourth generation: advanced and new reactor systems (2030 and beyond).

The **G**eneration IV **I**nternational **F**orum, or **GIF**, was chartered in July 2001 to lead the collaborative efforts of the world's leading nuclear technology nations to develop next generation nuclear energy systems to meet the world's future energy needs.

Eight technology goals have been defined for Generation IV systems in four broad areas:

1. Sustainability,
2. Economics,
3. Safety and Reliability, and finally,
4. Proliferation resistance and Physical protection

A large number of countries share these ambitious goals as they aim at responding to economic, environmental and social requirements of the 21st century. They establish a framework and identify concrete targets for focusing GIF R&D efforts

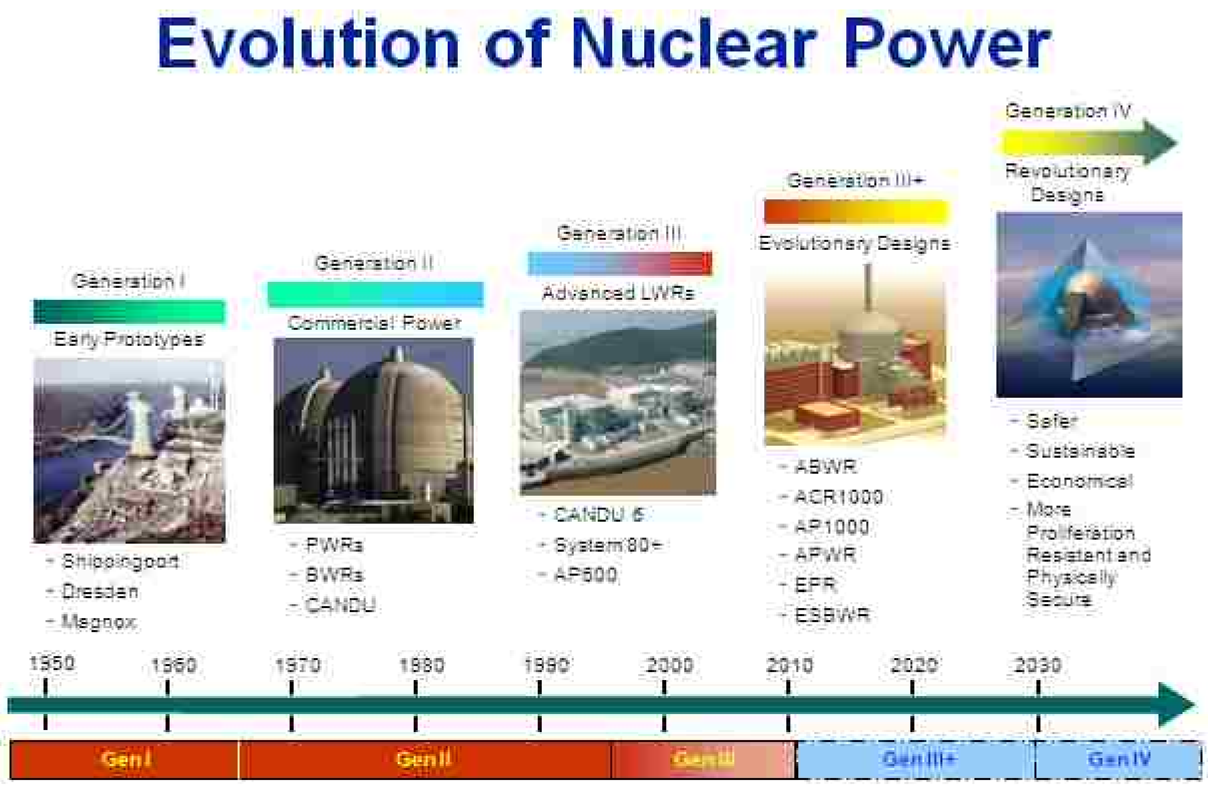


Figure 2-5: Evolution of Nuclear Power Plants

2.9 Goals for Generation IV Nuclear Energy Systems

The next generation ("Generation IV") of nuclear energy systems is intended to meet the below goals (while being at least as effective as the "third" generation in terms of economic competitiveness, safety and reliability) in order to provide a sustainable development of nuclear energy.

Sustainability – 1	Generation IV nuclear energy systems will provide sustainable energy generation that meets clean air objectives and provides long term availability of systems and effective fuel utilization for worldwide energy production.
Sustainability – 2	Generation IV nuclear energy systems will minimize and manage their nuclear waste and notably reduce the long term stewardship burden, thereby improving protection for the public health and the environment.
Economics – 1	Generation IV nuclear energy systems will have a clear life cycle cost advantage over other energy sources.
Economics – 2	Generation IV nuclear energy systems will have a level of financial risk comparable to other energy projects.
Safety and Reliability - 1	Generation IV nuclear energy systems operations will excel in safety and reliability.
Safety and Reliability – 2	Generation IV nuclear systems will have a very low likelihood and degree of reactor core damage.
Safety and Reliability – 3	Generation IV nuclear energy systems will eliminate the need for offsite emergency response.
Proliferation resistance and Physical Protection	Generation IV nuclear energy systems will increase the assurance that they are very unattractive and the least desirable route for diversion or theft of weapons usable materials, and provide increased physical protection against acts of terrorism.

Table 2-1: Goals for Generation IV Nuclear Energy Systems

In principle, the Generation IV Systems should be marketable or deployable from 2030 onwards. The systems should also offer a true potential for new applications compatible with an expanded use of nuclear energy, in particular in the fields of hydrogen or synthetic hydrocarbon production, seawater desalination and process heat production.

It has been recognized that these objectives, widely and officially shared by a large number of countries, should be the basis of an internationally shared R&D program, which allows keeping open and consolidating the technical options, and avoiding any early or premature down selection.

In fact, because the next generation nuclear energy systems will address needed areas of improvement and offer great potential, many countries share a common interest in advanced R&D that will support their development. The international research community should explore such development benefits with the identification of promising research areas and collaborative efforts. The collaboration on R&D by many nations on the development of advanced next generation nuclear energy systems will in principle aid the progress toward the realization of such systems, by leveraging resources, providing synergistic opportunities, avoiding unnecessary duplication, and enhancing collaboration.

2.10 A Technology Roadmap for Generation IV Nuclear Energy Systems

The technology roadmap defines and plans the necessary research and development (R&D) to support the next generation of innovative nuclear energy systems known as Generation IV. The roadmap has been an international effort of ten countries, including Argentina, Brazil, Canada, France, Japan, Republic of Korea, South Africa, Switzerland, the United Kingdom, and the United States, the International Atomic Energy Agency, and the OECD Nuclear Energy Agency.

Beginning in 2001, over 100 experts from these countries and international organizations began work on defining the goals for new systems, identifying many promising concepts, and evaluating them, and defining the R&D needed for the most promising systems. By the end of 2002, the work resulted in a description of the six most promising systems and their associated R&D needs and they are listed below.

1. Gas-cooled Fast Reactor (**GFR**): Features a fast-neutron-spectrum, helium-cooled reactor and closed fuel cycle;
2. Very-High-Temperature Reactor (**VHTR**): A graphite-moderated, helium-cooled reactor with a once-through uranium fuel cycle;
3. Supercritical-Water-cooled Reactor (**SCWR**): A high-temperature, high-pressure, water-cooled reactor that operates above the thermodynamic critical point of water;
4. Sodium-cooled Fast Reactor (**SFR**): Features a fast-spectrum, sodium-cooled reactor and closed fuel cycle for efficient management of actinides and conversion of fertile uranium;
5. Lead-cooled Fast Reactor (**LFR**): Features a fast-spectrum, lead/bismuth eutectic liquid-metal-cooled reactor and a closed fuel cycle for efficient conversion of fertile uranium and management of actinides;
6. Molten Salt Reactor (**MSR**): Produces fission power in a circulating molten salt fuel mixture with an epithermal-spectrum reactor and a full actinide recycling fuel cycle.

These systems offer significant advances in sustainability, safety and reliability, economics, proliferation resistance and physical protection. These six systems feature increased safety, improved economics for electricity production and new products such as hydrogen for transportation applications, reduced nuclear wastes for disposal, and increased proliferation resistance.

In 2009, the Experts Group published an outlook on Generation IV R&D, to provide a view of what GIF members hope to achieve collectively in the period 2010-2014. All Generation IV systems have features aiming at performance improvement, new applications of nuclear energy, and/or more sustainable approaches to the management of nuclear materials. High-temperature systems offer the possibility of efficient process heat applications and eventually hydrogen production. Enhanced sustainability is achieved primarily through adoption of a closed fuel cycle with reprocessing and recycling of plutonium, uranium and minor actinides using fast reactors; this approach provides significant reduction in waste generation and uranium resource requirements.

The following Table summarizes the main characteristics of the six Generation IV systems.

System	Neutron Spectrum	Coolant	Temp. ^{0}C	Fuel Cycle	Size (MWe)
VHTR (Very High Temperature gas Reactor)	Thermal	Helium	900 to 1000	Open	250 - 300
SFR (Sodium-cooled Fast Reactor)	Fast	Sodium	550	Closed	30 - 150, 300 - 1500 1000 - 2000
SCWR (Supercritical Water – cooled Reactor)	Thermal/Fast	Water	510 - 625	Open/Closed	300 – 700 1000 - 2000
GFR (Gas – cooled Fast Reactor)	Fast	Helium	850	Closed	1200
LFR (Lead – cooled Fast Reactor)	Fast	Lead	480 - 800	Closed	20 – 180 300 – 1200 600 - 1000
MSR (Molten Salt Reactor)	Epithermal	Fluoride Salt	700 - 800	Closed	1000

Table 2-2: Summary of the main characteristics of the six Generation IV systems

2.11 The Description of the Six most Promising Nuclear Power Systems

A brief summary of each Generation-IV nuclear power systems are as follows:

VHTR: The very - high temperature reactor is a next step in the evolutionary development of high-temperature reactors. The VHTR is a helium gas-cooled, graphite-moderated, thermal neutron spectrum reactor with a core outlet temperature greater than 900 °C, and a goal of 1000 °C, sufficient to support production of hydrogen by thermo-chemical processes. The reference reactor thermal power is set at a level that allows passive decay heat removal, currently estimated to be about 600 MWth. The VHTR is primarily dedicated to the cogeneration of electricity and hydrogen, as well as to other process heat applications. It can produce hydrogen from water by using thermo-chemical, electro-chemical or hybrid processes with reduced emission of CO_2 gases. At first, a once-through LEU (<20% U^{235}) fuel cycle will be adopted, but a closed fuel cycle will be assessed, as well as potential symbiotic fuel cycles with other types of reactors (especially light-water reactors) for waste reduction.

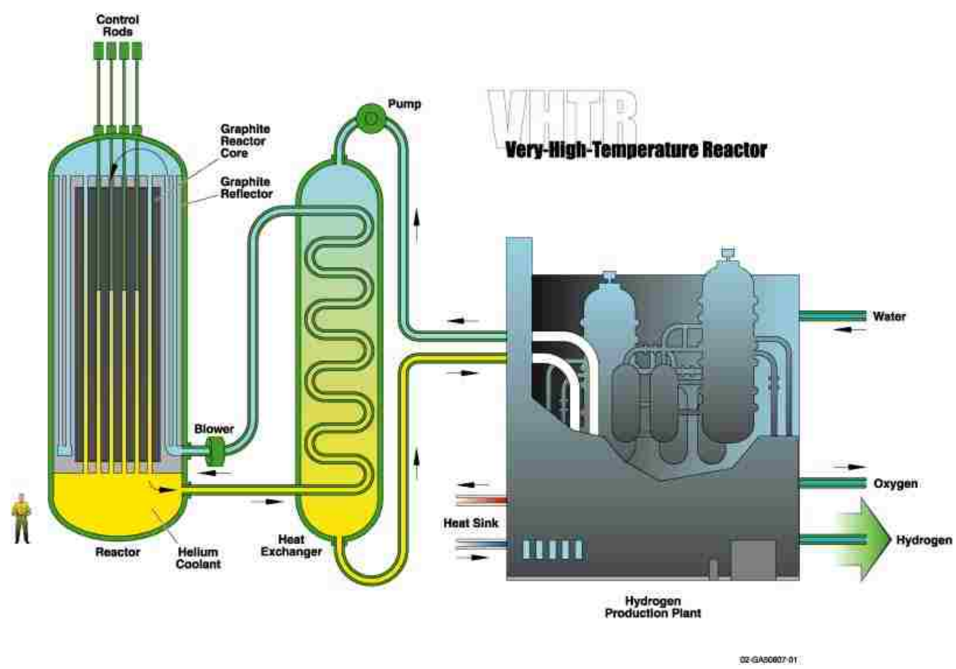


Figure 2-6: Very High Temperature Reactor

SFR: The sodium - cooled fast reactor system uses liquid sodium as the reactor coolant, allowing high power density with low coolant volume fraction. The reactor can be arranged in a pool layout or a compact loop layout. Reactor size options under consideration range from small (50 to 300 MWe) modular reactors to larger reactors (up to 1500 MWe). The two primary fuel recycle technology options are advanced aqueous and pyrometallurgical processing. A variety of fuel options are being considered for the SFR, with mixed oxide preferred for advanced aqueous recycle and mixed metal alloy preferred for pyrometallurgical processing. Owing to the significant past experience accumulated with sodium cooled reactors in several countries, the deployment availability of SFR systems is targeted for 2020.

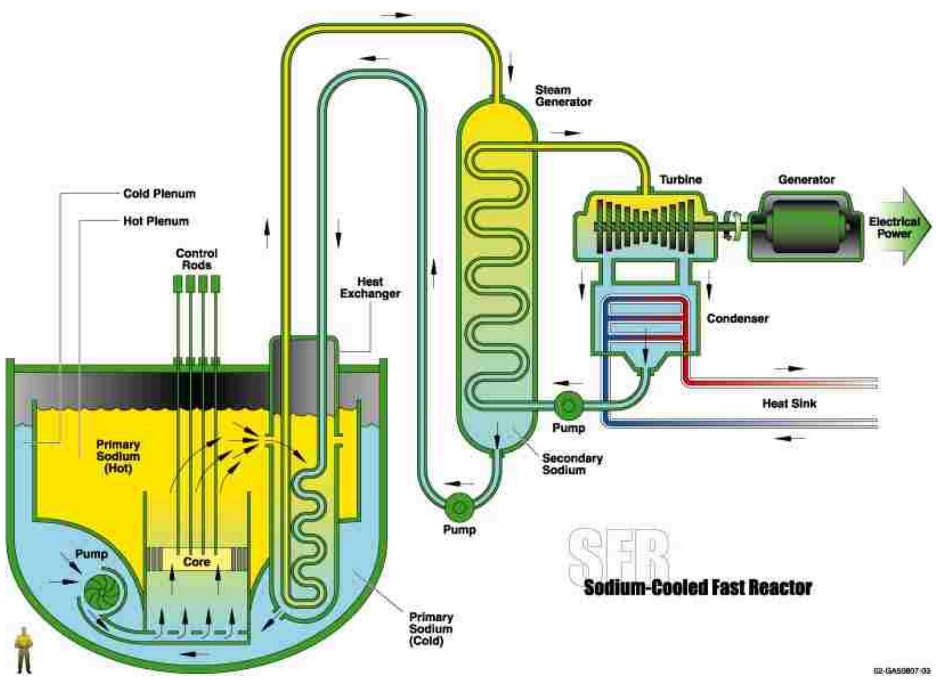


Figure 2-7: Sodium Cooled Fast Reactor

SCWR: Supercritical – water – cooled reactors are a class of high - temperature, high - pressure water - cooled reactors operating with a direct energy conversion cycle and above the thermodynamic critical point of water (374 °C, 22.1 MPa). The higher thermodynamic efficiency and plant simplification opportunities afforded by a high - temperature, single - phase coolant translate into improved economics. A wide variety of options are currently considered: both thermal - neutron and fast - neutron spectra are envisaged and both pressure vessel and pressure tube configurations are considered. The operation of a 30 to 150 MWe technology demonstration is targeted for 2022.

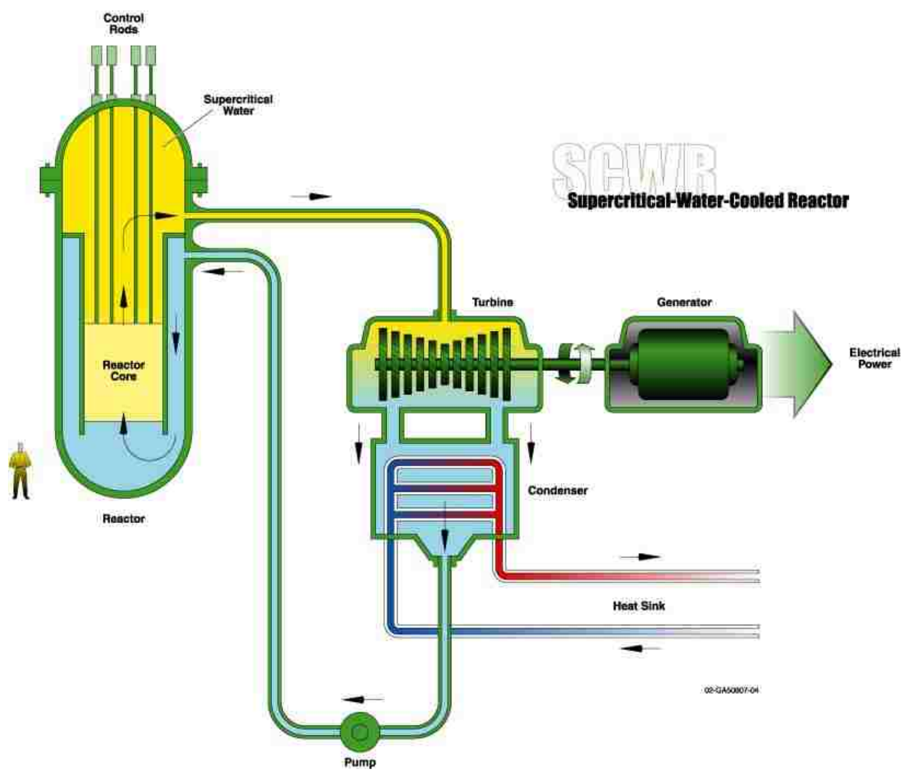


Figure 2-8: Supercritical Water Cooled Temperature Reactor

GFR: The main characteristics of the gas - cooled fast reactor are fissile self - sufficient cores with fast neutron spectrum, robust refractory fuel, high operating temperature, high efficiency electricity production, energy conversion with a gas turbine and full actinide recycling possibly associated with an integrated on - site fuel reprocessing facility. A technology demonstration reactor needed to qualify key technologies could be put into operation by 2020.

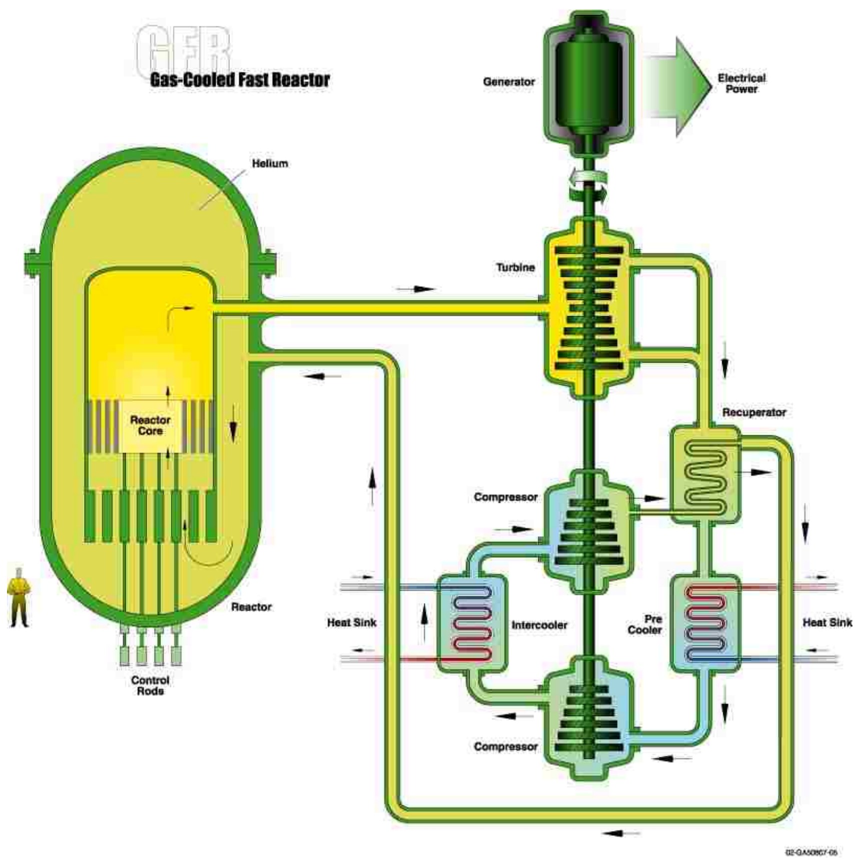


Figure 2-9: Gas Cooled Fast Reactor

LFR: The lead cooled fast reactor system is characterized by a fast - neutron spectrum and a closed fuel cycle with full actinide recycling, possibly in central or regional fuel cycle facilities. The coolant could be either lead (preferred option), or lead/bismuth eutectic. The LFR can be operated as a breeder; a burner of actinides from spent fuel, using inert matrix fuel; or a burner/breeder using thorium matrices. Two reactor size options are considered: a small transportable system of 50 to 150 MWe with a very long core life and a medium system of 300 to 600 MWe. In the long term a large system of 1200MWe could be envisaged. The LFR system may be deployable by 2025.

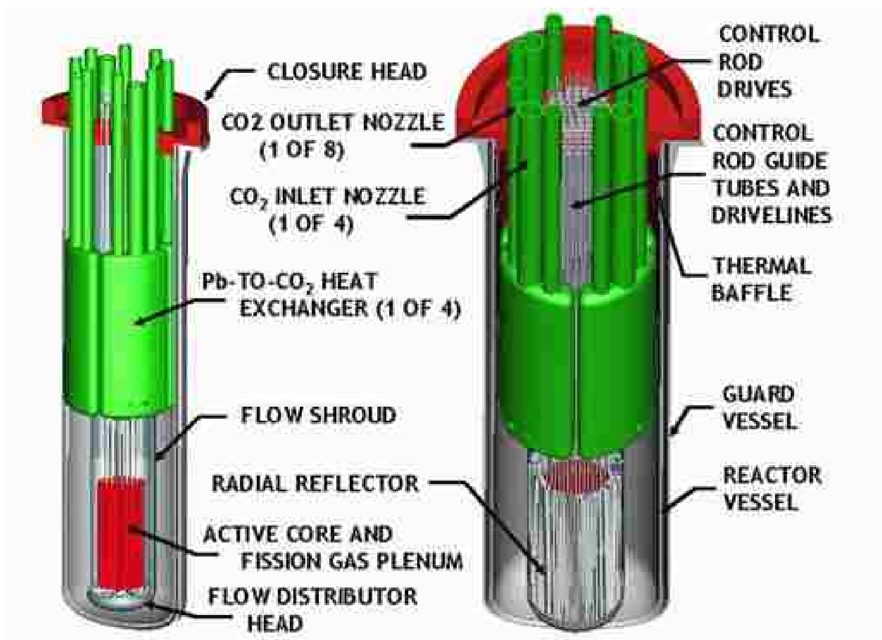


Figure 2-10: Lead Cooled Fast Reactor

MSR: The molten salt reactor system embodies the very special feature of a liquid fuel. MSR concepts, which can be used as efficient burners of TRU from spent LWR fuel, have also a breeding capability in any kind of neutron spectrum ranging from thermal (with a thorium based fuel cycle) to fast (with the U - Pu fuel cycle). Whether configured for burning or breeding, MSRs have considerable promise for the minimization of radiotoxic nuclear waste.

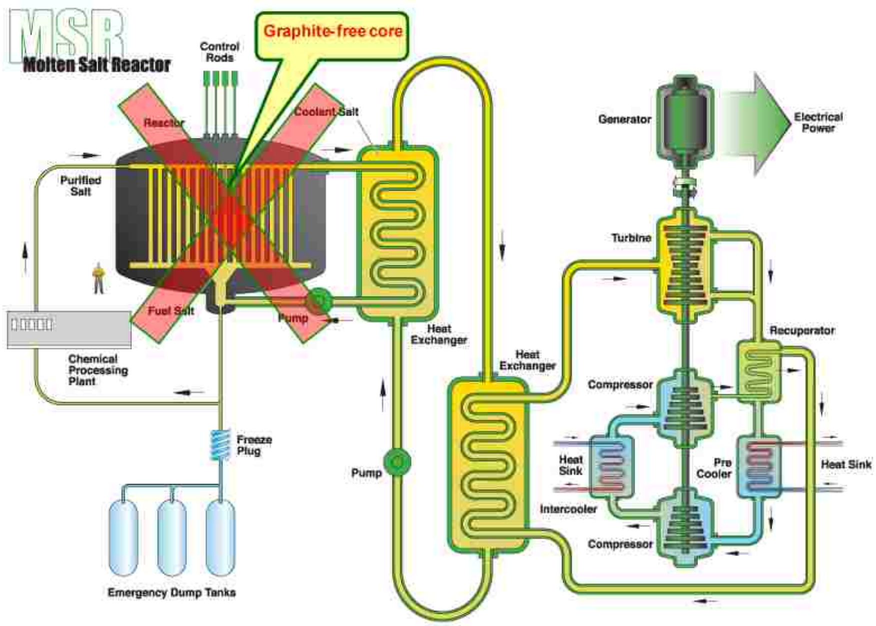


Figure 2-11: Molten Salt Reactor

CHAPTER 3 POWER CONVERSION AND TECHNOLOGY OPTIONS ASSESSMENT

Operating temperatures of conventional light water reactors, 280-320 ^{0}C limit power conversion systems to producing pressurized steam that drives a condensing steam turbine. After employing thermal recovery measures, nuclear plants using this Rankine cycle see a net plant efficiency of around 32% to 34%. Comparatively, gas turbines with turbine inlet temperatures of up to and greater than 1400 ^{0}C have simple cycle efficiencies of around 40% that can be boosted to around 60% in a combined cycle. The ability of the combined cycle or Brayton with recuperator cycle to drastically improve net plant efficiency is an especially appealing feature to employ with a nuclear power source, given the very low fuel costs for nuclear energy, but has previously been technically infeasible given the high operating temperature requirements of the combined cycle. See Figure 3-1 below.

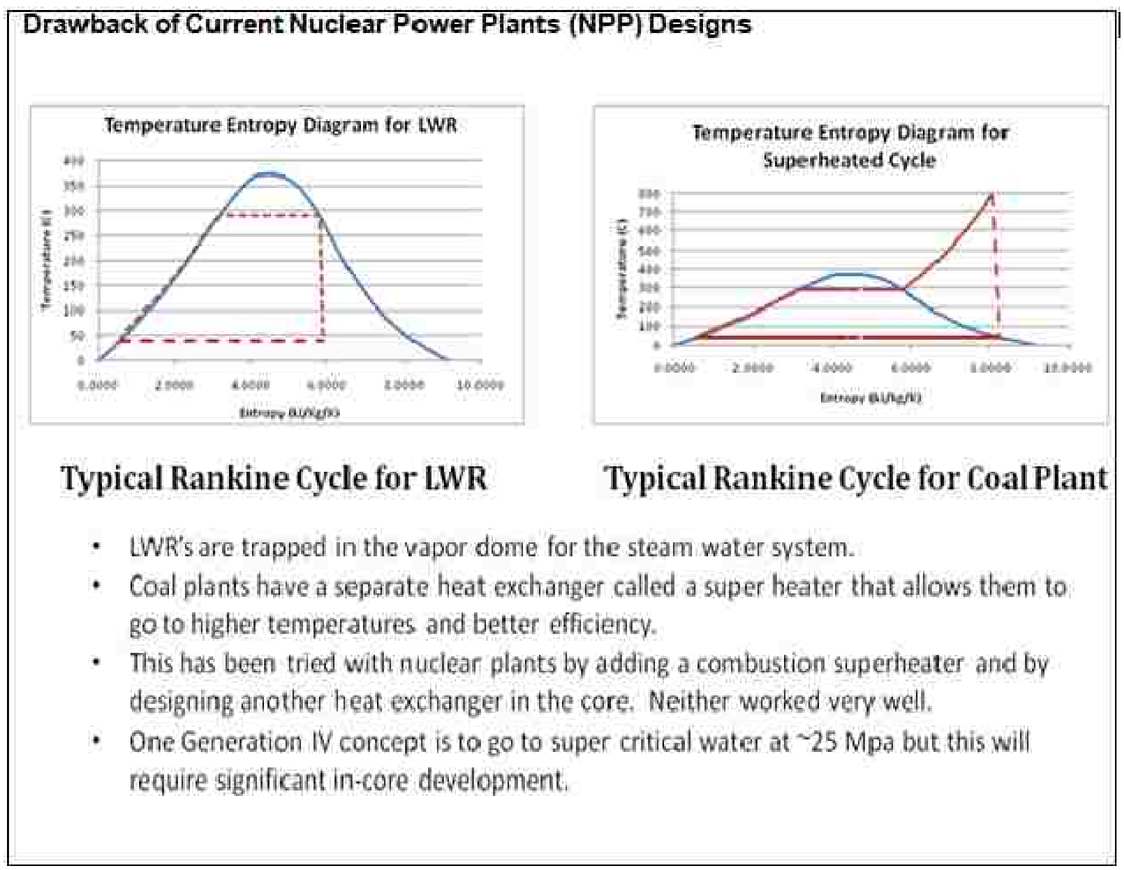


Figure 3-1: Drawback of Current Nuclear Power Plants (NPP) Designs

One of the differentiating features of the fluoride salt high temperature reactors is the operating temperature range of the primary coolant loop, 600 ^{0}C to 700 ^{0}C (reactor inlet and outlet temperatures, respectively). Although other advanced, high-temperature reactors have been developed, the high temperature characteristics of the lithium fluoride and beryllium fluoride eutectic (flibe) molten salt primary coolant used in fluoride salt high temperature reactors enables an operating temperature range that is uniquely suited to driving an open air combined cycle which sees proportional increases in efficiency and power generation with elevated turbine inlet temperatures.

A study was done in September of 2004 by a team of experts at the University of California, Nuclear Engineering Department [27]. The executive summary shows that the electrical Power Conversion System **(PCS)** for the Next Generation Nuclear Plant **(NGNP)** will take advantage of a significantly higher reactor outlet temperature to provide greater efficiency than can be achieved by the current generation of light water reactors. In anticipation of the design, development and procurement of an advanced power conversion system for NGNP, the study was initiated to identify the major design and technology options and their tradeoffs that must be considered in the evaluation of PCS options to support future research and procurement decisions. These PCS technology options affect cycle efficiency, capital cost, system reliability and maintainability and technical risk, and therefore the cost of electricity from Generation IV systems. A reliable evaluation and estimate of actual costs requires an optimized, integrated PCS design. At that early stage of the NGNP project it was useful to identify the technology options that would be considered in the design of proposed PCS systems, identify the system performance and cost implications of these design options, and provide a general framework for evaluating the design choices and technology tradeoffs.

The ultimate measure of the value of power conversion options is the cost of electricity produced, which is a function of capital and operating cost recovery and the system efficiency and reliability. Evaluating cost is difficult to do without detailed integrated designs, but it is possible to identify the factors that influence component and system performance, cost and technical risk. In this study, several existing Brayton conversion system designs were studied to illustrate and evaluate the implications of the major design choices to assess performance against the Generation IV economics and sustainability goals, and to identify areas of technical incompleteness or weakness. Several reference system designs were considered to provide a semi-quantitative basis for performing comparisons. The reference systems included the GT-MHR, PBMR, GTHTR-300, Framatome indirect cycle design, and AHTR high temperature Brayton cycle designs. Where appropriate, Generation II, III, and III+ light water reactors (two 1970's designs, the EPR, and the ESBWR) were also considered.

The design choices and technology options considered relevant for the assessment of NGNP power conversion options included the cycle types and operational conditions, such as working fluid choices, direct vs. indirect, system pressure and Interstage cooling and heating options. The cost and maintainability of the PCS is also influenced by the PCS layout and configuration including distributed vs. integrated PCS designs, single vs. multiple shafts, shaft orientation, and the implications for the pressure boundary.

From the summary table below, it is apparent that high temperature gas reactor power conversion design efforts to date have resulted in very different design choices based on project-specific requirements and performance or technical risk requirements.

Feature	PBMR (Horizontal)	GT-MHR	GTHTR300	Framatome Indirect	AHTR-IT
Thermal Power (MWt)	400	600	600	600	2400
Direct vs. Indirect Cycle	Direct	Direct	Direct	Indirect	Indirect
Recuperated vs. Combined Cycle	Recuperated	Recuperated	Recuperated	Combined	Recuperated
Intercooled vs. Non-Intercooled	Intercooled	Intercooled	Non-Intercooled	Intercooled	Intercooled/ Reheat
Integrated vs. Distributed PCS	Distributed	Integrated	Distributed	Distributed	Distributed (modular)
Single vs. Multiple TM Shafts	Single (previously Multiple)	Single	Single	Single	Multiple (modular)
Synchronous vs. Asynchronous	Reduction to synchronous	Asynchro-nous	Synchronous	Synchronous	Synchronous
Vertical vs. Horizontal TM	Horizontal	Vertical	Horizontal	Horizontal	Vertical
Submerged vs. External Generator	External	Submerged	Submerged	External	Submerged

Table 3-1: Summary of PCS Design Features for Representative Gas Reactor Systems

In the review of existing designs and the evaluation of the major technology options, it immediately becomes apparent that the optimized design involves a complex tradeoff of diverse factors, such as cost, efficiency, development time, maintainability, and technology growth path that must be considered in an integrated PCS system context before final evaluation. General observations derived from the review of the reference systems, including comparisons with light water reactor systems where applicable, include:

- There are key PCS design choices that can have large effects on PCS power density and nuclear island size, making careful and detailed analysis of design tradeoffs important in the comparison of PCS options.
- Considering the major construction inputs for nuclear plants—steel and concrete—high temperature reactors appear to be able to break the economy-of-scale rules for LWRs, and achieve similar material-input performance at much smaller unit sizes.
- For HTR’s, a much larger fraction of total construction inputs go into the nuclear island. To compete economically with LWRs, HTRs must find approaches to reduce the relative costs for nuclear-grade components and structures.

PCS technology options also include variations on the cycle operating conditions and the cycle type that can have an important impact on performance and cost. These options include:

- Working fluid choice – He, N_2, CO_2 or combinations have been considered. Working fluid physical characteristics influence cycle efficiency and component design.
- System pressure. – higher pressures lead to moderate efficiency increases and smaller PCS components, but increase the pressure boundary cost—particularly for the reactor vessel—which introduces a component-design, and a system cost and performance tradeoff.
- Direct vs. Indirect – Indirect cycles involve an intermediate heat exchanger (IHX) and resulting efficiency reduction, and more complex control requirements, but facilitate maintenance.
- Interstage cooling (or heating) results in higher efficiency but greater complexity.

Some of the observations from this assessment of these factors include:

- Differences between He versus N_2 working fluids were not considered critical for turbomachinery design, because both involve similar differences from current combustion turbines, with the primary difference being in the heat exchanger size to compensate for the lower N_2 thermal conductivity.
- N_2 allows 3600-rpm compressor operation at thermal powers at and below 600 MW(t), while He compressors must operate at higher speeds requiring reduction gears, asynchronous generators, or multiple-shaft configurations. However, power up-rating to approximately 800 MW(t) would permit 3600-rpm He compressor operation, providing a potentially attractive commercialization approach. Turbo machinery tolerances for He systems do not appear to be a key issue.
- Direct /Indirect – Efficiency loss can be 2 to 4 %, depending on design, and the IHX becomes a critical component at high temperatures. Direct cycles have an extended nuclear grade pressure boundary. Maintainability is considered a key design issue for direct cycles.
- Interstage cooling, as well as bottoming cycles (Rankine), can result in significant efficiency improvements, but at a cost of complexity and lower temperature differences for heat rejection, affecting the potential for dry cooling and reduced environmental impact from heat rejection.

The PCS configuration and physical arrangement of the system components has important effects on the volume and material inputs into structures, on the pressure boundary volume and mass, on gas inventories and storage volume, on the uniformity of flow to heat exchangers, on pressure losses, and on maintainability. The major factors considered in this study included:

- Distributed vs. integrated PCS design approach - PCS components can be located inside a single pressure vessel (e.g. GT-MHR), or can be divided between multiple pressure vessels (e.g., PBMR, HTR-300). This is a major design choice, with important impacts in several areas of design and performance.
- Shaft orientation (vertical/horizontal) -- Orientation affects the compactness of the system, the optimal design of ducting between turbo machinery and heat exchangers. Vertical turbo machinery provides a reduced PCS footprint area and building volume, and can simplify the ducting arrangement to modular recuperator and intercooler heat exchangers.
- Single vs. multiple shafts – Single shafts may include flexible couplings or reduction gears. In multiple-shaft systems, turbo-compressors are separated from synchronous turbo-generators, allowing the compressors to operate at higher speed and reducing the number of compressor stages required. Multiple shafts and flexible couplings reduce the weight of the individual turbo machines that bearings must support.
- Pressure boundary design – The pressure vessels that contain the PCS typically have the largest mass of any PCS components, and provides a significant (~33%) contribution to the total PCS cost.

3.1 Power Conversion System Components

The effectiveness or efficiency of the major PCS components, primarily the heat exchangers and turbo machinery, is clearly a major factor in system cost and performance. Observations and implications derived from this study include:

3.2 Heat Exchanger Components

Heat Exchanger Components are defined and the required designs are summarized as follow:

- Heat exchangers designs have significant impacts on both the efficiency and cost of the PCS. For a given heat exchanger type, higher effectiveness must be balanced against the increased size or pressure drop implications. Using small passages increases heat exchanger surface area per unit volume, but those same small passages tend to reduce the heat transfer coefficient due to laminar flow. Higher pressures may be utilized to force those flows back into the turbulent region, but those higher pressures force construction of a more robust pressure boundary and increase pumping power.
- The recuperator effectiveness and total Heat Exchanger (HX) pressure drop is a significant impact on the cycle efficiency and there is significant leverage in optimizing the recuperator design for both high heat transfer effectiveness and minimum pressure drop. For modular recuperators, careful attention must be paid to the module configuration and duct design to obtain equal flow rates to each module.
- Material limitations may limit the operating temperatures for many components, including the reactor vessel and heat exchangers. But because of the large flexibility of the Brayton cycle, high efficiency systems can still be designed within these limitations. Fabrication techniques will probably differ between intermediate, pre and intercooler, and recuperator heat exchangers, because of their operating temperature ranges. It would appear that transients could be tolerated by most of these heat exchanger designs.

