Performance Analysis of a 3 Pressured Combined Cycle Power Plant

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3압 복합 발전 플랜트 사이클에 대한 성능해석

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초 록

복합발전 사이클은 가스터빈이나 스팀터빈으로부터의 출력을 이용하여 전기를 생산하기 위한 발전기를 구동시키고 배열회수기로부터 나온 증기를 스팀터빈에서 팽창시킴으로서 부가적인 동력을 얻는 장치를 가리킨다. 보통 가스터빈 배기로 부터의 온도는 400~650℃ 정도로서 배열회수기에서 효과적으로 스팀을 생산할 수 있는 수준의 온도이다. 복합 사이클은 일반적으로 상부사이클과 하부사이클로 구분하는데 대부분의 열에너지 공급이 이루어지는 상부사이클을 브레이돈사이클 이라하며 브레이돈사이클에서 소비되는 에너지는 보다 낮은 온도 수준인 하부사이클에서 회수된다. 이러한 복합사이클은 최근 들어 더욱 보편적으로 적용되고 있는데 그 이유는 첫째, 가스터빈이나 스팀터빈이 독자적으로도 충분히 기술적인 검증을 받은 열기관으로서 초기에 비해 개발비가 저렴해졌다는 데 있고, 둘째, 작동유체인 공기가 1000℃ 이상에서도 별다른 문제 없이 적용될 수 있는 안전한 유체이고 비용이 전혀 들지 않는다는 점이다. 그 뿐아니라 스팀터빈에 사용되는 물도 중저온에서 매우 저가로 공급할 수 있고 쉽게 공급이 가능하다는 이점으로 하부사이클에의 적용이 매우 양호하다는 점이다. 최근 소재기술의 개발에 따른 터빈입구온도의 향상은 이러한 복합발전 사이클의 기술적, 경제적 이점을 더욱 강화시켜주고 있다. 본 연구에서는 3압에 의한 복합사이클에 대한 성능해석을 통하여 상부사이클이 전체 복합발전 성능에 미치는 영향을 조사하였으며 그 결과를 서인천 복합발전 인수 성능시험결과와 비교하였다. 본 연구결과는 현재 개념설계가 이루어지고 있는 장차 150~200 MW 수준의 산업용 가스터빈 개발에 중요한 방향제시를 할 수 있을 것으로 판단된다.

ABSTRACT

Combined cycle power plant is a system where a gas turbine or a steam turbine is used to produce shaft power to drive a generator for producing electrical power and the steam from the HRSG is expanded in a steam turbine for additional shaft power. The temperature of the

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exhaust gases from a gas turbine ranges from 400℃~650℃, and can be used effectively in a heat recovery steam generator to produce steam. Combined cycle can be classed as a topping and bottoming cycle. The first cycle, to which most of the heat is supplied, is a Brayton gas turbine cycle. The wasted heat it produces is then utilized in a second process which operates at a lower temperature level is a steam turbine cycle. The combined gas and steam turbine power plant have been widely accepted because, first, each separate system has already proven themselves in power plants as an independent cycle, therefore, the development costs are low. Secondly, using the air as a working medium, the operation is relatively non- problematic and inexpensive and can be used in gas turbines at an elevated temperature level over 1000oC. The steam process uses water, which is likewise inexpensive and widely available, but better suited for the medium and low temperature ranges. It therefore, is quite reasonable to use the steam process for the bottoming cycle. Recently gas turbine attained inlet temperature that make it possible to design a highly efficient combined cycle. In the present study, performance analysis of a 3 pressured combined cycle power plant is carried out to investigate the influence of topping cycle to combined cycle performance. Present calculation is compared with acceptance performance test data from SeoInchon combined cycle power plant. Present results is expected to shed some light to design and manufacture 150~200MW class heavy duty gas turbine whose conceptual design is already being undertaken.

