Viability of Stirling-based Combined Cycle Distributed Power Generation

A Thesis Presented to

The Faculty of the

Frits J. and Dolores H.Russ

College of Engineering and Technology

Ohio University

In Partial Fulfillment

Of the Requirements for the Degree

Master of Science

By

Hua Liang

November, 1998

Acknowledgements

The author is indebted to my advisor, Dr. David J. Bayless for his precious advice, continuous encouragement, and financial support.

The author wishes to thank Neill W. Lane, President of Sun Power Inc., who has provided invaluable help in designing the heat exchanger for the Stirling engine.

Thanks also go to Dr. M. Khairul Alam, Dr. Israel Urieli, and Dr. Herman W. Hill for their valuable comments and assistance.

The author is obliged to Tieyi Guo, my old classmate in China, for his very helpful suggestions and continuous inspiration throughout this project.

To all the professors who have taught me and all the students with whom I associated throughout my graduate study, I appreciate you for your support, help, and friendship.

Finally, this thesis could not see the light of the day without my parents' constant love and encouragement.

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Definitions of Symbols and Abbreviations

A	annualized capital cost of a device
A_1	area of the combustion gas side surface
$A_{attached}$	annualized capital cost of the attached devices
A_c	free flow area of the strip fins of the heat exchanger
A_{com}	annualized capital cost of the compressor
A_{cond}	surface area of the condenser
$A_{c,e}$	cross area of the evaporator
A_{exch}	annualized capital cost of the heat exchanger
A_{fan}	annualized capital cost of the fan of the Stirling engine combined cycle
A_{gast}	annualized capital cost of the gas turbine
A_G	annualized capital cost of the gas turbine combined cycle system
A_{steam}	annualized capital cost of the steam turbine
A_{Stir}	annualized capital cost of the Stirling engine
A_S	annualized capital cost of the Stirling engine combined cycle system
AF	air-to-fuel ratio
C_h	characteristic dimension of the liquid /vapor interface
C_{gas_sys}	total annual cost of a gas turbine combined cycle system

C_p constant pressure specific heat

C_{Stir_sys} total annualized cost of a Stirling engine combined cycle

d working days of a combined cycle system in a year

d_w wire diameter of the wick

D_h hydraulic diameter of the offset strip fins

err relative error

f cost of the fuel ($/Ton)

F present value of the future cost (x years later) for a m-years analysis

g acceleration due to gravity

h height of offset strip fins

h_a convection heat transfer coefficient for the combustion gas side design

h_c calculated heat transfer coefficient for the combustion gas side design

h_{cond} convection heat transfer coefficient of the condenser

h_1 condensed water enthalpy

h_2 preheated water enthalpy

h_3 main vapor enthalpy

HHV higher heating value of the coal

i interests rate of the money

j indices

k	specific heat ratio C_p/C_v
k_m	number of times the machine need to be replaced during m years
K_{eff}	effective thermal conductivity coefficient of the wick-liquid interface
K_l	liquid thermal conductivity of the heat pipe working fluid
K_w	thermal conductivity of the wick material
l	length of the fin
L_{eff}	effective vapor section length
m	number of years for an economic analysis
$\dot{m}$	mass flow rate of the combustion gas going into the heat exchanger
$\dot{M}_{air}$	mass flow rate of the air
$\dot{M}_{coal}$	mass flow rate of the coal
$\dot{M}_{extract}$	mass flow rate of the extracted main steam
$\dot{M}_{fuel}$	mass flow rate of the fuel
$\dot{M}_{\max\_flow}$	the maximum flow through the gas turbine or compressor
$\dot{M}_{steam}$	mass flow rate of the steam
n	lifetime of the machine
n_s	number of the slot of the evaporator
N	number of Stirling engine or heat exchanger
N_u	Nusselt number

N_{com}	stage of the compressor
N_x	number of fins in the x direction
N_y	number of fins in the y direction
N_z	number of fins in the z direction
N_w	mesh density of the wick
P_1	inlet pressure of the compressor
P_2	inlet pressure of the boiler for the gas turbine combined cycle
P_3	inlet pressure of the gas turbine
P_4	outlet pressure of the gas turbine
P_V	vapor pressure
PT	Total investment of the machine for a $m-$years analysis
PV	current capital cost of the machine
$PV_{attached}$	current capital cost of the attached devices
PV_{com}	current capital cost of the compressor
PV_{exch}	current capital cost of the heat exchanger
PV_{fan}	current capital cost of the fan of the Stirling engine combined cycle
PV_{fuel}	annual payment for fuels for a combined cycle system
PV_{gast}	current capital cost of the gas turbine
PV_{gas_sys}	current capital cost of the gas turbine combined cycle system

PV_{steam}	current capital cost of the steam turbine
PV_{Stir}	current capital cost of the Stirling engine
PV_{Stir_sys}	current capital cost of the Stirling engine combined cycle
ΔP	pressure drop of the heat exchanger
$\Delta P_{c,m}$	the maximum capillary pressure difference generated within capillary wicks of the heat pipe
ΔP_i	inertial pressure gradient
ΔP_l	pressure drop required to return the liquid from the condenser to the evaporator of the heat pipe
ΔP_g	pressure required lifting the liquid through the wick
ΔP_V	pressure drop necessary to cause the vapor to flow from the evaporator to condenser
$\dot{Q}_{air}$	heat flow rate of the air
$\dot{Q}_b$	boiling limit of the heat pipe
$\dot{Q}_c$	capillary limit of the heat pipe
$\dot{Q}_{coal}$	rate of heat flow produced by the coal
$\dot{Q}_{economizer}$	heat flow rate to the economizer
$\dot{Q}_{ent}$	entrainment limit of the heat pipe
$\dot{Q}_{extract}$	rate of heat flow to be extracted from the main steam

$\dot{Q}_{inh}$	rate of heat flow to the heat exchanger
$\dot{Q}_{ins}$	rate of heat flow to the Stirling engine
$\dot{Q}_{outh}$	rate of heat flow coming out from the heat exchanger
$\dot{Q}_{outs}$	rate of heat flow coming out from the Stirling engine
$\dot{Q}_{preheater}$	rate of heat flow to the preheater
Q_s	sonic limit of the heat pipe
$\dot{Q}_{steam}$	rate of heat flow to the steam
$r_{c,e}$	effective capillary radius
$r_{h,V}$	hydrolic radius of the evaporator
r_n	nucleation site radius
R	universal gas constant
R_1	thermal resistance of the evaporator
R_2	thermal resistance of the heat pipe
R_3	thermal resistance of the condenser
Re	Reynold number
s	transverse spacing of offset strip fins
t	thickness of the offset stripped fins
t_1'	inlet temperature of the hot fluid of the heat exchanger
t_2'	inlet temperature of the cold fluid of the heat exchanger

t_1''	outlet temperature of the hot fluid of the heat exchanger
t_2''	outlet temperature of the cold fluid of the heat exchanger
Δt	logarithms mean temperature difference of the heat exchanger
Δt_{cond}	temperature difference between the wall of the condenser and the working fluid of the Stirling engine
T_1	inlet temperature of the compressor
T_2	inlet temperature of the boiler
T_3	temperature of the hot gas, or inlet temperature of the gas turbine or heat exchanger of the Stirling engine
T_4	outlet temperature of the gas turbine
T_{4s}	ideal outlet temperature of the gas turbine
$T_{4'}$	outlet temperature of the heat exchanger of the Stirling engine
T_5	outlet temperature of the economizer
T_6	outlet temperature of the preheater
T_{cold}	temperature of the cold source of the Stirling engine
T_{hot}	temperature of the heater header of the Stirling engine in base load
V	velocity of the combustion gas
$\dot{W}_{com}$	rate of work done by the compressor
$\dot{W}_{design}$	design output of the entire combined cycle system
$\dot{W}_{fan}$	rate of work done by the fan of the Stirling engine combined cycle system

$\dot{W}_{gas\_turbine}$	rate of work done by the gas turbine
$\dot{W}_{steam}$	rate of work done by the steam turbine
$\dot{W}_{Stilring}$	rate of work done by the Stirling engine
$\dot{W}_{out}$	output of the whole system
w	percentage of the working flow rate to the maximum flow rate of the gas turbine or compressor
x	number of years after which a device has to be replaced
α	excess air coefficient
α_f	geometric parameter of the offset strip fin, $\alpha_f = \frac{s}{h}$
δ_f	geometric parameter of the offset strip fin, $\delta_f = \frac{t}{l}$
δ_w	thickness of the wick
ε	effectiveness of the heat exchanger
ε_w	wick porosity
γ	ratio of specific heats of the air
γ_f	geometric parameter of the offset strip fin, $\gamma_f = \frac{t}{s}$
γ_V	specific heats ratio of the heat pipe working fluid
γ_Q	heat ratio of air, which is equal to $\dot{Q}_{air}/\dot{Q}_{coal}$
$\eta_{assumme}$	initial efficiency of the whole combined cycle system

η_{com}	efficiency of the compressor
$\eta_{gas\_turbine}$	efficiency of the gas turbine
$\eta_{heat\_exchanger}$	efficiency of the heat exchanger of the preheater and the economizer
$\eta_{steam\_turbine}$	efficiency of the steam turbine
$\eta_{Stirling}$	thermal efficiency of the Stirling engine
η_{system}	efficiency of the whole combined cycle system
λ	conductivity of the air
λ_V	latent heat of vaporization
u_l	liquid viscosity
u_V	vapor viscosity
π	compressing ratio of each stage of the compressor
ρ_l	liquid density
ρ_V	vapor density
σ	surface tension

Chapter 1 Introduction

1.1 Future of Distributed Power Generation

Distributed power generation is defined as the integrated or stand-alone use of small modular power generating resources (ranging from a few kilowatts to 50MW) by utilities, utility customers, and third parties [EPRI, 1996]. Compared with conventional, centralized power generation, distributed power generation is built near the end user such as hospitals, shopping centers, or residential areas. As a result, it can defer significant transmission and distribution (T&D) investments and improve power quality and reliability. As electric utilities in the United States become more deregulated, distributed power generation will become more significant and competitive with centralized power generation. Presently, more than 30 percent of the power plants currently under construction are distributed units being built by independent power producers [Mukherjee, 1997]. Recent research by Pfeifenberger [1997] shows that distributed power generation will be a significant sector of power generation in the future.

One factor contributing to the competitiveness of distributed power generation is the introduction of new technologies that have significantly improved the operating characteristics of distributed resources. In recent years, new technologies, such as combined cycle technology, fuel cells, solar panels, and new storage facilities have been implemented. Such innovations have increased the thermal efficiency of small power plants to approximately 50-60%, thus greatly decreasing the cost of distributed generation

[Pfeifenberger, 1997]. Innovative development and creative thinking will continue to contribute to distributed electric generation systems.

1.2 Principle of Combined Cycles

Combined cycle technology presently is a very competitive technologies for distributed power generation. By combining a steam turbine with a gas turbine or other energy conversion devices, combined cycle technology can produce thermal efficiencies well in excess of 50%. Gas turbines typically operate at relatively high temperatures 1100-1650°C, with outlet temperatures of 400-650°C, whereas steam turbines are relatively low temperature machines with inlet temperature of 540-650°C and outlet temperature of around 30°C. Cycles with only one type of turbine can not reach a high thermal efficiency (>40%) when running alone, because the average temperature difference of the heat source and the cold sink is not very large. However, when the two turbines are used in combination, a significant amount of the waste heat in the gas turbine exhaust gases can be used to heat the water in the steam cycle. The average temperature difference between the heat source and cold sink of the whole system is much larger than the single cycle; thus, greatly improving the overall thermal efficiency [Shen, 1990].

1.3 Gas Turbine Combined Cycle

The gas turbine combined cycle is a very common and successful combined cycle. It couples the gas turbine single cycle and the steam turbine single cycle in such a way that most thermal "losses" from the gas turbine cycles are used by the steam turbine cycle. Gas turbine combined cycle offers not only the advantage of high thermal efficiency, but also the advantages of rapid start and flexible operation over a wide range of load. In addition, gas turbine combined cycle systems require low investment (capital) and operating costs, shorter plant construction times, offer high availability, and create low emissions [Mukherjee, 1997].

The main drawback of the gas turbine combined cycle is the decrease in reliability when using low quality fuels. This is because low quality fuels will generate a higher density of impurities such as sulfur, vanadium, sodium, and ash in the high temperature combustion gas, causing high temperature corrosion of the metallic structure components such as turbine blades. To lower the density of impurities, very expensive cleanup devices are required, often reducing the thermal efficiency. Furthermore, to achieve higher thermal efficiency with the gas turbine, expensive compressors, cooling system design, materials, and thermal barrier coatings are required, all raising the overall unit cost.

1.4 Stirling Engine Combined Cycle

The Stirling engine is an externally heated engine, which converts heat to work or vice versa with a very high thermal efficiency. The temperature of exhausted combustion gas from the Stirling engine is 400-800°C or even higher, which can provide heat to the steam turbine cycle. As the Stirling engine is an externally heated machine, the fuel source does not affect its working performance. For example, because moving parts of the Stirling engine is not exposed to the exhausted gas, particulate removal of the ash before the engine is not necessary. In addition, there is no need to compress the combustion gas to a high pressure as with the gas turbine. These characteristics make the Stirling engine a good candidate for combined cycle applications. It may be possible that the Stirling engine combined cycle, which uses the Stirling engine single cycle and the stream turbine single cycle in a combination, could compete with gas turbine combined cycle for distributed power generation, especially when using low quality fuels.

However, there are potential challenges in constructing Stirling engine combined cycles. First, a heat exchanger would be required to efficiently transport significant amounts of heat from the heat source to the Stirling engine heater head. Furthermore, the heater head of the Stirling engine would have to be able to endure very high temperatures. Finally, the low output of the Stirling (no more than 100kW) means the scale of the Stirling engine combined cycle is limited [Lane, 1997].

1.5 Scope of Study

The plausibility of using large-scale Stirling engines in combination with steam turbines to generate electricity for distributed power generation raises numerous questions. For example, what thermal efficiency can be obtained by the Stirling engine combined cycle and at what cost? Is the Stirling engine combined cycle competitive with the gas turbine combined cycle? Can the Stirling engine combined cycle technology compete with technologies employed for distributed power generation? The objective of this study is to explore the viability of the Stirling engine combined cycle system from both technical and economic aspects. The Stirling engine combined cycle is analyzed by designing a heat exchanger, building a combined cycle system model, and developing the system computational program to simulate it. Finally, a methodology of cost estimation for both the Stirling engine combined cycle system and the gas turbine combined cycle system is presented to address the economic viability of the Stirling engine combined cycle.

1.6 Thesis Organization

The remainder of this thesis is organized into five chapters. Chapter 2 reviews the literature of the Stirling engine combined cycle. Chapter 3 designs the heat exchanger for a large scale Stirling engine whose output is 100kW. Chapter 4 builds models for Stirling and the gas turbine combined cycles and analyzes the two cycles by a computer program. Chapter 5 discusses the methodology for the economic analysis. Finally, Chapter 6 presents conclusions and recommendations for future research.

Chapter 2 Stirling Engine Combined Cycle Analysis

2.1 Stirling Engine Background

First invented in 1816 by Robert Stirling, Stirling engines were widely used throughout the 19^{th} century. The invention of the internal combustion engine in middle of 19^{th} century, such as gasoline and fuel-oil engines, caused the use of Stirling engines to largely diminish until, by 1914, they were no longer available commercially in any quantity [Walker, 1980]. In recent years, Stirling engines have aroused much interest because of their many favorable characteristics including [Berchowitz, 1986]:

i) High thermal efficiency. Ideally speaking, the Stilring engine can achieve the maximum thermal efficiency of a thermal-energy conversion device.

ii) Minimal pollution. Because Stirling engines are externally heated, high efficiency burners, such as fluidized beds, can be used.

iii) Multi-fuel capacity. Stirling engines can use almost any form of fuel, as long as it can provide a sufficiently high temperature.

iv) Silent and practically vibrationless operation in certain configurations, such as Rhombic engine.

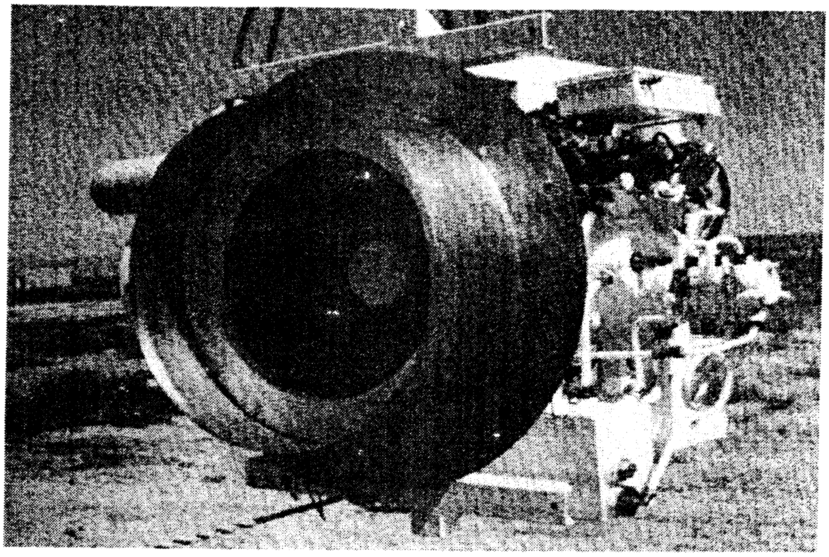
v) Ease of maintenance. The combustion products do not contact the moving parts, consequently, minimizing wear, increasing life and lowering maintenance.

Currently, Stirling engines of different types are widely used in many applications, such as small power generation, heat pump drivers, automobile engines, solar thermal conversion, and so on. [West, 1986]. Figure 2-1 and Figure 2-2 shows different Stirling engines used in different applications.

2.2 Stirling Engine Combined Cycle: Advantages and Disadvantages

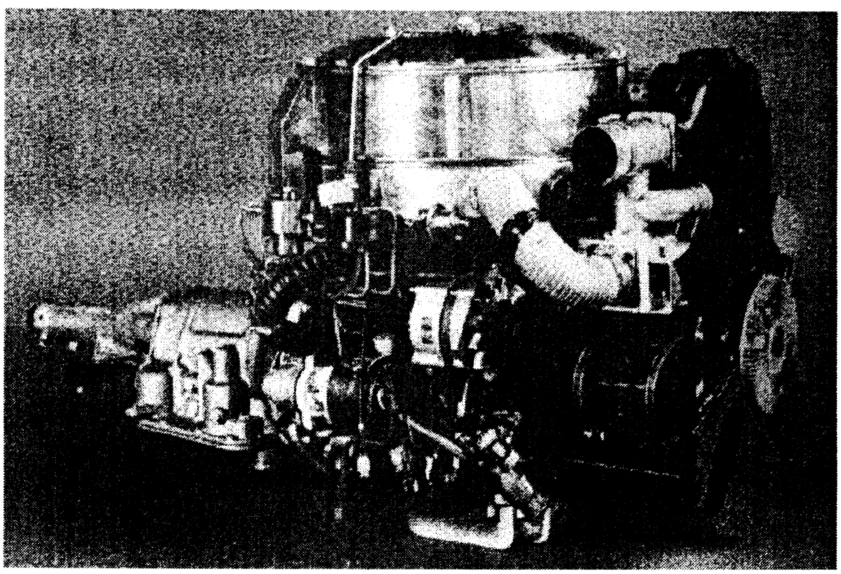
The idea of using the Stirling engines combined with other energy conversion devices to generate electricity is not new. Some authors and Stirling engine experts have discussed its advantages and disadvantages since the 1980s. In theory, the Stirling engine has a higher thermal efficiency than almost any other energy conversion device; thus its use in combined cycles may improve the overall efficiency compared with other configurations [Benvenuto, 1989]. In addition, Stirling engines have excellent low-load performance and respond quickly to sudden changes in load. They also have the potential to operate for very long periods with minimal maintenance and low lubricant oil consumption [Walker, 1980]. Finally, the fact that the Stirling engine can be fired by relatively less expensive and abundant energy sources, such as coal and biomass, offers a more important role for the Stirling engine in power generation. [Dunn, 1991].

