

A Study of thermodynamic Cycle and System Configurations of Hydrogen Combustion Turbines

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ABSTRACT

A hydrogen combustion turbine is powered by steam generated from the internal combustion of hydrogen as a fuel mixed with stoichiometric oxygen. As it is possible to use a closed cycle system, benefits in cycle efficiency and a reduction of environmental pollution effects.

Three different closed hydrogen combustion turbine cycles, are evaluated, These are the Bottoming reheat cycle (A), the Topping extraction cycle (B) designed by H. Jericha, et al. [1], and the Rankine cycle (C).

Calculations have been carried out to investigate the best cycle. This investigation consists of the comparison of thermodynamic efficiency, first stage turbine vane height of the high temperature, high pressure turbine, and maximum operating temperature of the heat exchangers. In these investigations, the component efficiencies are assumed to be the values which are expected to be achieved in the near future. As a result, the thermal efficiency of cycles (A) and (B) is the same value of 61.5%. That of cycle (C), which has the feed water heating with optimized pressure ratio of the intermediate turbine, is 58.8%. Cycle (B) has the largest first stage turbine vane height of the high temperature high pressure turbine. The larger vane height has an advantage from the point of view of both the manufacturing of the complex cooling passage inside the vane and the turbine aerodynamic efficiency. The maximum operating temperature of the heat exchanger of cycles (A) and (B) is 870°C. While that of cycle (C) is more than 1000°C where some problems are anticipated in the feasibility of this heat exchanger.

This investigation shows that the Topping extraction cycle (B) is considered to be the best cycle from the point of view of both the thermal efficiency and the feasibility of manufacturing.

1 INTRODUCTION

“World Energy Network”, a part of “New Sunshine Project” is a Japanese national project aimed at solving energy and environmental problems in the world. “World Energy Network”, which began in 1993 and is expected to last for 30 years, has the aim of constructing a clean energy network in the world, firstly by producing Hydrogen by electric dissolution of water using hydro-electric-power, solar energy, geothermal energy, wind

energy, and so on, and then transporting, storing and generating electricity using this produced Hydrogen. In this paper we present some evaluations of "Hydrogen Combustion Turbine Cycle and System Configurations", which is a sub-task of "World Energy Network".

Applying hydrogen as a fuel to a conventional air-breathing gas turbine cycle (Brayton Cycle), would not generate CO₂. But Nox would be generated because N₂ is present in the air. On the other hand, the hydrogen combustion turbine cycles studied in this paper are closed cycles, where Nox is not generated because air does not mix with the internal fluid. Thus the possibility of both oxidation and corrosion in the high temperature working fluid is small, and it is comparatively easier to achieve high thermal efficiencies.

Cycle calculations have been carried out to investigate the best cycle. The thermal efficiency, the first stage turbine vane height of the high temperature high pressure turbine, and the maximum operating temperature of heat exchangers are compared respectively. In these investigations, the component efficiencies are assumed to be the values which are expected to be achieved in the near future. The operating pressure and temperature of each component (such as turbines, compressors, and heat exchangers) are assumed to be almost the same values in the three different cycles. As a result, the thermal efficiency of cycle (A) and (B) has the same value of 61.5%. That of cycle (C), which has the feed water heating with optimized pressure ratio of the intermediate turbine, has a value of 58.8%. Cycle (B) has the largest turbine vane height of the high temperature high pressure turbine. The larger vane height has the advantage from the point of view of both the manufacturing of the complex cooling passage inside the vane and the turbine aerodynamic efficiency. The maximum operating temperature of the heat exchanger of cycles (A) and (B) is 870°C. While, that of cycle (C) is more than 1000°C, where some problems are anticipated in the feasibility of this heat exchanger.

This investigation shows that the Topping extraction cycle (B) is considered to be the best cycle from the view point of both the thermal efficiency and the feasibility of manufacturing.

2 CYCLE CONFIGURATIONS

The bottoming reheat cycle (A) shown in Figure 1 is a kind of gas turbine combined cycle whose working fluid is steam. The topping cycle is a closed Brayton cycle comprising of compressor 1, turbine 3, heat exchanger 4,5, and combustor 2. Steam is extracted from between heat exchanger 4 and 5, is expanded by turbine 6, and pumped out. The bottoming cycle is a closed reheat Rankine cycle composed formed by heat exchanger 4,5,17, turbine 9,11,16, feed water heater 14,15, and combustor 17.