3.3 Turbo Machinery

Turbo machinery that is used in the new generation of nuclear power systems (GEN-IV) plays a significant role in commercial applications in order to produce the electricity of the future.

- First order estimates of key turbine and compressor design and performance characteristics can be made with low level analysis. For the reference systems, key turbo machinery design parameters, (speed, stages, stage diameters, blade heights, blade clearances) will be similar to current commercial gas turbine engines.

- At lower reactor thermal powers He compressors will require greater than 3600 rpm operation to achieve efficiency goals (800 MW(t) allows 3600 rpm operation).
- Maximum system temperatures in the reference designs are near the limit for uncooled turbines.
- For both direct and indirect designs, the seals, housing and bearing components will be fundamentally different than current gas turbines, requiring extensive development with the associated cost and risk.

These observations illustrate the complex interactions of the many design choices that will be considered in the NGNP PCS. It is clear that detailed and integrated design efforts must be performed on candidate designs before quantitative evaluations are possible. The assessment described in that study helped illuminate those critical design choices and the resulting implications for the cost and performance of the future NGNP PCS design.

3.4 Combined Cycle Power Plant

A combined cycle gas turbine power plant is essentially an electrical power plant in which a gas turbine and a steam turbine are used in combination to achieve greater efficiency than would be possible independently. The gas turbine drives an electrical generator while the gas turbine exhaust is used to produce steam in a heat exchanger, called a Heat Recovery Steam Generator (HRSG) to supply a steam turbine whose output provides the means to generate more electricity. If the steam were used for heat then the plant would be referred to as a cogeneration plant.

It is important first to distinguish between a closed cycle power plant (or heat engine) and an open cycle power plant. In a closed cycle, fluid passes continuously round a closed circuit, through a thermodynamic cycle in which heat is received from a source at higher temperature, and heat rejected to a sink at low temperature and work output is delivered usually to drive an electric generator.

A gas turbine power plant may simply operate on a closed circuit as shown in Figure 3-2 below.

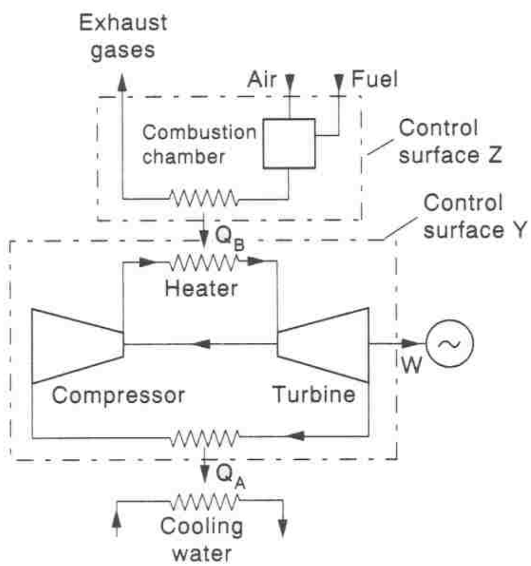


Figure 3-2: Closed Circuit Gas Turbine Plant

Most gas turbine plants operate in “open circuit”, with an internal combustion system as shown in Figure 3-3. Air fuel pass cross the single control surface into the compressor and combustion chamber respectively, and combustion products leave the control surface after expansion through the turbine.

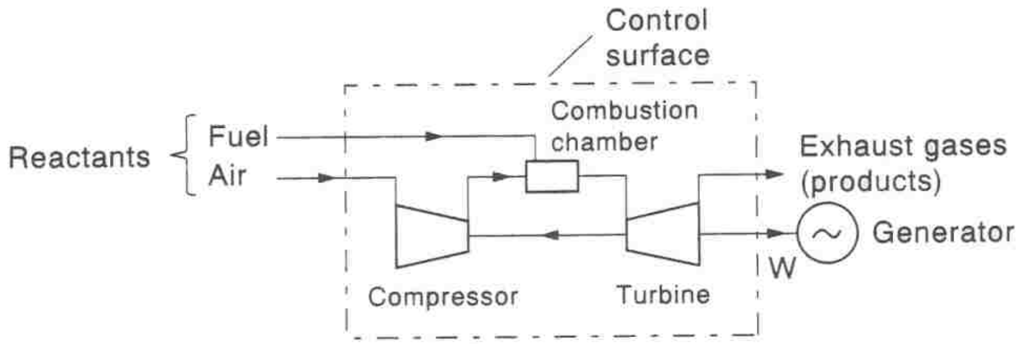


Figure 3-3: Open Circuit Gas Turbine Plant

The classical combined cycle for power production in a gas turbine and steam plant are normally associated with the names of Brayton and Rankine respectively.

Figure 3-4 below is simple representation of Combined Cycle Gas Turbine (CCGT) system. It demonstrates the fact that a CCGT system is two heat engines in series. The upper engine is the gas turbine. The gas turbine exhaust is the input to the lower engine (a steam turbine). The steam turbine exhausts heat to a circulating water system that cools the steam condenser.

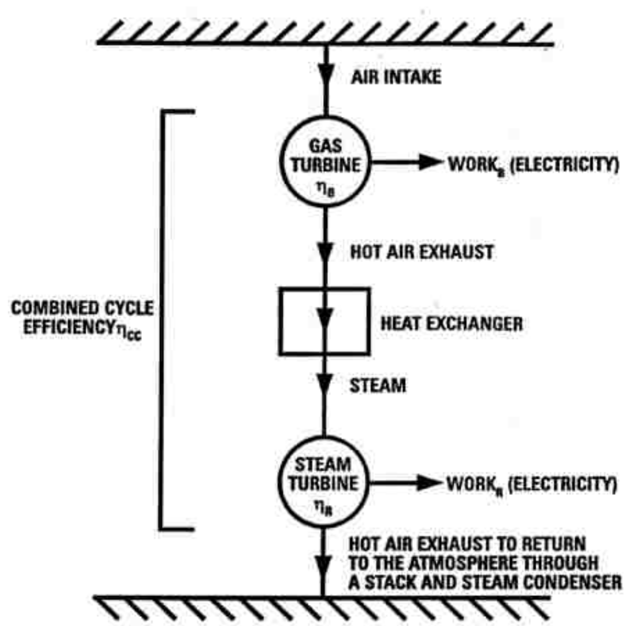


Figure 3-4: Schematic of Combined Cycle (CCGT) plant

An approximate combined cycle efficiency (η_{CC}) is given by the equation

$$\eta_{CC} = \eta_B + \eta_R - (\eta_B * \eta_R) \qquad \text{Eq. 3-1}$$

Equation 3-1 states that the sum of the individual efficiencies minus the product of the individual efficiencies equals the combine cycle efficiency. This simple equation gives significant insight to why combine cycle systems are successful.

For example, suppose the gas turbines efficiency η_B is 40% (a reasonable value for a today's gas turbines) and that the steam turbine efficiency η_R is 30% (a reasonable value for a Rankine Cycle steam turbine).

Utilizing Equation 3-1 would lead to the following conclusion.

$$\eta_{CC} = 0.4 + 0.3 - (0.4 * 0.3)$$

$$\eta_{CC} = 0.58$$

$$\eta_{CC} = 58\%$$

The combined cycle efficiency of 58% is much greater than either the gas turbine or the steam turbines efficiencies separately. The 58% value is slightly misleading in that system losses were ignored. However, efficiency values in the 60% range have been recorded for CCGT systems in the past few years [7].

CCGT power plants come in many different configurations. Some companies choose to treat the gas turbine exhaust bypass stack as a commodity; others choose to incorporate a diverter damper into the turbine exhaust gas path. The diverter damper allows for the rapid configuration of the power plant as a combined cycle or simple cycle system. The initial cost of the diverter damper is much higher than the cost of treating the gas turbine exhaust stack as a commodity. However, the diverter damper allows for the gas turbines to be operated in simple cycle when HRSG or steam turbine repair or maintenance is required.

3.5 Advanced Computational Materials Science Proposed for Gen IV Systems

A renewed interest in nuclear reactor technology has developed in recent years, in part as a result of international interest in sources of energy that do not produce CO_2 as a by-product. One result of this interest was the establishment of the Generation IV International Forum, which is a group of international governmental entities whose goal is facilitating bilateral and multilateral cooperation related to the development of new nuclear energy systems.

Historically, both the fusion and fission reactor programs have taken advantage of and built on research carried out by the other program. This leveraging can be expected to continue over the next ten years as both experimental and modeling activities in support of the Gen-IV program grow substantially. The Gen-IV research will augment the fusion studies (and vice versa) in areas where similar materials and exposure conditions are of interest. However, in addition to the concerns that are common to both fusion and advanced fission reactor programs, designers of a future DT fusion reactor have the unique problem of anticipating the effects of the 14 MeV neutron source term. For example, advances in computing hardware and software should permit improved (and in some cases the first) descriptions of relevant properties in alloys based on *ab initio* calculations. Such calculations could provide the basis for realistic inter-atomic potentials for alloys, including alloy-He potentials that can be applied in classical molecular dynamics simulations. These potentials must have a more detailed description of many-body interactions than accounted for in the current generations which are generally based on a simple embedding function. In addition, the potentials used under fusion reactor conditions (very high PKA energies) should account for the effects of local electronic excitation and electronic energy loss. The computational cost of using more complex potentials also requires the next generation of massively parallel computers. New results of *ab initio* and atomistic calculations can be coupled with ongoing advances in kinetic and phase field models to dramatically improve predictions of the non-equilibrium, radiation-induced evolution in alloys with unstable microstructures. This includes phase stability and the effects of helium on each micro-structural component.

However, for all its promise, computational materials science is still a house under construction. As such, the current reach of the science is limited. Theory and modeling can be used to develop understanding of known critical physical phenomena, and computer experiments can, and have been used to, identify new phenomena and mechanisms, and to aid in alloy design. However, it is questionable whether the science will be sufficiently mature in the foreseeable future to provide a rigorous scientific basis for predicting critical materials' properties, or for extrapolating well beyond the available validation database.

Two other issues remain even if the scientific questions appear to have been adequately answered. These are licensing and capital investment. Even a high degree of scientific confidence that a given alloy will perform as needed in a particular Gen-IV or fusion environment is not necessarily transferable to the reactor licensing or capital market regimes. The philosophy, codes, and standards employed for reactor licensing are properly conservative with respect to design data requirements.

Experience with the U.S. Nuclear Regulatory Commission suggests that only modeling results that are strongly supported by relevant, prototypical data will have an impact on the licensing process. In a similar way, it is expected that investment on the scale required to build a fusion power plant (several billion dollars) could only be obtained if a very high level of confidence existed that the plant would operate long and safely enough to return the investment.

These latter two concerns appear to dictate that an experimental facility capable of generating a sufficient, if limited, body of design data under essentially prototypic conditions (i.e. with ~14 MeV neutrons) will ultimately be required for the commercialization of fusion power. An aggressive theory and modeling effort will reduce the time and experimental investment required to develop the advanced materials that can perform in a DT fusion reactor environment. For example, the quantity of design data may be reduced to that required to confirm model predictions for key materials at critical exposure conditions. This will include some data at a substantial fraction of the anticipated end-of-life dose, which raises the issue of when such an experimental facility is required. Long lead times for construction of complex facilities, coupled with several years irradiation to reach the highest doses, imply that the decision to build any fusion-relevant irradiation facility must be made on the order of 10 years before the design data is needed.

Two related areas of research can be used as reference points for the expressed need to obtain experimental validation of model predictions. Among the lessons learned from ASCI, the importance of code validation and verification has been emphasized at the workshops among the courtiers involved with such research.

Because of the significant challenges associated with structural materials applications in these advanced nuclear energy systems, the *Workshop on Advanced Computational Materials Science: Application to Fusion and Generation IV Fission Reactors* was convened by the U.S. Department of Energy's Office of Science and the Office of Nuclear Energy, Science and Technology to ensure that research funded by these programs takes full advantage of ongoing advancements in computational science and the Department's investment in computational facilities. In particular, participants in the workshop were asked to:

1. Examine the role of high-end computing in the prediction of materials behavior under the full spectrum of radiation, temperature, and mechanical loading conditions anticipated for advanced structural materials that are required for future Generation IV fission and fusion reactor environments, and
2. Evaluate the potential for experimentally-validated computational modeling and simulation to bridge the gap between data that that is needed to support the design of these advanced nuclear technologies and both the available database and data that can be reasonably obtained in currently-available irradiation facilities.

Like the requirements for advanced fusion reactors, the need to develop materials capable of performing in the severe operating environments expected in Generation IV reactors represents a significant challenge in materials science. There is a range of potential Gen-IV fission reactor design concepts and each concept has its own unique demands. Improved economic performance is a major goal of the Gen-IV designs. As a result, most designs call for significantly higher operating temperatures than the current generation of LWRs to obtain higher thermal efficiency. In many cases, the desired operating temperatures rule out the use of the structural alloys employed today. The very high operating temperature (up to 1000 ^{0}C) associated with the NGNP is a prime example of an attractive new system that will require the development of new structural materials.

The operating temperatures, neutron exposure levels and thermo-mechanical stresses for proposed Gen-IV fission reactors are huge technological challenges among material scientists and engineers. In addition, the transmutation products created in the structural materials by the high energy neutrons produced in this generation of nuclear power reactors can profoundly influence the micro-structural evolution and mechanical behavior of these materials.

3.6 Material Classes Proposed for Gen IV Systems

The types of materials that were proposed in a DOE workshop in March of 2004 are tabulated as follows;

	STRUCTURAL MATERIALS						
System	FERRITIC-MARTENSITIC STAINLESS STEEL ALLOYS	AUSTENITIC STAINLESS STEEL ALLOYS	OXIDE DISPERSION STRENGTHENED STEELS	NI-BASED ALLOYS	GRAPHITE	REFRACTORY ALLOYS	CERAMICS
GFR	P	P	P	P		P	P
LFR	P	P	S			S	S
MSR				P	P	S	S
SFR	P	P	P				
SCWR-Thermal Spectrum	P	P	S	S			
SCWR-Fast Spectrum	P	P	S	S			
VHTR	S			P	P	S	P

P=Primary, S=Secondary

Table 3-2: Structural Materials

3.7 Generation IV Materials Challenges

A summary of these challenges for the next generation of nuclear power plants are presented here. They are;

- **Higher Temperature/Larger Temperature Ranges**
    - Examples

        - VHTR coolant outlet temperature near 1000 ^{0}C.
        - GFR transient temps to 1600-1800 ^{0}C, gradient across core of ~400 ^{0}C.
        - LFR to 800 ^{0}C steady-state outlet.
    - Issues
        - Creep.
        - Fatigue.
        - Toughness.
        - Corrosion/SCC.
- *Must drive modeling toward a predictive capability of materials properties in complex alloys across a wide temperature range*.

- **High Flounce dose**
    - Examples
        - LFR, SFR Cladding
        - SCWR Core Barrel
        - GFR Matrix
    - Issues
        - Swelling
        - Creep, stress relaxation
- *Must drive modeling toward a predictive capability of materials properties in complex alloys to large radiation dose.*

- **Unique Chemical Environments**
    - Examples
        - Pb and Pb-Bi Eutectic.

- Supercritical Water.
- High temperature oxidation in gas-cooled systems.
- Molten Salts.

o Issues

- Corrosion.
- SCC/IASCC.
- Liquid Metal Embrittlement.

- *Must drive modeling toward a predictive capability of chemical interactions in complex alloys to large radiation dose*.

3.8 Generation IV Materials Fundamental Issues

The co-evolution of all components of the microstructure, and their roles in the macroscopic response in terms of swelling, anisotropic growth, irradiation creep, and radiation-induced phase transformations should be studied within the science of complex systems.

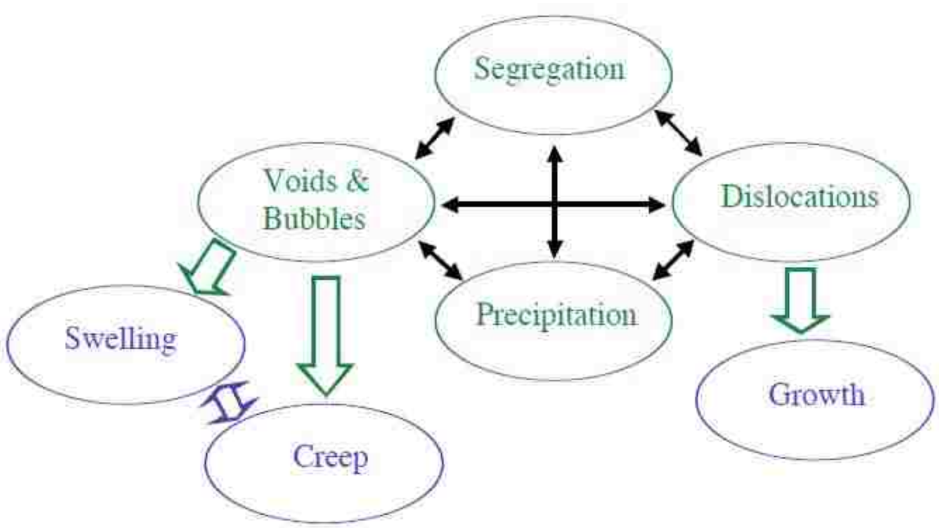


Figure 3-5: Flow Chart of Materials Fundamental Issues

In summary, we can conclude that;

- Six concepts have been identified with the potential to meet the Generation IV Goals
- Concepts operate in more challenging environments than current LWRs and significant material development challenges must be met for any of the Generation IV systems to be viable.
- Experimental programs cannot cover the breadth of materials and irradiation conditions for the proposed Gen IV reactor designs
- Modeling and micro-structural analysis can provide the basis for a material selection that is performed based on an incomplete experimental database and that requires considerable judgment to carry out the necessary interpolation and extrapolation

3.9 Capital Cost of Proposed Gen IV Reactors

Different PCS design trades may have substantial effects on the system capital cost. System optimization is typically complex, because, for example, increased PCS cost can increase cycle efficiency, reducing the reactor capital cost. The Generation IV Economic Modeling Working Group [EMWG, 2004] recommends two methodologies for modeling economics costs, a top-down method based on scaling and detailed information about similar systems, and a bottom-up method based on detailed accounting for all construction commodities, plant equipment, and labor-hours. For top-down methods, the EMWG recommends [EMWG, 2004]:

> The first task is to develop a reference design to which cost estimating techniques can be applied. The cost estimating part of this task generally is accomplished by considering the costs of equipment used for similar type projects and then scaling the equipment upwards or downwards. As an example, one might start cost estimating work on the Very High

Temperature Reactor (VHTR) by scaling reactor plant equipment from a project for which detailed estimates are available, such as the General Atomics HTGR.

For the purpose of system comparison, the top-down method was adopted to estimate PCS parameters that are important in scaling relative capital costs.

The measures selected were those typically calculated to provide input for system cost estimates, and thus provide a basis for rough comparisons of system options. To provide an approximate baseline for comparison, where possible, comparisons were made with Gen II and Gen III+ light water reactor values. Figure 3-6 below, shows such a comparison, quantifying steel and concrete inputs for the reference systems considered in the study. Several insights can be drawn from Figure 6-5. For example, the 1500-MWe passive ESBWR light-water reactor has slightly smaller inputs than the1970's light water reactors, as well as the evolutionary EPR.

But Figure 6-5 also shows that it is possible to build high-temperature gas-cooled reactors, e.g. the 286 MWe GT-MHR, with smaller construction material inputs than for light water reactors, due to the higher thermodynamic efficiency and power density. This shows that it is possible, with high-temperature gas power cycle technology, to break the economic scaling of the large light water reactors. This study also suggests that high-temperature; high efficiency gas-cycle power conversion can be adapted to other advanced reactor systems. For example, the even smaller inputs for the high-temperature, liquid-cooled, 1235-MWe AHTR-IT show that scaling economies may exist for high-temperature reactors. However, the material inputs for high-temperature reactors can be sensitive to equipment design choices and configurations, as shown by the differences in Figure 6-5 between the GT-MHR and the PBMR. Thus careful attention to design tradeoffs is clearly important in the design of power conversion systems.

The selected Capital Costs which have been calculated for the reference systems in the study and are presented by the report from the UC Berkeley team [27] in Chapter 3 in more detail are based on the volumes of materials used:

- Structures costs:
  - Building volume ($m^3/MW(e)_{ave}$) (nuclear/non-nuclear)
  - Concrete volume ($m^3/MW(e)_{ave}$) (nuclear/non-nuclear)
- Reactor and PCS cost:
  - Reactor power density ($m^3/MW(e)_{ave}$)
  - PCS power density ($m^3/MW(e)_{ave}$) (nuclear/non-nuclear)
  - System specific steel ($MT/MW(e)_{ave}$) (nuclear/non-nuclear)
  - Turbo machinery specific volume ($m^3/MW(e)_{ave}$)
  - System specific helium ($kg/MW(e)_{ave}$) (nuclear/non-nuclear) (nonrenewable resource, correlates with building volume (blow-down))

For each of these figures of merit, the values for the nuclear and non-nuclear portions of the plant were estimated. This division recognizes the difference in costs for procuring and installing nuclear-grade materials. For example, for concrete and reinforcing steel, material costs are estimated to be 65% greater for nuclear-grade materials, and installation costs 30% greater

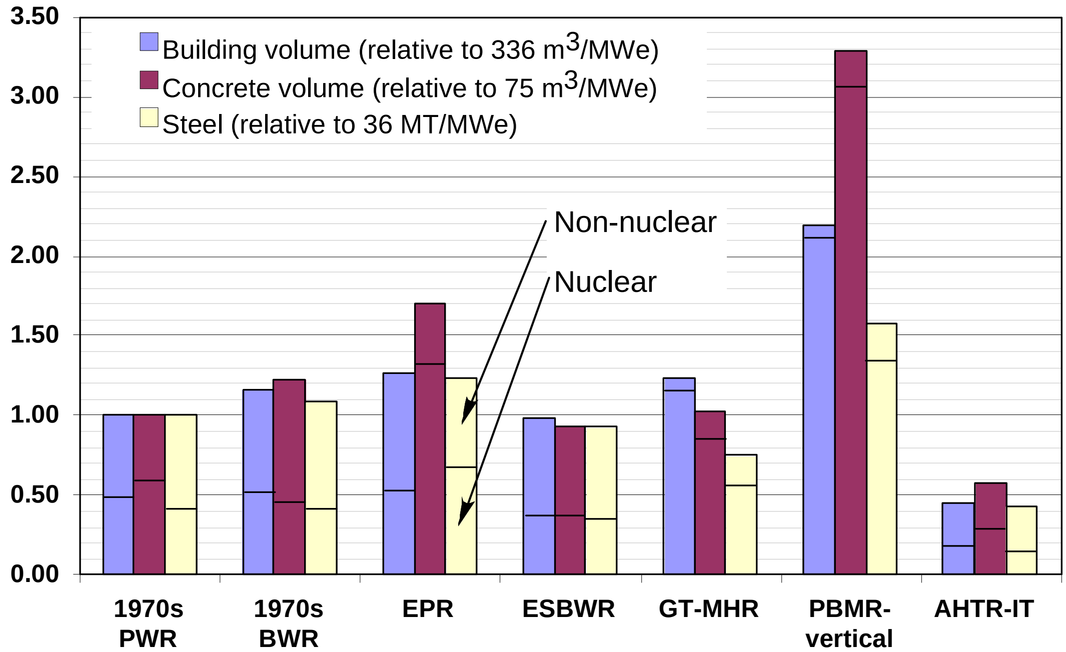


Figure 3-6: Comparison of the Total Building Volumes, and Total Plant Steel and Concrete inputs, for the reference HTR and LWR Systems Considered

3.9.1 Economic and Technical Consideration for Combined Cycle Performance

The output and efficiency of combined cycle plants can be increased during the design phase by selecting the following features[23]:

- Higher steam pressure and temperature
- Multiple steam pressure levels
- Reheat cycles

Additional factors are considered if there is a need for peak power production. They include gas turbine power augmentation by water or steam injection or a supplementary-fired Heat Recovery Steam Generator (HRSG). If peak power demands occur on hot summer days, gas turbine inlet evaporative cooling and chilling should be considered. Fuel heating is another technique that has been used to increase the efficiency of combined cycle plants.

The ability of combined cycle plants to generate additional power beyond their base capacity during peak periods has become an important design consideration. During the last decade, premiums were paid for power generated during the summer peak periods. The cost of electricity during the peak periods can be 70 times more expensive than off-peak periods. Since the cost during the peak periods is much higher, most of the plant’s profitability could be driven by the amount of power generated during these peak periods. Thus, plants that can generate large quantities of power during the peak periods can achieve the highest profits.

3.9.2 Economic Evaluation Technique

Plant output and efficiency are carefully considered during the initial plant design because they impact the cost of electricity in combination with fuel costs, plant capital cost, cost of capital and electricity sales. These factors will drive the gas turbine selection as well as the bottoming cycle design in combined-cycle operation.

As fuel costs increase, cycle selections typically include higher steam pressures, multiple steam pressure levels, reheat cycles and higher steam temperatures. Once these selections have been made, other factors are addressed. Is there a need for peak power production with premiums paid for the resulting power? If so, gas turbine power augmentation by way of water or steam injection or a supplementary fired heat recovery steam generator (HRSG) may be the solution. Do peak power demands occur on a hot day (summer peaking)? This may suggest a potential benefit from some form of gas turbine inlet evaporative cooling or chilling.[25]

For existing plants, some performance enhancement options can also be economically retrofitted to boost power output and efficiency. Although this research's primary focus is on options that enhance output, a brief discussion of fuel gas heating, which is a technique used to enhance combined-cycle plant efficiency, is provided.

The ability of utilities and independent power producers (IPPs) to generate additional power beyond a plant’s base capacity during summer peak power demand periods has become an important consideration in the design of combined- cycle plant configurations. In recent years, utilities and IPPs within the United States have received premiums for power generation capacity during summer peak power demand periods. The price of electricity varies greatly as a function of annual operating hours. The variation is also highly region dependent. With price-duration curves that are sharply peaked, implying a few hours annually with very high rates, the majority of a plant’s profitability could be driven by the high peak energy rates that can be achieved over a relatively short period of time. Thus, a plant that can economically dispatch a large quantity of additional power could realize the largest profits.

While current market trends should be considered during the design and development phase of a combined-cycle facility, forecasts of future market trends and expectations are equally important and warrant design considerations.

One of the primary challenges facing developers of new combined-cycle plants, as well as owner/operators of existing plants, is the optimization of plant revenue streams. As a result of escalating peak energy rates and peak demand duration, significant emphasis has been placed on developing plant designs that maximize peak power generation capacity while allowing for cost-effective, efficient operation of the plant during non-peak power demand periods.

In addition to maximizing plant profitability in the face of today’s marketplace, expectations of future market trends must be considered. Therefore the goal is to determine which performance-enhancement options or combination of options can be applied to a new or existing combined-cycle plant to maximize total plant profits on a plant life-cycle basis.

With very few exceptions, the addition of power-enhancement techniques to a base plant configuration will impact base load performance negatively and, hence, affect a plant's net revenue generating capability adversely during nonpeak periods.[25]

In general, efficiency is the predominating economic driver during non-peak generating periods, while capacity dominates the economic evaluation during peak power demand periods. Thus, it is extremely important to develop an economic model that considers both the cost of electricity (COE) during non-peak periods while taking into consideration expectations of peak energy rates.

After having established baseline peak and nonpeak period performance levels for the various power-enhancement alternatives, a COE analysis technique is applied to determine alternatives that would afford the best overall life-cycle benefit. In addition to including both peak and non-peak performance levels, the COE model includes the split between annual peak and non-peak operating hours, the premium paid for peak power generation capacity, the cost of fuel, plant capital cost, the incremental capital cost of the enhancements and the cost to operate and maintain the plant. This COE model then can be used to determine the sensitivity of a given power-enhancement alternative with respect to the economic parameters included within it.[25]

Most peak power enhancement opportunities exist in the topping cycle (gas turbine) as opposed to the bottoming cycle (HRSG/steam turbine). In general, with the exception of duct firing within the HRSG, there are few independent design enhancements that can be made to a bottoming cycle that has already been fully optimized to achieve maximum plant performance. However, in general, performance enhancements to the gas turbines will carry with them an increase in bottoming cycle performance due to an associated increase in gas turbine exhaust energy.[25]

3.9.3 Output Enhancement

The two major categories of plant output enhancements are;

1. Gas Turbine inlet air cooling and
2. Power augmentation

3.9.3.1 Gas Turbine Inlet Air Cooling

Industrial gas turbines operating at constant speed have a constant volumetric flow rate. Since the specific volume of air is directly proportional to temperature, cooler air has a higher mass flow rate. It generates more power in the turbine. Cooler air also requires less energy to be compressed to the same pressure as warmer air. Thus, gas turbines generate higher power output when the incoming air is cooler.[23]

A gas turbine inlet air cooling system is a good option for applications where electricity prices increase during the warm months. It increases the power output by decreasing the temperature of the incoming air. In combined cycle applications, it also results in improvement in thermal efficiency. A decrease in the inlet dry-bulb temperature by 10 ^{0}F (5.6 ^{0}C) will normally result in around a 2.7 percent power increase of a combined cycle using heavy duty gas turbines. The output of simple-cycle gas turbines is also increased by the same amount.

Figure 3-7 below, shows that a 10 ^{0}F (5.6 ^{0}C) reduction in gas turbine inlet dry-bulb temperature for heavy-duty gas turbines improves combined cycle output by about 2.7%. The actual change is somewhat dependent on the method of steam turbine condenser cooling being used. Simple cycle output is improved by a similar percentage.

Several methods are available for reducing gas turbine inlet temperature. There are two basic systems currently available for inlet cooling. The first and perhaps the most widely accepted system is evaporative cooling. Evaporative coolers make use of the evaporation of water to reduce the gas turbine's inlet air temperature. The second system employs various ways to chill the inlet air. In this system, the cooling medium (usually chilled water) flows through a heat exchanger located in the inlet duct to remove heat from the inlet air.

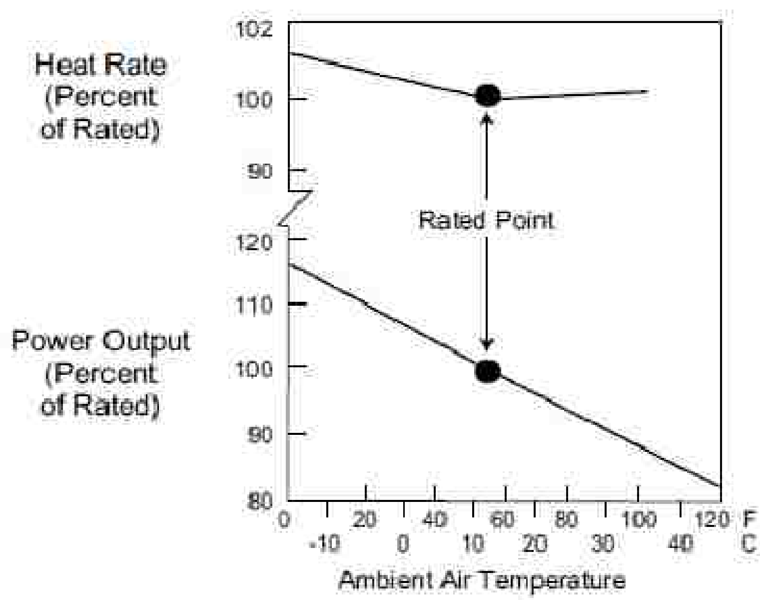


Figure 3-7: Combined-Cycle System Performance Variation with Ambient Air Temperature.[25]

Evaporative cooling is limited by wet-bulb temperature. Chilling, however, can cool the inlet air to temperatures that are lower than the wet bulb temperature, thus providing additional output albeit at a significantly higher cost.

Depending on the combustion and control system, evaporative cooling may reduce NOx emissions; however, there is very little benefit to be gained from current dry low NOx technology. This is another avenue that requires further analysis and investigation as well as collaboration between scientific communities, national laboratories and industries.

3.9.3.2 Power Augmentation

Three basic methods are available for power augmentation: water or steam injection, HRSG supplementary firing and peak firing.

1. Gas Turbine Steam/Water Injection
2. Supplementary Fired HRSG
3. Peak Firing

These are the three methods that General Electric is suggesting and they need to be investigated further by nuclear power manufacturers and the community involved with enhancing Nuclear Power Energy Efficiency [23] using combined cycle technology.

Others aspects of the cost of producing electricity are generally expressed in US$/MWh or US cts/kWh, depending on following parameters [22]:

- Capital cost of the project.
- Fuel cost
- Operation and maintenance cost

The capital cost per unit of electricity for a given power plant depends on following elements:

- Investment cost
- Financing structure
- Interest rate and return on equity
- Load factor of the plant (or equivalent utilization time)

The investment costs are the sum of the following positions:

- Power plant contract prices(s)
- Interest during construction (depending upon the construction time)
- Owner's cost for the realization of the project (project manager, owner's engineer, land cost, etc.)

The financing structure is defined by the debt-to-equity ratio of the financing and the return on equity is the return expected by the investors on their capital. Both are linked to the risks of the project.

The load factor results from the type of application the plant is intended for: Base, intermediate or peak load operation, and the availability and reliability of the power station.

Fuel costs per unit of electricity are proportional to the specific price of the fuel, and inversely proportional to the average electrical efficiency of the installation. This average electrical efficiency must not be mixed up with the electrical efficiency at rated load. It is defined as follows:

$$\bar{\eta} = \eta \cdot \eta_{\mathrm{Oper}}$$

Where:

η is the electrical net efficiency at rated load. (This is the % of the fuel that is converted into electricity at rated load for a new and clean condition)

η_{Oper} is the operating efficiency, which takes into account the following losses:

- Start-up and shutdown losses
- Higher fuel consumption for part load operation
- Aging and fouling of the plant

3.10 Gas Turbine Working Principle

Gas turbine engines derive their power from burning fuel in a combustion chamber and using the fast flowing combustion gases to drive a turbine in much the same way as the high pressure steam drives a steam turbine. Figure 3-8

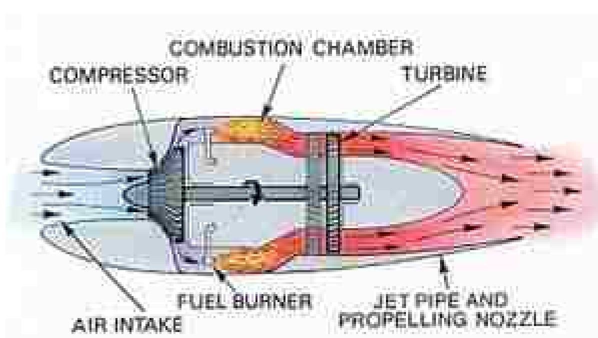


Figure 3-8: A Gas Turbine Power Plant

One major difference however is that the gas turbine has a second turbine acting as an air compressor mounted on the same shaft. The air turbine (compressor) draws in air, compresses it and feeds it at high pressure into the combustion chamber increasing the intensity of the burning flame.

It is a positive feedback mechanism. As the gas turbine speeds up, it also causes the compressor to speed up forcing more air through the combustion chamber which in turn increases the burn rate of the fuel sending more high pressure hot gases into the gas turbine increasing its speed even more. Uncontrolled runaway is prevented by controls on the fuel supply line which limit the amount of fuel fed to the turbine thus limiting its speed.

The thermodynamic process used by the gas turbine is known as the Brayton cycle. Analogous to the Carnot cycle in which the efficiency is maximized by increasing the temperature difference of the working fluid between the input and output of the machine, the Brayton cycle efficiency is maximized by increasing the pressure difference across the turbine The gas turbine is comprised of three main components: a compressor, a combustor, and a turbine. The working fluid, air, is compressed in the compressor (adiabatic compression - no heat gain or loss), then mixed with fuel and burned by the combustor under relatively constant pressure conditions in the combustion chamber (constant pressure heat addition). The resulting hot gas expands through the turbine to perform work (adiabatic expansion). Much of the power produced in the turbine is used to run the compressor and the rest is available to run auxiliary equipment and do useful work. The system is an open system because the air is not reused so that the fourth step in the cycle, cooling the working fluid, is omitted.

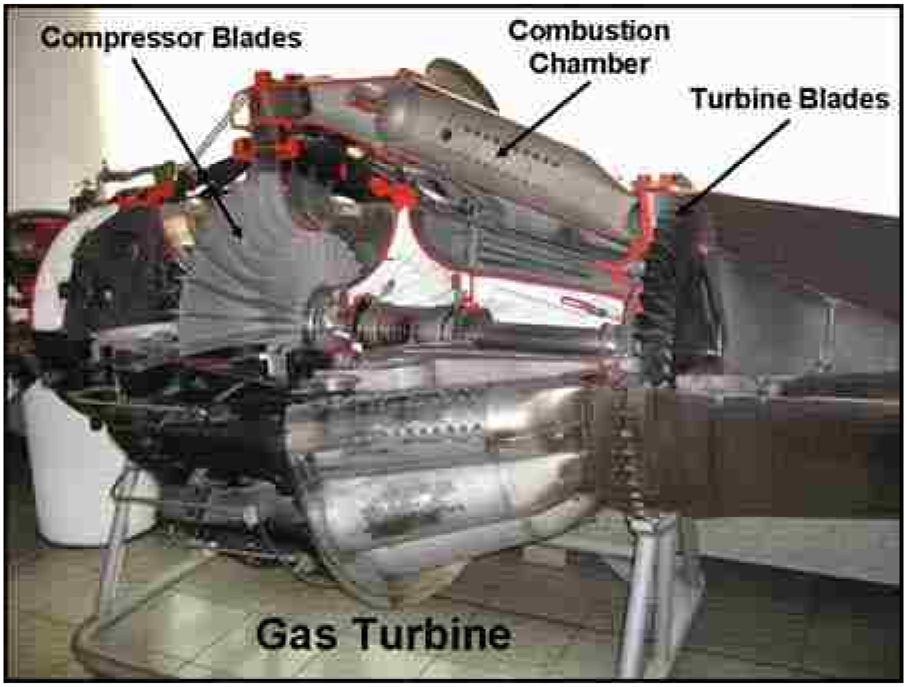


Figure 3-9: Gas Turbine Aero Engine

Gas turbines have a very high power to weight ratio and are lighter and smaller than internal combustion engines of the same power. Though they are mechanically simpler than reciprocating engines, their characteristics of high speed and high temperature operation require high precision components and exotic materials making them more expensive to manufacture. Figure 3-9.

