



Innovative Application of Thermodynamic Cycles to Computer Hardware Cooling

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ABSTRACT

To address waste-heat recovery and cooling-energy reduction in high-temperature liquid-cooled data centers under active refrigeration, a vapor-compression refrigeration–organic Rankine cycle coupled system was constructed, and a selective waste-heat recovery method based on thermal quality and marginal net power benefit was proposed. By introducing a continuous recovery termination temperature T_{cut} , a vapor-compression refrigeration energy model, an organic Rankine cycle net power model, a heat-exchanger pinch and UA model, a component exergy-loss model, and a system evaluation framework centered on the net cooling power consumption ratio and the cooling power compensation rate were established. Under baseline computational/simulation conditions, the vapor-compression refrigeration + organic Rankine cycle scheme reduced the net cooling power consumption ratio from 0.300 to 0.230 and net cooling power consumption from 0.90 kW to 0.69 kW relative to the vapor-compression refrigeration-only scheme; the lowest net cooling power consumption ratio was obtained when T_{cut} was approximately 68 °C, and a favorable balance between temperature-lift cost and organic Rankine cycle power-generation benefit was achieved at a vapor-compression refrigeration condensing temperature of approximately 70 °C; under the constraint that the maximum chip temperature did not exceed 95 °C, the feasible preferred range for the vapor-compression refrigeration evaporating temperature was 35–40 °C. As indicated by comparisons of representative working-fluid pairs, the lowest net cooling power consumption ratio and cooling power compensation rate were obtained with the R1234yf/R1233zd(E) pair. From these results, it can be concluded that greater vapor-compression refrigeration waste-heat recovery is not necessarily more favorable; the optimal recovery boundary of the system is jointly determined by thermal quality, organic Rankine cycle marginal net power, heat-exchanger scale, and chip thermal safety.

1. INTRODUCTION

With continued increases in per-rack computing capacity driven by artificial intelligence training, high-performance computing, and high-density servers [1], chip-level heat flux and cooling loads are increased simultaneously [2], and temperature control, energy consumption, and facility space utilization are increasingly difficult to reconcile by conventional low-temperature air-based cooling methods [3]. The heat-transfer path from the chip to the coolant can be shortened by cold-plate and immersion liquid cooling [4], and chip heat can be removed by the coolant at relatively high temperature levels [5]; consequently, not only is heat-dissipation capability improved, but more favorable heat-source conditions are also created for subsequent waste-heat recovery [6]. However, it should not be assumed that heat rejection can be accomplished by natural cooling under all operating conditions [7]; at high ambient temperatures [8], under constrained cooling-water temperatures, where stable control of the supply-liquid temperature is required [9], or during year-round continuous operation of data centers [10], active refrigeration duties still have to be borne by the vapor-

compression refrigeration cycle [11]. The strong dependence of cooling-system performance on ambient conditions further emphasizes the importance of evaluating active cooling configurations from a system-level energy-efficiency perspective [12]. The practical engineering problem of interest is how rejected heat can be further recovered and net cooling power consumption can be reduced within this necessary active-refrigeration boundary.

From the perspective of energy flow, while chip cooling is being accomplished by the vapor-compression refrigeration, both the chip heat absorbed by the evaporator and the compressor input power are upgraded to a higher condensing-temperature level, and a heat source with a pronounced temperature gradient is formed during compressor discharge and condensation [13]. When suitable high-temperature rejected heat is introduced into an organic Rankine cycle [14], the electrical power generated by the expander can be used, through a generator and power-electronics interface, to compensate for part of the electricity consumed by the compressor, pumps, and auxiliary equipment [15]. It should be clarified that the organic Rankine cycle is not intended to construct so-called self-powered refrigeration by additionally

inducing a temperature lift in the vapor-compression refrigeration [16]; rather, under the premise that the vapor-compression refrigeration must operate because of cooling-service boundary conditions, additional recovery is performed on its rejected heat [17]. Therefore, the effectiveness of the coupled system should not be evaluated solely by comparing the organic Rankine cycle power output or thermal efficiency [18]; rather, it should be evaluated in terms of net system power consumption under equivalent chip thermal safety and equivalent cooling-service conditions [19] so that whether the incremental power benefit provided by the organic Rankine cycle can offset the corresponding equipment and operating costs can be examined [20].

Existing studies on vapor-compression refrigeration–organic Rankine cycle and heat-pump-assisted waste-heat utilization have provided a foundation for the cascade utilization of low-grade thermal energy [21]; however, when such approaches are applied to high-temperature liquid-cooled chip scenarios, three interrelated issues remain. First, system designs are often oriented toward “recovering as much condensing heat as possible” [22], whereas less consideration is given to whether lower-temperature heat can continue to yield sufficient marginal net power after being incorporated into a single-pressure organic Rankine cycle [23]. As the minimum heat-source temperature is progressively reduced, the organic Rankine cycle evaporation temperature may be suppressed, incremental heat absorption and incremental power generation no longer remain synchronized, and the required heat-exchanger UA and area increase rapidly [24]. Second, the compressor pressure ratio and power consumption are affected by the vapor-compression refrigeration condensing temperature, while the discharge temperature and the temperature quality of the heat source available to the organic Rankine cycle are also determined by it; consequently, a pronounced coupling between temperature-lift cost and power-recovery benefit is present [25]. Previous energy and exergy analyses of power-generation systems have likewise shown that operating temperatures and pressure conditions can substantially alter both component-level irreversibility and overall thermodynamic performance [26]. Third, vapor-compression refrigeration and organic Rankine cycle working fluids cannot be regarded as two variables that are optimized independently of each other; final system performance is jointly determined by their saturation-temperature curves, pressure levels, critical parameters, volumetric flow rates, expansion ratios, and heat-exchange pinch points [27].

Based on these considerations, the research focus is shifted from maximizing heat recovery to minimizing net cooling power consumption under equivalent cooling service, and a continuous recovery termination temperature, T_{cut} is introduced to describe the heat-recovery boundary on the vapor-compression refrigeration condensation side. Around this core variable, four schemes—direct liquid-cooling heat rejection, vapor-compression refrigeration-only, direct liquid-cooling organic Rankine cycle, and vapor-compression refrigeration–organic Rankine cycle—are first compared, and the application prerequisites for the coupled architecture are clarified. Subsequently, a vapor-compression refrigeration energy model, a segmented condensation and recoverable heat model, an organic Rankine cycle net power model, a heat-exchanger pinch and UA model, and a component exergy-loss model are established, and the net cooling power consumption ratio and cooling power compensation rate are adopted as primary system-level evaluation metrics. On this basis, the

coupling relationships among T_{cut} , vapor-compression refrigeration evaporation/condensation temperatures, organic Rankine cycle evaporation state, and working-fluid combinations are further examined so that thermal quality, marginal net power, heat-exchanger scale, and chip thermal safety can be treated within a unified optimization framework. Such an integrated formulation is consistent with recent energy-system studies in which thermodynamic performance, system configuration, and economic or operational criteria are considered jointly rather than optimized in isolation [28].

Three principal contributions are made. First, a selective waste-heat recovery method based on thermal quality and marginal net power benefit is proposed, in which the superheated section is not presumed to be inherently superior to the two-phase condensation section; instead, the critical boundary at which incremental heat still retains system-level value is identified by continuously varying T_{cut} . Second, a coordinated optimization method is established for vapor-compression refrigeration temperature-lift cost and organic Rankine cycle power-compensation benefit, in which net cooling power consumption ratio minimization is taken as the primary objective and the feasible operating domain is determined under constraints on UA , equipment pressure, and chip temperature. Third, vapor-compression refrigeration and organic Rankine cycle working fluids are screened in a unified manner as a “working-fluid pair,” and the optimal system combination is interpreted across multiple dimensions, including pressure ratio, expander compatibility, heat-source temperature trajectory, pinch point, UA , environmental performance, and safety class. Accordingly, the system configuration and operating principles are presented in Section 2; thermodynamic models and the coordinated optimization method are established in Section 3; the experimental platform and validation scheme are described in Section 4; results are discussed in Section 5 with respect to key operating parameters, working-fluid matching, dynamic response, and long-term energy efficiency; and the main conclusions and applicable boundaries are summarized in Section 6.

2. SYSTEM CONFIGURATION AND OPERATING PRINCIPLES

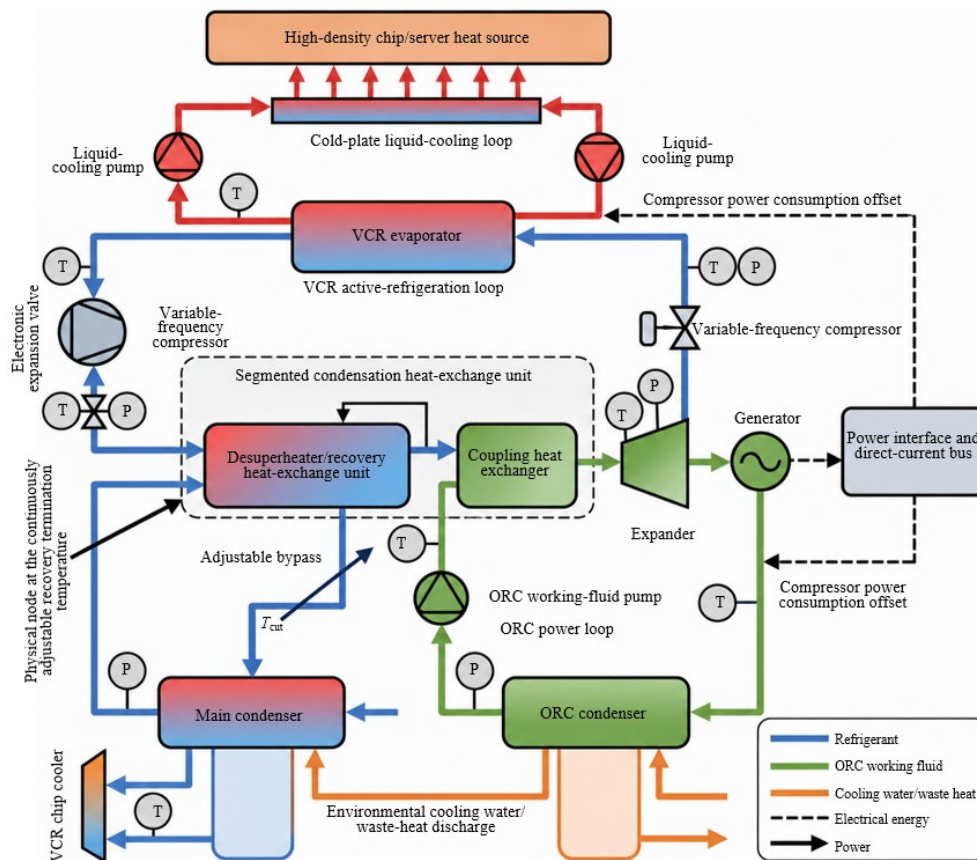
The system is composed of a chip/server heat source, a high-temperature liquid-cooling loop, a vapor-compression refrigeration active-refrigeration loop, a segmented condensation heat-exchange unit, an organic Rankine cycle power loop, and a power interface. Heat generated by the chips is first carried away by a cold plate or immersion liquid-cooling medium and is transferred to the refrigerant in the vapor-compression refrigeration evaporator. When the external heat-dissipation capacity is insufficient or the supply-liquid temperature must be maintained within a given range, the refrigerant pressure and temperature are elevated by the compressor, and heat is transported to the condensation side. Unlike conventional vapor-compression refrigeration, in which all condensing heat is rejected directly to the environment, an independent desuperheating/recovery heat-exchange unit is arranged at the compressor discharge end, so that rejected heat at a higher temperature with power-recovery potential is directed into the ORC, while the remaining heat is discharged by the main condenser or used for other low-temperature heat demands. The electrical power produced by the organic Rankine cycle is fed, through a generator and