The topping extraction cycle (B) [1] shown in Figure 2 is also a kind of gas turbine combined cycle like cycle (A). The difference between these cycles' system configuration is mainly that the topping and the bottoming are separated in cycle (A), while the topping and the bottoming are connected in cycle (B). The topping cycle of cycle (B) is almost the same configuration as cycle (A). The bottoming cycle of cycle (B) is a Rankine cycle. After steam extracted between heat exchanger 4 and 5 is expanded by turbine 6 and the same flow rates as the hydrogen fuel and stoichiometric oxygen are discharged, the rest of the steam is pumped up, recuperated by heat exchanger 4 and 5, expanded by turbine 8 and mixed with the topping steam in the outlet of compressor 1.

The Rankine cycle (C) shown in Figure 3 is a kind of recuperated and reheat Rankine cycles. The difference between this cycle and the other cycles is that there is no compressor in cycle (C). Rankine cycle (C) is equivalent to a high temperature cycle of a conventional industrial steam cycle.

3 ASSUMPTIONS & CONDITIONS

In investigating the best cycle, component efficiencies are assumed to the values which are expected in the near future. They are tabulated in Table 1. The total output of all cycles is a constant 500MW. In this examination, the operating pressures and temperatures of each component (such as turbines, compressors, and heat exchangers) are assumed to be the same values in the three different cycles and are shown in Table 2. Because of these assumptions, the thermal differences of cycles can be clarified. Topping and Bottoming combustor outlet temperature are assumed 1700°C and 1500°C respectively because the firing temperature of a recent high efficient industrial gas turbine has already been being realized to 1500°C. Cooling steam flow rate ratio is 0%, and 15%. The combustor outlet pressure is assumed to be 47.5bar(10^5 Pa). As cycle (C) has two combustors in the same route, the outlet pressure of the upstream one is set to 47.5bar(10^5 Pa), and that of the downstream one is set to 13bar(10^5 Pa). The outlet pressure of downstream 13bar(10^5 Pa) is confirmed to be the optimized pressure by parametric studies. The high pressure turbine inlet temperature, which is turbine 9 in cycle (A), turbine 6 in cycle (B), turbine 9 in cycle (C), is about 850°C, and the high pressure turbine inlet pressure is assumed to be 194bar(10^5 Pa). The outlet pressure downstream of the condensers is fixed at 0.05bar(10^5 Pa). The compressor outlet pressure in cycle (A) and (B) is set to 50bar(10^5 Pa). As the components' assumptions and conditions above are almost same, the manufacturing realization of the components (turbines, heat exchangers, compressors, combustores and so on) would be nearly equivalent.

4 CYCLE PERFORMANCES

Cycle calculations are carried out to estimate the cycle performances of the three cycles. The typical results of cycle calculations, assuming that combustor outlet temperature is 1700°C and cooling steam flow rate ratio is 0% are shown in the Figure 4. The thermal efficiencies of both cycle (A) and (B) are the same value of 61.5%(HHV). Cycle (C) has a value of 58.8%(HHV), which is a relative change of 4.4% lower than other cycles. This is because the downstream combustor outlet pressures of cycle (C) is a lower value than the other cycles, which is a reason why it is not high pressure in the high temperature point. The results of other cases are shown in Table 3. In all cases it seems that the thermal efficiencies of both cycle (A) and (B) are almost same values and the one of cycle (C) is a lower value than other cycles.

5 FEASIBILITY OF COMPONENTS

In order to examine the feasibility of each of the three cycles, the first stage turbine vane height of the high temperature high pressure turbine and the maximum operating temperature of the heat exchangers are compared. The results are summarized in Table 4, Figure 5 and Figure 6. It is shown that topping extraction cycle (B) has the largest first stage turbine vane height of 77mm which the advantage from the point of view of both manufacturing the complex cooling passage inside the vane and the turbine

aerodynamic efficiency. It is especially important for the aerodynamic efficiency of the high temperature turbines in each cycle to be the higher thermal efficiencies, because the high temperature turbine generates a bigger power compared with other turbines, and therefore has a larger weighting on the overall thermal efficiency. When new materials such as C/C composite which do not need the vane cooling are introduced, smaller vane heights may be preferable from a manufacturing view point. In general, metal blades are adopted in the high temperature turbine, which has complicated cooling passages, and so larger first stage turbine vane heights should be used for such high temperature turbines.

