리큐퍼레이터를 고려한 50KW급 터보제너레이터 가스터빈 엔진의 성능해석

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Performance Analysis of a 50kw Turbo-Generator Gas Turbine Engine with a Recuperator

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ABSTRACT

Performance analysis of a 50KW turbo-generator gas turbine engine with a recuperator was studied. Recuperated cycle has been employed to meet maximum fuel economy and ultra low emissions especially for military and vehicular engines. From thermodynamic stand point, it is known that recuperative cycle can contribute most to enhance thermal cycle efficiency for the pressure ratios under 10 and of comparatively low turbine inlet temperature. Efficiency of a simple cycle with a recuperator increases relatively about 30% than without one at effectiveness of 0.5. Pressure losses in the heat exchanger less than 5.2% is considered in the design process. A tubular type heat exchanger is selected for this particular engine because it can provide simple construction as well as structural sturdiness and excellent leak tightness.

초 록

50kw급 터보제너레이터 가스터빈 엔진에 리큐퍼레이터가 부착되는 경우 성능 변화를 조사하였다. 리큐퍼레이터는 군사 및 소형 자동차용 엔진에 경제적 연비와 배기가스 저감의 목적으로 많이 적용되어 왔다. 열역학적 관점에서 볼 때 재생사이클은 압축비 10이하 및 비교적 낮은 터빈 입구온도에서 사이클의 효율 상승에 기여하는 바가 큰 것으로 나타난다. 1축단순사이클 터보제너레이터 가스터빈 엔진에 리큐퍼레이터를 부착하는 경우 리큐퍼레이터 효율 ε = 0.5에서 엔진의 효율은 상대적으로 약 30% 증가하는 것으로 나타났고 이때 열교환기내의 압력 손실은 5.2%로 설계하였다. 용이한 제작, 구조적 견고성, 최소의 누출 등의 장점으로 튜브형의 열교환 시스템이 본 가스터빈 엔진에 적합한 것으로 판단되었다.

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NOMENCLATURE

A,a	Area, Air
c	cold, compressor
ε	recuperator effectiveness
h	Enthalpy, hot
LHV	Low Heating Value
π_c	Compressor pressure ratio
Psfc	Power specific fuel consumption
TIT	Turbine inlet temperature
η	Thermal efficiency
σ	Pressure loss ratio

1. Introduction

In an actual cycle, the temperature of the air leaving the compressor is considerably lower than the temperature of the gas leaving the turbine, and some degree of waste heat recovery in the gas turbine is imperative to improve operating effieicncy. Two commonly used ways to utilize this exhausting heat energy of high temperature gas are to equip with a recuperator or a regenerator. A recuperative heat exchanger is comparatively easier to build and rugged in design than regenerators and uses common shell and tube heat exchangers, although recent trends have been toward plate-fin arrangements[1][2]. Employing the recuperator essentially enables increase in the cycle efficiency without undergoing design changes in the turbine and compressor. Fig. 1 shows a symbolic representation of a gas turbine cycle with a recuperator. The heat exchanger is located in an exhaust channel behind the turbine and allows return of a part of wasted heat into a cycle. Here, air after the compressor(C), having temperature T_3 (refer to Fig. 2), before entering the combustion chamber(CC),

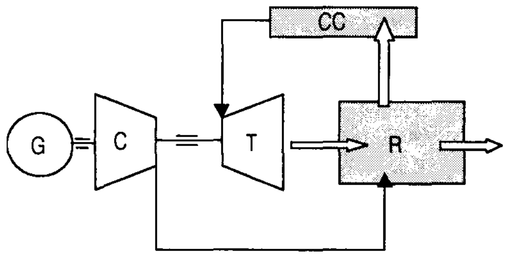

Fig. 1. Principal diagram of a recuperative cycle.