power-electronics interface, into the direct-current bus or alternating-current side, thereby offsetting the electrical power required for active refrigeration at the system boundary. A complete energy path is thus formed: liquid-cooling heat extraction – vapor-compression refrigeration temperature lift – selective waste-heat recovery – organic Rankine cycle power compensation. As shown in Figure 1, the physical architecture and cross-domain energy-flow topology of the vapor-compression refrigeration–organic Rankine cycle coupled system intuitively present this cascade energy flow. By introducing a continuously adjustable recovery termination temperature, the segmented condensation heat-exchange unit is enabled to flexibly partition useful recoverable heat from waste heat, and the system-level advantage of the coupled architecture over conventional direct-rejection schemes is thereby highlighted.

The most fundamental cooling service is provided by the high-temperature liquid-cooling loop, and its operating objective is not merely to increase the return-liquid temperature; rather, the available heat-source temperature is to be increased as much as possible while constraints on chip junction temperature, cold-plate temperature, and local hot spots are satisfied. After chip heat is absorbed by the liquid-cooling medium, the medium enters the vapor-compression refrigeration evaporator, where the evaporation temperature is jointly determined by the coolant supply and return temperatures, the minimum pinch point of the evaporator, the coolant flow rate, and the compressor suction state. Therefore, a low evaporation-temperature setpoint decoupled from the high-temperature liquid-cooling boundary should not be adopted. The maximum chip temperature or maximum cold-plate temperature is treated as a hard constraint, and a vapor-compression refrigeration evaporation-temperature search

interval is set within the allowable range of the equipment, so that the cold-side temperature level can reduce the compressor pressure ratio without eroding the thermal safety margin of the chips. With this treatment, both the heat-source temperature obtained by the organic Rankine cycle and the compression power required by the vapor-compression refrigeration can be calculated from the actual liquid-cooling boundary, and apparent thermodynamic benefits obtained by artificially enlarging the temperature difference are avoided.

The vapor-compression refrigeration subcycle consists of an evaporator, a variable-frequency compressor, a desuperheater, a main condenser, and an electronic expansion valve. Heat released by the liquid-cooling loop is absorbed by the refrigerant in the evaporator; after pressure boosting by the compressor, a high-temperature superheated vapor is formed, which is then cooled and condensed sequentially through the desuperheater and the main condenser. The purpose of the segmented condensation structure is not simply to mechanically separate the superheated section from the saturation section; rather, an adjustable physical interface is provided for the heat-recovery boundary. The higher-temperature heat exchange is preferentially handled by the desuperheater, and when the optimized T_{cut} is lower than the saturated-vapor temperature, part of the two-phase condensation heat can be progressively included by enlarging the recovery heat-exchange region or by using an adjustable bypass; the remaining heat below T_{cut} continues to enter the main condenser. Consequently, the vapor-compression refrigeration condensation side is transformed from a fixed binary all-recovery/no-recovery structure into a continuously adjustable heat source, providing a clear equipment basis for subsequent investigation of the marginal recovery benefit.



(a) Physical architecture of the vapor-compression refrigeration–organic Rankine cycle coupled system

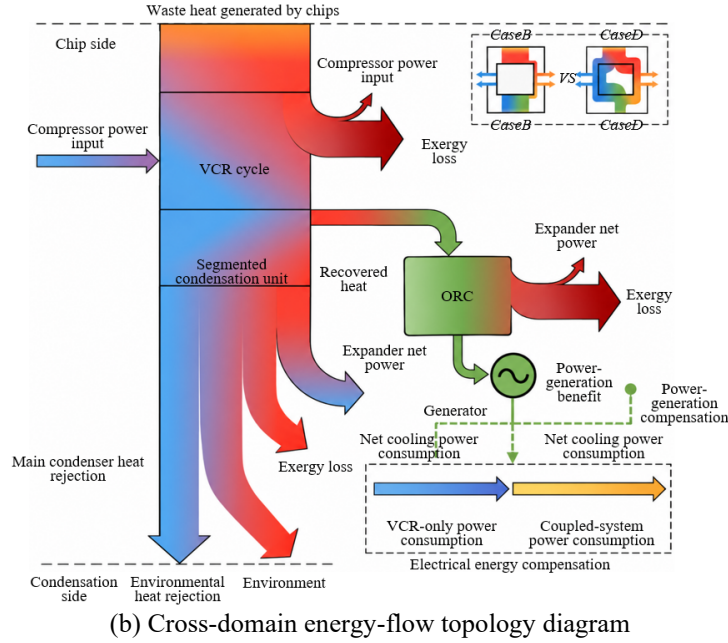


Figure 1. Physical architecture and cross-domain energy-flow topology of the vapor-compression refrigeration–organic Rankine cycle coupled system

The organic Rankine cycle subcycle is composed of a working-fluid pump, an evaporator/coupling heat exchanger, an expander, a generator, and an organic Rankine cycle condenser. The liquid organic Rankine cycle working fluid is pressurized by the pump and then absorbs vapor-compression refrigeration-rejected heat and evaporates in the coupling heat exchanger. After expansion power is produced in the expander, electrical power is generated by driving the generator, and a closed cycle is completed through the condenser and working-fluid pump. Electrical coupling, rather than direct mechanical belt coupling, is used, so that expander output power, generator efficiency, compressor electrical power, and pump and auxiliary-equipment power consumption can be measured and verified separately; modular matching among different vapor-compression refrigeration and organic Rankine cycle components is also facilitated. The electrical-side output power of the organic Rankine cycle is determined jointly by expander mechanical power, generator efficiency, and power-electronics conversion efficiency, where η_{gen} and η_{pe} denote generator efficiency and power-electronics interface efficiency, respectively. System evaluation is ultimately performed uniformly using the net electrical power $P_{ORC,net}$ after deduction of the organic Rankine cycle working-fluid pump and necessary auxiliary equipment so that overestimation of power-compensation effectiveness caused by substituting expander shaft power for actually usable electrical power is avoided. The electrical-side conversion relationship is expressed as follows:

$$P_{ORC,e} = W_{exp} \eta_{gen} \eta_{pe} \quad (1)$$

To avoid direct comparison of the vapor-compression refrigeration–organic Rankine cycle coupled system with schemes under different cooling-service conditions, four baseline configurations are established, and all schemes are required to first satisfy identical chip thermal-safety and supply-liquid temperature boundaries. Case A denotes direct heat rejection to the environment after high-temperature liquid cooling, and is used to identify the region in which free cooling

or natural heat dissipation can bear the entire thermal load. Case B is vapor-compression refrigeration-only, which serves as the most direct engineering baseline under active-refrigeration conditions. Case C denotes direct introduction of liquid-cooling waste heat into the organic Rankine cycle, and is used to determine whether sufficient power-generation value is already possessed by the waste heat without vapor-compression refrigeration temperature lift. Case D is the vapor-compression refrigeration–organic Rankine cycle scheme investigated herein. Under equivalent cooling service, the engineering value of the coupled architecture is established only when Case A or Case C cannot satisfy the thermal-safety/environmental boundary and Case D can reduce net cooling power consumption relative to Case B. The system configurations and comparison purposes of the four schemes are listed in Table 1.

The key to selective waste-heat recovery is not simply to determine whether heat exchange is possible within a given temperature segment, but to determine whether the net system power benefit continues to increase when additional lower-temperature heat is further supplied to the organic Rankine cycle. For a single-pressure organic Rankine cycle, a decrease in the minimum heat-source temperature often suppresses the allowable evaporation temperature and the cycle-average heat-absorption temperature, so that the net electrical power produced per unit of recovered heat gradually decreases. At the same time, the UA and area required by the heat exchanger continue to increase in order to utilize low-temperature heat near the pinch point. Therefore, the marginal net power benefit, $MB(T)$, corresponding to incremental recovered heat near temperature T is used to describe this variation. As T_{cut} decreases from the high-temperature end, Q_{rec} increases continuously, but $MB(T)$ is not necessarily always positive. The critical recovery temperature $T_{cut,opt}$ adopted by the system should be determined by the combined variations in the net cooling power consumption ratio, organic Rankine cycle net power, and UA , rather than being artificially fixed at the phase interface between the superheated section and the saturation section.

Table 1. Four baseline schemes and their comparison purposes

Scheme	System Configuration	Purpose
Case A	High-temperature liquid cooling + direct heat rejection (dry cooler/cooling tower)	To determine whether active refrigeration is necessary, serving as the lowest-complexity baseline
Case B	High-temperature liquid cooling + vapor-compression refrigeration	To quantify the energy consumption of the active-refrigeration unit itself
Case C	High-temperature liquid cooling + direct organic Rankine cycle (if permitted by the heat-source temperature)	To compare the system-level benefit of direct organic Rankine cycle with that of recovery after vapor-compression refrigeration temperature lift
Case D	High-temperature liquid cooling + vapor-compression refrigeration + selective organic Rankine cycle recovery (proposed scheme)	To quantify the actual compensation of vapor-compression refrigeration power consumption by the organic Rankine cycle and its applicable range

With this treatment, the optimal recovery boundary can vary with chip load, ambient temperature, vapor-compression refrigeration condensing pressure, and working-fluid pair, which better matches the continuous operating characteristics of an actual coupled system. $MB(T)$ is defined as:

$$MB(T) = \frac{dP_{ORC,net}}{dQ_{rec}} \quad (2)$$

3. THERMODYNAMIC MODELING AND COORDINATED OPTIMIZATION

The thermodynamic model is used to describe steady-state energy conversion, parameter coupling, and system optimization, and is calibrated against experimental platform data for key component efficiencies, pressure drops, and heat-exchanger UA . In the baseline model, each main component is assumed to operate under a single steady-state condition; pipeline heat loss and local pressure drop are neglected in the initial calculation and are corrected during model validation by means of measured pressure-drop coefficients and UA . The throttle valve is treated as an isenthalpic process; the compressor and expander are modeled using isentropic efficiency models that vary with pressure ratio, rotational speed, and working fluid; and the heat exchanger must satisfy the minimum pinch temperature difference as well as the maximum allowable pressure and temperature constraints of the equipment. The enthalpy, entropy, density, saturation pressure, and critical parameters of all state points are calculated uniformly by NIST REFPROP 10.0, and a consistent reference state is maintained. In addition to thermodynamic performance, candidate working fluids are required to simultaneously satisfy requirements related to global warming potential, safety class, material compatibility, flammability, equipment pressure containment, and market availability, so that the optimization results remain feasible for subsequent experimental implementation and engineering scale-up.

