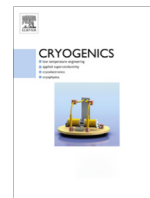




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Research paper

Integrated design of cryogenic refrigerator and liquid-nitrogen circulation loop for HTS cable

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ABSTRACT

A new concept of cryogenic cooling system is proposed and investigated for application to long-length HTS cables. One of major obstacles to the cable length of 1 km or longer is the difficulty in circulating liquid nitrogen (LN) along the cables, since the temperature rise and pressure drop of LN flow could be excessively large. This study attempts a breakthrough by integrating the refrigerator with the LN circulation loop in order to eliminate the cryogenic LN pumps, and generate a large LN flow with the power of compressors at ambient temperature. A variety of thermodynamic structures are investigated on standard and modified Claude cycles, where nitrogen is used as refrigerant and the LN circulation loop is included as part of the closed cycle. Four proposed cycles are fully analyzed and optimized with a process simulator (Aspen HYSYS) to evaluate the FOM (figure of merit) and examine the feasibility. The modified dual-pressure cycle cooled with expander stream is recommended for long HTS cables.

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1. Introduction

HTS (high-temperature superconductor) cables have been actively developed in Korea for over 15 years. Under a governmental program, the demonstration of 1 km HTS cable is now getting close to its completion at the transmission grid (3-phase 154 kV AC) in Jeju Island. In the following years, the KEPCO (Korea Electric Power Corporation) plans to replace old underground cables in selected urban area with new HTS cables, whose unit length is mostly in the range of 1–3 km. In accordance with the development of HTS cables, the cryogenic cooling technology for continuously circulating liquid nitrogen (LN) along the cables has evolved as well.

In early stages, a simple decompression system was adopted [1–3], as shown in Fig. 1(a). Nitrogen gas is pumped out from a liquid-nitrogen container (called the LN sub-cooler) at sub-atmospheric pressure, where heat is removed from the circulating LN flow. The decompression system is suitable for a short (100 m or less) cable or for demonstration purpose, as it is simple in operation and inexpensive for construction. However, a huge amount of LN must be supplied to this “open” system in a scheduled period.

For continuous operation of a “closed” system, cryogenic refrigerators should be employed. Two distinct types of refrigerators are

applicable to the HTS cable systems, mainly depending on the level of cryogenic load. When the thermal load is relatively small, multiple units of regenerative cryocoolers can be conveniently used. For example, a set of four stirling coolers shown in Fig. 1(b) is commercially available to cover the load of 1.5–2 kW at 70 K, as recently applied to 500 m DC cable in Korea and several other systems [4–10]. In some cases, the decompression units have been installed in parallel as a back-up or in preparation for emergency operation.

For long AC cable systems under a heavier thermal load, recuperative refrigerators (based on JT, Brayton, or Claude cycle) are more suitable, not only because the cooling capacity can be greater, but also much larger surface area can be provided for cooling LN with high-effectiveness heat exchangers. In case of 1 km AC cable, for example, a Brayton refrigerator is employed, as shown in Fig. 1(c) [11–16]. Neon Brayton refrigerators are commercially available to cover 2–9 kW at 70 K [12,13], and the helium Brayton refrigerator with a target capacity of 10 kW at 70 K is also under development for long HTS cables [14,15].

The LN circulation loop in these cryogenic systems can be represented by a triangular shape of liquid cycle on phase (pressure–temperature) diagram of nitrogen, as shown in Fig. 1(d). The three curves indicate (in counterclockwise direction) the cooling flow along HTS cable, the pressurization by pump, and the sub-cooling in heat exchanger, respectively. For the installation of long HTS cables in practice, the thermo-hydraulic design of LN circulation

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Nomenclature

<i>FOM</i>	figure of merit	<i>AC</i>	after-cooler
\dot{m}	mass flow rate	<i>C</i>	compressor
<i>P</i>	pressure	<i>E</i>	expander
\dot{Q}	heat transfer rate	<i>e</i>	LN exit of HTS cable
\dot{S}_{gen}	entropy generation rate	<i>H</i>	high pressure
<i>T</i>	temperature	<i>HTS</i>	HTS cable
\dot{W}	power or work rate	<i>HX</i>	heat exchanger
<i>x</i>	vapor fraction or quality	<i>I</i>	intermediate pressure
<i>y</i>	expander flow ratio	<i>i</i>	LN inlet of HTS cable
		<i>JT</i>	Joule–Thomson valve
		<i>L</i>	low pressure
		<i>min</i>	minimum
		<i>MIX</i>	mixing device

Subscripts

0	ambient
1, 2, 3, ...	location in refrigeration cycle

loop is a key challenge as well as the large-capacity refrigeration, because the temperature rise and pressure drop along the LN flow could be excessively large. If the flow rate of LN is relatively small, the triangular shape is horizontally sharp so that the system could be “thermally” hazardous. As the flow rate of LN increases, on the contrary, the triangular shape becomes vertically taller so that the system could be thermally more stable, but “hydraulically” more difficult. Together with the limited capacity and availability of commercial LN pumps, this thermo-hydraulic behavior of LN loop should be a severe constraint to the cryogenic cooling system for long HTS cables.

This study attempts a breakthrough by proposing and investigating a new concept of cryogenic cooling system that integrates the LN circulation loop into the refrigerator cycle. The main idea is to eliminate the LN pump, but generate a large LN flow rate with the power of compressors at ambient temperature. In the integrated design, the working fluid (refrigerant) of refrigeration cycle is nitrogen itself, and the LN circulation loop along HTS cable is a part of the closed cycle. An obvious advantage of the integrated system is to utilize the nitrogen (or air) compressors and expanders only, which are widely used in gas industry and commercially available at a variety of capacities. As first step, the structure of