One big advantage of gas turbines is their fuel flexibility. They can be adapted to use almost any flammable gas or light distillate petroleum products such as gasoline (petrol), diesel and kerosene (paraffin) which happen to be available locally, though natural gas is the most commonly used fuel. Crude and other heavy oils and can also be used to fuel gas turbines if they are first heated to reduce their viscosity to a level suitable for burning in the turbine combustion chambers.

Gas turbines can be used for large scale power generation. Examples are applications delivering 600 MW or more from a 400 MW gas turbine coupled to a 200 MW steam turbine in a co-generating installation. Such installations are not normally used for base load electricity generation, but for bringing power to remote sites such as oil and gas fields. They do however find use in the *major electricity grids* in *peak shaving* applications to provide emergency peak power.

Low power gas turbine generating sets with capacities up to 5 MW can be accommodated in transportation containers to provide mobile emergency electricity supplies which can delivered by truck to the point of need.

3.11 A Combined Cycle Power Conversion for the Generation IV Reactor Systems

A number of technologies are being investigated for the Next Generation Nuclear Plant that will produce heated fluids at significantly higher temperatures than current generation power plants. The higher temperatures offer the opportunity to significantly improve the thermodynamic efficiency of the energy conversion cycle. One of the concepts currently under study is the Molten Salt Reactor. The coolant from the Molten Salt Reactor may be available at temperatures as high as 800-1000 ^{0}C. At these temperatures, an open Brayton cycle combined with and Rankine bottoming cycle appears to have some strong advantages. Thermodynamic efficiencies approaching 50 % appear possible. Requirements for circulating cooling water will be significantly reduced. However, to realistically estimate the efficiencies achievable it is essential to have good models for the heat exchangers involved as well as the appropriate turbo-machinery. This study has concentrated on modeling all power conversion equipment from the fluid exiting the reactor to the energy releases to the environment.

Combined cycle power plants are currently commercially available. General Electric STAG^{TM} (Steam Turbine and Generator) systems have demonstrated high thermal efficiency, high reliability/availability and economic power generation for application in base load cyclic duty utility service.

Heat recovery type steam and gas turbine combined-cycle systems are the economic choice for gas or oil-fired power generation. Integration into nuclear power plants of the next generation is currently being studied and suggested by a team of universities including the University of New Mexico, Nuclear Engineering Department and this author, independent of others.

Incorporation with environmentally clean gasification system is extending their economic application to low cost solid fuel utilization. The features contributing to their outstanding generation economics are:

- High thermal efficiency
- Low installed cost
- Fuel flexibility – wide range of gas and liquid fuels
- Low operation and maintenance cost
- Operating flexibility – base, mid-range, daily start
- High reliability
- High availability
- Short installation time
- High efficiency in small capacity increments
- Minimum environment impact – low stack gas emissions and heat rejection

In electricity generating applications the turbine is used to drive a synchronous generator which provides the electrical power output but because the turbine normally operates at very high rotational speeds of 12,000 r.p.m or more it must be connected to the generator through a high ratio reduction gear since the generators run at speeds of 1,000 or 1,200 r.p.m. depending on the AC frequency of the electricity grid. Gas turbine power generators are used in two basic configurations

1. **Simple Systems:** This system consists of the gas turbine driving an electrical power generator. The following Figure 3-10 depicts such a configuration.

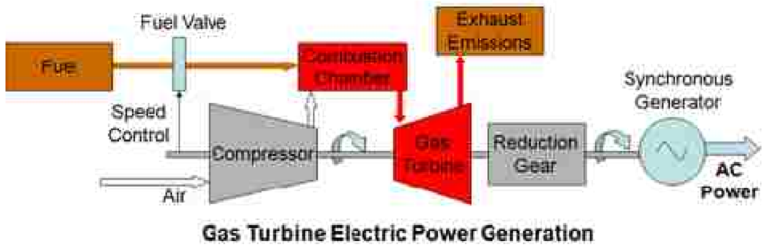


Figure 3-10: Simple Systems

2. **Combined Cycle Systems:** These systems are designed for maximum efficiency in which the hot exhaust gases from the gas turbine are used to raise steam to power a steam turbine with both turbines being connected to electricity generators. Figure 3-11.

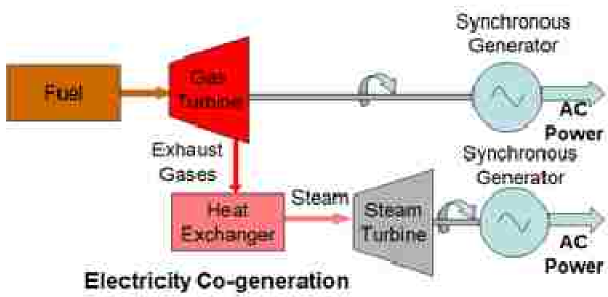


Figure 3-11: Combine Cycle Systems

In both cases as part of turbine performance and as turbine power output, we need to minimize the size and weight of the turbine for a given output power, the output per pound of airflow should be maximized. This is obtained by maximizing the air flow through the turbine which in turn depends on maximizing the compressor pressure ratio. The main factor governing this is the pressure ratio across the compressor which can be as high as 40:1 in modern gas turbines. In simple cycle applications, pressure ratio increases translate into efficiency gains at a given firing temperature, but there is a limit since increasing the pressure ratio means that more energy will be consumed by the compressor.

Some commercially available and installed combined cycles are presented below. Several of them that were looked at for purpose of benchmarking the code developed in this study are boxed. The particular one that was used to validate the CC code is identified as *S107FA* of General Electric.

Generation 3 GE Combined Cycle Systems

60 Hz. STAG Combined Cycle Experience with "F" Technology Gas Turbine

Country	Installation	Configuration	COD	Output (MW)
USA	Virginia Power #7	S107F	1990	214
USA	Virginia Power #8	S107F	1992	218
Korea	KEPCO Seo-Inchon #1 & #2	8 x S107F	1992	1887
USA	Sithe Independence	2 x S207FA	1995	1062
USA	Tampa Electric, Polk Co.	S107FA	1996	313
Korea	KEPCO Seo-Inchon #3 & #4	2 x S207FA	1996	1004
USA	US Gen. Co., Hermiston	2 x S107FA	1996	425
USA	Crockett Cogen	S107FA *	1996	202/248
Mexico	CFE Samalayuca	3 x S107FA *	1998	506
USA	Cogentrix, Clark Co.	S107FA *	1998	254
USA	Ft. St. Vrain	S207FA	1999	487
Korea	KEPCO, POSCO	S207FA	1999	498
Columbia	EPM LaSierra	S207FA	2001	478
USA	Bucksport Energy	S107FA	2001	176
USA	Westbrook	S207FA	2001	528
USA	Santee Cooper	S207FA	2001	600
Korea	Pusan	4 x S207FA	2003/4	2000

*Single-shaft Combined Cycle

Number of gas turbines = 45 Units

Installed Capacity = 12,411 MW

"Combined Cycle Development and Future", David L. Chase, GER-4206, 2001.

Table 3-3: Third Generation Combined Cycle Experience

3.12 System Efficiency

Thermal efficiency is important because it directly affects the fuel consumption and operating costs.

- **Simple Cycle Turbines**

  A gas turbine consumes considerable amounts of power just to drive its compressor. As with all cyclic heat engines, a higher maximum working temperature means greater efficiency (Carnot's Law), but in a turbine it also means that more energy is lost as waste heat through the hot exhaust gases whose temperatures are typically well over 500 ^{0}C. Consequently simple cycle turbine efficiencies are quite low. For heavy plant, design efficiencies range between 30% and 40%. (The efficiencies of aero engines are in the range 38% and 42% while low power micro-turbines (<100kW) achieve only 18% to 22%). Although increasing the firing temperature increases the output power at a given pressure ratio, there is also a sacrifice of efficiency due to the increase in losses due to the cooling air required to maintain the turbine components at reasonable working temperatures.

- **Combined Cycle Turbines**

  It is however possible to recover energy from the waste heat of simple cycle systems by using the exhaust gases in a combined cycle system to heat steam to drive a steam turbine electricity generating set. In such cases the exhaust temperature may be reduced to as low as 140 ^{0}C enabling efficiencies of up to 60% to be achieved in combined cycle systems. In combined-cycle applications, pressure ratio increases have a less pronounced effect on the efficiency since most of the improvement comes from increases in the Carnot thermal efficiency resulting from increases in the firing temperature.

Thus simple cycle efficiency is achieved with high pressure ratios. Combined cycle efficiency is obtained with more modest pressure ratios and greater firing temperatures.

3.13 Modeling the Brayton Cycle

Any external combustion or heat engine system is always at a disadvantage to an internal combustion system. The internal combustion systems used in current jet engine and gas turbine power systems can operate at very high temperatures in the fluid, and cool the structures containing the fluid to achieve high thermodynamic efficiencies. In an external energy generation system, like a reactor powered one, all of the components from the core to the heat exchangers heating the working fluid must operate at a higher temperature than the fluid. This severely limits the peak cycle temperature compared to an internal combustion system. This liability can be overcome to a certain extent by using multiple expansion turbines and designing highly efficient heat exchangers to heat the working fluid between expansion processes similar to reheaters in steam systems.

Typically the combustion chamber in a gas turbine involves a pressure drop of 3 to 5% of the total pressure. Efficient liquid salt to air heat exchangers can theoretically be designed with a pressure drop of less than 1%. This allows three to five expansion cycles to achieve a pressure drop comparable to a combustion system. Multiple turbines operating at different pressures have been common in steam power plants for a number of years. In this study one to five gas turbines operating on a common shaft were considered.

Multiple expansion turbines allow a larger fraction of the heat input to be provided near the peak temperature of the cycle. The exhaust from the last turbine is provided to the Heat Recovery Steam Generator (HRSG) to produce the steam used in the Rankine bottoming cycle. The hot air after it passes through the HRSG is exhausted to the atmosphere. A detailed comparison of this system was made with a recuperated stand alone Brayton cycle and the dual cycle appears to be more efficient for open systems.

3.14 Modeling the Rankine Cycle

The Rankine cycle was modeled with the standard set of components including the HSRG, a steam turbine, condenser and high-pressure pump. Multiple reheat processes were considered.

There is a slight efficiency advantage to include two reheat processes as per a fairly standard practice in today's power plants.

The major limitation on the size of the steam system is the enthalpy available from high temperature air above the pinch point where the high-pressure water working fluid starts to vaporize. Below this point, there is still a significant enthalpy in the air which is readily available to heat the high-pressure water. There does not appear to be an advantage to including feed water heaters in the cycle to bring the high pressure water up to the saturation point. The possibility that an intercooler could be inserted between the two stages of a split compressor was considered. The cooling fluid for the intercooler was the high-pressure water coming out of the water pump.

This process would combine the function of the traditional intercooler with the preheating of a typical feed water heater. The effect of this addition to the two cycles had a marginal effect on the overall system efficiency and likely is not worth the cost, or effort, to implement.

3.15 The Combined Brayton-Rankine Cycle

The combined-cycle unit combines the Rankine (Steam Turbine) and Brayton (Gas Turbine) thermodynamic cycles by using heat recovery boilers to capture the energy in the gas turbine exhaust gases for steam production to supply a steam turbine as shown in the Figure 3-12 "Combined-Cycle Cogeneration Unit". Process steam can be also provided for industrial purposes.

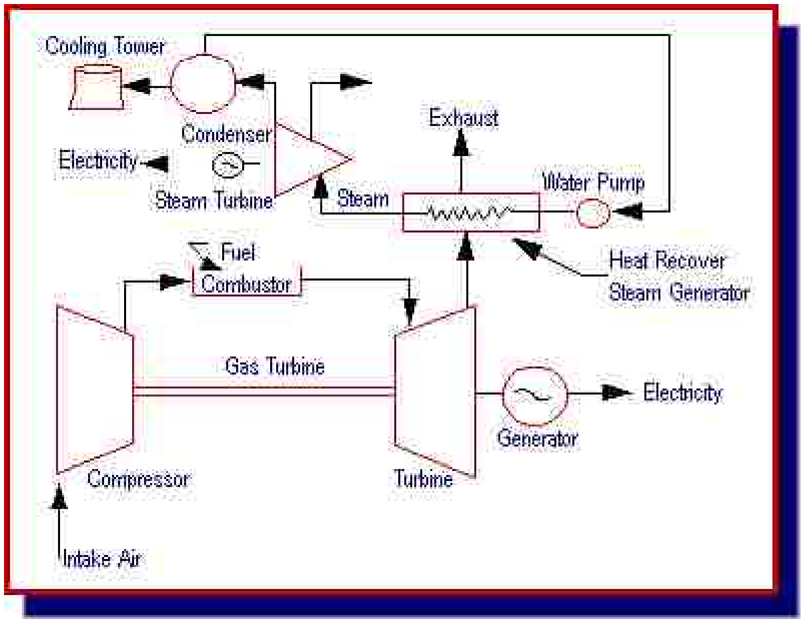


Figure 3-12: The Combined Cycle Brayton and Rankine Cycle Cogeneration Unit

Fossil fuel-fired (central) power plants use either steam or combustion turbines to provide the mechanical power to electrical generators. Pressurized high temperature steam or gas expands through various stages of a turbine, transferring energy to the rotating turbine blades. The turbine is mechanically coupled to a generator, which produces electricity.

The Brayton cycle efficiency is quite low primarily because a substantial amount of the energy input is exhausted to surroundings. This exhausted energy is usually at a relatively high temperature and thus it can be used effectively to produce power. One possible application is the combined Brayton Rankine cycle in which the high-temperature exhaust gases exiting the gas turbine are used to supply energy to the boiler of the Rankine cycle, as illustrated in Figure 3-12. Note that the temperature T_9 of the Brayton cycle gases exiting the boiler is less than the temperature T_3 of the Rankine cycle steam exiting the boiler; this is possible in the counter flow heat exchanger, the boiler.

3.16 Single and Multi-Shaft Design

The gas turbine can be designed in a single or multi-shaft configuration. In the single shaft case, the gas turbine is designed with roughly equal pressure ratios across all expansion stages which are mechanically coupled to the gas compressor and generator and operate at the generator speed (normally 3600 or 1800 rpm for 60-Hz electrical systems, and 3000 or 1500 rpm for 50-Hz electrical systems). In a multi-shaft configuration the compressor is mechanically driven by a set of expansion stages sized to produce the amount of mechanical work required by the compressor, so that this shaft is not connected to the electrical generator and can rotate at different speeds. The air produced from this gas-generator is heated and directed to a turbo-generator: a final expansion stage on a separate shaft that rotates at the optimal generator speed. **C**ombined **C**ycle **G**as **T**urbine (CCGT) Power Plant suppliers configure turbine generators in a number of different arrangements.

Multi-shaft and single-shaft configurations allow customization to optimize plant performance, capital investment, construction and maintenance access, operating convenience, and minimum space requirements.

The development of large F-class gas turbines during the past decade went hand in hand with manufacturers' efforts to standardize **C**ombined **C**ycle **P**ower **P**lant (CCPP) configurations, striving to best use the new technology. The **S**ingle-**S**haft **P**ower **T**rain (SSPT) arrangement was first conceived for applications using gas turbines over 250 megawatt. Only later the concept was extended to smaller units in the range of 60 megawatt. The new SSPT arrangement allowed single blocks of up to 450 megawatt to be built. SSPTs contributed the most to the power plants aiming at cost savings and project time reductions and thus at lower risk. In SSPT arrangements the gas turbine and the steam turbine are coupled to a common generator on a single shaft, whereas in Multi-Shaft Power Train blocks (MSPT) up to three gas turbines and their allocated boilers and generators share a common steam turbine (See Figure 3-13). SSPT and MSPT are both built for 50 and 60Hz markets. The main benefits of the new concept highlighted by manufacturers are higher operation flexibility, smaller footprint, simplified control, shorter run up time, more standardized peripheral systems and higher efficiency and availability. This development requires that, in addition to new technical issues related to the gas turbine design, insurers look at a great number of aspects when covering entire combined-cycle power plants.

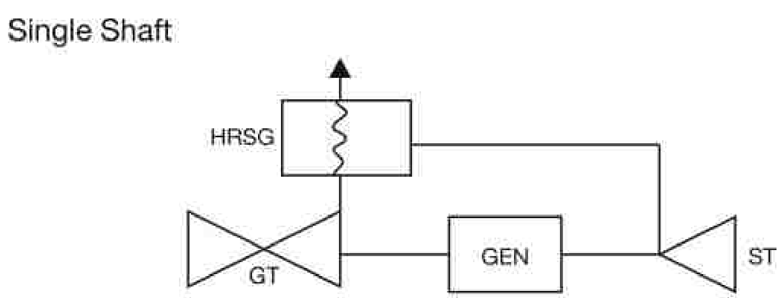


Figure 3-13: Combined Cycle Single Shaft Arrangements

In a multi-shaft combined cycle plant, there are generally several gas turbines with HRSGs generating steam for a single steam turbine. The steam and gas turbines use separate shafts, generators, set-up transformers, and so on. By combining the steam production of all the HRSGs, a large steam volume enters the steam turbine, which generally raises the steam turbine efficiency.

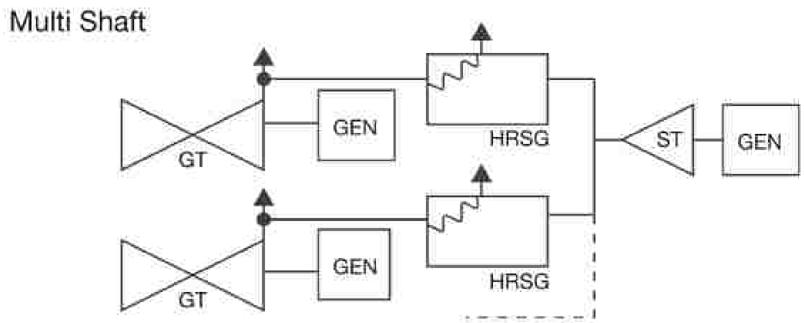


Figure 3-14: Combined Cycle Multi Shaft Arrangements

Modern gas turbines achieve higher output with higher exhaust temperatures. With the large gas turbines on the market, one steam turbine per gas turbine or one steam turbine for two gas turbines is common. Figure 3-14

If one steam turbine per gas turbine is installed, the single-shaft application is the most common solution—gas turbine and steam turbine driving the same generator.[22]

A plant with two gas turbines can be built either with two gas turbines on one steam turbine configuration (multi-shaft) or as a plant with two gas turbines, each in a single-shaft configuration. In either scenario usage of clutch as a synchronous self-shifting device, has a significant impact on components used in the respective combined cycle power plant (CCPP).

A combined cycle single shaft configuration with the generator between the two turbines enables installation of a clutch between steam turbine and generator. This means the clutch engages in that moment when the steam turbine speed tries to overrun the rigidly couple gas-turbine generator and disengages if the torque transmitted from the steam turbine to generator becomes zero.

The clutch allows startup and operation of the gas turbine without driving the steam turbine. This results in a lower starting power requirement and eliminates certain safety measures for the steam turbine (e.g., cooling steam or sealing steam).[22]

Furthermore, clutch implementation provides design opportunities for accommodating axial thermal expansion. The clutch itself compensates for a portion of axial displacement, and the two thrust bearings allow selective distribution of the remaining axial expansion (reducing tip clearance losses). In addition, it allows mores operational flexibility such as gas turbine simple cycle operation or early preventive maintenance (PM) activities on gas turbine during steam turbine cool down.

Generations of combined cycle power plant equipments that are manufactured by General Electric are also divided into two basic configurations;

1. **Single Shaft**
2. **Multiple Shaft**

The single –shaft combined cycle system consists of one gas turbine, one steam turbine, one generator and one **H**eat **R**ecovery **S**team **G**enerator **(HRSG)**, with the gas turbine and steam turbine coupled to a single generator in a tandem arrangement.[16]

Single shaft arrangements where a gas turbine and a steam turbine drive a single generator are often preferred because they offer a more compact plant at a lower cost. Many include an SSS (name of manufacture in United Kingdom) Clutch to disconnect the steam turbine and to allow the gas turbine/generator to be operated separately. Figure 3-15

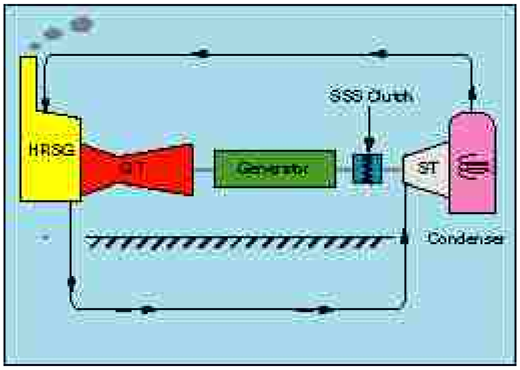


Figure 3-15: Single Shaft CCGT Arrangement using an SSS Clutch

Advantages of this arrangement are as follows:

A. Simple Startup

- Standard Gas Turbine Start (GT)
- Reduced time to generation
- No cooling steam required
- Reduced starting power
- Reduced emissions
- Standardized design
- Simplified torsion analysis

B. Increased Flexibility

- Simplified commissioning
- Steam turbine trips do not stop power generation
- Maintenance of Gas Turbine (GT) is possible during Steam Generation (ST) cooling

C. Optimized Shutdown

- Shutdown Steam Turbine (ST) at reduced Gas Turbine (GT) power

Multi-shaft combined-cycle systems have one or more gas turbine generators and HRSGs that supply steam through a common heater to a separate single steam turbine generator unit. Both configurations perform their specific functions, but the single shaft configuration excels in the base load and mid-range power generation applications.

The multi-shaft combined-cycle system configuration is most frequently applied in phased installations in which the gas turbines are installed and operated prior to the steam cycle installation and where it is desired to operate the gas turbines independent of the steam system. The multi-shaft configuration was applied most widely in the early history of heat recovery combined-cycles primarily because it was the least departure from the familiar conventional steam power plants. The single-shaft combined-cycle system has emerged as the preferred configuration for single phase applications in which the gas turbine and steam turbine installation and commercial operation are concurrent.

Multi-shaft systems have one or more gas turbine-generators and HRSGs that supply steam through a common header to a separate single steam turbine-generator. In terms of overall investment a multi-shaft system is about 5% higher in costs.

The ***primary disadvantage*** of multiple stage combined cycle power plant is the number of steam turbines, condensers and condensate systems, and perhaps the cooling towers and circulating water systems required by the bottoming cycle..

A Heat Recovery Steam Generator (HRSG) is a heat exchanger or series of heat exchangers that recovers heat from a hot gas stream and uses that heat to produce steam for driving steam turbines. or as process steam in industrial facilities, or as steam for district heating.

An HRSG is an important part of a Combined Cycle Power Plant (CCPP) or a cogeneration power plant.[10]. In both of those types of power plants, the HRSG uses the hot flue gas at approximately 500 to 650 °C from a gas turbine to produce high-pressure steam. The steam produced by an HRSG in a gas turbine combined cycle power plant is used solely for generating electrical power. However, the steam produced by an HRSG in a cogeneration power plant is used partially for generating electrical power and partially for district heating or for process steam.

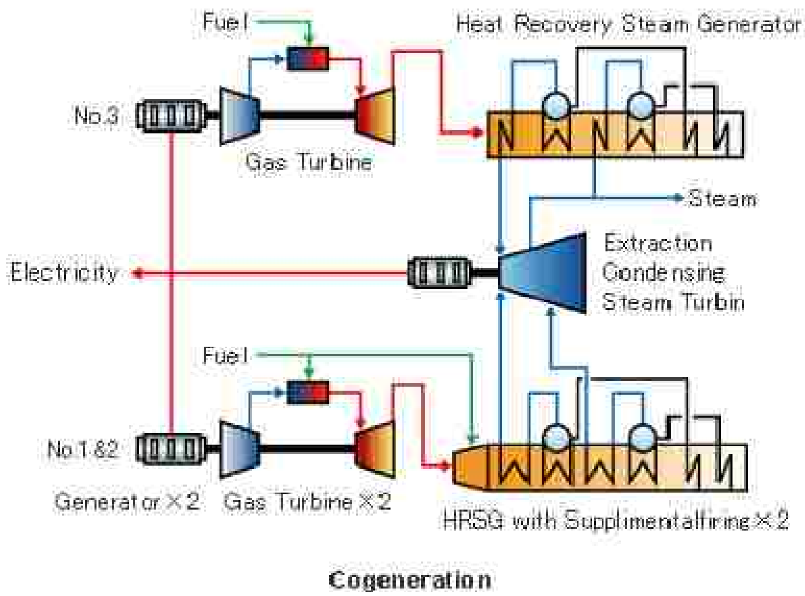


Figure 3-16: Combined Cycle Power Plant with Multi Shaft Configuration

The combined cycle power plant, schematically depicted in Figure 3-17 below, is so named because it combines the Brayton cycle for the gas turbine and the Rankine cycle for the steam turbines. About 60 percent of the overall electrical power generated in a CCPP is produced by an electrical generator driven by the gas turbine and about 40 percent is produced by another electrical generator driven by the high-pressure and low-pressure steam turbines. For large scale power plants, a typical CCPP might use sets consisting of a gas turbine driving a 400 MW electricity generator and steam turbines driving a 200 MW generator (for a total of 600 MW), and the power plant might have 2 or more such sets.

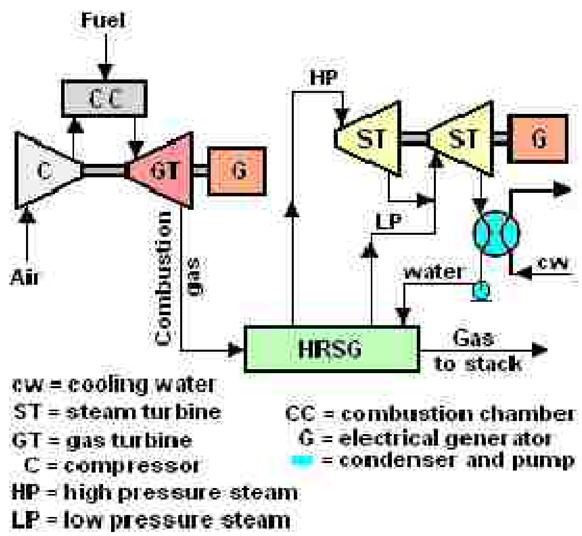


Figure 3-17: HRSG for Multi Shaft Combined Cycle Power Plant

The primary components of the heat exchangers in an HRSG are the economizer, the evaporator and its associated steam drum and the superheater as shown in Figure 3-18 below. An HRSG may be in horizontal ducting with the hot gas flowing horizontally across vertical tubes as in Figure 3-18 or it may be in vertical ducting with the hot gas flowing vertically across horizontal tubes. In either horizontal or vertical HRSGs, there may be a single evaporator and steam drum or there may be two or three evaporators and steam drums producing steam at two or three different pressures. Figure 3-18 depicts an HRSG using two evaporators and steam drums to produce high pressure steam and low pressure steam, with each evaporator and steam drum having an associated economizer and superheater. In some cases, supplementary fuel firing may be provided in an additional section at the front end of the HRSG to provide additional heat and higher temperature gas.

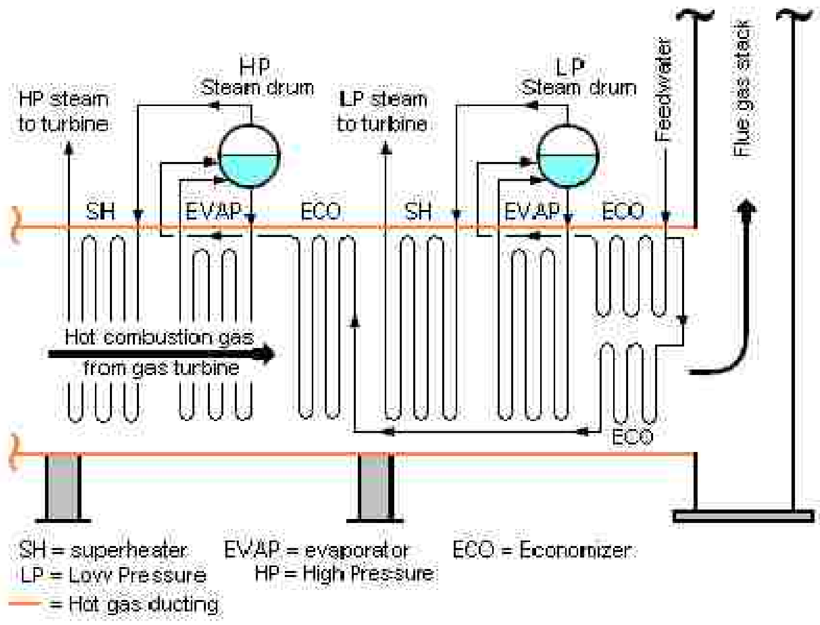


Figure 3-18: Heat Recovery Steam Generator (HRSG)

There are a number of other HRSG applications. For example, some gas turbines are designed to burn liquid fuels (rather than gas) such as petroleum naphtha or diesel oil[21] and others burn the syngas (synthetic gas) produced by coal gasification in an integrated gasification combined cycle plant commonly referred to as an IGCC plant. As another example, a combined cycle power plant may use a diesel engine rather than a gas turbine. In almost all such other applications, HSRGs are used to produce steam to be used for power generation.

3.17 Working Principle of Combined Cycle Gas Turbine (CCGT)

The first step is the same as the simple cycle gas turbine plant. An open cycle gas turbine has a compressor, a combustor and a turbine. For this type of cycle the input temperature to the turbine is very high. The output temperature of the flue gases is also very high. This is high enough to provide heat for a second cycle which uses steam as the working medium i.e. thermal power station.

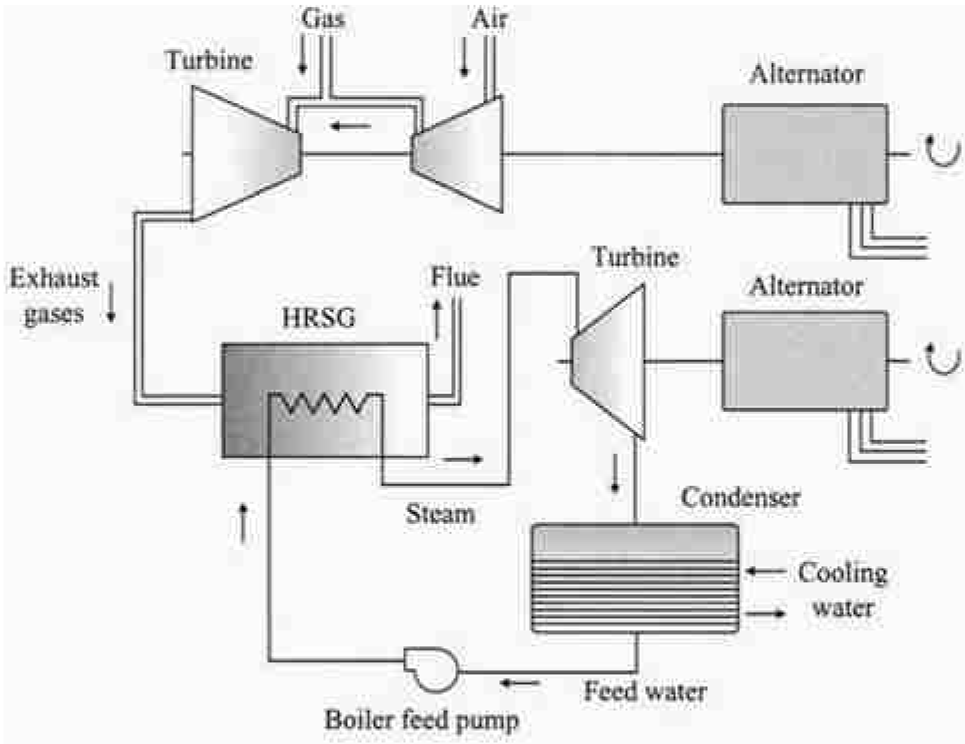


Figure 3-19: Working Principle of Combined Cycle Gas Turbine (CCGT) Plant

1. Air Inlet

This air is drawn though the large air inlet section where it is cleaned cooled and controlled. Heavy-duty gas turbines are able to operate successfully in a wide variety of climates and environments due to inlet air filtration systems that are specifically designed to suit the plant location.

Under normal conditions the inlet system has the capability to process the air by removing contaminants to levels below those that are harmful to the compressor and turbine.

In general the incoming air has various contaminants. They are:

<u>In Gaseous Sate Contaminants are:</u>

- ✓ Ammonia.
- ✓ Chlorine.
- ✓ Hydrocarbon gases.
- ✓ Sulfur in the form of H^2S, SO^2.
- ✓ Discharge from oil cooler vents.

<u>In Liquid State Contaminants are:</u>

- ✓ Chloride Salts dissolved in water (Sodium, Potassium).
- ✓ Nitrates.
- ✓ Sulfates.
- ✓ Hydrocarbons.

<u>In Solid State Contaminants are:</u>

- ✓ Sand, Alumina and Silica.
- ✓ Rust.
- ✓ Road Dust, Alumina and Silica.
- ✓ Calcium Sulfate.
- ✓ Ammonia compounds from fertilizer and animal feed operations.
- ✓ Vegetation, Airborne Seeds.

Corrosive Agents:

- ✓ Chlorides, nitrates and sulfates can deposit on compressor blades and may result in stress corrosion attack and/or cause corrosion Pitting. Sodium and potassium are alkali metals that can combine with Sulfur to form a highly corrosive agent and that will attack portions of the hot gas path. The contaminants are removed by passing through various types of filters which are present on the way.
- ✓ Gas phase contaminants such as ammonia or sulfur cannot be removed by filtration. Special methods are involved for this purpose.

2. Turbine Cycle

The air is purified then compressed and mixed with natural gas and ignited, which causes it to expand. The pressure created from the expansion spins the turbine blades, which are attached to a shaft and a generator, creating electricity.

In the second step, the heat of the gas turbine's exhaust is used to generate steam by passing it through a heat recovery steam generator (HRSG) with a live steam temperature between 420 and 580 ^{0}C.

3. Heat Recovery Steam Generator

In the Heat Recovery Steam Generator highly purified water flows in tubes and the hot gases pass around them to produce steam .The steam then rotates the steam turbine and coupled generator to produce electricity. The hot gases leave the HRSG at around 140 degrees centigrade and are discharged into the atmosphere.

The steam condensing and water pump systems are the same as in the steam power plant.

4. Typical Size and Configuration of CCGT

The combined-cycle system includes ***single-shaft*** and ***multi-shaft*** configurations. The single-shaft system consists of one gas turbine, one steam turbine, one generator and one Heat Recovery Steam Generator (HRSG), with the gas turbine and steam turbine coupled to the single generator on a single shaft.

Multi-shaft systems have one or more gas turbine-generators and HRSGs that supply steam through a common header to a separate single steam turbine-generator. In terms of overall investment a multi-shaft system is about 5% higher in costs.

5. Efficiency of CCGT Plant

Roughly the steam turbine cycle produces one third of the power and gas turbine cycle produces two thirds of the power output of the CCPP. By combining both gas and steam cycles, high input temperatures and low output temperatures can be achieved. The efficiency of the cycles adds, because they are powered by the same fuel source.

NOTE

To increase the power system efficiency, it is necessary to optimize the HRSG, which serves as the critical link between the gas turbine cycle and the steam turbine cycle with the objective of increasing the steam turbine output. HRSG performance has a large impact on the overall performance of the combined cycle power plant.

The electric efficiency of a combined cycle power station may be as high as 58 percent when operating new and at continuous output which are ideal conditions. As with single cycle thermal units, combined cycle units may also deliver low temperature heat energy for industrial processes, district heating and other uses. This is called cogeneration and such power plants are often referred to as a Combined Heat and Power (CHP) plant.

The efficiency of CCPT is increased by Supplementary Firing and Blade Cooling. Supplementary firing is arranged at HRSG and in gas turbine a part of the compressed air flow bypasses and is used to cool the turbine blades. It is necessary to use part of the exhaust energy through gas to gas recuperation. Recuperation can further increase the plant efficiency, especially when gas turbine is operated under partial load.

6. Fuels for CCPT Plants

The turbines used in Combined Cycle Plants are commonly fuelled with natural gas and it is more versatile than coal or oil and can be used in 90% of energy applications. Combined cycle plants are usually powered by natural gas, although fuel oil, synthesis gas or other fuels can be used.

7. Emission Control

Selective Catalytic Reduction (SCR):

- ✓ To control the emissions in the exhaust gas so that it remains within permitted levels as it enters the atmosphere, the exhaust gas passes though two catalysts located in the HRSG.
- ✓ One catalyst controls Carbon Monoxide (CO) emissions and the other catalyst controls Oxides of Nitrogen, (NOx) emissions. Aqueous Ammonia – In addition to the SCR, Aqueous Ammonia (a mixture of 22% ammonia and 78% water) is injected into system to even further reduce levels of NOx.

Advantages of Combined Cycle Power Plants are;

1. Fuel Efficiency

In conventional power plants turbines have a fuel conversion efficiency of 33% which means two thirds of the fuel burned to drive the turbine off. The turbines in combined cycle power plant have a fuel conversion efficiency of 50% or more, which means they burn about half amount of fuel as a conventional plant to generate same amount of electricity.

2. Low Capital Costs

The capital cost for building a combined cycle unit is two thirds the capital cost of a comparable coal plant.

3. Commercial Availability

Combined cycle units are commercially available from suppliers anywhere in the world. They are easily manufactured, shipped and transported.

4. Abundant Fuel Source

The turbines used in combined cycle plants are fuelled with natural gas, which is more versatile than a coal or oil and can be used in 90% of energy publications. To meet the energy demand now a day's plants are not only using natural gas but also using other alternatives like bio gas derived from agriculture.

5. Reduced Emission and Fuel Consumption

Combined cycle plants use less fuel per kWh and produce fewer emissions than conventional thermal power plants, thereby reducing the environmental damage caused by electricity production. Comparable with coal fired power plant burning of natural gas in CCPT is much cleaner.

6. Potential Applications in Developing Countries

The potential for combined cycle plant is with industries that requires electricity and heat or stem.

Disadvantages of Combined Cycle Power Plants are

1. The gas turbine can only use Natural gas or high grade oils like diesel fuel.

2. Because of this the combined cycle can be operated only in locations where these fuels are available and cost effective.