NORMENCLATURE

AMB	Ambient
COMP	Compressor
CC	Combined Cycle
D/A	Deaerator
GT	Gas Turbine
EXT	Exhaust
HHV	Higher Heating Value
HP	High Pressure
HPB	High Pressure Boiler
HPE	High Pressure Economizer
HPS	High Pressure Superheater
HRSG	Heat Recovery Steam Generator
IP	Intermediate Pressure
IPB	Intermediate Pressure Boiler
IPS	Intermediate Pressure Superheater
LHV	Lower Heating Value
LP	Low Pressure
LPB	Low Pressure Boiler
LPS	Low Pressure Superheater
LTE	Low Temperature Economizer
Psfc	Power Specific Fuel Consumption
Q	Heat Transfer Rate
RH	Reheater
SF	Suplementary Firing
ST	Steam Turbine
TCA	Turbine Cooling air
TET	Turbine Exhaust Temperature
TIT	Turbine Inlet Temperature
η	Efficiency

INTRODUCTION

The general definition of the thermal efficiency of a combined cycle plant is, as shown by Kehlhofer (1991),

$$\eta_{CC} = \frac{W_{GT} + W_{ST}}{\dot{Q}_{GT} + \dot{Q}_{SF}} \tag{1}$$

If there is no supplementary firing in the waste heat boiler, then this formula simplifies into,

$$\eta_{CC} = \frac{W_{GT} + W_{ST}}{\dot{Q}_{GT}} \tag{2}$$

Since, the efficiency for the gas turbine process is defined as,

$$\eta_{GT} = \frac{W_{GT}}{\dot{Q}_{GT}} \tag{3}$$

and for the steam turbine process,

$$\eta_{ST} = \frac{W_{ST}}{\dot{Q}_{SF} + \dot{Q}_{EXT}} \tag{4}$$

and,

$$\dot{Q}_{EXT} = \dot{Q}_{GT} \cdot (1 - \eta_{GT}) \tag{5}$$

combining eq. (4) and eq.(5) yields,

$$\eta_{ST} = \frac{W_{ST}}{\dot{Q}_{SF} + \dot{Q}_{ST} \cdot (1 - \eta_{GT})} \tag{6}$$

For a combined cycle, submitting eq.(3) and eq.(6) into eq.(1), one obtains

$$\eta_{CC} = \frac{\eta_{GT}\,\dot{Q}_{GT} + \eta_{ST}(\,\dot{Q}_{SF} + \dot{Q}_{GT}[1 - \eta_{GT}])}{\dot{Q}_{GT} + \dot{Q}_{SF}} \tag{7}$$

without additional firing eq.(7) becomes as,

$$\begin{aligned}\eta_{CC} &= \frac{\eta_{GT}\,\dot{Q}_{GT} + \eta_{ST}\,\dot{Q}_{GT}[1 - \eta_{GT}]}{\dot{Q}_{GT}} \\ &= \eta_{GT} + \eta_{ST}(1 - \eta_{GT})\end{aligned} \tag{8}$$

Differentiation of eq.(8) with respect to GT shows the effect of gas turbine efficiency on overall efficiency :

$$\frac{\partial \eta_{CC}}{\partial \eta_{GT}} = 1 + \frac{\partial \eta_{ST}}{\partial \eta_{GT}}(1 - \eta_{GT}) - \eta_{ST} \tag{9}$$

Increasing the gas turbine efficiency improves the overall efficiency only if :

$$\frac{\partial \eta_{CC}}{\partial \eta_{GT}} > 0 \tag{10}$$

And therefore, from eq.(9) one obtains,

$$\frac{\partial \eta_{ST}}{\partial \eta_{GT}} < \frac{1 - \eta_{ST}}{1 - \eta_{GT}} \tag{11}$$

This indicates that improving the gas turbine efficiency is helpful only if steam turbine efficiency drops due to gas turbine efficiency increase must not be too large. In a more plain words, the gas turbine with the highest efficiency does not necessarily produce the best overall efficiency of the combined cycle because higher efficiency of gas turbine at constant inlet temperature has higher pressure ratio in the turbine than moderate efficiency gas turbine. However, the efficiency of the combined cycle plant with a moderate gas turbine pressure is significantly better because the steam turbine operates far more efficiently with high exhaust temperature and produces a greater output. The main problem in laying out a combined cycle plant thus is making optimum use of the exhaust heat from the gas turbine in the waste heat boiler. Heat utilization can never be perfect, however, because evaporation process takes place under constant temperature. Figure 1 shows thermodynamic energy balance for combined cycle.

