A Study of Advanced Hydrogen/Oxygen Combustion Turbines

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ABSTRACT

Mitsubishi Heavy Industries Ltd. have proposed the advanced hydrogen/oxygen combustion turbine system which is an inter-cooled topping recuperation cycle as part of a Japanese government sponsored program WE-NET ("World Energy Network"). The efficiency of this cycle reaches more than 60% (HHV), not (LHV), with a power capacity of 500MW. This cycle is formed by a compressor, turbines, a combustor and heat exchangers. The combustor burns hydrogen/oxygen to make high temperature (1700 ℃) steam.

As a result of MHI research in 1994 [1], the topping extraction cycle (A) designed by Jericha et al. [2] was found to be the best cycle. MHI also designed the inter-cooled topping recuperation cycle (BI) in 1995 which modifed cycle (A) to be suitable for a high combustion temperature (1700 ℃).

This paper presents the details of the comparison between the above mentioned cycles. The investigation shows that cycle (BI) is considered to be better than cycle (A) from the point of view of both the thermal efficiency and the feasibility of manufacture.

1 INTRODUCTION

WE-NET, a part of "New Sunshine Project", is a Japanese government program aimed at solving energy and environmental problems in the world. WE-NET, which began in 1993 and is expected to last for 30 years, has the aim of constructing a clean energy network throughout the world; firstly by producing Hydrogen by electric dissolution of water using hydro-electric-power, solar energy, geothermal energy, wind energy, etc. and then transporting, storing and generating electricity using this produced Hydrogen. In this paper some evaluation of "Hydrogen/Oxygen Combustion Turbine Cycles", which is a sub-task of "World Energy Network", is presented.

In 1994, three different closed hydrogen combustion turbine cycles, were evaluated [1]. These are the Bottoming Reheat Cycle, the Topping Extraction Cycle (A) designed by Jericha et al. [2] and the Rankine Cycle. Results of this study showed the Topping Extraction Cycle (A) to be the best cycle.

Subsequent to this research, further work was undertaken to design a highly efficient and highly feasible cycle adapted to high combustion temperature (1700 ℃) - the Topping Recuperation Cycle(B). Cycles with an inter-cooler between low pressure and high pressure compressors in cycles (A) and (B) have also been investigated which are the Inter-cooled Topping Extraction Cycle (AI) and the Inter-cooled Topping Recuperation Cycle (BI).

2 CYCLE CONFIGURATIONS

The Inter-cooled Topping Extraction Cycle (AI) shown in Figure 1 is a kind of gas turbine combined cycle whose working fluid is steam. The topping cycle is a Brayton cycle comprising of low pressure compressor 1, high pressure compressor 2, turbine 4 ,heat exchanger 5,6, and combustor 3. The bottoming cycle of cycle (AI) is a Rankine cycle. Steam is extracted from between heat exchanger 5 and 6, is expanded by turbine 6, and condensed by condenser 8. The same flow rates as the hydrogen fuel and stoichiometric oxygen are discharged, the rest of the steam is pumped up by pump 12 and 13. The steam is economized by feed water heaters 10 and 11, recuperated by heat exchanger 5 and 6, expanded by turbine 9 and mixed with the topping steam in the outlet of compressor 2. The water to be inter-cooled is extracted after being condensed in condenser 13, and pumped up in pump 17. It is mixed with the the topping steam in the outlet of compressor 1. The Topping Extraction Cycle (A) is same as non-inter-cooled one of cycle (AI).

The Inter-cooled Topping Recuperation Cycle (BI) shown in Figure 2 is also a kind of gas turbine combined cycle like cycle (AI). Cycle (BI) has added heat exchangers 14 and 15 compared with cycle (AI). The heat exchanger 14 recuperates the heat from the exhaust of turbine 4 to the exit of compressor2.

3 ASSUMPTIONS CONDITIONS

The investigations are carried out preliminarily in the operating conditions in Table 1 and the component efficiencies in Table 2. Table 1 shows the operating pressures and temperatures of each component (such as turbines, compressors, and heat exchangers) which are assumed to be the almost the same values in cycle (A), (AI), (B) and (BI). The values in Table 1 and 2 are expected in the near future. Because of these assumptions, the thermal differences of the cycles can be clarified. The total output of all cycles is a constant 500MW. Combustor outlet temperatures are assumed to be both 1700 ℃ and 1500 ℃, because the firing temperatures of current high efficiency industrial gas turbines have already reached 1500 ℃. Cooling steam flow rate ratio is 0%, and 15%. The outlet pressure downstream of the condensers is fixed at 0.05bar(10^5Pa). As the components' assumptions and conditions above are almost the same, the manufacturing realization of the components (turbines ,heat exchangers, compressors, combustors and so on) should be nearly equivalent.

