



Development of Refrigeration and Air-Conditioning Technologies for
Practical Use of Next-Generation Low-GWP Refrigerants

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Refrigerants

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1. Research and Development Overview and Outcomes (Waseda University)

1.1 Overall Objectives of the Project

The objective of this project was to develop technologies capable of accurately evaluating the performance of equipment employing next-generation refrigerants through both theoretical analysis and experimental validation. Specifically, the project pursued two main goals:

- ① To develop general-purpose heat exchanger and refrigeration cycle simulators applicable not only to existing refrigerants but also to a wide variety of newly developed next-generation refrigerants.
- ② To establish internationally applicable performance evaluation criteria and methods based on the simulators and performance evaluation apparatus developed in FY2022.

To achieve these goals, the research and development effort was divided into the following four items:

(1) Eval. A-1-1: Research and Development of Simulation Technology

A general-purpose refrigeration cycle simulator was designed to accommodate both existing and newly developed refrigerants.

(2) Eval. A-1-2: Research and Development of Optimization Technology

A heat exchanger simulator was created to analyze refrigerant distribution paths and optimize heat exchanger performance for each refrigerant.

(3) Eval. A-1-3: Research and Development of Refrigeration Cycle System Evaluation Technology

Experimental validation was conducted using an “optimization verification device” to confirm the accuracy of simulation results from the heat exchanger and refrigeration cycle simulators.

(4) Eval. A-2: Development of an AI Optimization Algorithm for the Refrigeration Cycle Simulator (Subcontracted)

An algorithm was developed to identify optimal refrigerant distribution paths for each refrigerant within the simulator framework.

A major research objective was to establish a system evaluation platform enabling fair and highly accurate performance comparisons of various refrigerants, including low-GWP blends, at the equipment level—something difficult to achieve with conventional approaches. For example, direct “drop-in” evaluations (e.g., applying R454C to a heat exchanger originally designed for R32) cannot properly assess the intrinsic performance of a refrigerant.

Accordingly, the basic policy of this study was to evaluate refrigerant performance under heat exchanger conditions optimized for each refrigerant. Full optimization of physical dimensions and structural design was excluded, as the number of design parameter combinations would be impractically large for a university-scale project. Instead, “optimal heat exchanger” was defined as optimization of the refrigerant flow path while keeping the fundamental structure (size, number of rows) fixed. This approach balances practicality with potential for standardization. Discussions with the Japan Refrigeration and Air Conditioning Industry Association provided a certain degree of industry recognition of this evaluation concept.

In this study, refrigerant paths were optimized using the heat exchanger simulator, and refrigeration cycle simulations were then performed based on these results to compare refrigerant performance. These efforts were not intended to verify optimization methods for the heat exchanger design itself, but rather to establish an evaluation platform that enables fair performance comparisons among refrigerants.

Figure 1.1-1 presents a conceptual diagram of the overall structure of the research and development conducted in this study.

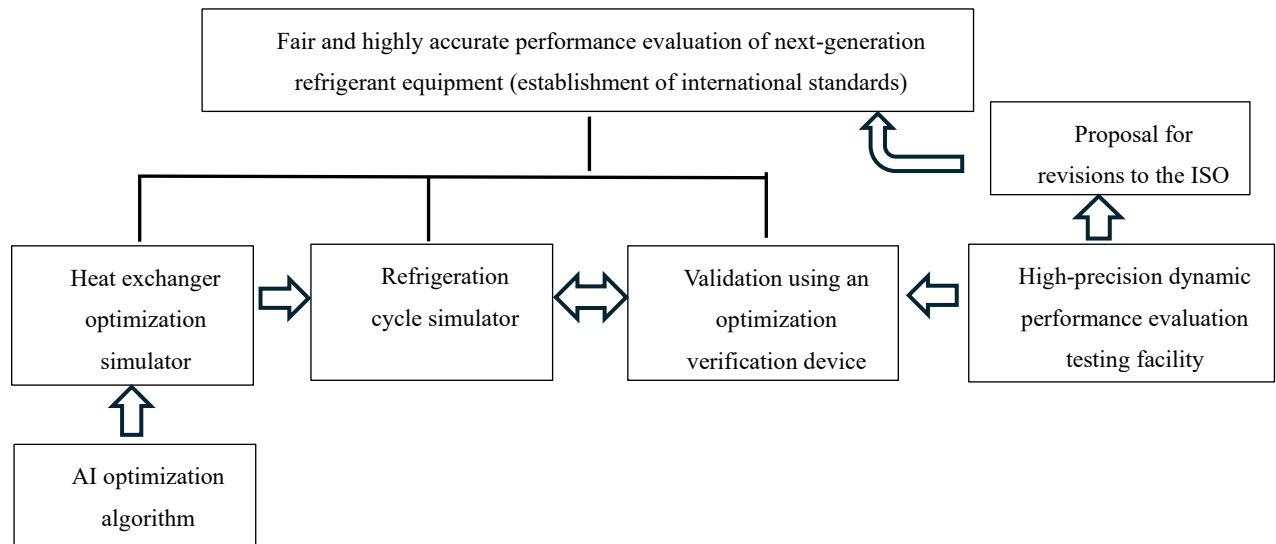


Figure 0-1: Overall structure of the research and development conducted in this study

1.2 Research Outcomes

(1) Eval. A-1-1: Research and Development of Simulation Technology

By incorporating detailed device models, including an accumulator model, the simulator reproduced refrigerant charge and cycle behavior with high accuracy. A general-purpose analytical platform applicable to both pure and zeotropic mixed refrigerants was realized. As a result, a simulator capable of predicting equipment-level performance for conventional refrigerants with an error of approximately 5% was established.

(2) Eval. A-1-2: Research and Development of Optimization Technology

Using a genetic algorithm, a detailed heat exchanger simulator for evaporators and condensers was developed. The simulator optimizes design elements such as branching structure (flow distribution and pressure-loss control) and circuit arrangement (temperature and phase distributions), while accounting for conditions feasible in actual equipment. Consequently, optimal circuit structures—such as cross-parallel and cross-counter configurations—can be derived automatically, enabling designs that reflect the temperature-glide characteristics of zeotropic mixed refrigerants.

(3) Eval. A-1-3: Research and Development of Refrigeration Cycle System Evaluation Technology

To confirm applicability to conventional refrigerants, experimental verification using the optimization verification device was conducted for R32 and R454C. The simulation and experimental results agreed closely, thereby validating the optimization method. Performance improvements were demonstrated: evaporator optimization increased COP by up to 20%, condenser optimization by up to 11%, and simultaneous evaporator and condenser optimization by up to 23%. The optimal solution varied with operating conditions, clarifying that heat exchanger optimization can improve the performance of R454C to a level close to that of R32. Furthermore, both evaporator and condenser characteristics are critical for zeotropic mixed refrigerants. Based on these results, an integrated method was established for achieving fair comparative performance evaluation among refrigerants, which had previously been difficult.

Extending this to LCCP evaluation, results from a 4 kW air-conditioning unit showed that heat exchanger optimization can reduce the LCCP of R454C to nearly the same level as that of R32. This demonstrates the possibility of simultaneously achieving both environmental and energy performance for low-GWP refrigerants.

(4) Eval. A-2: Development of an AI Optimization Algorithm for the Refrigeration Cycle Simulator

A practical algorithm was created to handle more than 2,000 combinations of constraint conditions by modeling the

pipng structure as a directed acyclic graph (DAG) and incorporating manufacturing constraints. The implemented functions include constraint-embedded optimization, an automatic repair mechanism, acceleration through evaluation caching, improved search efficiency through structural duplicate elimination, and enhanced local searches via a swap function. These improvements enabled high-speed optimization restricted to manufacturable designs, significantly increasing search efficiency compared with conventional methods. The algorithm made it possible to quantitatively and automatically optimize heat exchanger circuit design, which had previously relied heavily on designer experience.

Together, the outcomes of (1)–(4) established an analytical and verification platform for fair comparative evaluation of refrigerant performance under conditions close to actual equipment operation, using conventional refrigerants as benchmarks. These outcomes constitute a foundational technology for future equipment development with next-generation refrigerants and for advancing performance evaluation criteria. In particular, the concept established in this study —namely, evaluating performance under conditions optimized for each refrigerant—will serve as an important guiding principle in the future development of international performance evaluation standards.

1.3 Spillover Effects and Future Prospects

This research and development effort has enabled the following achievements:

- (1) Advancement of design guidelines for next-generation refrigerant equipment
- (2) Contribution to the standardization of international performance evaluation criteria
- (3) Acceleration of the social implementation of low-GWP refrigerants

These achievements directly contribute to improving the reliability of equipment performance evaluations during the transition to new refrigerants.

1.4 Awards and Honors

At the 35th International Symposium on Transport Phenomena (ISTP35), held in Busan, Republic of Korea, on September 21–24, 2025, the presentation entitled “Evaporator Circuitry Optimization in a Three-dimensional Case” received the Best Presentation Award.

2. Research and Development on Performance Evaluation of Refrigeration Cycles Employing Next-Generation Refrigerants (Waseda University)

2.1 Research and Development of Refrigeration Cycle Simulation Technology for Actual Equipment

2.1.1 Accumulator model

The refrigerant entering the accumulator undergoes gas–liquid separation. Most of the liquid refrigerant is separated and stored in the accumulator, while a portion flows into the outlet pipe together with the refrigeration oil through the oil-return hole provided in the outlet pipe (Figure 2.1-1).

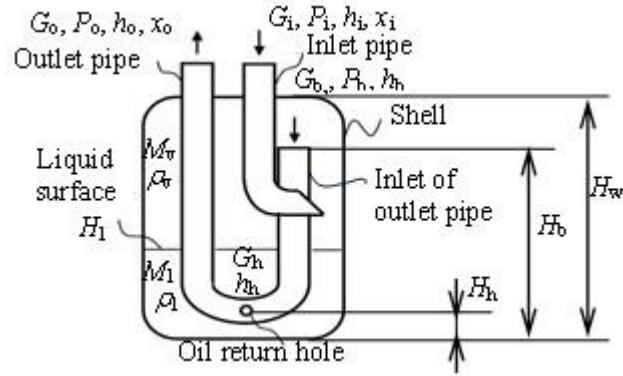


Figure 0-1: Schematic diagram of the accumulator

The calculation model developed for the accumulator is presented below.

$$\Delta P_{im} = k_{im} \left(1 - \frac{S_i}{A_y} \right)^2 \frac{\rho_i}{2} \left(\frac{G_i}{\rho_i S_i} \right)^2 \quad (2.1-1)$$

$$\Delta P_p = \rho_l g (H_1 - H_h) \quad (2.1-2)$$

$$G_h = \rho_l C_h S_h \sqrt{\frac{2(\Delta P_{bc} + \Delta P_{bf} + \Delta P_d + \Delta P_p)}{\rho_l}} \quad (\text{if } H_1 \leq H_h \rightarrow G_h = 0) \quad (2.1-3)$$

$$\frac{dM_l}{dt} = \phi G_i (1 - x_i) - G_h - G_i \frac{h_i - h_{iv}}{h_{iv} - h_{il}} - \frac{A_{sl} K \Delta T}{h_{iv} - h_{il}} \quad (\text{if } h_i < h_{iv} \rightarrow h_i - h_{iv} = 0) \quad (2.1-4)$$

$$H_1 = M_l / (\rho_l A_y) \quad (2.1-5)$$

(if $H_1 > H_b \rightarrow H_1 = H_b$, The amount exceeding H_b flows into the outlet pipe)

$$M_v / \rho_v + M_l / \rho_l = V \quad (2.1-6)$$

$$\phi = (G_{il} - G_{bl}) / G_{il} \quad (2.1-7)$$

$$G_{bv} = G_i x_i, \quad G_{bl} = (1 - \phi) G_i (1 - x_i), \quad G_b = G_{bv} + G_{bl}, \quad G_o = G_b + G_h, \quad M = M_v + M_l \quad (2.1-8)$$

$$x_o = \left[G_o - \{ G_h + (1 - \phi) G_i (1 - x_i) \} \right] / G_o \quad (\text{if } H_1 > H_b \rightarrow x_o = x_i, \quad x_o = x_i \text{ in static calculation}) \quad (2.1-9)$$

$$\Delta P = (\Delta P_{if} + \Delta P_{im}) + (\Delta P_{bc} + \Delta P_{bf} + \Delta P_{of}) \quad (2.1-10)$$

$$h_o = h_{ol} + x_o (h_{ov} - h_{ol}) \quad (\text{in } H_I > 0), \quad h_o = h_i \quad (\text{in } H_I = 0 \text{ or in static calculation}) \quad (2.1-11)$$

Using an accumulator with an inner diameter of 130 mm and a height of 273.6 mm, the mass flow rate and the inlet quality were varied to identify the flow coefficient C_h of the oil-return hole. C_h was assumed to take the same value at a given flow rate. A comparison between the measured and simulated results at $0.011 \text{ kg} \cdot \text{s}^{-1}$ is shown in Figure 2.1-2. The experimental and simulation results agreed well. The identified values of the flow coefficient C_h for each mass flux are illustrated in Figure 2.1-3.