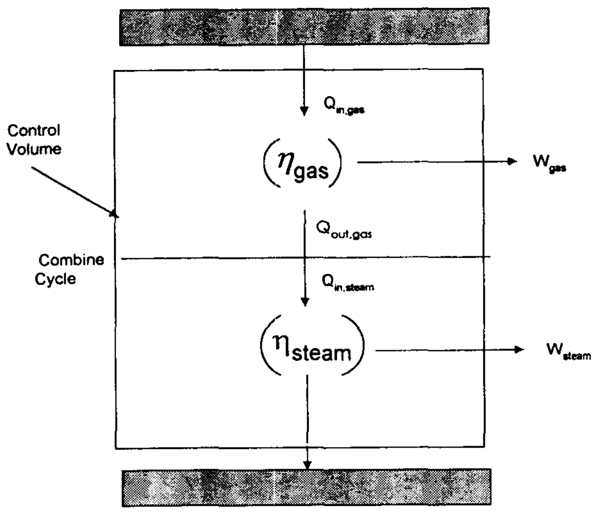

Fig. 1 Thermodynamic energy balance in combined cycle

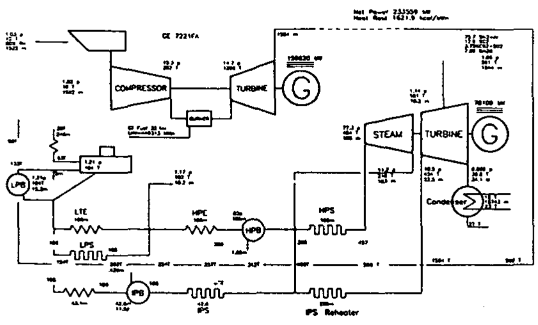

Fig. 2. System with 3 pressured HRSG for Steam Injection into the Gas turbine

3. PRESSURED POWER PLANT SYSTEM

Figure 2 shows the outline of the present power system studied. Basically, the system consists of a gas turbine to generate power and high temperature exhaust gas, heat recovery boiler, steam turbine and condenser. In the Fig. 2, heat recovery boiler houses superheater and economizer, boiler of high, intermediate and low pressure, and deaerator. Gas turbine data used for the present analysis are shown in Table 1. Gas Turbine inlet flow condition of Table 1, Tin = 10.16oC, Pin = 1.0332ata, are the inlet flow condition at the time of acceptance test, and are used here as reference condition for fair

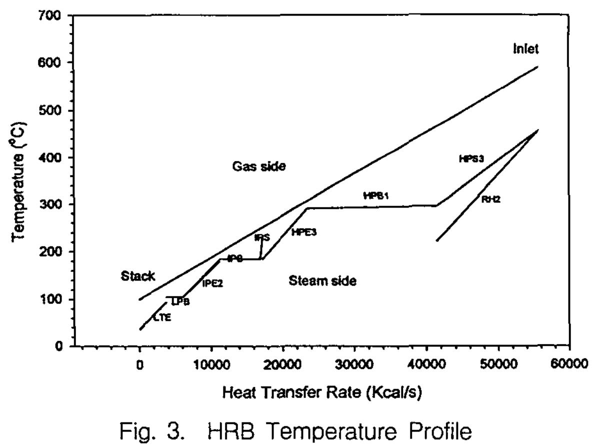

Fig. 3. HRB Temperature Profile

comparison. A combined cycle with multi pressure steam reduces the temperature of the gas leaving the heat recovery boiler and hence results in increased efficiency of the plant. In Fig. 2 exhaust gas from the gas turbine enters heat recovery steam generator(HRSG) leaving it to the stack. Low temperature economizer(LTE) close to the stack heats the water before reaching the deaerator using the flue gas energy. It then is followed by low pressure boiler and superheater thus producing steam at three pressure levels. Figure 2 also shows the configuration that IPS behind the HPE and IPS reheater in parallel with HPS and recovers more heat and additional extra steam from LPB is injected into low pressure steam turbine thus increasing power. High pressure superheated steam enters the steam turbine first stage. The D/A is heated from both LPB steam from an LPT steam bleed. The amount of the steam flow is determined by enthalpy consideration in the steam expansion path at the bleed. Figure 3 is the HRB temperature profile of the present calculation. In the figure one can see that low pressure steam boils at temperature below that of intermediate pressure steam and intermediate pressure steam boils at lower temperature than high pressure steam, respectively hence there exist three pinch points between the gas line and the saturated steam lines. In a 3 pressure system, pinch point of the high pressure boiler is less critical than one ot two pressure system as the heat that is not utilized can be recovered in the lower pressure boiler. Pinch point in the present calculation occurs at intermediate pressure heater at 18.7℃. Generally, it is known that a triple pressure arrangement attains a slightly higher efficiency at full load, but it costs more.