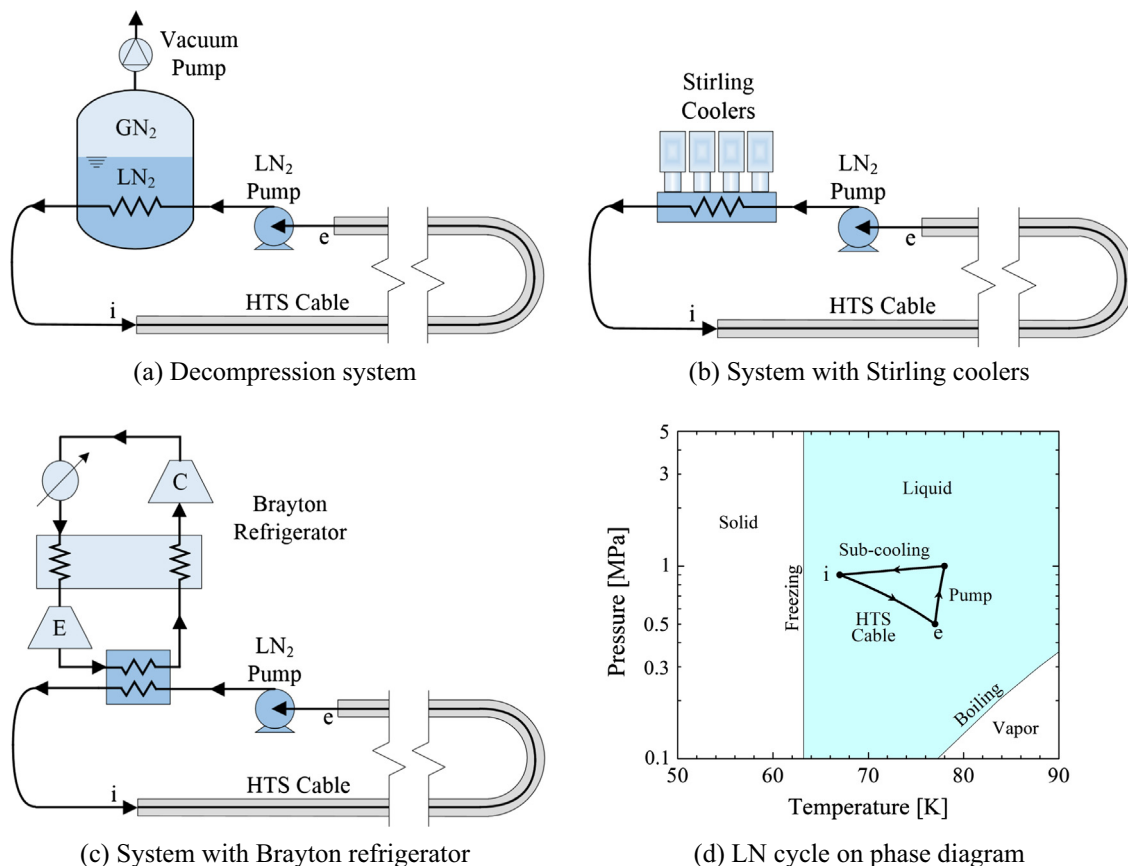


Fig. 1. Three existing cryogenic cooling systems for HTS cables and liquid-nitrogen cycle on phase diagram.

refrigeration cycle is explored in a systematic way, and then the detailed cycle analysis is performed towards an efficient thermodynamic system.

2. Proposed cycles for integrated system

Three basic recuperative cycles for cryogenic refrigeration are JT (Joule–Thompson) cycle, Brayton cycle, and Claude cycle, as distinguished by the expansion process involved [17]. A JT cycle uses JT expansion, which is the throttling process through a valve at the cold end, as shown in Fig. 2(a). On the contrary, a Brayton cycle uses adiabatic expansion, which is the work-producing process with a turbine or an expander at the cold end, as shown in Fig. 2(b). A Claude cycle uses both JT and adiabatic expansions, as shown in Fig. 2(c), where the JT valve is located at the cold end like JT cycle, and the expander is located at an intermediate stage in the branched stream.

Among these cycles, the standard or modified JT cycles are difficult to use for this application, because the cooling effect of nitrogen JT cycles is not large enough for HTS cables. The standard or modified Brayton cycles cannot be used for the integrated cycle either, because the refrigerant (nitrogen) must remain in gas (or vapor-rich) phase for expander, but a liquid cooling is required for HTS cables. On the other hand, the Claude cycle is a reasonable candidate for this application, because the expander could produce a large refrigeration and the JT valve could be effective in reaching down to LN temperatures [17]. Standard Claude cycle is therefore the starting point of this thermodynamic design, and various modifications in structure are considered.

A simplest structure for the integrated system is the standard cycle cooled with JT stream (Cycle I), where the high-pressure LN is supplied to HTS cable (i) and returned to JT valve (e), as shown in Fig. 3(a). In this cycle, the high-pressure nitrogen (at state 3) branches into two streams (called “expander stream” and “JT stream”), and the two streams merge (into state 7) at the same level of low-pressure. The pressure levels of expander stream and JT stream may be set differently in several ways, and Fig. 3(b) shows an example [17], where the JT stream has a higher supply pressure than the expander stream. This cycle is called the dual-pressure cycle cooled with JT stream (Cycle II). Two different pressure levels allow flexibility in the design of operating conditions and the selection of components. HX1 in Cycle II is a triple-

stream heat exchanger, where a cold stream is in thermal contact with two hot streams.

Another way of structural modification is to use the expander stream (instead of the JT stream) for the cooling of HTS cable, as shown in Fig. 3(c). Since sub-cooled liquid is required all along the cable, the cold vapor from the expander (at state 9) is further cooled (to liquid) in HX3, and then supplied to the HTS cable (i). Since the return pressure of expander stream is higher than the low pressure of JT stream, two streams merge at ambient temperature. This cycle is called the dual-pressure cycle cooled with expander stream (Cycle III). HX1 and HX2 are triple-stream heat exchangers, and HX3 is a quadruple-stream heat exchanger.

Finally, Cycle III is modified by adding a counter-flow HX (HX4) between the expander and JT streams at the cold end, as shown in Fig. 3(d). The role of HX4 is to further sub-cool the LN flow (state 11) with the coldest 2-phase counter-flow from JT stream. As described later, HX4 allows an opportunity to significantly reduce the temperature difference in HX3 for better thermodynamic performance. This cycle is called the modified dual-pressure cycle cooled with expander stream (Cycle IV).

3. Cooling requirements and analysis model

3.1. Cooling requirements

The cooling requirements of LN for HTS cables are described with the “trapezoidal” (shaded) area in liquid region of phase (pressure–temperature) diagram, as shown in Fig. 4. The primary requirement is the upper limit of cooling temperature for HTS conductors (for example, 80 K), as indicated by the right vertical line (denoted by “ T_c Limit”). This temperature determines the critical current density of HTS conductors, and therefore should be a basis of the electrical design of HTS cables. Second and another major requirement is the “Sub-cooling Limit”, which is indicated by the curve at bottom. The sub-cooled liquid state is crucial for high-voltage application, because any bubbles produced by internal or external heat source must be suppressed to prevent the deterioration in electrical insulation of LN. The sub-cooling requirement can be typically given by a temperature margin (for example, 10 K below the boiling temperature) [18], as graphically represented by a horizontal shift of the saturation (boiling) curve to left in phase diagram.