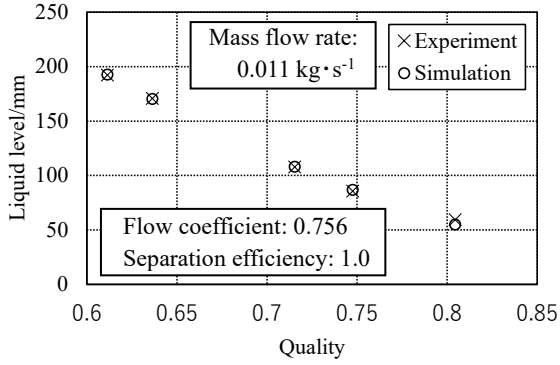


Figure 2.1-2: Comparison of liquid level results at a mass flow rate of $0.011 \text{ kg} \cdot \text{s}^{-1}$

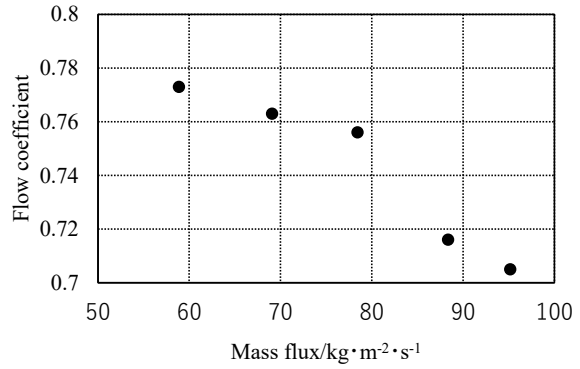
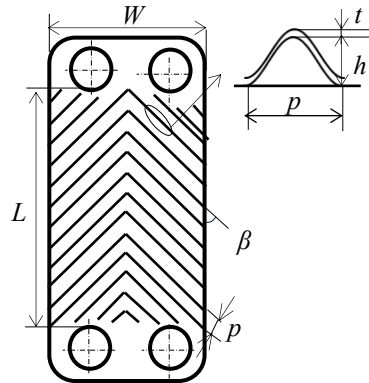


Figure 2.1-3: Relationship between mass flux and flow coefficient

(a) Plate heat exchanger model

A generalized model for the heat transfer coefficient and pressure loss of a plate heat exchanger was constructed. The generalized equations were formulated to incorporate the corrugation angle β and area enlargement factor ϕ . A schematic diagram of the plate heat exchanger is shown in Figure 2.1-4.



A	:	Area of heat transfer
D_{eq}	:	Equivalent diameter ($= 2 \cdot D_{eq}$)
h	:	Corrugation height
L	:	Length of corrugated section
n	:	Number of plates
p	:	Corrugation pitch
t	:	Thickness of plate
W	:	Width of corrugated section
β	:	Corrugation angle
δ	:	Plate spacing
ϕ	:	Area enlargement factor

Figure 0-2: Schematic diagram of the plate heat exchanger

For the single-phase heat transfer coefficient, the equation proposed by Martin (1996) was adopted.

$$Nu = \phi \cdot 0.122 \cdot Pr^{1/3} \cdot (\mu/\mu_{wall}) \cdot \left[\xi \cdot (Re/\phi)^2 \cdot \sin(2 \cdot \beta) \right]^{0.374} \quad (2.1-12)$$

$$\frac{1}{\sqrt{\xi}} = \frac{\cos \beta}{\sqrt{0.18 \cdot \tan \beta + 0.36 \cdot \sin \beta + \xi_0 / \cos \beta}} + \frac{1 - \cos \beta}{\sqrt{3.8 \cdot \xi_{1,0}}} \quad (2.1-13)$$

$$\xi_0 = 64 \cdot \phi / Re \quad (Re < 2000), \quad \xi_0 = \{1.8 \cdot \log_{10}(Re/\phi) - 1.5\}^{-2} \quad (Re \geq 2000) \quad (2.1-14)$$

$$\xi_{1,0} = (597 \cdot \phi / Re) + 3.85 \quad (Re < 2000), \quad \xi_{1,0} = 39 / \{(Re/\phi)^{0.289}\} \quad (Re \geq 2000) \quad (2.1-15)$$

For the single-phase friction loss, the equation proposed by Martin (1996) was also adopted. The parameters ξ , ξ_0 , and $\xi_{1,0}$ were calculated using the same expressions as those employed in Martin's single-phase heat transfer coefficient, and the pressure loss was calculated using the Fanning equation.

$$f = \phi \cdot \xi / 4 \quad (2.1-16)$$

For the condensation heat transfer coefficient, the correlation proposed by Okazaki and Manno (2006) was adopted. The parameters ξ , ξ_0 , and $\xi_{1,0}$ were calculated using the same expressions as those in Martin's equation. The corrugation angle β was expressed in rad. Re_{LO} was calculated by assuming that only liquid refrigerant flowed through the plate gap, using the liquid refrigerant properties and the total mass flux of the refrigerant.

$$Nu = \phi \cdot 1.172 \cdot Pr_L^{1/3} \cdot \left[\xi \cdot (Re_{LO}/\phi)^2 \cdot \sin(2 \cdot \beta) \right]^{0.28} \quad (2.1-17)$$

Because no existing correlation simultaneously accounts for both the corrugation angle β and the area enlargement factor ϕ for the evaporation heat transfer coefficient, a new correlation was constructed based on experimental results. Re_{Lm} was calculated using the average quality x_m between the inlet and outlet of the two-phase region, and the parameters ξ , ξ_0 , and $\xi_{1,0}$ were calculated using the same expressions as those in Martin's equation.

$$Nu = \phi \cdot 0.0290 \cdot Pr_L^{1/3} \cdot \left[\xi \cdot (Re_{Lm}/\phi)^2 \cdot \sin(2 \cdot \beta) \right]^{0.470} \quad (2.1-18)$$

For the two-phase friction loss, the equation proposed by Asano and Takenaka (2007) was adopted. The liquid single-phase friction factor f_L was calculated using Martin's equation.

$$\Delta P = 4 \cdot f_L \cdot \Phi_L^2 \cdot \left\{ \rho_L \cdot (G_L / \rho_L)^2 / 2 \right\} (L / D_{eq}) \quad (2.1-19)$$

$$\Phi_L^2 = 1 + C / X_{tt} + 1 / X_{tt}^2 \quad [C=3.14 \quad (\text{for upward flow}), \quad C=2.70 \quad (\text{for downward flow})] \quad (2.1-20)$$

Experiments were conducted using a plate heat exchanger with a length of 118.5 mm, a width of 62 mm, a corrugation angle of 60°, and 10 plates. Water was circulated on both the primary and secondary sides, and the single-phase Nusselt number and friction factor f were obtained (Figure 2.1-5 (i), (ii)). For comparison, the Muley and Manglik (1999) equation is also shown in the figure. R32 was circulated on the primary side, and the two-phase Nusselt number (Figure 2.1-5 (iii), (iv)) and the two-phase friction loss (Figure 2.1-5 (v)) were obtained.

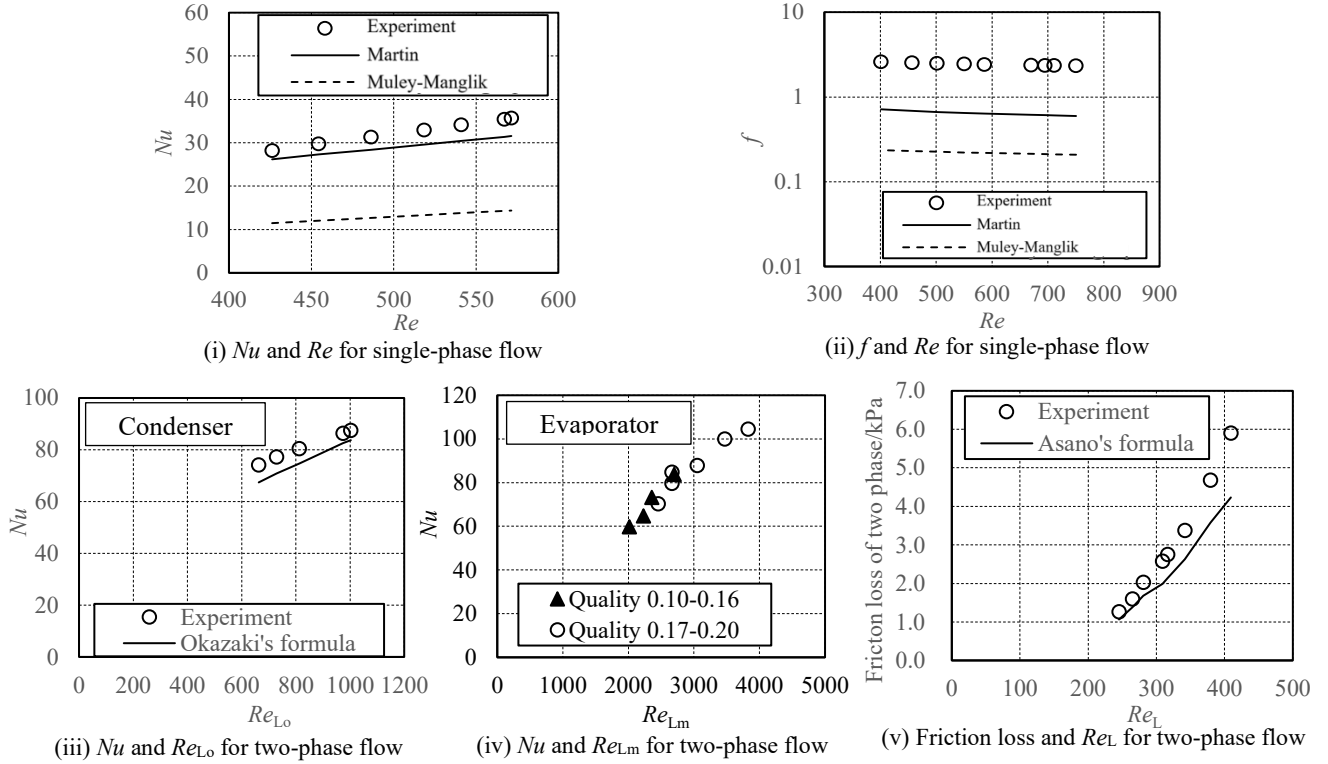


Figure 0-3: Relationship between Nusselt number and friction loss versus Reynolds number for single- and two-phase flows

(b) Refrigerant charge calculation model for the refrigeration cycle

The amount of refrigerant dissolved in refrigeration oil is expressed by the following equation:

$$M_{\text{ref}} = \frac{R}{1-R} M_{\text{oil}} = \frac{R}{1-R} (V_{\text{oil}} \cdot \rho_{\text{oil}}) \quad (2.1-21)$$

The solubility curves of refrigerants in refrigeration oil are proprietary to each company and are not disclosed externally. However, for the R22 refrigerant, mineral oil is used, and the data are publicly available (Figure 2.1-6). In the simulator, solubility was calculated from pressure and temperature based on the solubility curve data.

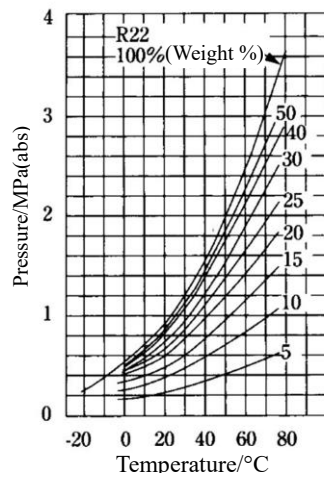


Figure 0-4: Sample solubility curve of a refrigerant in refrigeration oil

The newly developed equation for void fraction, together with a calculation example for R22, is displayed in Figure 2.1-7.

$$\alpha = \frac{U_{sg}}{C_0 \cdot U_M + U_{gm}} \quad (2.1-22)$$

$$C_0 = 1 + 0.12 \cdot (1 - x) \quad (2.1-23)$$

$$U_M = G \cdot \left(\frac{x}{\rho_{sg}} + \frac{1-x}{\rho_{sl}} \right) \quad (2.1-24)$$

$$U_{sg} = G \cdot \left(\frac{x}{\rho_{sg}} \right) \quad (2.1-25)$$

$$U_{gm} = K \cdot \frac{1.18 \cdot (1-x)}{\rho_{sl}^{0.5}} \left[g \cdot \sigma \cdot (\rho_{sl} - \rho_{sg}) \right]^{0.25} \quad (2.1-26)$$

$$K = 0.0178 \cdot \left\{ \left(\frac{1-x}{x} \right)^{0.9} \cdot \left(\frac{\rho_{sg}}{\rho_{sl}} \right)^{0.5} \right\}^{-0.736} \cdot Re^{0.493} \quad (\text{for grooved tube}) \quad (K = 1 \text{ for bare tube}) \quad (2.1-27)$$

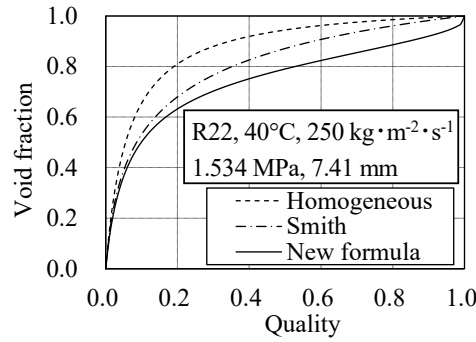


Figure 0-5: Relationship between quality and void fraction

A calculation example of the simulator for an R22 2.2 kW room air conditioner is shown below. The compressor was a high-pressure shell rotary type, and the amount of refrigeration oil and effective internal volume in the compressor were set to the values of the actual unit. The indoor heat exchanger had 2 rows, 10 stages, and a width of 550 mm, while the outdoor heat exchanger had 1 row, 20 stages, and a width of 750 mm. Table 2.1-1 presents the calculation results when the subcooling at the condenser outlet was 5 K, the compressor suction superheat was 5 K, and the evaporator capacity was 2.2 kW. In the refrigeration cycle model used for the calculation, the refrigerant charge inside the compressor was set to zero. It was confirmed that the calculated refrigerant charge became nearly equal to that of the actual unit when the slip ratio coefficient k in Smith's equation was set to 3; these results are presented as Case 1. Case 2 shows the results when the amount of refrigerant dissolved in the refrigeration oil was calculated using the solubility curve illustrated in Figure 2.1-7, together with the newly developed void fraction equation. The refrigerant charge was 0.6511 kg in Case 1 and 0.6517 kg in Case 2, indicating that the overall refrigerant charges in the two cases were nearly identical.