The paramatric studies based on the above investigations are carried out to assess the characteristics of pressure ratio and the inlet temperature of turbine 4. The pressure ratio should be selected carefully if the operating temperature is decided because a lot of factors are affected by the pressure ratio. The merits and demerits of increasing the pressure ratio are displayed in Table 3. From Table 3, it would be beneficial to achieve a high cycle thermal efficiency at the lowest possible pressure ratio.

4 PERFORMANCES AND FEASIBILITY OF COMPONENT

Cycle calculations for cycle (A), (AI), (B) and (BI) were carried out to estimate the performance of the cycles. The typical results of cycle calculations, assuming that combustor outlet temperature is 1700 ℃ and cooling steam flow rate ratio is 0% are shown in Table 4. Comparison of cycle (A) and (BI) shows the thermal efficiency of cycle (BI) to have a relative change of 4.0% higher than cycle (A). The first vane height of the high pressure, high temperature turbine of cycle (BI) has a relative change of 7% higher than cycle (A). The exhaust temperature of the turbine 4 and the maximum operating temperature of the heat exchanger in each cycle are almost the same at 800 ℃ . The compressor outlet temperature is about 700 ℃ in cycles (A) and (B) and 550 ℃ in cycles (AI) and (BI). Therefore, cycle (BI) is judged to be the best cycle from the point of view of the feasibility of the components. The results of other cases are shown in Table 5.

The results of the parametric studies are shown in Figure 3 and 4. The results show that the pressure ratio for the maximum thermal efficiency is about 130 in cycle (A) and 50 in cycle (BI). The thermal efficiency of cycle (BI) is higher than cycle (A) for a pressure ratio of less than 70. If the pressure ratio becomes higher, a lot of problems shown in Table 3 are anticipated. Cycle (BI), therefore, has an advantage in terms of thermal efficiency at a more feasible pressure ratio. The pressure ratio should ,therefore, be set to 50.

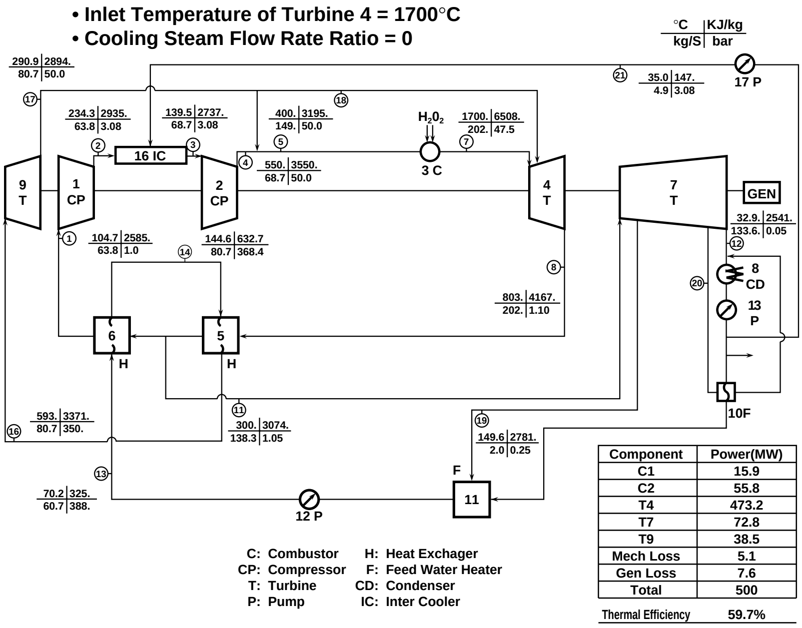
The thermal efficiency of cycle (BI) is higher than cycle(A) when the turbine 4 inlet temperature is more than 1500 °C. Cycle (BI) is, therefore, suitable for the high operational temperature cycle of more than 1500 °C. The features of the Inter-cooled Topping Recuperation Cycle are shown in Table 6.

5 CONCLUSIONS

1) The thermal efficiency of cycle (BI) is 62.2%(HHV), which is a relative change of 4.0% higher than cycle(A).
2) The cycle (BI) has a relative change of 7% larger first stage turbine vane height than cycle (A), and 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 to have high aerodynamic efficiency of the high temperature turbines in each cycle to achieve higher overall thermal efficiency.
3) The pressure ratio for the maximum thermal efficiency is about 50 in cycle (BI), which is lower than that of cycle (A) at 130. Cycle (BI), therefore, has an advantage of high thermal efficiency at a more feasible pressure ratio.
4) Cycle (BI) is suitable for a high operational temperature cycle of more than 1500 °C.
5) The inter-cooled topping recuperation cycle (BI) in Figure 2 is considered to be the best cycle from the view point of the thermal efficiency and the feasibility of manufacturing.