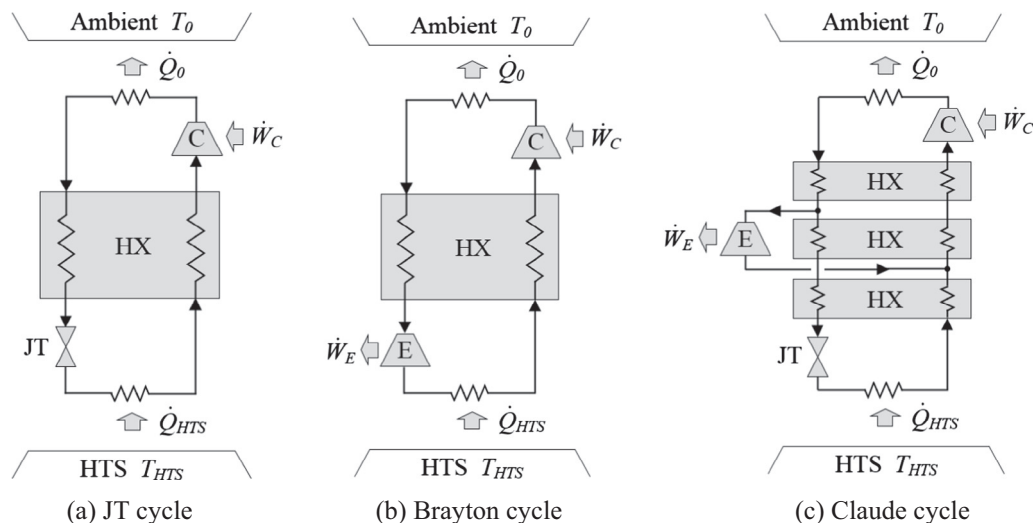


Fig. 2. Three basic recuperative refrigeration cycles (C: compressor, HX: heat exchanger, JT: Joule–Thomson valve, E: expander).

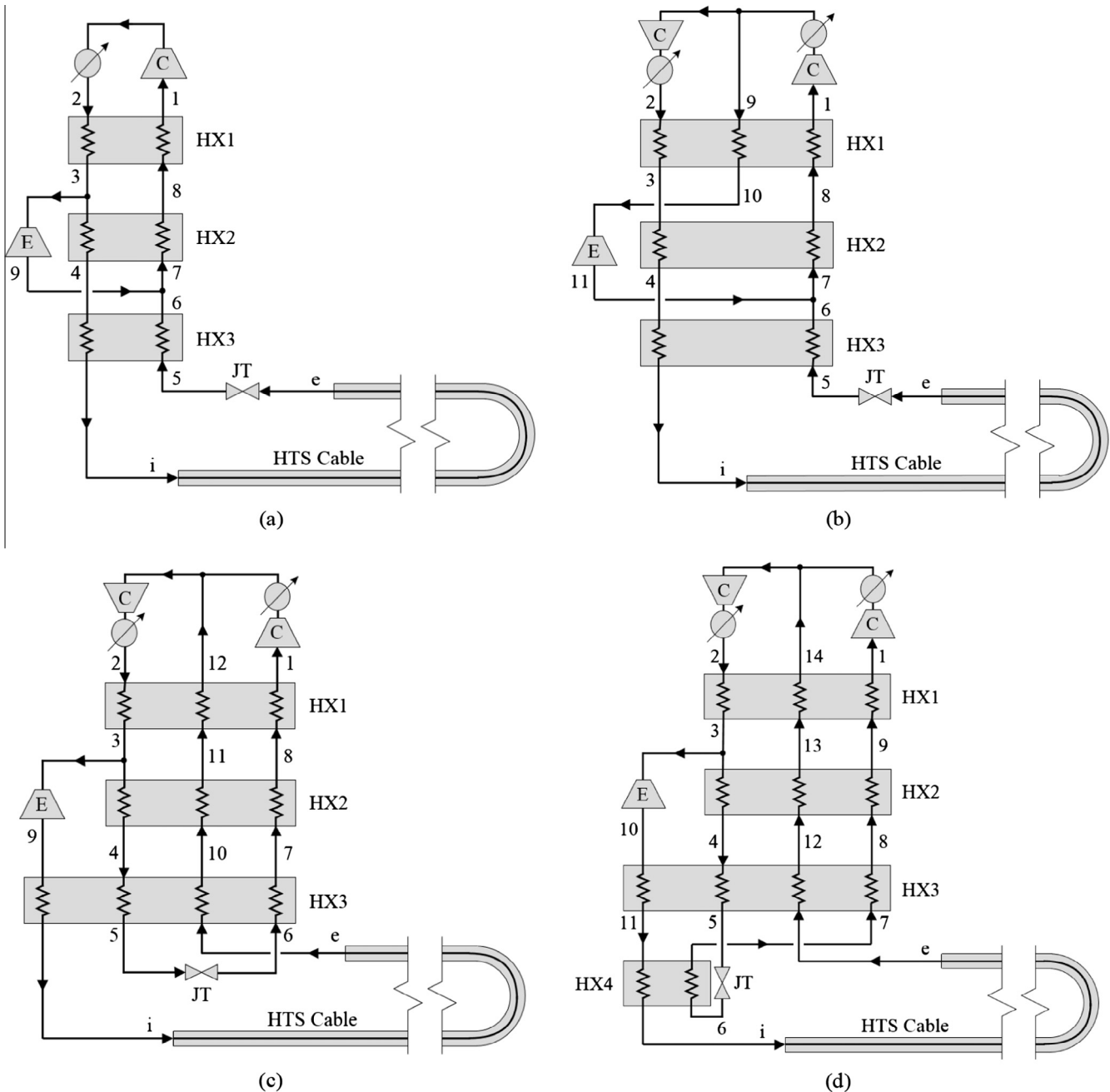


Fig. 3. Standard or modified Claude cycles proposed for integrated design. (a) Standard cycle cooled with JT stream (Cycle I), (b) dual-pressure cycle cooled with JT stream (Cycle II), (c) dual-pressure cycle cooled with expander stream (Cycle III), (d) modified dual-pressure cycle cooled with expander stream (Cycle IV).

Third requirement is the lower limit of temperature, as indicated by the left vertical line (denoted by “Freezing Limit”). The possibility of LN freeze-out is a significant safety issue, when the coldest temperature of refrigerant (helium or neon) is lower than the freezing temperature of LN (63.4 K). Even though several anti-freezing schemes have been developed about sub-cooling heat exchangers [19,20], a safety margin should be secured above the freezing temperature (for example, 65 K or higher). The last requirement is the “Pressure Limit”, as indicated by the horizontal line at top. This is the maximum allowable pressure, which should be determined by the mechanical strength of cable cryostat (for example, 1.2 MPa), depending on the material and dimension of cryostat wall and the manufacturing process.

As mentioned above, the LN flow in HTS cables is represented by a descending curve in phase diagram, whose slope is determined by the relative magnitude of temperature rise and pressure drop of the flow [21–23]. In order to fairly compare the proposed cycles by the same criteria, a specified condition of cooling requirement is defined here, as indicated by the arrow in the trapezoidal area. It will be assumed in this study that the LN at inlet and exit are at (69 K, 0.65 MPa), and (78 K, 0.5 MPa), respectively, which are securely fitted inside the required region.