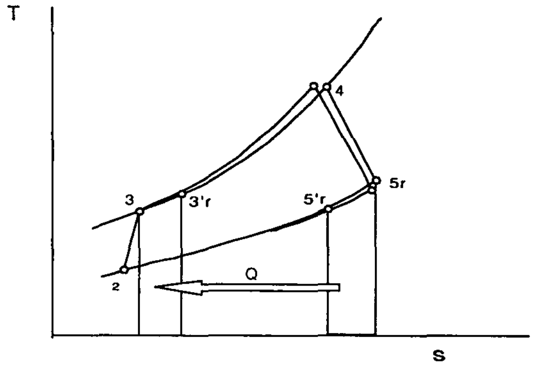

Fig. 2. T-s diagram of a recuperative Brayton cycle.

passes through the recuperator(R), is heated up due to exhaust gas from the turbine(T) with temperature T_{5r}. Characteristics of a recuperative cycle can be described with two terms such as effectiveness and total pressure drop. Recuperative effectiveness can be represented by the ratio of compressor air temperature rise, and the maximum difference between turbine exit temperature and that of compressor inlet, expressed as percentage. The definition of effectiveness is as follows.[3]

$$\varepsilon = \frac{(T'_{3r} - T_3)}{(T_{5r} - T_3)} \tag{1}$$

where, subscripts 3 and 5 indicate compressor exit and turbine exit, respectively whereas r indicates recuperator. The second parameter is total pressure losses on gas and air in the channel:

$$\frac{\Delta P}{P} = \frac{\Delta P_g}{P_5} + \frac{\Delta P_a}{P_3} \qquad (2)$$

Pressure drops in a gas turbine reduce engine performance because increased pumping power is required by the compressor to drive the flows through the resistances with the associated pressure drops. The engine consists of a compressor, permanent magnetic generator, turbine and combustor with a heat exchanger to recover the exhaust heat from the turbine. The engine rotates as a single shaft with 80,000 rpm. The permanent magnetic generator is coupled into the same shaft to make a gear box unnecessary and bring advantages such as reducing considerable amounts of component parts otherwise necessary.[4][5] Also this *arrangement will increase the system's* credibility and cost reduction. Present engine has a design point pressure ratio of 4.0, turbine inlet temperature of 1100°K with compressor and turbine efficiencies of 80.0, 85.2 percent, respectively. Table 1 shows the design point data. Detailed calculation of turbine indicate it's efficiency must be equal to 0.852 and a new design point is made here. Partload performance analysis of the turbogenerator gas turbine engine was made[6] and is not repeated here, but is scheduled to be presented in the separate paper.

Table 1. Design point of a 50kw turbo-generator engine.

Design Point	Values	Unit
Ambient conditions		
Temperature	288.15	K
Pressure	101.325	KPa
Intake pressure loss	0.98	
Compressor		
Temperature exit	462.19	K
Pressure exit	397.194	KPa
Pressure ratio	4.0	
Efficiency(Isentropic)	0.8	
Mass flow rate	0.5	kg/sec
Δhc	87.88	KW
Bleed	0.0	%
Combustor		
Temperature exit	1100.0	K
Pressure exit	373.362	KPa
Pressure loss	6.0	%
Efficiency	0.99	
Fuel LHV	43.124	MJ/kg
Mf/Ma	0.01584	
Turbine		
Temperature exit	850.53	K
Pressure exit	106.495	KPa
Expansion ratio	3.506	
Efficiency(Isentropic)	0.852	
Mass flow rate	0.508	kg/sec
Δht	140.076	KW
Rotor dynamics		
Ratating speed	79,750	rpm
Heat exchanger		
Effectiveness	(0.5)	
Exit air temperaturre	660.9	K
Heat transferred to air	205974	J/kg
Performance w/o Recup.		
Shaft horse power	54.3	KW
Psfc	0.6106	kg/(kw.h)
Thermal efficiency	15.1	%
Performance w. Recup		
Shaft horse power	49.4	KW
Effectiveness	0.6	
Thermal efficiency	19.4	%

2. Influence of a recuperator effectiveness on cycle performance

Thermal efficiency and specific power are two parameters that quantify the performance of gas turbine cycles. A high performance engine has high thermal efficiency and specific power. For an identical flow condition, the engine with larger specific power produces more power and for a same power output, the engine with larger specific power is small. Fig. 3 shows the influence of turbine inlet temperature on thermal efficiency of simple cycle. As explained in the previous paper[6], TIT of 1100°K was set up to obviate the inclusion of turbine cooling air for simpler aerodynamic and structural configuration

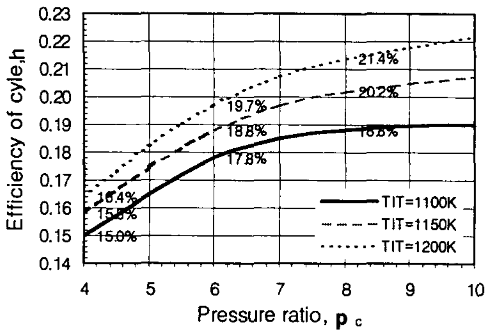

Fig. 3. Influence of TIT on smple cycle turbogenerator efficiency.