The maximum operating temperature of the heat exchangers is compared, and the results are summarized in Table 4 and Figure 6. The temperature of cycle (A) and cycle (B) is 870°C. While that of cycle (C) is 1367°C in the heat exchanger 2, which is 500°C higher than other cycles. Some problems are anticipated in the feasibility of the heat exchanger in cycle (C).

Therefore, the study above shows that the Topping extraction cycle (B) is the best cycle from the view point of the feasibility of components.

6 SELECTION OF THE BEST CYCLE

The results of the investigation of cycle performance and the feasibility of components, which are mentioned above is shown in Table 5. The Topping Extraction Cycle (B) is considered to be the best cycle from the view point of the thermal efficiency and the feasibility of manufacturing. Bottoming Reheat Cycle (A) is slightly inferior to cycle (B) because thermal efficiency and the feasibility of components are almost the same as cycle (B), however cycle (A) has more number of components than cycle (B), therefore initial costs are judged to be more expensive. Rankine Cycle (C) falls behind other cycles in a few aspects. Particularly, the maximum operating temperature of heat exchangers of cycle (C) is really high at 1367°C which is about 500°C higher than other cycles. As cycle (C) has two combustors in the same route, the outlet pressure of the downstream combustor has to be supposed to a lower value than the outlet pressure of the upstream one. This seems to be a main reason why the thermal efficiency of cycle (C) is the lowest.

7 CONCLUSION

We carried out cycle calculations to compare the thermal efficiencies and the feasibility of three different closed hydrogen combustion turbine cycles. From the results presented here, we have some conclusions below.

1 The thermal efficiencies both Bottoming Reheat cycle (A) and Topping Extraction Cycle (B) are the same value 61.5%(HHV), the one of Rankine cycle (C) is the value of 58.8%(HHV), which is a relative change of 4.4% lower than other cycles.

2 Topping extraction cycle (B) has the largest turbine first stage turbine vane height of 77mm and has the advantage from the point of view of both the manufacturing of the complex cooling passage inside the vane and the turbine aerodynamic efficiency. It is especially important for the aerodynamic efficiency of the high temperature turbines in each cycle to achieve higher overall thermal efficiency.

3 The maximum operating temperature of heat exchangers of cycle (A) and cycle (B) is 870°C. While, that of cycle (C) is 1367°C. Some problems are anticipated in the feasibility of the heat exchanger in cycle (C).

4 The Topping Extraction Cycle (B) is considered to be the best cycle from the view point of the thermal efficiency and the feasibility of manufacturing.

We succeeded in getting some preliminary useful results in selecting the best cycle for Hydrogen gas turbine in this research, but this is an interim report for the World Energy Network, and this study is continuing even at present, and better cycles are being designed by modifying cycle (B). Details of these results will be presented in future papers.

8 ACKNOWLEDGMENT

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References

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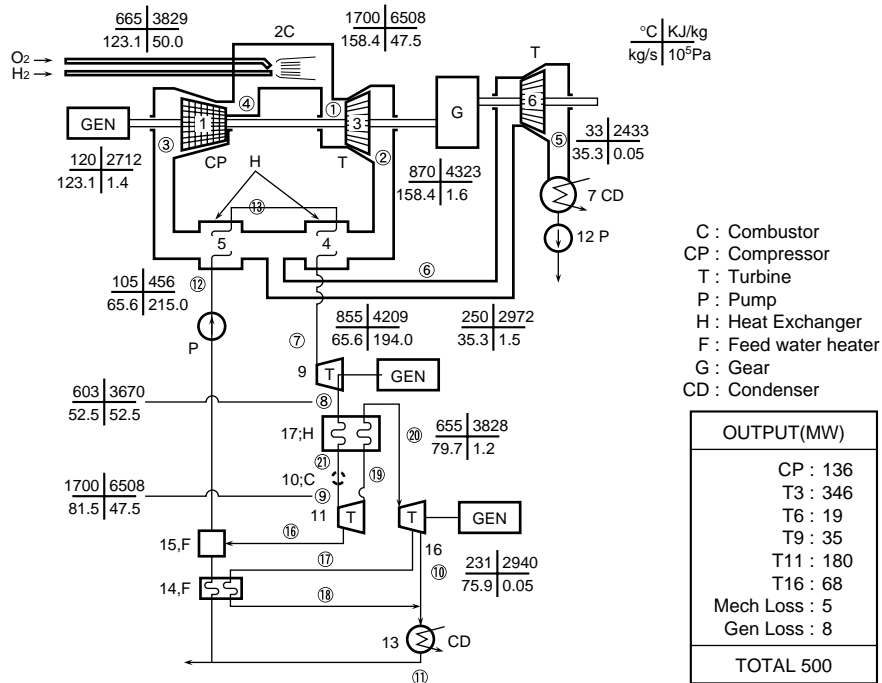