Another important factor to consider regarding the Stirling engine combined cycle system is the decreasing cost of the Stirling engine. Current manufacturing cost of the Stirling engine is equal to that of a similar sized Diesel or gasoline engine [Stirling



[West, 1986]

Figure 2-1 Stirling solar engine ready for installation on the collector dish



[West, 1986]

Figure 2-2 A 4-95 engine fully equipped for automobile use

Thermal Motors Inc., 1997]. Therefore, the Stirling engine combined cycle system could prove to be competitive with other combined cycles in economics.

According to Neill W. Lane, President of Sun Power Inc. USA (1997), there are also disadvantages associated with the Stirling combined cycle system, including:

i) At the present time, the largest unit output of the Stirling engine is only 100 kW, while the gas turbine has much larger generation capacity.

ii) The technology of large-scale Stirling engines (>25kW) is less reliable.

iii) Large scale Stirling engines of high thermal efficiency and high specific output pose formidable problems in heat transfer methods, both in terms of material and design.

The major deterrent of an effective Stirling engine combined cycle is the fact that in order to achieve high efficiency, the heater head of the Stirling engine must operate continuously at a maximum cycle temperature, which can be 800°C or even higher. At the same time, the internal volume must be limited, which leads to a small area for heat transfer. Thus, a very high heat flux must be transferred. A successful design of a heat exchanger that can transport such a high heat flux in a limited volume is the most difficult challenge for the Stirling engine combined cycle.

2.3 Existing Investigation on Stirling Engine Combined Cycle

Since 1980, some Stirling engine combined cycle models have been proposed to investigate the possibility of Stirling engine combined cycle for power generation. S.G. Carqvist presented and discussed a Diesel/Stirling combined cycle in 1986 [Carqvist, 1986]. A number of problems posed by the Diesel/Stirling combined cycle were explained and treated. For example, how to create ideal conditions of temperature in the diesel exhaust gas that feeds the Stirling engine cycle; how to achieve energy balance in the turbo-charging system; and how to arrive at a desirable low temperature of the rest flue gases that leave the system. After that, the resulting thermodynamic engine system was presented. A simplified mathematical model was built and the performance of the cycle was calculated. The result shows that the overall thermal efficiency of the Diesel/Stirling combined cycle was near 60%.

In 1989, Giovanni Benvenuto proposed the Rankine/Stirling and Brayton/Stirling combined cycles for power plants [Benvenuto, 1989]. Simplified mathematical models were built to evaluate and compare the thermal efficiencies of the systems with and without Stirling engines. The thermodynamic advantages arising from the cycle combinations were analytically investigated as a function of the most relevant parameters such as temperature, pressure, and volumes. The results show that the Stirling engine combined cycle could lead to a substantial improvement of the overall performance.

However, the literature is not complete in the area of Stirling-based combined cycles. For example, the formidable problem of heat exchanger design has not been addressed in such analysis. Also, the performance of the combined cycle was only evaluated by mathematical derivations, thus lacking detailed and dynamic quantitative analysis. Furthermore, there was no comparison of the Stirling-based combined cycle to other, competitive cycles, such as gas turbine combined cycles. Finally, no economic analyses were attempted, leaving the question of viability unanswered. All these concerns lead to the development of this work.

Chapter 3 Heat Exchanger Design

3.1 Introduction

This chapter focuses on the design of the heat exchanger for a large scale Stirling engine. Before designing the heat exchanger, the type of Stirling engine to be used in the combined cycle system is selected. The heat flow rate that should be transported by the heat exchanger is calculated, and the description of the design problem is given. The design requirements and the design scheme are then discussed. The calculational method is also presented. Specific design calculations are attached in Appendix I.

3.2 Heat Exchanger Design Problem

3.2.1 Stirling Engine Selection

For the Stirling engine to be used in the combined cycle system for distributed generation, the unit output should larger than 25kW. A smaller output will require a larger number of Stirling engines to ensure the entire system output. In addition, the Stirling engine should have a good performance and be reliable. Finally, the cost of the Stirling engine must be as low as possible. According to those properties, the STM 4-120 Stirling engine with unit output of 100 kW was selected for this research.

The STM 4-120 engine with unit output of 100 kW is the largest Stirling engine in the world. This engine is the result of fifteen years of Stirling engine development at Stirling Thermal Motors, Inc. (STM) in Ann Arbor, Michican, U.S.A. Although it is not yet in commercial production, it has a good reputation for its design, performance, and competitive manufacturing cost [Lane, 1997].

Table 3-1 lists the parameters of the STM 4-120 Stirling engine [Hargreaves, 1991].

Table 3-1 Summary of Technique Features of STM 4-120 Stirling Engine

Type	STM 4-120
Number of cylinders	4
Combustion system	External
Fuel to shaft efficiency	36%
Thermal efficiency	42%
Working gas	Hydrogen
Mean cycle pressure	12 Mpa
Heater temperature	800°C
Header area	1.2 m^2
Cooling water temperature	45°C
Material for engine block	Iron base CRM-6D

3.2.2 Heat Flow Rate Calculation

The energy distribution and convertion between the Stirling engine and heat exchager can be illustrated in Figure 3-1. According to the output and the thermal efficiency of the STM-120 Stirling engine, the heat flow rate, which the heat exchanger must deliver, can be calculated as fol!owings:

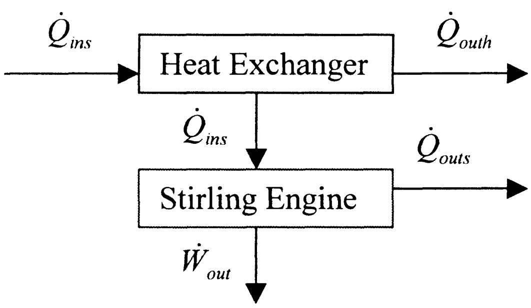


Figure 3-1 Heat and work flow chart

Output of the Stirling engine

$$\dot{W}_{out} = 100kW$$

Thermal Efficiency of the Stirling engine

$$\eta_{Stirling} = 42\ \%$$

Heat flow rate

$$\dot{Q}_{ins} = \frac{\dot{W}_{out}}{\eta_{Stirling}} = \frac{100}{0.42} = 238kW$$

Thus, the heat exchanger should be able to deliver a heat flow rate of at least 238 kW to the Stirling engine heater head. As the heat exchanger will not be 100% effective, additional heat should be added to the calculated heat flow. In this research, the design heat flow rate of the heat exchanger is 240 kW.

3.2.3 Design Problem Description

The design problem can be described as follows: The heat exchanger is designed for the STM-120 Stirling engine with output of 100 kW. It should be capable of transferring a heat flow rate of 240 kW. The hot working fluid is the combustion gas from the boiler, at a temperature of approximately 1100°C and at atmospheric pressure. The cold working fluid is the hydrogen inside the Stirling engine, whose temperature is 720°C. The convection heat transfer coefficient of the hydrogen inside the Stirling engine heater head is $5100 W/(m^2 \cdot K)$. The heater head of the Stirling engine must be maintained at 800°C in order to keep the Stirling engine to work in base load. (See Figure 3-2.)

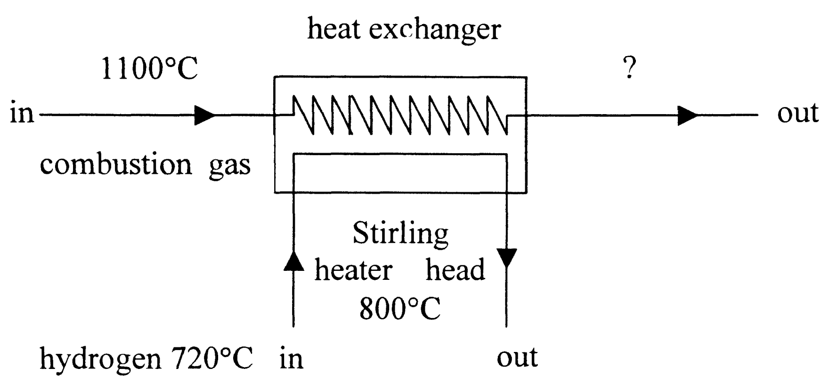


Figure 3-2 Heat exchanger design problem illustration chart

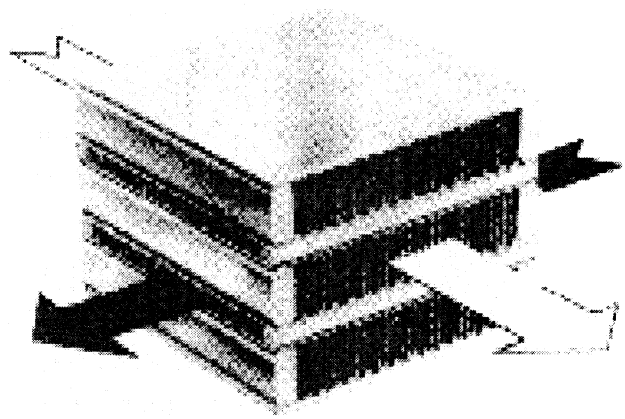
3.3 Heat Exchanger Design Schemes

The major difficulty with this design is that the temperature difference between the hot and cold working fluid is not very large (1100°C and 720°C respectively), while the heat exchanger must be able to transport a very high heat rate. That means a very high heat transfer coefficient must be attained. In this research, the following design schemes were used to achieve this requirement.

First, the problem of low temperature drop and high heat flow rate can be addressed by using the heat pipe, which can transport a very high heat rate through evaporation- condensation processes at solid surfaces, far higher than heat transfer between a gas and solid surface, even with forced convection. Second, in order to obtain a better heat transfer coefficient, a cross flow heat exchanger design, which is illustrated as Figure 3-3, was chosen. Finally, to achieve a large surface-area-to-volume ratio and a high heat transfer coefficient, the offset strip fins are used. Offset strip fins not only provide a high degree of compactness, but also help to obtain a substantial heat transfer enhancement [Shah, 1990]. Figure 3-4 shows the geometry of the offset strip fins.

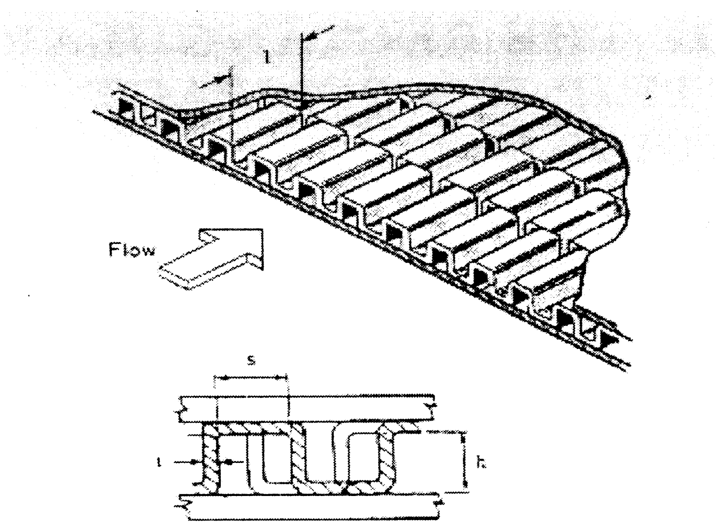
3.4 Design Procedures

There are three parts in the design of the heat pipe heat exchanger: the combustion gas side side surface design, the heat pipe design, and the condenser design. In this research, because the thermal conductance of heat pipe is high (several hundred times of that of the copper) and the convection inside the condenser has a very high heat transfer coefficient (over 5000 $W/m^2 \cdot K$ for the hydrogen under high pressure [Lane, 1997]), the heat resistance of these two parts can be ignored. The main heat transfer resistance of the heat exchanger lies in the combustion gas side of the evaporator. Therefore, the main concern is how to arrange the combustion gas side surface so that a higher comprehensive heat transfer coefficient can be achieved.



[Saunders, 1988]

Figure 3-3 Cross Flow



[Shah, 1990]

Figure 3-4 Offset strip fins

3.4.1 Combustion Gas Side Surface Design

The heat of the combustion gas is transferred to the heat pipe through the combustion gas surface. To design the combustion gas side surface, the outlet temperature of the combustion gas should be first determined by using the definition of the effectiveness of the heat exchanger, which can be expressed as

$$\varepsilon = \frac{t_1' - t_1''}{t_1' - t_2'} \qquad (3\text{-}1a)$$

where t_1' and t_1'' are the inlet and outlet temperature of the combustion gas, respectively, t_2' is the outlet the temperature of the working fluid of the heat pipe, which is taken as the 800°C (the same temperature as the heat head). The outlet temperature of the hot fluid is

$$t_1'' = t_1' - \varepsilon(t_1' - t_2') \qquad (3\text{-}1b)$$

In this study, the effectiveness of the heat exchanger is chosen as 96.67%. This is because a higher effectiveness will require more surface area, while a lower effectiveness can not take advantage of the temperature gap of the cold and hot fluids. Therefore, the outlet temperature is

$$t_2'' = 1100 - 0.967 \times (1100 - 800) = 810\,°\mathrm{C}$$

After the outlet temperature of the combustion gas is determined, the calculation of the combustion gas side design can be done using the following steps:

1. Calculate the mass flow of the combustion gas $\dot{m}$.
2. Calculate the logarithm mean temperature difference Δt.
3. Assume a convection coefficient h_a; calculate the surface area required A_1.

4. Choose the offset strip fin and arrange the surface.

5. Calculate the velocity V and Reynold number Re.

6. Calculate the Nusselt number Nu and friction factor f.

7. Calculate the convection heat transfer coefficient h_c and pressure drop ΔP.

8. Go back to 2, iterate 2—7 until the convection heat transfer coefficient satisfies the error requirement, namely, $\left|\frac{h_c - h_a}{h_a}\right| < err$ and the pressure drop is indurable large. In this study, the relative error err is 0.01.

This can be calculated as follows: (The method and formulas used to calculate the heat transfer of the offset strip fins are referred to in Shah, 1990).

1) Mass flow of the combustion gas $\dot{m}$

From the energy equation $\dot{Q} = \dot{m}C_p(t_1'' - t_1')$, therefore

$$\dot{m} = \frac{\dot{Q}}{C_p(t_1'' - t_1')} \tag{3-2}$$

2) Logarithm mean temperature difference Δt

The average temperature difference across the heat exchanger can be expressed as

$$\Delta t = \frac{(t_1' - t_2') - (t_1'' - t_2'')}{\ln\frac{t_1' - t_2'}{t_1'' - t_2''}} \tag{3-3}$$

3) Surface area required

The heat transfer equation can be expressed as

$$\dot{Q} = \frac{1}{R_1 + R_2 + R_3}\Delta t$$

where R_1, R_2, R_3 are the thermal resistances of the combustion gas side surface, the heat pipe and the condenser respectively. Since the thermal resistance of the heat pipe and the condenser can be ignored, the former equation can be simplified as

$$\dot{Q} = R_1 \Delta t$$

Since $R_1 = \dfrac{1}{h_a A_1}$,

$$A_1 = \frac{\dot{Q}}{h_a \Delta t} \tag{3-4}$$

4) Arrangement of the surface

Referring to Figure 3-4, the geometric parameters of the offset strip fin are

$$\alpha_f = \frac{s}{h}, \qquad \gamma_f = \frac{t}{s}, \qquad \delta_f = \frac{t}{l} \tag{3-5}$$

Let

N_x — number of fins in the x direction

N_y — number of fins in the y direction

N_z — number of the fins in the z direction

The surface area could be expressed as

$$A_1 = 2N_x N_y N_z l(s+h) \tag{3-6}$$

The free flow area is

$$A_c = N_x N_y (s+h) \tag{3-7}$$

The hydraulic diameter of the fin is expressed as

$$D_h = 4shl\left[2(sl + hl + th) + ts)\right] \tag{3-8}$$

5) Velocity V and Reynold number Re

$$V = \frac{\dot{m}}{\rho A_c} \tag{3-9}$$

$$\mathrm{Re} = \frac{VD_h}{\nu} \tag{3-10}$$

The reference Reynold number is

$$\mathrm{Re}' = 257(\frac{l}{s})^{1.23}(\frac{t}{s})^{0.58} D_h \left[t + 1.328(\frac{\mathrm{Re}}{lD_h})^{-0.5} \right]^{-1} \tag{3-11}$$

If $\mathrm{Re} > \mathrm{Re}' + 1000$, the flow in the offset fin is turbulent [Shah, 1990]. As turbulent flow has much higher heat transfer coefficient than laminar flow, the offset strip fins were arranged in such a way that a turbulence can be obtained.

6) Nusselt number Nu and the friction factor f

For turbulent flow in the offset stripped fins, Nu and f could be calculated by the following formulas, respectively [Shah, 1990]:

$$Nu = B(\mathrm{Re})^{b1}(\alpha_f)^{b2}(\delta_f)^{b3}(\gamma_f)^{b4} \tag{3-12}$$

where, $B = 0.2162$, $b1 = +0.5937$, $b2 = -0.1037$, $b3 = +0.1955$, $b4 = -0.1733$

$$f = C(\mathrm{Re})^{c1}(\alpha_f)^{c2}(\delta_f)^{c3}(\gamma_f)^{c4} \tag{3-13}$$

where, $C = 1.8699$, $a1 = -0.2993$, $a2 = -0.0936$, $a3 = +0.6820$, $a4 = -0.2423$

7) The convection heat transfer coefficient h_c and the pressure drop ΔP

Since $Nu = \dfrac{h_c D_h}{\lambda}$, therefore

$$h_c = \frac{Nu\lambda}{D_h} \tag{3-14}$$

The pressure drop can be calcuated as

$$\Delta P = f\left(\frac{1}{2}\rho V^2\right)\left(\frac{A_1}{A_c}\right) \tag{3-15}$$

3.4.2 Heat Pipe Design

The working fluid evaporates after absorbing a significant amount of heat on the evaporating surface of the evaporator and condenses on the surface of the condenser. The condensed liquid falls down to the evaporator by gravity and distributes all over the evaporating surface through the wick on the surface of the evaporator. The evaporation-condensation process repeats inside the heat pipe and the heat flux is transferred continuously from the combustion gas to the Stirling engine working fluid. The design of heat pipe is to select a suitable working fluid, determining the wick used in the evaporator and the geometry of the heat pipe, so that all the heat flow rate limits are larger than the design heat flow rate. In designing the heat pipe, the following procedures were followed:

1. Determine the geometry for the heat pipe heat exchanger.
2. Select the working fluid, wall material and wick for the heat pipe.
3. Calculate the capillary, sonic, entrainment, boiling, and viscous limits of the heat pipe. If one of those limits is less than the heat exchanger capacity, go back to step 2.