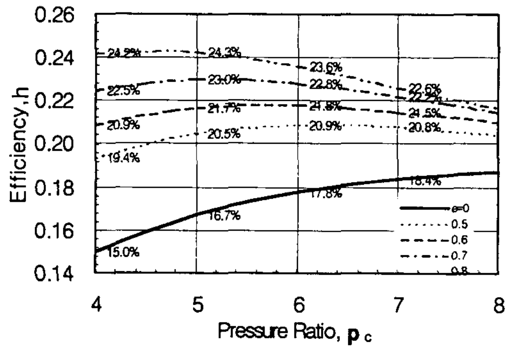

Fig. 4. Influence of a regeneration degree on the cycle efficiency(h_c=0.8, h_t=0.852, TIT=1100°k, $\Delta P = f(\varepsilon)$)

besides material confinement. Lower pressure ratio of 4.0 compared to optimum 5.8 is selected to avoid the final product becomes too large for the shaft power intended and based on the fact that the specific fuel comsumption's decreasng rate after pressure ratio 4.0 remains minute. From the figure, at constant pressure ratio of 4.0, TIT inrcrease upto 1150°K achieve 15.8% with effective power of 60KW. However a significant increase in cycle efficiency by means of TIT increasing will require full modification of compressor and turbine parts and is not considered here.[7][8][9]

On Fig. 4, the influence of a rgeneration degree on cycle efficiency with total pressure losses taken into account is considered. Regeneration ratio with pressure loss can be given as[10]:

$$\sigma_r = (1 - 0.012/(1-\varepsilon))^2 \tag{3}$$

It is visible from Fig. 4 that at ε = 0.5, the increase in thermal efficiency is equal to 30% relative and 4.3% absolute compared to simple cycle. It is necessary to note that at ε = 0.8 and TIT = 1100°K the optimum pressure ratio is close to 4.2. And cycle efficiency in this case is equal to 24.2%. This is a rather high thermal efficiency considering a modest turbine inlet temperature. However, with further detailed calculations, it becomes obvious that rational arrangement of recuperator with effectiveness 0.7 - 0.8 is impossible, because the volume for the heat transfer matrix is too small even for the plate-fin type heat exchanger. For a tubular construction, the effectiveness is confined to be less than 0.6 (Fig. 5). Fig. 5 shows the volume calculation results of tube, prime and plate fin type heat exchangers of different effectiveness. From the figure, tubular type recuperator requires more surface area per volume compared with other types of recuperator and upper limit for regeenrator effectiveness is close to 0.63. Tubular approach of gas turbine recuperator design is characterized by the fact that high pressure, low temperature air is being carried inside small diameter tubes which are subjected on the outside to the low pressure and high temperature exhausted gas, and is preferred here as it can provide structural integrity and excellent leak tightness[11][12]. The thick horizontal line indicates admissible volume of the recuperator considering the geometrical

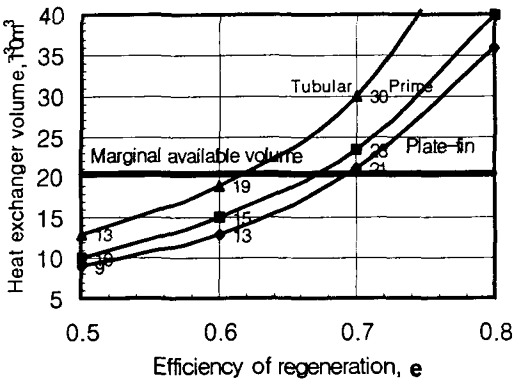

Fig. 5. Volume of heat exchange matrix variation for different types of heat exchangers with regeneration ratio.