The vapor-compression refrigeration model is based on evaporator heat absorption, compressor power consumption, and cycle refrigeration performance. The refrigerant is throttled at state 4 and enters the evaporator, where it is evaporated to state 1 under the action of the liquid-cooling-side heat source, and is then compressed to state 2 by the compressor. The evaporator heat absorption and the actual compressor power are calculated from the refrigerant mass flow rate $\dot{m}_{VCR}$ and the specific enthalpy h_i at each state point, and the vapor-compression refrigeration coefficient of performance is obtained from these two quantities. The compressor isentropic efficiency is not taken as a fixed

constant but is expressed as $\eta_{comp,is} = f(PR, N, fluid)$, and is preferentially obtained from equipment performance maps, experimental calibration, or reliable semi-empirical models, so that the combined effects of pressure ratio, rotational speed, and working fluid on compression efficiency can still be reflected when the condensing and evaporating temperatures vary over a wide range. By this treatment, underestimation of the power-consumption penalty at high condensing temperatures due to a fixed efficiency is avoided, and a consistent calculation basis is provided for subsequently comparing the trade-off between increasing the condensing temperature to obtain higher organic Rankine cycle heat-source quality and the additional compressor power consumption. The corresponding energy relationships are:

$$\dot{Q}_{eva,VCR} = \dot{m}_{VCR}(h_1 - h_4) \quad (3)$$

$$\dot{W}_{comp} = \dot{m}_{VCR}(h_2 - h_1) \quad (4)$$

$$COP_{VCR} = \frac{\dot{Q}_{eva,VCR}}{\dot{W}_{comp}} \quad (5)$$

To describe the continuous heat recovery process on the vapor-compression refrigeration condensation side, a recovery termination temperature T_{cut} is introduced. The compressor discharge is cooled from state 2 in the recovery heat-exchange unit, and heat supply to the organic Rankine cycle is terminated when the temperature drops to state 2 corresponding to T_{cut} ; the remaining rejected heat is then transferred to the main condenser. T_{cut} can be higher than the saturated vapor temperature, corresponding to utilization of only part of the superheated sensible heat; it can be equal to the saturation temperature, corresponding to full utilization of the superheated section; or it can extend into the two-phase region to examine partial latent heat recovery. Each time T_{cut} is changed, the heat-source temperature trajectory on the vapor-compression refrigeration side is altered, so the organic Rankine cycle evaporation pressure, working-fluid flow rate, and pinch-point location must be re-solved, and heat cannot be mechanically superimposed on the original organic Rankine cycle condition. Through this continuous modeling, it can be determined whether the optimal recovery boundary lies near the superheated endpoint, within the two-phase region, or migrates toward higher temperature regions with load and ambient conditions. For a given condensing pressure, the recoverable heat and remaining rejected heat are expressed respectively as:

$$\dot{Q}_{rec}(T_{cut}) = \dot{m}_{VCR}[h_2 - h(T_{cut}, p_{cond})] \quad (6)$$

$$\dot{Q}_{rej} = \dot{Q}_{cond,total} - \dot{Q}_{rec} \quad (7)$$

On the organic Rankine cycle side, a basic cycle structure consisting of pump–evaporator–expander–condenser is adopted. The condensate at state 8 is pressurized to state 5 by the working-fluid pump, heat is absorbed to state 6 in the vapor-compression refrigeration–organic Rankine cycle coupling heat exchanger, and the fluid is then expanded to state 7 in the expander to output shaft power. To maintain uniformity in the comparison of different working-fluid pairs, the organic Rankine cycle net output is defined as electrical-side net power, and the power consumption of the working-fluid pump and organic Rankine cycle auxiliary equipment is explicitly deducted. The organic Rankine cycle evaporation pressure is not an independently and freely set variable; rather, it is jointly constrained by the vapor-compression refrigeration discharge temperature, T_{cut} , the T–Q curves on the hot and cold sides, the minimum pinch point, the expander outlet state, and the critical temperature of the working fluid. For different candidate working fluids, it must also be checked whether the inlet volumetric flow rate, expansion ratio, and outlet dryness or superheat fall within the allowable operating range of the expander so that a mathematical optimum with high thermal efficiency but unachievable by actual equipment is avoided. The main energy relationships are:

$$\dot{W}_p = \dot{m}_{ORC}(h_5 - h_8) \quad (8)$$

$$\dot{Q}_{ORC,in} = \dot{m}_{ORC}(h_6 - h_5) \quad (9)$$

$$\dot{W}_{exp} = \dot{m}_{ORC}(h_6 - h_7) \quad (10)$$

$$P_{ORC,net} = \dot{W}_{exp} \eta_{gen} \eta_{pe} - \dot{W}_p - \dot{W}_{aux,ORC} \quad (11)$$

The vapor-compression refrigeration–organic Rankine cycle coupling heat exchanger is the core component in which thermodynamic coupling between the two subcycles occurs, and its physical feasibility is first determined by the temperature trajectories of the hot and cold fluids and by the minimum pinch point. The heat-exchange process is discretized into segments according to thermal load, and the total UA is obtained by relating the heat transfer rate of each segment to the local logarithmic mean temperature difference and summing the contributions so that the heat-exchanger scale required by different T_{cut} values and working-fluid combinations is quantified. On the basis of energy conservation, exergy analysis is further employed to identify irreversible losses in the system: the exergy loss of adiabatic steady-state components is determined by the ambient temperature and entropy generation rate, whereas for the coupling heat exchanger, heat-transfer irreversibility is calculated from the complete entropy changes on the hot and cold sides. The pinch temperature difference, UA , and exergy loss serve different evaluation functions in the present framework. A smaller heat-transfer temperature difference is generally beneficial for reducing heat-transfer irreversibility but may significantly increase the heat-exchange area; therefore, a small pinch point cannot be directly interpreted as a large exergy loss. By calculating UA and the exergy loss of each component simultaneously, equipment-scale cost and thermodynamic perfection can be reflected separately, and the combined value under their joint effect is evaluated through net system power consumption. The corresponding

expressions are:

$$UA = \sum_i \frac{\Delta \dot{Q}_i}{\Delta T_{lm,i}} \quad (12)$$

$$\dot{I}_k = T_0 \dot{S}_{gen,k} \quad (13)$$

$$\dot{I}_{HX} = T_0 [\dot{m}_h(s_{h,out} - s_{h,in}) + \dot{m}_c(s_{c,out} - s_{c,in})] \quad (14)$$

To enable direct comparison of different configurations under the same cooling-service boundary, net cooling power consumption is adopted as the primary core quantity in system-level evaluation. The electrical power of the vapor-compression refrigeration compressor, liquid-cooling pump, vapor-compression refrigeration-side pump, and other auxiliary equipment is included in the system input, while the organic Rankine cycle electrical-side net power is deducted from it. On this basis, the net cooling power consumption ratio and cooling power compensation rate are defined. The net cooling power consumption ratio represents the net electrical power required per 1 kW of chip heat removed; a lower value indicates a smaller power demand for accomplishing an equivalent cooling task. The cooling power compensation rate represents the proportion of total active-cooling power consumption that can be covered by the organic Rankine cycle net electrical power, and is used to quantitatively describe power compensation, thereby avoiding the description of a system as self-cooling when the refrigeration power consumption is not yet fully covered. When linkage with data-center energy-efficiency metrics is required, cooling partial power usage effectiveness is further adopted only under the conditions that the P_{IT} measurement boundary is clear and the energy consumption of other facilities, such as power distribution, is not mixed into the cooling subsystem; this metric reflects only the additional energy consumption of the cooling subsystem relative to the information technology load and is strictly distinguished from the full data-center power usage effectiveness. The metrics are defined as follows:

$$\dot{W}_{net,cool} = \dot{W}_{comp} + \dot{W}_{p,VCR} + \dot{W}_{p,coolant} + \dot{W}_{aux} - P_{ORC,net} \quad (15)$$

$$NCPR = \frac{\dot{W}_{net,cool}}{\dot{Q}_{chip}} \quad (16)$$

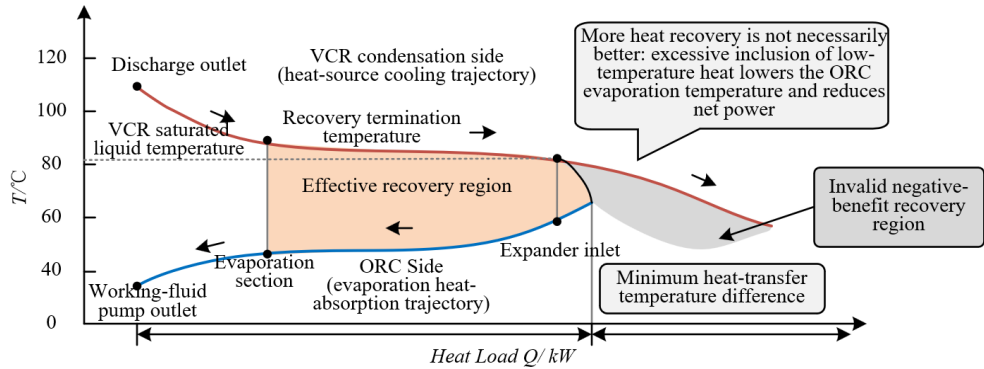
$$CPCR = \frac{P_{ORC,net}}{\dot{W}_{comp} + \dot{W}_{p,VCR} + \dot{W}_{p,coolant} + \dot{W}_{aux}} \quad (17)$$