Table 0-1: Calculation results for refrigerant charge

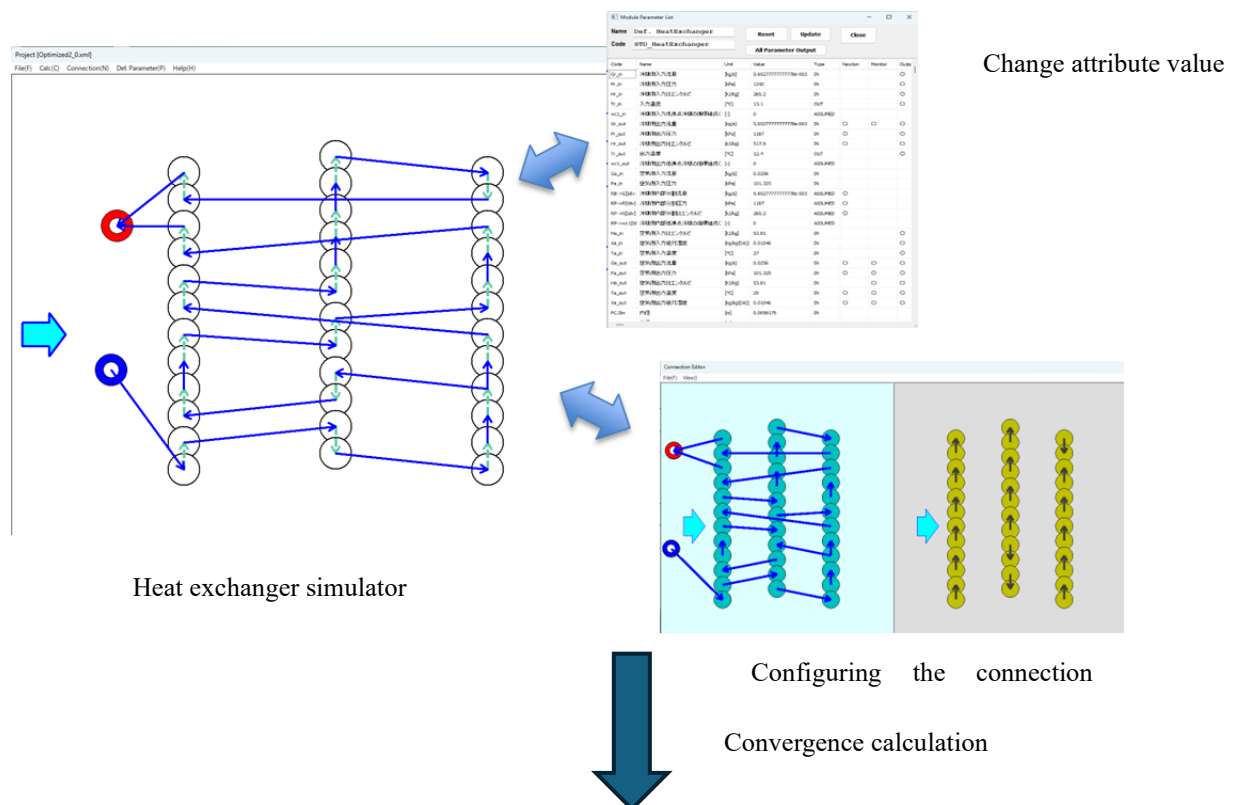
			Case 1	Case 2
Refrigerant charge		(kg)	0.6511	0.6517
	Condenser	(kg)	0.5063	0.4322
	Evaporator	(kg)	0.0833	0.0609
Distribution of Refrigerant amount	Compressor	(kg)	0.0	0.0970
	Discharge gas pipe	(kg)	0.0014	0.0014
	Two-phase pipe	(kg)	0.0458	0.0459
	Suction gas pipe	(kg)	0.0143	0.0143

(c) Development of the heat exchanger simulator

A heat exchanger simulator capable of analyzing the performance of various low-GWP refrigerants was developed. The main features of the simulator are as follows:

- A simulator reflecting the physical structure of the heat exchanger was constructed.
- The internal connection system of the heat exchanger can be freely configured, and physical structures such as fins can also be modified.
- The heat exchanger can be optimized independently by changing the connection system and attribute values.

An overview is shown in Figure 2.1-8.



2.2 Research and Development of Optimization Technology for the Refrigeration Cycle Simulator

2.2.1 Research outcomes

In this study, research was conducted on the optimization of refrigerant circuitry in heat exchangers for air-conditioning equipment, and the following outcomes were obtained:

- (1) A heat exchanger simulator for evaporators and condensers was constructed, and a performance evaluation method for pure and zeotropic mixed refrigerants was established.
- (2) A circuitry optimization method using a genetic algorithm was developed, clarifying the effects of branching structure and flow path arrangement on heat exchanger performance.
- (3) Studies targeting R32 and R454C confirmed that the proposed optimization method is effective for actual heat exchanger design.

2.2.2 Heat exchanger simulator

This study aimed to construct a simulation method for evaluating and optimizing refrigerant circuitry design in heat exchangers with high accuracy to realize high-efficiency air conditioners using low-GWP refrigerants.

After defining the refrigerant circuitry to be evaluated, the refrigeration cycle performance under specified operating conditions was calculated using independent simulators for the evaporator and condenser (Figure 2.2-1). In this simulator, the overall heat exchanger performance was obtained by iteratively calculating the local heat and mass balances in each tube within the heat exchanger. The calculations employed a combination of correlations for refrigerant-side and air-side heat transfer coefficients and pressure losses.

Furthermore, to appropriately reproduce the characteristics of zeotropic mixtures, transport-property correlations for mixed refrigerants were introduced in both the evaporator and condenser simulators.

In the evaporator simulator, the two-phase heat transfer of zeotropic mixed refrigerants was modeled using the correlation of Shah (2015). This correlation accounts for the influence of mass transfer resistance caused by temperature glide.

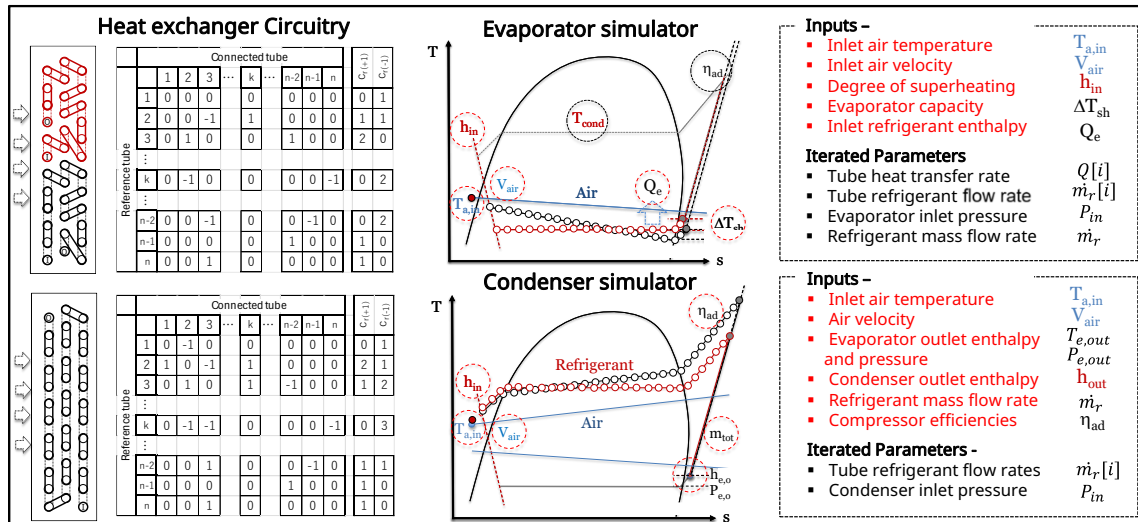


Figure 0-1: Conceptual framework of the simulator for evaporator and condenser

The condenser simulator employed the model of Deng (2019), in which the reduction in condensation heat transfer caused by temperature glide was represented as an equivalent thermal resistance.

Accordingly, the present simulator was configured to consistently evaluate the thermal and flow behaviors within heat exchangers for both pure and zeotropic mixed refrigerants.

Optimization of refrigerant circuitry

In this study, a search algorithm based on evolutionary computation was used to optimize the heat exchanger circuitry. Specifically, the calculated cycle COP was adopted as the fitness function, and the circuit configuration was determined using a genetic algorithm (Figure 2.2-2).

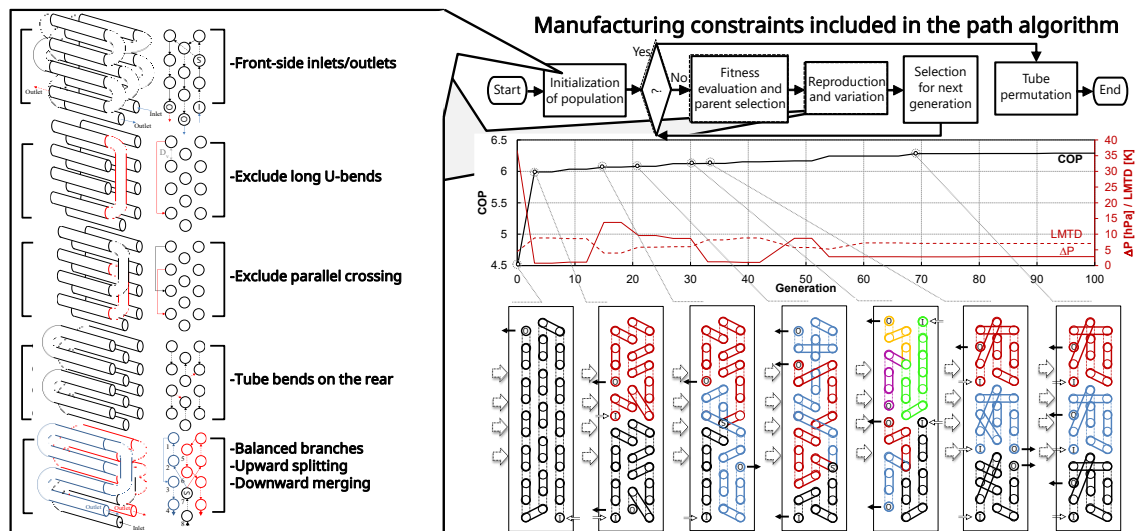


Figure 0-2: Conceptual framework of the evolutionary progression in the path generation algorithm

In this optimization method, the following manufacturing constraints were incorporated into the algorithm in collaboration with industry partners, ensuring that the resulting circuit configuration can be realized in actual manufacturing processes.

- The refrigerant inlet and outlet were arranged on the front face of the heat exchanger.
- Long U-bends were not used.
- Parallel crossings between tubes were avoided.
- Tube-bending geometry on the rear side of the heat exchanger was predefined.

In addition, to suppress maldistribution of the flow rate, a structure was adopted in which branching was directed upward and merging downward, thereby homogenizing the flow distribution.

By incorporating these constraints into the algorithm, circuit configurations with high thermal performance and feasible implementation in actual manufacturing processes can be derived automatically.

In the circuitry optimization conducted in this study, attention was focused mainly on two design elements: branching structure and circuit arrangement (Figure 2.2-3).

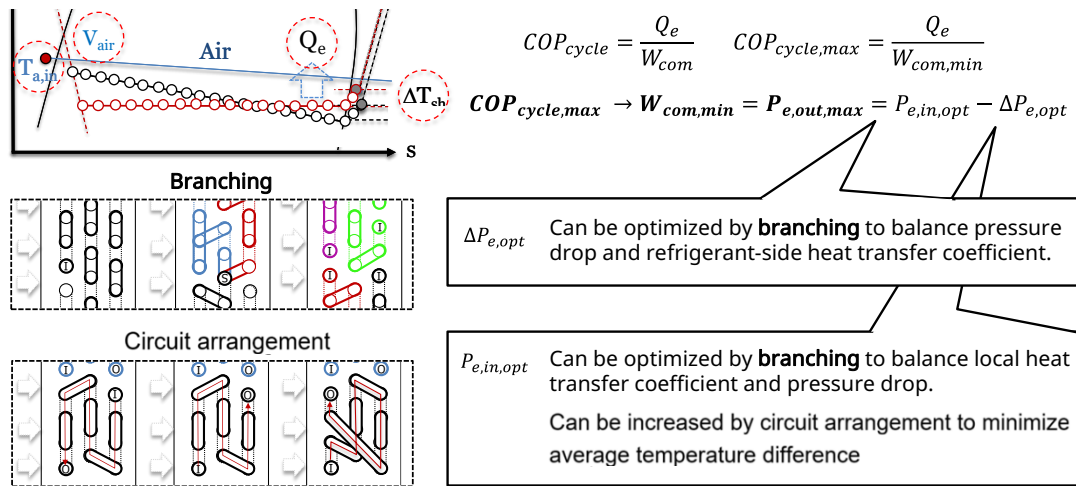


Figure 0-3: Fundamental topological features of optimized circuitry

The branching structure optimizes the balance between refrigerant-side pressure loss and the local heat transfer coefficient by adjusting the number and arrangement of parallel flow paths.

In contrast, the circuit arrangement is intended to improve the temperature distribution throughout the heat exchanger and reduce the mean temperature difference by adjusting the spatial arrangement of the refrigerant and air flows.

Thus, the branching structure is a design element that primarily affects the hydraulic and local heat transfer characteristics, whereas the circuit arrangement influences the overall temperature and phase distributions within the heat exchanger.

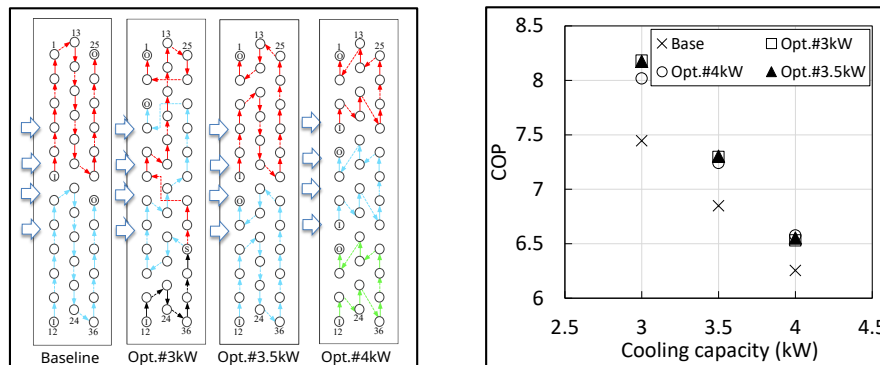
Investigation of evaporator optimization

To verify the effectiveness of the optimization method, a case study of an evaporator using R32 as the reference refrigerant was conducted (Figure 2.2-4).