The following formulas [Peterson, 1994] were used to calculate the capillary, sonic, entrainment, boiling, and viscous limits:

1) The capillary limits. In order for the heat pipe to operate, the maximum capillary pumping head $\Delta P_{c,m}$, which is the maximum pressure head produced by the capillary in the evaporator, must be greater than the total pressure drop in the pipe. That is,

$$\Delta P_{c,m} \geq \Delta P_l + \Delta P_V + \Delta P_i + \Delta P_g \tag{3-16}$$

where, $\Delta P_{c,m}$ is the maximum capillary pumping head, given by Equation (3-17)

$$\Delta P_{c,m} = \frac{2\sigma}{r_{c,e}} \tag{3-17}$$

ΔP_V is the pressure drop necessary to cause the vapor to flow from the evaporator to the condenser. ΔP_i is the pressure drop necessary to overcome the inertial pressure gradient of the vapor flow. For laminar flow, ΔP_V and ΔP_i are given by Equation (3-18a) and Equation (3-19a), respectively.

$$\Delta P_V = \frac{16\mu_V L_{eff} \dot{Q}_c}{2r_{h,V} A_V \rho_V \lambda_V} \tag{3-18a}$$

$$\Delta P_i = 0 \tag{3-19a}$$

For turbulent flow, ΔP_V and ΔP_i are given by Equation (3-18b) and Equation (3-19b), respectively.

$$\Delta P_V = 0.038\left(\frac{d_V \dot{Q}_c}{A_V \mu_V \lambda_V}\right)^{\frac{3}{4}} \times \frac{2\mu_V L_{eff} \dot{Q}_c}{d_V{}^2 A_V \rho_V \lambda_V} \tag{3-18b}$$

$$\Delta P_i = \frac{1.22\dot{Q}_c^{\ 2}}{g\rho_V A_V^{\ 2}\lambda_V^{\ 2}} \tag{3-19b}$$

ΔP_g is the pressure requires to lift the liquid to the highest place of the evaporator, given by Equation (3-19)

$$\Delta P_g = \rho_l gh \tag{3-20}$$

ΔP_l is the pressure requires to drag the liquid from the condenser to the evaporator. For a gravity-assisted heat pipe, gravity drags the liquid from the condenser to the evaporator; no extra pressure head is required to return the liquid from the condenser to the evaporator. Therefore,

$$\Delta P_l = 0 \tag{3-21}$$

By substituting Equations (3-17), (3-18), (3-19), (3-20) and (3-21) into Equation (3-16), the capillary limit $\dot{Q}_c$ can be solved.

2) The sonic limit is the upper heat flow rate limited to the heat transport capability set by the high temperature choking at the evaporator exit, given by Equation (3-22)

$$\dot{Q}_s = \rho_V \lambda_V \sqrt{\frac{\gamma_V R T_V}{2(\gamma_V + 1)m_m}} A_V \tag{3-22}$$

3) The entrainment limit is the upper heat flow rate, which is set by the shear force on the liquid of the wick of the heat pipe, given by Equation (3-23)

$$\dot{Q}_{ent} = n_s \pi r_{h,V}^{\ 2} \lambda_V \sqrt{\frac{2\pi \rho_V \sigma_l \cos\theta}{C_n}} \tag{3-23}$$

4) The boiling limit is maximum radial heat flow rate at which burnout will not occur at the evaporator, given by Equation (3-24)

$$\dot{Q}_b = \frac{2\pi L_{eff} K_{eff} T_V}{\lambda_V \rho_V \delta_w}\left(\frac{2\sigma}{r_n} - \Delta P_{cm}\right) \tag{3-24}$$

5) The viscous limit is maximum heat flow rate occurring when the pressure is reduced to zero and viscous forces are dominant. It is determined by ΔP_V. If $\frac{\Delta P_V}{P_V} < 0.1$, the viscous limit is avoided.

Detailed results from this procedure are attached in Appendix I.

3.4.3 The Condenser Design

The vapor in the heat pipe condenses on the surface of the condenser, while a significant amount of the latent heat of the vapor is transferred to the Stirling engine working fluid. The design of the condenser is to determine the relatively small surface of the condenser and arrange it compactly in the Stirling engine heater head.

As the convection heat transfer coefficient of the hydrogen inside the Stirling engine is known ($5100 W/(m^2 \cdot K)$, the surface area required can be calculated easily from the following equation:

$$\dot{Q}_{cond} = h_{cond} A_{cond} \Delta t_{cond} \tag{3-25a}$$

where $\dot{Q}_{cond}$ is the heat flow rate should be transported by the condenser and is equal to the capacity of the heat exchanger, 240 kW .The temperature difference between the wall of the condenser and the working fluid of the Stirling engine is given by Δt_{cond}. In this research, as the wall temperature of the condenser will be 800°C, and the temperature of the working fluid is 720°C, therefore Δt_{cond} is equal to 80°C. Thus, the surface area is given by

$$A_{cond} = \frac{\dot{Q}_{cond}}{h_{cond}\Delta t_{cond}} \tag{3-25b}$$

The surface of the condenser should be arranged according to geometry of the Stirling engine and requirements the Stirling engine cycle.

3.5 Design Results

The design calculations (in Appendix I) show that a volume of $0.270m^3$ for the heat exchanger can be attained for the design requirements. Therefore, it is possible to design a heat exchanger that can transport 240 KW heat flow from the combustion gas to the Stirling engine; and it is possible to use the large-scale Stirling engine such as 100 KW STM-120 to generate electricity with the combustion gas temperature being 1100°C. The design picture of the heat exchanger is attached in Part III of Appendix I.

Chapter 4 Simulation of Stirling Engine and Gas Turbine Combined Cycles

4.1 Introduction

The first section of this chapter describes models for the gas turbine combined cycle and the Stirling engine combined cycle. The second section presents the calculational methods for the two systems, based on the models built in the first section. The computational program for the system calculations is also introduced. In the last section of this chapter, a simulation for a 20MW Stirling engine combined cycle is presented and the simulation results are analyzed.

4.2 Models of Gas Turbine vs. Stirling Engine Combined Cycles

4.2.1 Model of Gas Turbine Combined Cycle

Figure 4-1 shows the schematic flow diagram of the gas turbine combined cycle system. The gas turbine cycle, including an air compressor and a gas turbine, uses the exhausted gas from the gas turbine (4) for feed-water heating (steam cycle) at the economizer (4-5) and for cold air heating at the air preheater (5-6). The standard steam cycle consists of steam turbine, condenser, and pump. The HRB (Heat Recovery Boiler) consists of a boiler, an ash separator at the outlet of the boiler, which is used to separate the ash from the combustion gas before it reaches the gas turbine, an economizer, and an air preheater. Both the gas turbine and the steam turbine drive electric generators.

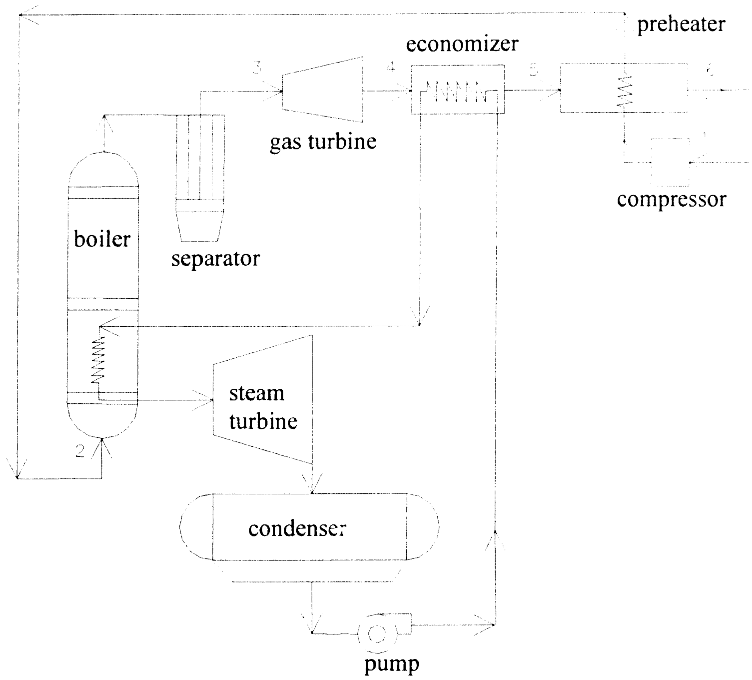


Figure 4-1 Gas turbine combined cycle system schematic

In addition to the main devices, there are some detached devices for the gas turbine cycle. They include the air cleanup device, which is used to lower the density of ashes in the combustion gas, and the blade-cooler, which can cool the blades of the gas turbine and protect the blades from high temperature. Both of those detached devices are expensive and will increase the overall cost of the gas turbine combined cycle system.

4.2.2 Model of Stirling Engine Combined Cycle

Figure 4-2 shows the schematic flow diagram of such a Stirling engine combined cycle system. From the diagram, we can see that the system is similar to that of the gas

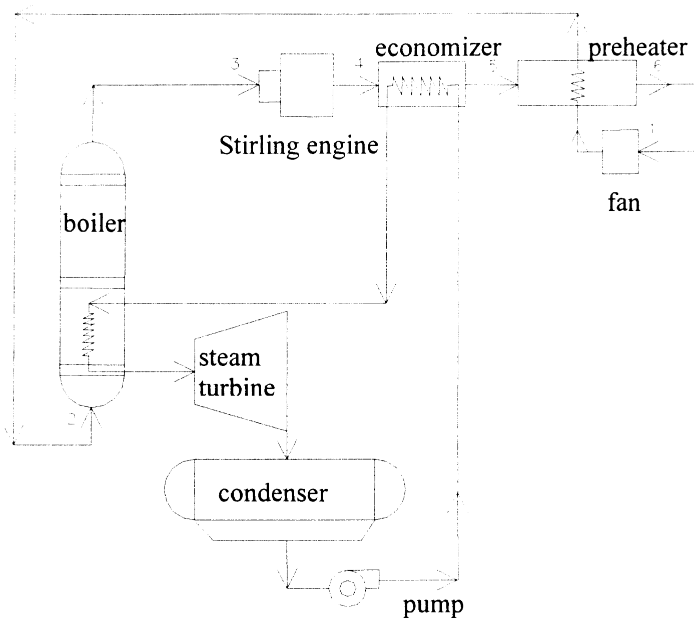


Figure 4-2 Stirling engine combined cycle system schematic

turbine combined cycle system except that the Stirling engine with a heat exchanger replaces the gas turbine. The other difference between the two models is that a fan replaces the compressor, because the Stirling engine does not need to compress the combustion gas. This difference is very important, because it not only helps to save the cost of the devices for the Stirling engine cycle (as a fan is much cheaper than a compressor with the same quantity of air output), but also helps to decrease a significant amount of work consumed by the devices (as the fan will consume much less work than the compressor).

4.3 Calculations for Stirling Engine vs. Gas Turbine Combined Cycles

4.3.1 Parameters for the Combined Cycles

To perform the calculation of a combined cycle system with the system output $\dot{W}_{out}$, the following list of parameters of the system should be known:

i. Properties of the air and combustion gas (C_p, k, R, T_1, T_2, P_2, P_3, P_4);

ii. Heat ratio of the combined cycle (γ_Q);

iii. Properties of the water and steam (h_1, h_2, h_3);

iv. Characteristicsof the compressor (N_{com}, π, η_{com}), gas turbine ($\eta_{gas\_turbine}$), and steam turbine ($\eta_{steam\_turbine}$), boiler (η_{boiler}) and Stirling engine ($\eta_{Stirling}$);

v. Properties of the fuel (composition of the fuel, HHV etc.);

Generally, those parameters are practically chosen based on the operational parameters of the existing power plants. For example, the inlet temperature of the compressor (T_1) can be chosen between 20-30°C; the inlet temperature of the boiler can be 230°C (T_2); the outlet temperature of the preheater (T_6) can be chosen as 140°C. Later simulation for a 20MW combined cycle will give a set of such parameters.

4.3.2 Calculation Formulas

Assuming the system output of a combined cycle is $\dot{W}_{out}$, and the system efficiency is η_{system}, the mass flow, heat flow, and work for the system can be determined based on the former known parameters. The calculational formulas are provided as follows: (The numerical labels for some parameters in the formulas are referred to in the system flow charts. See Figure 4-1 and Figure 4-2.)

1) Mass flow calculations ($\dot{M}_{coal}$, $\dot{M}_{air}$)

The heat flow needs to be produced by the coal is

$$\dot{Q}_{coal} = \frac{\dot{W}_{out}}{\eta_{system}} \tag{4-1}$$

Mass flow of the coal ($\dot{M}_{coal}$)

$$\dot{M}_{coal} = \frac{\dot{Q}_{coal}}{HHV} \tag{4-2}$$

Mass flow of the air ($\dot{M}_{air}$)

$$\dot{M}_{air} = AF * \alpha * \dot{M}_{coal} \tag{4-3}$$

where AF is the air to fuel ratio, α is the excess air coefficient.

2) Heat flow rate calculation ($\dot{Q}_{air}$, $\dot{Q}_{steam}$)

$$\dot{Q}_{air} = \gamma_Q * \dot{Q}_{coal} \tag{4-4a}$$

$$\dot{Q}_{steam} = (1-\gamma_Q) * \dot{Q}_{coal} \tag{4-4b}$$

where γ_Q is the heat ratio, which determines the percentages of the energy provided by the fuel to the gas turbine cycle or Stirling engine cycle.

3) Work done by the compressor ($\dot{W}_{com}$)

Assume the compressor works isothermally, which requires less comsumed work than other procedures, the work consumed by the compressor can be determined by [Shen, 1990]

$$\dot{W}_{com} = \frac{1}{\eta_{com}} N_{com} \dot{M}_{air} \frac{k}{k-1} R T_1 [\pi^{\frac{k-1}{k}} - 1] \tag{4-5}$$

where η_{com} is the working effiecincy of the compressor

N_{com} is the number of stages of the compressor

k is the specific heat ratio of the air

π is the compressing ratio of each stage of the compressor

4) Work done by the gas turbine ($\dot{W}_{gas\_turbine}$)

The work done by the gas turbine is

$$\dot{W}_{gas\_turbine} = \dot{M}_{air} \int_{T_4}^{T_3} C_p dT \tag{4-6a}$$

To simplify the calculation, C_p is taken as a constant throughout all the calculations. Thus, Equation (4-6a) is changed to

$$W_{gas\_turbine} = \dot{M}_{air} C_p (T_3 - T_4) \tag{4-6b}$$

To calculate $\dot{W}_{gas\_turbine}$, T_3 and T_4 (the inlet and outlet temperature of the gas turbine respectively) should be known. These two temperatures can be calculated as follows: First, the heat flow that the air absorbed from the boiler can be expressed as

$$\dot{Q}_{air} = \dot{M}_{air} C_p (T_3 - T_2) \tag{4-7}$$

As T_2 is practically chosen, T_3 can be expressed as

$$T_3 = \frac{\dot{Q}_{air}}{\dot{M}_{air} C_p} + T_2 \tag{4-8}$$

T_4 can be calculated by using the isotropic efficiency of the gas turbine which is expressed as [Shen, 1990]

$$\eta_{gas\_turbine} = \frac{T_3 - T_4}{T_3 - T_{4s}} \tag{4-9}$$

where T_{4s} represents the temperature at the condition of state 4 (in Figure 4-1) assuming it is an isentropic. T_{4s} can be calculated by using the specific entropy across the gas turbine, which is expressed as [Shen, 1990]

$$\Delta S_{3-4} = C_p \ln\frac{T_{4s}}{T_3} - R\ln\frac{P_4}{P_3} \tag{4-10}$$

As procedure 3-4 is an isentropic procedure, $\Delta S = 0$, therefore

$$T_{4s} = T_3 e^{(\frac{R}{C_p}\ln\frac{P_4}{P_3})} \tag{4-11}$$

From Equation (4-9) and Equation (4-11), the gas turbine outlet temperature T_4 can be attained.

$$T_4 = T_3 - \eta_{gas\_turbine} T_3[1 - e^{(\frac{R}{C_p}\ln\frac{P_4}{P_3})}] \tag{4-12}$$

Once T_3 and T_4 are known, the work done by the gas turbine can be determined by Equation (4-6).

6) Work done by the steam turbine ($\dot{W}_{steam}$)

The mass flow of the steam can be calculated as

$$\dot{M}_{steam} = \frac{\eta_{boiler}\dot{Q}_{steam}}{h_3 - h_2} \tag{4-13}$$

where h_2 is the enthalpy of the preheated water, h_3 is the the entralpy of the main vapor.

The heat flow required to heat the water in the economizer is

$$\dot{Q}_{economizer} = \dot{M}_{steam}(h_2 - h_1) \tag{4-14}$$

where h_1 is the entropy of the condensed water.

The heat flow required to preheat the cold air in the preheater is

$$\dot{Q}_{preheater} = \dot{M}_{air} C_p (T_2 - T_1) \tag{4-15}$$

The heat flow that needed to heat the water and the cold air is

$$\dot{Q}_{preheat} = \frac{(\dot{Q}_{economizer} + \dot{Q}_{preheater})}{\eta_{heat\_exchanger}} \tag{4-16}$$

The heat flow that can be provided by the exhausted combustion gas from the outlet of the gas turbine is determined by

$$\dot{Q}_{4\_6} = \dot{M}_{air} C_p (T_4 - T_6) \tag{4-17}$$

If $\dot{Q}_{4\_6} < \dot{Q}_{preheat}$, some main vapor should be extracted to make these two parts of heat balance. The extracted heat could be expressed as

$$\dot{Q}_{extract} = \frac{(\dot{Q}_{preheat} - \dot{Q}_{4\_6})}{\eta_{heat\_exchanger}} \tag{4-18}$$

The extracted mass flow of the vapor is

$$\dot{M}_{extract} = \frac{\dot{Q}_{extract}}{h_3} \tag{4-19}$$

The actual mass flow of the vapor to the steam turbine is

$$\dot{M}_{asteam} = \dot{M}_{steam} - \dot{M}_{extract} \tag{4-20}$$

The work done by the steam turbine is

$$\dot{W}_{steam} = \eta_{steam\_turbine} \dot{M}_{steam} (h_3 - h_1) \tag{4-21}$$

7) Work done by the Stirling engine ($\dot{W}_{Stirling}$)

$$\dot{W}_{Stirling} = \eta_{Stirling} \dot{M}_{air} C_p (T_3 - T_{4'}) \tag{4-22}$$

in which, $T_{4'}$ is the design outlet temperature the Stirling engine. In this research, $T_{4'}$ is 810°C.

8) Efficiency of the compressor, fan, gas turbine and Stirling engine

As the efficiencies of all these machines will change with the output, it is necessary to give out the efficiencies for these machines in different working conditions.

a) Efficiency of the compressor (η_{com})

The efficiency of the compressor is a function of the mass flow. Assuming the compressor will have maximum efficiency in the base load, the efficiency curve can be expressed as follows [AEP, 1988]:

$$\eta_{com} = 1.933428417 \times 10^{-4} w^3 - 4.18334900 \times 10^{-2} w^2 + 3.098533254 w + 5.879970 \tag{4-23}$$

b) Efficiency of the fan (η_{fan})

In order to simplify the calculation, the efficiency curve of the fan can be the same as that of the compressor.

c) Efficiency of the gas turbine ($\eta_{gas\_turbine}$)

The efficiency of the gas turbine is a function of mass flow of the working fluid. The relation between the efficiency of the gas turbine and the mass flow can be expressed as [Mattingly, 1996]

$$\eta_{gas\_turbine} = -8.4280303 \times 10^{-4} w^2 + 1.6940530 \times 10^{-1} w + 81.742500 \quad (4\text{-}24)$$

d) Efficiency of the Stirling engine ($\eta_{Stirling}$)

The efficiency of the Stirling engine is a function of temperature and the mass flow rate. But as temperature is the main factor that causes the change in the efficiency, the effect of the mass flow on the efficiency can be ignored. Therefore, the efficiency curve of the Stirling engine can be expressed as follows [Lane, 1997]:

$$\eta_{Stirling} = \eta_{\max\_Stir} \times \frac{1 - \dfrac{T_{cold}}{T}}{1 - \dfrac{T_{cold}}{T_{hot}}} \quad (4\text{-}25)$$

where $\eta_{\max\_Stir}$ is the maximum thermal efficiency of the Stilring engine. For the STM-120, it is equal to 42%.