Table 2. Shows the LNG fuel data used for the present calculation.

Table 1 Gas turbine data.

Variables	value	unit
Inlet pressure	1.0332	ata
Inlet temperature	10.16	℃
Pressure ratio	14.8	
TIT	1288	℃
TCA	11.15	%
TET	587	℃
M_{air}	421.7	kg/sec
Shafts	1	
Heat rate	2372	KJ/KWh
Rotating speed	3600	RPM
%LHV	36.9	
Power	161650.0	KW

Table 2. LNG Fuel Data

HigherHeat Value(HHV)	Kcal/kg	13,073(±10%)
Lower heat Value(LHV)	Kcal/kg	11,786(±10%)
HHV/LHV		1.109
CH_4	mole %	89.006
C_2H_6	mole %	8.735
C_3H_8	mole %	1.665
$I\text{-}C_4H_{10}$	mole %	0.266
$N\text{-}C_4H_{10}$	mole %	0.32
N_2	mole %	0.008

INFLUENCE OF AMBIENT CONDITIONS ON OUTPUT AND EFFICIENCY

The design of the combined cycle plant is influenced mainly by the air temperature, air pressure, and cooling water temperature. The inlet air temperature is known to have a strong influence on the power output and efficiency of an open cycle gas turbine with following two reasons. First, the density of air will decrease with increased inlet gas temperature, and thereby reduces the air mass flow drawn in. Secondly, the compressor will consume more power in proportion to the intake temperature, though the turbine doesnt produce a corresponding increase in power output. Since the absorption capacity of the turbine remains constant the pressure before the turbine is reduced with the mass flow decrease due to the air temperatures increase. Figure 4 shows the effect of ambient temperature on gas turbine and combined cycle efficiencies at electrical generator terminal. All gas turbine efficiencies are of percent of HHV. In the figure, one can see that gas turbine efficiency decreases rapidly with ambient temperature increase whereas combined cycle efficiency show slow increase and decrease with maximum at around Tin = 15℃. Steam turbine efficiency turned out to grow with constant rate. This indicates as mentioned earlier that maximum gas turbine efficiency does not necessarily guarantee the best overall combined cycle efficiency. Figure 5 shows the influence of ambient temperature on power output, mass flow, and heat rate of the gas turbine. Values of power output, specific power and mass flow rate, normalized by those of base load temperature, 11.16℃,

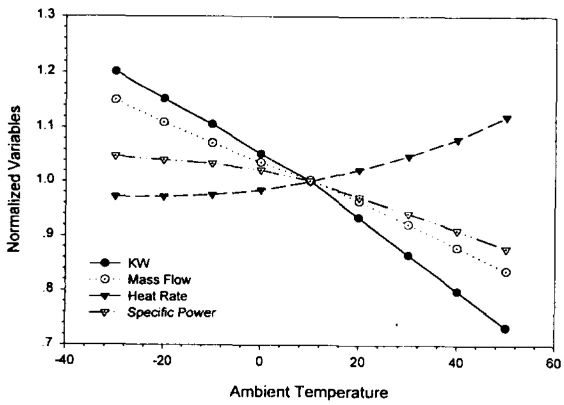

Fig. 4. Effect of Ambient Temperature on Efficiency

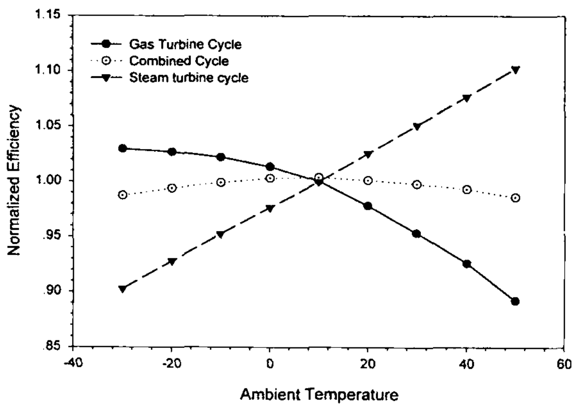

Fig. 5. Effect of ambient temperature on power output, mass flow and heat rate.