Under part-load operation (3 kW), where the influence of pressure loss is relatively small, circuit configurations that enhance the local heat transfer coefficient exhibited high performance. Under this condition, it was confirmed that a design improving refrigerant-side heat transfer by adjusting the branching structure is effective.

By contrast, under rated-load operation (4 kW), where the influence of pressure loss becomes more significant, circuit configurations that reduce pressure loss demonstrated higher performance, even if the local heat transfer coefficient decreased somewhat.

Optimized arrangement: cross-parallel (two-phase pressure drop) to cross-counter (superheating)



Optimized branching: increasing branches at higher design capacity (larger pressure drop)

Figure 0-4: Baseline and optimized circuitries for R32 at different operating conditions

2.2.3 Characteristics of circuit structures

In the optimized evaporator circuitry, the circuit arrangement varied along the refrigerant flow direction. Two distinct configurations were adopted:

- Cross-parallel in the two-phase region, selected to suppress pressure loss.
- Cross-counter in the superheated region, selected to improve heat exchange efficiency.

In addition, when the design capacity was large, the refrigerant mass flow rate increased and accordingly, the number of branches also increased.

These results indicate that the proposed optimization algorithm can appropriately adjust the circuit structure in accordance with local thermodynamic and fluid-dynamic conditions (Figure 2.2-5).

The circuit configurations selected in this study were also prototyped, and their validity was verified experimentally through comparison with model calculations.

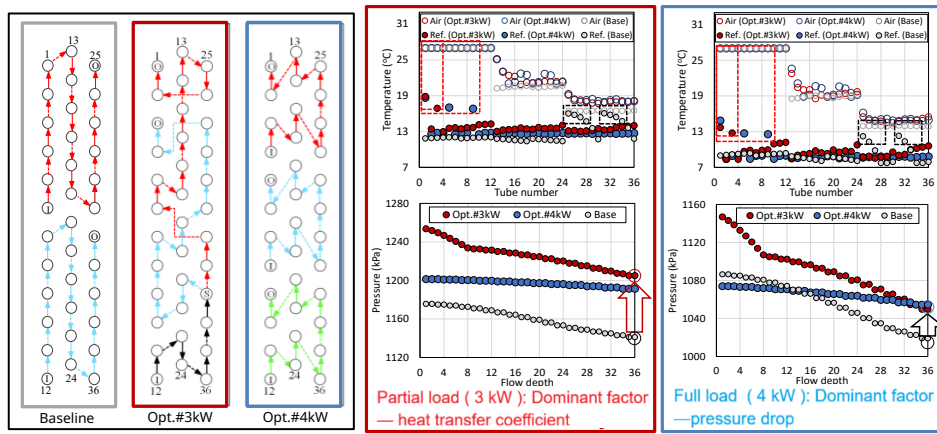


Figure 0-5: Analysis of the optimized circuitry features for R32 at 3 and 4 kW operation

2.2.4 System-level investigation

When the optimization results for the evaporator circuitry were evaluated at the system level, performance improvements were confirmed under all operating conditions. Notably, zeotropic mixed refrigerants exhibited remarkable gains.

When the evaporator optimization results were assessed within a vapor compression cycle combined with an ideal condensation process, the performance of R454C improved to a level comparable to that of R32 (Figure 2.2-6).

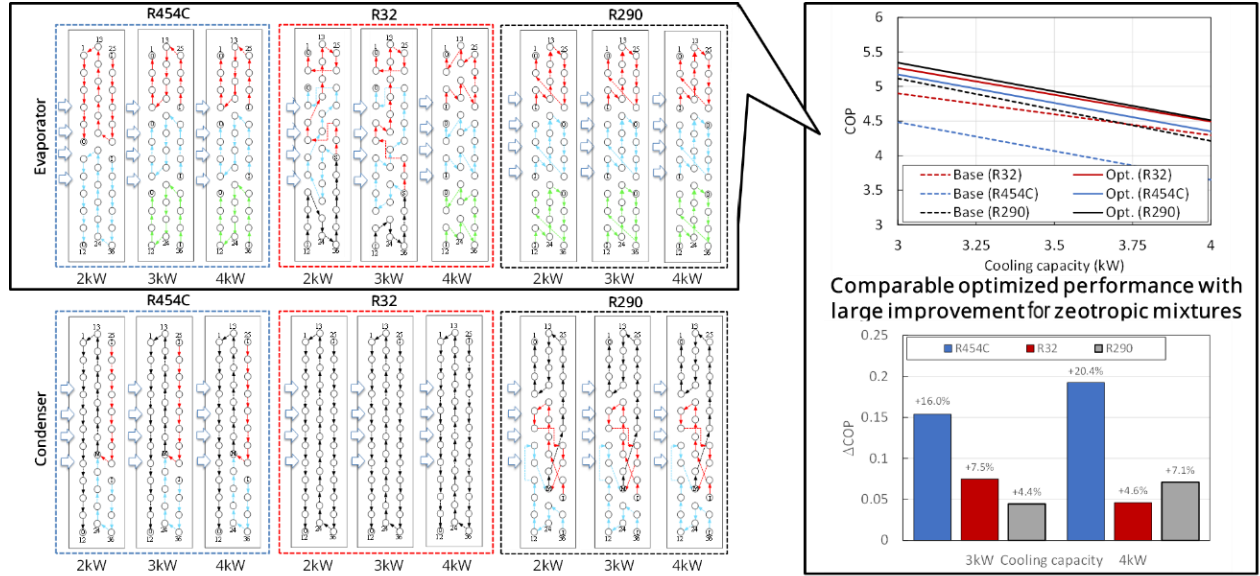


Figure 0-6: Evaporator and condenser circuitry optimization results for different refrigerants

However, for zeotropic mixed refrigerants, a performance difference remains. This is mainly attributed to the transport characteristics on the condenser side and cannot be completely eliminated through evaporator circuit design alone (Figure 2.2-7).

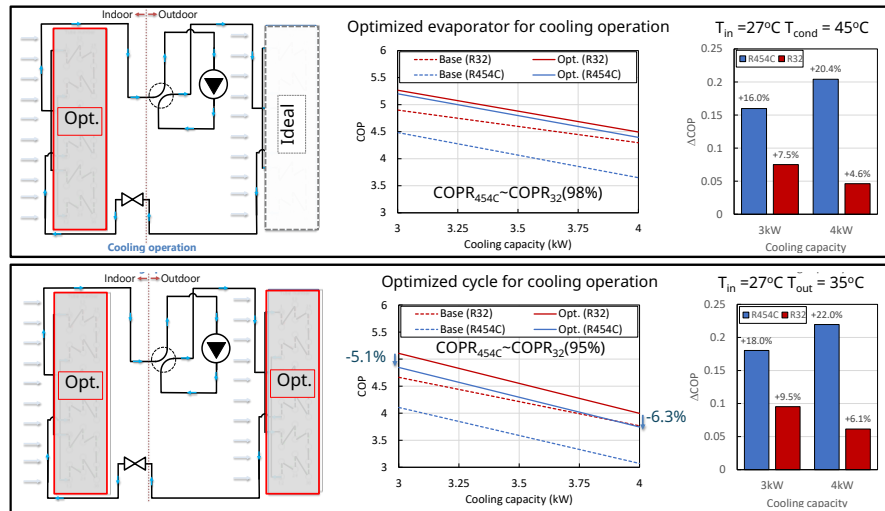


Figure 0-7: Optimized cycle performance comparison for different refrigerants

These results demonstrate that achieving a high-performance heat exchanger design requires integrated optimization that considers both refrigerant circuit configuration and refrigerant properties.

2.3 Research and Development of Refrigeration Cycle System Evaluation Technology

2.3.1 Validation using the optimization verification device

An optimization verification device was fabricated, and experiments were conducted to validate the heat exchanger optimized by the AI-based design method. A schematic diagram of the verification device is illustrated in Figure 2.3-1,

and the heat exchanger in Figure 2.3-2. Photographs of the apparatus are shown in Figures 2.3-3 and 2.3-4, and the main specifications are listed in Table 2.3-1.

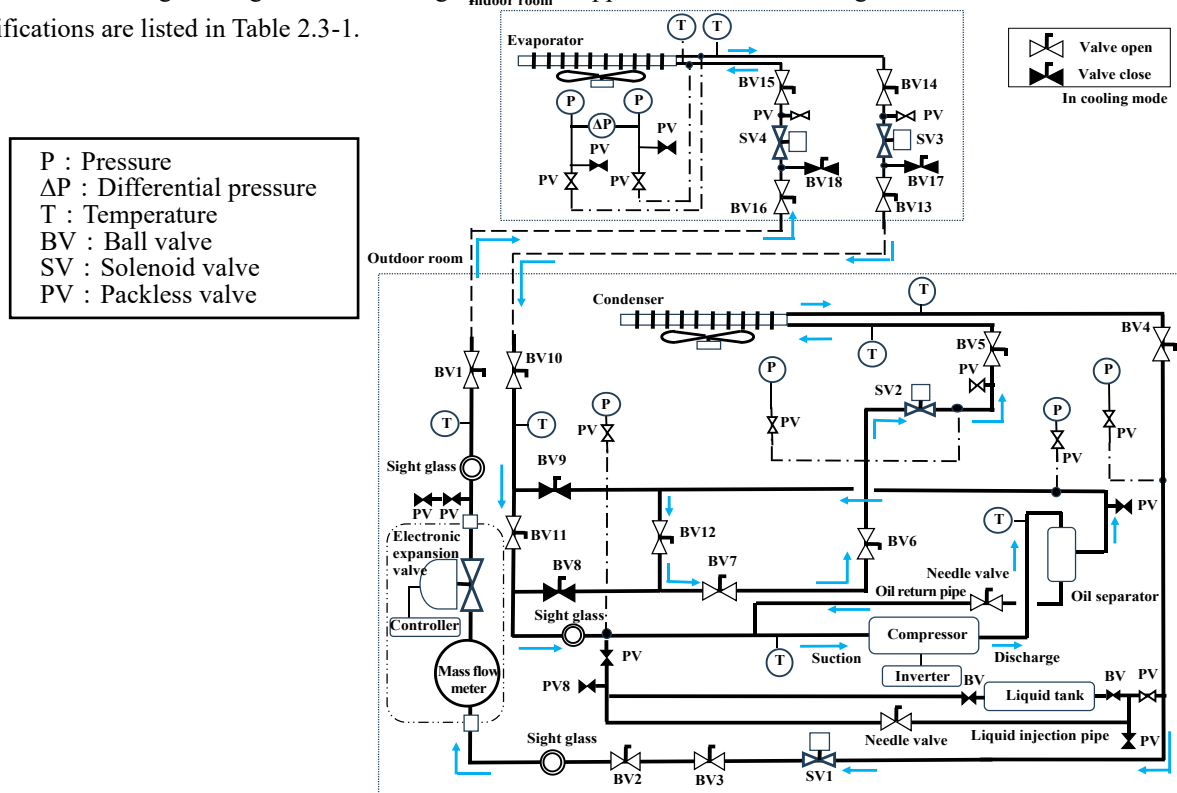


Figure 0-1: Schematic diagram of the optimization verification device

Table 0-1: Main specifications of the optimization verification device

Name	Specification
Compressor	Horizontal scroll type, stroke volume 15.9 cc, refrigerant oil: POE
Inverter	C1-022LF2 (Hitachi Industrial Equipment Systems)
Expansion valve	PKV-14BS (Saginomiya Seisakusho) [Cv value: 0.05, Max. opening: 480 pulse]
Condenser fan	EMR1865-A (Oriental Motor) [Max. flow volume: 8.2 m ³ /min × 2 units]
Liquid tank	Swagelok, volume: 500 cc
Tube	φ12.7 mm (t 0.8 mm) (insulated with 13 mm thick Aeroflex insulation)
Mass flow meter	ALTI mass II type U(Oval) [Measurement range: 3.6-720 kg/h, accuracy: ±0.1%RS]
Pressure	FP101(Yokogawa Electric)[Measurement range: 0-5 MPa, accuracy: ±0.25%FS]
Differential pressure	EJX110J (Yokogawa Electric) [Measurement range: 2.5-500 kPa, accuracy: ±0.075%FS]
Temperature	T-type thermocouple (calibrated with ice water at 0°C)

Table 0-2: Specifications of the heat exchanger (unit: mm)

Tube type	Outer diameter	Tube thickness	Height	Width	Row pitch	Stage pitch	Number of rows	Number of stages	Fin pitch	Fin thickness	Length
Bare	6.35	0.5	232	48	16	19.05	3	12	1.2	0.115	500