CHAPTER 4 OPEN AIR BRAYTON GAS POWER CYCLE

Power generation is an important issue today, especially on the West Coast. Demand outweighs supply because of the lack of incentives for the utility industry to build additional power plants over the past 10-20 years.

The major growth in the electricity production industry in the last 30 years has centered on the expansion of natural gas power plants based on gas turbine cycles. The most popular extension of the simple Brayton gas turbine has been the combined cycle power plant with the Air-Brayton cycle serving as the topping cycle and the Steam-Rankine cycle serving as the bottoming cycle. The Air-Brayton cycle is an open air cycle and the Steam-Rankine cycle is a closed cycle. The air-Brayton cycle for a natural gas driven power plant must be an open cycle, where the air is drawn in from the environment and exhausted with the products of combustion to the environment. The hot exhaust from the Air-Brayton cycle passes through a Heat Recovery Steam Generator (HSRG) prior to exhausting to the environment in a combined cycle. The HRSG serves the same purpose as a boiler for the conventional Steam-Rankine cycle.

In 2007 gas turbine combined cycle plants had a total capacity of 800 GW and represented 20% of the installed capacity worldwide. They have far exceeded the installed capacity of nuclear plants, though in the late 90's they had less than 5% of the installed capacity worldwide [22]. There are a number of reasons for this. First natural gas is abundant and cheap. Second combined cycle plants achieve the greatest efficiency of any thermal plant. And third, they require the least amount of waste heat cooling water of any thermal plant.

A typical gas turbine plant consists of a compressor, combustion chamber, turbine, and an electrical generator. A combined cycle plant takes the exhaust from the turbine and runs it through a Heat Recovery Steam Generator (HRSG) before exhausting to the local environment. The HRSG serves the function of the boiler for a typical closed cycle steam plant. The steam plant consists of a steam turbine, a condenser, a water pump, an evaporator (boiler), and an electrical generator. In a combined cycle plant the gas turbine and steam turbine can be on the same shaft to eliminate the need for one of the electrical generators. However the two shafts, two generator systems provide a great deal more flexibility at a slightly higher cost. In addition to the closed loop for the steam, an open loop circulating water system is required to extract the waste heat from the condenser. The waste heat extracted by this 'circulating' water system is significantly less per megawatt for a combined cycle system as the open Brayton cycle exhausts its waste heat directly to the air.

4.1 Computer Code Development

This effort was undertaken to investigate the possibility of using a nuclear reactor driven heat exchanger, or group of heat exchangers, to drive a Brayton-like cycle gas turbine as an open cycle power conversion system. Since in a nuclear reactor driven system, the core fuel elements of the reactor must exist at a higher temperature than any other component in the system, such a system is usually severely limited in the peak temperatures that can be produced in the gas turbine working fluid. For this study, a peak temperature of 660 ^{0}C (933 K) was chosen as a reasonable upper limit on the temperature that the working fluid could attain prior to being expanded through a turbine. This turbine inlet temperature is roughly half of the state of the art aircraft jet engine turbine inlet temperatures. Given this limitation, it is necessary to use several turbines with a reheat heat exchanger between each. The baseline for this study was four turbines operating with an inlet temperature of 660 ^{0}C, and an exit temperature of 537 ^{0}C. To demonstrate that as many as four turbines may be required, the analysis started with a system using only one turbine. The effects of a bottoming steam cycle and a recuperator were investigated.

GE currently markets a system that will produce 61% efficiency at design power and better than 60% efficiency down to 87% of design power [28] for gas turbine combined cycle plants.

An approximate efficiency can be calculated for a combined cycle power plant by the following simple argument [29].

Brayton cycle efficiency = $\frac{W_B}{Q_{in}} = \eta_B$

Heat to Rankine cycle = $Q_R = (1-\eta_B)Q_{in}$

Rankine cycle efficiency = $\frac{W_R}{Q_R} = \eta_R$

Overall efficiency = $\frac{W_B + W_R}{Q_{in}} = \eta_T = \frac{\eta_B Q_{in} + \eta_R Q_R}{Q_{in}} = \frac{\eta_B Q_{in} + \eta_R (1-\eta_B) Q_{in}}{Q_{in}} = \eta_B + \eta_R - \eta_B \eta_R$

$$\eta_T = \eta_B + \eta_R - \eta_B \eta_R$$

This efficiency has to be corrected for pressure losses and assumes that all of the heat in the Brayton exhaust is used in the HSRG. For a combustion gas turbine this is not usually possible if condensation of the water in the exhaust products is to be avoided. The detailed models developed in this effort give a more accurate answer.

For a nuclear system to take advantage of combined cycle technology, there are a number of changes to the plant components that have to be made. The most significant of course is that the combustion chamber has to be replaced by a heat exchanger in which the working fluid from the nuclear reactor secondary loop is used to heat the air. The normal Brayton cycle is an internal combustion one where the working fluid is heated by the combustion of the fuel with the air in the combustion chamber. The walls of the combustion chamber can be cooled and peak temperatures in the working fluid can be significantly above the temperature that the walls of the chamber can tolerate for any length of time.

For the nuclear reactor system the heat transfer is in the opposite direction. All reactor components and fluids in the primary and secondary loops must be at a higher temperature than the peak temperature of the gas exiting the heat exchanger. This severely restricts the peak temperature that can be achieved for the air entering the turbine. However all is not lost.

In a typical combustion system, there are pressure losses approaching 5% of the total pressure to complete the combustion process [30]. Heat exchangers can be built with significantly lower pressure drops than 5% approaching 1% [34]. So the most straightforward method to overcome this severe temperature limitation is to borrow a technique from steam power plants and implement multiple reheat cycles. That is the first heat exchanger heats the air to its peak temperature. Then the air is expanded through the first turbine. The air is then reheated to the same peak temperature and expanded through the second turbine. Based on the relative pressure losses that appear possible, up to five turbines might be considered. All five turbines will be driving the same compressor. Multiple compressors on concentric shafts [30] driven by different sets of turbines might be possible, but that has not been considered here.

Multiple reheat cycles allow more heat to be put into the working fluid at a higher temperature. This improves the efficiency of the overall cycle. Interestingly enough it does not improve the efficiency of the Brayton cycle, but because the exit temperature from the last turbine is higher, it does improve the efficiency of the Rankine part of the cycle with a net gain for the overall cycle. For this to work, the reactor coolant temperature must reach temperatures significantly higher than current Light Water Reactor temperatures. Even sodium cooled reactor exit temperatures in the 550^{o}C range are not quit high enough to get a nuclear Air Brayton combined cycle to compete with the efficiency of a pure Steam Rankine cycle. But when the coolant exit temperatures reach the 650^{o}C to 700^{o}C range, the combined cycle systems with multiple turbines surpass the performance of Steam Rankine systems. So the analysis that follows is targeted at a molten salt reactor, or a lead coolant reactor. A pressurized sodium reactor that could reach these temperatures is another possibility but no one is proposing such a system at this time. It could also apply to a High Temperature Gas Reactor but the heat exchangers would be quite different. Gas to gas heat exchangers have not been considered as the primary heat exchangers at this point, but will be addressed in the recuperated systems.

Liquid metal and molten salt heat exchangers were developed and tested successfully during the Aircraft Nuclear Propulsion program in the late 1950's. [33, 34, 35]. They were conventional tube and plate exchangers and were tested for over 1000 hours at temperatures up to 1100 K. The largest size tested transferred 55 MW of heat in a package of approximately 1.2 m^3. The heat transfer area on the air side had a surface area per unit volume of 1180 m^2/m^3. Certainly some development will be needed to bring this technology up modern standards and get NRC approval for a power producing reactor. But the tasks involved do not appear insurmountable. A number of additional heat exchangers were designed in this work in an attempt to estimate sizes of components and validate that pressure drop criteria could be met. The heat from the exhaust of the Brayton cycle transfers heat to vaporize the steam in the Rankine cycle in an HRSG of fairly conventional design. This includes air to steam superheaters as well as an economizer and evaporator section. A condenser of conventional design is included. For this work all heat exchangers were considered to be counterflow designs.

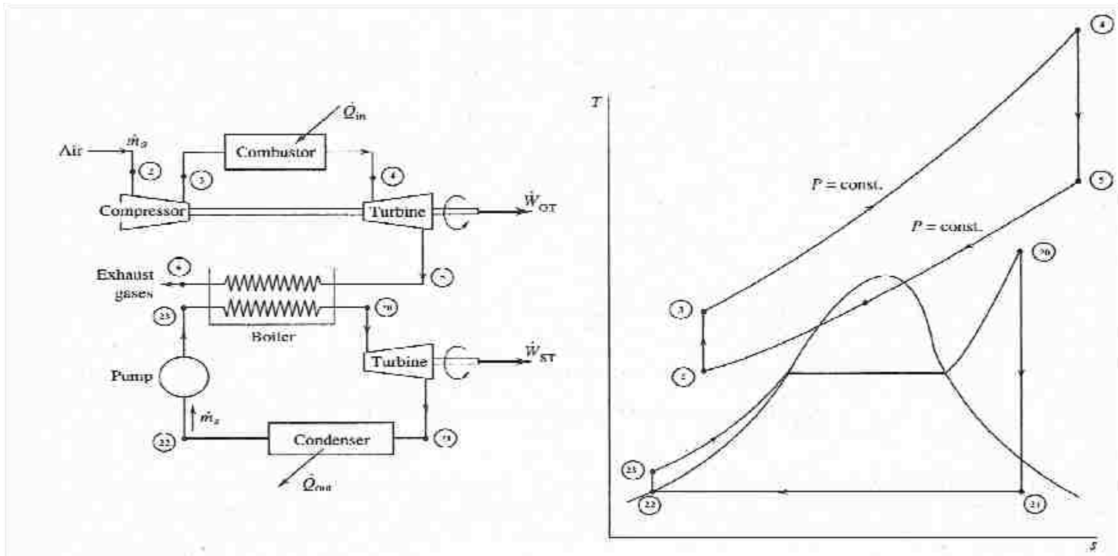
The heat exchanger design procedures and experimental data were taken from the text by Kays and London [36]. All of the data presented in this text was developed from steam to air heat exchangers and should be particularly applicable to the types of heat exchangers developed here. The only ones not using these two fluids are the molten salt or liquid metal to air heat exchangers referenced above, thus there is a reality basis for all of the design calculations performed here to estimate power conversion system performance and sizing.

One of the significant advantages of the combined cycle power system over current LWR power systems is its reduced requirement for circulating water in the waste heat rejection loop for the Rankine cycle. The typical combined cycle plant considered here produces approximately 50% of its power from the Rankine cycle and 50% of its power from the Brayton cycle. This automatically reduces the cooling water requirement by half. In addition the combined cycle plant achieves 45% efficiency so that only 55% of the heat generated has to be released as waste. So a typical 25 MW system will only need to get rid of 6.9 MW of heat via a circulating water system. A current LWR plant generating 25 MW at an efficiency of 33% would need to dump 16.8 MW of heat. This represents a major savings in circulating water requirements.

Since the combined cycle reduced the circulating water requirements so significantly the natural question arises as to whether they can be eliminated completely. With a recuperated Air Brayton cycle they can be. All of the waste heat can be rejected directly to the atmosphere. At first it was thought that the efficiency of a multi-turbine recuperated cycle could not compete with a combined cycle plant. However, after performing the detailed analysis, the efficiencies of a recuperated cycle come within one or two percent of predicted combined cycle efficiencies. This would seem to be a minor penalty to pay for being free of a circulating water requirement. However, achieving these high efficiencies requires a very effective recuperator which can become quite large.

4.2 System Description

Figure 4-1 below shows a simple bottoming Rankine and Brayton cycle layout along with its thermodynamic T-s Diagram



a **b**

Figure 4-1 (a, b): Simplified Brayton and Rankine Combined System Layout with Thermodynamic Cycle Plot

The systems of interest are focused on ***Open Cycle*** (Figure 4-3) mode where air is taken in from the atmosphere (point 1 in Figure 4-1a and 4-1b) and discharged back into the atmosphere (point 4). With the hot air being cooled naturally after it exits the turbine. In a ***Closed Cycle*** (Figure 4-3) gas turbine facility the working fluid (air or other gas) is continuously recycled by cooling the exhaust air (point 4 in Figure 4-1b) through a heat exchanger shown schematically in Figure 4-3 below and directing it back to the compressor inlet (point 1 Figure 4-1b). Because it is a confined, fixed amount of gas, the closed cycle gas turbine is not an internal combustion system. It is also good to note that, in the closed cycle system, combustion cannot be sustained and the normal combustor is replaced with a second heat exchanger to heat the compressed air before it enters the turbine. In this case, the heat is supplied by an external source such as a nuclear reactor, the fluidized bed of a coal combustion process, or some other heat source.

Closed cycle systems using gas turbines have been proposed for missions to Mars and other long term space applications.[24]

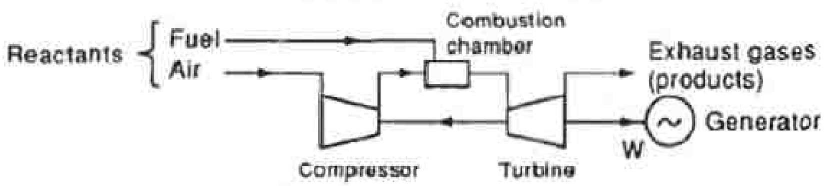


Figure 4-2: Open Cycle System for Brayton Cycle

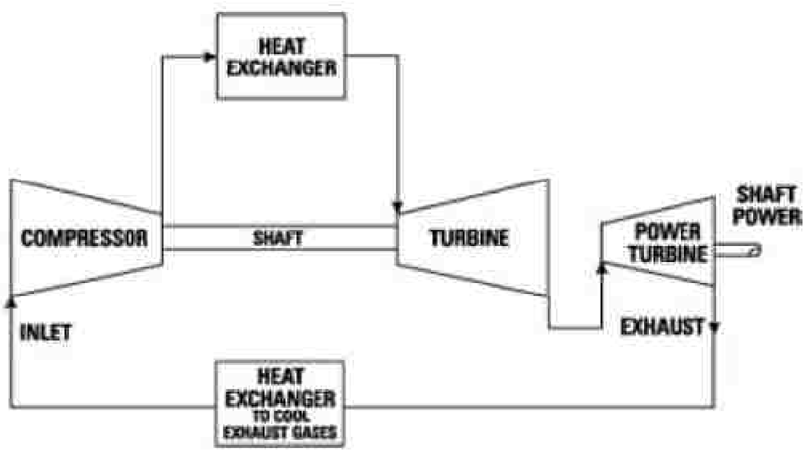


Figure 4-3: Closed Cycle System for Brayton Cycle

A gas turbine that is configured and operated to closely follow the Brayton cycle (Figure 4-2) is called a simple cycle gas turbine. Most aircraft gas turbines operate in a simple configuration since attention must be paid to engine weight and frontal area. However, in land or marine applications, additional equipment can be added to the simple cycle gas turbine, leading to increases in efficiency and/or the output of a unit. Three such modifications are regeneration, intercooling and reheating.

Regeneration involves the installation of a heat exchanger (recuperator) through which the turbine exhaust gases pass. The compressed air is then heated in the exhaust gas heat exchanger, before the flow enters the combustor.

If the regenerator is well designed (i.e., the heat exchanger effectiveness is high and the pressure drops are small) the efficiency will be significantly increased over the simple cycle value. However, the relatively high cost of such a regenerator must also be taken into account. Regenerators are being used in the gas turbine engines of the M1 Abrams main battle tank of Desert Storm fame, and in experimental gas turbine automobiles. Regenerated gas turbines increase efficiency 5-6% and are even more effective in improved part-load applications.[24]

Intercooling also involves the use of a heat exchanger. An intercooler is a heat exchanger that cools compressor gas during the compression process. For instance, if the compressor consists of a high and a low pressure unit, the intercooler could be mounted between them to cool the flow and decrease the work necessary for compression in the high pressure compressor. The cooling fluid could be atmospheric air or water (e.g., sea water in the case of a marine gas turbine). It can be shown that the output of a gas turbine is increased with a well-designed intercooler.

Reheating occurs in the turbine and is a way to increase turbine work without changing compressor work or melting the materials from which the turbine is constructed. If a gas turbine has a high pressure and a low pressure turbine at the back end of the machine, a reheater (usually another combustor) can be used to "reheat" the flow between the two turbines This can increase efficiency by 1-3%. Reheat in a jet engine is accomplished by adding an afterburner at the turbine exhaust, thereby increasing thrust, at the expense of a greatly increased fuel consumption rate.

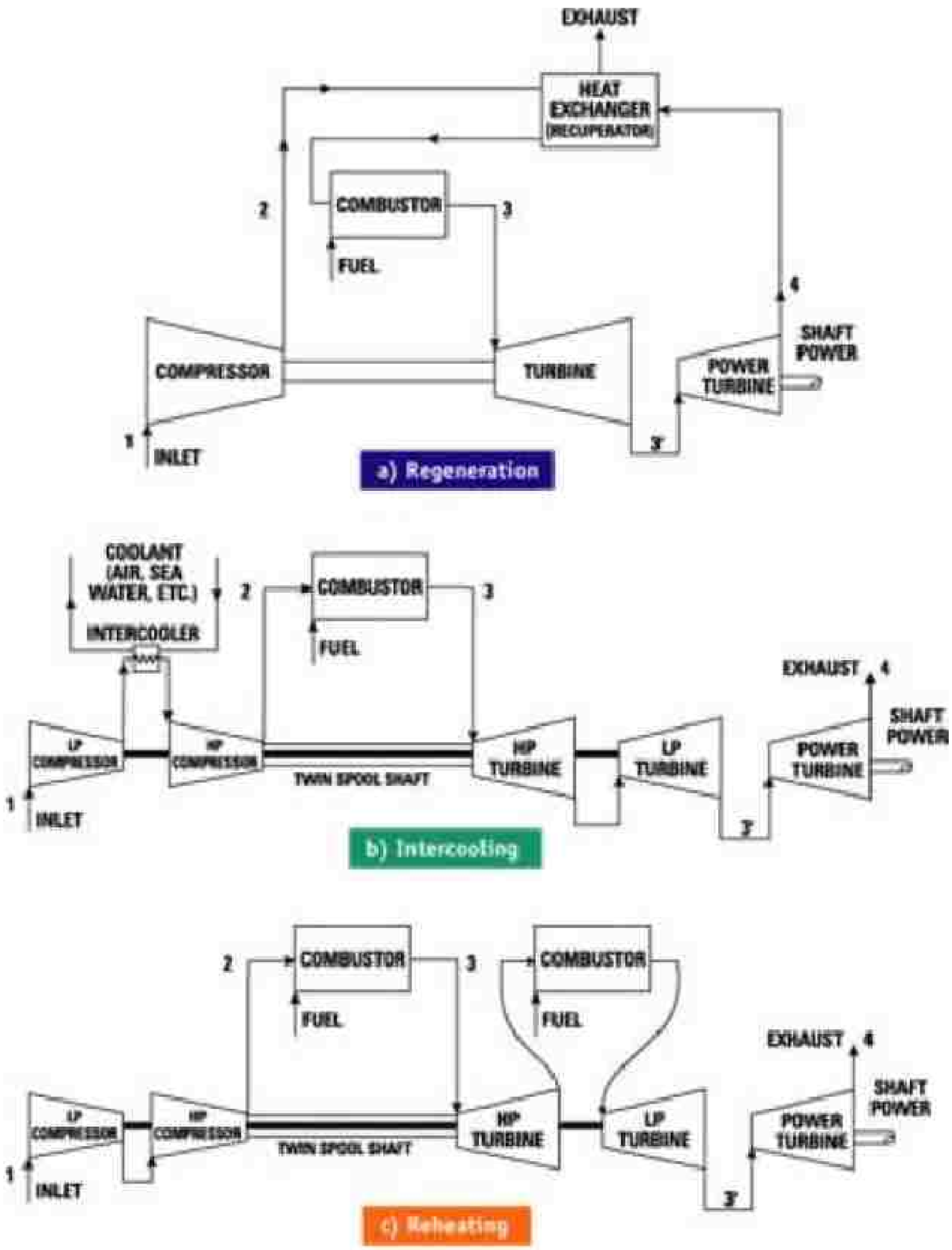


Figure 4-4: Modifications available for the simple Brayton Cycle.

A *combined cycle* gas turbine power plant, frequently identified by the abbreviation CCGT, is essentially an electrical power plant in which a gas turbine and a steam turbine are used in combination to achieve greater efficiency than would be possible independently. The gas turbine drives an electrical generator. The gas turbine exhaust is then used to produce steam in a heat exchanger (called a heat recovery steam generator) to supply a steam turbine whose output provides the means to generate more electricity. If the steam is used for heat (e.g. heating buildings), the unit would be called a cogeneration plant or a CHP (Combined Heat and Power) plant. Figure 4-5 is a simplified representation of a CCGT and shows it to be two heat engines coupled in series. The "upper" engine is the gas turbine. It expels heat as the input to the "lower" engine (the steam turbine). The steam turbine then rejects heat by means of a *steam condenser*.

In today technology of efficient power plant, an important field of study for power plants is that of the ‘combined plant’. A broad definition of the combined power plant (Figure 4-5) is one in which a higher (upper or topping) thermodynamic cycle produces power, but part or all of its heat rejection is used in supplying heat to a ‘lower’ or bottoming cycle. The ‘upper’ plant is frequently an open circuit gas turbine while the ‘lower’ plant is a closed circuit steam turbine; together they form a combined cycle gas turbine (CCGT) plant.

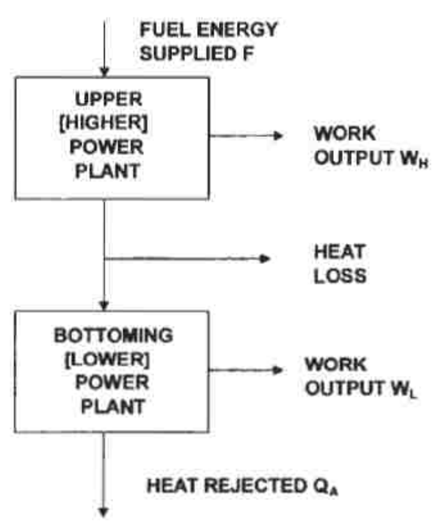


Figure 4-5: Combined Power Plant

The objective of combining two power plants in this way is to obtain greater work output for a given supply of heat or fuel energy. This is achieved by converting some of the heat rejected by the upper plant into extra work in the lower plant.

The term ‘cogeneration’ is sometimes used to describe a combined power plant, but it is better used for a Combined Heat and Power (CHP) plant such as the one shown in Figure 4-6 (see Ref. [26] for a detailed discussion on CHP plants).

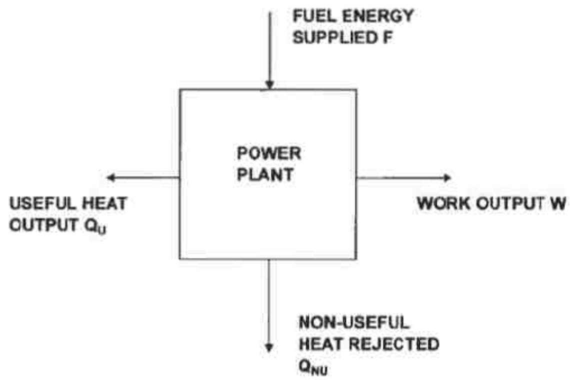


Figure 4-6: Cogeneration Power Plant

4.3 Combined-Cycle Features

The combination of the gas turbine Brayton Cycle and the steam power system Rankine Cycle complement each other to form efficient combined-cycles. The Brayton Cycle has high source temperature and rejects heat at a temperature that is conveniently used as the energy source for the Rankine Cycle. The most commonly used working fluids for combined cycles are air and steam. Other working fluids (organic fluids, potassium vapor, mercury vapor, and others) have been applied on a limited scale.

Combined-cycle systems that utilize steam and air-working fluids have achieved widespread commercial application due to:

- **a.** High thermal efficiency through application of two complementary thermodynamic cycles.
- **b.** Heat rejection from the Brayton Cycle (gas turbine) at a temperature that can be utilized in a simple and efficient manner.
- **c.** Working fluids (water and air) those are readily available, inexpensive, and nontoxic.

Integration of CCGS with environmentally clean gasification systems and their advantages were described in section 7-1 of this thesis.

4.4 Nature of Problem Solved

GE currently markets a system that will produce 61% efficiency at design power and better than 60% efficiency down to 87% of design power [28] for gas turbine combined cycle plants.

An approximate efficiency can be calculated for a combined cycle power plant by the following simple argument [29].

Brayton cycle efficiency = $\frac{W_B}{Q_{in}} = \eta_B$

Heat to Rankine cycle = $Q_R = (1-\eta_B)Q_{in}$

Rankine cycle efficiency = $\frac{W_R}{Q_R} = \eta_R$

Overall efficiency = $\frac{W_B + W_R}{Q_{in}} = \eta_T = \frac{\eta_B Q_{in} + \eta_R Q_R}{Q_{in}} = \frac{\eta_B Q_{in} + \eta_R (1-\eta_B) Q_{in}}{Q_{in}} = \eta_B + \eta_R - \eta_B \eta_R$

$$\eta_T = \eta_B + \eta_R - \eta_B \eta_R$$

This efficiency has to be corrected for pressure losses and assumes that all of the heat in the Brayton exhaust is used in the HSRG. For a combustion gas turbine this is not usually possible if condensation of the water in the exhaust products is to be avoided. The detailed models developed in this effort give a more accurate answer.

For a nuclear system to take advantage of combined cycle technology, there are a number of changes to the plant components that have to be made. The most significant of course is that the combustion chamber has to be replaced by a heat exchanger in which the working fluid from the nuclear reactor secondary loop is used to heat the air. The normal Brayton cycle is an internal combustion one where the working fluid is heated by the combustion of the fuel with the air in the combustion chamber. The walls of the combustion chamber can be cooled and peak temperatures in the working fluid can be significantly above the temperature that the walls of the chamber can tolerate for any length of time.

For the nuclear reactor system the heat transfer is in the opposite direction. All reactor components and fluids in the primary and secondary loops must be at a higher temperature than the peak temperature of the gas exiting the heat exchanger. This severely restricts the peak temperature that can be achieved for the air entering the turbine. However all is not lost.

In a typical combustion system, there are pressure losses approaching 5% of the total pressure to complete the combustion process [30]. Heat exchangers can be built with significantly lower pressure drops than 5% approaching 1% [31]. So the most straightforward method to overcome this severe temperature limitation is to borrow a technique from steam power plants and implement multiple reheat cycles. That is the first heat exchanger heats the air to its peak temperature. Then the air is expanded through the first turbine. The air is then reheated to the same peak temperature and expanded through the second turbine. Based on the relative pressure losses that appear possible, up to five turbines might be considered. All five turbines will be driving the same compressor. Multiple compressors on concentric shafts [32] driven by different sets of turbines might be possible, but that has not been considered here.

Multiple reheat cycles allow more heat to be put into the working fluid at a higher temperature. This improves the efficiency of the overall cycle. Interestingly enough it does not improve the efficiency of the Brayton cycle, but because the exit temperature from the last turbine is higher, it does improve the efficiency of the Rankine part of the cycle with a net gain for the overall cycle. For this to work, the reactor coolant temperature must reach temperatures significantly higher than current Light Water Reactor temperatures. Even sodium cooled reactor exit temperatures in the 550^{o}C range are not quit high enough to get a nuclear Air Brayton combined cycle to compete with the efficiency of a pure Steam Rankine cycle. But when the coolant exit temperatures reach the 650^{o}C to 700^{o}C range, the combined cycle systems with multiple turbines surpass the performance of Steam Rankine systems. So the analysis that follows is targeted at a molten salt reactor, or a lead coolant reactor. A pressurized sodium reactor that could reach these temperatures is another possibility but no one is proposing such a system at this time. It could also apply to a High Temperature Gas Reactor but the heat exchangers would be quite different. Gas to gas heat exchangers have not been considered as the primary heat exchangers at this point, but will be addressed in the recuperated systems.

Liquid metal and molten salt heat exchangers were developed and tested successfully during the Aircraft Nuclear Propulsion program in the late 1950's [33-35]. They were conventional tube and plate exchangers and were tested for over 1000 hours at temperatures up to 1100 K. The largest size tested transferred 55 MW of heat in a package of approximately 1.2 m^3. The heat transfer area on the air side had a surface area per unit volume of 1180 m^2/m^3. Certainly some development will be needed to bring this technology up modern standards and get NRC approval for a power producing reactor. But the tasks involved do not appear insurmountable. A number of additional heat exchangers were designed in this work in an attempt to estimate sizes of components and validate that pressure drop criteria could be met. The heat from the exhaust of the Brayton cycle transfers heat to vaporize the steam in the Rankine cycle in an HRSG of fairly conventional design. This includes air to steam superheaters as well as an economizer and evaporator section. A condenser of conventional design is included. For this work all heat exchangers were considered to be counterflow designs.

The heat exchanger design procedures and experimental data were taken from the text by Kays and London [36]. All of the data presented in this text was developed from steam to air heat exchangers and should be particularly applicable to the types of heat exchangers developed here. The only ones not using these two fluids are the molten salt or liquid metal to air heat exchangers referenced above, thus there is a reality basis for all of the design calculations performed here to estimate power conversion system performance and sizing.

One of the significant advantages of the combined cycle power system over current LWR power systems is its reduced requirement for circulating water in the waste heat rejection loop for the Rankine cycle. The typical combined cycle plant considered here produces approximately 50% of its power from the Rankine cycle and 50% of its power from the Brayton cycle. This automatically reduces the cooling water requirement by half. In addition the combined cycle plant achieves 45% efficiency so that only 55% of the heat generated has to be released as waste. So a typical 25 MW system will only need to get rid of 6.9 MW of heat via a circulating water system. A current LWR plant generating 25 MW at an efficiency of 33% would need to dump 16.8 MW of heat. This represents a major savings in circulating water requirements.

Since the combined cycle reduced the circulating water requirements so significantly the natural question arises as to whether they can be eliminated completely. With a recuperated Air Brayton cycle they can be. All of the waste heat can be rejected directly to the atmosphere. At first it was thought that the efficiency of a multi-turbine recuperated cycle could not compete with a combined cycle plant. However, after performing the detailed analysis, the efficiencies of a recuperated cycle come within one or two percent of predicted combined cycle efficiencies. This would seem to be a minor penalty to pay for being free of a circulating water requirement. However, achieving these high efficiencies requires a very effective recuperator which can become quite large.

4.5 Typical Cycles

The following Figures 4-7 and 4-8 provide a schematic of a four turbine combined cycle system and its thermodynamic cycles on a Temperature-Entropy plot. This turns out to be the near optimum Combined Cycle system.

Figure 4-7: Layout for Four Turbines Combined Cycle

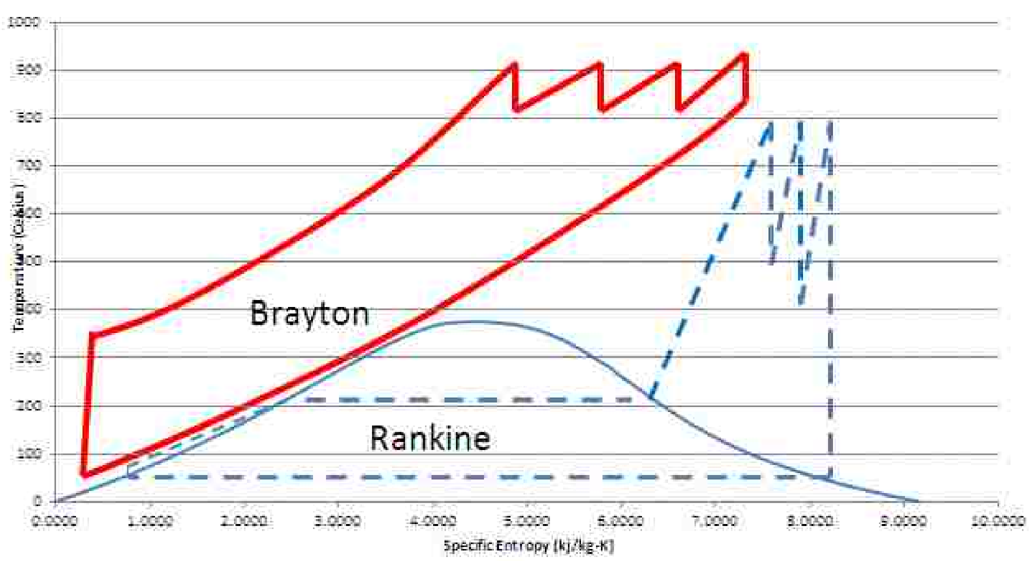


Figure 4-8: Temperature Entropy Diagram for Combined Cycle

The following Figures 4-9 and 4-10 provide a schematic layout and a Temperature Entropy diagram for a three turbine Recuperated system. This turns out to be the optimum Recuperated system

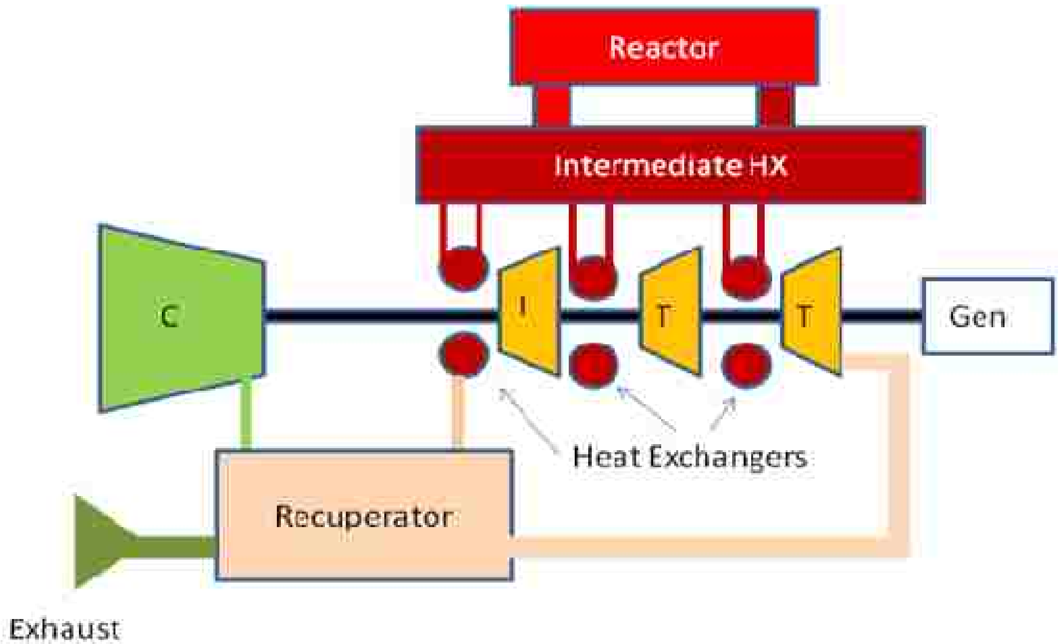


Figure 4-9: Recuperated System Layout Schematic

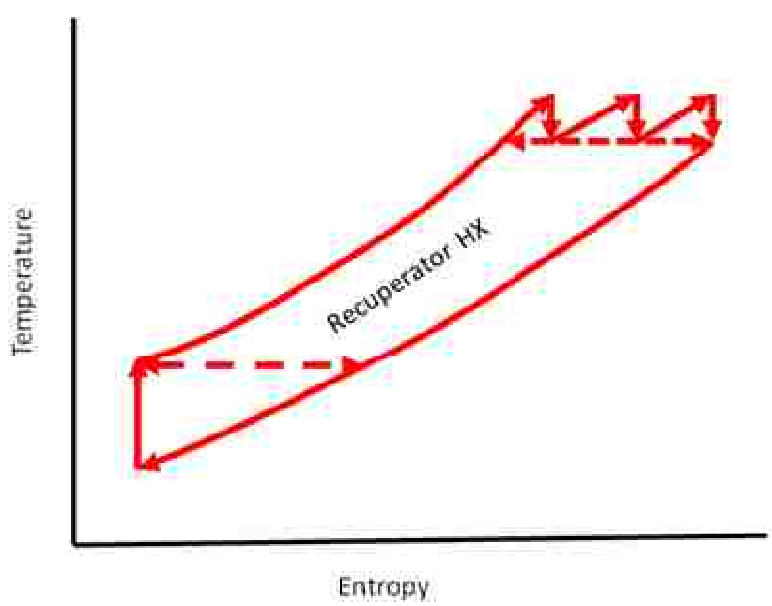


Figure 4-10: Recuperated System Temperature Entropy Diagram

4.6 Methodology

The approach taken in the Combined Cycle (CC) code developed for this effort is to model the thermodynamics of the components making up the power conversion systems as real components with non-ideal efficiencies. Pressure drops are included for every component except the connected piping. The compressor design is modeled with a small stage polytropic efficiency to take into account state of the art designs [37]. The gas turbines are likewise modeled with polytropic efficiency. The steam turbines are modeled with a simple overall thermal efficiency. Pressure drops in each of the heat exchangers are included. The input files specify the pressure drops and the heat exchangers are designed to meet these specifications if possible.

The code begins with an estimated compressor pressure ratio and then calculates the state points for both the Brayton and Rankine cycles. Then the code iterates on the compressor ratio to deliver an exit air pressure slightly above atmospheric. In a sub-iteration, it calculates the ratio of mass flows in the Rankine and Brayton cycles. Once the cycle state points have been identified, the output from the cycle is normalized to the desired power level. This sets the total mass flows of air and water. Once the mass flow rates have been calculated, it is possible to size all of the components. The compressor and turbines are sized based on a correlation with state of the art components and simple scaling rules. Each of the heat exchangers is designed based on the configuration chosen from the Kays and London text. Finally the volumes of all of the components are summed to get an estimate of system size.

In order to optimize the efficiency of the combined cycle, there are two main parameters that must be varied. These are the gas turbines' exit temperatures and the steam cycle peak pressure. The peak gas turbine inlet temperature is set as an input parameter. It can be varied, but it is obvious that the higher the gas turbine inlet temperature, the better the efficiency will be the same can be said for most of the components. If their efficiency is higher, the efficiency of the cycle will be higher. The same is true of atmospheric conditions. The colder the input air and the circulating water, the better the efficiency will be. Therefore most of the input parameters are chosen based on nominal values. It is not obvious though what the values of gas turbine exit temperatures and steam pressure should be to get an optimum efficiency. These must be varied to identify the peak efficiency achievable.