$$pPUE_{cooling} = 1 + \frac{\dot{W}_{net,cool}}{P_{IT}} \quad (18)$$

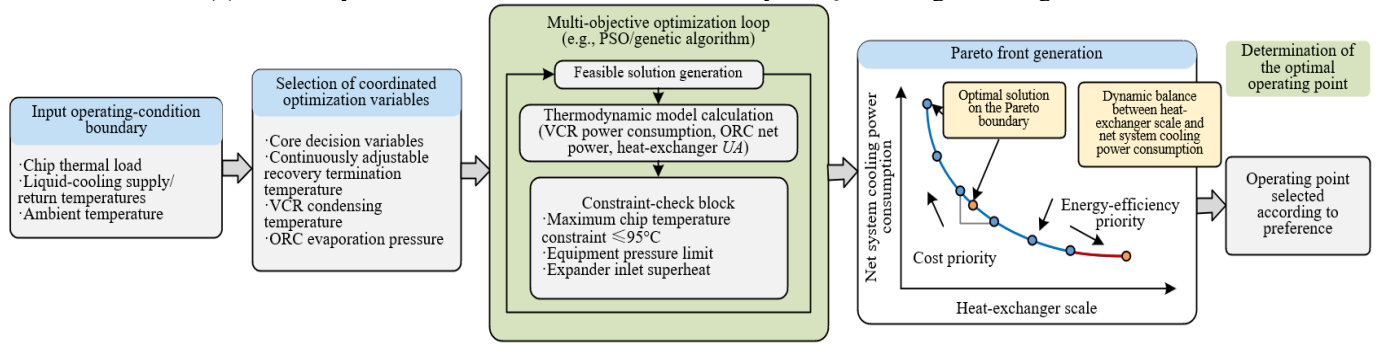
In selective recovery optimization, T_{cut} is treated as the core decision variable and is solved jointly with the vapor-compression refrigeration evaporation temperature, vapor-compression refrigeration condensing temperature, organic Rankine cycle evaporation pressure, and working-fluid flow rates on both sides. For any given T_{cut} , constraints on chip thermal safety, equipment pressure, heat-exchange pinch point, and expander operating region are first checked by the model, after which the system net cooling power consumption ratio, organic Rankine cycle net power, and UA are calculated. If only net cooling power consumption ratio minimization is pursued, a scheme with an excessively large heat-exchanger

area may be obtained; therefore, in addition to the single-objective optimum, a Pareto relationship between the net cooling power consumption ratio and UA is constructed, and the trade-off between thermodynamic performance and equipment scale is described by a heat-exchanger scale penalty

parameter λ_{UA} . To reveal the underlying mechanism of this trade-off, the T–Q matching relationship for selective recovery and the logic of the coordinated optimization algorithm are presented in Figure 2.



(a) Thermodynamic mechanism of selective recovery: T–Q matching and marginal benefit



(b) Logic of the Pareto-front-based multi-objective coordinated optimization algorithm

Figure 2. Flowchart of selective recovery T–Q matching and Pareto multi-objective coordinated optimization

The thermodynamic reason why excessive recovery of low-temperature heat leads to a negative marginal net power benefit is intuitively clarified in this figure, and the manner in which the optimal operating point is locked between heat-exchanger scale and net system cooling power consumption by means of the Pareto front is also shown. The final $T_{cut,opt}$ can be taken either at the minimum net cooling power consumption ratio point or in an advantageous region on the Pareto front where the net power benefit is near its limit while UA has not yet increased rapidly, so that the critical recovery temperature possesses both a clear energy-efficiency objective and engineering implementation significance. The optimization objective is written as:

$$\min J = NCPR + \lambda_{UA} UA_{norm} \quad (19)$$

Working-fluid screening is performed using a “working-fluid pair” rather than by independently optimizing the two subcycles. On the vapor-compression refrigeration side, attention must be paid to evaporation/condensation pressures, compression ratio, discharge temperature, volumetric refrigeration capacity, and environmental and safety attributes; on the organic Rankine cycle side, the critical temperature, evaporation pressure, expansion ratio, expander volumetric flow rate, outlet state, and net electrical power must be examined simultaneously. More importantly, the minimum pinch point and UA are directly determined by the T–Q trajectories of the two working fluids in the coupling heat exchanger; therefore, a working fluid that performs excellently in a standalone vapor-compression refrigeration or organic Rankine cycle may lose its advantage in the coupled system because of mismatched temperature trajectories.

Table 2. Collaborative evaluation dimensions for vapor-compression refrigeration–organic Rankine cycle working-fluid pairs

Dimension	Evaluation metric	Physical meaning
Heat-source matching	ΔT_{min} , T–Q curve deviation, and UA	To determine whether the heat-source and organic Rankine cycle temperature trajectories are matched
Vapor-compression refrigeration performance	Pressure ratio, discharge temperature, and W_{comp}	To determine the temperature-lift penalty
Organic Rankine cycle performance	Expansion ratio, outlet state, and $P_{ORC,net}$	To determine the power-conversion capability per unit heat
System performance	Net cooling power consumption ratio, cooling power compensation rate, and total exergy loss	To determine the system-level net benefit
Engineering constraints	Maximum pressure, critical temperature, safety class, and global warming potential	To determine equipment and environmental feasibility

These factors are uniformly incorporated into the evaluation dimensions listed in Table 2, and the net cooling power consumption ratio, $P_{ORC,net}$, UA , and equipment compatibility are used as the final ranking basis.

The complete solution procedure is cyclically executed in the order of “specified external boundary — vapor-compression refrigeration state solution — segmented heat-source generation — organic Rankine cycle optimization — equipment and pinch constraint checking — system metric calculation.” First, the chip thermal load, liquid-cooling supply/return targets, ambient cooling boundary, and candidate working-fluid pair are input, and the mass flow rate, compression power, and discharge state of the vapor-compression refrigeration are obtained at given evaporation/condensation temperatures. Subsequently, T_{cut} is swept, the recoverable heat and temperature trajectory on the vapor-compression refrigeration side are established, and the organic Rankine cycle evaporation pressure and flow rate are re-optimized for each heat-source boundary. Finally, UA , exergy loss, net cooling power consumption ratio, and cooling power compensation rate are calculated, and schemes that fail to satisfy the chip temperature, compressor discharge temperature, equipment pressure, pinch-point, or expander operating-region requirements are eliminated. Through this layer-by-layer constraint enforcement rather than isolated parameter scanning, a feasible operating domain jointly determined by $T_{eva,VCR} - T_{cond,VCR} - T_{cut}$ and the working-fluid pair can be formed, thereby providing a unified calculation basis for the parameter sensitivity analysis, working-fluid matching, and engineering operating map presented in Section 5.

4. EXPERIMENTAL SYSTEM AND VALIDATION METHOD

The experimental platform is composed of a chip-simulated

heat source, a high-temperature liquid-cooling loop, a vapor-compression refrigeration loop, a desuperheater/main condenser, an organic Rankine cycle loop, a cooling-water system, an electrical power measurement system, and a data acquisition system. A controllable electrical heating module is adopted for the chip-simulated heat source, and multipoint temperature measurement is configured to obtain the average temperature, peak temperature, and temperature uniformity. The organic Rankine cycle expander is connected to the generator by electrical coupling. Electrical power measurements are separately arranged for the vapor-compression refrigeration compressor, liquid-cooling pump, organic Rankine cycle pump, generator, and power-electronics interface, so that complete system energy balance closure can be established. The model, range, accuracy, sampling frequency, and calibration method of the measuring instruments are determined by the actual experimental platform configuration and are maintained consistent in the experimental records and result tables. Table 3 shows the main measured parameters, instrument types, and technical requirements.

In experimental data processing, uncertainty propagation analysis is first performed, and energy closure errors for key heat flows and electrical powers are reported. Heat quantities independently calculated on both sides of the vapor-compression refrigeration evaporator and the organic Rankine cycle evaporator are used for cross-checking.

$$\varepsilon_{balance} = \frac{|\dot{Q}_{hot} - \dot{Q}_{cold}|}{(\dot{Q}_{hot} + \dot{Q}_{cold})/2} \quad (20)$$

Only operating conditions for which the energy closure error of the key heat exchangers falls within a preset acceptable range are included in model validation and result analysis. Table 4 shows the experimental matrix and evaluation objectives for the vapor-compression refrigeration–organic Rankine cycle coupled system.

Table 3. Main measured parameters, instrument types, and technical requirements

Measured Quantity	Instrument Type	Key Requirement
Temperature	T-type/K-type thermocouple or Pt100	Multipoint temperature measurement is performed at working-fluid inlets/outlets and on chip surfaces; calibration uncertainty is included in error propagation.
Pressure	Piezoresistive pressure sensor	High- and low-pressure sides of the vapor-compression refrigeration and organic Rankine cycle are covered; actual uncertainty is converted from the full-scale error.
Mass flow rate	Coriolis flowmeter	Separate measurements are performed for the vapor-compression refrigeration and organic Rankine cycle.
Liquid-cooling flow rate	Electromagnetic/Coriolis flowmeter	Used for independent verification of $\dot{Q}_{chip}$.
Electrical power	High-precision power analyzer	Electrical power of the compressor, pumps, and generator/inverter is measured separately.
Rotational speed/torque (optional)	Encoder + torque sensor	Used for decomposition of expander mechanical-to-electrical efficiency.

Table 4. Experimental matrix and evaluation objectives for the vapor-compression refrigeration–organic Rankine cycle coupled system

Experiment	Control Variable	Main Output	Purpose
E1 Architecture baseline comparison	Identical $\dot{Q}_{chip}$ and ambient/cooling boundaries; Cases A–D	$W_{net,cool}$, net cooling power consumption ratio, cooling power compensation rate, and $P_{ORC,net}$	To verify the boundaries under which a net benefit is yielded by the vapor-compression refrigeration–organic Rankine cycle architecture
E2 Selective recovery	T_{cut} or bypass ratio is varied stepwise.	$\dot{Q}_{rec}$, $P_{ORC,net}$, the net cooling power consumption ratio, $UA/\Delta T_{min}$	To determine the critical recovery temperature and marginal-benefit attenuation

E3 Condensing-temperature sweep	$T_{cond,VCR}$ at multiple levels; other boundaries fixed	W_{comp} , T_{dis} , $P_{ORC,net}$, and the net cooling power consumption ratio	To reveal the competition between temperature-lift cost and power-generation benefit
E4 Evaporation temperature/thermal safety	$T_{eva,VCR}$ at multiple levels; upper T_{chip} constraint	COP_{VCR} , $T_{chip,max}$, and the net cooling power consumption ratio	To obtain the thermal-efficiency–chip-thermal-safety feasible domain
E5 Working-fluid/working-fluid pair validation	Representative vapor-compression refrigeration/organic Rankine cycle working-fluid combinations	ΔT_{min} , pressure ratio, expansion ratio, $P_{ORC,net}$, and the net cooling power consumption ratio	To validate the collaborative screening model
E6 Dynamic thermal load	Q_{chip} step or periodic variation	Response time, temperature overshoot, and power tracking	To validate load adaptability
E7 Long-term stability	Continuous operation under optimal conditions for ≥ 72 hours (if conditions permit)	Mean, standard deviation, drift, and leak/fault records	To validate engineering stability

Model parameters are identified from selected steady-state operating conditions, including UA , pressure drop, and component-efficiency correction parameters. In the validation set, combinations of condensing temperature, evaporation temperature, and thermal load that are independent of the calibration set are adopted, and mean absolute percentage error and root mean square error are reported separately for temperature, pressure, W_{comp} , $P_{ORC,net}$, and the net cooling power consumption ratio.