3.2. Performance index

For a closed refrigeration cycle for HTS cable, the energy and entropy balance equations are written as

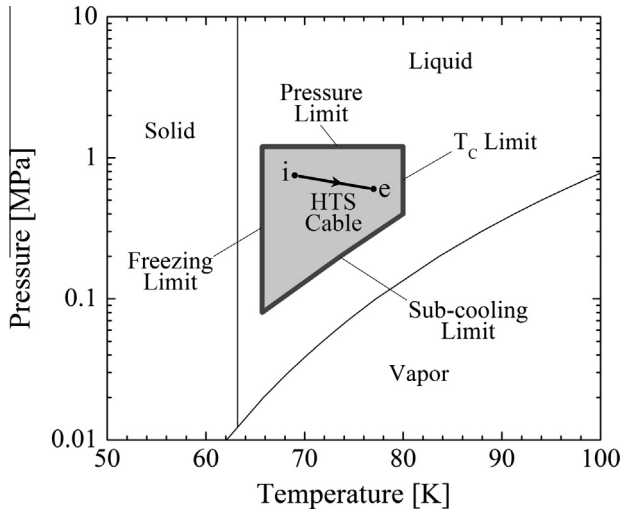


Fig. 4. Cooling requirements on phase diagram of liquid nitrogen for HTS cables.

$$\dot{Q}_o = (\dot{W}_C - \dot{W}_E) + \dot{Q}_{HTS} \quad (1)$$

$$\frac{\dot{Q}_o}{T_0} = \frac{\dot{Q}_{HTS}}{T_{HTS}} + \dot{S}_{gen} \quad (2)$$

where \dot{Q}_{HTS} and \dot{Q}_o are the thermal load from HTS cable at T_{HTS} and the heat rejected to ambient at T_0 , respectively, and \dot{W}_C and \dot{W}_E are the power input to compressors and the power output from expanders, respectively. \dot{S}_{gen} is the entropy generation rate in entire system, which can be itemized for individual components, such as compressors (C), expanders (E), heat exchangers (HX), JT valves (JT), and mixing devices (MIX).

By combining Eqs. (1) and (2), the net input power can be expressed as

$$\dot{W}_C - \dot{W}_E = \dot{Q}_{HTS} \left(\frac{T_0}{T_{HTS}} - 1 \right) + T_0 \dot{S}_{gen} \quad (3)$$

where the first term is the theoretical minimum of input power that can be obtained when $\dot{S}_{gen} = 0$. Eq. (3) is called the exergy (or availability) balance equation, since $\dot{W}_C - \dot{W}_E$ and the minimum input power are regarded as the exergy input and the exergy output, respectively, and their difference is the exergy loss (or exergy destruction) due to the thermodynamic irreversibility [24]. A dimensionless index to evaluate the thermodynamic performance is the FOM (figure of merit, or percentage of Carnot) [17,24], defined as the ratio of minimum to actual power input

$$FOM = \frac{\dot{Q}_{HTS}(T_0/T_{HTS} - 1)}{\dot{W}_C - \dot{W}_E} \quad (4)$$

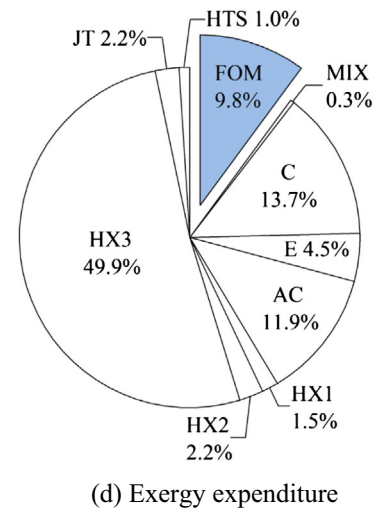
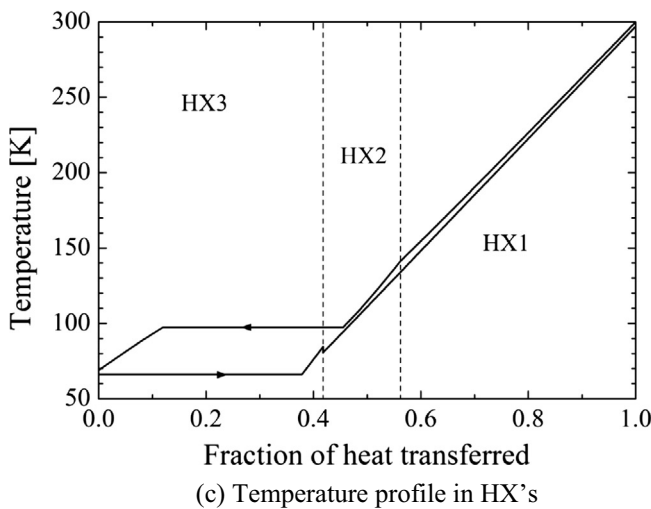
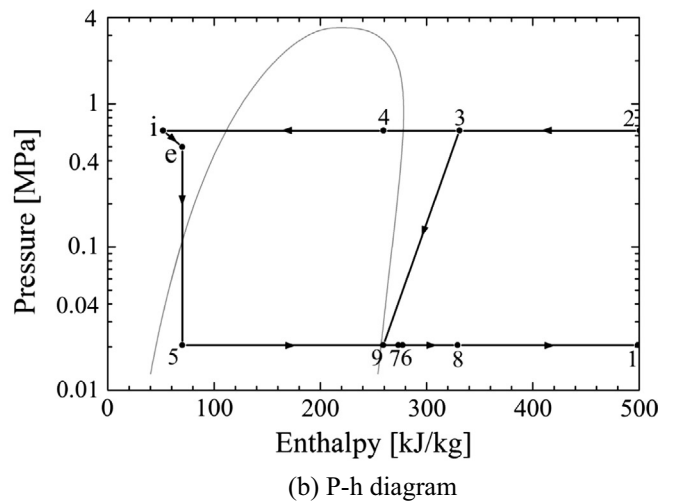
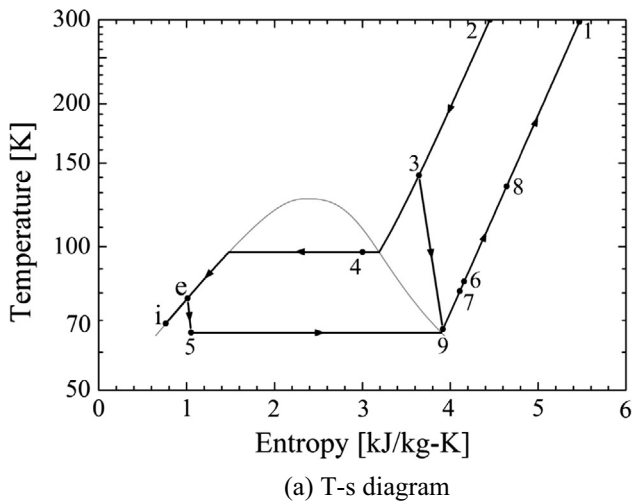


Fig. 5. Optimized standard cycle cooled with JT stream (Cycle I).