The coding for the recuperated system is much simpler than the combined cycle coding because the steam cycle does not have to be modeled. The compressor pressure ratio iteration is much simpler, but still sets the pressure ratio so as to meet an exit pressure slightly above atmospheric. The new calculation in the recuperated system is the air to air recuperator itself. Since the recuperator will be the largest component in the system it does not make sense to prescribe anything other than a counterflow heat exchanger. In this case the pressure drops for the hot and cold fluids cannot be set independently for simple heat transfer correlations. So the pressure drop on the hot air side was chosen as the flow path setting parameter. In this case, also, most of the parameters in the code models are set to nominal values. The only parameter that must be varied to optimize the efficiency of the system is the gas turbines' exit temperatures. The choice of the exit temperature can be made to achieve the peak efficiency.

4.7 Equations

The CC code tracks the conversion of thermal energy to mechanical energy as the working fluid moves through the system. It does this by calculating the temperatures, or enthalpies, and pressures at a sequence of state points between the major system components. The following are the major state points.

Air Path

State 0: Ambient air

State 1: Inlet to compressor

State 2: Exit from compressor (Inlet to recuperator)

State 3: Inlet to first heater (Exit from recuperator)

State 4: Exit from heater, Turbine inlet

State 5: Turbine exit, Recuperator inlet

State 6: Recuperator exit, Superheater inlet

State 7: Superheater exit, Evaporator inlet

State 8: Evaporator exit, Economizer inlet

State 9: Economizer exit, Exhaust to atmosphere

Note: The Heat Recovery Steam Generator contains the superheaters, evaporator, and economizer. State 5 & 6 are the same for the combined cycle and State 6 & 9 are the same for the recuperated cycle.

Steam Path

State 20: Pump inlet, Condenser exit (saturated water)

State 21: Pump exit, Economizer inlet (liquid water)

State 23: Economizer exit, Pinch Point, Evaporator inlet (saturated water)

State 24: Evaporator exit, First Superheater inlet (saturated steam)

State 25: First Superheater exit, High Pressure Turbine inlet (superheated steam)

State 26: High Pressure Turbine exit, Second Superheater inlet (superheated steam)

State 27: Second Superheater exit, Medium Pressure Turbine inlet (superheated steam)

State 28: Medium Pressure Turbine exit, Third Superheater inlet (superheated steam)

State 29: Third Superheater Exit, Low Pressure Turbine inlet (superheated steam)

State 30: Low Pressure Turbine exit, Condenser inlet (superheated steam)

The major equations are:

$$p_2 = p_1 * CPR \qquad CPR = Compressor\ pressure\ ratio$$

$$T_2 = T_1 * CPR^{\frac{\gamma - 1}{\eta_{pc}\gamma}} \qquad \gamma = Ratio\ of\ specific\ heats,\ \eta_{pc} = Compressor\ polytropic\ efficiency$$

$$W_c = \dot{m}_{air} C_p (T_2 - T_1) \qquad W_c = Compressor\ work$$

$$\dot{m}_{air} = Mass\ flow\ rate\ of\ air$$

$$C_p = Constant\ pressure\ specific\ heat\ for\ air$$

$$\varepsilon = \frac{T_3 - T_2}{T_5 - T_2} = \frac{T_5 - T_9}{T_5 - T_2} \qquad \varepsilon = Recuperator\ effectiveness$$

$$Q_1 = \dot{m}_{air} C_p (T_4 - T_3) \qquad Q_1 = Heat\ input\ in\ first\ heater$$

$$W_{t1} = \dot{m}_{air} C_p (T_4 - T_5) \qquad W_{t1} = Work\ from\ first\ turbine$$

$$p_4 = PR_{h1} * p_3 \qquad PR_{h1} = Pressure\ ratio\ across\ first\ heater$$

$$p_5 = p_4 * \left(\frac{T_5}{T_4} \right)^{\frac{\eta_{pt}\gamma}{(\gamma - 1)}} \qquad \eta_{pt} = Turbine\ polytropic\ efficiency$$

$p_9 = p_5 * PR_{rec}$ $\quad PR_{rec}$ = *Pressure ratio across the recuperator*

$T_{23} = T_8 - PPDT$ $\quad PPDT$ = *Pinch point delta temperature*

$T_{25} = T_5 - STDT$ $\quad STDT$ = *Superheater terminal temperature difference*

$\dot{m}_w (h_{25} - h_{22} + h_{27} - h_{26} + h_{29} - h_{28}) = \dot{m}_{air} C_p (T_5 - T_8)$ $\quad \dot{m}_w$ = *Mass flow rate of water*

$\dot{m}_w (h_{30} - h_{29}) = \dot{m}_{air} C_p (T_8 - T_9)$

$\mathrm{W}_p = \dot{m}_w \dfrac{(p_{21} - p_{22})}{\rho_w}$ $\quad \rho_w$ = *Water density*, $\quad W_p$ = *Pump work*

$W_{hp} = \dot{m}_w (h_{25} - h_{26})$ $\quad W_{hp}$ = *Work from high pressure turbine*

$W_{mp} = \dot{m}_w (h_{27} - h_{28})$ $\quad W_{mp}$ = *Work from medium pressure turbine*

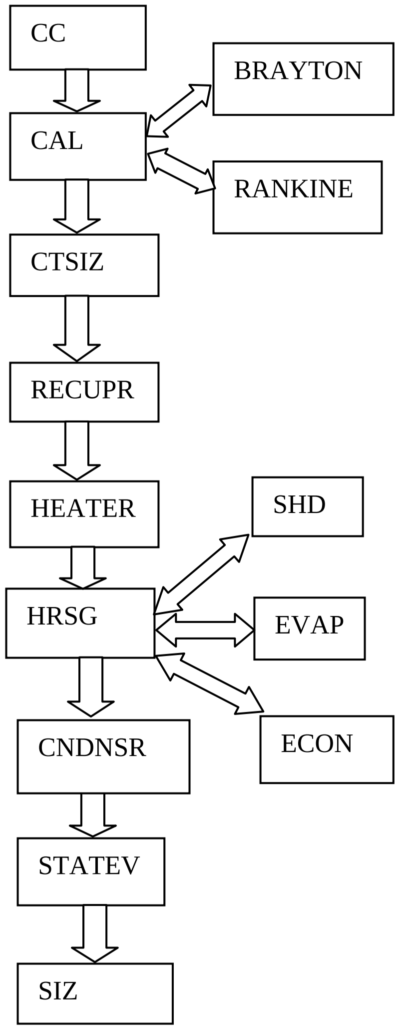
$W_{lp} = \dot{m}_w (h_{29} - h_{30})$ $\quad W_{lp}$ = *Work from low pressure turbine*

$\eta_B = \dfrac{\sum_i W_{ti} - W_c}{\sum_i Q_{hi}}$ $\quad \eta_B$ = *Brayton cycle efficiency*

$\eta_R = \dfrac{W_{hp} + W_{mp} + W_{lp} - W_p}{(h_{25} - h_{24}) + \left(h_{27} - h_{26}\right) + (h_{29} - h_{28})}$ $\quad \eta_R$ = *Rankine cycle efficiency*

$\eta_{overall} = \dfrac{\sum_i W_{ti} - W_c + \dfrac{\dot{m}_w}{\dot{m}_{air}} \left[W_{hp} + W_{mp} + W_{lp} - W_p \right]}{\sum_i Q_{hi}}$ $\quad \eta_{overall}$ = *Overall efficiency*

4.8 Flow Chart



CC-reads input and controls the start and stopping process

CAL - Manages the calculation

BRAYTON- Calculates the Brayton cycle

RANKINE-Calculates the Rankine cycle

CTSIZ - Sizes the cycles to meet the desired power

RECUPR - Designs the recuperator

HEATER - Designs the primary heaters

HRSG-Designs the HRSG

SHD - Designs a superheater

EVAP - Designs the evaporator

ECON -Designs the economizer

CNDNSR - Designs the condenser

STATEV - Summarizes all state variables

SiZ - Summarizes the volumes of all components

Figure 4-11: Flow Chart of Combined Cycle (CC) Computer Code

There an additional set of small routines that calculate material properties as a function temperature. The property tables have been taken from Incropera et al [38]. In addition a new set of steam properties were calculated as needed based on the IAPWS 1997 formulation [39].

4.9 Validation of methodology

Before proceeding to estimate the performance of advanced systems, it is useful to validate the methodology by estimating the performance of a currently deployed system. By the year 2000 GE had over 893 combined cycle and cogeneration systems installed worldwide producing more than 67,397 MW(e) [40], so one of their systems was chosen to model. The system chosen was an S107FA 41 60 Hz, 250 MW Gas Turbine with a three pressure reheat steam cycle. It is a single shaft system and most of the nominal performance parameters are available in the GE literature. The only performance parameter not readily available was the gas turbine inlet temperature. The listed efficiency for the combined cycle plant is 56.5% at standard sea level conditions.

In order to estimate the turbine inlet temperature (T4) a set of given turbine exit temperatures (T5) and pressure ratios were available for 12 GE engines. The T4's were then estimated assuming a polytropic efficiency of 0.95. The results are presented in the following table.

GE Engines	CPR	T5		T4	
GE10-1	15.5	482	C	1449	K
PGT16	20.2	491	C	1562	K
PGT20	15.7	475	C	1440	K
PGT25	17.9	525	C	1585	K
PGT25+	21.5	500	C	1604	K
PGT+G4	23.2	510	C	1654	K
LM6000	28	455	C	1608	K
LMS100	40	417	C	1659	K
MS5002E	17	511	C	1538	K
MS6001B	12.2	548	C	1488	K
MS7001EA	12.6	537	C	1480	K
MS9001E	12.6	543	C	1491	K
	Average =			1546	K
	Range =			1440-1659 K	

Table 4-1: Turbine Temperatures for Several GE Engines

The average temperature of 1546 K was chosen for the baseline comparison. With this temperature and a single turbine model the CC code gives an estimated efficiency of 56.55%. If instead of the average temperature, the two extremes of 1440 K and 1659 K are considered, the code estimates efficiencies of 53.90% and 58.97%, so the calculation is within 2% at its worst and within 0.05% at nominal conditions. It should be pointed out that the simple model for the efficiency,

$$\eta_T = \eta_B + \eta_R - \eta_B \eta_R = 0.426 + 0.421 - 0.426 * 0.421 = 0.668$$

Or it overestimates the combined cycle efficiency by 10.3%. The detailed model developed here does significantly better.

CHAPTER 5 MODELING THE OPEN AIR NUCLEAR BRAYTON COMBINED CYCLE

Having demonstrated that the CC code calculation predicts a reasonable efficiency for current gas turbine combined cycle systems, it can be used to predict the performance of Nuclear Air Brayton Combined Cycle Systems and Nuclear Recuperated Air Brayton Cycle Systems. A nominal set of conditions has been chosen as a best estimate for environmental conditions and component performance. A peak turbine inlet temperature of 933 K was chosen as the baseline condition. It is anticipated that this will be achievable by both the molten salt reactors and the liquid lead, or lead-bismuth reactors. The High Temperature Gas Reactors will do better, but helium to air heat exchangers have not been considered here. For the combined cycle system, the number of turbines, the turbine exit temperature and the steam pressure in the Rankine bottoming cycle were varied to achieve the maximum thermodynamic efficiency. After the optimum efficiency was determined, the sensitivity of this result to important parameters was estimated. For the Recuperated cycle the number of turbines and the turbine exit temperatures were varied to achieve the maximum efficiency. The nominal input parameters for the two systems follow.

5.1 Combined Cycle System Baseline

Number of Turbines - Varied

Turbine Inlet Temperature - 933 K

Turbine Exit Temperature - Varied

Turbine Polytropic Efficiency - 0.90

Compressor Pressure Ratio - Calculated

Compressor Polytropic Efficiency - 0.90

Main Heater Pressure Ratios - 0.99

Atmospheric Pressure - 1 atm

Atmospheric Temperature - 288 K (15^{o}C)

Circulating Water Input Temperature - 288 K (15^{o}C)

Ratio of Exhaust Pressure to Atmospheric - 0.98

Air Pressure Ratios Across Each Superheater - 0.99

Air Pressure Ratio Across the Evaporator - 099

Air Pressure Ratio Across the Economizer - 0.99

Pinch Point Temperature Difference - 10 K

Terminal Temperature Difference at Steam Exit to Superheaters -15 K

Peak Rankine Cycle Pressure - Varied

Intermediate Rankine Cycle Pressure - Varied, 1/4 of Peak

Low Rankine Cycle Pressure - Varied, 1/16 of Peak

Condenser Pressure - 9 kPa

Steam Turbines Thermal Efficiency - 0.90

Power Level - 25 MW

5.2 Nominal Results for Combined Cycle Model

The turbine exit temperatures and the peak pressure in the steam bottoming cycle were varied for systems using 1 to 5 turbines. The best efficiency achievable in each case is plotted in Figure 5-1. The efficiency is a monotonic function of the number of turbines, with the five turbine system only slightly better than the four turbine system. So the four turbine case was chosen as the baseline representative combined cycle system. The peak efficiency for the

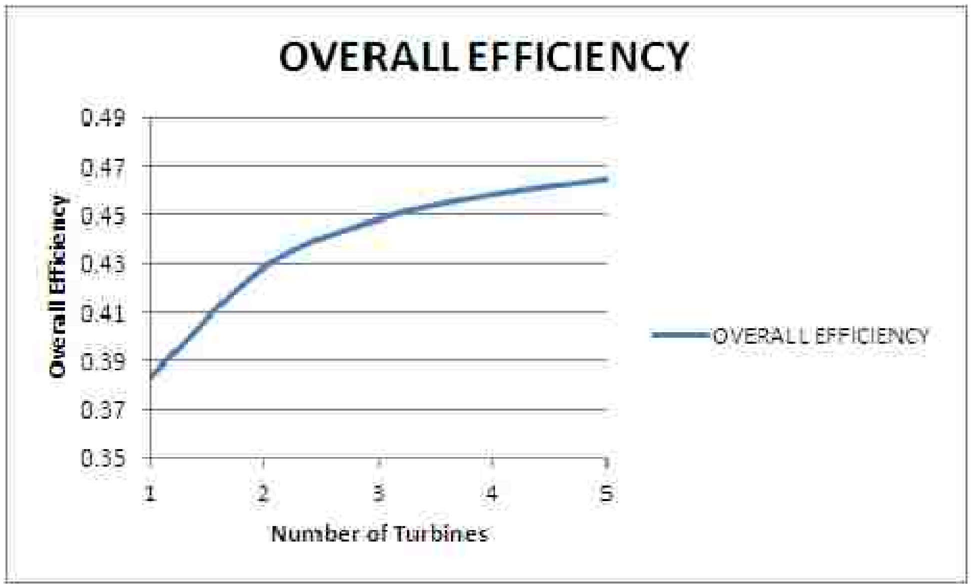


Figure 5-1: Peak Overall Efficiency vs. Number of Turbines

four turbine case is 45.88% and for the five turbine case is 46.52%. The optimum turbine exit temperature for the four turbine case is 810 K with a best steam pressure in the bottoming cycle of 3 MPa. The detailed results are presented in Figure 5-2. The system underperforms with a steam pressure in the bottoming cycle of only 1 MPa, but from 2 to 4 MPa, the results are very similar to the optimum exhaust temperature shifts slightly from 800 to 810 K.

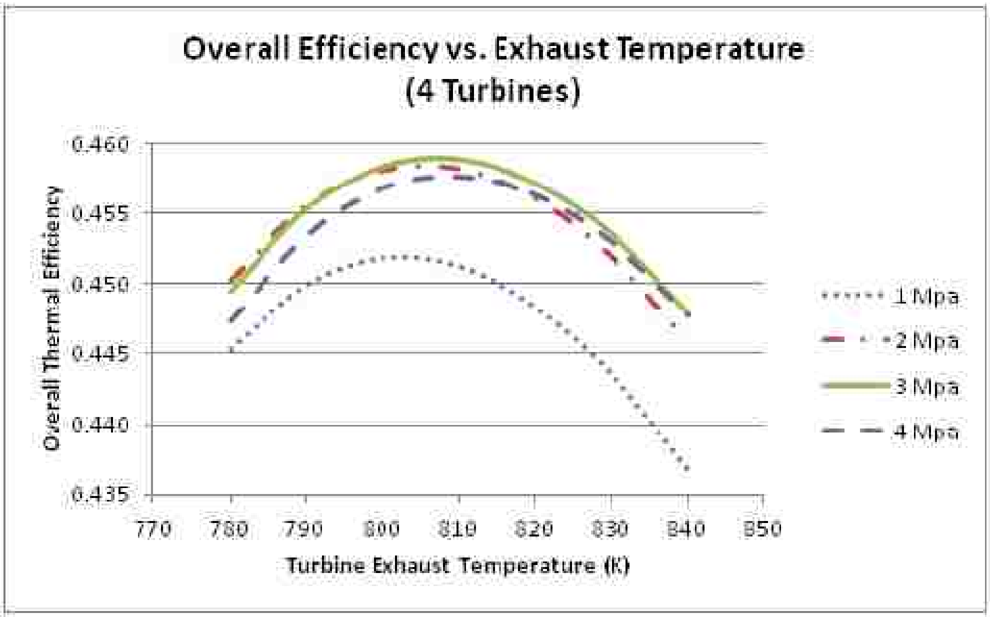


Figure 5-2: Overall Efficiency vs. Exhaust Temperature for Various Steam Pressures

The optimum compressor ratio to achieve peak efficiency and keep the exhaust pressure above the environmental pressure is plotted in Figure 5-3 for each of the turbine models.

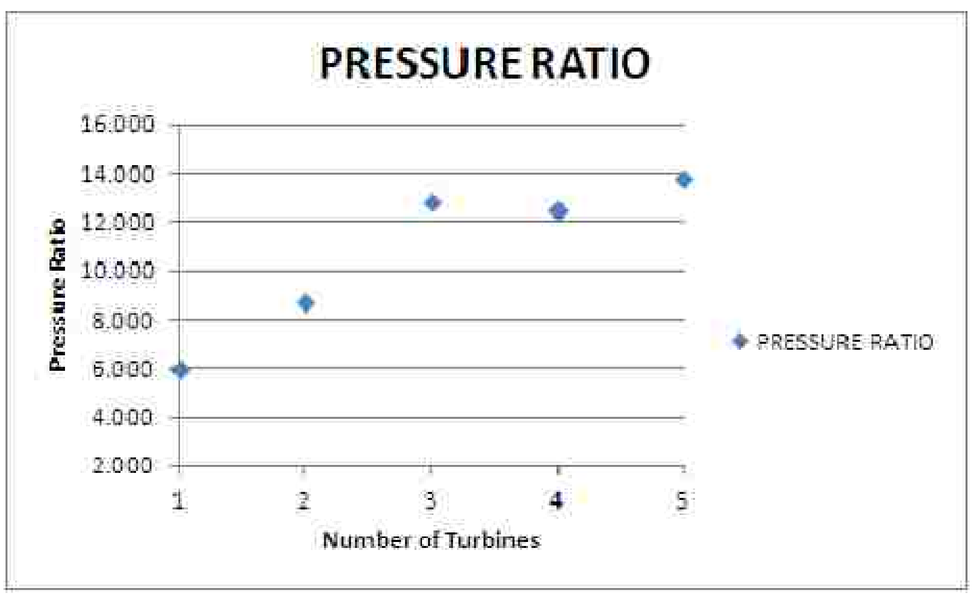


Figure 5-3: Optimum Pressure Ratio vs. Number of Turbines

Since the exhaust temperature and the steam pressure in the bottoming cycle are varying in a discrete manner in fairly large steps, the calculated points do not follow a smooth curve, so a power fit to the data has been included.

Three major sensitivities were considered - compressor polytropic efficiency, pressure drops (or ratios) in the main heaters, and the environmental temperature. Figure 6-4 presents the dependence of the overall thermal efficiency on the gas compressor polytropic efficiency.

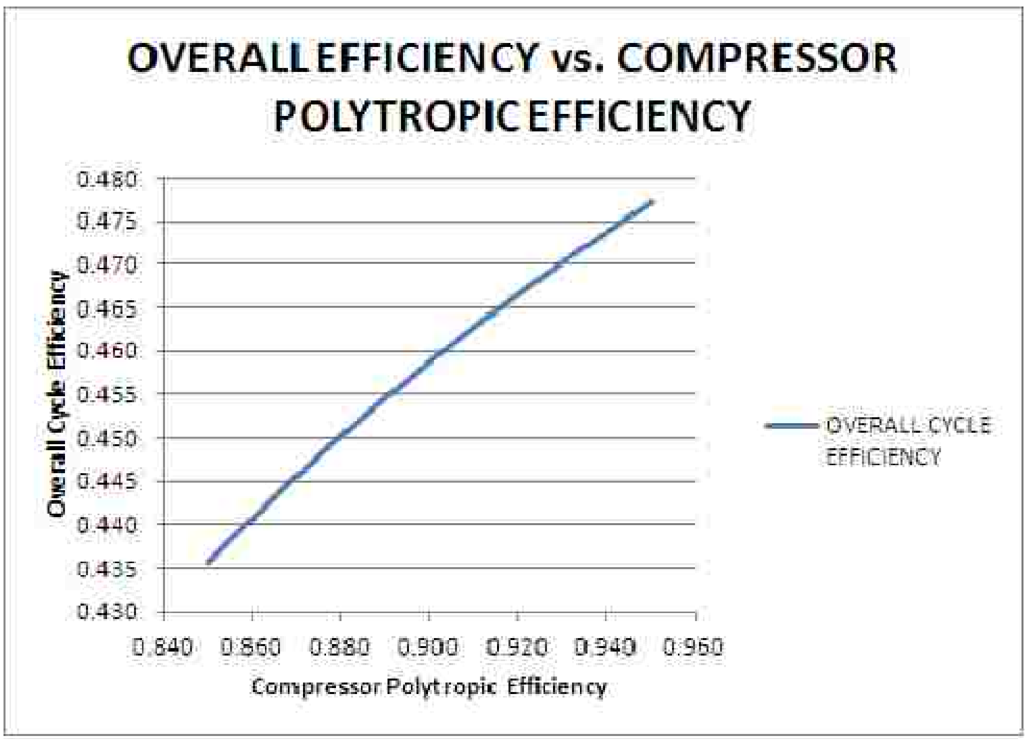


Figure 5-4: Overall Cycle Efficiency vs. Compressor Polytropic Efficiency

The overall thermal efficiency increases about 0.4% for every 1% increase in the compressor polytropic efficiency.

The pressure drops in the main heaters are analogous to the pressure drop in a combustion chamber. Nominally the pressure drop in each heater was set to 1%, but the effect of these pressure drops on the overall efficiency was estimated.

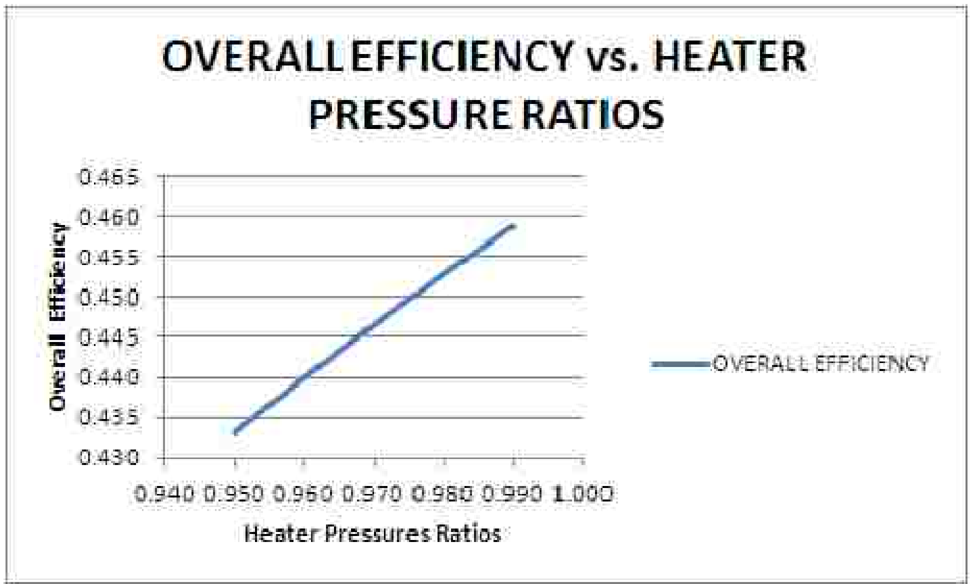


Figure 5-5: Overall Efficiency vs. Heater Pressure Ratio (4 turbines)

The overall efficiency drops approximately 0.5% for every 1% increase in the pressure drop in the main heater for the four turbine system.

The standard day conditions of 15^{o}C and one atmosphere pressure are not likely to be met very often during the operation of a typical power plant, so the variation of efficiency with the ambient temperature was estimated. The results are presented in Figure 6-6 below.

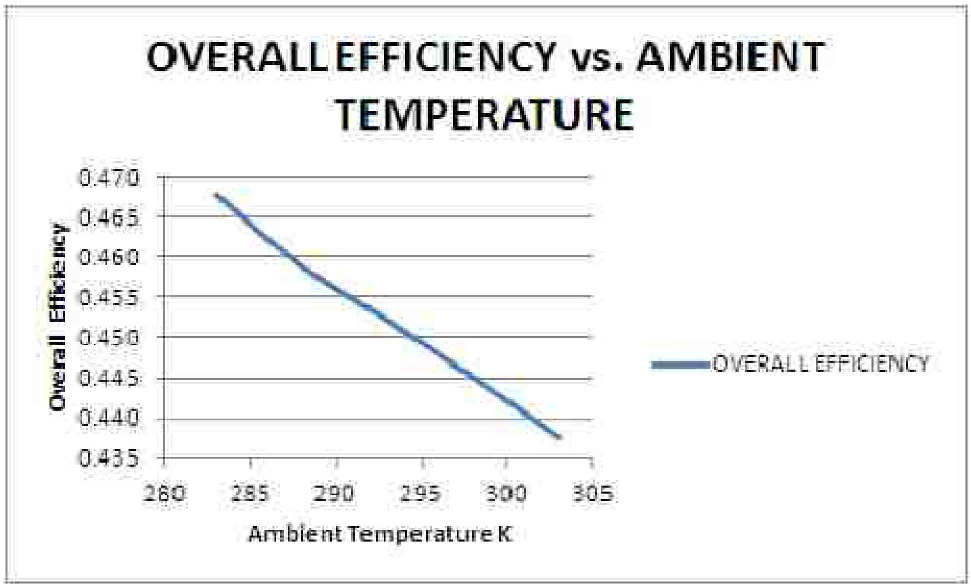


Figure 5-6: Overall Efficiency vs. Ambient Temperature (4 Turbines)

The overall efficiency drops about 1.4% for every 10 K increase in temperature.

The main purpose of designing the heat exchangers and estimating sizes for pump, compressor, and turbine components was to get an overall estimate of the size for the complete power conversion system. This estimate is provided in Figure 5-7 below. Note that for the 25 MW(e) system the air turbines produce 57% of the power and the steam turbines produce 43% of the power. The steam bottoming cycle fills 77% of the estimated volume and the air topping cycle fills 23% of the volume. These percentages appear to remain relatively constant with increasing or decreasing power levels.

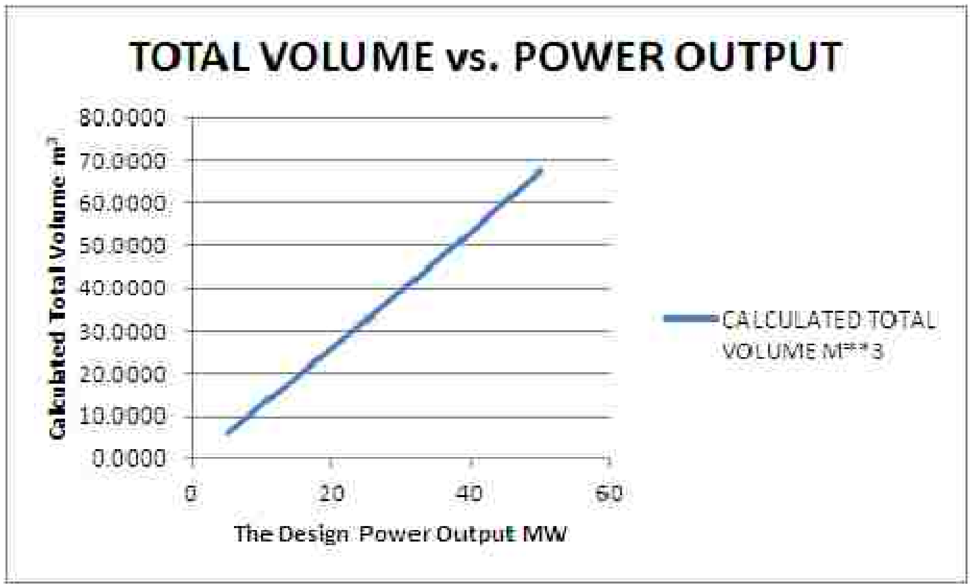


Figure 5-7: System Volume vs. Power Output

5.3 Extension of results vs. peak turbine temperatures

The turbine inlet temperature of 933 K chosen for this study is aggressive but within the range projected for the molten salt reactor and the lead or lead-bismuth cooled reactor. Should it be possible to achieve even higher temperatures in the future, the CC code was used to estimate the efficiencies that might be achieved. Figure 5-8 gives the anticipated efficiencies that can be achieved by a nuclear combined cycle system up to about 1100 K turbine inlet temperature.

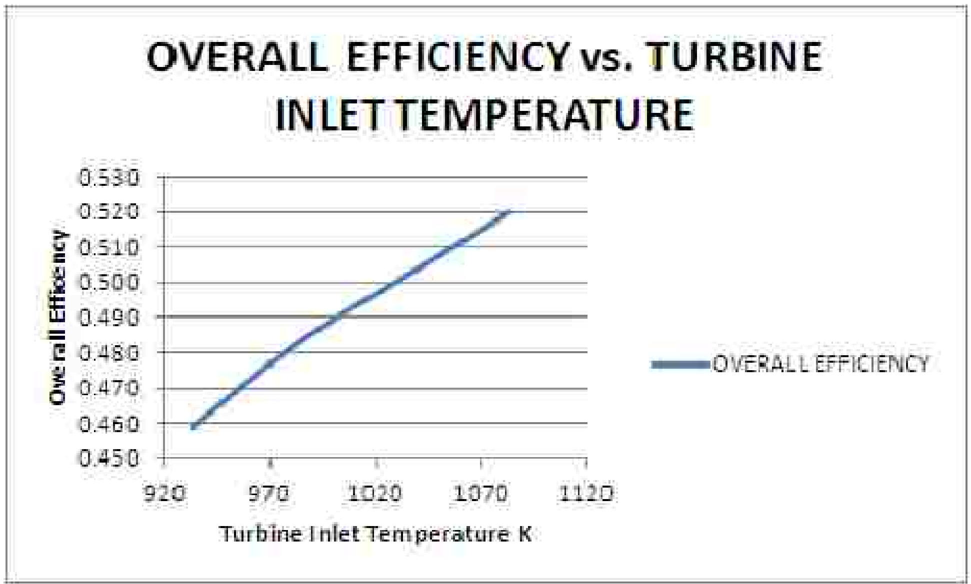


Figure 5-8: Overall Efficiency vs. Increases in Turbine Inlet Temperature

CHAPTER 6 MODELING THE OPEN AIR NUCLEAR RECUPERATED BRAYTON CYCLE

The following section is description of modeling of Combined Cycle (CC) computer code using the open air nuclear recuperated Brayton Cycle

6.1 Recuperated Cycle System Baseline

Number of Turbines - Varied

Turbine Inlet Temperature - 933 K

Turbine Exit Temperature - Varied

Turbine Polytropic Efficiency - 0.90

Compressor Pressure Ratio - Calculated

Compressor Polytropic Efficiency - 0.90

Main Heater Pressure Ratios - 0.99

Recuperator Effectiveness - 0.95

Recuperator Pressure Ratio on Hot Side - 0.99

Atmospheric Pressure - 1 atm

Atmospheric Temperature - 288 K (15^{o}C)

Circulating Water Input Temperature - 288 K (15^{o}C)

Ratio of Exhaust Pressure to Atmospheric - 0.98

Power Level - 25 MW

6.2 Nominal Results for Recuperated Cycle

For this case, one to five turbines were considered also. However the only parameter of interest for optimizing the overall cycle efficiency is the turbine exhaust temperature. The results for all five turbine models are presented in Figure 6-1 below.

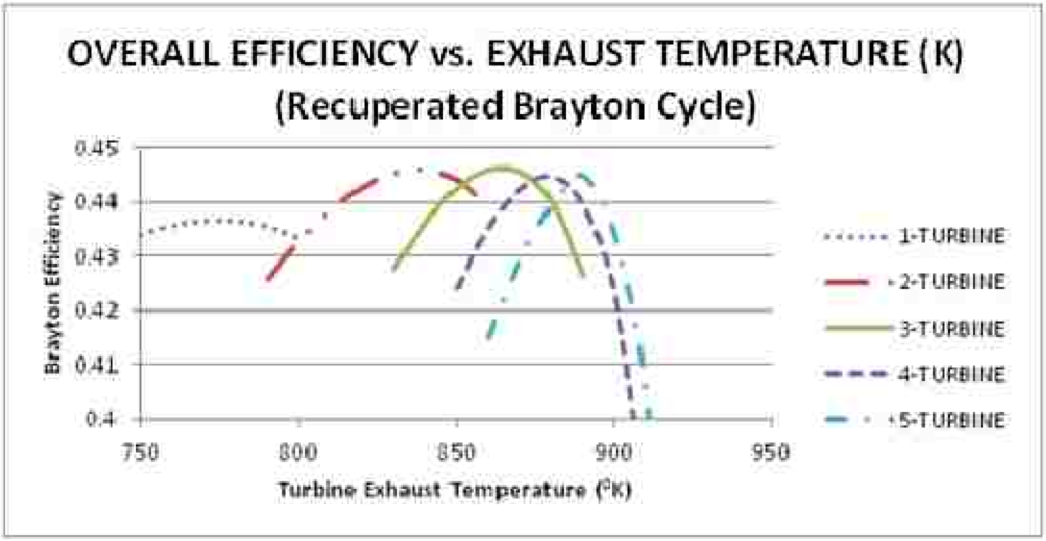


Figure 6-1: Recuperated Overall Efficiency vs. Turbine Exhaust Temperatures

For the recuperated Brayton Cycles, the efficiency peaks with three turbines under the nominal conditions, but the peak efficiencies for two and four turbines are very close. The actual numbers are given in Table 6-1.

Turbines	Overall Efficiency
1	0.4366
2	0.4458
3	0.4461
4	0.4447
5	0.4447

Table 6-1: Peak Cycle Efficiencies for Recuperated Brayton System

So all sensitivity studies were done on the three turbine model and it is interesting to note that the peak efficiency achieved for the recuperated system of 44.61% is only 1.27% less than the peak efficiency obtained for the combined cycle system.

The compressor pressure ratios for the peak efficiencies are plotted in Figure 6-2 below.

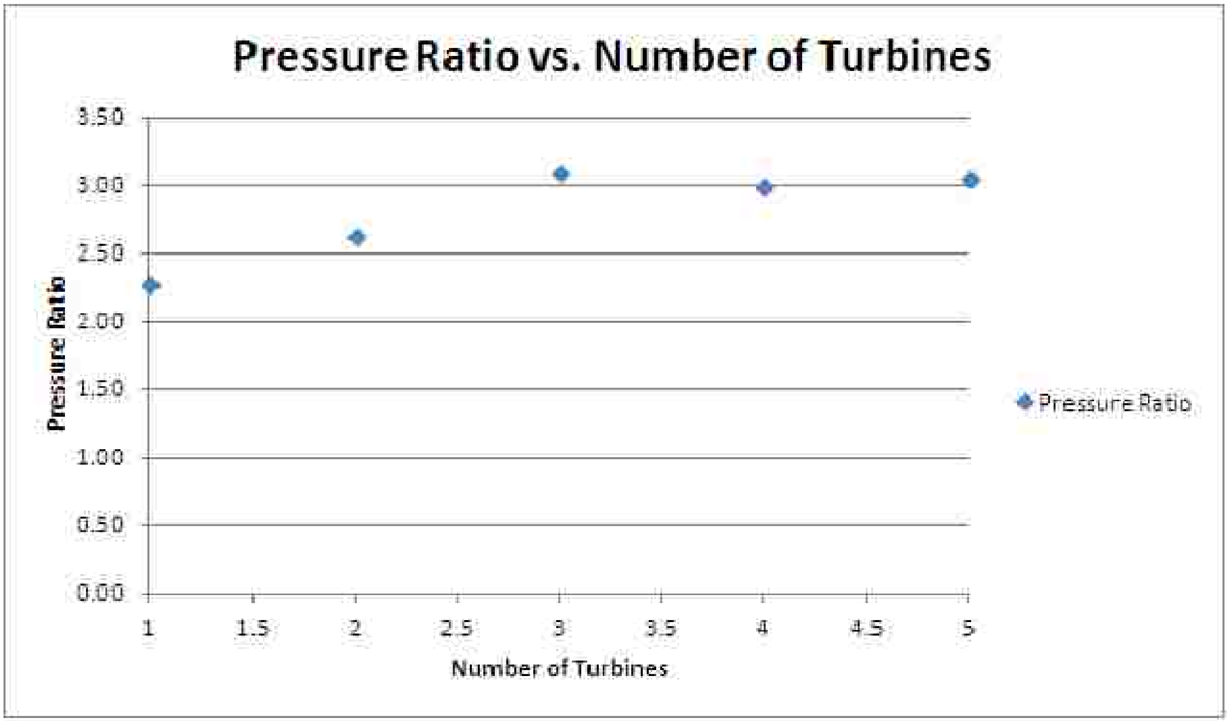


Figure 6-2: Compressor Pressure Ratios for Peak Efficiency vs. Number of Turbines

As expected for a recuperated system, the compressor pressure ratios are significantly less than those for an unrecuperated system.

The sensitivity of the results to recuperator pressure drops and recuperator effectiveness were estimated. The sensitivity to the recuperator pressure drops is presented in Figure 6-3 below.