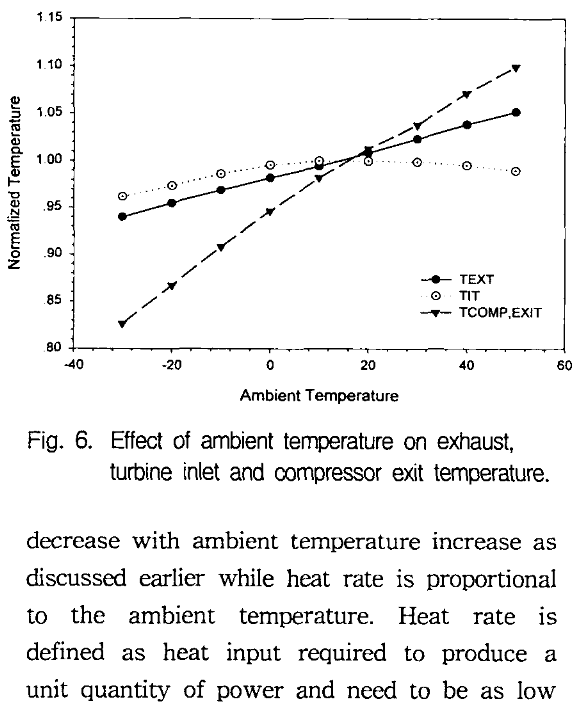

Fig. 6. Effect of ambient temperature on exhaust, turbine inlet and compressor exit temperature.

decrease with ambient temperature increase as discussed earlier while heat rate is proportional to the ambient temperature. Heat rate is defined as heat input required to produce a unit quantity of power and need to be as low as ossible for a given system. Turbine inlet and exhaust temperature increase with compressor inlet temperature as shown in Fig. 6. It is generally known that if the gas turbine inlet temperature remains constant then a gas turbine with a higher exhaust temperature will results in better combined cycle efficiency. If the gas turbine exhaust temperature is lowered, both the thermodynamic quality of the steam process and the energy utilization rate of the waste heat boiler will deteriorate.

INFLUENCE OF STEAM INJECTION ON GAS TURBINE PERFORMANCE

Environmental protection laws among many industrial countries such as USA, Japan, and most European countries require that the Nox levels in the exhaust be very low. One way to reduce the Nox levels in the exhaust gas is to lower the flame temperature in the combustor. The oxides of nitrogen are derived from nitrogen in the combustion air and any nitrogen in the fuel itself. And the formation of Nox during combustion increases as the speed of reaction producing Nox is noticeably high at high temperature. Figure 7 shows turbine inlet temperature as function of steam injection rate. Turbine inlet temperature decreases with steam injection and ambient temperature decrease. Steam injection will preserve the material temperature limitations and system longevity will be secured. It is interesting to note in the figure that TIT reaches maximum at base load ambient temperature then decreases with further increasing the ambient temperature. This is considered because of inherent TIT control in the program. Figure 8 shows the relation between gas turbine power and steam injection mass flow. Power increases with steam injection for all compressor inlet temperatures studied. Gas turbine exhaust temperature always decreases for the ambient temperature decrease and increase with steam injection rate as shown in Fig. 9. Steam injection pressure in this calculation was 45.7ata.

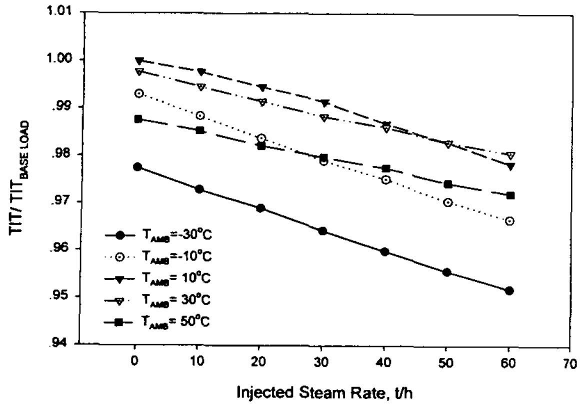

Fig. 7. Turbine inlet temperature vs. steam injection rate.