4.3.3 Computational Algorithm

In order to simulate the gas turbine combined cycle and the Stirling engine combined cycle systems under different working conditions, a computational program to calculate the heat, mass, work, and efficiency for the systems was developed. The flowchart of the program is shown in Figure 4-3.

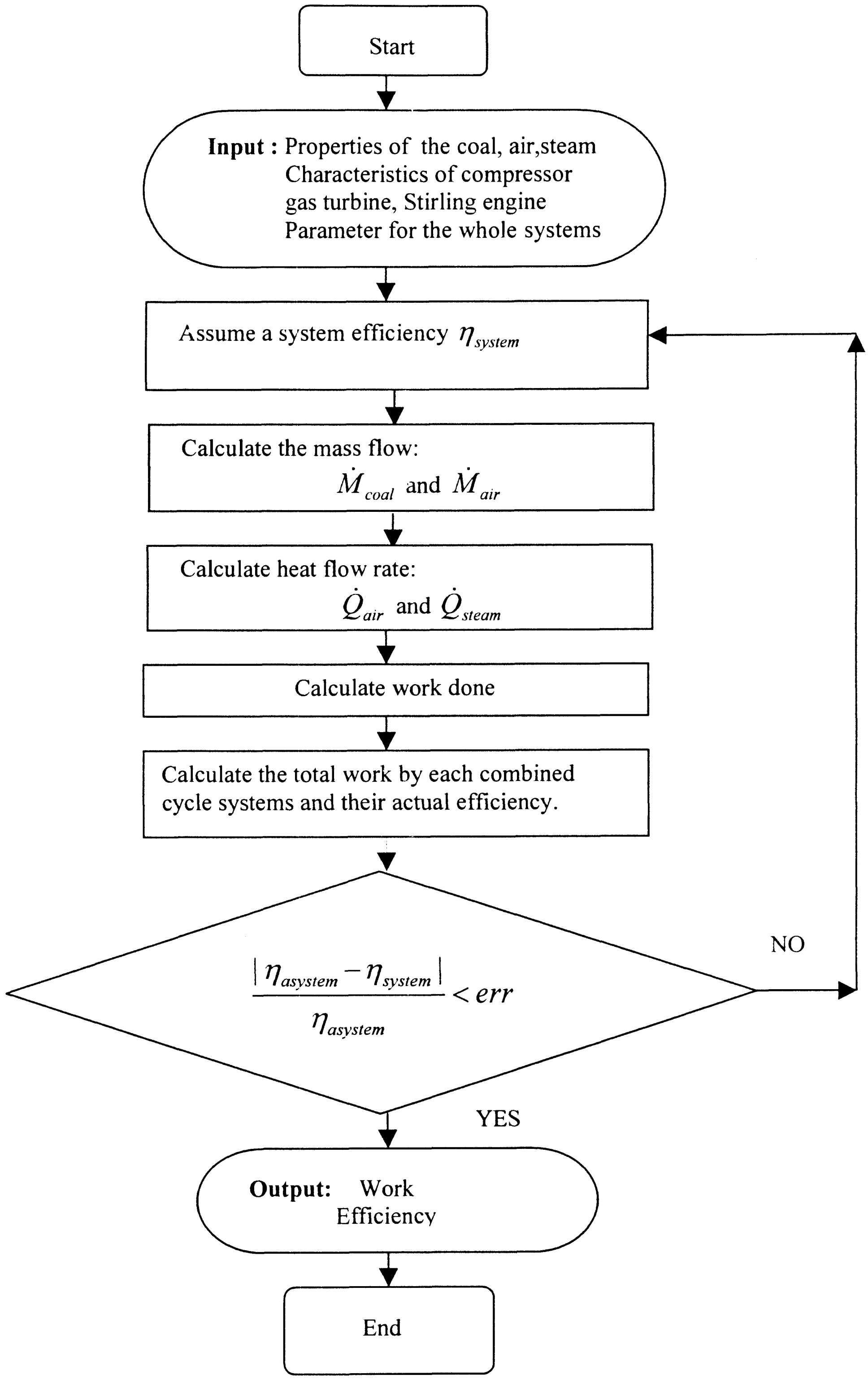


Figure 4-3 System calculation flow chart

At the beginning of the program, all the known parameters (as listed in the last section) of the system are entered into the input file. Assuming the system efficiency is η_{assume}, let the system efficiency equal the assumed efficiency; that is $\eta_{system} = \eta_{assume}$. According to the system output $\dot{W}_{out}$ and the system efficiency η_{system}, the heat, mass, and work of the whole system can be calculated using the formulas introduced in the last section. Finally, the actual system efficiency $\eta_{asystem}$ can be calculated according to the heat flow and work. Using $\eta_{asystem}$ instead of η_{system}, iterate the whole procedure described until the relative error of the system efficiency is less than the maximum error permitted. In this research, the maximum relative error is 0.005, that is $\left|\frac{\eta_{asystem} - \eta_{ystem}}{\eta_{asystem}}\right| < 0.005$.

Both the Stirling engine combined cycle system and the gas turbine combined cycle system have the same calculation procedure, except that the work done by the Stirling is different from that of the gas turbine.

4.4 Simulation of A 20MW Stiring Engine Combined Cycle

4.4.1 Properties for 20MW Combined Cycle System

The following tables (Table 4-1 through Table 4-4) give the key parameters to perform the calculations for a 20MW Stirling engine combined cycle system and the gas turbine combined cycle system. They are the properties of the air, coal, steam, water, and the characteristics of the main devices in the system. All of them are practically chosen.

Table 4-1 Properties of the Coal

Item	Symbol	Mass (%)
Carbon	C	80.7
Hydrogen	H_2	4.5
Sulfur	S	1.8
Oxygen	O_2	2.4
Nitrogen	N_2	1.1
Water	H_2O	3.3

Table 4-2 Characteristics of the Compressor, Gas Turbine and Steam Turbine

Item	Symbol	Value
Number of stages	N_{com}	4
Compressing ratio of each stage	π	1.8
Gas turbine isotropic efficiency	$\eta_{gas\_turbine}$	By Equation (4-24)
Steam turbine efficiency	$\eta_{steam\_turbine}$	.62

Table 4-3 Properties of the Air

Item	Symbol	Unit	Value
Universal gas constant	R	$kJ/kg \cdot K$	2.870×10^{-1}
Specific heat at constant pressure	C_p	$kJ/kg \cdot K$	1.004
Specific heat ratio C_p/C_v	k		1.400
Compressor inlet temperature	T_1	K	293
Compressor outlet temperature	T_2	K	503
Stack inlet temperature	T_6	K	413
Stirling engine outlet temperature	$T_{4'}$	K	1083
Compressor inlet pressure	P_1	Pa	1.013×10^5
Compressor outlet pressure	P_2	Pa	1.113×10^6
Gas turbine inlet pressure	P_3	Pa	1.016×10^6
Gas turbine outlet pressure	P_4	Pa	1.013×10^5
Extra air coefficient	α	Pa	1.20

Table 4-4 Properties of Steam and Water

Item	Symbol	Unit	Value
Condensed water enthalpy	h_1	kJ/kg	100
Preheated water enthalpy	h_2	kJ/kg	860.4
Main vapor enthalpy	h_3	kJ/kg	3394.0

The technique parameters of the Stirling engine can be seen in Table 3-1 in page 11.

4.4.2 Computational Results and Analysis

The computational results for Stirling engine and gas turbine combined cycles are shown in from Figure 4-4 through Figure 4-6. The detailed output data can be referred to in the output file in Part II of Appendix II.

Figure 4-4 shows the working efficiency curves of the gas turbine and the compressor. From this graph, the gas turbine and compressor exhibit maximum efficiencies when the whole system is running at base load (100%), and their efficiencies decrease when the system output decreases. This characteristic of gas turbines and compressors is important to the efficiency of the gas turbine combined cycle system, especially when running on a fraction of maximum load, know as reduced load. The thermal efficiency of the Stirling engine will not change with the system output if the heat ratio is maintained at a constant value. Its efficiency only changes with the temperature of its heater head, which is directly affected by the temperature of the

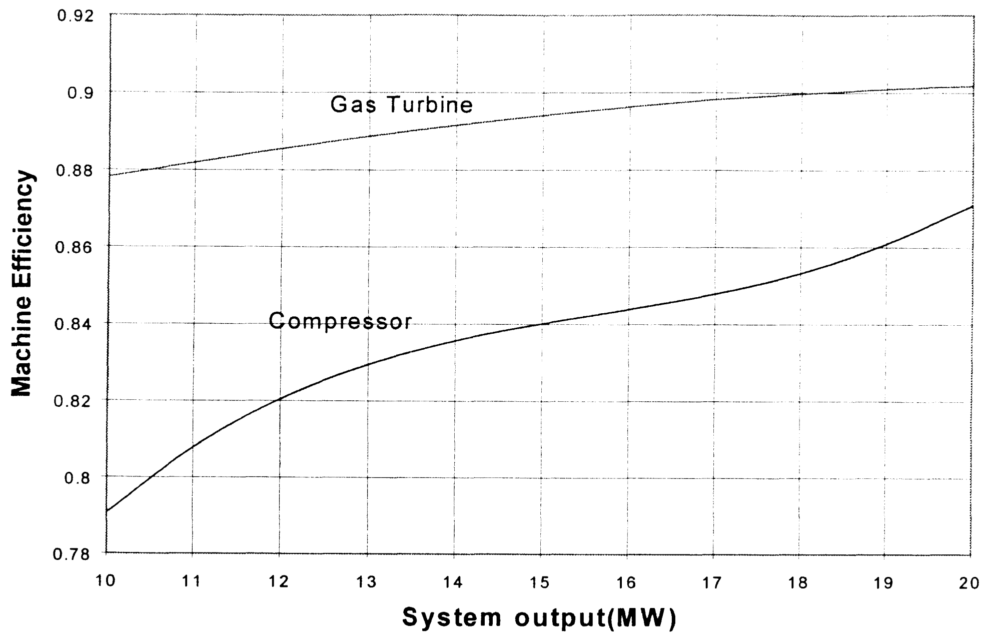


Figure 4-4 Efficiency curves of compressor, gas turbine vs. system output

combustion gas. The relation between the temperature of the combustion gas and the heat ratio could be deduced from the following equations:

$$\dot{M}_{coal} = \frac{\dot{Q}_{coal}}{HHV} \tag{4-2}$$

$$\dot{M}_{air} = AF * \alpha * \dot{M}_{coal} \tag{4-3}$$

$$\dot{Q}_{air} = \gamma_Q * \dot{Q}_{coal} \tag{4-4a}$$

$$T_3 = \frac{\dot{Q}_{air}}{\dot{M}_{air} C_p} + T_2 \tag{4-7}$$

Substituting equation (4-2), (4-3) and (4-4a) into equation (4-7) yields

$$T_3 = \frac{HHV}{AF * \alpha * C_p} \gamma_Q + T_2 \tag{4-26a}$$

Let $m = \dfrac{HHV}{AF * \alpha * C_p}$, therefore,

$$T_3 = m\gamma_Q + T_2 \tag{4-26b}$$

As m is a constant, Equation (4-26b) shows that the temperature of the combustion gas is linear with the heat ratio γ_Q. If γ_Q is constant, T_3 will remain the same; thus the efficiency of the heat exchanger will be unchanged. Therefore, even in reduced load, the Stirling engine can maintain a high thermal efficiency.

In Figure 4-5 the combustion gas temperature curve vs. the heat ratio is illustrated. This figure exhibits that when the heat ratio is larger than 0.4, the temperature of the hot gas will exceed 1100°C (1373K). But according to the design calculation of the heat exchanger in Chapter 4, the hot gas design temperature is 1100°C. A higher temperature will hurt the heater head of the Stirling engine. For the gas turbine, if the working gas temperature exceeds 1100°C, the metal blades could sustain damage. Therefore, the optional heat ratio of both combined cycle systems is assumed to be 0.4.

In Figure 4-6 the system efficiency curves of 20MW combined cycle system vs. the system output when the heat ratio is 0.4 is plotted. The figure shows that when running with base load, the efficiency of the Stirling engine combined cycle is 0.5262 -- higher than that of the gas turbine combined cycle system, which is 0.5178. In addition, when the system output is lower than the base load, both the system efficienciesof Stirling engine combined cycle and gas turbine combined cycle decrease. But the Stirling engine combined cycle still has a higher efficiency than the gas turbine combined cycle system. Finally, when the system output differs from 10MW to 20MW, the slope of the

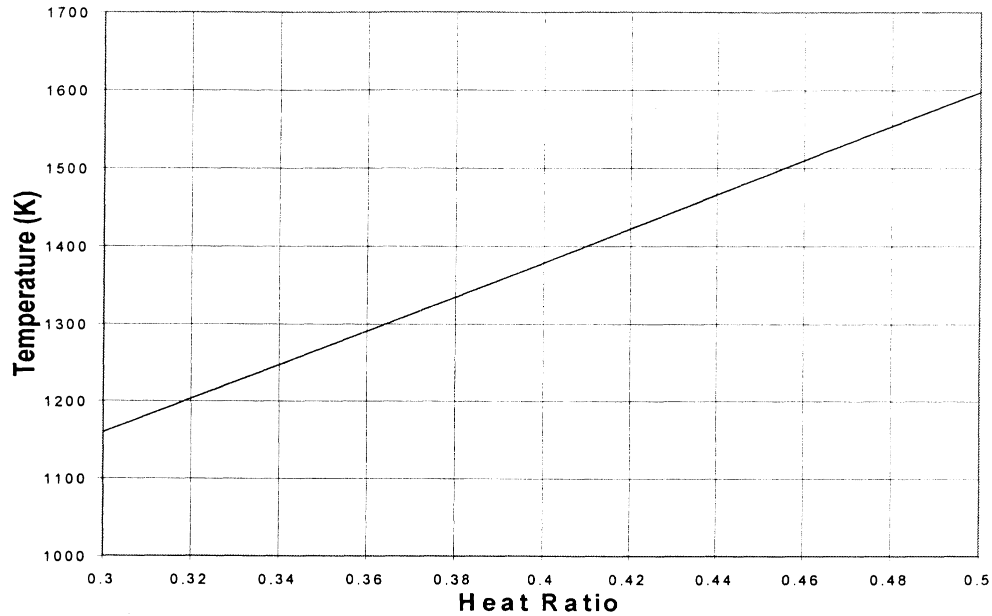


Figure 4-5 Curve of hot gas temperature vs. heat ratio

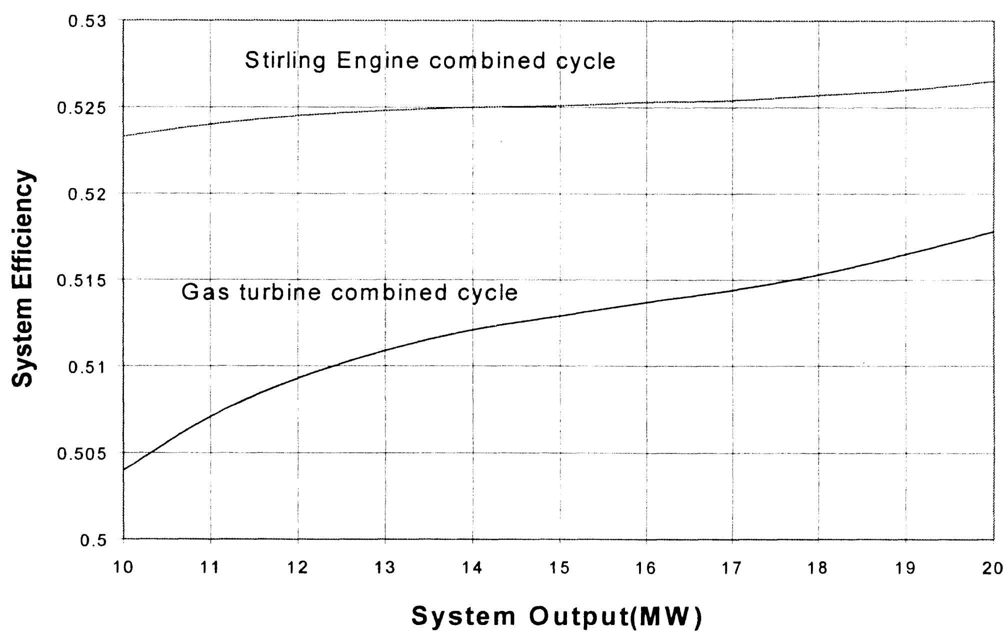


Figure 4-6 System efficiency curves vs. system output

efficiency curve of the Stirling engine combined cycle system is much less than that of the gas turbine combined cycle system. This means that the Stirling engine combined cycle system has a better reduced-load efficiency than that of the gas turbine combined cycle system. This is because that the Stirling engine maintains constant efficiency in reduced-loads, while the efficiency of the gas turbine and the compressor will significantly change.

Generally speaking, for a 20MW combined cycle system with heat ratio of 0.4, Stirling engine combined cycle will have higher system efficiency than gas turbine combined cycle in both base load and reduced load. The Stirling engine combined cycle has a better reduced-load performance than that of the gas turbine combined cycle system.

Chapter 5 Economic Analysis

5.1 Introduction

This chapter presents the methodology for estimating the cost of a Stirling engine combined cycle system and gas turbine combined cycle system, which is based on the calculated results of the computational program. The economic analysis includes the estimation of the annualized capital costs, the operating costs (including maintenance), the fuel costs, and the opportunity generation costs.

5.2 Machine Selections

Before doing a cost estimation for a system, the machines used in the system should be selected. The selections of the machines are based on the system calculation results, such as the output of the gas turbine and steam turbine, the capacity of the compressor, the total output of the Stirling engine, etc. Also the market supply of the machines has to be considered, because there is a great possibility that the machines with the exact parameters attained from the calculation are not supplied in the current market.

5.2.1 Selection of the Steam Turbine

Assume the capabilities of the steam turbine are the same for both the gas turbine combined cycle system and the Stirling engine combined cycle system. From the computational program, the steam turbine base outputs for both the two systems can be calculated. By choosing the maximum output of the two systems as the reference capacity of the steam turbine and rounding it up to a value whose corresponding steam turbine could be purchased in the market, the steam turbine can be determined. For example, if the calculated base output of the steam turbines were $12.5\,MW$ and $13.5\,MW$ for the gas turbine combined cycle and the Stirling engine combined cycle respectively, 13.5MW should be chosen as the reference capacity of the steam turbine and generator. If there are only a series of the steam turbines whose capacities are $10\,MW$, $12\,MW$, $15\,MW$ 20MW etc, the $15\,MW$ steam turbine and generator system should be chosen.