Fig1. Bottoming Reheat Cycle (A)

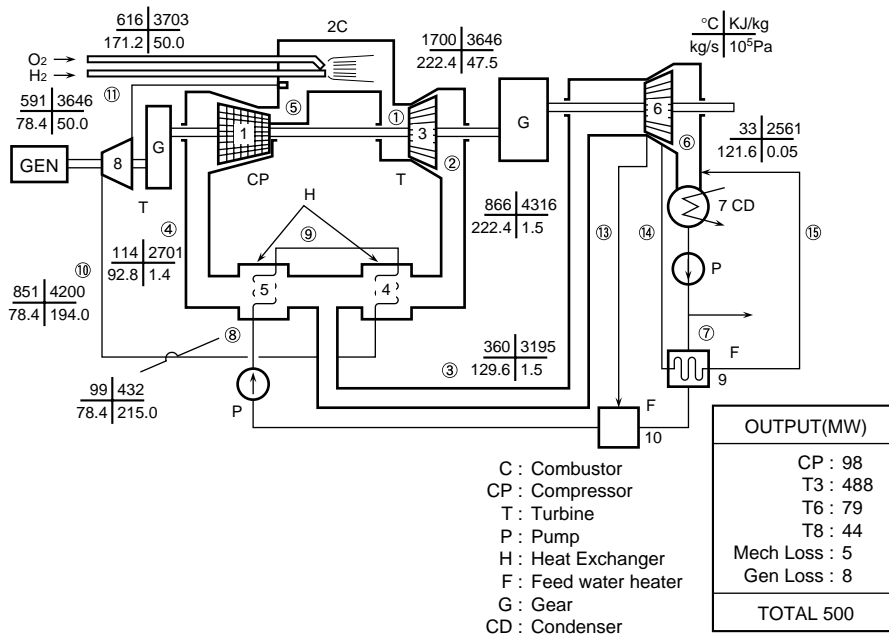


Fig2. Topping Extraction Cycle(B)

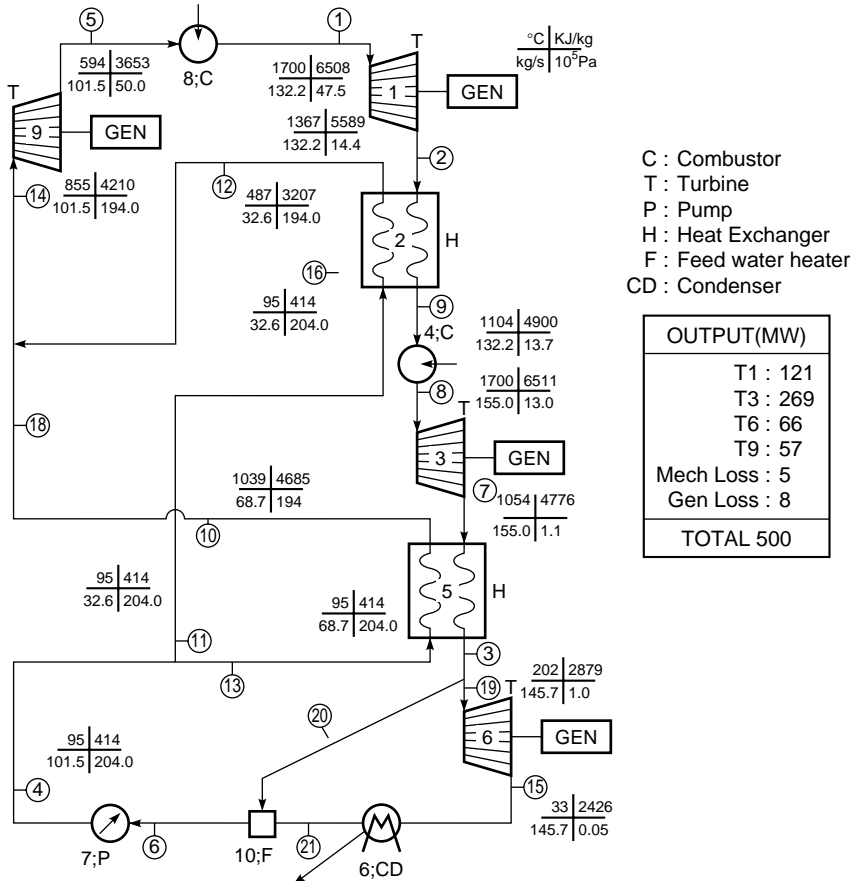


Fig3. Rankine Cycle (C)