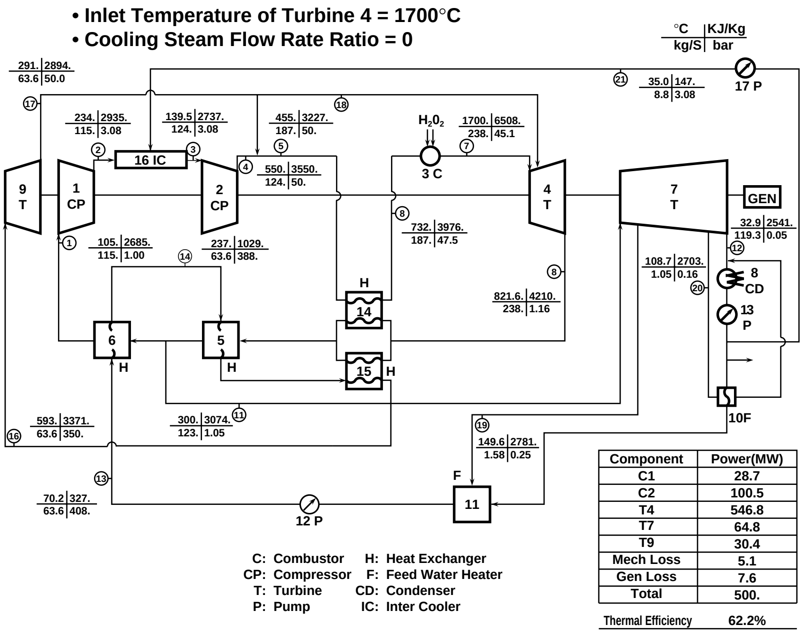
References

[1] H.Sugishita, H.Mori and K.Uematsu "A Study of Thermodynamic Cycle and System Configurations of Hydrogen Combustion Turbines", 11th World Hydrogen Energy Conference, Stuttgart Germany, June 23-26,1996, pp1851-1860.
[2] H. Jericha, R.Ratzesberger, "A Novel Thermal Peak Power Plant", ASME Cogen-Turbo Nice France August 30 - September 1 ,1989.



Component	Power(MW)
C1	15.9
C2	55.8
T4	473.2
T7	72.8
T9	38.5
Mech Loss	5.1
Gen Loss	7.6
Total	500
Thermal Efficiency	59.7%

Fig.1 A Typical Calculation Result of Intercooled Topping Extraction Cycle(AI)



Component	Power(MW)
C1	28.7
C2	100.5
T4	546.8
T7	64.8
T9	30.4
Mech Loss	5.1
Gen Loss	7.6
Total	500.
Thermal Efficiency	62.2%

Fig.2 A Typical Calculation Result of Intercooled Topping Recuperation Cycle (BI)