5.2.2 Machine Selections for the Gas Turbine Combine Cycle

The gas turbine output is determined by $\dot{W}_{gas\_turbine}$, the calculated output work of the gas turbine on base load. Often, there is no gas turbine - generator whose output is exactly equal to $\dot{W}_{gas\_turbine}$. The one that could be purchased in the present market and whose output is the nearest to $\dot{W}_{gas\_turbine}$ should be selected.

The type of the compressor depends on its capacity and compressing pressure ratio. The selected compressor should be able to be purchased in the market and its capacity, and compressing pressure ratio is consistent with the calculation results.

The main attached devices of the gas turbine cycle include the blade cooler (which is used to lower the temperature of the blade), and the air clean-up device (which is used to lower the density of the ash in the combustion air). They can be selected according to the chosen gas turbine, the temperature, mass flow, and composition of the combustion gas. Both of these devices are expensive and will raise the overall cost.

5.2.3 Machines Selection for the Stirling Engine Combined Cycle

As described in Chapter 3, the $100\,kW$ STM-120 Stirling engine was chosen for this research. The number of the Stirling engines is a step function of the total output of the Stirling engine. That is,

$$N = \left\lceil \frac{\dot{W}_{Stirling}}{100} \right\rceil \tag{5-1}$$

The heat exchanger was designed in Chapter 3. The number of heat exchangers is equal to the number of Stirling engines used in the overall system.

Once the types and numbers of the devices are determined, their current purchase price, and lifetimes are determined.

5.3 Annualized Cost Estimation

5.3.1 General Method to Estimate Annualized Costs

Assume the number of years for the analysis of a system is m and the lifetime for a certain machine is n years. If $n < m$, the number of times the machine should be replaced during m years is

$$k_m = \left\lceil \frac{m}{n} \right\rceil -1 \tag{5-2}$$

Not only must the current cost of this machine be considered, but also its future cost, as it will be replaced. In order to make the annualization of cost easier, the following equation is used to estimate the future cost as a present value.

$$F = \frac{PV}{(1+i)^x} \tag{5-3}$$

where, F is the present value of the future cost (x years later) for a m -years analysis

PV is the current capital cost of the machine

i is the interests of the money

x is the number of years after which the machine has to be replaced

So the total investment should be

$$PT = \sum_{j=0}^{j=k_m} \frac{PV}{(1+i)^{jn}} \tag{5-4}$$

where, PT is the total investment of the machine for a m-years analysis

PV is the current capital cost of the machine

j is the indice

k_m is the number of times the machine need to be replaced during m years

n is the lifetime of the machine

The capital cost can be annualized using the following formula:

$$A = PT \frac{i \cdot (1+i)^m}{(1+i)^m - 1} \tag{5-5}$$

where, A is the annualized cost

m is the number of year of the analysis.

Substitute (5-4) into (5-5) yielding

$$A = \left(\sum_{j=0}^{j=k} \frac{PV}{(1+i)^{jn}} \right) \cdot \frac{i(1+i)^m}{(1+i)^m - 1} \tag{5-6}$$

Equation (5-6) can be used to estimate the annualized cost for both the Stirling engine and gas turbine system.

5.3.2 Annualized Costs of Combined Cycle Systems

After the machines for both the gas turbine combined cycle system and the Stirling engine combined cycle system are chosen, the capital cost and the lifetime for each machine should be determined. Their replacement times during m years can be calculated by using Equation (5-2). Table 5-1 lists parameters of the cost, lifetime and replacement time for the machines in term of generic parameters, including PV, n, k_m

For a m-years analysis with an interest rate of i, the annualized cost of the gas turbine combine cycle system is

$$A_G = A_{steam} + A_{gast} + A_{com} + A_{attatched} \tag{5-7a}$$

where, A_{steam}, A_{gast}, A_{com}, $A_{attached}$ are the annualized capital costs of the steam turbine, gas turbine, compressor and the attached device (such as air clean-up device) respectively. Applying Equation (5-6) to each machine in (5-7a) yields

$$A_G = \left\{ \left(\sum_{j=0}^{j=k_{m1}} \frac{PV_{steam}}{(1+i)^{jn_1}} \right) + \left(\sum_{j=0}^{j=k_{m2}} \frac{PV_{gast}}{(1+i)^{jn_2}} \right) + \left(\sum_{j=0}^{j=k_{m3}} \frac{PV_{com}}{(1+i)^{jn_3}} \right) + \left(\sum_{j=0}^{j=k_{m4}} \frac{PV_{attached}}{(1+i)^{jn_4}} \right) \right\} \left(\frac{i(1+i)^m}{(1+i)^m - 1} \right) \tag{5-7b}$$

Table 5-1 Capital Cost, Lifetime and Replacement Time of the Devices

Machine Name	Capital Cost ($)	Lifetime(year)	Rep. Times
Steam turbine	PV_{steam}	n_1	k_{m1}
Gas turbine	PV_{gast}	n_2	k_{m2}
Compressor	PV_{com}	n_3	k_{m3}
Gas turbine attached dev.	$PV_{attached}$	n_4	k_{m4}
Stirling engines(N units)	PV_{Stir}	n_5	k_{m5}
Heat exchanges(N units)	PV_{exch}	n_6	k_{m6}
Fan	PV_{fan}	n_7	k_{m7}

For the Stirling engine combined cycle system, the annualized cost is given as

$$A_S = A_{steam} + A_{Stir} + A_{exch} + A_{fan} \tag{5-8a}$$

where, A_{steam}, A_{Stir}, A_{exch}, A_{fan} are the annualized cost of the steam turbine, Stirling engine, the heat exchanger, and the fan respectively. Applying Equation (5-6) to each machine in (5-8a) yields

$$A_S = \left\{ \left(\sum_{j=0}^{j=k_{m1}} \frac{PV_{steam}}{(1+i)^{jn_1}} \right) + \left(\sum_{j=0}^{j=k_{m5}} \frac{PV_{Stir}}{(1+i)^{jn_5}} \right) + \left(\sum_{j=0}^{j=k_{m6}} \frac{PV_{exch}}{(1+i)^{jn_6}} \right) + \left(\sum_{j=0}^{j=k_{m7}} \frac{PV_{fan}}{(1+i)^{jn_7}} \right) \right\} \left(\frac{i(1+i)^m}{(1+i)^m - 1} \right) \tag{5-8b}$$

5.4 Other Cost Estimations

5.4.1 Fuel Cost Estimation

If the capacity of a system is $\dot{W}$ (MW), its efficiency is η, and the higher heating value of the fuel is HHV (kJ/kg), then the mass flow rate of the fuel should be

$$\dot{M}_{fuel} = \frac{\dot{W}}{\eta} \cdot \frac{1}{HHV} \qquad \text{(Ton/s)} \qquad (5\text{-}9)$$

Assume the system will work for d days each year. As there is 24 ×3600 second in a day, the total fuel used each year will be

$$\dot{M} = \dot{M}_{fuel} \cdot (d \times 24 \times 3600) \qquad \text{(Ton/year)} \qquad (5\text{-}10)$$

If the cost of the fuel is f (\$/Ton), the total cost of the fuel will be

$$PV_{fuel} = f \cdot \dot{M}_{fuel} \cdot (d \times 24 \times 3600) \qquad \text{(\$/year)} \qquad (5\text{-}11)$$

Substitute Equation (5-9) into (5-11) yielding

$$PV_{fuel} = f \cdot \frac{\dot{W}}{\eta} \cdot \frac{1}{HHV} (d \times 24 \times 3600) \qquad \text{(\$/year)} \qquad (5\text{-}12)$$

Equation (5-12) can be used to calculate the yearly fuel cost for a system whose output is $\dot{W}$ and whose efficiency is η. Substituting the system output and system efficiency into Equation (5-12), the fuel cost of the gas turbine combined cycle system and the Stirling turbine combined cycle system can be calculated.

5.4.2 Operating Cost and Downtime Cost

The cost used to operate and maintain a power plant system is called the operating cost. When the equipment at a power plant system breaks and the system can not generate electricity, the power plant must pay to buy electricity from another source. This cost is called downtime cost or opportunity cost. Both must be included in the economic analysis for combined cycle systems. Their values can be practical chosen referring to existing power plants whose output is similar to the one in the research.

5.5 Economic Analysis Example

In this section, an example is given to illustrate the methodology to do the economic analysis for the Stirling engine combined cycle system and the gas turbine combined cycle system using the methodology provided in this chapter. The total capacity for each of the two systems is $15\,MW$. The capacity of the steam turbine for each of the systems is $12\,MW$. The capacity of the gas turbine is $4\,MW$, while the capacity for Stirling engines is $3.5\,MW$. The thermal efficiencies of the two systems are 51.2% for the gas turbine combined cycle system and 51.8% for the Stirling engine combined cycle. The purchase price and lifetime cost for the relative machines are listed in Table 5-2.

Assume the two systems will use coal as fuel. The HHV of the coal is 14,800 kJ/kg. The price of the coal is 100\$/Ton. The operating cost and the downtime cost of the gas turbine combined cycle system are $0.5\,M\$/year$ and $0.25\,M\$/year$, respectively.

For the Stirling engine combined cycle system, they are $0.42\,M\$/year$ and $0.28\,M\$/year$, respectively. These costs of the Stirling engine combined cycle may be smaller than gas turbine because Stirling engines are easier to maintain. The two systems are assumed to work for 340 days each year. The analysis will be for 30 years with an interest rate of 9%. (Note: All the numbers given in this problem is only for illustration purposes.)

Table 5-2 Capital Cost, Lifetime and Replacement Time of Selected Devices

Machine Name	Capital Cost(M \$)	Lifetime(year)	Rep. Times
Steam turbine	10	30	0
Gas turbine	2.5	30	0
Compressor	1	15	1
Gas turbine attached devices	0.8	10	2
Stirling engines (35 units)	35×0.08	15	1
Heat exchangers (35 units)	35×0.02	15	1
Fan	0.3	30	0

Solution:

From Table 5-2, the capital cost of the steam turbine is $10\,M$ \$ for both systems. The capital cost of the gas turbine is $2.5\,M$ \$. The capital cost of the Stirling engine is $35\times0.08 = 2.8\,M$ \$. The total annualized cost of the gas turbine combined cycle system is \$1,465,000 obtained by substituting the known values in Table 5-2 into Equation (5-7b).

$A_G =$

$$\left\{PV_{steam} + PV_{gast} + \left(PV_{com} + \frac{PV_{com}}{(1+i)^{n_3}}\right) + \left(PV_{attached} + \frac{PV_{attached}}{(1+i)^{n_3}} + \frac{PV_{attached}}{(1+i)^{2n_3}}\right)\right\}\left(\frac{i(1+i)^m}{(1+i)^m - 1}\right)$$

$$= \left\{10 + 2.5 + \left(1 + \frac{1}{(1+0.09)^{15}}\right) + \left(0.8 + \frac{0.8}{(1+0.09)^{10}} + \frac{0.8}{(1+0.09)^{20}}\right)\right\}\left(\frac{0.09(1+0.09)^{30}}{(1+0.09)^{30} - 1}\right)$$

$$= \{10 + 2.5 + 1.274 + 1.281\} \times 0.097336$$

$$= 1.465(M\$/year)$$

The annualized cost of the Stirling engine combined cycle system is \$1,437,000 obtained by substituting the known values in Table 5-2 into Equation (5-8b).

$$A_S = \left\{PV_{steam} + \left(PV_{Stir} + \frac{PV_{Stir}}{(1+i)^{n_5}}\right) + \left(PV_{exch} + \frac{PV_{exch}}{(1+i)^{n_6}}\right) + PV_{fan}\right\}\left(\frac{i(1+i)^m}{(1+i)^m - 1}\right)$$

$$= \left\{10 + \left(2.8 + \frac{2.8}{(1+0.09)^{15}}\right) + \left(0.7 + \frac{0.7}{(1+0.09)^{15}}\right) + 0.3\right\}\left(\frac{0.09(1+0.09)^{30}}{(1+0.09)^{30} - 1}\right)$$

$$= (10 + 3.569 + 0.892 + 0.3) \cdot 0.097336$$

$$= 1.437(M\$/year)$$

Using equation (5-12), the total cost of fuel for the gas turbine combined cycle system is \$5,820,000 each year.

$$PV_{fuel\_gast} = f \cdot \frac{W}{\eta_{gast}} \cdot \frac{1}{HHV}(d \times 24 \times 3600)$$

$$= 100 \times \frac{15}{0.512} \cdot \frac{1}{1.48 \times 10^4}(340 \times 24 \times 3600)$$

$$= 5.82(M\$/year)$$

Using equation (5-12), the total fuel cost for the Stirling engine combined cycle system is \$5,750,000 each year.

$$PV_{fuel\_Stir} = f \cdot \frac{W}{\eta_{Stir}} \cdot \frac{1}{HHV}(d \times 24 \times 3600)$$

$$= 100 \times \frac{15}{0.518} \frac{1}{1.48 \times 10^4}(340 \times 24 \times 3600)$$

$$= 5.75(M\$/year)$$

The total cost for the two systems should be the sum of the annualized cost, annually fuel cost, annual operating cost and annually downtime cost.

For the gas turbine combined cycle system, the total annual cost is found to be \$8,035,000.

$$C_{gast\_sys} = A_G + PV_{fuel\_gast} + PV_{oper\_gast} + PV_{dtime\_gast}$$

$$= 1.465 + 5.82 + 0.5 + 0.25$$

$$= 8.035(M\$/year)$$

For the Stirling engine combined cycle system, the total annual cost is found to be \$7,887,000.

$$C_{Stir\_sys} = A_S + PV_{fuel\_Stir} + PV_{oper\_Stir} + PV_{dtime\_Stir}$$

$$= 1.437 + 5.75 + 0.42 + 0.28$$

$$= 7.887(M\$/year)$$

The calculation results show that the Stirling engine combined cycle system has a lower annual cost than the gas turbine combined cycle system.

Chapter 6 Conclusions

6.1 Conclusions

This thesis presents a combined cycle model -- the Stirling engine/Steam turbine combined cycle model -- for distributed power generation and explores its viability by comparing it, from technical aspects, with the gas turbine combined cycle system. It also provides a methodology for comparison of the economics of the two combined cycles.

The ideas of the Stirling engine combined cycle system are as follows. First, the Stirling engine is a device that has very high thermal efficiency; it is also possible for the Stirling engine and steam turbine combined cycle to achieve a high thermal efficiency to generate electricity. Second, the Stirling engine is an externally heated machine. Therefore, the fuel type has less of an effect on its operation. It is also possible to use low quality fuels as the energy resource for the system. Using an external combustion system also offers possibilities for a greater level of pollution control compared to an internal combustion system. And third, the Stirling engine combined cycle does not require compressing cold air, cleaning up the combustion gas, or cooling of internal components, all of which may possibly make the system simpler than the gas turbine combined cycle.

The disadvantages of this modeled Stirling engine combined cycle include the need for an extraordinary heat exchanger, capable of transporting a very high heat flux, and limited power output. This limited output restriction may necessitate several engines in a given system, for even a small amount of generation. Finally, the large-scale Stirling engines are not well proven and continued research and development is needed.

However, by using heat pipe techniques, it is possible to design a compact heat exchanger to transport a very high heat flux for large-scale Stirling engines. Thus it is possible to couple the large-scale Stirling engines, such as a 100kW STM-120 machine, with a steam turbine to generate electricity. The simulation results of a 20MW Stirling engine combined cycle shows that the efficiency of the Stirling engine combined cycle is competitive with that of the gas turbine combined cycle for both the base load and reduced-load generation when the temperature of the combustion gas from the boiler is 1100°C. Also, the Stirling engine combined cycle system shows good reduced load performance when the system load changes.

For small-scale power generation, the Stirling engine combined cycle system can compete with gas turbine combined cycle system on such aspects as fuel flexibility, thermal efficiency, and reduced load performance, but it can not compete with gas turbine on scale. Stirling engine combined cycles will have great potential for distributed power generation, especially when using low quality fuels.

6.2 Recommendations

This thesis studies the viability of the large-scale Stirling engine combined cycle for distributed power generation by comparing it with the gas turbine combined cycle. The results show that the Stirling engine combined cycle is competitive with the gas turbine combined cycle for cases of small power generation. Further work on this project is needed to compare the Stirling engine combined cycle with micro-turbine-based combined cycles. The micro-turbine is very similar to the gas turbine. Compared with the gas turbine, micro-turbine can minimize the compressing work and maintain very high thermal efficiencies (>58%). Therefore, the micro-turbine combined cycle system may be more economical than gas turbine combined cycles for distributed power generation. Comparing the Stirling engine combined cycle with the micro-turbine combined cycle will further the study of the viability of the large scale Stirling engine combined cycle for the distributed power generation.

Although this thesis builds models for Stirling engine combined cycle and gas turbine combined cycle, and develops a computational program to simulate its work, the models are simplified. In order to get more exact calculation results, a more detailed model should be built, and compound simulations of the system performance are required.

Finally, a specific application case study should be done to prove the viability of the Stirling engine combined cycle. This application can be, for example, distributed power generation for a shopping mall, a hospital, or a hotel. The performance of the application should be evaluated by computer simulation. A complete economic analysis based on its simulation results is also needed. Examination of the potential markets of this application is also required in further research.

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Appendix I Design Calculation for the Heat Exchanger

Part I Combustion Gas Side Surface Design

No.		Description	Formula or equation	Unit	Value
1.	$\dot{Q}$	Heat flow	$\dot{Q} = \dot{m}C_p(t_1'' - t_1')$	kW	240
2.	$\dot{m}$	Mass flow of the air	$\dot{m} = \dfrac{\dot{Q}}{C_p(t_1'' - t_1')}$	kg/s	0.625
3.	Δt	Logarithm mean temperature	$\Delta t = \dfrac{(t_1' - t_2') - (t_1'' - t_2'')}{\ln \dfrac{t_1' - t_2'}{t_1'' - t_2''}}$	°C	85.25
4.	h_a	Convection heat transfer coefficient	Assumed	$W/m^2 \cdot K$	150
5.	$\dot{Q}$	Heat flow	$\dot{Q} = A_1 h_a \Delta t$	kW	240
6.	A_1	Surface area required, Assume $R_1 = R_2 = 0$	$A_1 = \dfrac{\dot{Q}}{h_a \Delta t}$	m^2	18.69
7	α_f	Geometry parameter of the fin	$\alpha_f = \dfrac{s}{h}$		0.672
8	γ_f	Geometry parameter of the fin	$\gamma_f = \dfrac{t}{s}$,		0.084
9	δ_f	Geometry parameter of the fin	$\delta_f = \dfrac{t}{l}$		0.04

10	N_x	Number of fins in the x direction			15
11	N_y	Number of the fins in the z direction			35
12	N_z	Number of the fins in the y direction			34
13	A_1	Actual surface area	$A_1 = N_x N_y N_z l \times 2(s+h)$	m^2	18.667
14	A_c	Across area in the flow direction:	$A_c = N_x N_y (s+h)$	m	0.0782
15	D_h	Hydraulic diameter	Equation (3-8)	m	0.0116
16	V	Velocity V	$V = \frac{\dot{m}}{\rho A_c}$	m/s	27.69
17	Re	Reynold number	$\mathrm{Re} = \frac{VD_h}{\nu}$		1945
18		Ref. Reynold number	Equation (3-7)		876
19	B	Coefficient	$B = 0.2162$		0.2162
20	$b1$	Coefficient	$b1 = +0.5937$		0.5937
21	$b2$	Coefficient	$b2 = -0.1037$		-0.104
22	$b3$	Coefficient	$b3 = +0.1955$		+0.196
23	$b4$	Coefficient	$b4 = -0.1733$		−0.173
24	Nu	Nusselt number	$B(\mathrm{Re})^{b1}(\alpha_f)^{b2}(\delta_f)^{b3}(\gamma_f)^{b4}$		16.539

25	C	Coefficient	$C = 1.8699$		1.8699
26	$a1$	Coefficients	$a1 = -0.299$		-0.299
27	$a2$	Coefficient	$a2 = -0.094$		-0.094
28	$a3$	Coefficient	$a3 = 0.682$		0.682
29	$a3$	Coefficients	$a3 = +0.6820,$		-0.243
30	h_c	Convection heat transfer coefficient	$h_c = \frac{Nu\lambda}{D_h}$	$W/m^2 \cdot K$	150.69
31	err	Relative error of h_c	$err = \frac{\lvert h_c - h_a \rvert}{h_c}$		0.0045
32	f	Friction factor	$C(\mathrm{Re})^{c1}(\alpha_f)^{c2}(\delta_f)^{c3}(\gamma_f)^{c4}$		0.0571
33	ΔP	Pressure drop	$\Delta P = f\left(\frac{1}{2}\rho V^2\right)\left(\frac{A_1}{A_c}\right)$	Pa	1570.7

Part II Heat Pipe Design

The heat pipe can be design by using the following steps:

1. **Determine the geometry of the heat pipe.**

   Based on the combustion gas side surface design, the heat pipe heat exchanger is configured as shown in Figure Appendix I-1. The geometric parameters of the heat pipe, as labled in the design picture, are dynamically determined. They may change when calculating the heat flow rate limits. For example, if one of the heat flow rate limits is smaller than the heat exchanger capacity, those geometric parameters may change to achieve a higher heat flow rate for the heat pipe.