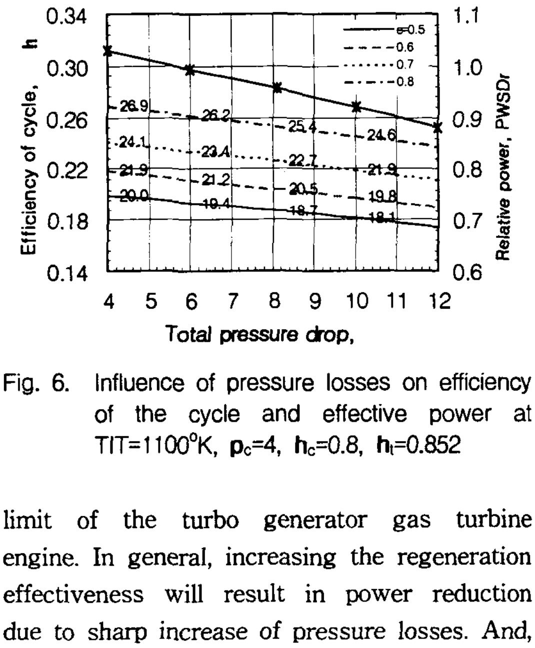

Fig. 6. Influence of pressure losses on efficiency of the cycle and effective power at TIT=1100°K, p_c=4, h_c=0.8, h_t=0.852

limit of the turbo generator gas turbine engine. In general, increasing the regeneration effectiveness will result in power reduction due to sharp increase of pressure losses. And, at ε = 0.7 (it requires ΔP = 8 - 9%) useful power decreases 7 - 8%. as illustrated in Fig. 6. Increasing the regeneration degree within the confined space will result in nonuniform heat transfer matrix size and that will cause severe decrease in thermal efficiency. This is why the regeneration degree is decreased to 50% so that unform heat carriers distribution in the recuperator channels is obtained. The influence of total pressure losses in the recuperator on the cycle efficiency is shown in Fig. 6. The calculations are made for ε = 0.5 - 0.8 with total pressure losses from 4 to 12%. Referenced data for this analysis is of ε = 0.5 and pressure losses 5.2%. In this case, cycle efficiency increases up to 19.4%.

Further increasing the cycle efficiency is possible by increasing the regeneration degree of the recuperator[11][12]. In a tubular construction this can be reached both by intensification of heat exchange inside the tubes, and by increasing the heat transfer surface. The former, saving constant overall dimensions of a turbular heat exchange matrix, keads to growth of pressure loss(direction A) and results in the decrease of thermal efficiency because heat exchange intensification inside the tubes brings in pressure lossesthus liquidating the effect of regeneration. The second way to improve thermal efficiency is to increase the heat exchange matrix. This can, in connection with the boundedness of available volume, increase non-uniformity of stream distribution in the input and output of manifolds for both gas and air sides, and brings positive thermal effect. On the other hand, increase of turbogenerator efficiency is possible by using compact plate-fin type heat exchangers. And, in this case, cycle efficiency can be directed by lines B and C. A plate-fin type recuperator with ε=0.7 and pressure losses 7-8% would allow to achieve an efficiency of cycle 23%. However, at the present phase of turbogenerator creation, the simplicity and reliability of tubular version is favored in this low thermal ratio. Furthermore, introduction of a recuperator with ε=0.7 the temperature of air exiting compressor, as contrasted with a cycle without regeneration, will increases about 260°K(from 190°K to 450°K) and this will certainly require significant modernizing of the combustion chamber. And a rather high outlet gas temperatureor or inlet temperature of a recuperator(T_{5r} = 857°K for ε = 0.5, with higher degree of regeneration this temperature will increase) press the selection of the modest regenerative ratio and tubular construction of heat exchanger. Fig. 7 shows tubular heat exchanger inside the turbogenerator casing. Heat exchanger has cross flow scheme and the heat exchanger matrix consists of tubes bundle with internal tubulators. The casing is thin walled as the pressure difference between inside and poutside is small. It is necessary to indicate that with in-service experience of a turbogenerator and further enhancement of regenerative scheme of higher parameters, it will be necessary to consider plate-fin type construction which require higher technological level and careful experimental improvement. Fig. 8 shows the conceptual design of a recuperator for the 50kw turbo generator gas turbine engine. The shell and tube type recuperator consists of the cylindrical casing, where tubular matrix is located, transversal baffles and collector ring. The air comes in through the windows, which are attached to the casing, and passes through the first section of tube bundles, then turns around the partition baffle. After then it passes through the second tube bundles before finally exits to the collector ring which is connected to the combustor chamber. The conceptual design of this type of a heat exchanger ensures a degree of regeneration of 50% and total relative pressure losses approximately 5.2%. The tube has the dimensions of inner and outer diameters 7.4mm and 6.6mm, respectively. Table 2 shows the calculation of the volume for different heat exchanger effectiveness of recuperators considered in the present study. As noticed in Fig. 5, the results show that the tubular heat exchanger requires a larger volumetric space than prime and plate types for a given surface area. The main reasons for selecting a tubular type heat exchanger despite the disadvantages of less efficiency and large volume were the reliability, structural confinement and easy maintenance. Table 3 shows the calculation results of tubular recuperator for this particular study.