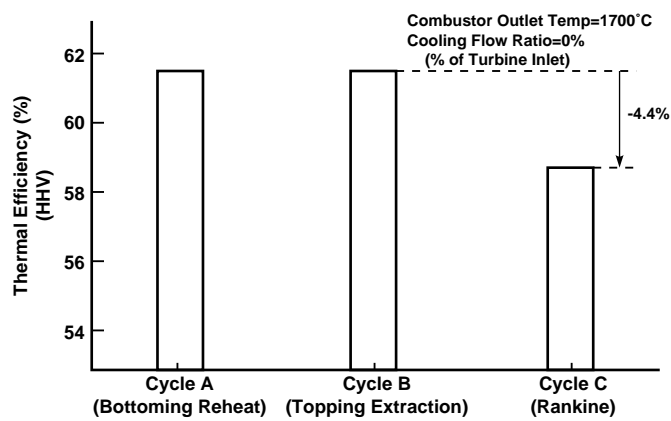


Fig4. Comparison of Thermal Efficiencies

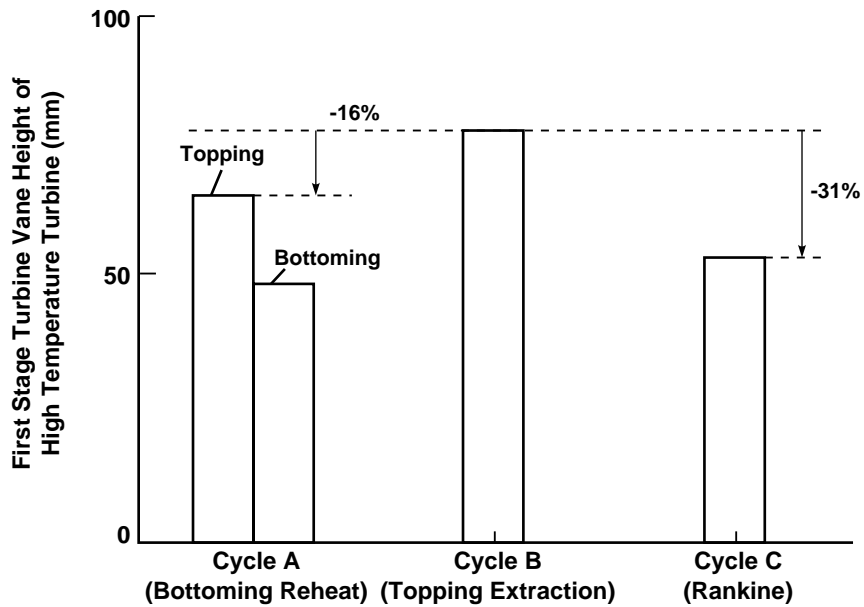


Fig5.Comparison of First Stage Turbine Vane Height of High Temperature Turbines

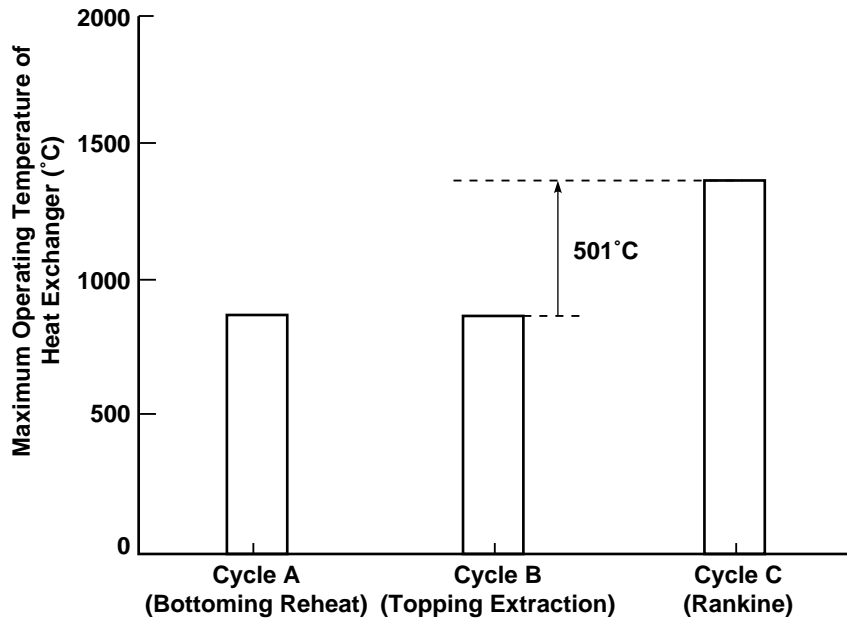


Fig6.Comparison of Maximum Operating Temperature of Heat Exchanger

Table 1. Componets Efficiency

Compressor Efficiency	0.89
Turbine Efficiency	0.93
Combustion Efficiency	1.0
Combustor Pressure Loss	5% of inlet pressure
Heat Exchanger Pressure Loss	5% of inlet pressure
Pump Loading	0
Mechanical Efficiency	0.99
Generater Efficiency	0.985

Table 2. Conditions of Cycle Calculations

Cycle	Bottoming Reheat Cycle(A)	Topping Extraction Cycle(B)	Rankine Cycle (C)
Topping Pressure Ratio	35.7	35.7	–
Topping Combustor Outlet Pressure (10 ⁵ Pa)	47.5	47.5	47.5 (Up Stream)
Topping Combustor Outlet Temperature (°C)	1700,1500	1700,1500	1700,1500 (Up Stream)
Bottoming Combustor Outlet Pressure (10 ⁵ Pa)	47.5	–	13 (Down Stream)
Bottoming Combustor Outlet Temperature (°C)	1700,1500	–	1700,1500 (Down Stream)
High PressureTurbine Inlet Pressure (10 ⁵ Pa)	194 (Turbine 9)	194 (Turbine 8)	194 (Turbine 9)
High PressureTurbine Inlet Temperature (°C)	855 (Turbine 9)	851 (Turbine 8)	855 (Turbine 9)
Cooling Steam Flow Rate Ratio (% of Tubine Inlet)	0,15		
TOTAL OUTPUT (MW)	500		

Table3. Cycle Calculation Results

Cycle	Bottoming reheat Cycle (A)				Topping Extraction Cycle (B)				Rankine Cycle (C)			
	1700		1500		1700		1500		1700		1500	
Combustor Outlet Temperature (°C)	1700		1500		1700		1500		1700		1500	
Cooling Steam Flow Rate Ratio (%of Turbine Inlet)	0	15	0	15	0	15	0	15	0	15	0	15