5. RESULTS AND DISCUSSION

On the basis of consistency verification between the experimental measurement chain and the thermodynamic model, the system configuration, selective waste-heat recovery boundary, vapor-compression refrigeration operating temperatures, working-fluid matching, dynamic load, and long-term operating characteristics are comprehensively discussed in this section. Parametric extension results are obtained by calculation using the validated model. Complete operating-condition data are listed in the Appendix, while representative results that reflect performance variation trends, key transitions, and applicable boundaries are focused on in the main text.

5.1 Model reliability and experimental uncertainty

The credibility of parameter extension and operating-boundary analysis is directly determined by model reliability; accordingly, the model is examined at three levels: prediction error, energy closure error, and uncertainty of key measured quantities. Under independent validation conditions, mean absolute percentage error values for compressor power W_{comp} , organic Rankine cycle net output power $P_{ORC,net}$, maximum chip temperature $T_{chip,max}$, and net cooling power consumption ratio are obtained as 2.8%, 4.2%, 1.1%, and 3.4%, respectively, with corresponding root-mean-square error values of 0.024 kW, 0.010 kW, 0.92 °C, and 0.009. The mean absolute percentage error for each key output is controlled within 5%; the relative error of organic Rankine cycle net output power is slightly higher, mainly because of its relatively small absolute magnitude, whereas its root mean square error of 0.010 kW remains low. Average energy closure errors for the vapor-compression refrigeration evaporator and the organic Rankine cycle coupling heat exchanger are calculated as 3.1% and 4.0%, respectively, indicating that measurement errors in mass flow rate, temperature, and pressure do not cause pronounced system energy imbalance.

Prediction errors and energy closure results are found to be

consistent, indicating that the coupling relationships among vapor-compression refrigeration compression power, organic Rankine cycle power recovery, and chip temperature can be stably reproduced by the established model. In particular, net cooling power consumption ratio, as the core evaluation metric of system net benefit, is maintained at a low prediction error, and a reliable basis is thereby provided for expanding the parameter space of $T_{cond,VCR}$, $T_{eva,VCR}$, T_{cut} , and working-fluid combinations. Subsequent discussion is therefore extended from a single operating point to multi-parameter coupling regions, so that inflection points, constraint boundaries, and preferred operating intervals in performance variation can be identified.

5.2 Thermodynamic feasibility of the coupled architecture: Cases A–D

The comparison among the four system configurations is premised on identical cooling-service boundaries. Under the specified high ambient temperature condition, auxiliary power consumption for direct liquid-cooling heat rejection in Case A is only 0.13 kW, and the net cooling power consumption ratio is 0.043, whereas the maximum chip temperature reaches 101.2 °C. In Case C, after liquid-cooling waste heat is used to directly drive the organic Rankine cycle, net cooling power consumption is further reduced to 0.10 kW, and the net cooling power consumption ratio is 0.033, but $T_{chip,max}$ still reaches 99.8 °C. Although lower apparent power consumption is exhibited by the two schemes, the 95 °C thermal safety limit is exceeded by both, and therefore direct energy-efficiency ranking against active-refrigeration schemes capable of stably maintaining the target chip temperature cannot be performed.

After active temperature control is introduced through the vapor-compression refrigeration, the maximum chip temperature in Case B is reduced to 82.3 °C, but 0.90 kW of electrical power is consumed by the compressor and auxiliary equipment, corresponding to a net cooling power consumption ratio of 0.300. When selective organic Rankine cycle recovery is configured under identical vapor-compression refrigeration operating boundaries, an organic Rankine cycle net output power of 0.21 kW is achieved in Case D so that net system cooling power consumption is reduced from 0.90 kW to 0.69 kW, the net cooling power consumption ratio is correspondingly decreased to 0.230, and the maximum chip temperature is maintained at 82.5 °C. The compensation rate of active-cooling power consumption by the organic Rankine cycle reaches 23.3%, and net cooling power consumption is reduced by 23.3% relative to the vapor-compression refrigeration-only scheme. These results indicate that the advantage of the vapor-compression refrigeration–organic

Rankine cycle coupled system has a clear boundary: when thermal safety can be satisfied by free cooling, a vapor-compression refrigeration temperature-lift stage should not be artificially introduced; only when active refrigeration becomes a necessary condition because of ambient conditions or supply-liquid temperature requirements does power recovery from vapor-compression refrigeration rejected heat possess a stable system-level value.

5.3 Selective recovery strategy and critical recovery temperature

The temperature range of vapor-compression refrigeration rejected heat that is incorporated into the organic Rankine cycle is determined by the recovery termination temperature T_{cut} . As T_{cut} is gradually reduced from 85 °C to 68 °C, the recoverable heat Q_{rec} is increased from 0.55 kW to 1.78 kW, the organic Rankine cycle net output power is raised from 0.080 kW to 0.225 kW, and the net cooling power consumption ratio is continuously decreased from 0.273 to 0.225. Within this range, the incremental heat can still be converted into effective net power; therefore, the system-level net energy efficiency can be continuously improved by expanding the recovery temperature interval. However, as lower-temperature heat is progressively included, the organic Rankine cycle evaporation temperature is decreased from 70 °C to 56 °C, and the marginal net power benefit per unit of incremental heat is attenuated accordingly, being reduced to approximately 0.031 at $T_{cut} = 68$ °C.

When T_{cut} is further reduced to 65 °C and 62 °C, the recoverable heat is increased to 2.05 kW and 2.28 kW, respectively, but the organic Rankine cycle net output power is instead decreased to 0.220 kW and 0.205 kW, and the net cooling power consumption ratio is increased again to 0.227 and 0.232. At the same time, the required heat-exchanger UA is rapidly increased from 118 W/K to 165 W/K and 240 W/K, and the marginal benefit MB is changed to -0.019 and -0.065 , respectively. Although the total amount of recovered heat is increased by the additional low-temperature heat, the average organic Rankine cycle heat-absorption temperature is simultaneously lowered and the heat-exchanger scale is significantly enlarged; consequently, effective system-level net power benefit can no longer be generated. Thus, approximately 68 °C is identified as a relatively clear critical recovery temperature under the present operating conditions.

It is revealed by further comparison of the exergy loss distributions under the $T_{cut} = 68$ °C and 62 °C operating conditions that the coupling heat-exchanger exergy loss is not increased by the inclusion of low-temperature heat; instead, it is slightly reduced from 0.070 kW to 0.062 kW, whereas the total system exergy loss is only slightly increased from 0.590 kW to 0.602 kW. The degradation in recovery performance within the low-temperature segment is therefore attributed not to a sharp increase in heat-transfer irreversibility induced by a small temperature difference, but to the weakened net power benefit of the organic Rankine cycle after the average heat-source temperature is lowered, together with the rapid increase in UA required to accomplish the transfer of additional low-temperature heat. The physical basis of selective recovery is thus manifested as a comprehensive balance among heat quality, marginal net power benefit, and heat-exchanger scale, rather than as a simple division between recoverable and non-recoverable heat based on the condenser phase-change boundary.

5.4 Competition mechanism between vapor-compression refrigeration temperature-lift cost and organic Rankine cycle power-generation benefit

The compressor discharge temperature and the organic Rankine cycle heat-source quality can be elevated by increasing the vapor-compression refrigeration condensing temperature, but additional compression power is simultaneously required, and whether a net system benefit can be obtained depends on whether the incremental organic Rankine cycle output power can cover the incremental vapor-compression refrigeration power. As the condensing temperature is raised from 60 °C to 70 °C, compressor power is increased from 0.65 to 0.80 kW, while organic Rankine cycle net output power is increased from 0.02 to 0.21 kW. Because the organic Rankine cycle power increment is larger than the compressor power increment, net system cooling power consumption is decreased from 0.73 to 0.69 kW, net cooling power consumption ratio is reduced from 0.243 to 0.230, and the power compensation rate is increased from 2.7% to 23.3%, indicating that actual system-level benefit can be generated by heat-source quality upgrading within this temperature range.

Once the condensing temperature exceeds 70 °C, the relationship between the two power increments is reversed. When the condensing temperature is raised to 75 °C, organic Rankine cycle net output power is further increased to 0.30 kW, but compressor power has already risen to 0.90 kW, net cooling power consumption rises back to 0.70 kW, and the incremental net benefit becomes negative for the first time. After further increases to 80 °C and 85 °C, compressor power reaches 1.04 and 1.22 kW, respectively, while organic Rankine cycle net power increases only to 0.36 and 0.39 kW, causing the net cooling power consumption ratio to rise to 0.260 and 0.310. Meanwhile, compressor discharge temperature is continuously increased from 78 °C under the 60 °C condition to 118 °C under the 85 °C condition, and equipment thermal load and reliability constraints are simultaneously intensified. Therefore, the system optimization objective should not be set as maximum organic Rankine cycle power generation; rather, minimum net electrical consumption for active cooling should be adopted as the criterion. Under the present operating conditions, the optimal balance between temperature-lift cost and power-generation benefit is identified at approximately 70 °C, and 68–72 °C constitutes a reasonable condensing-temperature operating interval.

$$\Delta P_{benefit} = \Delta P_{ORC,net} - \Delta W_{comp} \quad (21)$$

5.5 Feasible operating domain of evaporation temperature and chip thermal safety

A mechanism different from that of the condensing temperature is exerted by the vapor-compression refrigeration evaporation temperature on system performance. As $T_{eva,VCR}$ is increased from 25 °C to 45 °C, the suction pressure is elevated and the compression ratio is reduced, so compressor power is continuously decreased from 0.98 to 0.67 kW. Although organic Rankine cycle net output power is also gradually reduced from 0.24 to 0.15 kW, the compressor energy saving is larger, so net system cooling power consumption is decreased from 0.84 to 0.62 kW and the net cooling power consumption ratio is reduced from 0.280 to 0.207. From the perspective of net system energy

consumption, the vapor-compression refrigeration temperature-lift cost can be continuously reduced as the evaporation temperature is increased.

