Layout Study of Supercritical CO_2 Power System for Waste Heat Recovery Application

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1. Introduction

As a part of supercritical CO_2 cycle development, interests on various cycle layout to increase the system power from the waste heat are growing. Several layouts including simple recuperated, preheating recuperated and cascaded recuperated cycle are investigated with some component parameters and LM-2500 exhaust gas condition. Thermal power, system efficiency and cycle mass flow rate are assessed depending on the cycle layouts.

2. Supercritical CO_2 Cycle Layout Design

In this section some characteristics of supercritical CO_2 cycle layouts and waste heat recovery system condition are introduced. Several major parameters are described and the realistic performance is assessed.

2.1 Simple Recuperated Layout

Table I. Supercritical CO_2 cycle design condition

sCO_2 Assumption	Value
Turbine isentropic efficiency	85%
Compressor isentropic efficiency	80%
Compressor inlet pressure	7.739 MPa
Compressor inlet temperature	34.1℃
Recuperator effectiveness	90%
Waste heat exchanger effectiveness	90%
Recuperator pressure loss	0.5% (cold side) 1.5% (hot side)
Waste heat exchanger pressure loss	4%
Precooler pressure loss	3%
Pipe pressure loss	0.5% (each)
Primary gas turbine (LM-2500)	Value
Exhaust gas flow rate	70.5kg/s
Exhaust gas temperature	566℃
Gas turbine efficiency	35.5%
Thermal combustion power	61.6 MWth
Waste heat power (@ 15℃)	39.7 MWth

Before discussing the cycle layout, some component parameters and waste heat condition are introduced. The component (turbomachineries and heat exchangers) parameters are carefully assumed based on the realistic assumption and industrial references as shown in Table I. The pressure losses of heat exchangers and pipes are roughly considered. Waste heat thermal power considering 15℃ atmospheric condition is assessed as well.

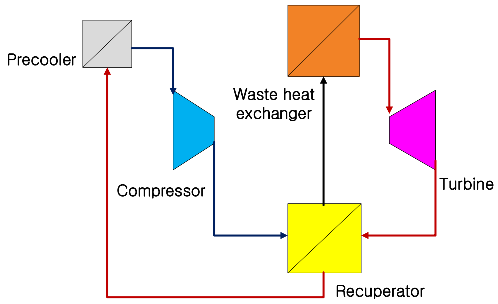

Fig. 1. Simple recuperated layout

Fig. 1 shows the representative layout (simple recuperated layout). As the turbine outlet temperature is still high, some portion of heat is recuperated in the system to increase the system efficiency. Considering the supercritical CO_2 cycle design, compressor inlet condition is controlled because the cycle efficiency increases as the compressor inlet condition approaches to the critical point. In this system, the static condition of compressor inlet condition is critical point (30.98℃, 7.3773 MPa).

The thermal power and cycle efficiency gradually increase as the maximum pressure increases. Therefore the cycle pressure ratio can be determined based on the turbomachinery suppliers. Generally high pressure condition increases capital cost as well. In this study, the compressor outlet condition of 20 MPa is considered.

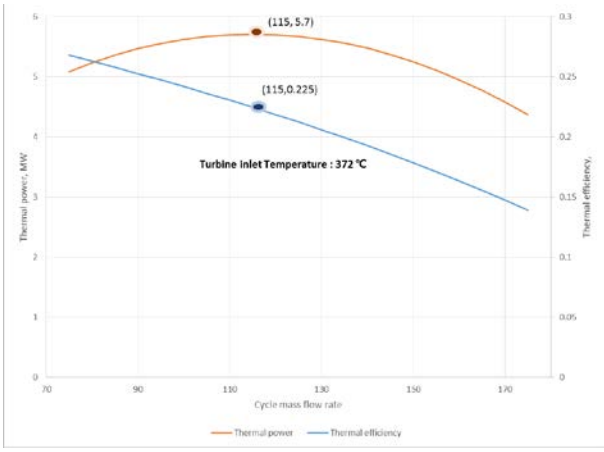

Fig. 2. Thermal power and cycle efficiency of simple recuperated cycle

Cycle mass flow rate can be selected to produce the maximum thermal power from the heat source. 41% of waste heat is recovered to the supercritical CO_2 system and 5.7MW power is additionally produced.

However, due to the narrow temperature gradient in the waste heat exchanger, small portion of heat is recovered from the heat source. Therefore, additional waste heat exchanger or recuperator. In this study, the preheating recuperated system and cascaded recuperated system are further investigated as shown in Fig.3 and Fig. 4.

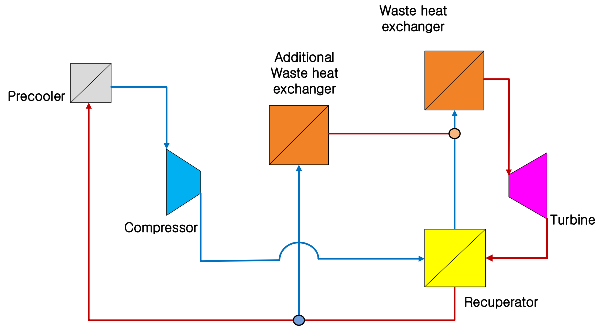

Fig. 3. Preheating recuperated layout

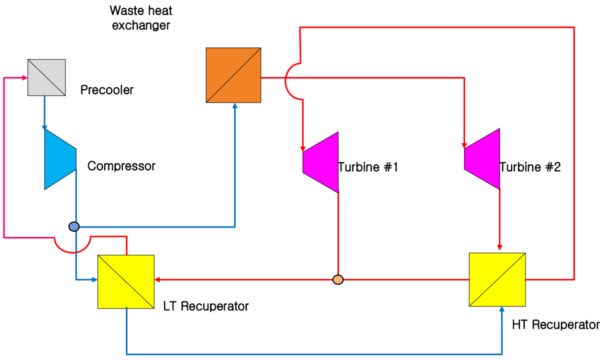

Fig. 4. Cascaded recuperated layout

In preheating recuperated layout, some portion of compressed CO_2 is preheated and merged with the recuperated flow. Therefore, the outlet temperature of exhaust gas significantly low compared to the simple recuperated layout. The sensitivity study of flow split ratio and cycle mass flow is investigated as shown in Fig. 5 and Fig. 6.