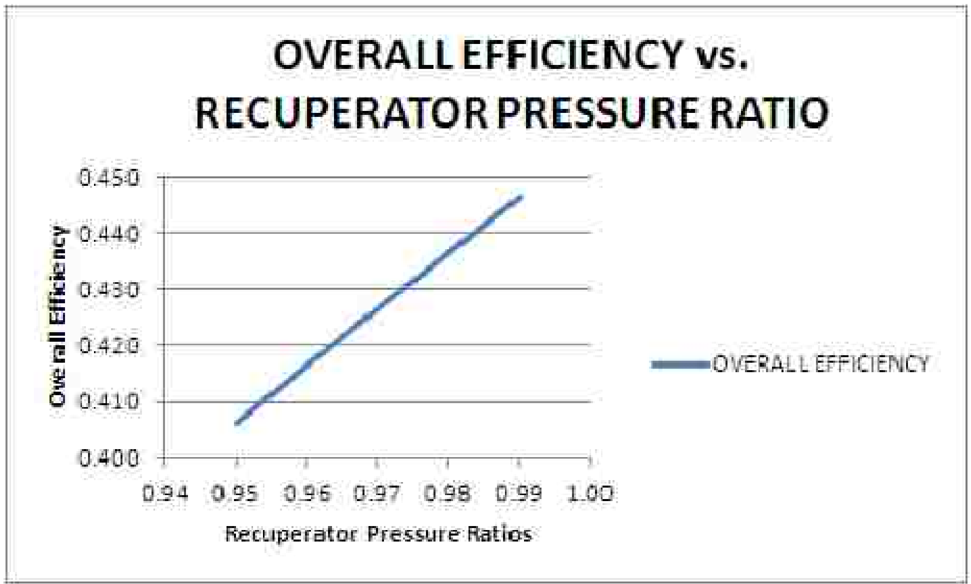


Figure 6-3: System Efficiency vs. Recuperator Pressure Ratio

The cycle efficiency drops 0.6% for every 1% increase in the recuperator pressure drop. This is one of the more sensitive parameters because the pressure drops occur on both the cold leg and the hot leg, and the compressor pressure ratio has to compensate for both of them.

The recuperator effectiveness of 0.95 for the baseline cases produces a very large recuperator, so the effectiveness of the recuperator was varied to determine its effect on the overall cycle efficiency and the volume of the resulting system. The effectiveness of the recuperator's impact on the overall efficiency is plotted in Figure 6-4 below.

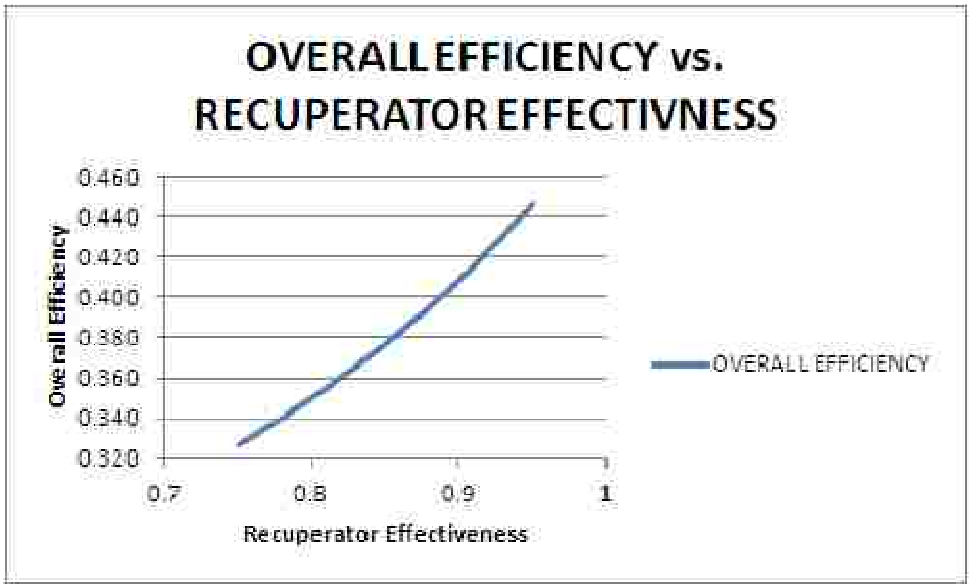


Figure 6-4: Efficiency vs. Recuperator Effectiveness (3 Turbines)

The overall system efficiency also drops approximately 0.6% for every 1% decrease in the recuperator's effectiveness. The recuperator volume estimates for the baseline 25 MW system are plotted in Figure 6-5.

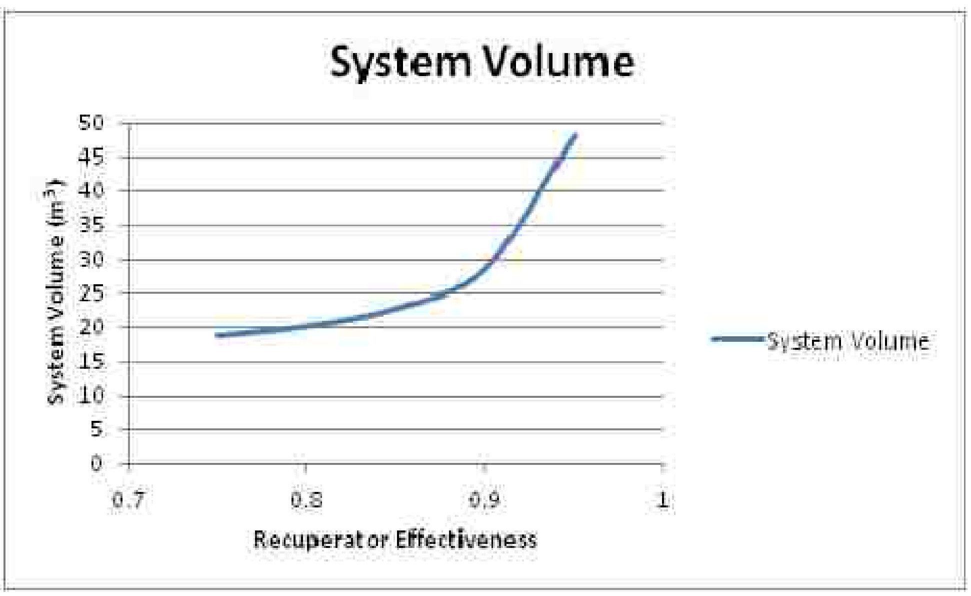


Figure 6-5: Recuperated System Volume vs. Recuperator Effectiveness

It appears that asking for an effectiveness of 0.95 causes a very significant size increase compared to an effectiveness of 0.9. Giving up 5% in effectiveness causes a 4% loss in system efficiency, but reduces the size of the system by 40%.

Figure 6-6 below gives the Recuperator system sizes as a function of total system power for the range from 5 to 50 MW.

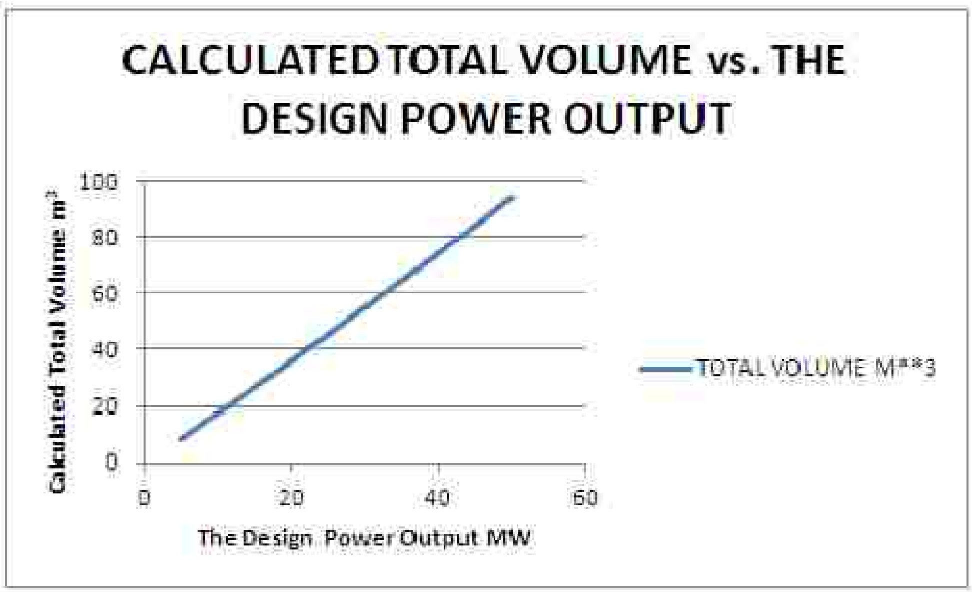


Figure 6-6: System Volume vs. System Power for the Recuperator System

Clearly the recuperated system requires a larger volume than the Combined Cycle system, but it has fewer components and is free of the circulating water requirements.

6.3 Extension of results vs. peak turbine temperatures

The turbine inlet temperature of 933 K chosen for this study was aggressive but within the range projected for the molten salt reactor and the lead or lead-bismuth cooled reactor. Should it be possible to achieve even higher temperatures in the future, the CC code was used to estimate the efficiencies that might be achieved. Figure 6-7 gives the anticipated efficiencies that can be achieved by a nuclear open air recuperated cycle up to about 1100 K turbine inlet temperature. It is worth remembering that these temperatures are still about 500 K below state of the art gas turbine inlet temperatures.

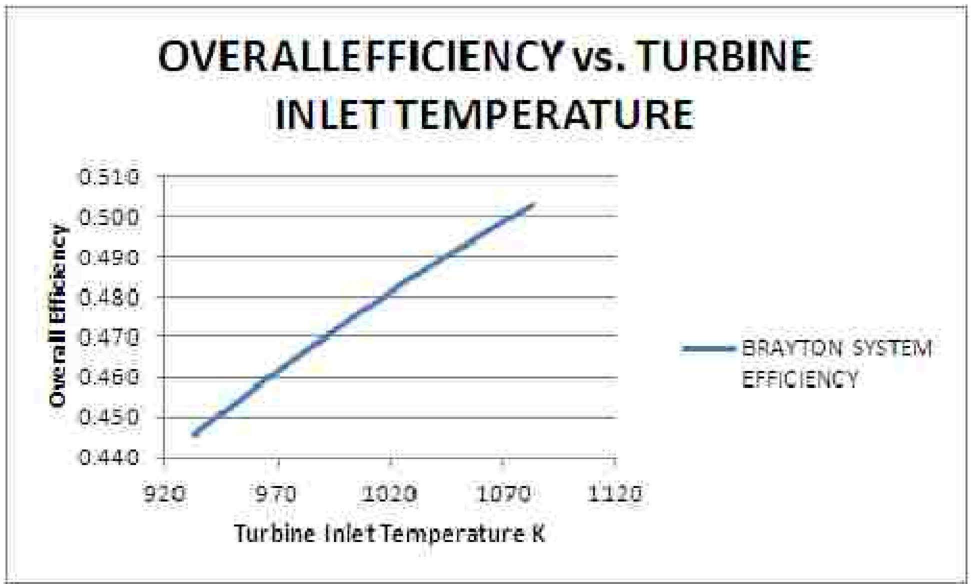


Figure 6-7: Recuperated System Efficiency vs. Turbine Inlet Temperature

Perhaps a more interesting comparison is to compare the Open Air Recuperated Brayton cycle with a Supercritical Water Cycle reactor. The Supercritical Water Reactor is projecting a pressure of 25 MPa and an outlet temperature of 500 ^{0}C. The estimated efficiency is 0.40. Figure 6-8 compares the CC estimated thermal efficiency of the Open Air Recuperated Brayton Cycle against a Supercritical Water Reactor operating at 25 MPa as the Turbine Inlet Temperature is varied.

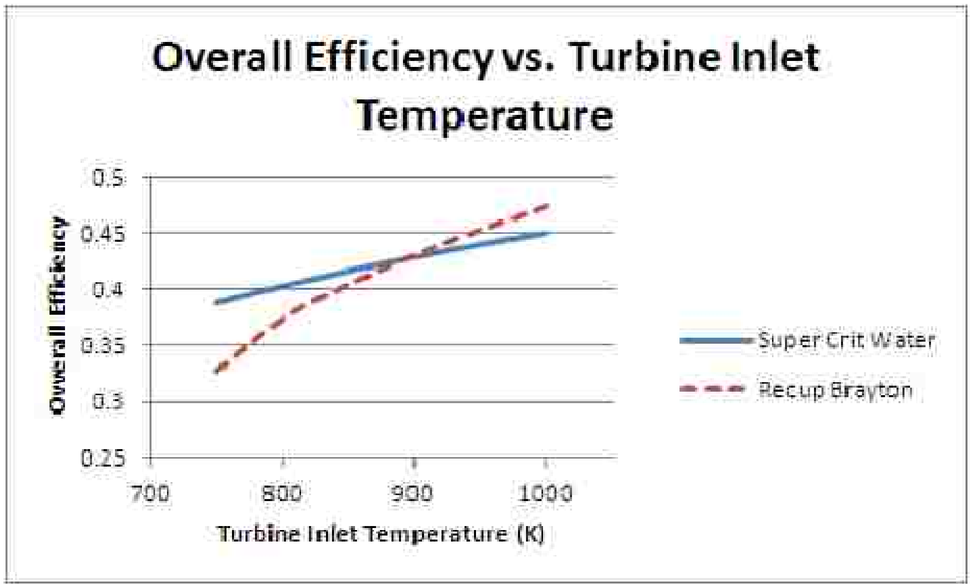


Figure 6-8: Comparison of Open Air Recuperated Brayton Reactor vs. Supercritical Water Reactor

The Recuperated Brayton system produces a nominal 44.64% efficiency at the projected 933 K. The Supercritical Water cycle produces 43.45% efficiency at this temperature if it were to reach it. The Supercritical Water cycle produces a 40% efficiency at 773 K, but the Recuperated Brayton cycle only produces an efficiency of 35% at this temperature. The obvious conclusion is, that to achieve significantly higher efficiencies the turbine inlet temperatures must increase. If that is the direction of development, then the Recuperated Brayton and Combined Cycle Brayton-Rankine cycle systems are the way to go.

CHAPTER 7 CONCLUSION

The Combined Cycle code methodology does an excellent job of modeling combined cycle plants and allows prediction of performance based on a number of plant design features. It predicts plant efficiencies above 45% for reactor temperatures predicted to be in the 700^{o}C range. It also predicts reduction in circulating water requirements by over 50% compared to current generation reactors. This will be a major saving in water resources and should reduce the capital investment in large cooling towers. In fact the circulating water requirements can be eliminated completely if a recuperator is used instead of the bottoming steam cycle. The efficiency penalty for going to a recuperated system appears to be only about 1-2%. The size of the system increases by about 50% however.

No attempt to analyze a gas cooled reactor was undertaken in this study. Given that efficiencies will continue to increase with even higher reactor outlet temperatures, it could prove fruitful to consider this possibility in the future. However, the primary heaters will be large, but they will operate at higher pressure so they will not grow like the recuperator does. This brings up another possibility for improving the recuperated system's performance. It might be possible to exhaust at higher pressure and reduce the size of a 0.95 effective recuperator. Little has been done in this study to optimize the recuperated design. This study started as a combined cycle effort and the recuperator was investigating as additional issue. Future work may well be able to reduce the size of the recuperator and keep a high efficiency.

Though this study focused on a 25 MW(e) power conversion system, it only considered one loop. Typically an LWR system has 3 or 4 loops and so the results are easily extended to a 100 MW(e) plant. This should cover the range for most of the Generation IV Small Modular Reactors that are being considered. There is also talk of replicating Small Modular Reactors to build up to a significant power station. So the 25 MW(e) system components appear to be a reasonable point to start hardware development.

System efficiencies around 40% appear very achievable with conservative performance on the various heat exchangers required for each of these systems. Two turbine systems and three turbine systems appear to achieve performances comparable to a four turbine system.

Reheat cycles in the bottoming steam cycle do not appear to be advantageous at the pressures (10 MPa) currently being used in combined cycles. Lower pressures in the order of 1 to 2 MPa may reverse this result and let efficiencies see an improvement with steam reheat cycles.

It would appear that the analysis accomplished here only scratched the surface of the possibilities for this type of system and thermal efficiencies easily above 40% should be readily achievable.

Suggested future work can be seen as an effort with the following possible scenarios,

- Apply an Optimization Framework to the Combined Cycle Code
    - Will allow finer discretazation on variables like pressure ratio
    - Can vary turbine exit temperatures independently
- Investigate off design performance for a fixed design
- Investigate optimum way to implement a power turbine
- Investigation best method of hybridization with natural gas – possibly 60+% efficient
- Redevelop the present model in Modelica

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APPENDIX A GLOSSARY

Glossary of Nuclear Terms (US NRC)

AHTR	Advanced High Temperature Reactor
AECL	Atomic Energy of Canada Ltd.
AECB	Atomic Energy Control Board
AESOP	Atomic Energy Simulation of Optimization (computer code)
ASDV	Atmospheric Steam Discharge Valve
ASSERT	Advanced Solution of Sub-channel Equations in Reactor Thermal hydraulics (computer code)
ASTM	American Society for Testing Materials
BLC	Boiler Level Control
BLW	Boiling Light Water
BPC	Boiler Pressure Controller
CBA	Core Barrel Assembly
CCP	Critical Channel Power
CHF	Critical Heat Flux
CPR	Critical Power Ratio
CRL	Chalk River Laboratories
CRT	Cathode Ray Tube
CSA	Canadian Standards Association
CSDV	Condenser Steam Discharge Valve
CSNI	Canadian Standards for the Nuclear Industry
DBE	Design Base Earthquake
DCC	Digital Control Computer
DF-ET	Drift Flux-Equal Temperature
DF-UT	Drift Flux-Unequal Temperature
DNB	Departure from Nucleate Boiling
ECC	Emergency Core Cooling
ECI	Emergency Core Injection
EFPH	Effective Full Power Hours
EVET	Equal Velocity Equal Temperature
EVUT	Equal Velocity-Unequal Temperature
EWS	Emergency Water Supply
FBR	Feed, Bleed and Relief
FP	Full Power

HEM	Homogeneous Equilibrium Model
HTS	Heat Transport System
HWP	Heavy Water Plant
HYDNA	Hydraulic Network Analysis (extinct computer code)
I&C	Instrumentation and Control
IBIF	Intermittent Buoyancy Induced Flow
ICRP	International Commission on Radiological Protection
LOC	Loss of Coolant
LOCA	Loss of Coolant Accident
LOC/LOECC	Loss of Coolant with Coincident Loss of Emergency Core Cooling
LOP	Loss of Pumping
LOR	Loss of Regulation
GT-MHR	Gas Turbine-Modular Helium Reactor
GTHTR	Gas Turbine High Temperature Reactor
MCCR	Ministry of Corporate and Consumer Relations
MCS	Maintenance Cooling System
MHD	Magneto hydrodynamics
milli-k	Unit of reactivity for reactor physics
NGNP	Next Generation Nuclear Plant
NPD	Nuclear Power Demonstration
NPSH	Net Positive Suction Head
NUCIRC	Nuclear Circuits (computer code)
OECD	Organization for Economic Co-operation & Development
OH	Ontario Hydro
PBMR	Pebble Bed Modular Reactor
PCS	Power Conversion System
PGSA	Pickering Generating Station A
PHTS	Primary Heat Transport System
PHW	Pressurized Heavy Water
PHWR	Pressurized Heavy Water Reactor
PRESCON2	Pressure Containment (computer code)
QA	Quality Assurance
RAMA	Reactor Analysis Implicit Algorithm
R&M	Reliability and Maintainability
RB	Reactor Building
RCS	Reactivity Control System

RIH	Reactor Inlet Header
ROH	Reactor Outlet Header
RSS	Reserve Shutdown System
RTD	Resistance Temperature Detectors
RU	Reactor Unit
SDM	Safety Design Matrices
SOPHT	Simulation of Primary Heat Transport (computer code)
SRV	Safety Relief Valve
TMI	Three Mile Island
TOFFEA	Two Fluid Flow Equation Analysis (computer code)
TRIS	Triple-Coated Isotropic
UVUT	Unequal Velocity Unequal Temperature
VB	Vacuum Building
VC	Vacuum Chamber
VHTR	Very High Temperature Reactor
WRE	White-shell Research Establishment
LEU-TRISO	Low Enriched Uranium Triple-Coated Isotropic.
PBMR	Pebble Bed Modular Reactor
HTR	High Temperature Reactor
CSC	Core Structure Ceramics
CBA	Core Barrel Assembly
CS	Core Structures
THTR	Thorium High Temperature Reactor
HTGR	High Temperature Gas-Cooled Reactor
FHSS	Fuel Handling and Storage System
HRSG	Heat Recovery Steam Generators
CHP	Combined Heat and Power
NGNP	Next Generation Nuclear Power Plant
NERI	Nuclear Energy Research Initiative
LFR	Lead-Alloy Cooled Fast Reactor
GFR	Gas-Cooled Fast Reactor
AFCI	Advanced Fuel Cycle Initiative
EBR-II	Experimental Breeder Reactor
LWR	Light Water Reactor
TRISO	Tristructural-Isotropic
DRACS	Direct Reactor Auxiliary Cooling System

DHX	Dump Heat Exchanger
IHX	Intermediate Heat Exchanger
PCS	Power Conversion System

APPENDIX B: INPUT AND OUTPUT OF COMPUTER CODE CC (COMBINED CYCLE)

The following is a summary description of Combined Cycle Computer code as well as list of input and output of the code.

B-1: Summary List of Combined Cycle Code

```
      PROGRAM CC
C
C   CC ANALYZES AND DESIGNS AN EXTERNALLY HEATED BRAYTON CYCLE WITH A RANKINE
C   BOTTOMING CYCLE. 1 TO 5 GAS TURBINES ARE CONSIDERED, WITH 1 OR 2 COMPRESSORS.
C   IF THE COMPRESSOR IS SPLIT, AN INTERCOOLER THAT ALSO SERVES AS A FEEDWATER
C   HEATER IS INSERTED BETWEEN THE TWO COMPRESSORS.  THE RANKINE CYCLE CAN HAVE
C   FROM 1 TO 3 STEAM TURBINES IN THE SUPERHEATER SECTION.
C
C   THE INPUT DATA IS SET UP TO RUN A SET OF DEFAULT VALUES SO THAT ONLY THE
C   DEVIATION FROM THESE VALUES NEED BE ENTERED AS INPUT.  THE DEFAULT VALUES
C   ARE SPECIFIED IN THE 'RESET' SUBROUTINE.  THE PARAMETERS REQUIRED TO
C   SPECIFY A DESIGN ARE CONTAINED IN THE LABELED COMMONS BELOW.  THE
C   MINIMUM INPUT REQUIRED TO COMPLETE THE DEFAULT CALCULATION IS A TITLE
C   RECORD FOLLOWED BY THE WORD START IN THE FIRST FIVE CHARATERS OF THE NEXT
C   LINE.
C
C   INPUT CHANGES TO THE DEFAULT VALUES ARE READ IN BLOCKS CORRESPONDING TO
C   EACH OF THE LABELED COMMONS.  THE NEW VALUES ARE ENTERED BY ENTERING THE
C   NAME OF THE LABELED COMMON AS THE FIRST FIVE CHARACTERS OF A RECORD.  THE
C   NEXT RECORD SHOULD THEN CONTAIN THE NEW VALUES THE VARIABLES IN THAT
C   COMMON.  THE COMMONS CAN BE ENTERED IN ANY ORDER.  THE LISTING BELOW DEFINES
C   THE COMMONS AND THE VARIABLES CONTAINED IN EACH.  IF THE COMMON BLOCK
C   IS ENTERED MORE THAT ONCE, THE LAST ENTRY IS USED IN THE COMPUTATION.
C
C  A COMPLETE INPUT WOULD BE
C
C LINE 1: TITLE CARD 1-80 CHARACTERS
C
C LINE 2: 'TRBIN'
C
C LINE 3: 'LPRINT,NTURB'
C
C           LPRINT=0, MINIMUM PRINT OUT
C                 =1, PRINTS INPUT PLUS MINNIMUM OUTPUT
C                 =2, PRINTS INPUT PLUS SUMMARY PERFORMANCE
```

```
C                  =3, PRINTS QUANTITATIVE DESIGN
C
C            NTURB=THE NUMBER OF AIR TURBINES CONSIDERED (1-5)
C
C LINE 4:(REPEATED NTURB TIMES) 'T4(I),T5(I),PET(I)'
C
C           T4(I)=AIR TEMPERATURE AT INLET TO ITH TURBINE (K)
C
C           T5(I)=AIR TEMPERATURE AT EXIT OF ITH TURBINE (K)
C
C           PET(I)=POLYTROPIC EFFICIENCY FOR ITH TURBINE
C
C LINE 5: 'COMPR'
C
C LINE 6: 'NCOMP,CPRZ1,CPRZ2,PEC'
C
C           NCOMP=NUMBER OF COMPRESSORS (1 OR 2)
C
C           CPRZ1=PRESSURE RATIO FOR FIRST COMPRESSOR
C
C           CPRZ2=PRESSURE RATIO FOR SECOND COMPRESSOR(1.0 IF ONLY 1)
C
C           PEC=POLYTROPIC EFFICIENCY FOR THE COMPRESSOR(S)
C
C LINE8: 'HEATR'
C
C LINE9: '(HPR(I), I=1,NTURB'
C
C           HPR(I)=ESTIMATED PRESSURE RATIO FOR ITH HEATER
C
C LINE 10: 'RCUPR'
C
C LINE 11: 'RCEFF,PRRCP,KRC'
C
C            RCEFF=RECUPERATOR EFFECTIVENESS
C
C            PRRCP=MINIMUM ACCEPTABLE PRESSURE RATIO FOR THE RECUPERATOR
C
C            KRC=1, COUNTERFLOW RECUPERATOR
C               =2, CROSSFLOW RECUPERATOR WITH BOTH FLUIDS MIXED
C               =3, CROSS FLOW RECUPERATOR WITH ONE FLUID MIXED
C
```

```
C LINE 12: 'ICOOL'
C
C LINE 13: 'CIPR,CIEFF'
C
C            CIPR=ESTIMATED PRESSURE RATIO FOR THE INTERCOOLER
C
C            CIEFF=INTERCOOLER EFFICIENCY
C
C LINE 14: 'ATMOS'
C
C LINE 15: 'PATM,TATM,TCWIN,EXPR'
C
C            PATM=ATMOSPHERIC PRESSURE WHERE SYSTEM IS OPERATING
C
C            TATM=AMBIENT TEMPERATURE WHERE SYSTEM IS OPERATING
C
C            TCWIN=TEMPERATURE OF THE INLET COOLING WATER
C
C            EXPR=RATIO OF EXIT PRESSURE TO ATMOSPHERIC PRESSURE
C
C LINE 16: 'STGEN'
C
C LINE 17: 'PRSG(3),PPDT,STDT,PSG1,PSG2,PSG3,CNDT,STEFF'
C
C            PRSG(1)=PRESSURE RATIO FOR THE SUPERHEATER
C
C            PRSG(2)=PRESSURE RATIO FOR THE EVAPORATOR
C
C            PRSG(3)=PRESSURE RATIO FOR THE ECONOMIZER
C
C            PPDT=TEMPERATURE DIFFERENCE AT PINCH POINT (K)
C
C            STDT=TEMPERATURE DIFFERENCE AT INPUT TO SUPEHEATER
C
C            PSG1=PRESSURE FOR ECONOMIZER,EVAPORATOR AND HP SUPERHEATER (PAS)
C
C            PSG2=PRESSURE FOR FIRST REHEAT (PAS)
C
C            PSG3=PRESSURE FOR SECOND REHEAT (PAS)
C
C            CNDT=CONDENSER TEMPERATURE (K)
C
```

```
C            STEFF=STEAM TURBINE(S) EFFICIENCY
C
C LINE 18: 'POWER'
C
C LINE 19: 'PTOT,TFIN,WDOTF,PRF'
C
C            PTOT=TOTAL SYSTEM POWER GENERATED (W)
C
C            TFIN=SECONDARY LOOP FLUID INPUT TEMPERATURE (K)
C
C            WDOTF=MASS FLOW RATE OF SECONDARY LOOP FLUID (KG/SEC)
C
C            PRF=FLUID PRESSURE IN THE REACTOR COOLING LOOP (PAS)
C
C LINE 20: 'CSIZE'
C
C LINE 21: 'RRAT,SOLID,RMIC,RMEC,(RMET(I),I=1,NTURB)
C
C            RRAT=RADIUS RATIO FOR THE TURBINES (RMIN/RMAX FOR BLADES)
C
C            SOLID=FRACTION OF TURBINE DISK NOT OPEN TO FLOW
C
C            RMIC=MACH NUMBER AT ENTRANCE TO COMPRESSOR
C
C            RMEC=MACH NUMBER AT EXIT OF COMPRESSOR
C
C            RMET(I)=MACH NUMBER AT EXIT OF ITH TURBINE
C
C LINE 22: 'MATER'
C
C LINE 23: 'MFL,MHT,MIC,MEC,MEV,MSH,MCD,MRC'
C
C            MFL=MATERIAL IDENTIFIER FOR SECONDARY LOOP FLUID
C
C            MHT=MATERIAL IDENTIFIER FOR THE GAS/AIR HEATERS
C
C            MIC=MATERIAL IDENTIFIER FOR THE INTERCOOLER
C
C            MEC=MATERIAL IDENTIFIER FOR THE ECONOMIZER
C
C            MEV=MATERIAL IDENTIFIER FOR THE EVAPORATOR
C
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C            MSH=MATERIAL IDENTIFIER FOR THE SUPERHEATER
C
C            MCD=MATERIAL IDENTIFIER FOR THE CONDENSER
C
C            MRC=MATERIAL IDENTIFIER FOR THE RECUPERATOR
C
C LINE 24: "HEATX'
C
C LINE 25: 'KHT,KSH,KEV,KEC,KCD,KIC,KRC'
C
C           KHT=HEAT TRANSFER & FRICTION CORRELATION FOR MAIN HEATERS
C
C            KSH=HEAT TRANSFER & FRICTION CORRELATION FOR SUPERHEATERS
C
C            KEV=HEAT TRANSFER & FRICTION CORRELATION FOR EVAPORATOR
C
C            KEC=HEAT TRANSFER & FRICTION CORRELATION FOR ECONOMIZER
C
C            KCD=HEAT TRANSFER & FRICTION CORRELATION FOR CONDENSER
C
C            KIC=HEAT TRANSFER & FRICTION CORRELATION FOR INTERCOOLER
C
C            KRC=HEAT TRANSFER & FRICTION CORRELATION FOR RECUPERATOR
C
C LINE 26: 'START'
C
C  NOTE: THE ONLY REQUIRED ENTRY IS THE 'START' RECORD AFTER THE TITLE
C   RECORD TO EXECUTE THE DEFAULT CASE.
C
```

B-2: Input File and Output Results for Four Turbine Combined Cycle

INPUT FILE

```
Advanced System w/4 turbines
TRBIN
 4 4
 933.0 810.0 0.90
 933.0 810.0 0.90
 933.0 810.0 0.90
 933.0 810.0 0.90
RCUPR
 0.0 0.99 1
STGEN
 0.99 0.99 0.99 0.999 10.0 15.0 3000000.0 750000.0 187500.0 313.0 0.90
START
```