When the evaporation temperature is increased from 25 °C to 40 °C, the maximum chip temperature rises from 70.8 °C to 88.9 °C, which remains below the 95 °C limit. When the evaporation temperature is further increased to 45 °C, the maximum chip temperature reaches 96.8 °C, and the thermal safety boundary is exceeded. It can therefore be seen that the optimization direction of solely pursuing low net power consumption is altered by the chip thermal safety constraint. After the constraint $T_{chip,max} \leq 95$ °C is introduced, 40 °C becomes a more reasonable upper limit of the evaporation temperature within the present parameter range, corresponding to a net cooling power consumption of 0.65 kW and a net cooling power consumption ratio of 0.217. In a high-temperature liquid-cooling system, the vapor-compression refrigeration temperature-lift cost can be significantly reduced by increasing the evaporation temperature, but the available temperature-increase space is ultimately constrained by chip thermal safety; therefore, evaporation-temperature optimization must be performed synchronously with the chip temperature boundary.

5.6 Collaborative matching of vapor-compression refrigeration–organic Rankine cycle working-fluid pairs

The influence of working-fluid combinations on system performance cannot be judged by a single cycle efficiency or a single pressure metric. Among six representative working-fluid pairs, the lowest net cooling power consumption ratio, 0.222, is achieved when R1234yf is adopted on the vapor-compression refrigeration side and R1233zd(E) on the organic Rankine cycle side, and the highest power compensation rate, 24.4%, is simultaneously obtained. Under this combination, compressor power is 0.78 kW, organic Rankine cycle net output power reaches 0.215 kW, heat-exchanger UA is 115 W/K, the compression ratio and expansion ratio are 2.88 and 3.31, respectively, and the expander isentropic efficiency reaches 0.73. Although none of the individual metrics is absolutely optimal in isolation, a more coordinated overall matching relationship is formed at the system level.

By comparison, although the R134a/R1233zd(E) combination requires a slightly lower heat-exchanger UA of 112 W/K, compressor power is increased to 0.80 kW, and the final net cooling power consumption ratio is 0.230. When R1234ze(E) is adopted as the organic Rankine cycle working fluid, organic Rankine cycle net output power is markedly reduced regardless of whether R134a or R1234yf is used on the vapor-compression refrigeration side, and the net cooling power consumption ratio rises to 0.245 and 0.240, respectively. The superiority of a working-fluid pair is jointly determined by compression power, heat-source and evaporation-temperature trajectories, expansion ratio, expander compatibility, and heat-exchanger scale. The advantage of R1234yf/R1233zd(E) originates from the favorable system-level coordination among these factors rather than from being dominated by a single thermophysical parameter.

5.7 Dynamic thermal load response

Within a chip thermal load range of 1.0–4.0 kW, all energy branches of the system are enhanced as the load increases.

Compressor power is increased from 0.30 to 1.12 kW, organic Rankine cycle net output power is increased from 0.05 to 0.32 kW, and the power compensation rate is raised from 13.9% to 25.6%. As the chip thermal load increases, both the amount of rejected heat available to the organic Rankine cycle and the heat quality are simultaneously elevated; therefore, the compensation effect of the organic Rankine cycle on active-cooling power consumption is more pronounced in the medium-to-high load range. The net cooling power consumption ratio is rapidly decreased from 0.310 under the 1 kW condition, reaching 0.240 and 0.230 at 2 and 3 kW, respectively, indicating that the waste-heat recovery capability of the coupled system is not yet fully exploited under low-load conditions.

When the thermal load is further increased to 4 kW, although the cooling power compensation rate continues to increase to 25.6%, the net cooling power consumption ratio rises back to 0.233, and the previous downward trend is no longer maintained. Meanwhile, the maximum chip temperature is increased from 68.5 °C to 89.7 °C, and the steady-state response time is extended from 3.8 to 7.6 minutes. High load increases the recoverable organic Rankine cycle power, but also synchronously increases the compressor burden and the thermal inertia of the liquid-cooling loop; consequently, the net system energy efficiency gradually enters a plateau region. Pronounced benefit saturation characteristics are exhibited near 3–4 kW; when the thermal load is further increased, greater attention should be paid to the chip temperature margin and dynamic response speed, and operating quality should not be judged solely by the organic Rankine cycle power compensation rate.

5.8 Long-term stability and engineering energy efficiency

During 72 hours of continuous operation, the mean chip thermal load was maintained at 3.00 kW, with a standard deviation of 0.04 kW. The maximum chip temperature averaged 82.6 °C, with a standard deviation of 1.2 °C; the entire sequence ranged from 79.8 °C to 85.4 °C and remained consistently below the 95 °C thermal safety limit. Mean values of compressor power, organic Rankine cycle net output power, and auxiliary power were 0.800, 0.210, and 0.100 kW, respectively, corresponding to a mean net cooling power consumption of 0.690 kW with a standard deviation of 0.035 kW. The net cooling power consumption ratio was maintained at 0.230 ± 0.012 over the 72 hour period, and the cooling power compensation rate was $23.3\% \pm 1.4\%$, indicating that no pronounced attenuation or large fluctuation in organic Rankine cycle power recovery occurred during continuous operation.

Under identical cooling-service boundaries, with the 0.90 kW net cooling power consumption of Case B taken as the baseline, approximately 0.21 kW of continuous cooling power consumption was reduced by Case D, corresponding to a reduction of 23.3%. Based on an equivalent annual operating time of 8000 h/a, an annual cooling electricity saving of approximately 1680 kWh can be achieved for a single 3 kW thermal-load system. When calculated within the cooling-subsystem boundary, the mean $pPUE_{cooling}$ was 1.230, with a standard deviation of 0.012; this metric describes only the additional energy consumption of the cooling subsystem relative to the information technology thermal load and does not replace the full facility-level power usage effectiveness. The stable power compensation and temperature-control

capability demonstrated by the long-term sequence indicate that the engineering value of the vapor-compression refrigeration–organic Rankine cycle coupled scheme is mainly manifested in high-temperature environments or high supply-liquid temperature scenarios where active refrigeration has become a necessary condition, rather than in replacing low-energy-consumption conditions in which free cooling can be directly adopted.

$$E_{\text{save,annual}} = (W_{\text{net,cool,B}} - W_{\text{net,cool,D}}) t_{\text{annual}} \quad (22)$$

5.9 High-power-density scenarios and applicable boundaries

A gradual tightening of the system's applicable boundary with increasing power density is revealed by the load-extension results. Within the 1–4 kW range, the maximum chip temperature is increased from 68.5 °C to 89.7 °C, remaining within the 95 °C constraint, but the thermal safety margin has been markedly narrowed. At the same time, the steady-state response time is extended from 3.8 to 7.6 minutes, and the net cooling power consumption ratio, after reaching 0.230 at 3 kW, rises back to 0.233 at 4 kW. As power density is further increased, the system performance limitation will gradually be shifted from waste-heat recovery capability alone to a joint constraint imposed by chip thermal safety, compressor discharge temperature, heat-exchanger heat-transfer capacity, and dynamic thermal inertia.

Three operating regions with distinct technical characteristics are further formed by the aforementioned parameter sweeps. When chip thermal safety can be directly satisfied by external heat-dissipation conditions, Case A should be preferentially adopted in the free-cooling region so that unnecessary compression power is avoided. When the target supply-liquid temperature cannot be maintained by direct heat dissipation, the system enters the active vapor-compression refrigeration necessary region, in which Case B constitutes the minimum active-cooling baseline. Only when a positive marginal net power benefit is jointly satisfied by the vapor-compression refrigeration rejected-heat temperature, the recovery termination temperature, and the organic Rankine cycle conditions is Case D allowed to enter the effective organic Rankine cycle recovery region with system-level advantage. Under the current baseline conditions, this region is jointly constrained by a critical recovery temperature of approximately 68 °C, a preferred condensing temperature of approximately 70 °C, and a feasible evaporation temperature not exceeding approximately 40 °C. Thus, the system applicable boundary is not a fixed temperature point, but a multi-parameter operating domain jointly defined by heat quality, compression power cost, organic Rankine cycle marginal net power, heat-exchanger scale, and chip thermal safety.

6. CONCLUSIONS

Based on the computational/simulation data and the system model under the specified operating conditions, the following quantitative conclusions were obtained:

(i) Under the given high ambient temperature and supply-liquid temperature constraints, the equivalent cooling-service boundary of a maximum chip temperature ≤ 95 °C could not be satisfied by direct liquid-cooling heat rejection (Case A) or

direct liquid-cooling organic Rankine cycle (Case C). Thermal safety was satisfied by vapor-compression refrigeration-only (Case B), but a net cooling power consumption ratio of 0.300 was obtained. When selective organic Rankine cycle recovery was incorporated (Case D), the net cooling power consumption ratio was reduced to 0.230, and net cooling power consumption was reduced from 0.90 to 0.69 kW, corresponding to a reduction of 23.3%.

(ii) When the recovery termination temperature T_{cut} was continuously reduced, the recoverable heat was increased from 0.55 to 2.28 kW; however, the organic Rankine cycle net electrical power no longer increased after approximately 0.225 kW was reached at $T_{\text{cut}} \approx 68$ °C, and the corresponding minimum net cooling power consumption ratio was 0.225. When T_{cut} was further reduced below 65 °C, the marginal power-generation benefit became negative, while UA rapidly increased from 118 W/K to 165–240 W/K, indicating that further inclusion of lower-temperature condensation heat no longer provides a system-level advantage.

(iii) When the vapor-compression refrigeration condensing temperature was increased from 60 °C to 70 °C, the increment in organic Rankine cycle net electrical power was greater than the incremental compressor power consumption, and the net cooling power consumption ratio was reduced from 0.243 to 0.230. When the condensing temperature was further increased to 75 °C and above, the incremental net benefit $\Delta P_{\text{benefit}}$ became negative. Therefore, the optimal condensing temperature under the baseline condition was approximately 70 °C, and 68–72 °C could be adopted as the key search interval for subsequent experiments.

(iv) Under a condensing temperature of 70 °C, compressor power consumption could be continuously reduced by increasing the vapor-compression refrigeration evaporation temperature; however, when the evaporation temperature was increased from 40 °C to 45 °C, the maximum chip temperature rose from 88.9 °C to 96.8 °C, exceeding the 95 °C thermal safety constraint. Consequently, the thermodynamic optimum under the baseline condition was truncated by thermal safety, and a feasible evaporation temperature of approximately 35–40 °C was recommended, with a net cooling power consumption ratio of 0.217 at 40 °C.

(v) Among six representative working-fluid pairs, the lowest net cooling power consumption ratio of 0.222 and the highest cooling power compensation rate of 24.4% were obtained with the R1234yf/R1233zd(E) pair, while heat-exchanger UA was maintained at a relatively low level of 115 W/K. This advantage resulted from the combined effects of compression power, temperature-trajectory matching, expander compatibility, and heat-exchanger scale, and could not be simply attributed to a single pressure parameter.