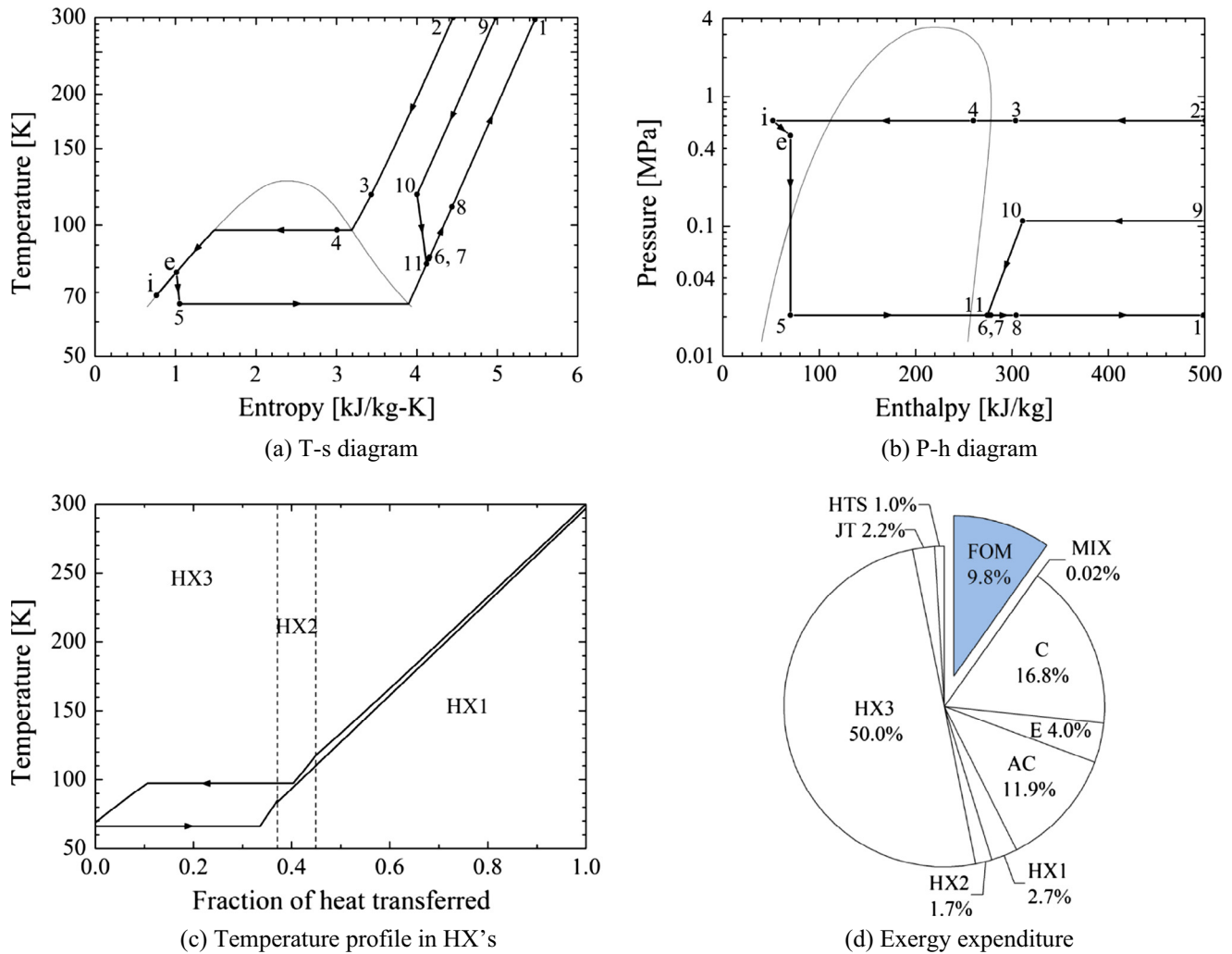


Fig. 6. Optimized dual-pressure cycle cooled with JT stream (Cycle II).

3.3. Assumptions and analysis model

The following simplifying assumptions are made for the cycle analysis and optimization.

- ① The thermal load at HTS cable is 10 kW at 78 K.

$$\dot{Q}_{HTS} = 10 \text{ kW} \quad (5)$$

- ② Liquid nitrogen (LN) enters the HTS cable at (69 K, 0.65 MPa), and exits at (78 K, 0.5 MPa). It follows that the temperature rise and pressure drop through HTS cables are 9 K and 0.15 MPa, respectively.

$$T_e - T_i = (78 \text{ K}) - (69 \text{ K}) = 9 \text{ K} \quad (6)$$

$$P_i - P_e = (0.65 \text{ MPa}) - (0.5 \text{ MPa}) = 0.15 \text{ MPa} \quad (7)$$

- ③ The ambient temperature is 300 K.
 ④ The adiabatic efficiency of all compressors and expanders is 80%.
 ⑤ The minimum temperature difference between hot and cold streams is 3 K in all heat exchangers.

$$\Delta T_{\min} = 3 \text{ K} \quad (8)$$

- ⑥ The pressure drop in all heat exchangers and after-coolers is zero.

A general-purpose process simulator (Aspen HYSYS) is used for cycle analysis. Thermodynamic properties of nitrogen are calculated with the equation of state linked to the simulator. Assumptions ① and ② are made with reference to the 154 kV HTS cables under development and the cooling capacity of Brayton refrigerator for future 1–3 km cables [14,15]. The input power to compressors is calculated with Assumption ④ and an additional assumption that the compression process is multi-staged with inter-cooling such that the pressure ratio at each stage is in the range of 2–2.5 and the intermediate pressure levels are optimally determined so as to minimize the total power. It is noted that Assumption ⑥ is taken for simplicity, but the pressure drop of LN flow through long HTS cable is taken into account as a source of thermodynamic irreversibility with Eq. (7).

4. Results and discussion

Four proposed cycles are quantitatively analyzed on the basis and assumptions given in the previous section. In Cycle I, the num-