OUTPUT FILE

```
Advanced System w/4 turbines


 THE PRINT LENGTH PARAMETER IS  4

 THE NUMBER OF TURBINES =  4
    DATA FOR TURBINE NUMBER  1
    TURBINE INLET TEMPERATURE     =         933.0 K          1679.4 R
    TURBINE EXIT TEMPERATURE      =         810.0 K          1458.0 R
    TURBINE POLYTROPIC EFFICIENCY =         0.900
    DATA FOR TURBINE NUMBER  2
    TURBINE INLET TEMPERATURE     =         933.0 K          1679.4 R
    TURBINE EXIT TEMPERATURE      =         810.0 K          1458.0 R
    TURBINE POLYTROPIC EFFICIENCY =         0.900
    DATA FOR TURBINE NUMBER  3
    TURBINE INLET TEMPERATURE     =         933.0 K          1679.4 R
    TURBINE EXIT TEMPERATURE      =         810.0 K          1458.0 R
    TURBINE POLYTROPIC EFFICIENCY =         0.900
    DATA FOR TURBINE NUMBER  4
    TURBINE INLET TEMPERATURE     =         933.0 K          1679.4 R
    TURBINE EXIT TEMPERATURE      =         810.0 K          1458.0 R
    TURBINE POLYTROPIC EFFICIENCY =         0.900

 THE NUMBER OF COMPRESSORS = 1
    FIRST COMPRESSOR PRESSURE RATIO  =      10.00
    SECOND COMPRESSOR PRESSURE RATIO =       1.00
    COMPRESSOR POLYTROPIC EFFICIENCY =      0.900

 RECUPERATOR DESIGN INFORMATION
    RECUPERATOR EFFECTIVENESS =             0.0000
    MAX RECUPERATOR PRESSURE RATIO =        0.9900
    RECUPERATOR CONFIGURATION =                  1


    THE PRESSURE RATIO FOR HEATER 1 =       0.990
    THE PRESSURE RATIO FOR HEATER 2 =       0.990
    THE PRESSURE RATIO FOR HEATER 3 =       0.990
    THE PRESSURE RATIO FOR HEATER 4 =       0.990

 THE OPERATING ATMOSPHERIC DATA ARE
    ATMOSPHERIC PRESSURE =                0.10132 MPA       14.70 PSI
    ATMOSPHERIC TEMPERATURE =               288.0 K         518.4 R
    INLET WATER TEMPERATURE =               288.0 K         518.4 R
    EXHAUST PRESSURE RATIO =              0.9800

 THE HEAT RECOVERY STEAM GENERATOR DATA ARE
    SUPERHEATER PRESSURE RATIO =          0.9900
    EVAPORATOR PRESSURE RATIO =           0.9900
    ECONOMIZER PRESSURE RATIO =           0.9900
    CONDENSER PRESSURE RATIO =            0.9990
    PINCH POINT TEMP DIFF=                  10.0 K          18.0 R
    HRSG EXIT TEMP DIFF  =                  15.0 K          27.0 R
    PEAK HRSG PRESSURE   =              3.00000 MPA      435.23 PSI
    1ST REHEAT PRESSURE  =              0.75000 MPA      108.81 PSI
    2ND REHEAT PRESSURE  =              0.18750 MPA       27.20 PSI
    CONDENSER TEMPERATURE=                  40.0 C         104.0 F
    STEAM TURBINES EFFIC.=                0.900


    THE DESIGN POWER OUTPUT IS =            25.00 MW
    FLUID TEMPERATURE IN =                  973.0 K
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   THE FLUID MASS FLOW RATE IS =           40.00 KG/SEC
   THE FLUID PRESSURE IS =              1000.000 KPA,


   DESIGN RADIUS RATIO =                   0.600
   TURBINE BLADE SOLIDITY =                0.050
   COMPRESSOR INLET MACH NO. =             0.300
   COMPRESSOR EXIT MACH NO. =              0.100
   TURBINE 1 EXIT MACH NO. =               0.100
   TURBINE 2 EXIT MACH NO. =               0.100
   TURBINE 3 EXIT MACH NO. =               0.100
   TURBINE 4 EXIT MACH NO. =               0.100

MATERIAL IDENTIFIERS
   HEAT TRANSFER FLUID IDENTIFIER =            1
   AIR HEATER MATERIAL IDENTIFIER =           40
   INTERCOOLER MATERIAL IDENTIFIER =          20
   ECONOMIZER MATERIAL IDENTIFIER =           30
   EVAPORATOR MATERIAL IDENTIFIER =           30
   SUPERHEATER MATERIAL IDENTIFIER =          40
   CONDENSER MATERIAL IDENTIFIER =            60
   RECUPERATOR MATERIAL IDENTIFIER =          40

HEAT TRANSFER & FRICTION CORRELATIONS
   MAIN HEATER COLD CORRELATION =        27
   MAIN HEATER HOT CORRELATION =         27
   SUPERHEATER COLD CORRELATION =        27
   SUPERHEATER HOT CORRELATION =         27
   EVAPORATOR COLD CORRELATION =         27
   EVAPORATOR HOT CORRELATION =          27
   ECONOMIZER COLD CORRELATION =         27
   ECONOMIZER HOT CORRELATION =          27
   CONDENSER COLD CORRELATION =          27
   CONDENSER HOT CORRELATION =           27
   INTERCOOLER COLD CORRELATION =        27
   INTERCOOLER HOT CORRELATION =         27
   RECUPERATOR COLD CORRELATION =        25
   RECUPERATOR HOT CORRELATION =         25

THE NEW PRESSURE RATIO IS=    12.498


COMPRESSOR PERFORMANCE DATA
   OVERALL CPR =                           12.498
   COMPR EXIT PRESS=                       1.2664 MPA        183.72 PSI
   COMPR EXIT TEMP =                        629.4 K          1132.9 R
   COMPRESSOR WORK =                        352.2 KJ/KG      151.4 BTU/LBM
   COMPRESSOR ISENTROPIC EFFICIENCY=       0.8614

HEATER  1 AND TURBINE PERFORMANCE DATA
   1ST HTR EX PRESS=                       1.2537 MPA         181.89 PSI
   HEATR CP SPEC HT=                        1.094 KJ/KG/K      0.261 BTU/LBM/R
   1ST HEATR INPUT =                       332.12 KJ/KG       142.79 BTU/LBM
   TURBN CP SPEC HT=                        1.113 KJ/KG/K      0.266 BTU/LBM/R
   RATIO OF SP HEAT=                        1.348
   TURB EXIT PRESS =                       0.6821 MPA          98.95 PSI
   1ST TURBINE WORK=                        136.8 KJ/KG         58.8 BTU/LBM
   1ST TURBINE PRESSURE RATIO=             0.5440
   TURBINE ISENTROPIC EFFICIENCY=          0.9069

HEATER  2 AND TURBINE PERFORMANCE DATA
   HTR ENTER PRESS=                        0.6752 MPA          97.96 PSI
   HEATR CP SPEC HT=                        1.113 KJ/KG/K      0.266 BTU/LBM/R
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   HEAT INPUT =                            136.84 KJ/KG        58.83 BTU/LBM
   TURBINE CP SP HT=                        1.113 KJ/KG/K      0.266 BTU/LBM/R
   RATIO OF SP HEAT=                       1.348
   TURB EXIT PRESS =                       0.3673 MPA        53.29 PSI
   TURBINE WORK =                           136.8 KJ/KG        58.8 BTU/LBM
   TURBINE PRESSURE RATIO=                 0.5440
   TURBINE ISENTROPIC EFFICIENCY=          0.9069

HEATER  3 AND TURBINE PERFORMANCE DATA
   HTR ENTER PRESS=                        0.3637 MPA          52.76 PSI
   HEATR CP SPEC HT=                        1.113 KJ/KG/K      0.266 BTU/LBM/R
   HEAT INPUT =                            136.84 KJ/KG        58.83 BTU/LBM
   TURBINE CP SP HT=                        1.113 KJ/KG/K      0.266 BTU/LBM/R
   RATIO OF SP HEAT=                       1.348
   TURB EXIT PRESS =                       0.1978 MPA        28.70 PSI
   TURBINE WORK =                           136.8 KJ/KG        58.8 BTU/LBM
   TURBINE PRESSURE RATIO=                 0.5440
   TURBINE ISENTROPIC EFFICIENCY=          0.9069

HEATER  4 AND TURBINE PERFORMANCE DATA
   HTR ENTER PRESS=                        0.1959 MPA          28.42 PSI
   HEATR CP SPEC HT=                        1.113 KJ/KG/K      0.266 BTU/LBM/R
   HEAT INPUT =                            136.84 KJ/KG        58.83 BTU/LBM
   TURBINE CP SP HT=                        1.113 KJ/KG/K      0.266 BTU/LBM/R
   RATIO OF SP HEAT=                       1.348
   TURB EXIT PRESS =                       0.1066 MPA        15.46 PSI
   TURBINE WORK =                           136.8 KJ/KG        58.8 BTU/LBM
   TURBINE PRESSURE RATIO=                 0.5440
   TURBINE ISENTROPIC EFFICIENCY=          0.9069


BRAYTON SYSTEM PERFORMANCE
   TOTAL HEAT INPUT=                        742.7 KJ/KG       319.3 BTU/LBM
   TOTAL TURB WORK =                        547.4 KJ/KG       235.3 BTU/LBM
   COMPRESSOR WORK =                        352.2 KJ/KG       151.4 BTU/LBM
   NET SYSTEM WORK =                        195.2 KJ/KG        83.9 BTU/LBM
   BRAYTON SYS EFF =                       0.2628


HRSG PUMP PERFORMANCE
   PUMP INLET ENTHALPY     =                166.9 KJ/KG
   CONDENSER EXIT PRESSURE =               7.3256 KPAS
   PUMP WORK =                                3.0 KJ/KG
   PUMP EXIT ENTHALPY =                     169.9 KJ/KG

PINCH POINT STATE DATA
   WATER TEMPERATURE AT SATURATION =           507.0 K
   PINCH POINT DELTA TEMPERATURE =              10.0 K
   AIR TEMPERATURE AT PINCH POINT =            517.0 K
   TOTAL WATER ENTHALPY REQ TO REACH SAT =     838.4 KJ/KGW
   WATER TO AIR FLOW RATIO =                  0.0949 KGW/KGA

HIGH PRESSURE TURBINE PERFORMANCE
   THROTTLE TEMPERATURE FOR HP TURBINE =     795.0 K
   THROTTLE ENTHALPY FOR HP TURBINE =       3506.2 KJ/KG
   HEAT INPUT TO HP TURBINE ABOVE PP =      1794.9 KJ/KG
   STEAM ENTROPY AT THROTTLE POINT =          7.30 KJ/KG/K
   QUALITY AT END OF TURBINE EXPANSION =     1.000
   ACTUAL HP TURBINE WORK =                  386.4 KJ/KG
   ACTUAL ENTHALPY AT END OF EXPANSION =    3119.8 KJ/KG

MEDIUM PRESSURE TURBINE PERFORMANCE
   THROTTLE ENTHALPY FOR MP TURBINE =       3529.0 KJ/KG
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   HEAT INPUT FOR MP TURBINE =                 702.9 KJ/KG
   STEAM ENTROPY AT THROTTLE POINT =            7.96 KJ/KG/K
   QUALITY AT END OF TURBINE EXPANSION =       1.000
   ACTUAL MP TURBINE WORK =                    392.5 KJ/KG
   ACTUAL ENTHALPY AT END OF EXPANSION =      3136.5 KJ/KG

LOW PRESSURE TURBINE PERFORMANCE
   THROTTLE ENTHALPY FOR LP TURBINE =         3534.6 KJ/KG
   HEAT INPUT TO LP TURBINE =                  409.2 KJ/KG
   STEAM ENTROPY AT THROTTLE POINT =            8.60 KJ/KG/K
   QUALITY AT END OF TURBINE EXPANSION =       1.000
   ACTUAL LP TURBINE WORK =                    758.3 KJ/KG
   ACTUAL ENTHALPY AT END OF EXPANSION =      2776.4 KJ/KG


OVERALL SYSTEM PERFORMANCE
   CHANGE IN WATER ENTHALPY ABOVE PP =        3305.2 KJ/KGW
   CHANGE IN AIR ENTHALPY ABOVE PP =           313.6 KJ/KGA
   MASS FLOW RATIO - WATER/AIR =             0.09488 KGW/KGA
   HEAT EXTRACTED BY RANKINE CYCLE =          4143.6 KJ/KGW
   WORK PERFORMED BY RANKINE CYCLE =          1534.1 KJ/KGW
   RANKINE CYCLE EFFICIENCY =                 0.3702
   BRAYTON CYCLE EFFICIENCY =                 0.2628
   OVERALL CYCLE EFFICIENCY =                 0.4588



SIZING DATA
   TOTAL SYSTEM POWER =                      25.000 MW
   BRAYTON SYSTEM POWER =                    14.308 MW
   RANKINE SYSTEM POWER =                    10.692 MW
   TOTAL HEAT INPUT =                        54.442 MW
   MASS FLOW RATE(AIR) =                     73.308 KGM/SEC
   MASS FLOW RATE(WATER) =                    6.956 KGM/SEC
   TPRIME(WILSON)=                            3.240


HEATER SIZING DATA
   HEAT INPUT FOR HEATER NO. 1 =             24.347 MW
   HEAT INPUT FOR HEATER NO. 2 =             10.032 MW
   HEAT INPUT FOR HEATER NO. 3 =             10.032 MW
   HEAT INPUT FOR HEATER NO. 4 =             10.032 MW

COMPRESSOR SIZING DATA
   MACH NUMBER INTO COMPRESSOR =             0.3000
   COMPRESSOR INLET AREA =                   0.8064 M**2
   COMPRESSOR OUTER RADIUS =                 0.6498 M
   MACH NUMBER AT COMPRESSOR EXIT =          0.1000
   COMPRESSOR EXIT AREA =                    0.2761 M**2
   NUMBER OF STAGES =                             7
   COMPRESSOR VOLUME =                        0.965 M**3
   ESTIMATED POLYTROPIC EFFICIENCY=          0.9130


GAS TURBINE SIZING DATA
   DATA FOR AIR TURBINE NUMBER         1
   AIR VELOCITY AT INLET =                    599.7 M/SEC
   AIR TURBINE INLET AREA =                  0.0585 M**2
   AIR TURBINE OUTER RADIUS =                 0.321 M
   EXIT MACH NUMBER =                         0.100
   AIR VELOCITY AT EXIT =                      56.1 M/SEC
   AIR TURBINE EXIT AREA =                    0.585 M**2
   AIR TURBINE EXIT RADIUS =                  0.321 M
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AIR TURBINE NO OF STAGES =                   3
AIR TURBINE VOLUME =                     0.009 M**3
ESTIMATED POLYTROPIC EFFICIENCY=        0.9010

DATA FOR AIR TURBINE NUMBER         2
AIR VELOCITY AT INLET =                  599.7 M/SEC
AIR TURBINE INLET AREA =                0.1087 M**2
AIR TURBINE OUTER RADIUS =               0.438 M
EXIT MACH NUMBER =                       0.100
AIR VELOCITY AT EXIT =                    56.1 M/SEC
AIR TURBINE EXIT AREA =                  1.087 M**2
AIR TURBINE EXIT RADIUS =                0.438 M
AIR TURBINE NO OF STAGES =                   3
AIR TURBINE VOLUME =                     0.023 M**3
ESTIMATED POLYTROPIC EFFICIENCY=        0.9103

DATA FOR AIR TURBINE NUMBER         3
AIR VELOCITY AT INLET =                  599.7 M/SEC
AIR TURBINE INLET AREA =                0.2018 M**2
AIR TURBINE OUTER RADIUS =               0.597 M
EXIT MACH NUMBER =                       0.100
AIR VELOCITY AT EXIT =                    56.1 M/SEC
AIR TURBINE EXIT AREA =                  2.017 M**2
AIR TURBINE EXIT RADIUS =                0.597 M
AIR TURBINE NO OF STAGES =                   3
AIR TURBINE VOLUME =                     0.058 M**3
ESTIMATED POLYTROPIC EFFICIENCY=        0.9196

DATA FOR AIR TURBINE NUMBER         4
AIR VELOCITY AT INLET =                  599.7 M/SEC
AIR TURBINE INLET AREA =                0.3747 M**2
AIR TURBINE OUTER RADIUS =               0.813 M
EXIT MACH NUMBER =                       0.100
AIR VELOCITY AT EXIT =                    56.1 M/SEC
AIR TURBINE EXIT AREA =                  3.746 M**2
AIR TURBINE EXIT RADIUS =                0.813 M
AIR TURBINE NO OF STAGES =                   3
AIR TURBINE VOLUME =                     0.146 M**3
ESTIMATED POLYTROPIC EFFICIENCY=        0.9288

DATA FOR HIGH PRESSURE STEAM TURBINE
HP STEAM VELOCITY AT INLET =             679.5 M/SEC
HP STEAM TURBINE INLET AREA =           0.0028 M**2
HP STEAM TURBINE OUTER RADIUS =          0.070 M
HP STEAM EXIT MACH NUMBER =              0.100
HP STEAM VELOCITY AT EXIT =               58.7 M/SEC
HP STEAM TURBINE EXIT AREA =             0.056 M**2
HP STEAM TURBINE EXIT RADIUS =           0.171 M
HP STEAM TURBINE PRESSURE RATIO =        4.000
HP STEAM TURBINE NUMBER OF STAGES =          7
HP STEAM TURBINE VOLUME =                0.002 M**3

DATA FOR MEDIUM PRESSURE STEAM TURBINE
MP STEAM VELOCITY AT INLET =             682.7 M/SEC
MP STEAM TURBINE INLET AREA =           0.0111 M**2
MP STEAM TURBINE OUTER RADIUS =          0.140 M
MP STEAM EXIT MACH NUMBER =              0.100
MP STEAM VELOCITY AT EXIT =               59.0 M/SEC
MP STEAM TURBINE EXIT AREA =             0.222 M**2
MP STEAM TURBINE EXIT RADIUS =           0.341 M
MP STEAM TURBINE PRESSURE RATIO =        4.000
MP STEAM TURBINE NUMBER OF STAGES =          7
MP STEAM TURBINE VOLUME =                0.014 M**3
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   DATA FOR LOW PRESSURE STEAM TURBINE
   LP STEAM VELOCITY AT INLET =            683.5 M/SEC
   LP STEAM TURBINE INLET AREA =          0.0443 M**2
   LP STEAM TURBINE OUTER RADIUS =         0.279 M
   LP STEAM TURBINE EXIT MACH NUMBER =     0.100
   LP STEAM VELOCITY AT EXIT =              47.8 M/SEC
   LP STEAM TURBINE EXIT AREA =            4.517 M**2
   LP STEAM TURBINE EXIT RADIUS =          1.538 M
   LP STEAM TURBINE PRESSURE RATIO =      25.595
   LP STEAM TURBINE NUMBER OF STAGES =        15
   LP STEAM TURBINE VOLUME =               2.092 M**3


DESIGN PARAMETERS FOR HEAT EXCHANGER  1

   AIR TEMPERATURE AT INLET =              629.4 K
   AIR PRESSURE AT INLET =                1.2664 MPA
   AIR TEMPERATURE AT EXIT =               933.0 K
   AIR MASS FLOW RATE =                    73.31 KG/SEC
   AIR SIDE PRESSURE RATIO =               0.990
   TOTAL HEAT TRANSFER RATE =            24372.0 KW
   FLUID TEMPERATURE AT INLET =            973.0 K
   INPUT FLUID MASS FLOW RATE=             40.00 KG/SEC
   FLUID MASS FLOW RATE =                 105.79 KG/SEC
   FLUID TEMPERATURE AT EXIT =             858.5 K
   LM AVERAGE AIR TEMPERATURE =            737.7 K
   AIR PRANDTL NUMBER =                    0.703
   AIR VISCOSITY =                      0.000035 PAS-SEC
   AIR SPECIFIC HEAT =                     1.086 KJ/KG/K
   AIR THERMAL CONDUCTIVITY =               0.00 KW/M/K
   AVERAGE FLUID TEMPERATURE =             915.7 K
   FLUID THERMAL CONDUCTIVITY =           0.0009 KW/M/K
   FLUID VISCOSITY =                    3.80E-03 PAS-SEC
   FLUID SPECIFIC HEAT =                   1.951 KJ/KG/K
   FLUID PRANDTL NUMBER =                  8.450
   FLUID SPECIFIC VOLUME =              4.85E-04 M**3/KG

FLUID HEAT TRANSFER PROPERTIES
   AIR MASS FLOW RATE/AREA =               43.49 KG/SEC/M**2
   AIR REYNOLDS NUMBER =                  3880.5
   FLUID MASS FLOW RATE/AREA =             62.76 KG/SEC/M**2
   FLUID REYNOLDS NUMBER =                  50.9
   AIR STANTON NUMBER =                  0.00939
   AIR SIDE FRICTION FACTOR =            0.03026
   FLUID STANTON NUMBER =                0.01202
   FLUID FRICTION FACTOR =               0.39820
   AIR FILM HEAT TRANSFER COEF  =           0.44 KW/M**2/K
   FLUID FILM HEAT TRANSFER COEF =          1.47 KW/M**2/K
   FIN EFFECTIVENESS =                    0.3410
   OVERALL SURFACE EFFECTIVENESS =        0.5018
   OVERALL HEAT TRANSFER COEF =            214.1 W/M**2/K
   AIR HEAT CAPACITY RATE =                 79.6 KW/K
   FLUID HEAT CAPACITY RATE =              206.3 KW/K
   MINIMUM HEAT CAPACITY RATE =             79.6 KW/K
   MAXIMUM HEAT CAPACITY RATE =            206.3 KW/K
   NUMBER OF HEAT TRANSFER UNITS =         2.920
   HEAT CAPACITY RATE RATIO =              0.386
   HEAT EXCHANGER EFFICIENCY =             0.891
   HEATER EXIT TEMPERATURE =               935.5 K
   AIR INLET SPECIFIC VOLUME =          1.43E-01 M**3/KG
   AIR EXIT SPECIFIC VOLUME =           2.14E-01 M**3/KG
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   AVERAGE AIR SPECIFIC VOLUME =         1.68E-01 M**3/KG
   FLUID PRESSURE DROP =                 1.23E+02 PAS
   REQUIRED FLUID PUMPING POWER =        6.29E-03 KW
   HEATER AIR PRESSURE DROP =            1.27E+04 PAS
   HEATER AIR PRESSURE RATIO =             0.9900
   HEATER VOLUME =                          1.946 M**3
   HEATER FRONTAL AREA=                     0.979 M**2
   HEATER DEPTH =                           1.987 M
   HEATER WIDTH =                           0.990 M
   HEATER HEIGHT =                          0.990 M



DESIGN PARAMETERS FOR HEAT EXCHANGER  2

   AIR TEMPERATURE AT INLET =               810.0 K
   AIR PRESSURE AT INLET =                 0.6821 MPA
   AIR TEMPERATURE AT EXIT =                933.0 K
   AIR MASS FLOW RATE =                     73.31 KG/SEC
   AIR SIDE PRESSURE RATIO =                0.990
   TOTAL HEAT TRANSFER RATE =             10033.8 KW
   FLUID TEMPERATURE AT INLET =             973.0 K
   INPUT FLUID MASS FLOW RATE=             105.79 KG/SEC
   FLUID MASS FLOW RATE =                  105.79 KG/SEC
   FLUID TEMPERATURE AT EXIT =              925.8 K
   LM AVERAGE AIR TEMPERATURE =             881.3 K
   AIR PRANDTL NUMBER =                     0.701
   AIR VISCOSITY =                       0.000039 PAS-SEC
   AIR SPECIFIC HEAT =                      1.115 KJ/KG/K
   AIR THERMAL CONDUCTIVITY =                0.00 KW/M/K
   AVERAGE FLUID TEMPERATURE =              949.4 K
   FLUID THERMAL CONDUCTIVITY =            0.0009 KW/M/K
   FLUID VISCOSITY =                     3.23E-03 PAS-SEC
   FLUID SPECIFIC HEAT =                    1.986 KJ/KG/K
   FLUID PRANDTL NUMBER =                   7.210
   FLUID SPECIFIC VOLUME =               4.91E-04 M**3/KG

FLUID HEAT TRANSFER PROPERTIES
   AIR MASS FLOW RATE/AREA =                29.17 KG/SEC/M**2
   AIR REYNOLDS NUMBER =                   2327.3
   FLUID MASS FLOW RATE/AREA =              42.09 KG/SEC/M**2
   FLUID REYNOLDS NUMBER =                   40.2
   AIR STANTON NUMBER =                   0.01118
   AIR SIDE FRICTION FACTOR =             0.03551
   FLUID STANTON NUMBER =                 0.01495
   FLUID FRICTION FACTOR =                0.47093
   AIR FILM HEAT TRANSFER COEF  =            0.36 KW/M**2/K
   FLUID FILM HEAT TRANSFER COEF =           1.25 KW/M**2/K
   FIN EFFECTIVENESS =                     0.3781
   OVERALL SURFACE EFFECTIVENESS =         0.5298
   OVERALL HEAT TRANSFER COEF =             187.5 W/M**2/K
   AIR HEAT CAPACITY RATE =                  81.7 KW/K
   FLUID HEAT CAPACITY RATE =               210.1 KW/K
   MINIMUM HEAT CAPACITY RATE =              81.7 KW/K
   MAXIMUM HEAT CAPACITY RATE =             210.1 KW/K
   NUMBER OF HEAT TRANSFER UNITS =          1.723
   HEAT CAPACITY RATE RATIO =               0.389
   HEAT EXCHANGER EFFICIENCY =              0.753
   HEATER EXIT TEMPERATURE =                932.8 K
   AIR INLET SPECIFIC VOLUME =           3.41E-01 M**3/KG
   AIR EXIT SPECIFIC VOLUME =            3.97E-01 M**3/KG
   AVERAGE AIR SPECIFIC VOLUME =         3.73E-01 M**3/KG
   FLUID PRESSURE DROP =                 3.06E+01 PAS
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   REQUIRED FLUID PUMPING POWER =        1.59E-03 KW
   HEATER AIR PRESSURE DROP =            6.82E+03 PAS
   HEATER AIR PRESSURE RATIO =             0.9900
   HEATER VOLUME =                          1.345 M**3
   HEATER FRONTAL AREA=                     1.460 M**2
   HEATER DEPTH =                           0.921 M
   HEATER WIDTH =                           1.208 M
   HEATER HEIGHT =                          1.208 M


DESIGN PARAMETERS FOR HEAT EXCHANGER  3

   AIR TEMPERATURE AT INLET =               810.0 K
   AIR PRESSURE AT INLET =                 0.3673 MPA
   AIR TEMPERATURE AT EXIT =                933.0 K
   AIR MASS FLOW RATE =                     73.31 KG/SEC
   AIR SIDE PRESSURE RATIO =                0.990
   TOTAL HEAT TRANSFER RATE =             10033.8 KW
   FLUID TEMPERATURE AT INLET =             973.0 K
   INPUT FLUID MASS FLOW RATE=             105.79 KG/SEC
   FLUID MASS FLOW RATE =                  105.79 KG/SEC
   FLUID TEMPERATURE AT EXIT =              925.8 K
   LM AVERAGE AIR TEMPERATURE =             881.3 K
   AIR PRANDTL NUMBER =                     0.701
   AIR VISCOSITY =                       0.000039 PAS-SEC
   AIR SPECIFIC HEAT =                      1.115 KJ/KG/K
   AIR THERMAL CONDUCTIVITY =                0.00 KW/M/K
   AVERAGE FLUID TEMPERATURE =              949.4 K
   FLUID THERMAL CONDUCTIVITY =            0.0009 KW/M/K
   FLUID VISCOSITY =                     3.23E-03 PAS-SEC
   FLUID SPECIFIC HEAT =                    1.986 KJ/KG/K
   FLUID PRANDTL NUMBER =                   7.210
   FLUID SPECIFIC VOLUME =               4.91E-04 M**3/KG

FLUID HEAT TRANSFER PROPERTIES
   AIR MASS FLOW RATE/AREA =                16.16 KG/SEC/M**2
   AIR REYNOLDS NUMBER =                   1289.3
   FLUID MASS FLOW RATE/AREA =              23.32 KG/SEC/M**2
   FLUID REYNOLDS NUMBER =                   22.2
   AIR STANTON NUMBER =                   0.01369
   AIR SIDE FRICTION FACTOR =             0.04537
   FLUID STANTON NUMBER =                 0.01975
   FLUID FRICTION FACTOR =                0.71411
   AIR FILM HEAT TRANSFER COEF  =            0.25 KW/M**2/K
   FLUID FILM HEAT TRANSFER COEF =           0.91 KW/M**2/K
   FIN EFFECTIVENESS =                     0.4367
   OVERALL SURFACE EFFECTIVENESS =         0.5741
   OVERALL HEAT TRANSFER COEF =             140.6 W/M**2/K
   AIR HEAT CAPACITY RATE =                  81.7 KW/K
   FLUID HEAT CAPACITY RATE =               210.1 KW/K
   MINIMUM HEAT CAPACITY RATE =              81.7 KW/K
   MAXIMUM HEAT CAPACITY RATE =             210.1 KW/K
   NUMBER OF HEAT TRANSFER UNITS =          1.723
   HEAT CAPACITY RATE RATIO =               0.389
   HEAT EXCHANGER EFFICIENCY =              0.753
   HEATER EXIT TEMPERATURE =                932.8 K
   AIR INLET SPECIFIC VOLUME =           6.33E-01 M**3/KG
   AIR EXIT SPECIFIC VOLUME =            7.37E-01 M**3/KG
   AVERAGE AIR SPECIFIC VOLUME =         6.92E-01 M**3/KG
   FLUID PRESSURE DROP =                 1.05E+01 PAS
   REQUIRED FLUID PUMPING POWER =        5.47E-04 KW
   HEATER AIR PRESSURE DROP =            3.67E+03 PAS
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   HEATER AIR PRESSURE RATIO =            0.9900
   HEATER VOLUME =                         1.795 M**3
   HEATER FRONTAL AREA=                    2.636 M**2
   HEATER DEPTH =                          0.681 M
   HEATER WIDTH =                          1.624 M
   HEATER HEIGHT =                         1.624 M



DESIGN PARAMETERS FOR HEAT EXCHANGER  4

   AIR TEMPERATURE AT INLET =              810.0 K
   AIR PRESSURE AT INLET =                0.1978 MPA
   AIR TEMPERATURE AT EXIT =               933.0 K
   AIR MASS FLOW RATE =                    73.31 KG/SEC
   AIR SIDE PRESSURE RATIO =               0.990
   TOTAL HEAT TRANSFER RATE =            10033.8 KW
   FLUID TEMPERATURE AT INLET =            973.0 K
   INPUT FLUID MASS FLOW RATE=            105.79 KG/SEC
   FLUID MASS FLOW RATE =                 105.79 KG/SEC
   FLUID TEMPERATURE AT EXIT =             925.8 K
   LM AVERAGE AIR TEMPERATURE =            881.3 K
   AIR PRANDTL NUMBER =                    0.701
   AIR VISCOSITY =                      0.000039 PAS-SEC
   AIR SPECIFIC HEAT =                     1.115 KJ/KG/K
   AIR THERMAL CONDUCTIVITY =               0.00 KW/M/K
   AVERAGE FLUID TEMPERATURE =             949.4 K
   FLUID THERMAL CONDUCTIVITY =           0.0009 KW/M/K
   FLUID VISCOSITY =                    3.23E-03 PAS-SEC
   FLUID SPECIFIC HEAT =                   1.986 KJ/KG/K
   FLUID PRANDTL NUMBER =                  7.210
   FLUID SPECIFIC VOLUME =              4.91E-04 M**3/KG

FLUID HEAT TRANSFER PROPERTIES
   AIR MASS FLOW RATE/AREA =                8.77 KG/SEC/M**2
   AIR REYNOLDS NUMBER =                   700.2
   FLUID MASS FLOW RATE/AREA =             12.66 KG/SEC/M**2
   FLUID REYNOLDS NUMBER =                  12.1
   AIR STANTON NUMBER =                  0.01829
   AIR SIDE FRICTION FACTOR =            0.06340
   FLUID STANTON NUMBER =                0.02634
   FLUID FRICTION FACTOR =               1.09814
   AIR FILM HEAT TRANSFER COEF  =           0.18 KW/M**2/K
   FLUID FILM HEAT TRANSFER COEF =          0.66 KW/M**2/K
   FIN EFFECTIVENESS =                    0.5022
   OVERALL SURFACE EFFECTIVENESS =        0.6237
   OVERALL HEAT TRANSFER COEF =            108.5 W/M**2/K
   AIR HEAT CAPACITY RATE =                 81.7 KW/K
   FLUID HEAT CAPACITY RATE =              210.1 KW/K
   MINIMUM HEAT CAPACITY RATE =             81.7 KW/K
   MAXIMUM HEAT CAPACITY RATE =            210.1 KW/K
   NUMBER OF HEAT TRANSFER UNITS =         1.723
   HEAT CAPACITY RATE RATIO =              0.389
   HEAT EXCHANGER EFFICIENCY =             0.753
   HEATER EXIT TEMPERATURE =               932.8 K
   AIR INLET SPECIFIC VOLUME =          1.18E+00 M**3/KG
   AIR EXIT SPECIFIC VOLUME =           1.37E+00 M**3/KG
   AVERAGE AIR SPECIFIC VOLUME =        1.29E+00 M**3/KG
   FLUID PRESSURE DROP =                3.36E+00 PAS
   REQUIRED FLUID PUMPING POWER =       1.74E-04 KW
   HEATER AIR PRESSURE DROP =           1.98E+03 PAS
   HEATER AIR PRESSURE RATIO =            0.9900
   HEATER VOLUME =                         2.325 M**3
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HEATER FRONTAL AREA=	4.854 M**2
HEATER DEPTH =	0.479 M
HEATER WIDTH =	2.203 M
HEATER HEIGHT =	2.203 M

THREE SUPERHEATER DESIGN

FIRST SUPERHEATER DESIGN

MASS FLOW RATE OF AIR =	34.12 KGA/SEC
AIR TEMPERATURE INTO SUPERHEATER =	810.0 K
AIR TEMPERATURE OUT OF SUPERHEATER =	670.6 K
AVERAGE AIR TEMPERATURE =	740.3 K
AIR SPECIFIC HEAT =	1.09 KJ/KG/K
AIR VISCOSITY =	3.46E-05 PAS-SEC
AIR THERMAL CONDUCTIVITY =	5.36E-05 KW/M/K
AIR PRANDTL NUMBER =	0.7028
AIR HEAT CAPACITY RATE =	37.08 KW/K
MASS FLOW RATE OF STEAM =	6.9557 KGW/SEC
STEAM TEMPERATURE INTO SUPERHEATER =	507.0 K
AVERAGE STEAM TEMPERATURE =	569.2 K
STEAM SPECIFIC HEAT =	2.44 KJ/KG/K
STEAM VISCOSITY =	1.95E-05 PAS-SEC
STEAM THERMAL CONDUCTIVITY =	3.96E-05 KW/M/K
STEAM PRANDTL NUMBER =	1.2052
STEAM HEAT CAPACITY RATE =	16.98 KW/K
SUPERHEATER EFFECTIVENESS =	0.9505
HEAT CAPACITY RATE RATIO =	0.4579
AIR MASS FLOW RATE PER UNIT AREA =	3.085 KG/S/M**2
AIR REYNOLDS NUMBER =	274.7
AIR STANTON NUMBER =	0.02852
AIR FRICTION FACTOR =	0.12141
AIR HEAT TRANSFER COEFFICIENT =	0.0956 KW/M**2/K
AIR HYDRAULIC DIAMETER =	0.0031 M
SUPERHEATER EXIT AIR TEMPERATURE =	678.1 K
STEAM MASS FLOW RATE PER UNIT AREA =	0.629 CU.M/S
STEAM REYNOLDS NUMBER =	99.3
STEAM STANTON NUMBER =	0.03216
STEAM FRICTION FACTOR =	0.24871
STEAM HEAT TRANSFER COEFFICIENT =	0.0494 KW/M**2/K
STEAM HYDRAULIC DIAMETER =	0.0031 M
SUPERHEATER EXIT STEAM TEMPERATURE =	795.0 K
REQUIRED HEAT TRANSFER UNITS =	4.490
STEAM PRESSURE DROP =	4.3 PAS
STEAM PRESSURE RATIO =	1.0000
AIR PRESSURE DROP =	1065.6 PAS
AIR SUPERHEATER PRESSURE RATIO =	0.9900
SUPERHEATER VOLUME REQUIRED =	4.611 M**3
SUPERHEATER FRONTAL AREA =	6.426 M**2
SUPERHEATER DEPTH =	0.718 M
SUPERHEATER HEIGHT =	2.535 M
SUPERHEATER WIDTH =	2.535 M

SECOND SUPERHEATER DESIGN

MASS FLOW RATE OF AIR =	19.86 KGA/SEC
AIR TEMPERATURE INTO SUPERHEATER =	810.0 K
AIR TEMPERATURE OUT OF SUPERHEATER =	670.6 K
AVERAGE AIR TEMPERATURE =	740.3 K
AIR SPECIFIC HEAT =	1.09 KJ/KG/K

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 AIR VISCOSITY =                       3.46E-05 PAS-SEC
 AIR THERMAL CONDUCTIVITY =            5.36E-05 KW/M/K
 AIR PRANDTL NUMBER =                    0.7028
 AIR HEAT CAPACITY RATE =                 21.59 KW/K
 MASS FLOW RATE OF STEAM =               6.9557 KGW/SEC
 STEAM TEMPERATURE INTO SUPERHEATER =     601.8 K
 AVERAGE STEAM TEMPERATURE =              637.1 K
 STEAM SPECIFIC HEAT =                     2.12 KJ/KG/K
 STEAM VISCOSITY =                     2.20E-05 PAS-SEC
 STEAM THERMAL CONDUCTIVITY =          4.53E-05 KW/M/K
 STEAM PRANDTL NUMBER =                  1.0324
 STEAM HEAT CAPACITY RATE =               14.77 KW/K
 SUPERHEATER EFFECTIVENESS =             0.9280
 HEAT CAPACITY RATE RATIO =              0.6841
 AIR MASS FLOW RATE PER UNIT AREA =       2.526 KG/S/M**2
 AIR REYNOLDS NUMBER =                    224.9
 AIR STANTON NUMBER =                   0.03134
 AIR FRICTION FACTOR =                  0.13978
 AIR HEAT TRANSFER COEFFICIENT =         0.0860 KW/M**2/K
 AIR HYDRAULIC DIAMETER =                0.0031 M
 SUPERHEATER EXIT AIR TEMPERATURE =       677.8 K
 STEAM MASS FLOW RATE PER UNIT AREA =     0.885 CU.M/S
 STEAM REYNOLDS NUMBER =                  123.8
 STEAM STANTON NUMBER =                 0.03214
 STEAM FRICTION FACTOR =                0.21293
 STEAM HEAT TRANSFER COEFFICIENT =       0.0604 KW/M**2/K
 STEAM HYDRAULIC DIAMETER =              0.0031 M
 SUPERHEATER EXIT STEAM TEMPERATURE =     795.0 K
 REQUIRED HEAT TRANSFER UNITS =           5.138
 STEAM PRESSURE DROP =                     42.2 PAS
 STEAM PRESSURE RATIO =                  0.9999
 AIR PRESSURE DROP =                     1065.6 PAS
 AIR SUPERHEATER PRESSURE RATIO =        0.9900
 SUPERHEATER VOLUME REQUIRED =            4.238 M**3
 SUPERHEATER FRONTAL AREA =               4.569 M**2
 SUPERHEATER DEPTH =                      0.928 M
 SUPERHEATER HEIGHT =                     2.137 M
 SUPERHEATER WIDTH =                      2.137 M


THIRD SUPERHEATER DESIGN

 MASS FLOW RATE OF AIR =                  19.32 KGA/SEC
 AIR TEMPERATURE INTO SUPERHEATER =       810.0 K
 AIR TEMPERATURE OUT OF SUPERHEATER =     670.6 K
 AVERAGE AIR TEMPERATURE =                740.3 K
 AIR SPECIFIC HEAT =                       1.09 KJ/KG/K
 AIR VISCOSITY =                       3.46E-05 PAS-SEC
 AIR THERMAL CONDUCTIVITY =            5.36E-05 KW/M/K
 AIR PRANDTL NUMBER =                    0.7028
 AIR HEAT CAPACITY RATE =                 21.00 KW/K
 MASS FLOW RATE OF STEAM =               6.9557 KGW/SEC
 STEAM TEMPERATURE INTO SUPERHEATER =     562.5 K
 AVERAGE STEAM TEMPERATURE =              609.6 K
 STEAM SPECIFIC HEAT =                     2.09 KJ/KG/K
 STEAM VISCOSITY =                     2.10E-05 PAS-SEC
 STEAM THERMAL CONDUCTIVITY =          4.30E-05 KW/M/K
 STEAM PRANDTL NUMBER =                  1.0210
 STEAM HEAT CAPACITY RATE =               14.51 KW/K
 SUPERHEATER EFFECTIVENESS =             0.9394
 HEAT CAPACITY RATE RATIO =              0.6910
 AIR MASS FLOW RATE PER UNIT AREA =       2.363 KG/S/M**2
 AIR REYNOLDS NUMBER =                    210.4
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   AIR STANTON NUMBER =                    0.03234
   AIR FRICTION FACTOR =                   0.14653
   AIR HEAT TRANSFER COEFFICIENT =          0.0830 KW/M**2/K
   AIR HYDRAULIC DIAMETER =                 0.0031 M
   SUPERHEATER EXIT AIR TEMPERATURE =        649.3 K
   STEAM MASS FLOW RATE PER UNIT AREA =      0.850 CU.M/S
   STEAM REYNOLDS NUMBER =                   124.6
   STEAM STANTON NUMBER =                  0.03228
   STEAM FRICTION FACTOR =                 0.21196
   STEAM HEAT TRANSFER COEFFICIENT =        0.0573 KW/M**2/K
   STEAM HYDRAULIC DIAMETER =               0.0031 M
   SUPERHEATER EXIT STEAM TEMPERATURE =      795.0 K
   REQUIRED HEAT TRANSFER UNITS =            5.683
   STEAM PRESSURE DROP =                     170.8 PAS
   STEAM PRESSURE RATIO =                   0.9991
   AIR PRESSURE DROP =                      1065.7 PAS
   AIR SUPERHEATER PRESSURE RATIO =         0.9900
   SUPERHEATER VOLUME REQUIRED =             4.805 M**3
   SUPERHEATER FRONTAL AREA =                4.753 M**2
   SUPERHEATER DEPTH =                       1.011 M
   SUPERHEATER HEIGHT =                      2.180 M
   SUPERHEATER WIDTH =                       2.180 M


EVAPORATOR DESIGN
   MASS FLOW RATE OF AIR =                   73.31 KGA/SEC
   EVAPORATOR AIR INLET TEMPERATURE =        670.6 K
   AVERAGE EVAPORATOR AIR TEMPERATURE =      593.8 K
   AIR SPECIFIC HEAT =                        1.06 KJ/KG/K
   AVERAGE AIR VISCOSITY =               2.99E-05 PAS-SEC
   AVERAGE AIR THERMAL CONDUCTIVITY =    4.50E-05 KW/M/K
   AIR PRANDTL NUMBER =                     0.7022
   AIR INLET SPECIFIC VOLUME =                1.82 M**3/KGA
   AIR AVERAGE SPECIFIC VOLUME =              1.62 M**3/KGA
   AIR EXIT SPECIFIC VOLUME =                 1.41 M**3/KGA
   AIR MASS FLOW RATE PER UNIT AREA =        5.550 KG/S/M**2
   AIR REYNOLDS NUMBER =                     571.6
   AIR STANTON NUMBER =                    0.02020
   AIR FRICTION FACTOR =                   0.07244
   AIR HEAT TRANSFER COEFFICIENT =            0.12 KW/M**2/K
   OVERALL HEAT TRANSFER COEFFICIENT =       65.42 KW/M**2/K
   MASS FLOW RATE OF WATER =                  6.96 KGW/SEC
   EVAPORATOR WATER INLET TEMPERATURE =      507.0 K
   EVAPORATOR EFFECTIVENESS =                0.950
   NUMBER OF REQ HEAT TRANSFER UNITS =       2.996
   EVAPORATOR AIR PRESSURE DROP =           1054.9 PAS
   EVAPORATOR AIR PRESSURE RATIO =          0.9900
   REQUIRED EVAPORATOR VOLUME =              3.545 M**3
   REQUIRED EVAPORATOR FRONTAL AREA =        7.673 M**2
   REQUIRED EVAPORATOR DEPTH =               0.462 M
   REQUIRED EVAPORATOR WIDTH =               2.770 M
   REQUIRED EVAPORATOR HEIGHT =              2.770 M


ECONOMIZER DESIGN
   MASS FLOW RATE OF AIR =                   73.31 KGA/SEC
   ECONOMIZER AIR INLET TEMPERATURE =        517.0 K
   AVERAGE ECONOMIZER AIR TEMPERATURE =      477.9 K
   ECONOMIZER EXIT AIR TEMPERATURE =         438.7 K
   AIR SPECIFIC HEAT =                        1.04 KJ/KG/K
   AVERAGE AIR VISCOSITY =               2.59E-05 PAS-SEC
   AVERAGE AIR THERMAL CONDUCTIVITY =    3.80E-05 KW/M/K
   AIR PRANDTL NUMBER =                     0.7083
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   AIR INLET SPECIFIC VOLUME =               1.42 M**3/KGA
   AIR AVERAGE SPECIFIC VOLUME =             1.32 M**3/KGA
   AIR EXIT SPECIFIC VOLUME =                1.22 M**3/KGA
   AIR MASS FLOW RATE PER UNIT AREA =       6.807 KG/S/M**2
   AIR REYNOLDS NUMBER =                    810.5
   AIR STANTON NUMBER =                   0.01688
   AIR FRICTION FACTOR =                  0.05777
   AIR HEAT TRANSFER COEFFICIENT =           0.12 KW/M**2/K
   MASS FLOW RATE OF WATER =                 6.96 KGW/SEC
   ECONOMIZER WATER INLET TEMPERATURE =     313.7 K
   ECONOMIZER WATER EXIT TEMPERATURE =      506.8 K
   WATER SPECIFIC HEAT =                     4.44 KJ/KG/K
   AVERAGE WATER VISCOSITY =             2.07E-04 PAS-SEC
   AVERAGE WATER THERMAL CONDUCTIVITY =  6.84E-04 KW/M/K
   WATER PRANDTL NUMBER =                  1.3461
   WATER MASS FLOW RATE PER UNIT AREA =     0.646 KG/S/M**2
   WATER REYNOLDS NUMBER =                    9.6
   WATER STANTON NUMBER =                 0.08983
   WATER FRICTION FACTOR =                1.29042
   WATER HEAT TRANSFER COEFFICIENT =         0.26 KW/M**2/K
   OVERALL HEAT TRANSFER COEFFICIENT =       0.08 KW/M**2/K
   AIR HEAT TRANSPORT CAPACITY =             76.2 KW/K
   WATER HEAT TRANSPORT CAPACITY =           30.9 KW/K
   HEAT TRANSPORT CAPACITY RATIO =         0.4054
   ECONOMIZER EFFECTIVENESS =               0.950
   NUMBER OF REQ HEAT TRANSFER UNITS =      4.220
   ECONOMIZER WATER PRESSURE DROP =           0.2 PAS
   ECONOMIZER WATER PRESSURE RATIO =     0.999998
   ECONOMIZER AIR PRESSURE DROP =          1044.4 PAS
   ECONOMIZER AIR PRESSURE RATIO =       0.990000
   REQUIRED ECONOMIZER VOLUME =             2.902 M**3
   REQUIRED ECONOMIZER FRONTAL AREA =       6.257 M**2
   REQUIRED ECONOMIZER DEPTH =              0.464 M
   REQUIRED ECONOMIZER HEIGHT =             2.501 M
   REQUIRED ECONOMIZER WIDTH =              2.501 M


CONDENSER DESIGN
   MASS FLOW RATE OF CIRC WATER =          182.79 KG/SEC
   CIRC WATER INLET TEMPERATURE =           288.0 K
   AVERAGE CIRC WATER TEMPERATURE =         299.9 K
   CIRC WATER EXIT TEMPERATURE =            311.8 K
   CIRC WATER SPECIFIC HEAT =                4.18 KJ/KG/K
   CIRC WATER VISCOSITY =                1.05E-03 PAS-SEC
   CIRC WATER THERMAL CONDUCTIVITY =     6.09E-04 KW/M/K
   CIRC WATER PRANDTL NUMBER =             7.2159
   CIRC WATER SPECIFIC VOLUME =          0.001004 M**3/KG
   CIRC WATER MASS FLOW RATE/AREA =        21.132 KG/S/M**2
   CIRC WATER REYNOLDS NUMBER =              62.0
   CIRC WATER STANTON NUMBER =            0.01217
   CIRC WATER FRICTION FACTOR =           0.34670
   CIRC WATER HEAT TRANSFER COEF =           1.08 KW/M**2/K
   OVERALL HEAT TRANSFER COEFFICIENT =     453.00 KW/M**2/K
   MASS FLOW RATE OF CONDENSING WATER =      6.96 KG/SEC
   CONDENSING WATER INLET TEMPERATURE =     313.0 K
   CONDENSER EFFECTIVENESS =                0.950
   NUMBER OF REQ HEAT TRANSFER UNITS =      2.996
   CIRC WATER PRESSURE DROP =               101.3 PAS
   CIRC WATER PRESSURE RATIO =             0.9990
   CIRC WATER PUMPING POWER =               0.019 KW
   REQUIRED CONDENSER VOLUME =             5.0540 M**3
   REQUIRED CONDENSER FRONTAL AREA =       5.0262 M**2
   REQUIRED CONDENSER DEPTH =              1.0056 M
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   REQUIRED CONDENSER WIDTH =              2.2419 M
   REQUIRED CONDENSER HEIGHT =             2.2419 M


THE FINAL STATE VARIABLE SUMMARY

   STATE(0)(ATMOSPHERE)  PRESS=  0.1013 MPA TEMP=  288.0 K

 BRAYTON SYSTEM STATE POINTS

   STATE(1)(COMP EXIT)   PRESS=  1.2664 MPA TEMP=  629.4 K

   STATE(3)(HEATR(1) IN) PRESS=  1.2664 MPA TEMP=  629.4 K
   STATE(4)(TURBN(1) EX) PRESS=  1.2537 MPA TEMP=  933.0 K
   STATE(5)(TURBN(1) EX) PRESS=  0.6821 MPA TEMP=  810.0 K

   STATE(3)(HEATR(2) IN) PRESS=  0.6821 MPA TEMP=  810.0 K
   STATE(4)(TURBN(2) EX) PRESS=  0.6752 MPA TEMP=  933.0 K
   STATE(5)(TURBN(2) EX) PRESS=  0.3673 MPA TEMP=  810.0 K

   STATE(3)(HEATR(3) IN) PRESS=  0.3673 MPA TEMP=  810.0 K
   STATE(4)(TURBN(3) EX) PRESS=  0.3637 MPA TEMP=  933.0 K
   STATE(5)(TURBN(3) EX) PRESS=  0.1978 MPA TEMP=  810.0 K

   STATE(3)(HEATR(4) IN) PRESS=  0.1978 MPA TEMP=  810.0 K
   STATE(4)(TURBN(4) EX) PRESS=  0.1959 MPA TEMP=  933.0 K
   STATE(5)(TURBN(4) EX) PRESS=  0.1066 MPA TEMP=  810.0 K

   STATE(6)(RECUP EXIT)  PRESS=  0.1066 MPA TEMP=  810.0 K

   STATE(7)(SPRHT EXIT)  PRESS=  0.1055 MPA TEMP=  670.6 K
   STATE(8)(EVAP EXIT)   PRESS=  0.1044 MPA TEMP=  517.0 K
   STATE(9)(ECON EXIT)   PRESS=  0.1034 MPA TEMP=  438.7 K

 RANKINE SYSTEM STATE POINTS

   STATE(20)(PUMP ENTR)  PRESS=  0.0073 MPA TEMP=  313.0 K
   STATE(21)(PUMP EXIT)  PRESS=  3.0000 MPA TEMP=  313.7 K

   STATE(23)(ECON EXIT)  PRESS=  3.0000 MPA TEMP=  506.8 K
   STATE(24)(EVAP EXIT)  PRESS=  3.0000 MPA TEMP=  507.0 K

   STATE(25)(HP TURB IN) PRESS=  3.0000 MPA TEMP=  795.0 K
   STATE(26)(HP TURB EX) PRESS=  0.7500 MPA TEMP=  582.0 K

   STATE(27)(MP TURB IN) PRESS=  0.7500 MPA TEMP=  795.0 K
   STATE(28)(MP TURB EX) PRESS=  0.1875 MPA TEMP=  583.3 K

   STATE(29)(LP TURB IN) PRESS=  0.1875 MPA TEMP=  795.0 K

   STATE(30)(CONDNSR IN) PRESS=  0.0073 MPA TEMP=  375.5 K


COMPONENT SIZING SUMMARY

   COMPRESSOR     9.6519E-01 M**3    1.3000  1.2547E+00 M**3
   AIR TURBINE 1  9.0275E-03 M**3    1.1000  9.9302E-03 M**3
   AIR TURBINE 2  2.2839E-02 M**3    1.1000  2.5123E-02 M**3
   AIR TURBINE 3  5.7783E-02 M**3    1.1000  6.3561E-02 M**3
   AIR TURBINE 4  1.4619E-01 M**3    1.1000  1.6081E-01 M**3
   HEATER      1  1.9459E+00 M**3    1.1000  2.1405E+00 M**3
   HEATER      2  1.3453E+00 M**3    1.1000  1.4798E+00 M**3
```

```
HEATER      3  1.7948E+00 M**3    1.1000  1.9743E+00 M**3
HEATER      4  2.3254E+00 M**3    1.1000  2.5580E+00 M**3
SUPRHEATER  1  4.6114E+00 M**3    1.2000  5.5337E+00 M**3
SUPRHEATER  2  4.2379E+00 M**3    1.2000  5.0854E+00 M**3
SUPRHEATER  3  4.8051E+00 M**3    1.2000  5.7662E+00 M**3
HP STM TURBIN  1.8055E-03 M**3    1.5000  2.7082E-03 M**3
MP STM TURBIN  1.4362E-02 M**3    1.5000  2.1542E-02 M**3
LP STM TURBIN  2.0919E+00 M**3    1.5000  3.1378E+00 M**3
EVAPORATOR     3.5451E+00 M**3    1.3000  4.6087E+00 M**3
ECONOMIZER     2.9016E+00 M**3    1.3000  3.7721E+00 M**3
CONDENSER      5.0540E+00 M**3    1.2000  6.0648E+00 M**3
INTER COOLER   0.0000E+00 M**3    1.4000  0.0000E+00 M**3
RECUPERATOR    0.0000E+00 M**3    1.0250  0.0000E+00 M**3

CALCULATED BRAYTON VOLUME =   8.6125E+00 M**3
CALCULATED RANKINE VOLUME =   2.7263E+01 M**3
CALCULATED TOTAL VOLUME =     3.5876E+01 M**3
ESTIMATED REQUIRED VOLUME =   4.3660E+01 M**3
```