(vi) In the 72-hour operating sequence, $Q_{\text{chip}} = 3.00 \pm 0.04$ kW, net cooling power consumption ratio = 0.230 ± 0.012 , cooling power compensation rate = $23.3\% \pm 1.4\%$, and cooling partial power usage effectiveness = 1.230 ± 0.012 . If the vapor-compression refrigeration-only net cooling power consumption of 0.90 kW was taken as the baseline and an annual equivalent operation of 8000 h was assumed, approximately 1680 kWh of cooling electricity could be saved annually by Case D.

REFERENCES

[1] Zhang, Y.L., Zhao, Y.Q., Dai, S.Y., et al. (2022).

- Cooling technologies for data centres and telecommunication base stations – A comprehensive review. *Journal of Cleaner Production*, 334: 130280. <https://doi.org/10.1016/j.jclepro.2021.130280>
- [2] Wang, N.B., Guo, Y.H., Huang, C.Q., Shao, S.Q. (2024). Advances in direct liquid cooling technology and waste heat recovery for data center: A state-of-the-art review. *Journal of Cleaner Production*, 477: 143872. <https://doi.org/10.1016/j.jclepro.2024.143872>
 - [3] Alkrush, A.A., Salem, M.S., Abdelrehim, O., Hegazi, A.A. (2024). Data centers cooling: A critical review of techniques, challenges, and energy saving solutions. *International Journal of Refrigeration*, 160: 246-262. <https://doi.org/10.1016/j.ijrefrig.2024.02.007>
 - [4] Heydari, A., Gharaibeh, A.R., Tradat, M., et al. (2024). Experimental evaluation of direct-to-chip cold plate liquid cooling for high-heat-density data centers. *Applied Thermal Engineering*, 239: 122122. <https://doi.org/10.1016/j.applthermaleng.2023.122122>
 - [5] Pambudi, N.A., Sarifudin, A., Firdaus, R.A., Ulfa, D.K., Gandidi, I.M., Romadhon, R. (2022). The immersion cooling technology: Current and future development in energy saving. *Alexandria Engineering Journal*, 61(12): 9509-9527. <https://doi.org/10.1016/j.aej.2022.02.059>
 - [6] Ljungdahl, V., Jradi, M., Veje, C. (2022). A decision support model for waste heat recovery systems design in Data Center and High-Performance Computing clusters utilizing liquid cooling and Phase Change Materials. *Applied Thermal Engineering*, 201: 117671. <https://doi.org/10.1016/j.applthermaleng.2021.117671>
 - [7] Zhou, W.N., Sun, Q., Luo, W.M., et al. (2023). Performance analysis and optimization of free cooling strategies for a liquid-cooled data center. *Building Simulation*, 16: 1317-1330. <https://doi.org/10.1007/s12273-023-1012-6>
 - [8] Aili, A., Long, W.J., Cao, Z.W., Wen, Y.G. (2024). Radiative free cooling for energy and water saving in data centers. *Applied Energy*, 359: 122672. <https://doi.org/10.1016/j.apenergy.2024.122672>
 - [9] Sun, X.Q., Zhang, C., Han, Z.W., et al. (2023). Experimental study on a novel pump-driven heat pipe/vapor compression system for rack-level cooling of data centers. *Energy*, 274: 127335. <https://doi.org/10.1016/j.energy.2023.127335>
 - [10] Yuan, X.L., Liang, Y.M., Hu, X.Y., Xu, Y.Z., Chen, Y.B., Kosonen, R. (2023). Waste heat recoveries in data centers: A review. *Renewable and Sustainable Energy Reviews*, 188: 113777. <https://doi.org/10.1016/j.rser.2023.113777>
 - [11] Zhang, Y.Q., Dong, J.X., Zhang, C., Han, Z.W. (2023). Dynamic model based study on mode switching characteristics and reliability analysis of data center composite air conditioning system. *Applied Thermal Engineering*, 227: 120442. <https://doi.org/10.1016/j.applthermaleng.2023.120442>
 - [12] Saleem, J., Chahrour, K.M.N. (2024). Enhancing the efficiency of air conditioning systems in high-temperature climates through direct evaporative cooling. *Journal of Sustainability for Energy*, 3(2): 119-138. <https://doi.org/10.56578/jse030205>
 - [13] Xia, X.X., Liu, Z.P., Wang, Z.Q., Zuo, Q.S., Sun, T. (2023). Energy, conventional and advanced exergy analysis for the organic Rankine cycle-vapor compression refrigeration combined system driven by low-grade waste heat. *Applied Thermal Engineering*, 220: 119650. <https://doi.org/10.1016/j.applthermaleng.2022.119650>
 - [14] Li, P., Xu, J.Q., Wang, B.B., Liu, J.Y., Zhao, W.S., Han, Z.H. (2025). Performance analysis of a coupled system based on organic Rankine cycle and double effect absorption refrigeration for waste heat recovery in data center. *Journal of Thermal Science*, 34: 188–205. <https://doi.org/10.1007/s11630-024-2043-8>
 - [15] Platt, B., Young, D., Bandhauer, T. (2025). Thermodynamic and turbomachinery analysis of a hybrid electric organic Rankine vapor compression system. *Applied Energy*, 387: 125554. <https://doi.org/10.1016/j.apenergy.2025.125554>
 - [16] Jiang, H.Y., Rong, Y., Zhou, X., et al. (2023). Performance assessment of an organic Rankine–Vapor compression cycle (ORC–VCR) for low-grade compression heat recovery. *Energy Conversion and Management*, 275: 116492. <https://doi.org/10.1016/j.enconman.2022.116492>
 - [17] Wang, Z.Q., Zhao, Y.B., Xia, X.X., et al. (2024). Experimental study of the thermodynamic performance of the ORC–VCC system with a zeotropic mixture. *Applied Thermal Engineering*, 250: 123534. <https://doi.org/10.1016/j.applthermaleng.2024.123534>
 - [18] Hao, Y.T., Zhou, H.J., Tian, T., et al. (2025). Data centers waste heat recovery technologies: Review and evaluation. *Applied Energy*, 384: 125489. <https://doi.org/10.1016/j.apenergy.2025.125489>
 - [19] Platt, B., Young, D., Bandhauer, T. (2025). Experimental validation of a hybrid electric organic Rankine vapor compression cooling system. *Applied Energy*, 392: 125956. <https://doi.org/10.1016/j.apenergy.2025.125956>
 - [20] Chen, X.T., Pan, M.Z., Li, X.Y., Zhang, K. (2022). Multi-mode operation and thermo-economic analyses of combined cooling and power systems for recovering waste heat from data centers. *Energy Conversion and Management*, 266: 115820. <https://doi.org/10.1016/j.enconman.2022.115820>
 - [21] Gupta, R., Puri, I.K. (2021). Waste heat recovery in a data center with an adsorption chiller: Technical and economic analysis. *Energy Conversion and Management*, 245: 114576. <https://doi.org/10.1016/j.enconman.2021.114576>
 - [22] Al-Sayyab, A.K.S., Mota-Babiloni, A., Navarro-Esbrí, J. (2023). Performance evaluation of modified compound organic Rankine-vapour compression cycle with two cooling levels, heating, and power generation. *Applied Energy*, 334: 120651. <https://doi.org/10.1016/j.apenergy.2023.120651>
 - [23] Liu, J., Xu, Y.T., Zhang, Y.N., Shuai, Y., Li, B.X. (2022). Multi-objective optimization of low temperature cooling water organic Rankine cycle using dual pinch point temperature difference technologies. *Energy*, 240: 122740. <https://doi.org/10.1016/j.energy.2021.122740>
 - [24] Miao, Z., Wang, Z.B., Varbanov, P.S., Klemeš, J.J., Xu, J.L. (2023). Development of selection criteria of zeotropic mixtures as working fluids for the trans-critical organic Rankine cycle. *Energy*, 278: 127811. <https://doi.org/10.1016/j.energy.2023.127811>
 - [25] Graubeger, A., Young, D., Bandhauer, T. (2022). Off-design performance of an organic Rankine-vapor compression cooling cycle using R1234ze(E). *Applied Energy*, 321: 119421.

- <https://doi.org/10.1016/j.apenergy.2022.119421>
- [26] Rezaei, R.A. (2023). Energy and exergy evaluation of a dual fuel combined cycle power plant: An optimization case study of the Khoi plant. *Power Engineering and Engineering Thermophysics*, 2(2): 97-109. <https://doi.org/10.56578/peet020204>
- [27] González, J., Llorell, F., Garrido, J.M., Quinteros-Lama, H. (2023). A study of the optimal conditions for organic Rankine cycles coupled with vapour compression refrigeration using a rigorous approach based on the Helmholtz energy function. *Energy*, 285: 129554. <https://doi.org/10.1016/j.energy.2023.129554>
- [28] Vasković, S., Gvero, P., Montel, N., Marijanović, I. (2025). Calculation and optimization of biomass energy production by the dignet energy platform. *Power Engineering and Engineering Thermophysics*, 4(2): 98-109. <https://doi.org/10.56578/peet040202>

APPENDIX

Appendix A: Computational/simulation result data

The following data are computational/simulation data constructed according to the system boundary adopted herein, and are used to verify the consistency of the evaluation metrics, calculation relationships, and argument chain. These data are not actual experimental measurements; conclusions involving experimental validation must be supported by actual experimental data or by experimentally validated model results.

Table A1. Performance comparison of different system configurations under baseline conditions

Metric	Case A: Direct Liquid-Cooling Heat Rejection	Case B: Vapor-Compression Refrigeration-Only	Case C: Direct Organic Rankine Cycle	Case D: Vapor-Compression Refrigeration + Organic Rankine Cycle
Q_{chip} (kW)	3.00	3.00	3.00	3.00
W_{comp} (kW)	—	0.80	—	0.80
W_{aux} (kW)	0.13	0.10	0.14	0.10
$P_{ORC,net}$ (kW)	—	—	0.04	0.21
$W_{net,cool}$ (kW)	0.13	0.90	0.10	0.69
Net cooling power consumption ratio	0.043*	0.300	0.033*	0.230
Cooling power compensation rate	0%	0%	28.6%*	23.3%
$T_{chip,max}$ (°C)	101.2	82.3	99.8	82.5
Feasibility under equivalent thermal safety boundary	No	Yes	No	Yes

A.2 Selective recovery and critical recovery temperature T_{cut}

As shown in Table A2, as T_{cut} is reduced from 85 °C to 68 °C, more rejected heat is incorporated into the organic Rankine cycle, $P_{ORC,net}$ is increased from 0.080 to 0.225 kW, and the net cooling power consumption ratio is simultaneously decreased. Below 68 °C, although Q_{rec} continues to increase,

A.0 Baseline computational conditions and nature of the data

All values in this appendix are computational/simulation data, and are used to examine the reproducibility of the system model and evaluation metrics. The baseline condition is specified as a chip thermal load of $Q_{chip} = 3.0$ kW, with a vapor-compression refrigeration evaporation temperature of 35 °C and a condensation temperature of 70 °C under active refrigeration. Unless a variable is specifically swept, the remaining boundary conditions are kept consistent, and the system auxiliary electrical power is accounted for according to the values specified in each table. The above data do not represent actual experimental measurements.