2. **Choose the working fluid**

As the heater head must maintain a temperature of 800°C, the sodium was selected as the working fluid of the heap pipe, because it has an operating temperature range of 600-1200°C and attractive thermodynamic properties such as high thermal conductivity, high latent heat, good thermal stability. The properties of sodium at 800°C are listed in Table Appendix I-1 [Dunn, 1994].

3. **Choose the wall material of the heat pipe**

The wall material was selected as Nb +1%Zr, which is compatible with the sodium.

4. **Choose the wick**

The selection of wick is mainly depends on the properties of the working fluid and the heat transport capacity of the heat pipe. For one hand, the material of the wick must be compatible with the working fluid--the sodium. On the other hand, the wick should be able to generate capillary pressure to transport the working fluid through the evaporator. The capacity of the wick is mainly determined by its pore size and thickness. The heat transport capacity of the heat pipe can be raised by increasing the wick thickness or decreasing its pore size [Dunn, 1994]. The numbers of the layers and the mesh density were determined during the calculations of the maximum heat flow rates. If the maximum heat flow rates related to the wick are smaller than the capacity of the heat pipe, more layers of wick or wick with higher density of mesh (smaller pore size) should be used. In this design, three layers of #500 mesh 304

Table Appendix I-1 Properties of Sodium at 800°C

Vapor pressure	$P_V = 0.47$ Bar
Latent heat of vaporization	$\lambda_V = 3977 kJ/kg$
Liquid density	$\rho_l = 757.3 kg/m^3$
Vapor density	$\rho_v = 0.134 kg/m^3$
Liquid thermal conductivity	$K_l = 57.81 w/(m^2.K)$
Liquid viscosity	$\mu_l = 0.18 \times 10^{-3} kg/(m.s)$
Vapor viscosity	$\mu_v = 0.22 \times 10^{-4} kg/(m.s)$
Surface tension	$\sigma = 1.23 \times 10^{-1} N/m$
Specific heats ratio	$\gamma_V = 1.358$

stainless steel wick were used. Later calculation will prove that it can provide a heat transport capacity higher than the design capacity of the heat pipe. Table Appendix I-2 lists the main properties of #500 mesh 304 stainless steel wick [Peterson, 1994].

5. **Calculate the heat flow rate limits**

According to the geometry of the heat pipe, the properties of the working fliud and the wick selected, the capillary limit, sonic limit, entrainment limit, boiling limit and the viscous limit can be calculated. Compare those limits with the capacity of the heat exchanger. If all the limits are larger than the required heat flow rate, the design works; otherwise, the geometry of the heat pipe or the wick has to be changed until

Table Appendix I-2 Properties of #500 mesh 304 stainless steel wick

Wire diameter	$d_w = 0.00085$ in
Conductivity	$K_w = 22.6 W/m^2 \cdot K$
Mesh density	$N_w = 500\, mesh/in$
Wick porosity	$\varepsilon_w = 0.6495$

all the limits are larger that the heat exchanger design capacity.

Those heat flow rate limits can be calculated as followings:

1) The capillary limit ($\dot{Q}_c$)

The capillary limit can be calculated by Equation (3-16)

$$\Delta P_{c,m} \geq \Delta P_l + \Delta P_V + \Delta P_i + \Delta P_g \tag{3-16}$$

Those items in Equation (3-16) can be calculated as followings:

a) $\Delta P_{c,m}$ --The maximum capillary pumping head

$$\Delta P_{c,m} = (\frac{2\sigma}{r_{c,e}}) = \frac{2 \times 1.23 \times 10^{-1}}{2.54 \times 10^{-5}} = 9685(Pa)$$

where σ is the surface tension of the working fluid liquid

$r_{c,e}$ is the effective capillary radius, which can be calculated as

$$r_{c,e} = \frac{1}{2N_w} = \frac{1}{2 \times 500 mesh/in} = 0.001 in = 2.54 \times 10^{-5} m$$

where N_w is the number of meshes in one inch of the wick. For the #500 mesh 304 stainless steel wick, N_w is equal to 500 $mesh/in.$

b) ΔP_V -- Pressure drop necessary to cause the vapor to flow from the evaporator to the condenser

Assume the vapor flow is a laminar flow, ΔP_V can be calculated by using Equation (3-18a)

$$\Delta P_V = \frac{16\mu_V L_{eff} \dot{Q}_c}{2r_{h,V} A_V \rho_V \lambda_V} \tag{3-18a}$$

where L_{eff} is the effective vapor section length which can be calculated as

$$L_{eff} = L_e + \frac{L_a + L_c}{2} = 0.25 + \frac{0.38 + 0.1}{2} = 0.49\ m$$

where, L_e, is the length of the evaporator section, In this design, it is equal to $0.38\,m$.

L_a is the length of the adiabatic section of the heat pipe. In this design, it is equal to $0.25\,m$.

L_c is the length of the condenser section and is equal to $0.1\,m$

$r_{h,V}$ in Equation (3-18a) is the hydraulic radius of the evaporator. It can be evaluated as

$$r_{h,V} = \frac{1}{2} \times \frac{4ab}{2(a+b)} = \frac{1}{2} \times \frac{4 \times 0.714 \times 0.02}{2(0.714 + 0.02)} = 0.0195m$$

where a, b are the length and width of the evaporator slot, respectively. They are equal to $0.714\,m$ and $0.02\,m$ respectively. (See Figure Appendix I-1)

A_V in Equation (3-18a) is the cross section area of the vapor space and can be calculated as

$$A_V = 16 \times ab = 16 \times 0.714 \times 0.02 = 0.2285 m^2$$

Substituting the values of L_{eff}, $r_{h,V}$, A_V and the other properties of the sodium into the Equation (3-18a) gives

$$\Delta P_V = \frac{16 \times 0.22 \times 10^{-4} \times 0.49 \times \dot{Q}_c}{2 \times 0.0195^2 \times 0.2285 \times 0.134 \times 3913000} = 1.893 \times 10^{-6} \dot{Q}_c (Pa)$$

c) ΔP_i -- The inertial pressure gradient

For a laminar flow,

$$\Delta P_i = 0 \qquad (3\text{-}19a)$$

d) ΔP_g -- Pressure required to lift the liquid to the highest place where there are wicks

ΔP_g can be calculated by Equation (3-20) as following:

$$\Delta P_g = \rho_l g h = 757.3 \times 9.81 \times 0.38 = 2823 Pa$$

e) ΔP_l -- Pressure requires to drag the liquid from the condenser to the evaporator

In this design, the heat pipe is a gravity-assisted heat pipe. Therefore,

$$\Delta P_l = 0 \qquad (3\text{-}21)$$

Substituting the former calculational results into Equation (3-16) yields

$$9685 \geq 0 + 1.893 \times 10^{-6} \dot{Q}_c + 0 + 2823$$

Solve this equation giving

$$\dot{Q}_c \geq 3{,}625 \times 10^3 kW$$

The result is based on the assumption that the vapor flow is a laminar flow. To check the assumption, calculate the Reynold number of the vapor flow as following:

$$\mathrm{Re}_V = \frac{2r_{h,V}\dot{Q}_c}{A_V \mu_V \lambda_V} = \frac{4\dot{Q}_c}{\pi d_V \mu_V \lambda_V} = 2749 \times 10^4 >> 2300$$

Therefore, the assumption that the vapor flow is laminar flow is wrong. ΔP_V and ΔP_i should be recalculated by Equation (3-18b) and Equation (3-19b), respectively.

$$\Delta P_V = 0.38\left(\frac{d_V \dot{Q}_c}{A_V \mu_V \lambda_V}\right)^{\frac{3}{4}} \cdot \frac{2\mu_V L_{eff} \dot{Q}_c}{d_V^{\,2} A_V \rho_V \lambda_V} = 0.1 \times 10^{-9} \dot{Q}_c^{\frac{7}{4}} \qquad \text{(a)}$$

$$\Delta P_i = \frac{1.22\dot{Q}_c^{\,2}}{g\rho_V A_V^{\,2} \lambda_V^{\,2}} = 1.125\dot{Q}^2{}_c \qquad \text{(b)}$$

Substituting Equation (a) and (b)into Equation (3-16) yields

$$9685 \geq 0.1 \times 10^{-9} \dot{Q}_c^{\frac{7}{4}} + 1.125 \times 10^{-9} \dot{Q}_c + 2823$$

Solving this equation gives

$$\dot{Q}_c \geq 2460 kW$$

Thus, the capillary limit is

$$\dot{Q}_c = 2460 kW > 240 kW$$

Therefore, the capillary limit satisfied.

2) The sonic limit ($\dot{Q}_s$)

The sonic limit could be calculated by Equation (3-22):

$$\dot{Q}_s = \rho_V \lambda_V \sqrt{\frac{\gamma_V R T_V}{2(\gamma_V + 1) m_m}} \times A_V \tag{3-22}$$

where, γ_V is the specific heat ratio of the sodium, $\gamma_V = 1.358$

R is the general gas constant, $R = 8.314 kJ/(kg \cdot mol \cdot K)$

T_V is the vapor temperature which is 1073 K (800°C)

m_m is the molecule mass of sodium, which is equal to 23 kg/mol

Substituting those values into Equation (3-22) yields

$$\dot{Q}_s = 1287 kW > 240 kW$$

Therefore, the sonic limit is satisfied.

3) The entrainment limit ($\dot{Q}_{ent}$)

The entrainment limit can be determined by equation (3-23):

$$\dot{Q}_{ent} = n_s \pi r_{h,V}{}^2 \lambda_V \sqrt{\frac{2\pi \rho_V \sigma \cos\theta}{C_h}} \tag{3-23}$$

where, n_s is the number of the slot of the evaporator

r_V is the radius of vapor space

C_h is the characteristic dimension of the liquid/vapor interface

θ is the contact angle

In this design, $n_s = 16$, $C_h = 0.036$ (for fine mesh), $\theta = 0$.

Substituting those value and the properties of sodium into Equation (3-23) yields

$$\dot{Q}_{ent} = 16 \times \pi \times 0.0195^2 \times 3977 \times \sqrt{\frac{2\pi \times 0.134 \times 0.123 \times 1}{0.036 \times 10^{-3}}} = 4077 kW > 240\, kW$$

Therefore, entrainment limit is satisfied.

4) Boiling limit ($\dot{Q}_b$)

The boiling limit can be calculated by Equation (3-24)

$$\dot{Q}_b = \frac{2\pi L_{eff} k_{eff} T_V}{\lambda \rho_V \delta_w} \left(\frac{2\sigma}{\gamma_n} - \Delta P_{c,m}\right) \qquad (3\text{-}24)$$

where K_{eff} is the effective conductivity coefficient of the wick-liquid interface. K_{eff} can be calculated as

$$K_{eff} = \frac{K_l(K_l + K_w) - (1-\varepsilon_w)(K_l - K_w)}{(K_l + K_w) + (1-\varepsilon_w)(K_l - K_w)}$$

where K_l is the liquid thermal conductivity

K_w is the thermal conductivity of the wick material

ε_w is the wick porosity

For the sodium, $K_l = 57.81\, W / m^2 \cdot K$

For the wick selected, $K_w = 22.6 W / m^2 \cdot K$, ε_w=0.6495

Substituting the values of K_l, K_w, ε_w into the equation of K_{eff} yields

$$K_{eff} = 42.43 W / m^2 \cdot K$$

δ_w is the sickness of the wick. Assume the thickness of each layer of screen is equal to twice the wire diameter, as there are three layers, the thickness can be calculated as

$$\delta_w = 3 \times 2 \times d_w = 3 \times 2 \times 0.00085 inch = 1.295 \times 10^{-4} m$$

r_n is the nucleation site radius, which is assumed to be $2.54 \times 10^{-7} m$

$\Delta P_{c,m}$ is the maximum capillary pressure difference generated within capillary, which is 9685 Pa by former calculation

Therefore, $\dot{Q}_b$ can be evaluated as

$$\dot{Q}_b = \frac{2\pi L_{eff} k_{eff} T_V}{\lambda_V \rho_V \delta_w} (\frac{2\sigma}{\gamma_n} - \Delta P_{c,m})$$

$$= \frac{2\pi \times 0.49 \times 42.43 \times (800 + 273)}{3977000 \times 0.134 \times 1.295 \times 10^{-4}} \times \left(\frac{2 \times 0.123}{2.54 \times 10^{-7}} - 9685 \right)$$

$$= 1947400 kW >> 240 kW$$

The boiling limit is much larger that the heat exchanger capacity, therefore, the boiling limit is satisfied.

5) Viscous limit

Former calculations yielded

$$\Delta P_V = 0.1 \times 10^{-9} \dot{Q}_c^{\frac{7}{4}} \quad \text{(a)}$$

and

$$\dot{Q}_c = 2460 kW = 2460 \times 10^3 W$$

Therefore,

$$\Delta P_V = 0.1 \times 10^{-9} \times (2460 \times 10^3)^{\frac{7}{4}} = 15.28 Pa$$

Thus

$$\frac{\Delta P_V}{P_V} = \frac{15.28}{0.47 \times 10^5} < 0.1$$

The viscous limit should be avoided.

The former results shows that all the limits to heat transport are larger than the design capacity of the heat exchanger. Therefore, the design should be sufficient.

Part III Condenser Design

The surface of the condenser can be calculated by using Equation (3-25b).

$$A_{cond} = \frac{\dot{Q}_{cond}}{h_{cond} \Delta t_{cond}} = \frac{240 \times 10^3}{5100 \times (800 - 720)} = 0.588 m^2$$

The condenser is actually the Stirling engine heater. The Stirling engine designer does the surface arrangement of the heater. Figure Appendix I-1 shows a typical Stirling engine heater. In this design, a short pipe was used to equip the Stirling engine heater into the heat pipe (see Figure Appendix I-2). The diameter and the length of this short pipe should be determined by the geometry of the Stirling engine heater.

[Harfreaves, 1991]

Figure Appendix I-1 The heater used in 30-15 Stirling engine

Part IV Checking Calculations for the Heat Exchanger

In the combustion gas side surface design, the thermal resistance of the heat pipe and the condenser are assumed to be ignoreable. After the design calculation, checking calculation should be performed to prove the rightness of this assumption. Table Appendix I-3 lists six different thermal resistances for the heat pipe heat exchanger, with their definitions and their referential calculated values (All the definitions and the calculation formulas for the thermal resistances are referred to in Peterson, 1994). Those values of the thermal resistances shows that the heat source side thermal resistance ($R_{ext,\,e}$) is much larger than the other thermal resistances. Therefore, the assumption that the thermal resistances of the heat pipe and the condenser are not significant is correct.

Table Appendix I-3 Thermal Resistances of the Heat Pipe Heat Exchanger

Item	Definition	Value(K/W)
$R_{ext,e}$	the contact resistance occurring between the heat source and the heat pipe evaporator	3.567×10^{-4}
$R_{p,e}$	the thermal resistance due to the pipe wall in the evaporator	1.671×10^{-6}
$R_{w,e}$	the resistance of the liquid-wick combination	3.42×10^{-7}
R_V	the thermal resistance of the vapor flow	≈ 0
$R_{p,c}$	The thermal resistance due to the pipe wall in the condenser	3.12×10^{-7}
$R_{\text{int},c}$	contact resistance occurring between the heat sink and the heat pipe condenser	1.36×10^{-6}

Part V Design Graph

Figure Appendix I-2 is the final design graph for the heat exchanger.