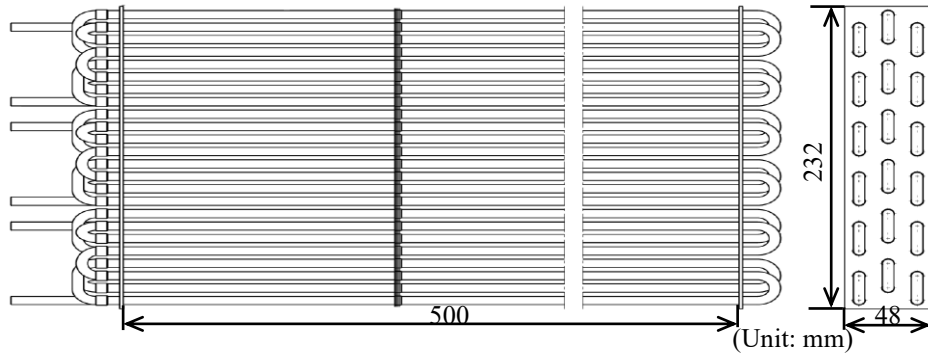


Figure 0-2: Overall view of the heat exchanger

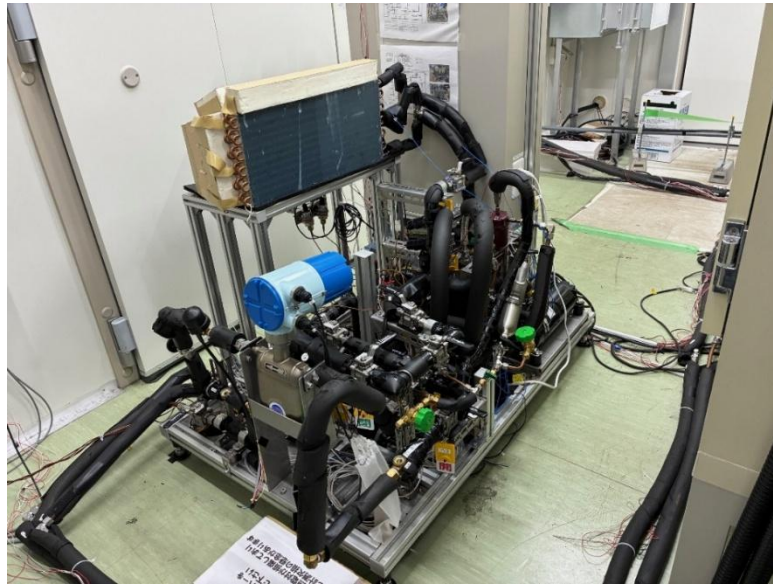


Figure 0-4: Installation view of the optimization verification equipment in the outdoor unit room

The specifications of the heat exchanger (evaporator, parallel-flow type) are listed in Table 2.3-2, and the path diagram is shown in Figure 2.3-5. In the case of parallel flow between the refrigerant and air, “In” indicates the refrigerant inlet while “Out” denotes the refrigerant outlet. When the suffix “-C” is attached (e.g., “Optimized 4-C”), it indicates the

counterflow case, in which the refrigerant enters from **Out** and exits from **In**, opposite to the direction shown in Figure 2.3-5. The types of fabricated heat exchangers are summarized in Table 2.3-3.

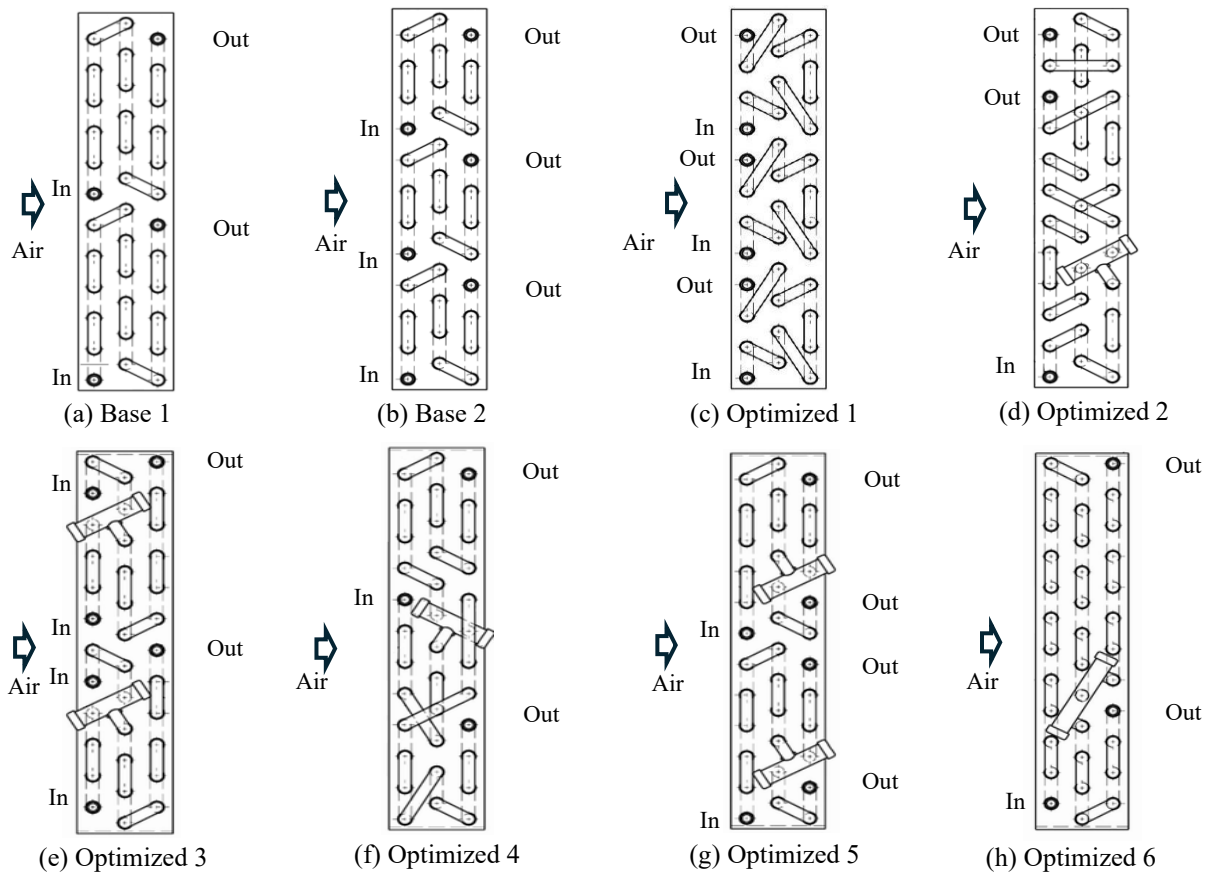


Figure 0-5: Path diagram of heat exchanger (evaporator, parallel flow)

Table 0-3: List of fabricated heat exchanger types

Symbol	Name	Number of passes	Type	Optimized refrigerant	Capacity (kW)
a	Base 1	2	Evaporator	R32	—
b	Base 2	3	Evaporator	R454C	—
c	Optimized 1	3	Evaporator	R32	4
d	Optimized 2	1-2	Evaporator	R32	3
e	Optimized 3	4-2	Evaporator	R454C	3-4
f	Optimized 4	1-2	Condenser	R32	4
g	Optimized 5	2-4	Condenser	R454C	5
h	Optimized 6	1-2	Condenser	R454C	3.7

The designed and fabricated heat exchangers were evaluated using R32 under various configurations:

• **Evaporator:** Base 1 (parallel flow), Optimized 1 (parallel flow–counterflow), and Optimized 2 (parallel flow–counterflow)

• **Condenser:** Base 1-C (counterflow) and Optimized 4-C (counterflow)

• **Combined systems:** Optimized (Condenser)–Optimized (Evaporator)

[Optimized (Condenser)–Optimized (Evaporator)] Optimized 4-C–Optimized 1

[Base (Condenser)–Base (Evaporator)] Base 1-C–Base 1

The evaporator evaluation results are presented in Figure 2.3-6. At 3.0 kW, the COP values were 4.608 for Base 1, 5.025 for Optimized 1 (+9.1% relative to Base 1), and 5.226 for Optimized 2 (+13.4% relative to Base 1). At 4.0 kW, the COP values were 3.213 for Base 1, 3.869 for Optimized 1 (+20.5% relative to Base 1), and 3.850 for Optimized 2 (+19.9% relative to Base 1). Thus, Optimized 2 achieved the highest COP at 3.0 kW, whereas Optimized 1 performed best at 4.0 kW.

The condenser evaluation results are displayed in Figure 2.3-7. At 3.7 kW, the COP was 4.667 for Base 1-C and 5.259 for Optimized 4-C (+12.7% relative to Base 1-C). At 4.0 kW, the values were 3.699 for Base 1-C and 3.779 for Optimized 4-C (+3.8% relative to Base 1-C).

The evaluation results combining evaporator and condenser optimization are shown in Figure 2.3-8. At 3.0 kW, the COP was 4.608 for Condenser Base 1-C–Evaporator Base 1 (Base–Base), 5.025 for Condenser Base 1-C–Evaporator Optimized 1 (evaporator-only optimization) (+9.1% relative to Base–Base), and 5.338 for Condenser Optimized 4-C–Evaporator Optimized 1 (optimization of both evaporator and condenser) (+15.8% relative to Base–Base). At 4.0 kW, the COP was 3.213 for Base–Base, 3.869 for evaporator-only optimization (+20.5% relative to Base–Base), and 3.959 for dual optimization (+23.3% relative to Base–Base). These results indicate that simultaneous optimization of both evaporator and condenser yields higher COP than optimizing the evaporator alone, with the relative improvement more pronounced at 3.0 kW.

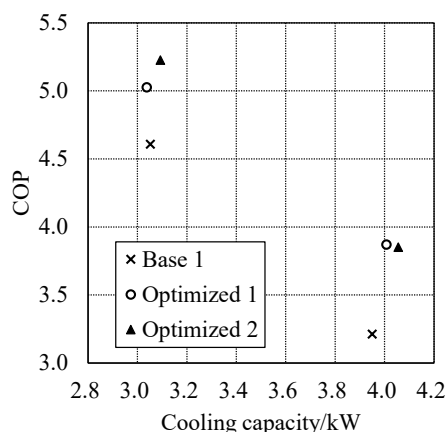


Figure 0-6: Relationship between cooling capacity and COP of R32

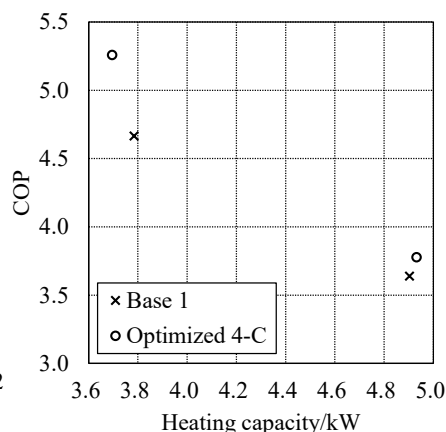


Figure 0-7: Relationship between heating capacity and COP of R32

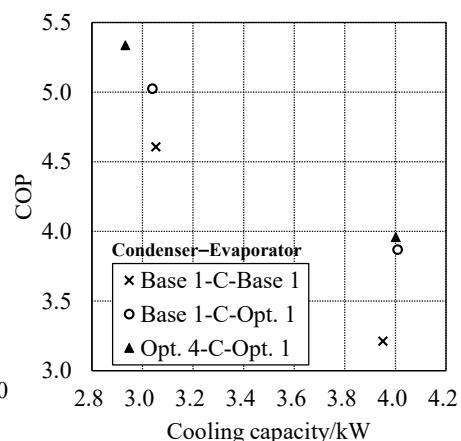


Figure 0-8: Relationship between cooling capacity and COP (R32, cooling, opt.-opt.)

The designed and fabricated heat exchangers were also evaluated using R454C under various configurations:

- **Evaporator:** Base 2 (parallel flow), Base 2-C (counterflow), and Optimized 3-C (counterflow)
- **Condenser:** Base 1-C (counterflow), Base 2-C (counterflow), Optimized 5-C (counterflow), and Optimized 6-C (counterflow)

The evaporator performance results are illustrated in Figure 2.3-9. At 3.0 kW, the COP values were 3.371 for Base 2, 3.983 for Optimized 3-C (+18.2% relative to Base 2), and 4.215 for Base 2-C (+25.0% relative to Base 2). At 3.6 kW, the COP values were 3.038 for Optimized 3-C and 3.297 for Base 2-C. Thus, Base 2-C exhibited the highest COP at both load conditions.

The condenser performance results are shown in Figure 2.3-10. At 3.7 kW, the COP was 4.891 for Base 2-C, 5.168 for Base 1-C (+5.7% relative to Base 2-C), 5.304 for Optimized 5-C (+8.4% relative to Base 2-C), and 5.391 for Optimized 6-C (+10.2% relative to Base 2-C). At 5.0 kW, the COP values were 4.108 for Base 2-C, 4.460 for Base 1-C (+8.6% relative to Base 2-C), 4.571 for Optimized 5-C (+11.3% relative to Base 2-C), and 4.562 for Optimized 6-C (+11.1% relative to Base 2-C). Thus, condenser optimization, particularly in Optimized 5-C and Optimized 6-C, yields notable COP improvements compared with the base designs.

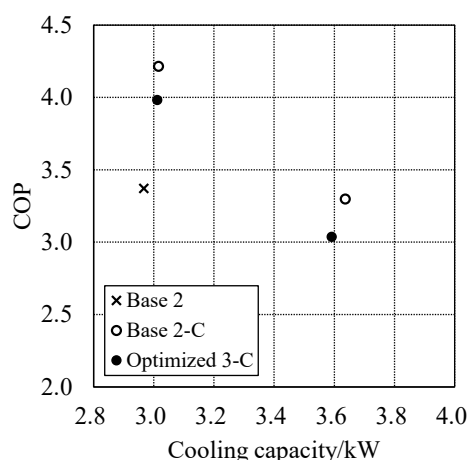


Figure 0-9: Relationship between cooling capacity and COP of R454C

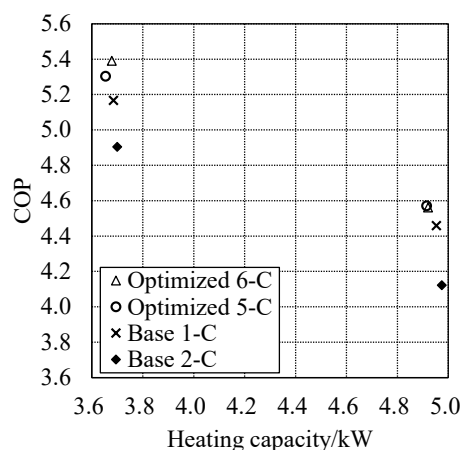


Figure 0-10: Relationship between heating capacity and COP of R454C

2.3.2 Validation results

To confirm the validity of the optimization, the performance of the evaporator circuitry optimized for R32 was evaluated under multiple operating conditions.