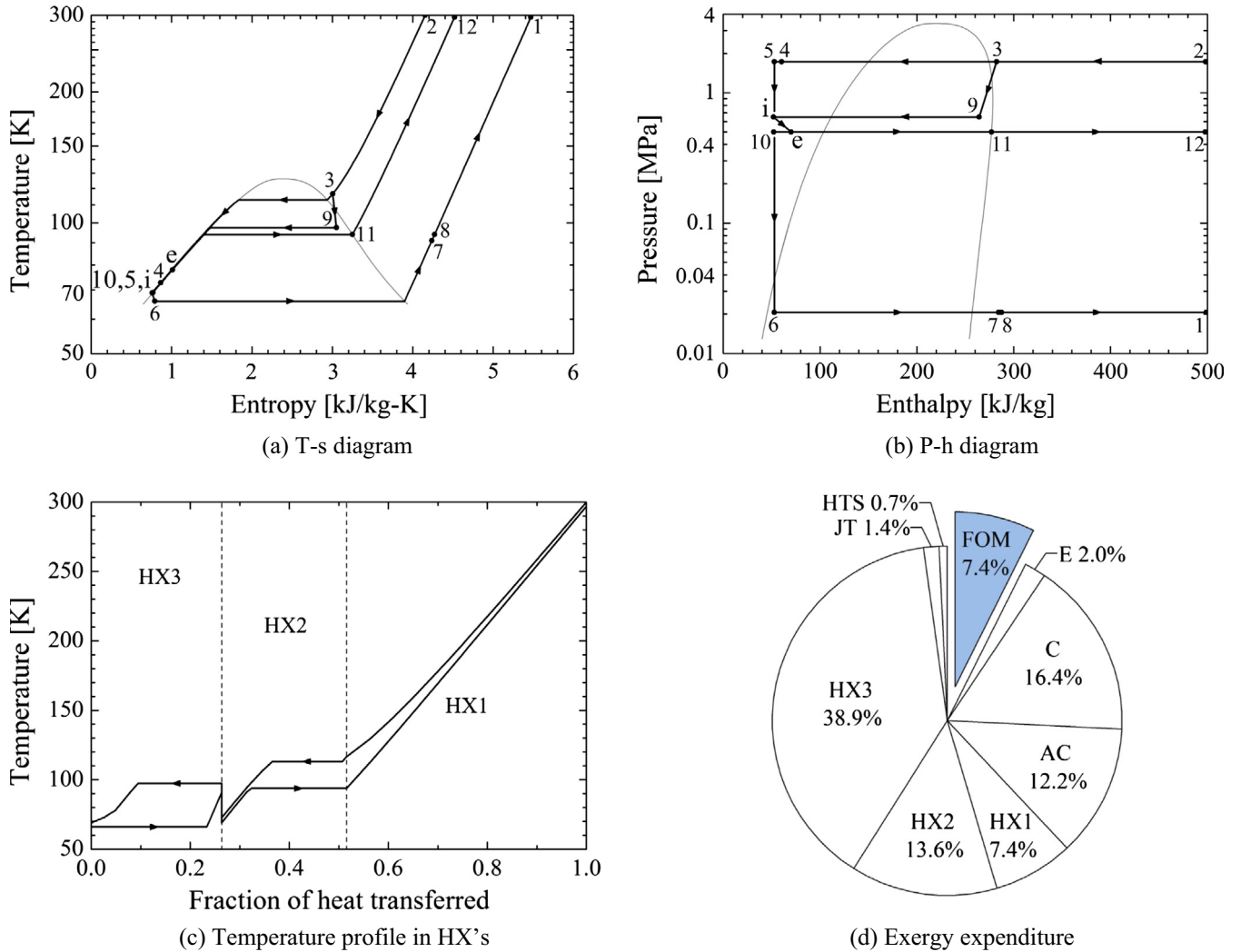


Fig. 7. Optimized dual-pressure cycle cooled with expander stream (Cycle III).

ber of unknowns is 10 (P_L , T_1 , T_3 , T_4 or x_4 , T_5 or x_5 , T_6 , T_7 , T_8 , T_9 , \dot{m}_E), because three states (2, i, e) are fixed, and \dot{m}_{HTS} and P_H are determined by Assumptions ① and ②. A key variable in Claude cycle is the “expander flow ratio” [17], which is defined as the ratio of expander (E) flow to compressor (C) flow.

$$y = \frac{\dot{m}_E}{\dot{m}_C} = \frac{\dot{m}_E}{\dot{m}_E + \dot{m}_{HTS}} \quad (9)$$

since $\dot{m}_C = \dot{m}_E + \dot{m}_{JT}$ and $\dot{m}_{JT} = \dot{m}_{HTS}$. On the other hand, the number of given equations is 9 (energy balance for 3 HX's + 1 JT + 1 MIX, 3 minimum ΔT 's for 3 HX's, 1 efficiency for E). For this cycle analysis, there is thus only one independent variable, which is selected as the expander flow ratio (y). This means that once y is given, the cycle is uniquely determined and the FOM can be calculated accordingly. After repeated analyses with different values of y , it is found that the maximum FOM is 9.8% when $y = 0.22$. The optimized cycle is plotted on temperature–entropy (T - s) and pressure–enthalpy (P - h) diagrams (excluding the compression and after-cooling processes) in Fig. 5(a) and (b), respectively. The P - h diagram is useful here, as the liquid processes (including the states i and e) are clearly displayed. In order to confirm Assumption ⑤, the temperature profile in HX's is also plotted in Fig. 5(c). The exergy expenditure is also plotted in Fig. 5(d) to compare the useful effect (i.e. FOM) with the irreversibility ratio of each component according to Eqs. (3) and (4).

The main source of irreversibility is the large temperature difference between the condensing and evaporating streams in HX3.

In Cycle II, the number of unknowns is 12 (P_L , P_L , T_1 , T_3 , T_4 or x_4 , T_5 or x_5 , T_6 , T_7 , T_8 , T_{10} , T_{11} , \dot{m}_E), and the number of given equations is 10 (energy balance for 3 HX's + 1 JT + 1 MIX, 4 minimum ΔT 's for 3 HX's, 1 efficiency for E). It is noted that 2 equations are counted as the minimum ΔT for HX1, since HX1 is a triple-stream heat exchanger. For this cycle analysis, there are two independent variables, which are selected as y and P_i (the intermediate pressure). The optimization of this cycle needs more calculations, as the FOM should be maximized in two-dimensional way. It is found that the maximum FOM is 9.8% when $y = 0.37$ and $P_i = 0.11$ MPa. The optimized cycle is presented in Fig. 6(a)–(d) in the same format as Fig. 5. The amount of gain in FOM with the dual-pressure cycle is insignificant, because the irreversibility in the cold heat exchanger (HX3) is still dominant. When compared with Cycle I, the expander stream has a larger flow rate and a smaller pressure ratio, but the overall thermodynamic performance is basically the same.

In Cycle III, the number of unknowns is 13 (P_L , T_1 , T_3 , T_4 , T_5 , T_6 or x_6 , T_7 , T_8 , T_9 or x_9 , T_{10} , T_{11} , T_{12} , \dot{m}_C), and the number of given equations is 12 (energy balance for 3 HX's + 1 JT, 7 minimum ΔT 's for 3 HX's, 1 efficiency for E). The minimum ΔT equations are counted as 2, 2, and 3 for HX1 (triple-stream), HX2 (triple-stream), and HX3 (quadruple-stream), respectively. For this cycle analysis, there is