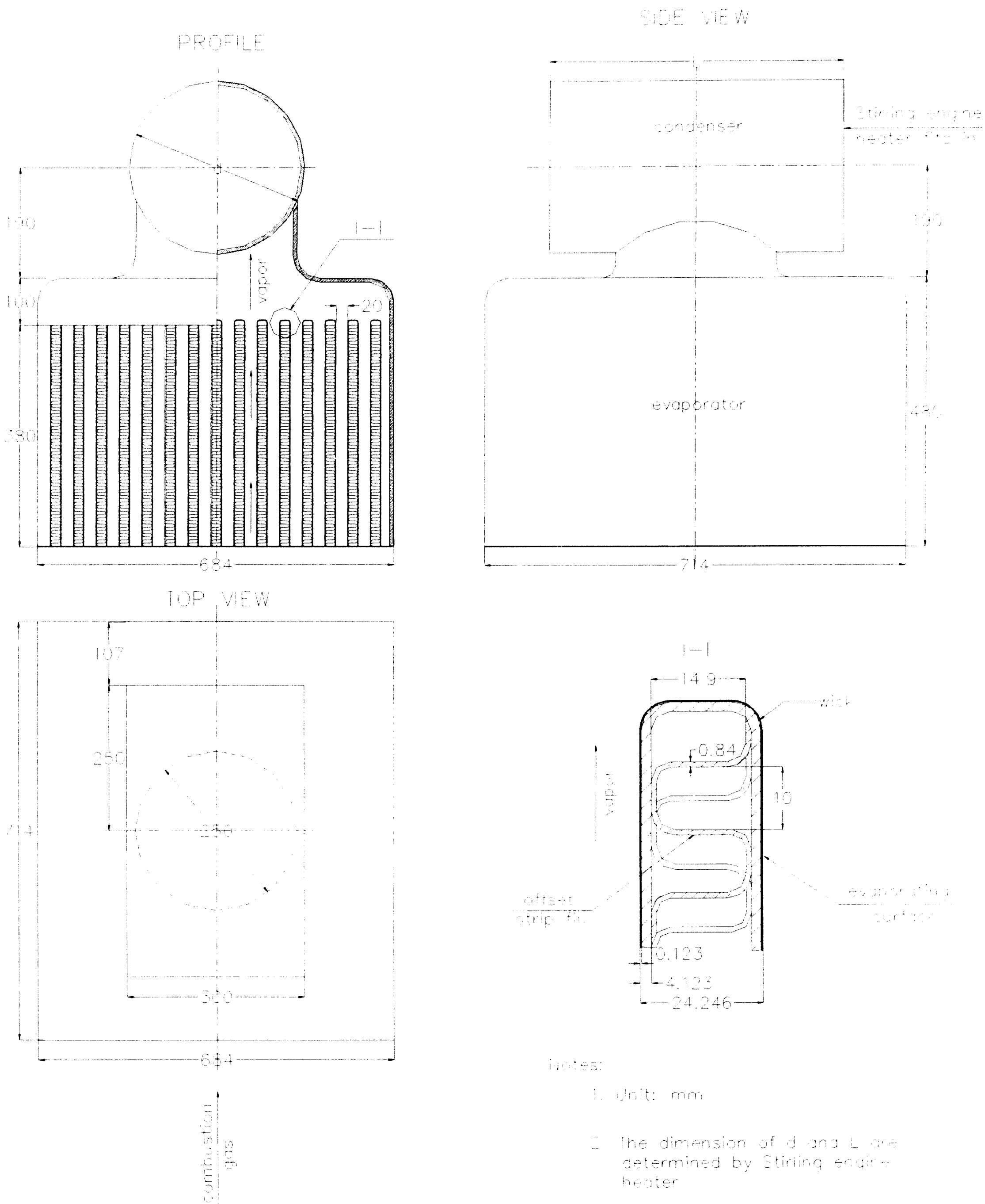

Figure Appendix I-2 Heat exchanger design picture

Appendix II Computer Code and Output

Part I Source Code for the Program

```
#     makefile for the progam power
#     By Hua Liang
#     Date: 21/10/97


OBJECT = define.o parameter.o calculation.o stirling.o print.o
power:power.c power.h ${OBJECT}
      gcc power.c ${OBJECT} -o power -lm

define.o: define.c
      gcc -c define.c

parameter.o: parameter.c
      gcc -c parameter.c

calculation.o: calculation.c
      gcc -c calculation.c

stirling.o: stirling.c
      gcc -c stirling.c

print.o: print.c
      gcc -c print.c
```

```
/**********************************************************************
 * FILENAME: power.h
 * PROGRAMMER: Hua Liang
 * DATE:14/10/97
 * DISCRIPTION: THis is the header file for the combined cycle system
  of the power system
 **********************************************************************/

#ifndef POWER_H
#include<stdio.h>
#include<stdlib.h>
#include<math.h>
#define POWER_H
#define WSTEP 100.00
#define RATIO_STEP 0.01
#define EERROR 0.005
#define WERROR 0.1
#define MAX_EFF 0.4500
#define MIN_EFF 0.3000

/********* FUNCTON PROTOTYPE ***********************/
void define(void);
void parameter(void);
void heat(double ratio);
void mass(double ratio);
void calAcr(void);
void comWork(void);
void gturbWork(void);
void steam_turb(void);
void print(void);
void efficiency(void);
void gast_steam();
void steam_stir();
void compareStirling(void);
void stir_output(void);
double calComEff(double M );    /*calculate the efficiency of the
                                compressor.*/
double calGastEff(double M);    /*calculate the efficiency of the gas
                                turbine. */
double calEffStirling(double T);/*calculate the efficiency of the
                                  stirling engine */
void stir_work(void);/* calculate the work done by stirling engine. */
void output(void);   /* output the results for a certain Wout */
void menu(void);
void base_load(void);
void diff_load(void);
void diff_heat_ratio(void);
void another(void);

/*****  GLOBAL VARIABLES - specified in function "define" *****/
FILE *infile;       /*pointer to the input data file */
FILE *outfile;      /*pointer to the output data file */
FILE *printfile;    /*pointer to the print output file */
```

```
FILE *printfile1;   /* pointer to the print outfile file1, which
                     output the gas turbine combined system efficiency.*/

/******* Prototype of the global parameter *********************/

int choice;         /* menu choice*/
int debug ;

/*** Parameter for the air.***/
double R ;    /* Gas constant      */
double Cp;    /* specific heat at const pressure (J/kg.K) */
double k;     /* ratio (cp / cv) */
double T1;    /* preheated air temperature before get into the
                 compressor.*/
double T2;    /* tempreture of the compressed air after the
                 compressor.*/
double T3;    /* air tempreture before getting into the gas turbine.*/
double T4;    /* air tempreture after the gas turbine. */
double T4s;   /* air tempreture after the gas turbine for an adiabatic
                 procedure. */
double T5;    /* air tempreture after the econominer  */
double T6;    /* air temptrture after the air preheater. */
double Tair_stirling; /* the hot air tempreture before get into the
                         stirling engine */
double P1;    /* air pressure before the compressor */
double P2;    /* air pressure after the compressor */
double P3;    /* gas pressure before the gas turbine or after the
                 furnace.*/
double P4;    /* gas pressure after the gas turbine. */
double alfa;         /* extra air coefficient */
double acRatio;      /* Air to coal ratio    */
double Mair ;        /* mass flow of the air */

/******** Parameter of the coal *******/

double carbon;      /* ingredient of carbon */
double hydrogen;    /* ingredient of hydrogen */
double sulfur;      /* ingredient of sulfur. */
double oxygen;      /* ingredinent of oxygen */
double nytrogen;    /* ingredient of nytrogen */
double water;       /* ingredient of water. */
double HHV;         /* High Heat Value of the coal. */
double Mcoal;       /* mass flow of the coal */

/********Paramter of the compressor for the gas turbine *******/

double Ecom;         /* efficiency of the compressor */
int Nstage;        /* stage of the compressor */
double  gama;        /* compressor ratio P2/P1  */
double Wcom;         /* work comsuption by the compressor. */
double maxFlow;      /* the maximum flow of the compressor. */
double percentOfMaxFlow; /* percentage of the maxFlow. */
```

```
/******* Gas turbine parameter **************/

double Egasturb;      /* The efficiency of the gas turbine. */
double Wgasturb ;     /* Work doing by the gas turbine. */
double Wgtnet;        /* gas turbine net work. */

/******* Steam turbine efficiency ***********.*/

double Esteamturb;    /* efficiency of the steam turbine. */
double Wsteamturb;    /* Work doing by the steam turbine. */

/********** Steam and water parameters ******************/

double h1;         /* The tempreture of the condensor water. */
double h2;         /* The tempreture of the preheated water. */
double h3;         /* The tempreture of the vapour */
double Cp_cwater; /* the specific heat of the cold water */
double Cp_hwater; /* The specific heat of the hot water. */
double Cp_vapor;  /* The specific heat of the vapor */
double Msteam;    /* The mass of the steam */
double Mextract;  /* The mass flow that extracted from the main vapor*/

/**** Parameters for Stirling Steam_turbine Combined System ******/

double Estirling;          /* Effiency of the stirling engine. */
double Wstirling;          /* Work done by the stirling engine. */
double Tin_stirling;       /* temperature of the inlet air to the
                              stirling engine */
double Mair_stirling;      /* the mass flow through the stirling
                              engine*/
double acRatio_stirling;   /* The air and coal mass ratio for the
                              stirling_steam combined system. */
double alfa_stirling;      /* The extra air coefficient */
int numStirling ;          /* the number of the stirling engine
                              required*/
double loadCoeff;          /* the load coefficient of the stirling
                              engine */
double Qin_stirling;       /* the heat get into the stirling engine */
double Wstirling_steam;    /* the tatal output work of the
                              stirling_steam combined system */
double Estir_system;        /* the efficiency of the stirling_steam
                               combined system. */
double Eastir_system;       /* The actual efficiency of the stir_steam
                                system.*/
double Efan;                /* the efficiency of the fan using in the
                               system*/
double Wfan_stirling;       /* work consumed by the fan of the
                               stirling_steam combined system. */
double Wfan;
double  gama_fan;           /* compressor ratio P2/P1 of the fan in the
                               stirling_steam combined cycle.  */
double effStep;             /* The iteration increasement of the
                                stirling system */
```

```
/****** Gas turbine _Steam turbine combined System parameter *****/

double Eassume;           /* the assumption efficiency of the system */
static double Esystem;    /* Efficiency of the system */
double Easystem;          /* actual system efficiency */

double Wdesign;           /* the designed output of the system  */
double Wout;              /* Total output work.(including baseload or
                             changed load. */
double Waout;             /* actual net work by the system. */
double Qassume_ratio;     /* the assuption heat ratio */
double Qagast_ratio;      /* the actual heat ratio of the gas turbine */
double Qgast_ratio;       /* the assumed heat ratio(Qair: Qsteam) */
double Qstir_ratio;       /* the assumed heat ratio of the Stirling */
double Qastir_ratio;      /* the acual heat ratio of the Stirling.*/
double Qfurnace;           /* Heat provided by the furnace. */
double Qcoal;              /* Heat generated from the coal. */
double Qextract;           /* Heat exttract from the vapour to heat the
                              condensor water */
double Q4_6;               /* Heat provided by the air to the
                              the economoner and the preheater*/
double Qpreheat;           /* Heat required in the economoner and the
                              preheater*/
double Qair;               /* Heat goes to the gas. */
double Qsteam;             /* Heat goes to the steam. */
double load_coeff;         /* The load coeff of the system. */
#endif
```

```
/*****************************************************
 * FILNAME: power.c
 * PROGRAMMER: Hua Liang
 * DATE: 14/10/1997
 * DISCRIPTION:
 *****************************************************/
#include<stdio.h>
#include"power.h"


/*********************************************************
 * FUNCTION: main()
 * DESCRIPTION: main function of the program
 *********************************************************/
main()
{
  define();
  system("clear");
  menu();
  return;
}

/************************************************************
 * FUNCTION: void menu(void)
 * DESCRITPTION: This function provides a menu for the user.
 ************************************************************/
void menu(void)
{
  int i;

   printf("\n\n\n\n");
   printf("\t\t ********** Please choose the calculation type
*********\n\n");
   printf("\t\t 1> Calculation for Base Load Differs From 1MK
   ~20MK\n\n");
   printf("\t\t 2> Calculation for Different Load with a Certain
   Output\n\n");
   printf("\t\t 3>Calculation for Different Heat Ratio\n\n");
   printf(" \t\t please input 1, 2, or 3  ====>  ");
   while(1)
       {
        scanf("%d", &choice);
        printf("\n");
        if(choice==1||choice==2||choice==3)
        break;
       }

     switch(choice)
         {
           case 1:base_load();break;
           case 2:diff_load();break;
           case 3:diff_heat_ratio(); break;
           default:perror("input error.\n"); exit(1);
```

```
        }
 return;
}


/**************************************************************
 * FUNCTION: void base_load()
 * DESCRIPTIUON: THis function calculate the system efficient for
 *    both Stirling engine and gas turbine combined cycle system
 *    for different base load.
 **************************************************************/
void base_load()
{
  int i;
  load_coeff = 1.0;

  printf("Wdesign\t\t Easystem\t\tEastir_system\n");
  for(i=1; i<=5; i++)
    {
      Wdesign  = i  * 1000;
      gast_steam( );
      stir_steam( );
      printf("%.4f\t\t%.4f\t\t%.4f\n", Wdesign, Easystem,
      Eastir_system);
    }
  Wdesign = 2000;
  gast_steam();
  printf("Wgast=%.4f\tWcom%.4f\t\tWsteamt=%.4f\n",
          Wgasturb, Wcom, Wsteamturb );
  stir_steam();
  printf("Wstir=%.4f\tWfan=%.4f\tWsteamt=%.4f\n",
          Wstirling, Wfan, Wsteamturb );

  another();
}


/******************************************************************
 * FUNCTION: vooid diff_load(void)
 * DESCRIPTION: This function calculates the efficiency of different
 *     load for a system with a fixed designed output.
 ******************************************************************/
void diff_load()
{
  double ratio;

  printf(" Please input the system design output:");
  scanf("%lf", &Wdesign);

  printf(" Please input the lowest load coefficient:");
  scanf("%lf", &load_coeff);
```

```
  printf(" Please input the heat ratio:");
  scanf("%lf", &ratio);

  Qgast_ratio = Qstir_ratio=ratio;

  printf("\n\n For the system design output of %.2f\n", Wdesign);
  fprintf(printfile, "\n\n For the system design output of %.2f\nKW",
          Wdesign);
  printf("\n For the gas turbine combined cycle system:\n\n");
  fprintf(printfile, "\n For the gas turbine combined cycle
  system:\n\n");
  printf("Wout\t\tEasystem\tEcom\tEgasturb\n\n");
  fprintf(printfile,"Wout\t\tEasystem\tEcom\tEgasturb\n\n");
  gast_steam( );

  printf("\n\nFor the stirling engine combined cycle system:\n\n");
  fprintf(printfile,"\n\nFor the stirling engine combined cycle
  system:\n\n");
  printf("Wout\t\tEastir_system\n");
  fprintf(printfile, "Wout\t\tEastir_system\n");
  stir_steam( );

  another();
}


/*******************************************************************
 * FUNCTION: diff_heat_ratio
 * DESCRIPTION: This function calculates the system efficient with
 *     different heat ratio

********************************************************************/
void diff_heat_ratio()
{
 int i;
 double ratio, ratio1, ratio2;

 printf("Please input the system design output:\n");
 scanf("%lf", &Wdesign);

 printf("Please input the highest heat ratio: \n");
 scanf("%lf", &ratio1);

 printf("Please input the lowest heat ratio:\n");
 scanf("%lf", &ratio2);

 load_coeff = 1.0;
 ratio = ratio2;

 printf(" For the system design output of %.2f\n", Wdesign);
 fprintf(printfile, "\n\nFor the system design output of %.2f\n",
 Wdesign);
 printf("ratio\t\tEasystem\tEastir_system\t T3\n");
```

```
 fprintf(printfile, "ratio\tEasystem\tEastir_system\t T3\n");

 while(1)
   {
     Qgast_ratio = Qstir_ratio=ratio;
     gast_steam( );

     stir_steam( );
     printf("%.4f\t\t%.4f\t\t%.4f\t %.2f\n", ratio, Easystem,
          Eastir_system, T3);
     fprintf(printfile, "%.4f\t  %.4f\t  %.4f\t  %.2f\n", ratio,
     Easystem,
           Eastir_system, T3);
     ratio = ratio + 0.01;
     if(ratio > ratio1 +0.01 )
       break;
   }
 another();
}

/****************************************************************
 * FUNCTION: void another()
 * DESCRIPTION:This function provides an interface for the user
 *     to  continue another kind of calculation.
 ****************************************************************/
void another()
{
  int ch;

  printf("\n\n Continue another calculation? 1) Yes 2) No ");
  scanf("%d", &ch);
  if(ch==1)
    {
      system("clear");
      menu();
    }
  else
    exit(0);
}
```

```
/************************************************************
 * FUNCTION define
 * FILENAME:      define.c
 * PROGRAMMER:     Dr. Israel Urieli
 * Modifier: Hua Liang
 * DATE:           15/10/97
 * DESCRITPTION: This function defines the parameters used in this
 *               program
 * ***********************************************************/
#include <stdio.h>
#include <stdlib.h>
#include "power.h"

void define(void)
{
  char new, filename[11];
  printfile = fopen("print.data", "w");
  if(printfile == NULL) {
    printf("\"printfile\" could not be opened\n");
    exit(1);
  }
  printfile1 = fopen("print1.data", "w");
  if(printfile == NULL) {
    printf("\"printfile\" could not be opened\n");
    exit(1);
  }
   printf("create a new data file? (y/n): ");
   new = getchar();
   fflush(stdin);
   if (new == 'y') {
     printf("enter the data filename (max 10 chars): ");
     scanf("%10s", filename);
     fprintf(printfile,"\nnew data filename:  %s \n",filename);
     outfile = fopen(filename, "w");
     if (outfile == NULL) {
     printf("\"%s\" could not be opened\n", filename);
     exit(2);
      }
      infile = stdin;
   }else {
     printf("enter the input data filename (max 10 chars): ");
     scanf("%10s", &filename);
     fprintf(printfile,"\ninput data filename:  %s\n", filename);
     infile = fopen(filename, "r");
     if (infile == NULL) {
       printf("\"%s\" could not be opened\n", filename);
       exit(3);
     }
     outfile = stdout;
   }
   parameter();
 }
```

```
/*****************************************************************
 * FILENAME: parameter.c
 * PROGRAMMER: Hua Liang
 * DATE: 15/10/97
 * DISCRIPTION: This file inputs the paramters air, water and steam
 *              of the gas, compressor, gas turbine and steam turbine.
 *****************************************************************/
#include"power.h"
#include<stdio.h>


/******************************************************************
*FUNCTION: void parameter()
*DESCRIPRION: This function defines the parameters for the whole
*systems
******************************************************************/
void parameter()
{
  printf("  The parameter of the air:\n");
  fprintf( printfile, "I.   The parameter of the air:\n\n");

  printf(" Enter the gas constant R: (kJ/kg.k)\n");
  fscanf(infile,"%lf", &R);
  fprintf(outfile,"    %.3e \n", R);
  fprintf(printfile, "Gas constant R (kJ/kg.k)\n");
  fprintf(printfile, "    %.3e \n", R);

  printf(" Enter the specific heat at constant pressure of the air
        Cp:(kJ/kg.k\n");
  fscanf(infile,"%lf", &Cp);
  fprintf(outfile,"    %.3e \n", Cp);
  fprintf(printfile, "specific heat Cp (kJ/kg.k)\n");
  fprintf(printfile, "    %.3e \n", Cp);

  printf(" Enter the specific heat ratio k \n");
  fscanf(infile,"%lf", &k);
  fprintf(outfile,"    %.3e \n", k);
  fprintf(printfile, "specific heat ratio :\n");
  fprintf(printfile, "    %.3e \n", k);

  printf(" Enter the air temperature before getting into the compressor
          T1: (K) \n");
  fscanf(infile,"%lf", &T1);
  fprintf(outfile,"    %.3e \n", T1);
  fprintf(printfile, "air temperature before compressing (K):\n");
  fprintf(printfile, "    %.3e \n", T1);


  printf(" Enter the air temperature before getting into the boiler T2:
  (K) \n");
  fscanf(infile,"%lf", &T2);
  fprintf(outfile,"    %.3e \n", T2);
  fprintf(printfile, "air tempreture before boiler (K):\n");
```

```
fprintf(printfile, "    %.3e \n", T2);


printf(" Enter the air temperature before outlet to the air T6: (K)
\n");
fscanf(infile,"%lf", &T6);
fprintf(outfile,"    %.3e \n", T6);
fprintf(printfile, "air temperature before boiler (K):\n");
fprintf(printfile, "    %.3e \n", T6);

printf(" Enter the air pressure before getting into the compressor
         P1 : (Pa) \n");
fscanf(infile,"%lf", &P1);
fprintf(outfile,"    %.3e \n", P1);
fprintf(printfile, "air pressure before compressing (Pa):\n");
fprintf(printfile, "    %.3e \n", P1);

printf(" Enter the air pressure after compressing P2: (Pa) \n");
fscanf(infile,"%lf", &P2);
fprintf(outfile,"    %.3e \n", P2);
fprintf(printfile, "air pressure after compressing (Pa):\n");
fprintf(printfile, "    %.3e \n", P2);

printf(" Enter the air pressure before gas turbine P3: (Pa) \n");
fscanf(infile,"%lf", &P3);
fprintf(outfile,"    %.3e \n", P3);
fprintf(printfile, "air pressure before gas turbine (Pa):\n");
fprintf(printfile, "    %.3e \n", P3);

printf(" Enter the air pressure after gas turbine P4: (Pa) \n");
fscanf(infile,"%lf", &P4);
fprintf(outfile,"    %.3e \n", P4);
fprintf(printfile, "air pressure after gas turbine (Pa):\n");
fprintf(printfile, "    %.3e \n", P4);

printf(" Enter the extra air coefficient in the furnace alfa:  \n");
fscanf(infile,"%lf", &alfa);
fprintf(outfile,"    %.3e \n", alfa);
fprintf(printfile, "extra air coefficient:\n");
fprintf(printfile, "    %.3e \n", alfa);

printf("\n Enter the coal parameters: \n");
fprintf(printfile,"\n II. coal parameters: \n\n");

printf(" Enter the coal's HHV: (Btu/h)  \n");
fscanf(infile,"%lf", &HHV);
fprintf(outfile,"    %.3e \n", HHV);
fprintf(printfile, "coal's HHV: (kJ/kg)");
fprintf(printfile, "    %.3e \n", HHV);

printf(" Enter the carbon :)  \n");
fscanf(infile,"%lf", &carbon);
fprintf(outfile,"    %.3e \n", carbon);
```

```
fprintf(printfile, "coal's carbon: ");
fprintf(printfile, "    %.3e \n", carbon);

printf(" Enter the hydrogen :)  \n");
fscanf(infile,"%lf", &hydrogen);
fprintf(outfile,"    %.3e \n", hydrogen);
fprintf(printfile, "coal's hydrogen: ");
fprintf(printfile, "    %.3e \n", hydrogen);

printf(" Enter the sulfur:)  \n");
fscanf(infile,"%lf", &sulfur);
fprintf(outfile,"    %.3e \n", sulfur);
fprintf(printfile, "coal's sulfur: ");
fprintf(printfile, "    %.3e \n", sulfur);

printf(" Enter the oxygen:)  \n");
fscanf(infile,"%lf", &oxygen);
fprintf(outfile,"    %.3e \n", oxygen);
fprintf(printfile, "coal's oxygen: ");
fprintf(printfile, "    %.3e \n", oxygen);

printf(" Enter the nitrogen:)  \n");
fscanf(infile,"%lf", &nitrogen);
fprintf(outfile,"    %.3e \n", nitrogen);
fprintf(printfile, "coal's nitrogen: ");
fprintf(printfile, "    %.3e \n", nitrogen);

printf(" Enter the water:)  \n");
fscanf(infile,"%lf", &water);
fprintf(outfile,"    %.3e \n", water);
fprintf(printfile, "coal's water: ");
fprintf(printfile, "    %.3e \n", water);

printf("\nIII. Enter the compressor's parameters:\n\n");
fprintf(printfile,"\n compressor parameters: \n");

printf(" Enter stage of  the compressor Nstage: \n");
fscanf(infile,"%lf", &Nstage);
fprintf(outfile,"    %.3e \n", Nstage);
fprintf(printfile, "Stage of the compressor");
fprintf(printfile, "    %.3e \n", Nstage);

printf(" Enter compressing ratio gama: \n");
fscanf(infile,"%lf", &gama);
fprintf(outfile,"    %.3e \n", gama);
fprintf(printfile, "compressing ratio");
fprintf(printfile, "    %.3e \n", gama);

printf("\n Enter the parameter of the gas turbine:\n");
fprintf(printfile,"\n IX gas turbine paramter:\n\n");

printf(" Enter gas turbine efficiency Egasturb: \n");
fscanf(infile,"%lf", &Egasturb);
```

```
  fprintf(outfile,"    %.3e \n", Egasturb);
  fprintf(printfile, "gas turbine efficiency:");
  fprintf(printfile, "    %.3e \n", Egasturb);

  printf("\n Enter the parameter of the steam turbine:\n");
  fprintf(printfile,"\n X. steam turbine paramter:\n");

  printf(" Enter steam turbine efficiency Esteamturb: \n");
  fscanf(infile,"%lf", &Esteamturb);
  fprintf(outfile,"    %.3e \n", Esteamturb);
  fprintf(printfile, "steam turbine efficiency:");
  fprintf(printfile, "    %.3e \n", Esteamturb);


  printf("\n Enter the parameter of the steam_water :\n");
  fprintf(printfile,"\n XI. steam_water  parameter:\n");

  printf(" Enter condensed water temperature: \n");
  fscanf(infile,"%lf", &h1);
  fprintf(outfile,"    %.3e \n", h1);
  fprintf(printfile, "Condensed water enthalpy:");
  fprintf(printfile, "    %.3e \n", h1);

  printf(" Enter water temperature before getting into the boiler:
\n");
  fscanf(infile,"%lf", &h2);
  fprintf(outfile,"    %.3e \n", h2);
  fprintf(printfile, "water enthalpy before getting into the boiler:");
  fprintf(printfile, "    %.3e \n", h2);

  printf(" Enter vapour temperature before getting into the turbine:
\n");
  fscanf(infile,"%lf", &h3);
  fprintf(outfile,"    %.3e \n", h3);
  fprintf(printfile, "vapor enthalpy before getting into the
turbine:");
  fprintf(printfile, "    %.3e \n", h3);

  printf(" Enter the parameter of the whole system:\n");
  fprintf(printfile,"\n XII. system parameters:\n");

  printf(" Enter system efficiency(guess) Eassume: \n");
  fscanf(infile,"%lf", &Eassume);
  fprintf(outfile,"    %.3e \n", Eassume);
  fprintf(printfile, "system efficiency(Assumed):");
  fprintf(printfile, "    %.3e \n", Eassume);

  printf(" Enter system designed output  Wdesign: (kw) \n");
  fscanf(infile,"%lf", &Wdesign );
  fprintf(outfile,"    %.3e \n", Wdesign);
  fprintf(printfile, "system designed output:(kw)");
  fprintf(printfile, "    %.3e \n", Wdesign);
```

```
  printf(" Enter system heat ratio of gast_steam system
           (Qair: Qsystem):\n");
  fscanf(infile,"%lf", &Qgast_ratio );
  fprintf(outfile,"    %.3e \n", Qgast_ratio);
  fprintf(printfile, "initial assumed  heat ratio of the gast_steam
  system
         (Qair: Qsystem):");
  fprintf(printfile, "    %.3e \n", Qgast_ratio);

   printf(" Enter sytem heat ratio of stir_steam system
                                        (Qair: Qsystem):\n");
  fscanf(infile,"%lf", &Qstir_ratio );
  fprintf(outfile,"    %.3e \n", Qstir_ratio);
  fprintf(printfile, "initial assumed  heat ratio of the stir_steam
  system
                       (Qair:Qsystem):");
  fprintf(printfile, "    %.3e \n", Qstir_ratio);
}
```

```
/**************************************************************
 * FILANAME: calculation.c
 * PROGRAMMER: Hua Liang
 * DATE: 15/10/97
 * DISCRIPTION: This file defines several modules to calculate the heat
 *              and works.
 ***************************************************************/
#include<stdlib.h>
#include"power.h"
#include <math.h>

/*****************************************************************
 *
 * FUNCTION : void gast_steam(void)
 *
 * DISCRIPTION: This function calculates the system efficiency for both
 *       the gasturb_steamturb and the stirling_steamturb combined
 *       cycle.
*****************************************************************/
void gast_steam( )
{
  int count = 1;
  double temp1;
  double Wstep;

  calAcr();              /* Claculate the air to coal ratio. */

  Wout = Wdesign;   /* Calculate from when the load is at least 40%  */

  Wstep = (1-load_coeff)/10 * Wdesign;

  while(1)
    {
      Easystem = Esystem = Eassume;
      while(1)
      {
             mass(Qgast_ratio);
             heat(Qgast_ratio);

             if(count==1)
             maxFlow = 1.1 * Mair;

             comWork();
             gturbWork();
             steam_turb();

             Waout = Wgtnet + Wsteamturb;
             Easystem = (Wgtnet + Wsteamturb)/Qcoal ;

             temp1 = fabs(Easystem - Esystem)/Easystem;

             if( temp1 <= EERROR)
             break;
```

```
            Esystem = Easystem;
      }
      if(choice == 2)
            {
            printf(" %.4f\t%.4f\t   %4f\t%4f\n", Wout, Easystem,
                                                Ecom, Egasturb);

             fprintf(printfile, " %.4f\t%.4f\t   %4f\t%4f\n", Wout,
             Easystem,Ecom, Egasturb);
           }

      count = 0;
      Wout = Wout - Wstep;

      if(choice==2 && Wout<load_coeff * Wdesign)
         break;

      if(Wout<=load_coeff * Wdesign)
        break;
    }

}


/*********************************************************************
 * FUNCTION: void calAcr(void)
 *
 * DESCRIPTION: This function calculates the air to coal ratio.
*********************************************************************/
void calAcr(void)
{
  acRatio =  (carbon/12.011 + hydrogen/2.016 + sulfur/32.06
           - oxygen/32.00) * 32.00 /100.00 /.29;

}

/*********************************************************************
 * FUNCTION : void mass(double ratio)
 * DESCRIPTION: This function mainly calculates heat provided by the
 *      furnace , heat to the air, the Mass flow of the coal and the
 *      mass flow of the air.
*********************************************************************/
void  mass( double ratio )
{
  Qcoal = Wout/Esystem;
  Mcoal = Qcoal/ HHV ;                    /*the unit is kg/s*/
  Mair = acRatio * alfa * Mcoal ;         /*kg/s*/
}
```

```
/*****************************************************************
 * FUNCTION :heat()
 * DESCRIPTION : THis function calculates the heat to gas and to steam;
 *
 *****************************************************************/
void heat(double ratio)
{
  Qair = ratio * Qcoal;
  Qsteam = (1 - ratio) * Qcoal;
}


/*****************************************************************
 *
 * FUNCTION : void comWork(void)
 *
 * DISCRIPTION: This function mainly calculates the work comsumed by
     the compressor.
 *****************************************************************/
void comWork(void)
{
  double temp1;
  double temp2;
  double temp3;

  temp1 = k/(k-1.0);
  temp3 = 1.0/temp1;
  temp2 = pow(gama, temp3);

  Ecom = calComEff(Mair);  /* calculate the compressor efficiency */

  Wcom = (1.0/Ecom) * 4.0 *  Mair * temp1 * R * T1 * (temp2 - 1.0);
}

/*****************************************************************
 * FUNCTION : void gturbWork(void)
 * DISCRIPTION: This function mainly calculates the work done by the
 *    gas turbine.
*****************************************************************/
void gturbWork(void)
{
  T3 = Qair / (Mair * Cp) +T2;
  T4s = exp( R/Cp * log(P4/P3) ) * T3;
                              /* 0 = Cp * log(T4s/T3) - R*log(P4/P3) */

  Egasturb = calGastEff(Mair);
  T4 = T3 - Egasturb *(T3 - T4s);
                              /* Egasturb = Cp*(T3-T4)/(Cp*(T3-T4s)) */
  Wgasturb = Mair * Cp * ( T3 - T4);
  Wgtnet = Wgasturb - Wcom;
 }
```

```
/******************************************************************
 * FUNCTION : void sturbWork(void)
 * DISCRIPTION: This function mainly calculates the work done by the
 *    steam turbine,
*****************************************************************/
void steam_turb(void)
{

  T5 = 0.92 *Mair * Cp * (T2-T1) + T6;
  Msteam = Qsteam * .92 /(h3-h2);
  Qpreheat= ( Mair * Cp * (T2-T1) + Msteam * (h2 - h1) )/0.9;

  /* Heat use to preheat the water and the cold air */

  Q4_6 = Mair * Cp * (T4 - T6);

  if(Q4_6 < Qpreheat)
    Qextract = (Qpreheat - Q4_6)/0.9;
  else
    Qextract = 0;

  Mextract = Qextract/h3;
  Msteam = Msteam - Mextract;
  Wsteamturb = Msteam * (h3 - h1 ) * Esteamturb;
}



/******************************************************************
 * FUNCTION : double calComEff(double M)
 * DISCRIPTION: This function mainly calculates Effciency of the
 *       compressor.
 *****************************************************************/
double calComEff(double M)
{
  double x, y;
  double E;

  percentOfMaxFlow = M/maxFlow * 100;
  x = percentOfMaxFlow;
  y = 1.933428417 * pow(10, -4) * pow(x, 3) -
       4.183340300 * pow(10, -2) * pow(x, 2) + 3.098533254 * x +
       5.879971870;

  E = y/100;
  return (E);
}
```

```
/*******************************************************************
 *  FUNCTION : double calGastEff(double M)
 * DISCRIPTION: This function mainly calculates efficiency of the
 *       compressor.
 *******************************************************************/
double calGastEff(double M)
{
  double x, y;
  double E;

  percentOfMaxFlow = M/maxFlow * 100;
  x = percentOfMaxFlow;
  y = -8.4280303 * pow(10, -4) * pow(x, 2) +
    1.6940530 * pow(10, -1) * x + 8.1742500 * 10;

  E = y/100;
  return (E);
}
```

```
/*******************************************************************
 * FILENAME: stirling.c
 * PROGRAMMER: Hua Liang
 * DATE: 06/12/1997
 * DESCRIPTION: This program is used to calculate the stirling and
 * steam turbine combined system.
*******************************************************************/

#include<stdlib.h>
#include"power.h"

/*******************************************************************
* FUNCTION: void stirling(void)
* DISCRIPTION: This function calculates the mass flow throught the
* stirling engine, the work need to be done by the fan in the system.
*******************************************************************/
void stir_steam(void)
{
  int count;
  double Wstep;

  T4 = 273 + 810;
  Mextract = 0;

  Wout =  Wdesign;  /* Calculate from when the load is at least 40%  */

  Wstep = (1-load_coeff)/10*Wdesign;

  while(1)
    {
      Eastir_system = Estir_system = Eassume;
      while(1)
      {
        mass(Qstir_ratio);
        heat(Qstir_ratio );
        steam_turb();
              steam_stir();
              Waout = Wstirling + Wsteamturb;
            if(count==1)
            {
              maxFlow = 1.1 * Mair;
                }
              comWork();
              Wfan = .25*Wcom;
              Eastir_system = (Wstirling + Wsteamturb - Wfan)/Qcoal ;

              if(fabs( (Eastir_system-Estir_system)/Eastir_system) <
                        EERROR)
               break;
              Estir_system = Eastir_system;
           }

            if(choice==2)
```

```
                {
              printf("%.4f\t%.4f\n", Wout, Eastir_system),
              fprintf(printfile, "%.4f\t%.4f\n", Wout, Eastir_system);
                }

              Wout = Wout - Wstep;
              if(Wout<=load_coeff*Wdesign)
                break;
      }
}

/*********************************************************************
 * FUNCTION: void steam_stir(void)
 * DESCRIPTION: This function calculates work done by the Stirling

*********************************************************************/
void steam_stir(void)
{
  T3 = Qair/(Mair*Cp) + T2;
  Estirling = calEffStirling(T3);
  Wstirling = Mair * Cp * (T3-810) * Estirling;
}


/*********************************************************************
* FUNCTION: calEffStirling(double T)
* DISCRIPTION: THis function calculate the efficiency of the stirling
*engine which changes with the tempreture.
*********************************************************************/
double  calEffStirling(double T)
{
  double save;

  save = .42 * ( 1.0 - 333.00/T ) /( 1.0 - 333/1373.00);

  return save;
}
```