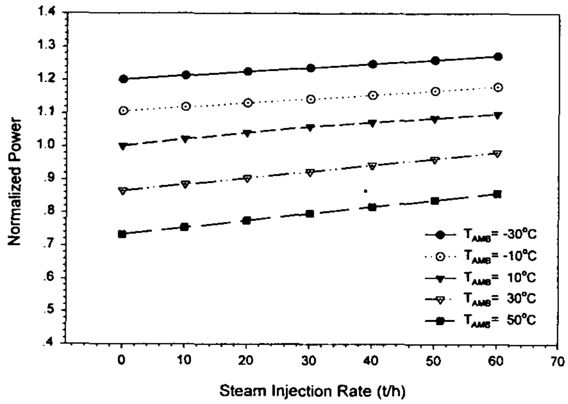

Fig. 8. Power output vs. injected steam mass flow rate.

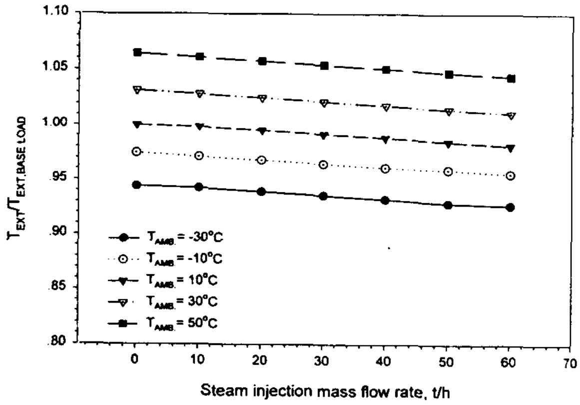

Fig. 9. Turbine exhaust temperature vs. injected steam mass rate

PART LOAD PERFORMANCE

It is very important to have accurate information on how the power plant will react to changes in outside conditions such as ambient conditions, part load. Analysis of steady state & dynamic operating behavior during part load condition will be vital to the safe operation of the plant system. Part load, herein, refers to. power reduction at levels below the base load of ambient temperature 11.16℃, which is the temperature of the acceptance test. Figure 10 shows part load performance of the present calculation with same input pressure loss of 129.5mm H_2O inlet and 85.3mm H_2O exhaust, respectively as measured during acceptance test and calculated performance results are compared with those of an actual plant(SeoInchon combined cycle). Power and efficiency prediction in this calculation agree very well with acceptance test data of SeoInchon gas turbine power plant for base and 75% part load, however, seemed to diverge at low range of part load. Turbine exhaust temperature calculated agrees very well with acceptance test data for the range of the load studied. Figure 11 shows heat rate and specific power variation during part load condition for different ambient temperatures. Specific power decreases with increasing ambient temperatures whereas heat rate increases with increasing ambient temperatures. This is obvious because heat rate is proportional to Psfc which is inversely proportional to specific power. More specific power per unit heat input means less heat rate. Figure 12 shows part load efficiency of the gas turbine and the combined cycle plant. Based on full load condition. Both the paths of gas turbine and combined cycle efficiencies are more or less similar but efficiency of the

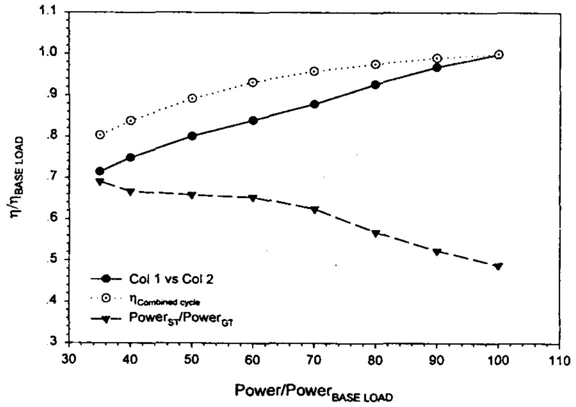

Fig. 10. Performance comparison of calculated values with test results, all normalized by the base load values.