Each circuit exhibited its highest performance under the respective design condition.

The circuit optimized for 3 kW exhibited superior part-load performance, characterized by an increase in evaporation temperature accompanied by a tendency for higher refrigerant-side pressure loss.

In contrast, the circuit optimized for 4 kW achieved the highest performance at rated operation. Here, the reduction of pressure loss and improvement of temperature distribution by means of circuit arrangement were identified as the principal factors contributing to performance enhancement (Figure 2.3-11).

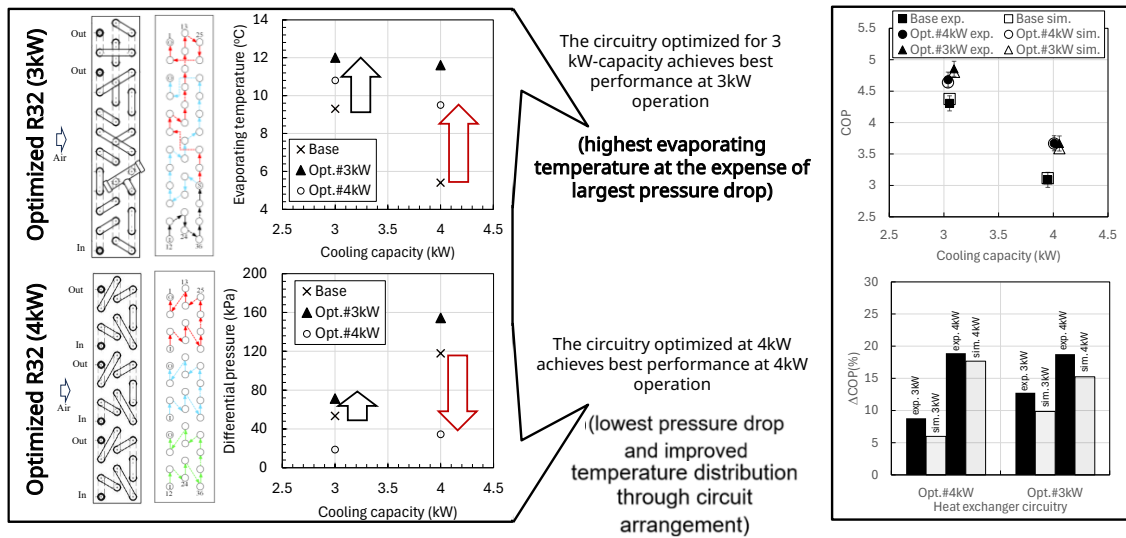
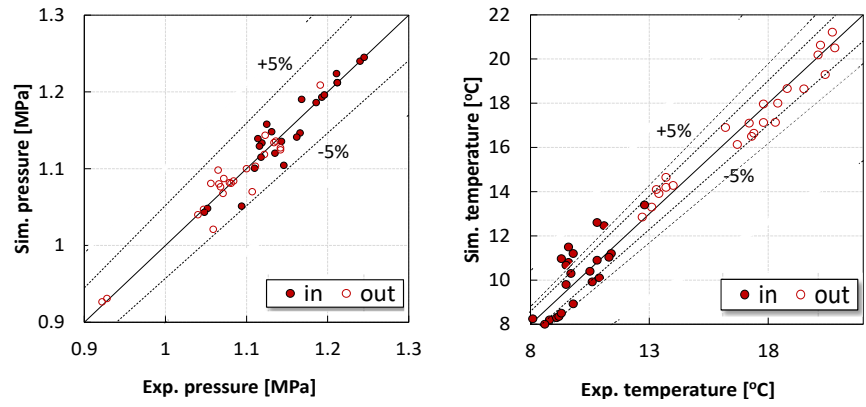


Figure 0-6: Validation of optimized evaporator features for R32 refrigerant

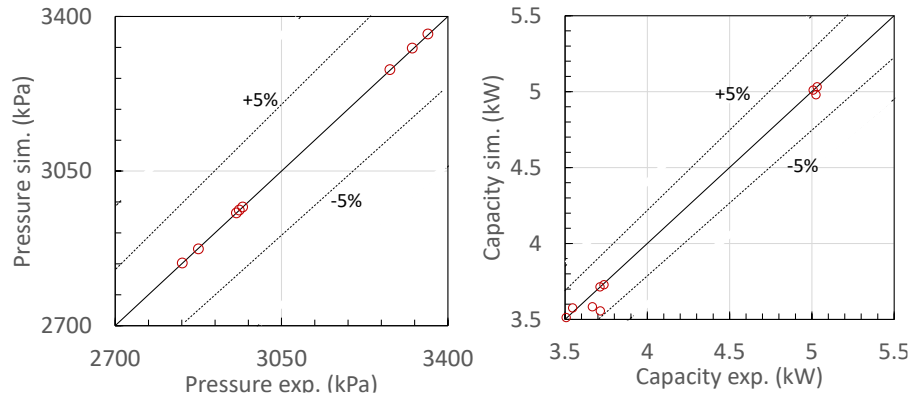
Furthermore, the validation was extended beyond evaporators to include condensers and the R454C mixed refrigerant. As a result, it was confirmed that the proposed optimization method is also effective for R454C.

In addition, model validation was performed for the evaporator and condenser simulators by comparison with experimental and reference data (Figures 2.3-12 and 2.3-13). Consequently, for both R32 and R454C, the calculated results agreed well with the experimental data, confirming that the prediction accuracy for heat transfer and pressure loss was sufficient.

From these findings, it is evident that the simulation and optimization methods established in this study provide a robust technological foundation for advancing heat exchanger circuit design toward the development of high-efficiency air conditioners.



(a) R32 evaporator



(b) R32 condenser

Figure 0-7: Validation of evaporator and condenser models for R32 refrigerant

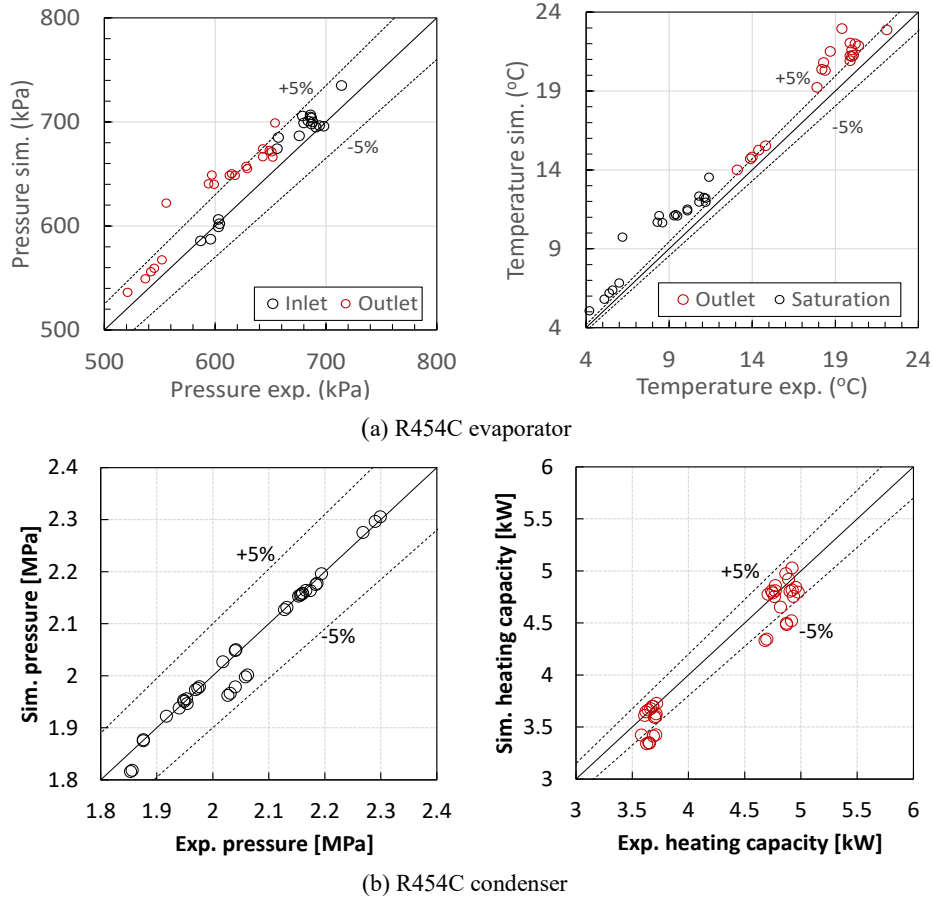


Figure 0-8: Validation of evaporator and condenser models for R454C refrigerant

2.3.3 LCCP evaluation through heat exchanger optimization

To assess the impact of heat pump systems on global warming, the LCCP, which is an integrated evaluation method that encompasses the effects of refrigerant leakage and energy consumption, was first proposed in the 1990s. In recent years, many studies have applied LCCP in accordance with the calculation equations described in the Guideline for LCCP v1.2 (2016). The relevant equations are given below:

$$LCCP = Direct\ Emission + Indirect\ Emission \quad (2.3-1)$$

$$Direct\ Emission = C \cdot (L \cdot ALR + EOL) \cdot (GWP + Adp.\ GWP) \quad (2.3-2)$$

$$Indirect\ Emission = L \cdot AEC \cdot EM + \Sigma(m \cdot MM) + \Sigma(mr \cdot RM) + C \cdot (1 + ALR) \cdot RFM + C \cdot (1 - EOL) \cdot RFD \quad (2.3-3)$$

As shown, LCCP consists of two components:

- **Direct impact** due to refrigerant leakage. The leakage rate over the service life is first calculated as the product of the annual leakage rate (ALR) and service life (L), and the refrigerant release rate at end-of-life (EOL) is then added to obtain the total leakage rate. The direct impact of refrigerant leakage is then obtained as the product of refrigerant charge (C) and global warming potential (GWP, Adp. GWP).

- **Indirect impact** due to energy consumption and manufacturing/recovery processes. This includes five elements: product manufacturing, electricity consumption of the product, material recovery, refrigerant manufacturing, and refrigerant recovery. For electricity consumption, the CO₂ emissions are calculated as the product of the annual energy consumption, service life, and the CO₂ emission factor. The global warming impacts of product manufacturing and

recovery are calculated as the products of the product mass (m) and recovered-material mass (mr) with the respective CO₂ emission factors for material manufacturing (MM) and recovered material (RM).

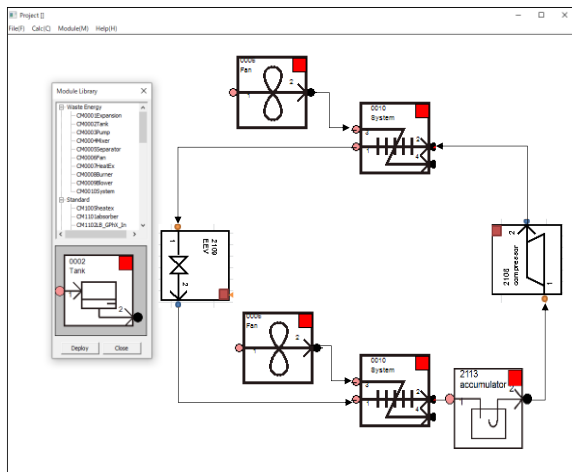
In prior studies, LCCP evaluations of R410A conducted at The University of Tokyo (Horie *et al.*, 2011) and trial calculations for various refrigerants carried out by the Japan Refrigeration and Air Conditioning Industry Association showed that the global warming impact associated with electricity consumption, which depends on operating conditions, is the dominant contributor to LCCP. In contrast, the global warming impacts of product and refrigerant manufacturing and recovery are comparatively small.

Table 2.3-4 summarizes the variables used in the LCCP calculations and their definitions.

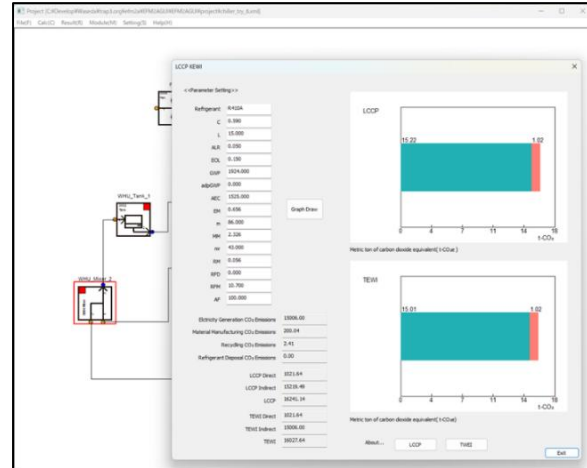
Table 0-4: Variables and their meanings in LCCP calculations

Symbol	Description	Unit
C	Refrigerant charge	kg
GWP	Global warming potential of refrigerant	kg-CO ₂ eq/kg
Adp. GWP	GWP of atmospheric degradation products of the refrigerant	kg-CO ₂ eq/kg
L	Product lifetime	year
ALR	Annual leakage rate during operation	%/year
EOL	Refrigerant release rate at end-of-life	%
AEC	Annual electricity consumption	kWh/year
EM	CO ₂ emission factor of electricity	kg-CO ₂ /kWh
MM	Impact of material manufacturing per unit mass	kg-CO ₂ /kg
m	Mass of material	kg
RM	Impact of material recycling per unit mass	kg-CO ₂ /kg
mr	Mass of recycled material	kg
RFM	Impact of refrigerant manufacturing	kg-CO ₂ /kg refrigerant
RFD	Impact of gases released during refrigerant disposal or recycling	kg-CO ₂ /kg refrigerant

Here, for electricity consumption, the results calculated by the system simulator “ENERGY FLOW +M II” (Jeong *et al.*, 2021), which was developed in this study, were transferred to the LCCP simulator. Examples of the simulator display screens and the calculation results generated by the LCCP simulator are shown in Figure 2.3-14. The target air conditioner, defined in consultation with the Japan Refrigeration and Air Conditioning Industry Association, was specified as a room air conditioner with a rated cooling capacity of 4 kW.