Partload performance analysis of a turbogenerator with a recuperator is reserved for future study. Influence of the climatic

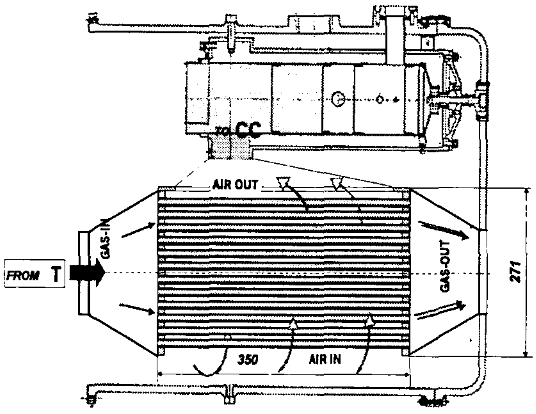

Fig. 7. Conceptual design of a recuperator for 50kw turbo-generator gas turbine engine.

Table 2. Characteristic of different types of heat exchanger.

ε	η	PWSD	Area	tubular type	prime type	plate type
	%	KW	m^2	dm^3	dm^3	dm^3
0.5	20.5	50.3	8.95	12.78	5.46	4.47
0.6	22.0	49.2	13.43	19.18	8.19	6.71
0.7	23.6	47.3	20.89	29.85	12.74	10.45
0.8	24.7	43.5	35.84	51.20	21.85	17.92

Table 3. Dimensions of the tubular heat exchanger.

No.	Name	Param.	Value	Unit
1	Outer diameter	d_2	0.0074	m
2	Inner diameter	d_1	0.0066	m
3	Cross tube spacing	S_1	0.009	m
4	Clearance between tubes	δ_{Cl}	0.0015	m
5	Relative tube cross spacing	S_1/d_2	1.22	
6	Tube length	L	0.365	m
7	Number of pipes	Z_1	701	
8	Recuperator external diameter	D	0.285	m
9	Casing thickness	$b_{cas.}$	0.002	m
10	Tube desk thickness	b_{desk}	0.008	m
11	Volume	V	0.0233	m3
12	Mass	M	31.75	kg
13	Compactness factor	K	255.0	m^2/m^3

temperature and speed incorporating components' performance characteristics on system's performance will be investigated in the future study.

3. Conclusions

For a simple cycle turbo-generator performance analysis with a recuperator is carried out in the present study. Followings are the results noticed from the study.

(1) Performance analysis of a turbo-generator with a recuperator was carried out to investigate the influence of a tubular type recuperator.

(2) Investigation shows that the relative cycle efficiency increases considerably (approx. 51%) with the inclusion of a recuperator at effectiveness of 0.6, however, power output decreases about 2.8%.

(3) *For this particular cycle, a tubular type* recuperator is selected for easier construction and design within the gemetrical confinement of the turbo generator casing resulted from the previous study.

(4) In view of operating and construction features of a turbo-generator, taking into account cost effects, and also initial stage of updating, the tubular heat exchangers with ε = 60% and pressure losses of 7% was slelcted.

(5) Even though a significant increase in cycle efficiency can be gained through the use of a recuperator, it must be weighed against the disadvantages of increased service problems, cost, size and weight for maximum economic benefit.

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