Thermal Efficiency (HHV) (%)	61.5	60.2	60.4	58.7	61.5	60.5	60.1	59.1	58.8	55.5	56.4	54.8

Table4. Feasibility of Components

Cycle	Bottoming reheat Cycle (A)	Topping Extraction Cycle (B)	Rankine Cycle (C)
Combustor Outlet Temperature (°C)	1700		
Cooling Steam Flow Rate Ratio (%of Turbine Inlet)	15		
First Stage Vane Height of High Temperature Turbine (mm)	65 (Topping) 48 (Bottoming)	77	53 (Up Stream)
Maximum Operating Temperature of Heat Exchanger (°C)	870 (Heat Exchanger4)	866 (Heat Exchanger4)	1367 (Heat Exchanger4)

Table5. Selection of Best Cycle

Bottoming Reheat Cycle (A)	<ul style="list-style-type: none"> <input type="radio"/> Highest Thermal Efficiency (61.5%) <input type="radio"/> Lowest Maximum Operating Temperature of Heat Exchanger (870°C)
Topping Extraction Cycle (B)	<ul style="list-style-type: none"> <input type="radio"/> Highest Thermal Efficiency (61.5%) <input type="radio"/> Biggest First Stage Turbine Vane Height of High Temperature Turbine. (77mm) <ul style="list-style-type: none"> • In case of using metal for vane material, easy to construct same complicated cooling construction. • Benefits for higher Turbine efficiency <input type="radio"/> Lowest Maximum Operating Temperature of Heat Exchanger (866°C) <input type="radio"/> Few Number of Components (Cheap initial costs)
Rankine Cycle (C)	<ul style="list-style-type: none"> × Lowest Thermal Efficiency (58.8%) × Lowest First Stage Turbine Vane Height of High Temperature Turbine (53mm) × Highest Maximum Operating Temperature of Heat Exchanger (1367°C) <ul style="list-style-type: none"> • Some problems in the feasibility of the Heat Exchanger)