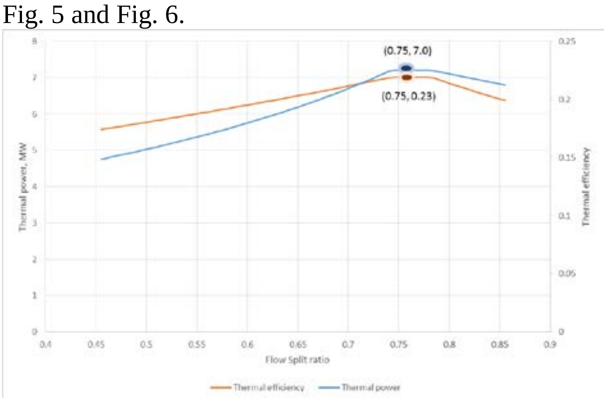

Fig. 5. Thermal power and cycle efficiency with respect to the flow split ratio (preheating recuperated layout)

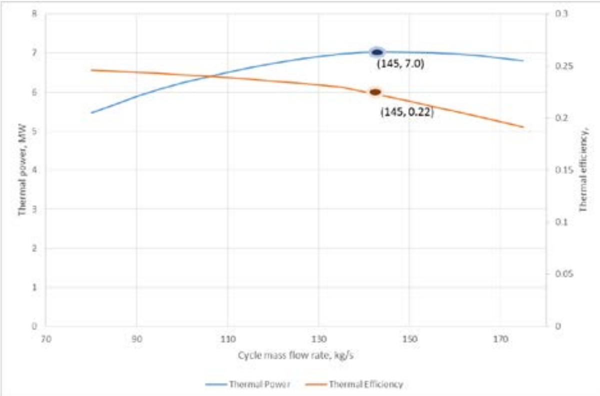

Fig. 6. Thermal power and cycle efficiency with respect to the cycle mass flow (preheating recuperated layout)

In cascaded recuperated layout, a turbine and a recuperator are added to increase the heat recovery from the heat source. One turbine expands with the recuperated heat inside the system and another turbine expands with the recovered heat from the heat source. The sensitivity study of flow split ratio and cycle mass flow is analyzed as shown in Fig. 7 and Fig. 8.

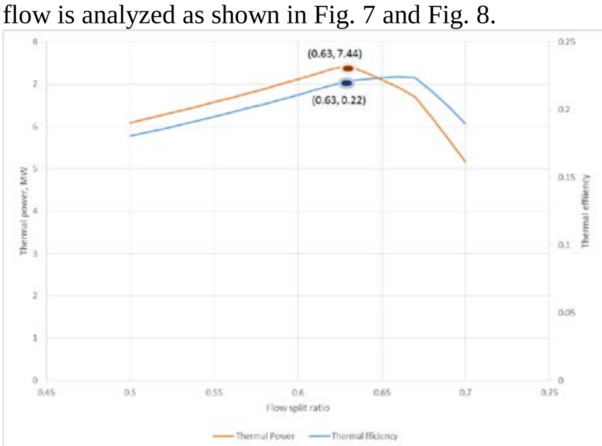

Fig. 7. Thermal power and cycle efficiency with respect to the flow split ratio (cascaded recuperated layout)

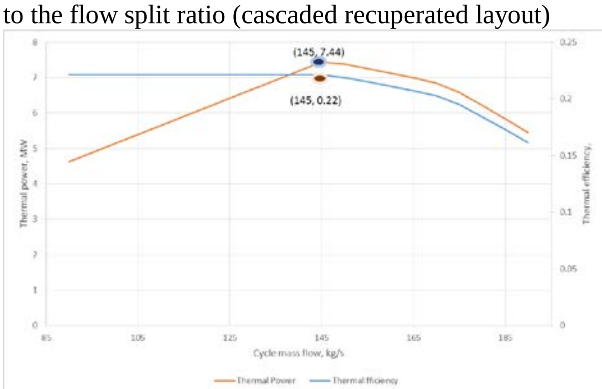

Fig. 8. Thermal power and cycle efficiency with respect to the cycle mass flow (cascaded recuperated layout)

The comparison of cycle performance is summarized in Table II.

	Simple recuperated	Preheating recuperated	Cascaded recuperated
Heat (WHR)	25.4 MW	31.8 MW	33.6 MW
Heat (Recuperator)	27.3 MW	34.3 MW	34.9 MW
Compressor work	3.0 MW	3.8 MW	3.8 MW
Turbine work	8.7 MW	10.8 MW	11.2 MW
Thermal power	5.7 MW	7.0 MW	7.4 MW
Cycle efficiency	22.5 %	22.1 %	22.1 %

3. Conclusion and further works

Compared to the simple recuperated layout, marginal power can be expected with the addition of heat exchangers and turbomachineries. If the overall system size is not strictly controlled, preheating recuperated layout or cascaded recuperated layout is desirable. Further studies related to the turbomachinery and heat exchanger designs are required.

ACKNOWLEDGEMENT

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