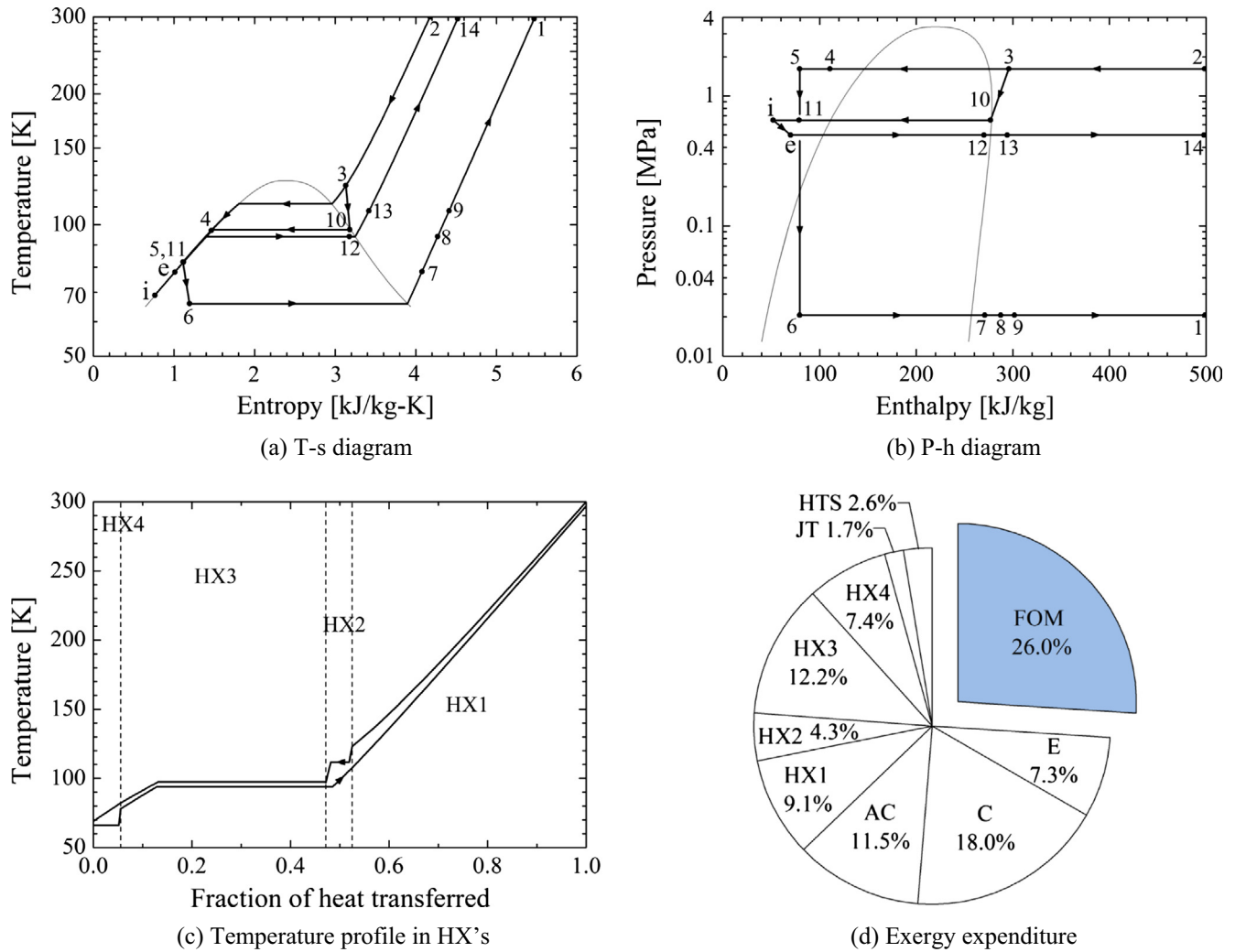


Fig. 8. Optimized modified dual-pressure cycle cooled with expander stream (Cycle IV).

thus only one independent variable like Cycle I, but the expander flow ratio is defined here as

$$y = \frac{\dot{m}_E}{\dot{m}_C} = \frac{\dot{m}_{HTS}}{\dot{m}_C} \quad (10)$$

since $\dot{m}_E = \dot{m}_{HTS}$. It is found that the maximum FOM is 7.4% when $y = 0.49$, as the optimized cycle is presented in Fig. 7(a)–(d). Cycle III has a lower FOM than Cycle I and Cycle II, mainly because the expander stream should be further cooled (to liquid) before the supply to HTS cable, resulting in more irreversibility in two cold heat exchangers (HX2 and HX3).

In Cycle IV, the number of unknowns is 15 ($P_L, T_1, T_3, T_4, T_5, T_6$ or $x_6, T_7, T_8, T_9, T_{10}, T_{11}, T_{12}, T_{13}, T_{14}, \dot{m}_C$), and the number of given equations is 14 (energy balance for 4 HX's + 1 JT, 8 minimum ΔT 's for 4 HX's, 1 efficiency for E). The minimum ΔT equations are counted as 2, 2, 3, and 1 for HX1, HX2, HX3, and, HX4, respectively. For this cycle analysis, there is only one independent variable, and the expander flow ratio (y) is defined also as Eq. (10). It is found that the maximum FOM is 26.0% when $y = 0.88$, as the optimized cycle is presented in Fig. 8(a)–(d). Cycle IV has a remarkably higher FOM than other cycles, since the temperature difference in HX3 is significantly reduced with the addition of HX4. It should be carefully observed that the condensing temperature of expander stream is closely matched with the evaporating temperature of JT stream, as shown in Fig. 8(c). A similar effect of small

temperature difference between phase-changing streams can be found, for example, in the parallel combination of two pre-cooling JT cycles for the natural gas liquefaction with mixed-refrigerant, as presented (denoted by b1J/e1J + M2J) in the recent review paper [25].

In summary, Fig. 9 graphically compares the four proposed cycles in terms of (a) FOM, (b) mass flow rate, and (c) heat multiplied by the number of streams. FOM is the key thermodynamic index to determine the power input (i.e. the operation cost), while two others are considered the indexes that are closely related with the capital cost of refrigeration system. In Fig. 9(b), the height of columns for total mass flow is stacked as sum for two streams ($\dot{m}_C = \dot{m}_E + \dot{m}_{JT}$) in reference to the capacity of compressors and expanders. In Fig. 9(c), the heat multiplied by the number of streams can be considered as an index for the estimated size of HX's [26]. The overall height of columns is also represented as stacked sum for respective HX's.

Cycle IV is dominantly superior to other cycles from the viewpoint of power consumption. In Cycle IV, the total flow rate is also the smallest, while the expander flow rate is the largest. It can be noted that the flow rate of JT stream in Cycle I and Cycle II is the same as the flow rate of expander stream in Cycle III and Cycle IV. For the requirements on HX's, Cycle I is expected to be the simplest and smallest, while Cycle II and Cycle IV have the same order of magnitude. It is recalled that the dual-pressure cycles (Cycle II,

