§20. Feasibility of Helium Gas Turbine System for Molten Salt Blanket

Shimizu, A., Yokomine, T., Ebara, S. (Interdisciplinary Grad. Sch. Eng. Sci., Kyushu Univ.),
Imagawa, S., Sagara, A.

1. Introduction

In the FFHR design activity, helium gas turbine cycle is thought to be a candidate energy conversion system, in which thermal energy extracted from blanket by molten salt FLIBE is transferred to a secondary helium flow through heat exchangers. Although the use of gas turbine is not necessarily attractive because of very restricted design window of the operating temperature of FLIBE, conventional power cycles such as light water Rankin cycle are disliked due to the difficulty in tritium handling. Accordingly, an improvement of thermal efficiency of gas turbine system is a key issue for the liquid blanket concept using FLIBE.

2. Previous estimation of thermal efficiency

Based on the above, numerical estimations have been performed in last several years on the thermal efficiency of multi-stage closed gas turbine cycle and it was concluded that when five stage compression/expansion is adopted, the gas turbine cycle efficiency up to 42 % is possible even for the narrow temperature design window of FFHR, namely between 450℃ and 550℃. However, that estimation has been based on very optimistic evaluation of pressure loss within the cycle path so that more realistic estimation is expected.

Following this, the research effort was turned to the direct examination of the Ericsson cycle itself in which heat addition and rejection in respective expansion and compression phases are performed isothermally. The isothermal expansion process was previously proposed and examined by several authors for the open-cycle combustion gas turbine. Meanwhile, there exists no corresponding investigation for the closed cycle gas turbine system. In the open cycle case, the isothermal expansion can be realized through continuous fuel addition in the expansion phase, while it must be via. heat transfer through any separating wall between FLIBE and helium flow for the closed cycle case.

If ideal Ericsson cycle is realized, it is equivalent to the Carnot one.

3. Rough estimation of Ericsson Cycle

In a previous year, the performance of the Ericsson cycle has been investigated, particularly for the expansion phase. The estimation was extended this year so as to cover the compression phase as well. The followings must be noticed.

(1) As high gas temperature as possible is favorable even for the Ericsson cycle while temperature level for the heat rejection was fixed.

(2) Heat transfer coefficient between operating gas and casing wall or turbine blade must be raised typically several orders higher compared with conventional equipments. Of possible heat transfer enhancement techniques available at present, the heat transfer enhancement by use of nano-particle coating is promising. However, reported heat transfer improvement is barely twice of conventional value so that more innovation is strongly expected. Moreover, there remains some uncertainty in the enhancement effect for gaseous heat transfer because reported enhancement effect is verified only for the liquid heat trasnfer. Another possibility is to ultra-fine heat transfer structure between turbine blades.

(3) Extension of heat transfer surface

The possible heat transfer surface for additional heat input in expansion stage is restricted to the casing wall and turbine blade surfaces. Of these, the former requires less change in the whole turbine geometry while the latter must accompany with a drastic change in the blade design.

Accordingly, a simple estimation was made first for the required heat transfer performance in the expansion phase.

Following assumptions were made.

(1) When inner diameter and length of casing are 2m×2m、the heat transfer area is $6.3m^2$.

(2) The order of heat transfer coefficient is at most 3000kcal/m^2hr℃. The conventional value for He is of 1000 kcal/m^2hr℃ order

(3) Temperature difference is about 200℃（gas-650℃、casing wall-850℃）

The rough estimation gives the heat transfer value of

6.3×3000×200×4.19/3600＝4400 kw (=4.4MW)

In fact, the value required for actual gas turbine is about 100 times of the above. Namely, simple heat input only through the casing wall will be unrealistic.

The similar estimation was performed for the expansion stage and conclusion was that the maximum attainable heat transfer rate was estimated to be barely 3.3MW for the casing size of 2m inner diameter×3m length. This value was compared with the actual required level and the conclusion was that the heat transfer enhancement of two order higher is required .

It is concluded therefore that the Erricson type expansion/compression gas turbine cannot be realized at least through heat addition/rejection by way of casing walls.

4. Conclusion and prospect

According to the above estimation, attainable heat input through the simple use of expander/compressor casing wall amounts to only hundredth of the required level of actual power cycle. Remaining possibility is heat addition or rejection by making use of the flow paths between static blades of expander/compressors as heat exchange region. Of possible candidates for it, the fine woven-fiber heat transfer structure may be attractive. However, drastic increase in pressure drop will be inevitable and an extensive research for verifying it is important for that purpose.

Therefore, the next step is to find out a compromising point between two concepts, one being the multistage compression/expansion system and other the use of fine fiber heat exchange structure.