Part II Computation Results for 20MW Combined Cycle

```
input data filename:  pow.data
I.   The parameter of the air:

Gas constant R (kJ/kg.k)
    2.870e-01
specific heat Cp (kJ/kg.k)
    1.004e+00
specific heat ratio :
    1.400e+00
air temperature before compressing (K):
    2.930e+02
air temperature before boiler (K):
    5.030e+02
air temperature before boiler (K):
    4.330e+02
air pressure before compressing (Pa):
    1.013e+05
air pressure after compressing (Pa):
    1.113e+06
air pressure before gas turbine (Pa):
    1.016e+06
air pressure after gas turbine (Pa):
    1.013e+05
extra air coefficient:
    1.200e+00

 II. coal parameters:

coal's HHV: (kJ/kg)    2.600e+04
coal's carbon:     8.070e+01
coal's hydrogen:     4.500e+00
coal's sulfur:     1.800e+00
coal's oxygen:     2.400e+00
coal's nitrogen:     1.100e+00
coal's water:     3.300e+00

compressor parameters:
Stage of the compressor    4.000e+00
compressing ratio    1.800e+00

 IX gas turbine parameter:

gas turbine efficiency:    9.000e-01

 X. steam turbine parameter:
steam turbine efficiency:    6.200e-01

 XI. steam_water  parameter:
Condensed water enthalpy:    1.000e+02
water enthalpy before getting into the boiler:    8.604e+02
```

vapor enthalpy before getting into the turbine: 3.394e+03

XII. system parameter:
system efficiency(Assumed): 3.500e-01
system designed output:(kw) 2.000e+03
initial assumed heat ratio of the gast_steam system(Qair: Qsystem): 4.000e-01
initial assumed heat ratio of the stir_steam system(Qair: Qsystem): 4.000e-01

For the system design output of 20000.00 KW
For the gas turbine combined cycle system:

Wout	Easystem	Ecom	Egasturb
20000.0000	0.5178	0.870955	0.901777
19000.0000	0.5165	0.860955	0.900976
18000.0000	0.5153	0.853242	0.899776
17000.0000	0.5144	0.847884	0.898232
16000.0000	0.5137	0.843861	0.896345
15000.0000	0.5129	0.840131	0.894117
14000.0000	0.5121	0.835651	0.891557
13000.0000	0.5109	0.829405	0.888673
12000.0000	0.5093	0.820424	0.885480
11000.0000	0.5071	0.807798	0.881989

For the stirling engine combined cycle system:

Wout	Eastir_system
20000.0000	0.5263
19000.0000	0.5259
18000.0000	0.5256
17000.0000	0.5254
16000.0000	0.5252
15000.0000	0.5251
14000.0000	0.5249
13000.0000	0.5247
12000.0000	0.5244
11000.0000	0.5239

For the system design output of 20000.00

ratio	Easystem	Eastir_system	T3
0.3000	0.4902	0.5378	1159.8598
0.3100	0.4930	0.5369	1181.7551
0.3200	0.4957	0.5360	1203.6505
0.3300	0.4985	0.5352	1225.5458
0.3400	0.5013	0.5344	1247.4411
0.3500	0.5040	0.5336	1269.3365
0.3600	0.5068	0.5329	1291.2318

0.3700	0.5095	0.5322	1313.1271
0.3800	0.5123	0.5315	1335.0224
0.3900	0.5151	0.5288	1356.9178
0.4000	0.5178	0.5261	1378.8131
0.4100	0.5206	0.5235	1400.7084
0.4200	0.5234	0.5208	1422.6037
0.4300	0.5261	0.5182	1444.4991
0.4400	0.5289	0.5157	1466.3944
0.4500	0.5317	0.5131	1488.2897
0.4600	0.5344	0.5106	1510.1851
0.4700	0.5372	0.5080	1532.0804
0.4800	0.5399	0.5055	1553.9757
0.4900	0.5427	0.5031	1575.8710
0.5000	0.5455	0.5000	