B-3 Input File and Output Results for 3 Turbines with Recuperated Brayton Cycle

INPUT FILE

```
Advanced System w/3 turbines
TRBIN
 4 3
 933.0 860.0 0.90
 933.0 860.0 0.90
 933.0 860.0 0.90
RCUPR
 0.95 0.99 1
STGEN
 0.99 0.99 0.99 0.999 10.0 15.0 0.0 0.0 0.0 313.0 0.90
START
```

OUTPUT FILE

```
 Advanced System w/3 turbines


  THE PRINT LENGTH PARAMETER IS  4

  THE NUMBER OF TURBINES =  3
     DATA FOR TURBINE NUMBER  1
     TURBINE INLET TEMPERATURE     =         933.0 K          1679.4 R
     TURBINE EXIT TEMPERATURE      =         860.0 K          1548.0 R
     TURBINE POLYTROPIC EFFICIENCY =         0.900
     DATA FOR TURBINE NUMBER  2
     TURBINE INLET TEMPERATURE     =         933.0 K          1679.4 R
     TURBINE EXIT TEMPERATURE      =         860.0 K          1548.0 R
     TURBINE POLYTROPIC EFFICIENCY =         0.900
     DATA FOR TURBINE NUMBER  3
     TURBINE INLET TEMPERATURE     =         933.0 K          1679.4 R
     TURBINE EXIT TEMPERATURE      =         860.0 K          1548.0 R
     TURBINE POLYTROPIC EFFICIENCY =         0.900

  THE NUMBER OF COMPRESSORS = 1
     FIRST COMPRESSOR PRESSURE RATIO  =      10.00
     SECOND COMPRESSOR PRESSURE RATIO =       1.00
     COMPRESSOR POLYTROPIC EFFICIENCY =      0.900

  RECUPERATOR DESIGN INFORMATION
```

```
   RECUPERATOR EFFECTIVENESS =             0.9500
   MAX RECUPERATOR PRESSURE RATIO =        0.9900
   RECUPERATOR CONFIGURATION =                  1


   THE PRESSURE RATIO FOR HEATER 1 =       0.990
   THE PRESSURE RATIO FOR HEATER 2 =       0.990
   THE PRESSURE RATIO FOR HEATER 3 =       0.990

THE OPERATING ATMOSPHERIC DATA ARE
   ATMOSPHERIC PRESSURE =                0.10132 MPA       14.70 PSI
   ATMOSPHERIC TEMPERATURE =               288.0 K         518.4 R
   INLET WATER TEMPERATURE =               288.0 K         518.4 R
   EXHAUST PRESSURE RATIO =               0.9800

   THE DESIGN POWER OUTPUT IS =            25.00 MW
   FLUID TEMPERATURE IN =                  973.0 K
   THE FLUID MASS FLOW RATE IS =           40.00 KG/SEC
   THE FLUID PRESSURE IS =              1000.000 KPA,


   DESIGN RADIUS RATIO =                   0.600
   TURBINE BLADE SOLIDITY =                0.050
   COMPRESSOR INLET MACH NO. =             0.300
   COMPRESSOR EXIT MACH NO. =              0.100
   TURBINE 1 EXIT MACH NO. =               0.100
   TURBINE 2 EXIT MACH NO. =               0.100
   TURBINE 3 EXIT MACH NO. =               0.100

MATERIAL IDENTIFIERS
   HEAT TRANSFER FLUID IDENTIFIER =            1
   AIR HEATER MATERIAL IDENTIFIER =           40
   INTERCOOLER MATERIAL IDENTIFIER =          20
   ECONOMIZER MATERIAL IDENTIFIER =           30
   EVAPORATOR MATERIAL IDENTIFIER =           30
   SUPERHEATER MATERIAL IDENTIFIER =          40
   CONDENSER MATERIAL IDENTIFIER =            60
   RECUPERATOR MATERIAL IDENTIFIER =          40

HEAT TRANSFER & FRICTION CORRELATIONS
   MAIN HEATER COLD CORRELATION =        27
   MAIN HEATER HOT CORRELATION =         27
   SUPERHEATER COLD CORRELATION =        27
   SUPERHEATER HOT CORRELATION =         27
   EVAPORATOR COLD CORRELATION =         27
   EVAPORATOR HOT CORRELATION =          27
   ECONOMIZER COLD CORRELATION =         27
   ECONOMIZER HOT CORRELATION =          27
   CONDENSER COLD CORRELATION =          27
   CONDENSER HOT CORRELATION =           27
   INTERCOOLER COLD CORRELATION =        27
   INTERCOOLER HOT CORRELATION =         27
   RECUPERATOR COLD CORRELATION =        25
   RECUPERATOR HOT CORRELATION =         25

THE NEW PRESSURE RATIO IS=     3.088


COMPRESSOR PERFORMANCE DATA
   OVERALL CPR =                            3.088
   COMPR EXIT PRESS=                       0.3129 MPA         45.40 PSI
   COMPR EXIT TEMP =                        411.3 K           740.4 R
   COMPRESSOR WORK =                        124.5 KJ/KG       53.5 BTU/LBM
```

COMPRESSOR ISENTROPIC EFFICIENCY= 0.8832

HEATER 1 AND TURBINE PERFORMANCE DATA

1ST HTR EX PRESS=	0.3067 MPA	44.50 PSI
HEATR CP SPEC HT=	1.115 KJ/KG/K	0.266 BTU/LBM/R
1ST HEATR INPUT =	106.43 KJ/KG	45.76 BTU/LBM
TURBN CP SPEC HT=	1.117 KJ/KG/K	0.267 BTU/LBM/R
RATIO OF SP HEAT=	1.346	
TURB EXIT PRESS =	0.2156 MPA	31.28 PSI
1ST TURBINE WORK=	81.6 KJ/KG	35.1 BTU/LBM
1ST TURBINE PRESSURE RATIO=	0.7030	
TURBINE ISENTROPIC EFFICIENCY=	0.9040	

HEATER 2 AND TURBINE PERFORMANCE DATA

HTR ENTER PRESS=	0.2135 MPA	30.97 PSI
HEATR CP SPEC HT=	1.117 KJ/KG/K	0.267 BTU/LBM/R
HEAT INPUT =	81.57 KJ/KG	35.07 BTU/LBM
TURBINE CP SP HT=	1.117 KJ/KG/K	0.267 BTU/LBM/R
RATIO OF SP HEAT=	1.346	
TURB EXIT PRESS =	0.1501 MPA	21.77 PSI
TURBINE WORK =	81.6 KJ/KG	35.1 BTU/LBM
TURBINE PRESSURE RATIO=	0.7030	
TURBINE ISENTROPIC EFFICIENCY=	0.9040	

HEATER 3 AND TURBINE PERFORMANCE DATA

HTR ENTER PRESS=	0.1486 MPA	21.55 PSI
HEATR CP SPEC HT=	1.117 KJ/KG/K	0.267 BTU/LBM/R
HEAT INPUT =	81.57 KJ/KG	35.07 BTU/LBM
TURBINE CP SP HT=	1.117 KJ/KG/K	0.267 BTU/LBM/R
RATIO OF SP HEAT=	1.346	
TURB EXIT PRESS =	0.1044 MPA	15.15 PSI
TURBINE WORK =	81.6 KJ/KG	35.1 BTU/LBM
TURBINE PRESSURE RATIO=	0.7030	
TURBINE ISENTROPIC EFFICIENCY=	0.9040	

RECUPERATOR PERFORMANCE

HEAT RECOVERED =	454.8 KJ/KG	195.5 BTU/LBM
HOT TEMP IN =	860.0 K	1548.0 R
HOT TEMP OUT =	433.7 K	780.7 R
COLD TEMP IN =	411.3 K	740.4 R
COLD TEMP OUT =	837.6 K	1507.6 R

BRAYTON SYSTEM PERFORMANCE

TOTAL HEAT INPUT=	269.6 KJ/KG	115.9 BTU/LBM
TOTAL TURB WORK =	244.7 KJ/KG	105.2 BTU/LBM
COMPRESSOR WORK =	124.5 KJ/KG	53.5 BTU/LBM
NET SYSTEM WORK =	120.3 KJ/KG	51.7 BTU/LBM
BRAYTON SYS EFF =	0.4461	

SIZING DATA

TOTAL SYSTEM POWER =	25.000 MW
BRAYTON SYSTEM POWER =	25.000 MW
RANKINE SYSTEM POWER =	0.000 MW
TOTAL HEAT INPUT =	56.038 MW
MASS FLOW RATE(AIR) =	207.870 KGM/SEC
MASS FLOW RATE(WATER) =	0.000 KGM/SEC
TPRIME(WILSON)=	3.240

```
HEATER SIZING DATA
   HEAT INPUT FOR HEATER NO. 1 =          22.125 MW
   HEAT INPUT FOR HEATER NO. 2 =          16.957 MW
   HEAT INPUT FOR HEATER NO. 3 =          16.957 MW

COMPRESSOR SIZING DATA
   MACH NUMBER INTO COMPRESSOR =          0.3000
   COMPRESSOR INLET AREA =                2.2866 M**2
   COMPRESSOR OUTER RADIUS =              1.0941 M
   MACH NUMBER AT COMPRESSOR EXIT =       0.1000
   COMPRESSOR EXIT AREA =                 3.6807 M**2
   NUMBER OF STAGES =                          3
   COMPRESSOR VOLUME =                     1.975 M**3
   ESTIMATED POLYTROPIC EFFICIENCY=       0.9361


GAS TURBINE SIZING DATA
   DATA FOR AIR TURBINE NUMBER        1
   AIR VELOCITY AT INLET =                 599.7 M/SEC
   AIR TURBINE INLET AREA =               0.6785 M**2
   AIR TURBINE OUTER RADIUS =              1.094 M
   EXIT MACH NUMBER =                      0.100
   AIR VELOCITY AT EXIT =                   57.7 M/SEC
   AIR TURBINE EXIT AREA =                 5.417 M**2
   AIR TURBINE EXIT RADIUS =               1.094 M
   AIR TURBINE NO OF STAGES =                  2
   AIR TURBINE VOLUME =                    0.237 M**3
   ESTIMATED POLYTROPIC EFFICIENCY=       0.9401

   DATA FOR AIR TURBINE NUMBER        2
   AIR VELOCITY AT INLET =                 599.7 M/SEC
   AIR TURBINE INLET AREA =               0.9749 M**2
   AIR TURBINE OUTER RADIUS =              1.311 M
   EXIT MACH NUMBER =                      0.100
   AIR VELOCITY AT EXIT =                   57.7 M/SEC
   AIR TURBINE EXIT AREA =                 7.783 M**2
   AIR TURBINE EXIT RADIUS =               1.311 M
   AIR TURBINE NO OF STAGES =                  2
   AIR TURBINE VOLUME =                    0.409 M**3
   ESTIMATED POLYTROPIC EFFICIENCY=       0.9456

   DATA FOR AIR TURBINE NUMBER        3
   AIR VELOCITY AT INLET =                 599.7 M/SEC
   AIR TURBINE INLET AREA =               1.4008 M**2
   AIR TURBINE OUTER RADIUS =              1.572 M
   EXIT MACH NUMBER =                      0.100
   AIR VELOCITY AT EXIT =                   57.7 M/SEC
   AIR TURBINE EXIT AREA =                11.183 M**2
   AIR TURBINE EXIT RADIUS =               1.572 M
   AIR TURBINE NO OF STAGES =                  2
   AIR TURBINE VOLUME =                    0.704 M**3
   ESTIMATED POLYTROPIC EFFICIENCY=       0.9510


RECUPERATOR DESIGN

   FLUID PROPERTIES
   MASS FLOW RATE OF HOT AIR =            207.87 KGA/SEC
   HOT AIR TEMP INTO RECUPERATOR =         860.0 K
   HOT AIR TEMP OUT OF RECUPERATOR =       433.7 K
   AVERAGE HOT AIR TEMPERATURE =           646.9 K
   HOT AIR SPECIFIC HEAT =                  1.07 KJ/KG/K
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HOT AIR VISCOSITY =                     3.17E-05 PAS-SEC
HOT AIR THERMAL CONDUCTIVITY =          4.82E-05 KW/M/K
HOT AIR PRANDTL NUMBER =                  0.7025
HOT AIR HEAT CAPACITY RATE =              221.77 KW/K
MASS FLOW RATE OF COLD AIR =            207.8700 KGW/SEC
COLD AIR TEMP INTO RECUPERATOR =           411.3 K
COLD AIR TEMP OUT OF RECUPERATOR=          837.6 K
AVERAGE COLD AIR TEMPERATURE =             624.4 K
COLD AIR SPECIFIC HEAT =                    1.06 KJ/KG/K
COLD AIR VISCOSITY =                    3.10E-05 PAS-SEC
COLD AIR THERMAL CONDUCTIVITY =         4.68E-05 KW/M/K
COLD AIR PRANDTL NUMBER =                 0.7025
COLD AIR HEAT CAPACITY RATE =             220.87 KW/K
RECUPERATOR EFFECTIVENESS =               0.9500
HEAT CAPACITY RATE RATIO =                0.9960
HOT AIR MASS FLOW PER UNIT AREA =          1.224 KG/S/M**2
HOT AIR REYNOLDS NUMBER =                   31.1
HOT AIR FRICTION FACTOR =                0.52644
HOT AIR HEAT TRANSFER COEFFICIENT =       0.1443 KW/M**2/K
HOT AIR HYDRAULIC DIAMETER =              0.0008 M
RECUPERATOR EXIT HOT AIR TEMP =            433.7 K
COLD AIR MASS FLOW PER UNIT AREA =         1.224 KG/S/M**2
COLD AIR REYNOLDS NUMBER =                  31.8
COLD AIR FRICTION FACTOR =               0.51444
COLD AIR HEAT TRANSFER COEFFICIENT =      0.1409 KW/M**2/K
COLD AIR HYDRAULIC DIAMETER =             0.0008 M
COLD AIR EXIT TEMPERATURE =                837.6 K
REQUIRED HEAT TRANSFER UNITS =            18.307
COLD AIR PRESSURE DROP =                   165.8 PAS
COLD AIR PRESSURE RATIO =                 0.9995
HOT AIR PRESSURE DROP =                   1044.4 PAS
HOT AIR RECUPERATOR PRESSURE RATIO =      0.9900
RECUPERATOR VOLUME REQUIRED =             34.711 M**3
RECUPERATOR FRONTAL AREA =               115.504 M**2
RECUPERATOR DEPTH =                        0.301 M
RECUPERATOR HEIGHT =                      10.747 M
RECUPERATOR WIDTH =                       10.747 M
COLD CHANNEL HEAT XFER AREA/VOLUME =      4372.0 M**2/M**3
HOT CHANNEL HEAT XFER AREA/VOLUME =       4372.0 M**2/M***3


DESIGN PARAMETERS FOR HEAT EXCHANGER  1

AIR TEMPERATURE AT INLET =                 837.6 K
AIR PRESSURE AT INLET =                   0.3098 MPA
AIR TEMPERATURE AT EXIT =                  933.0 K
AIR MASS FLOW RATE =                      207.87 KG/SEC
AIR SIDE PRESSURE RATIO =                  0.990
TOTAL HEAT TRANSFER RATE =               22127.3 KW
FLUID TEMPERATURE AT INLET =               973.0 K
INPUT FLUID MASS FLOW RATE=                40.00 KG/SEC
FLUID MASS FLOW RATE =                    243.67 KG/SEC
FLUID TEMPERATURE AT EXIT =                927.9 K
LM AVERAGE AIR TEMPERATURE =               899.3 K
AIR PRANDTL NUMBER =                       0.701
AIR VISCOSITY =                         0.000039 PAS-SEC
AIR SPECIFIC HEAT =                        1.118 KJ/KG/K
AIR THERMAL CONDUCTIVITY =                  0.00 KW/M/K
AVERAGE FLUID TEMPERATURE =                950.4 K
FLUID THERMAL CONDUCTIVITY =              0.0009 KW/M/K
FLUID VISCOSITY =                       3.22E-03 PAS-SEC
FLUID SPECIFIC HEAT =                      1.988 KJ/KG/K
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   FLUID PRANDTL NUMBER =                   7.177
   FLUID SPECIFIC VOLUME =              4.91E-04 M**3/KG

FLUID HEAT TRANSFER PROPERTIES
   AIR MASS FLOW RATE/AREA =                14.16 KG/SEC/M**2
   AIR REYNOLDS NUMBER =                  1115.2
   FLUID MASS FLOW RATE/AREA =              16.60 KG/SEC/M**2
   FLUID REYNOLDS NUMBER =                  15.9
   AIR STANTON NUMBER =                  0.01456
   AIR SIDE FRICTION FACTOR =            0.04862
   FLUID STANTON NUMBER =                0.02320
   FLUID FRICTION FACTOR =               0.90436
   AIR FILM HEAT TRANSFER COEF  =           0.23 KW/M**2/K
   FLUID FILM HEAT TRANSFER COEF =          0.77 KW/M**2/K
   FIN EFFECTIVENESS =                    0.4735
   OVERALL SURFACE EFFECTIVENESS =        0.6019
   OVERALL HEAT TRANSFER COEF =            130.4 W/M**2/K
   AIR HEAT CAPACITY RATE =                232.4 KW/K
   FLUID HEAT CAPACITY RATE =              484.3 KW/K
   MINIMUM HEAT CAPACITY RATE =            232.4 KW/K
   MAXIMUM HEAT CAPACITY RATE =            484.3 KW/K
   NUMBER OF HEAT TRANSFER UNITS =         1.543
   HEAT CAPACITY RATE RATIO =              0.480
   HEAT EXCHANGER EFFICIENCY =             0.703
   HEATER EXIT TEMPERATURE =               932.8 K
   AIR INLET SPECIFIC VOLUME =          7.76E-01 M**3/KG
   AIR EXIT SPECIFIC VOLUME =           8.73E-01 M**3/KG
   AVERAGE AIR SPECIFIC VOLUME =        8.38E-01 M**3/KG
   FLUID PRESSURE DROP =                5.73E+00 PAS
   REQUIRED FLUID PUMPING POWER =       6.86E-04 KW
   HEATER AIR PRESSURE DROP =           3.10E+03 PAS
   HEATER AIR PRESSURE RATIO =            0.9900
   HEATER VOLUME =                         4.927 M**3
   HEATER FRONTAL AREA=                    8.529 M**2
   HEATER DEPTH =                          0.578 M
   HEATER WIDTH =                          2.920 M
   HEATER HEIGHT =                         2.920 M



DESIGN PARAMETERS FOR HEAT EXCHANGER  2

   AIR TEMPERATURE AT INLET =              860.0 K
   AIR PRESSURE AT INLET =                0.2156 MPA
   AIR TEMPERATURE AT EXIT =               933.0 K
   AIR MASS FLOW RATE =                   207.87 KG/SEC
   AIR SIDE PRESSURE RATIO =               0.990
   TOTAL HEAT TRANSFER RATE =            16957.8 KW
   FLUID TEMPERATURE AT INLET =            973.0 K
   INPUT FLUID MASS FLOW RATE=            243.67 KG/SEC
   FLUID MASS FLOW RATE =                 243.67 KG/SEC
   FLUID TEMPERATURE AT EXIT =             938.4 K
   LM AVERAGE AIR TEMPERATURE =            917.1 K
   AIR PRANDTL NUMBER =                    0.701
   AIR VISCOSITY =                      0.000040 PAS-SEC
   AIR SPECIFIC HEAT =                     1.121 KJ/KG/K
   AIR THERMAL CONDUCTIVITY =               0.00 KW/M/K
   AVERAGE FLUID TEMPERATURE =             955.7 K
   FLUID THERMAL CONDUCTIVITY =           0.0009 KW/M/K
   FLUID VISCOSITY =                    3.14E-03 PAS-SEC
   FLUID SPECIFIC HEAT =                   1.993 KJ/KG/K
   FLUID PRANDTL NUMBER =                  7.009
   FLUID SPECIFIC VOLUME =              4.92E-04 M**3/KG
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```
FLUID HEAT TRANSFER PROPERTIES
   AIR MASS FLOW RATE/AREA =                10.80 KG/SEC/M**2
   AIR REYNOLDS NUMBER =                    840.8
   FLUID MASS FLOW RATE/AREA =              12.66 KG/SEC/M**2
   FLUID REYNOLDS NUMBER =                   12.4
   AIR STANTON NUMBER =                   0.01670
   AIR SIDE FRICTION FACTOR =             0.05659
   FLUID STANTON NUMBER =                 0.02648
   FLUID FRICTION FACTOR =                1.07597
   AIR FILM HEAT TRANSFER COEF  =            0.20 KW/M**2/K
   FLUID FILM HEAT TRANSFER COEF =           0.67 KW/M**2/K
   FIN EFFECTIVENESS =                     0.5029
   OVERALL SURFACE EFFECTIVENESS =         0.6242
   OVERALL HEAT TRANSFER COEF =             117.4 W/M**2/K
   AIR HEAT CAPACITY RATE =                 233.1 KW/K
   FLUID HEAT CAPACITY RATE =               485.7 KW/K
   MINIMUM HEAT CAPACITY RATE =             233.1 KW/K
   MAXIMUM HEAT CAPACITY RATE =             485.7 KW/K
   NUMBER OF HEAT TRANSFER UNITS =          1.274
   HEAT CAPACITY RATE RATIO =               0.480
   HEAT EXCHANGER EFFICIENCY =              0.644
   HEATER EXIT TEMPERATURE =                932.8 K
   AIR INLET SPECIFIC VOLUME =           1.15E+00 M**3/KG
   AIR EXIT SPECIFIC VOLUME =            1.25E+00 M**3/KG
   AVERAGE AIR SPECIFIC VOLUME =         1.23E+00 M**3/KG
   FLUID PRESSURE DROP =                 2.79E+00 PAS
   REQUIRED FLUID PUMPING POWER =        3.35E-04 KW
   HEATER AIR PRESSURE DROP =            2.16E+03 PAS
   HEATER AIR PRESSURE RATIO =             0.9900
   HEATER VOLUME =                          4.533 M**3
   HEATER FRONTAL AREA=                    11.180 M**2
   HEATER DEPTH =                           0.405 M
   HEATER WIDTH =                           3.344 M
   HEATER HEIGHT =                          3.344 M


DESIGN PARAMETERS FOR HEAT EXCHANGER  3

   AIR TEMPERATURE AT INLET =               860.0 K
   AIR PRESSURE AT INLET =                 0.1501 MPA
   AIR TEMPERATURE AT EXIT =                933.0 K
   AIR MASS FLOW RATE =                    207.87 KG/SEC
   AIR SIDE PRESSURE RATIO =                0.990
   TOTAL HEAT TRANSFER RATE =             16957.8 KW
   FLUID TEMPERATURE AT INLET =             973.0 K
   INPUT FLUID MASS FLOW RATE=             243.67 KG/SEC
   FLUID MASS FLOW RATE =                  243.67 KG/SEC
   FLUID TEMPERATURE AT EXIT =              938.4 K
   LM AVERAGE AIR TEMPERATURE =             917.1 K
   AIR PRANDTL NUMBER =                     0.701
   AIR VISCOSITY =                       0.000040 PAS-SEC
   AIR SPECIFIC HEAT =                      1.121 KJ/KG/K
   AIR THERMAL CONDUCTIVITY =                0.00 KW/M/K
   AVERAGE FLUID TEMPERATURE =              955.7 K
   FLUID THERMAL CONDUCTIVITY =            0.0009 KW/M/K
   FLUID VISCOSITY =                     3.14E-03 PAS-SEC
   FLUID SPECIFIC HEAT =                    1.993 KJ/KG/K
   FLUID PRANDTL NUMBER =                   7.009
   FLUID SPECIFIC VOLUME =               4.92E-04 M**3/KG

FLUID HEAT TRANSFER PROPERTIES
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   AIR MASS FLOW RATE/AREA =                  7.45 KG/SEC/M**2
   AIR REYNOLDS NUMBER =                    579.5
   FLUID MASS FLOW RATE/AREA =                8.73 KG/SEC/M**2
   FLUID REYNOLDS NUMBER =                    8.6
   AIR STANTON NUMBER =                   0.02010
   AIR SIDE FRICTION FACTOR =             0.07174
   FLUID STANTON NUMBER =                 0.03156
   FLUID FRICTION FACTOR =                1.39880
   AIR FILM HEAT TRANSFER COEF  =             0.17 KW/M**2/K
   FLUID FILM HEAT TRANSFER COEF =            0.55 KW/M**2/K
   FIN EFFECTIVENESS =                     0.5444
   OVERALL SURFACE EFFECTIVENESS =         0.6556
   OVERALL HEAT TRANSFER COEF =             100.6 W/M**2/K
   AIR HEAT CAPACITY RATE =                 233.1 KW/K
   FLUID HEAT CAPACITY RATE =               485.7 KW/K
   MINIMUM HEAT CAPACITY RATE =             233.1 KW/K
   MAXIMUM HEAT CAPACITY RATE =             485.7 KW/K
   NUMBER OF HEAT TRANSFER UNITS =          1.274
   HEAT CAPACITY RATE RATIO =               0.480
   HEAT EXCHANGER EFFICIENCY =              0.644
   HEATER EXIT TEMPERATURE =                932.8 K
   AIR INLET SPECIFIC VOLUME =           1.65E+00 M**3/KG
   AIR EXIT SPECIFIC VOLUME =            1.80E+00 M**3/KG
   AVERAGE AIR SPECIFIC VOLUME =         1.76E+00 M**3/KG
   FLUID PRESSURE DROP =                 1.39E+00 PAS
   REQUIRED FLUID PUMPING POWER =        1.66E-04 KW
   HEATER AIR PRESSURE DROP =            1.50E+03 PAS
   HEATER AIR PRESSURE RATIO =             0.9900
   HEATER VOLUME =                          5.291 M**3
   HEATER FRONTAL AREA=                    16.222 M**2
   HEATER DEPTH =                           0.326 M
   HEATER WIDTH =                           4.028 M
   HEATER HEIGHT =                          4.028 M


THE FINAL STATE VARIABLE SUMMARY

   STATE(0)(ATMOSPHERE)  PRESS=  0.1013 MPA TEMP=  288.0 K

 BRAYTON SYSTEM STATE POINTS

   STATE(1)(COMP EXIT)   PRESS=  0.3129 MPA TEMP=  411.3 K

   STATE(3)(HEATR(1) IN) PRESS=  0.3098 MPA TEMP=  837.6 K
   STATE(4)(TURBN(1) EX) PRESS=  0.3067 MPA TEMP=  933.0 K
   STATE(5)(TURBN(1) EX) PRESS=  0.2156 MPA TEMP=  860.0 K

   STATE(3)(HEATR(2) IN) PRESS=  0.2156 MPA TEMP=  860.0 K
   STATE(4)(TURBN(2) EX) PRESS=  0.2135 MPA TEMP=  933.0 K
   STATE(5)(TURBN(2) EX) PRESS=  0.1501 MPA TEMP=  860.0 K

   STATE(3)(HEATR(3) IN) PRESS=  0.1501 MPA TEMP=  860.0 K
   STATE(4)(TURBN(3) EX) PRESS=  0.1486 MPA TEMP=  933.0 K
   STATE(5)(TURBN(3) EX) PRESS=  0.1044 MPA TEMP=  860.0 K

   STATE(6)(RECUP EXIT)  PRESS=  0.1034 MPA TEMP=  433.7 K


COMPONENT SIZING SUMMARY

   COMPRESSOR     1.9751E+00 M**3    1.3000  2.5677E+00 M**3
   AIR TURBINE 1  2.3749E-01 M**3    1.1000  2.6124E-01 M**3
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AIR TURBINE 2  4.0904E-01 M**3    1.1000  4.4994E-01 M**3
AIR TURBINE 3  7.0450E-01 M**3    1.1000  7.7495E-01 M**3
HEATER      1  4.9266E+00 M**3    1.1000  5.4193E+00 M**3
HEATER      2  4.5327E+00 M**3    1.1000  4.9859E+00 M**3
HEATER      3  5.2911E+00 M**3    1.1000  5.8203E+00 M**3
SUPRHEATER  1  0.0000E+00 M**3    1.2000  0.0000E+00 M**3
SUPRHEATER  2  0.0000E+00 M**3    1.2000  0.0000E+00 M**3
SUPRHEATER  3  0.0000E+00 M**3    1.2000  0.0000E+00 M**3
HP STM TURBIN  0.0000E+00 M**3    1.5000  0.0000E+00 M**3
MP STM TURBIN  0.0000E+00 M**3    1.5000  0.0000E+00 M**3
LP STM TURBIN  0.0000E+00 M**3    1.5000  0.0000E+00 M**3
EVAPORATOR     0.0000E+00 M**3    1.3000  0.0000E+00 M**3
ECONOMIZER     0.0000E+00 M**3    1.3000  0.0000E+00 M**3
CONDENSER      0.0000E+00 M**3    1.2000  0.0000E+00 M**3
INTER COOLER   0.0000E+00 M**3    1.4000  0.0000E+00 M**3
RECUPERATOR    3.4711E+01 M**3    1.0250  3.5579E+01 M**3

CALCULATED BRAYTON VOLUME =   5.2788E+01 M**3
CALCULATED RANKINE VOLUME =   0.0000E+00 M**3
CALCULATED TOTAL VOLUME =     5.2788E+01 M**3
ESTIMATED REQUIRED VOLUME =   5.5858E+01 M**3
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