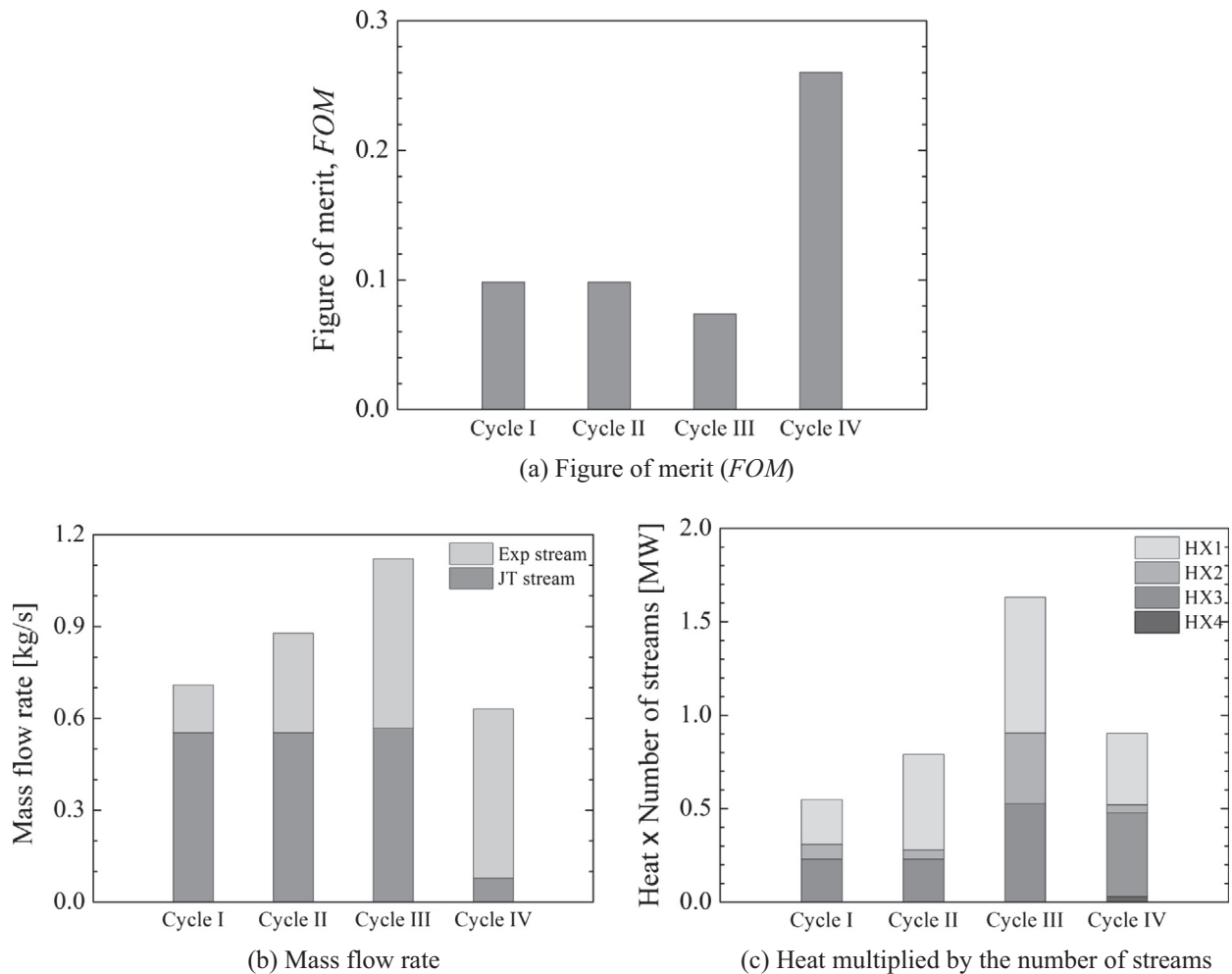


Fig. 9. Comparison of four proposed cycles.

Table 1
Temperature, pressure, vapor fraction, and mass flow rate for optimized Cycle IV.

	1	2	3	4	5	6	7	
Temperature (K)	297.0	300.0	123.0	97.0	82.2	66.0	78.2	
Pressure (MPa)	0.021	1.610	1.610	1.610	1.610	0.021	0.021	
Vapor fraction	1.000	1.000	1.000	0.000	0.000	0.160	1.000	
Flow rate (kg/s)	0.078	0.631	0.631	0.078	0.078	0.078	0.078	
8	9	10	11	<i>i</i>	<i>e</i>	12	13	14
94.0	107.6	97.5	82.2	69.0	78.0	94.0	107.6	297.0
0.021	0.021	0.650	0.650	0.650	0.500	0.500	0.500	0.500
1.000	1.000	1.000	0.000	0.000	0.000	0.961	1.000	1.000
0.078	0.078	0.553	0.553	0.553	0.553	0.553	0.553	0.553

Cycle III, and Cycle IV) have advantages over the standard cycle (Cycle I) in flexibility in the selection of compressors and expanders. It is noticeable in Cycle IV that the expected size of HX4 is quite small in spite of its crucial role in improving the thermodynamic performance, as mentioned above. Overall, the modified dual-pressure cycle cooled with expander stream (Cycle IV) is primarily recommended as the integrated system of refrigerator and LN circulation loop for long HTS cables. The detailed thermodynamic data for an optimized Cycle IV under the specified conditions are listed in Table 1. The supply (*i*) and return (*e*) conditions for HTS cables are highlighted with shaded background.

There will be a number of technical issues in realizing the proposed cycle for a practical HTS cable system. Some aspects are

briefly mentioned here, even though the details are beyond the scope of this thermodynamic study. First, sub-atmospheric operation is required for the proposed cycle. Since it was assumed in the analysis that the coldest temperature is 66 K (for the LN supply to HTS cable at 69 K), the corresponding P_L of nitrogen should be 21 kPa, as listed in Table 1. This sub-atmospheric system with nitrogen should be similar in structure, for example, with the helium refrigeration systems at temperatures below 4 K [27]. The required P_L can be set at a higher level, if the operation region shown in Fig. 4 is shifted to the upper and right direction, and the LN flow rate is large enough for the descending curve (from *i* to *e*) to be steep. In spite of the hardware complexity, the sub-atmospheric system may have an advantage that the possibility of LN freeze-out is fully removed.

Another question on the proposed cycle is the practical effectiveness of multiple-stream heat exchangers, as Cycle IV is composed of triple-stream and quadruple-stream HX's. It was assumed in this study that all hot streams have the same temperature and all cold streams have the same temperature at an axial location, as only two composite curves were plotted in Fig. 8(c). In the author's previous work, it was reported that the calculated *FOM* on the assumption is always an over-estimate, but can be closed achieved with a proper heat-exchanger design [28]. The elaboration with contemporary technology on plate-fin heat exchangers (PFHX's) will be essential to the realization of the proposed system for long HTS cables.

Finally, it should be mentioned that the proposed system will consume more compressor power than the existing pump-circulation systems. Since the power input to a cryogenic pump is much smaller by itself than the compressor power, the elimination of cryogenic pumps must accompany the penalty in power consumption. For a quantitative comparison, an ideal cooling system with Brayton refrigerator and LN pump shown in Fig. 1(c) can be simulated on the same assumptions above, and the estimated FOM is 34.5%, which is even higher than the case of optimized Cycle IV. Once again, the obvious merits of this proposed system are found in the reliability and investment cost of total system over the operational cost.

5. Conclusions

In accordance with the ongoing development of long-length HTS cables, a thermodynamic study is executed for the cryogenic refrigeration cycles that can be integrated with the liquid-nitrogen circulation loop. The integrated system could be configured in several ways with standard or modified Claude cycles such that the LN circulation loop along HTS cables is included as part of the cycle. It is concluded that the modified dual-pressure Claude cycle cooled by expander stream (Cycle IV) is primarily recommended for high efficiency and hardware requirements. Next steps works towards patent application and practical system design are underway.

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