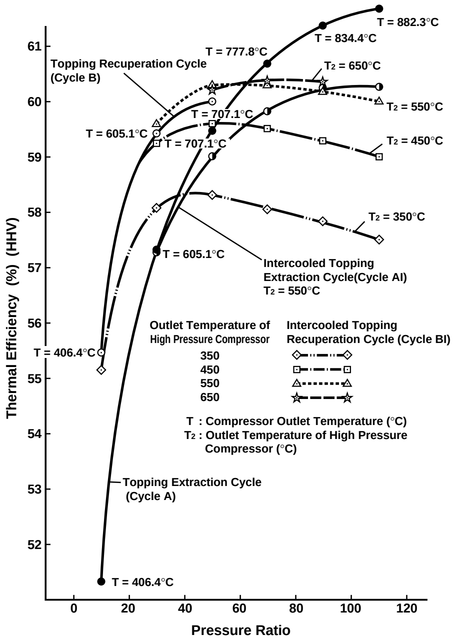


Fig.3 Effects of Topping Pressure Ratio

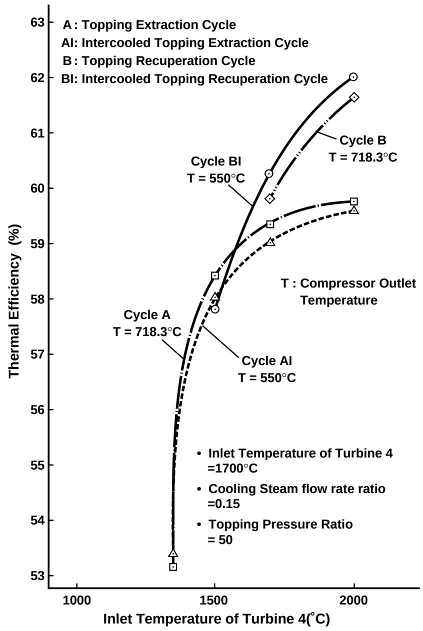


Fig.4 Effects of Inlet Temperature of Turbine 4

Table 1 Operating Conditions

• Compressor Inlet Pressure	1.0 bar
• Compressor Inlet Temperature	+5°C of a Saturated Temperature
• Compressor Outlet Pressure	50.0 bar
• Compressor Outlet Temperature	550°C (With inter-cooled)
• Turbine 4 Inlet Temperature	1700, 1500°C
• Cooling Air Flow Rate Ratio of Turbine 4	0, 0.15
• Turbine 4 Outlet Temperature	Less than 870°C
• Turbine 9 Inlet Temperature	593°C
• Turbine 9 Inlet Pressure	350 bar
• Maximum Operating Temperature of Heat Exchanger	Less than 900°C
• Temperature Efficiency of Heat Exchanger	Less than 0.85
• Total Output	500 MW

Table 2 Components 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 3 Merits and Demerits by increasing Pressure Ratio

Merits
∑ Higher Thermal Efficiency
Demerits
∑ Increase the number of stages of turbines and compressors
∑ Increase the leakage flow rate
∑ Higher compressor exhaust temperature
∑ Thicker casing of compressor outlet and combustor chamber
∑ Higher heat transfer coefficient on the surface of blades
∑ Higher energy to pump up a fuel
∑ Lower height of first stage turbine vanes

Table 4 Comparison of Results

	Thermal Efficiency (%)	Relative Change of Thermal Efficiency	Relative Change of First Stage Turbine Vane Height of Turbine 4	Compressor Outlet Temperature (°C)	Exhaust Temperature of Turbine 4 (°C)	Maximum Operating Temperature of Heat Exchanger (°C)
Cycle A	60.0	1	1	707	803	803
Cycle AI	59.7	0.995	0.98	550	803	803
Cycle B	62.4	1.04	1.09	707	822	822
CycleBI	62.2	1.04	1.07	550	822	822

∑ Inlet Temperature of Turbine 4=1700°C
∑ Cooling Steam flow Rate Ratio=0

Table 5 Results of Cycle Calculations

		Topping Extraction Cycle								Topping Recuperation Cycle							
Cycle		(A)				(AI)				(B)				(BI)			
Pressure ratio		50				50				50				50			
Inlet temperature of turbine 4		1700		1500		1700		1500		1700		1500		1700		1500	
Cooling Flow rateratio		0	0.15	0	0.15	0	0.15	0	0.15	0	0.15	0	0.15	0	0.15	0	0.15
Compressor Outlet Temperature	Low	707.0	707.0	707.0	707.0	234.3	234.3	234.3	234.3	707.0	707.0	707.0	—	234.3	234.3	234.3	234.3
	High					549.8	549.8	549.8	549.8					549.8	549.8	549.8	549.8
Inlet temperature of Turbine 4		803.1	699.4	672.7	578.2	803.1	699.4	672.7	579.8	821.6	716.8	689.6	—	821.6	716.8	689.6	595.7
Flow rate of Turbine 4		209.15	223.35	267.68	289.84	202.11	211.44	252.06	266.42	246.35	241.10	290.65		237.95	236.88	279.83	275.63
Thermal Efficiency (%)		60.0	59.5	59.6	58.2 (T15=538°C)	59.7	59.0	59.1	58.0	62.4	60.0	60.3		62.2	60.3	60.3	57.8 (T15=566°C)

Table 6 Features of Cycles

	Inter-cooled Topping Extraction Cycle (AI)	Inter-cooled Topping Recuperation Cycle (BI)
1. System Configurations	∑ Combined Cycle	←
2. Recuperation from Turbine Outlet to Combustor Inlet	×	○
3. Characteristic of Pressure Ratio	∑ Optimized Pressure Ratio is about 130	∑ Optimized Pressure Ratio is about 50
4. Characteristic of Temperature of Turbine 4 Inlet	∑ Higher Efficiency in less than 1500°C	∑ Higher Efficiency in more than 1500°C
5. Steam Flow Rate of Turbine 4 Inlet	∑ Small	∑ Large (Easier to construct complicated cooling passages)