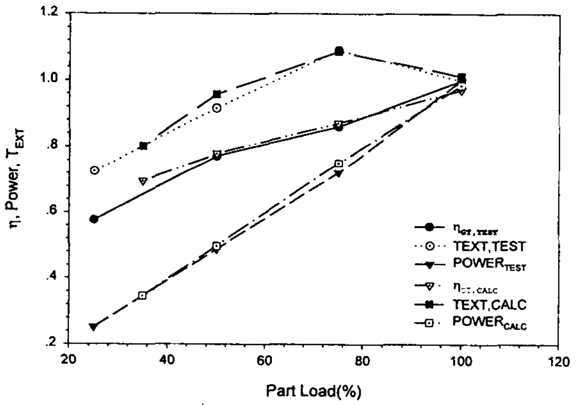

Fig. 11. Normalized heat rate and specific power during part load conditions.

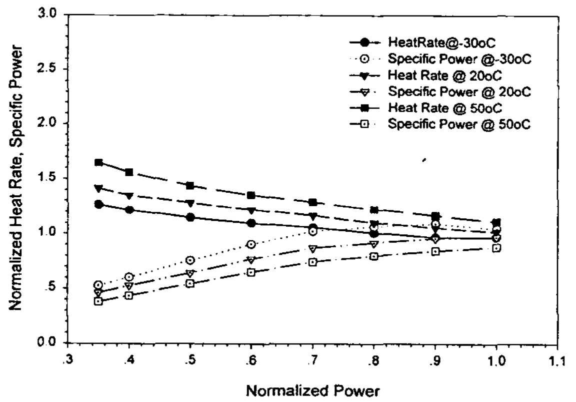

Fig. 12. Part load efficiencies of the gas turbine, combined cycle plant and ratio of steam turbine output to gas turbine output.

combined cycle are between 43~54% whereas that of gas turbine varies between 23~33% approximately. Figure 12 also shows that steam outputs more power than gas turbine in the low range of part load condition. This means that more gas turbines need be employed to improve part load efficiency of the combined cycle plant. This alefficiency of the combined cycle. Table 3 is the heat transfer calculation results of the plant system. Heat transfer calculations are governed by

$$Q = UA \cdot \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)} \tag{12}$$

where, ΔT_1, ΔT_2 are defined as

$$\begin{aligned} \Delta T_1 &= T_{h,i} - T_{c,o} \\ \Delta T_2 &= T_{h,o} - T_{c,i} \end{aligned} \tag{13}$$

and subscripts h, c, i, o indicate hot, cold and inlet, outlet, respectively.

Table 3. Heat transfer in the HRG

	UA (KCAL/S-C)	Q Kcal/s	Area m^2	Flow t/h
LPS	3.3	106	492	10.184
LPB	60.14	2261	6538	15.181
LTE	77.16	3803	8655	239.601
RH2	41.65	7624	5373	225.3
IPS1	7.87	334	1285	42.641
IPB	157.32	5711	15603	43.067
IPE2	27.45	923	2876	225.3
HPS3	43.43	6543	5672	184.504
HPB1	270.87	17995	25566	184.504
HPE3	147.81	6175	15219	186.349
HPE1	143.54	4224	15390	186.349

CONCLUSIONS

Followings are the results noticed from the present study.

1. The gas turbine with the highest efficiency does not necessarily produce the best overall efficiency of the combined cycle. Improving the gas turbine efficiency is helpful only if it does not bring too much of steam process efficiency.
2. Increase of ambient temperature reduces gas turbine efficiency whereas combined cycle efficiency increases. The increased gas turbine exhaust temperature raises the efficiency of the steam process enough to more than compensate for the reduced efficiency of the gas turbine unit.
3. During low range of part load condition, steam turbine output more power than gas turbine indicating more gas turbine employment is needed for a better combined cycle efficiency.
4. When evaluating the suitability of a gas turbine for a combined cycle process, consideration must given not only to its efficiency but also to its exhaust gas temperature.
5. Pinch points in the present study occur in both high pressure(HPB) and intermediate pressure boiler(IPB) at temperature difference of 16.67℃.

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