(i) Display screen of system simulator



(ii) Display screen of calculation result by LCCP simulator

Figure 0-9: Example display screen of the “ENERGY FLOW +M II” simulator

(a) Method for calculating annual electricity consumption (AEC)

① Aggregation of occurrence hours by outdoor air temperature

Outdoor temperature data from JIS C9612 (2013) were used. According to this standard, the operating time for room air conditioners for each outdoor temperature is 1,569 h for cooling and 2,707 h for heating, for a total of 4,276 h.

② Calculation of air-conditioning load at each outdoor temperature

The load at each outdoor temperature was obtained using the JIS air-conditioning load model. In this model, the cooling load is 100% at an outdoor temperature of 35°C and becomes zero at 23°C. The heating load is 100% at an outdoor temperature of 0°C and decreases to zero at 17°C. In addition, when the outdoor temperature is between 15°C and 21°C, the air conditioner is not operated, and the load is therefore zero.

③ Calculation of electricity consumption at each outdoor temperature

The power consumption was determined from the air-conditioning load at each outdoor temperature, with COP values calculated by simulation. Multiplying this by the occurrence time at each temperature yields the electricity consumption for that condition.

④ Calculation of annual electricity consumption

The annual electricity consumption was obtained by integrating the electricity consumption across all outdoor temperatures.

⑤ Optimization using the heat exchanger simulator and COP calculation

Figure 2.3-15 shows the optimization results for heat exchangers using R32 and R454C.

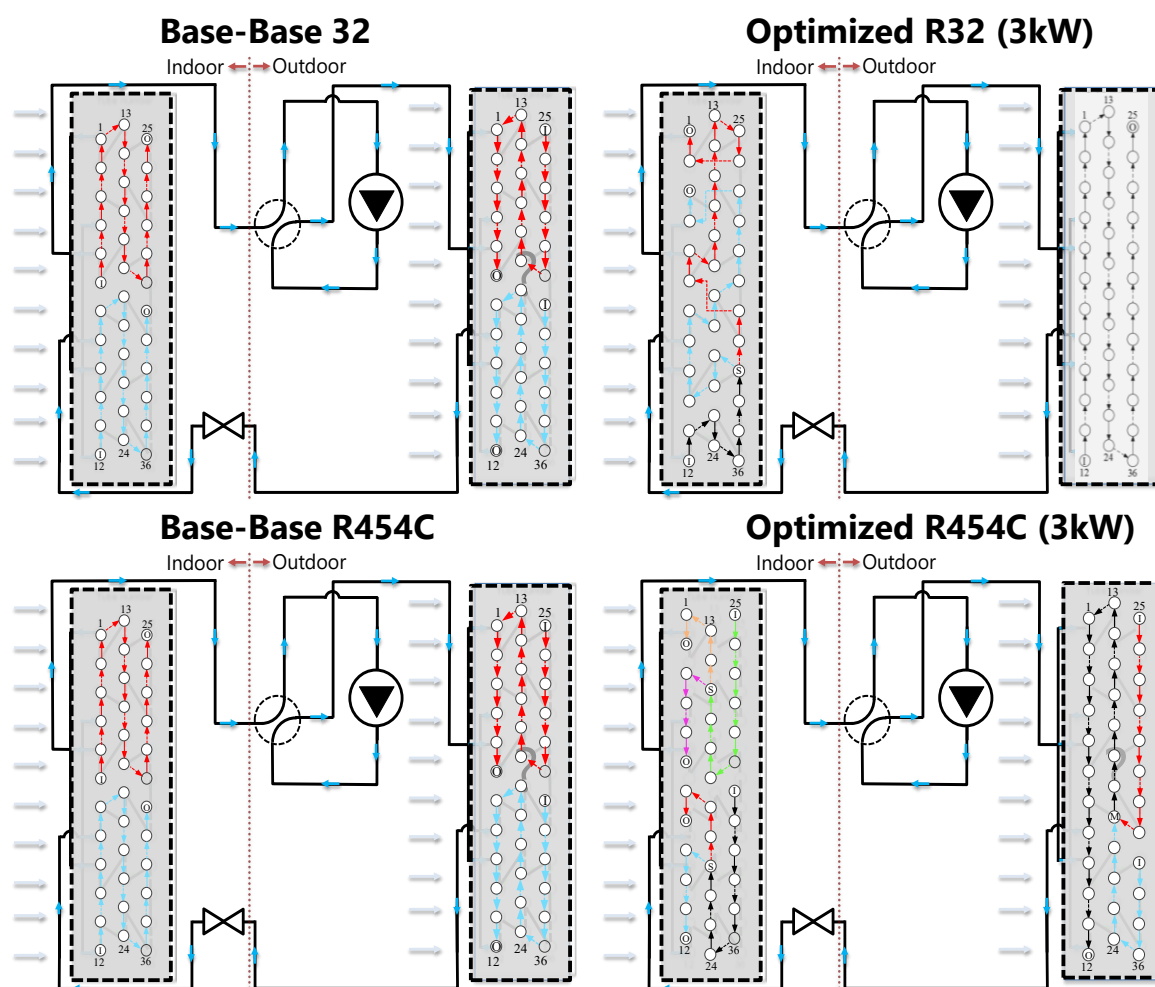


Figure 0-10: Optimization results for heat exchangers using R32 and R454C

The calculated COP values are listed in Table 2.3-5. Because only the heat exchangers were optimized, the reduction in COP associated with intermittent operation during low-load conditions in actual equipment was not taken into account.

The power consumption at each outdoor temperature was obtained from the COP values and multiplied by the corresponding occurrence time to determine the electricity consumption. The calculated electricity consumption results are summarized in Table 2.3-6.

Table 0-5: COP calculation results from heat exchanger optimization

Mode	PLR (%)	Outdoor Temp. (°C)	Indoor Temp. (°C)	COP (-)			
				R32		R454C	
				Base	Opt.	Base	Opt.
Cooling	100	35	27	3.77	3.94	3.07	3.69
	75	32	27	5.10	5.60	4.51	5.28
	50	29	27	7.72	9.02	6.44	8.04
	25	26	27	15.84	15.81	11.76	12.94
Heating	75	2	20	3.19	3.48	2.58	3.41
	50	7	20	4.57	4.96	3.74	4.68
	25	12	20	8.16	9.12	6.83	8.03
	10	15	20	14.18	20.36	12.12	16.40

Table 0-6: Calculation results of air-conditioning power consumption through heat exchanger optimization

Mode	Air-conditioning load (kWh)	R32 Base		R32 Optimized		R454C Base		R454C Optimized	
		COP (-)	AEC (kWh)	COP (-)	AEC (kWh)	COP (-)	AEC (kWh)	COP (-)	AEC (kWh)
Cooling	2498.33	7.72	323.77	8.53	293.02	6.43	388.44	7.65	326.48
Heating	5463.12	5.37	1016.87	5.59	977.58	4.42	1235.54	5.22	1046.76
Total	7961.45	5.94	1340.65	6.27	1270.60	4.90	1623.98	5.80	1373.25

A detailed dynamic test of a room air conditioner with a rated cooling capacity of 4 kW was conducted using the “emulator load test apparatus,” an environmental test system capable of highly accurate evaluation of actual operating performance. The APF value determined from this test was 3.97. The evaluated room air conditioner is shown in Figure 2.3-16.

Because the heat exchanger simulation yielded an APF of 5.94 for R32 Base, each electricity-consumption value was assumed to be scaled by a factor of $5.94/3.97 = 1.50$ relative to the actual unit. The adjusted comparison with the real machine is presented in Table 2.3-6.



Figure 0-11: Photograph of the tested 4kW room air conditioner

Table 0-7: Assumed values for a 4 kW room air conditioner using R32 and R454C(1/2)

Direct Emission	R32	R454C	
C	0.9	←	kg
GWP	675	148	kg CO2e
Adp. GWP	0	←	R454C
L	15	←	Year
ALR	5	←	%
EOL	15	←	%

Table 0-8: Assumed values for a 4 kW room air conditioner using R32 and R454C(2/2)

Indirect Emission	R32	R454C	
L	15	←	Year
AEC			kWh
EM	0.423	←	kg CO2e/kWh
MM	2.326	←	kg-CO2/kg
m	45.5	←	kg
RM	0.0562	←	kg-CO2/kg
mr	22.75	←	kg
RFM	7.2	←	kg-CO2/kg refrigerant

The numerical values required for the LCCP calculation are listed in Tables 2.3-7 and 2.3-8. The CO₂ emission factor (EM) per kWh of electricity was taken as the national average coefficient published by the Ministry of the Environment and Ministry of Economy, Trade and Industry (2026), in “Emission factors by electric utility (for calculating unadjusted emissions), FY2024 results”. In addition, according to a survey by the Japan Refrigeration and Air Conditioning Industry Association (Shimizu *et al.*, 2025), the annual operating time of a room air conditioner is 2,995 h and the product lifetime is 12 years. By inputting these values into the LCCP simulator, the following results were obtained.

⑥ Calculation results

The results are illustrated in Figure 2.3-17. It was clarified that, through optimization of the heat exchangers, the LCCP of R454C can be reduced to a level nearly equivalent to that of R32.

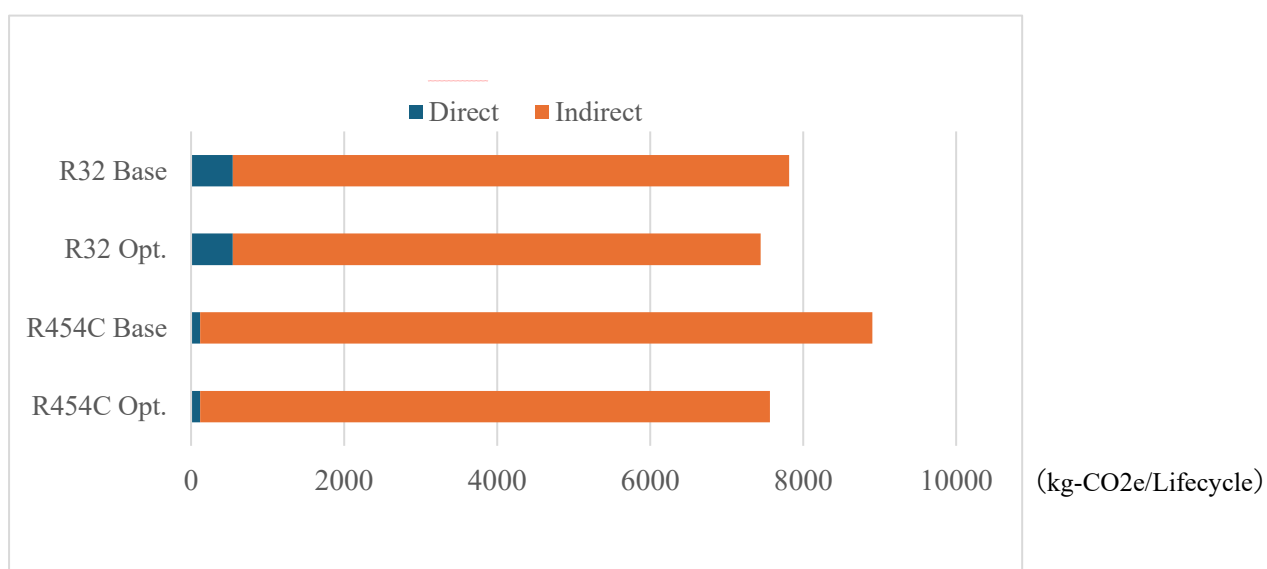


Figure 0-12: LCCP calculation results based on heat exchanger optimization
(air conditioner with cooling capacity of 4 kW for residential buildings)

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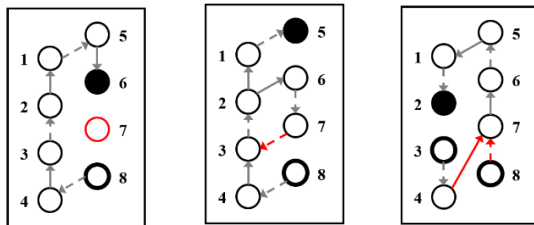
3. Development of an AI Optimization Algorithm for the Refrigeration Cycle

Simulator (The University of Electro-Communications)

To realize heat exchanger optimization at the actual equipment level, simulator development must account for various constraints inherent to the actual manufacturing process, such as physical dimensions of the heat exchanger and refrigerant piping branch paths. The objective of this study was to develop an AI optimization algorithm that takes these constraints into consideration.

3.1 Development of Constraint Conditions

Various constraint conditions in the actual heat exchanger manufacturing process were comprehensively identified. Through information exchange with the Japan Refrigeration and Air Conditioning Industry Association, ten constraints (one mandatory and nine optional) were organized. Figure 3.1-1 shows the constraints established in collaboration with the association.

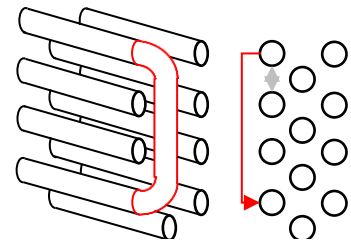


Restrictions on pipe connections

Left: Ensure no unconnected pipes are created.

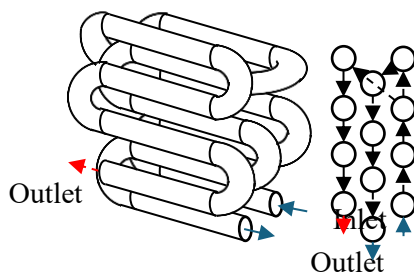
Center: Prevent loops from forming in pipe connections.

Right: Prevent pipes from merging at either end.



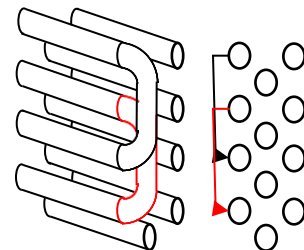
Length of connecting pipes

Up to L units



Direction of inflow and outflow

Inflow and outflow cannot be in the same direction.



Pipe crossing

Pipes cannot cross each other.

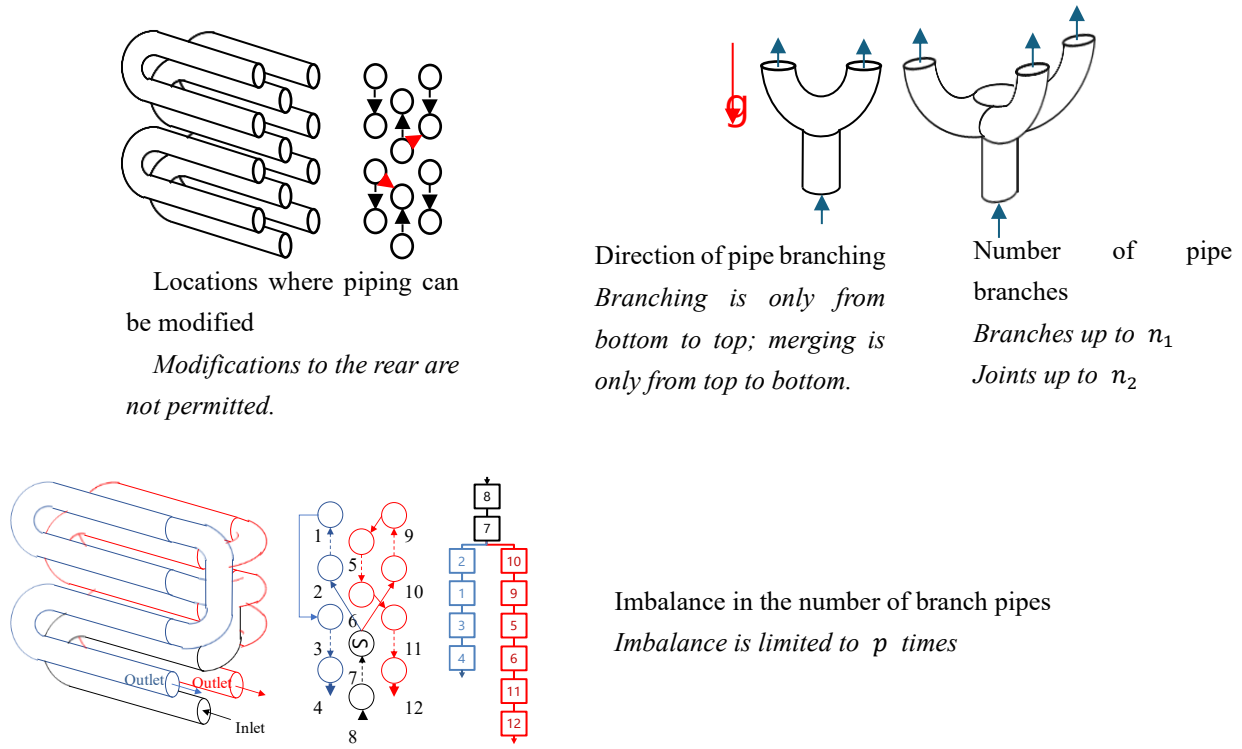


Figure 0-1: Ten restrictions agreed upon with the Japan Refrigeration and Air Conditioning Association

Further investigations were conducted to obtain an appropriate structure, and two additional optional constraints, as illustrated in Figure 3.1-2, were added.

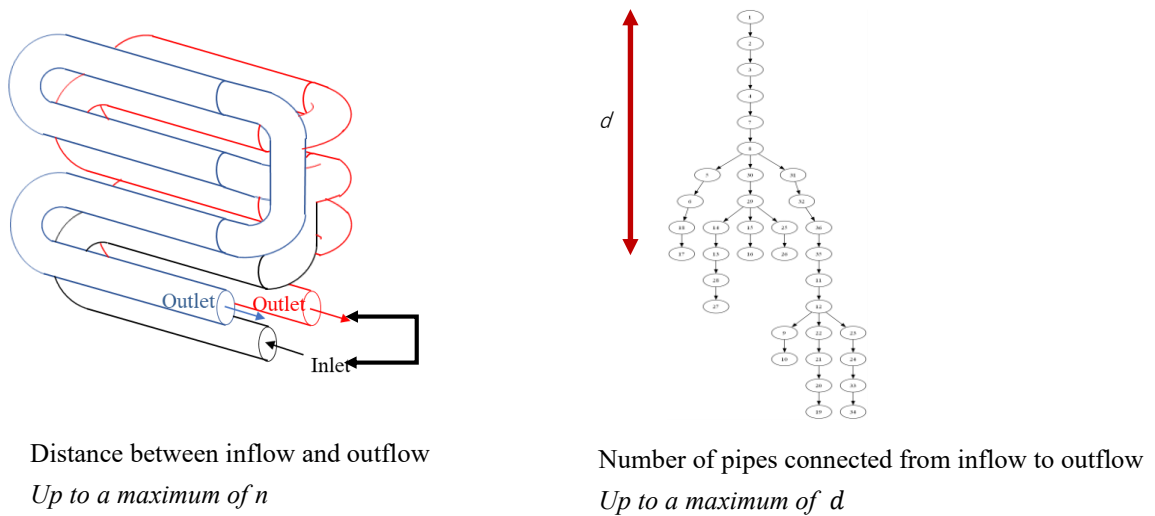


Figure 0-2: Two additional constraints (optional)

The AI program was designed so that the variables L, n_1, n_2, p, n, d shown in Figures 3.1-1 and 3.1-2 can each be freely set by the user.

3.2 Development of an AI Optimization Algorithm that Satisfies the Constraint Conditions

To allow each of the 11 optional constraints to be turned ON or OFF, $2^{11} = 2048$ optimization scenarios were envisaged. Physical constraints on bent piping in the heat exchanger were incorporated into the AI optimization algorithm. By introducing these constraints, the possibility of obtaining infeasible optimization results was reduced and the search space for optimization was significantly narrowed.

Furthermore, the program was improved to enable testing of as many diverse piping configurations as possible within the constraint range. Acceleration mechanisms were implemented both for constraint checking and for execution of the genetic algorithm to improve computational efficiency.

3.2.1 Characteristics of the AI optimization algorithm

Based on the genetic programming framework developed thus far, an AI optimization algorithm was constructed. Its distinctive feature is that, rather than merely generating a piping connection structure for a heat exchanger that satisfies the constraints, it integrates an optimization algorithm designed and implemented to also consider constraint handling, search efficiency, and operational usability. Various manufacturing constraints (e.g., upper limits on the number of branches, restrictions on merging, prohibition of crossings, length constraints, and directional constraints) were implemented as individual decision rules. A modular rule-based architecture was established, allowing each constraint to be independently switched ON or OFF. This design enables flexible combinations of multiple constraints. Importantly, this is not a simple condition-checking mechanism; it is a knowledge-driven design in which constraint knowledge is explicitly embedded within the algorithm.

In addition, the proposed system incorporates an automatic repair mechanism instead of simply rejecting constraint violations. For example, when a cyclic structure is detected, the corresponding link is removed and reconnection is attempted. When isolated nodes occur, their connections are reconstructed within the range that satisfies the constraints. By adopting this flow of “generation → judgment → repair,” the algorithm prevents the search from halting because of frequent generation of infeasible solutions and enables practical exploration of diverse configurations.

3.2.2 Acceleration of the AI optimization algorithm

Acceleration measures were implemented because obtaining an appropriate heat exchanger design still required considerable computation time, even when using genetic programming.

(a) Swap mechanism

A swap function was newly implemented to accelerate the optimization process. In this study, the piping connection structure within the heat exchanger was modeled as a DAG.

A DAG represents a network with a defined flow direction (from inlet to outlet) and no cycles (loops). Since refrigerant must flow in one direction, a closed-loop structure in which the flow returns to its origin is physically impossible. Therefore, in this study, piping connections were formulated as a one-directional network with the inlet as the starting point and the outlet as the endpoint, and only structures without cycles were permitted.

The swap function is a process that partially exchanges the connection relationships among pipes while preserving the DAG structure itself (i.e., the topological conditions in which the outlet is reachable from the inlet and no cycle is included) within the range satisfying the various manufacturing constraints. Specifically, while maintaining the overall flow direction and hierarchical relationships of branching and merging, new piping configurations were generated by randomly changing the combination of destinations at certain connection points.

In this way, by creating an operation that changes only the connections, while maintaining the skeleton of the structure, the following can be suppressed:

- Generation of cyclic structures
- Structures where the outlet becomes unreachable from the inlet
- Structures violating manufacturing constraints

As a result, while limiting the search space to a physically realizable range, a wide variety of candidate structures can be efficiently generated, thereby accelerating optimization calculations based on the genetic algorithm.

(b) Validity judgment at the structure generation stage

Local validity judgments were incorporated into the structure generation stage to reduce the search space in advance. Rather than performing a global judgment after generating a candidate structure, this method excludes configurations that are highly likely to be physically infeasible at the connection addition stage. Consequently, the mass generation of inappropriate solutions is suppressed, and the effective ratio of genetic operations (crossover and mutation) is increased.

(c) Caching mechanism

To improve the efficiency of performance evaluation calculations, a caching mechanism was introduced to avoid reevaluating identical structures. Because heat-exchanger simulations are computationally expensive, repeated evaluations of the same topology are inefficient. Therefore, a mechanism was established to identify structural identity, and the results were reused for individuals that had already been evaluated. This prevents wastage of computational resources and improves the overall throughput of the search.

(d) Structural duplicate elimination mechanism

A structural duplicate elimination mechanism was implemented to maintain diversity within the population. If superior individuals are selected solely on the basis of performance indices, similar structures may proliferate excessively and cause the search to converge to a local optimum. In this system, structural identity is judged and duplicate retention of the same topology is suppressed, thereby ensuring diversity in the search. This strategy considers coverage of the design space as a whole rather than simply pursuing an optimum value.

(e) Hybrid strategy

A hybrid strategy was adopted, in which global and local searches were assigned distinct roles. After a wide range of design candidates is generated by genetic programming, limited local operations are applied to promising solutions for fine adjustment. This makes it possible to explore a broad range in the initial stage of the search and efficiently improve performance during the convergence stage.

(f) Rule-based processing

Rule-based processing is positioned not merely as a constraint-checking function but also as a mechanism for stabilizing the search. Even when generated individuals contain constraint violations, repairs are attempted whenever possible to ensure search continuity. This avoids stagnation caused by the mass generation of infeasible individuals during evolutionary computation.

As described above, in this study, genetic programming was not applied in isolation; instead, the reduction of the search space, reuse of calculations, diversity control, local improvement, and rule-based stabilization were integrated to realize an accelerated design framework capable of addressing large-scale design problems premised on actual equipment manufacturing. This is not acceleration by mere parameter adjustment, but by structural improvement at the level of algorithm design.

3.2.3 Visualization of the AI optimization algorithm

Although the AI optimization program was handled internally as XML files, a visualization program was created to make it easier for users to inspect the results (Figure 3.2-1). Furthermore, a visualization program was developed for piping during the optimization process. For example, this makes it possible to understand how new piping structures 1' and 2' are generated from existing piping structures 1 and 2.

In the future, in addition to enhancing error handling and related functions, further efforts will be made to develop the system into a GUI tool that can be easily used, even by non-researchers.

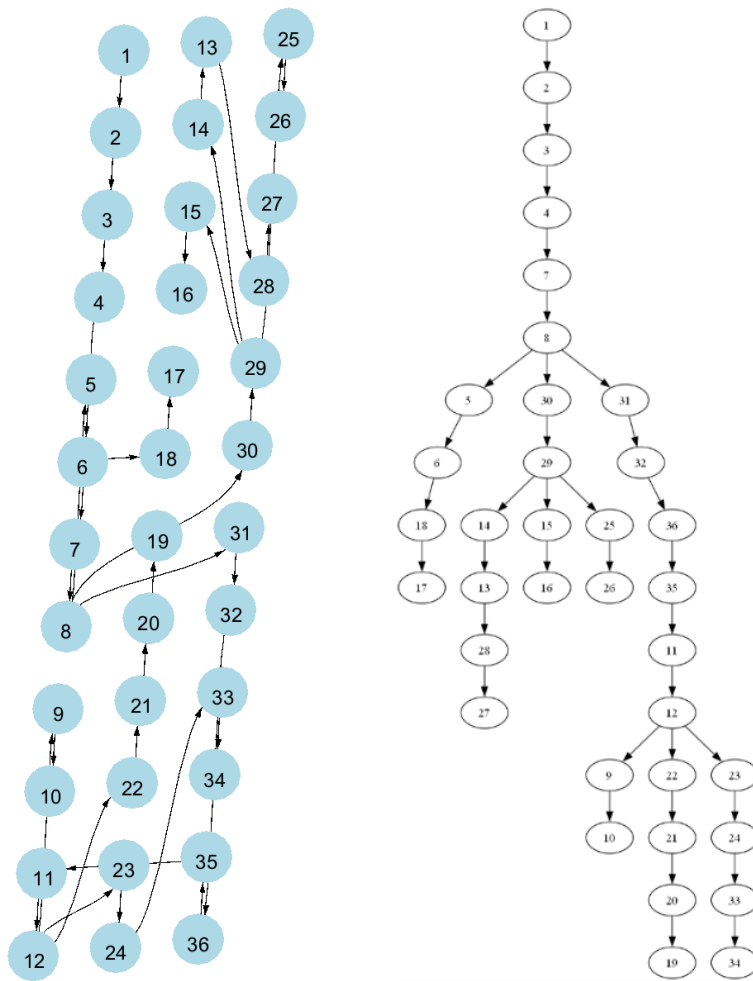


Figure 0-1: Examples of structural visualization

Left: Visualization maintaining physical spatial relationships

Right: Visualization of the flow from entrance to exit