A.1 Architecture baseline comparison: Cases A–D

As shown in Table A1, although lower net power consumption is exhibited by Cases A and C, the condition $T_{chip,max} \leq 95$ °C cannot be satisfied under the specified high ambient temperature boundary. Therefore, their net cooling power consumption ratio/cooling power compensation rate values are not included in the direct ranking under equivalent cooling-service boundaries. Relative to Case B, net cooling power consumption is reduced by Case D from 0.90 to 0.69 kW, corresponding to a reduction of 23.3%, which demonstrates the compensation effect of the organic Rankine cycle on cooling power consumption under conditions where active refrigeration is necessary.

the organic Rankine cycle evaporation temperature is further decreased, the marginal power-generation benefit of incremental heat is rapidly attenuated and becomes negative, and UA is significantly increased. Therefore, an optimal T_{cut} of approximately 68 °C is identified, and the low-temperature recovery boundary is mainly jointly constrained by the net power benefit per unit recovered heat and the heat-exchanger scale.

Table A2. Thermodynamic performance of the selective waste-heat recovery system under different recovery termination temperatures

T_{cut} (°C)	Q_{rec} (kW)	$T_{eva,ORC}$ (°C)	$P_{ORC,net}$ (kW)	UA (W/K)	Net Cooling Power Consumption Ratio	$MB=\Delta P/\Delta Q$
85	0.55	70	0.080	42	0.273	—
80	0.90	66	0.140	55	0.253	0.171

T_{cut} (°C)	$\dot{Q}_{rec}$ (kW)	$T_{eva,ORC}$ (°C)	$P_{ORC,net}$ (kW)	UA (W/K)	Net Cooling Power Consumption Ratio	$MB=\Delta P/\Delta Q$
75	1.25	62	0.190	72	0.237	0.143
70	1.62	58	0.220	98	0.227	0.081
68	1.78	56	0.225	118	0.225	0.031
65	2.05	52	0.220	165	0.227	-0.019
62	2.28	48	0.205	240	0.232	-0.065

A.3 Exergy loss distribution and mechanism analysis

The set of computational data shown in Table A3 indicates that, after T_{cut} is further reduced, the irreversibility caused by the finite temperature difference in the coupling heat exchanger is slightly decreased, whereas the deterioration in system performance mainly arises from the decrease in the average organic Rankine cycle heat-absorption temperature, the reduction in cycle net power, and the rapid increase in UA . Therefore, the advantage of selective recovery is mainly jointly determined by the marginal net power benefit and the heat-exchanger scale, and exergy analysis is used to explain the irreversibility distribution among components.

A.4 Vapor-compression refrigeration condensing temperature: Temperature-lift cost and organic Rankine cycle power-generation benefit

From the condensing-temperature sweep results shown in Table A4, it can be seen that, when the condensing temperature is increased from 60 °C to 70 °C, the increment in organic Rankine cycle net electrical power exceeds the increment in compressor power consumption, and $\Delta P_{benefit}$ is

positive. After the temperature is increased from 70 °C to 75 °C, this incremental relationship becomes negative for the first time. Although the cooling power compensation rate may continue to increase at 75–80 °C, the net cooling power consumption ratio reaches its minimum at approximately 70 °C. Therefore, system optimization is oriented toward minimum net cooling power consumption rather than maximum organic Rankine cycle power generation.

A.5 Vapor-compression refrigeration evaporation temperature and chip thermal safety

As shown in Table A5, if the thermal safety constraint is temporarily disregarded, the lowest net cooling power consumption ratio is obtained at an evaporation temperature of 45 °C; however, the maximum chip temperature reaches 96.8 °C at this point, exceeding the 95 °C limit. After a hard thermal safety constraint is introduced, 40 °C becomes the minimum net cooling power consumption ratio point within the computationally feasible domain, indicating that the vapor-compression refrigeration evaporation temperature must be jointly determined with the high-temperature liquid-cooling boundary and the chip thermal safety constraint.

Table A3. Exergy loss distribution of system components under different recovery termination temperatures

Component	$T_{cut} = 68$ °C Exergy Loss (kW)	$T_{cut} = 62$ °C Exergy Loss (kW)	Explanation
Vapor-compression refrigeration compressor	0.260	0.260	Condensation/evaporation boundaries are identical; essentially unchanged.
Vapor-compression refrigeration evaporator	0.120	0.120	Heat-source-side boundary is identical.
Coupling heat exchanger	0.070	0.062	After the low-temperature section is included, the average temperature difference is reduced; heat-exchanger exergy loss is not “sharply increased.”
Organic Rankine cycle expander	0.050	0.061	The lower evaporation temperature changes the cycle state and expansion process.
Organic Rankine cycle condenser	0.080	0.088	Loss is slightly increased due to changes in cycle flow rate/state.
Pump and others	0.010	0.011	Small variation
Total	0.590	0.602	The main problem of low-temperature recovery is manifested as decreased net power benefit and increased UA , rather than a sharp rise in heat-exchanger exergy loss.

Table A4. Effect of vapor-compression refrigeration condensing temperature on temperature-lift cost and organic Rankine cycle power-generation benefit

T_{cond} (°C)	T_{dis} (°C)	$\dot{W}_{comp}$ (kW)	$P_{ORC,net}$ (kW)	$\dot{W}_{net,cool}$ (kW)	Net Cooling Power Consumption Ratio	Cooling Power Compensation Rate	$\Delta P_{benefit}$ (kW)
60	78	0.65	0.02	0.73	0.243	2.7%	—
65	84	0.72	0.10	0.72	0.240	12.2%	+0.01
70	91	0.80	0.21	0.69	0.230	23.3%	+0.03
75	99	0.90	0.30	0.70	0.233	30.0%	-0.01
80	108	1.04	0.36	0.78	0.260	31.6%	-0.08
85	118	1.22	0.39	0.93	0.310	29.5%	-0.15

Table A5. Effect of vapor-compression refrigeration evaporation temperature on net system energy efficiency and chip thermal safety

$T_{eva,VCR}$ (°C)	$T_{chip,max}$ (°C)	W_{comp} (kW)	$P_{ORC,net}$ (kW)	$W_{net,cool}$ (kW)	Net Cooling Power Consumption Ratio	Cooling Power Compensation Rate	Thermal Safety
25	70.8	0.98	0.24	0.84	0.280	22.2%	Satisfied
30	76.2	0.88	0.23	0.75	0.250	23.5%	Satisfied
35	82.3	0.80	0.21	0.69	0.230	23.3%	Satisfied
40	88.9	0.73	0.18	0.65	0.217	21.7%	Satisfied
45	96.8	0.67	0.15	0.62	0.207	19.5%	Not satisfied

A.6 Collaborative matching of vapor-compression refrigeration–organic rankine cycle working-fluid pairs

As shown in Table A6, among the compared working-fluid pairs, the lowest net cooling power consumption ratio of 0.222 is obtained with the R1234yf/R1233zd(E) combination. Lower compression power, a more suitable expansion ratio, higher expander efficiency, and lower UA are simultaneously exhibited by this working-fluid pair, and a system-level advantage is thereby formed. η_{exp} is determined by actual expander performance maps, experimental calibration, or a reliable equipment model.

A.7 Dynamic thermal load response

As shown in Table A7, as the thermal load is increased from 1 to 4 kW, the compensation rate of cooling power consumption by the organic Rankine cycle is raised from 13.9% to 25.6%, indicating that both the quantity and quality of recoverable rejected heat are more favorable under high load. However, the net cooling power consumption ratio under

the 4 kW condition is not further decreased to a pronounced extent, while the response time is extended to 7.6 minutes. Therefore, although the organic Rankine cycle compensation ratio can be increased by high load, a plateau region exists in the net system energy efficiency, and the system is simultaneously constrained by thermal inertia and thermal safety.

A.8 72-Hour operating sequence and engineering energy efficiency

As shown in Table A8, the 72-hour operating sequence indicates that the maximum chip temperature was always maintained below 95 °C, the mean net cooling power consumption ratio was 0.230, and the mean cooling power compensation rate was 23.3%. If $P_{IT} \approx Q_{chip} = 3.0$ kW, then cooling partial power usage effectiveness = 1.230. With the 0.90 kW net cooling power consumption of Case B taken as the baseline under the same service boundary, 0.21 kW is saved by Case D; at 8000 h/a, an annual electricity saving of approximately 1680 kWh is obtained.

Table A6. Collaborative thermodynamic performance comparison of different vapor-compression refrigeration–organic Rankine cycle working-fluid pairs

Vapor- Compression Refrigeration Working Fluid	Organic Rankine Cycle Working Fluid	W_{comp} (kW)	$P_{ORC,net}$ (kW)	UA (W/K)	Compression Ratio	Expansion Ratio	η_{exp}	Net Cooling Power Consumption Ratio	Cooling Power Compensation Rate
R134a	R245fa	0.80	0.190	122	2.75	3.10	0.68	0.237	21.1%
R134a	R1233zd(E)	0.80	0.210	112	2.75	3.35	0.72	0.230	23.3%
R134a	R1234ze(E)	0.80	0.165	131	2.75	3.70	0.64	0.245	18.3%
R1234yf	R245fa	0.78	0.185	126	2.88	3.08	0.68	0.232	21.0%
R1234yf	R1233zd(E)	0.78	0.215	115	2.88	3.31	0.73	0.222	24.4%
R1234yf	R1234ze(E)	0.78	0.160	136	2.88	3.66	0.63	0.240	18.2%

Table A7. Response performance of the vapor-compression refrigeration–organic Rankine cycle coupled system under dynamic chip thermal load

Q_{chip} (kW)	$T_{chip,max}$ (°C)	W_{comp} (kW)	$P_{ORC,net}$ (kW)	W_{aux} (kW)	$W_{net,cool}$ (kW)	Net Cooling Power Consumption Ratio	Cooling Power Compensation Rate	Steady-State Response Time (min)
1.0	68.5	0.30	0.05	0.06	0.31	0.310	13.9%	3.8
2.0	74.6	0.52	0.12	0.08	0.48	0.240	20.0%	5.1
3.0	82.3	0.80	0.21	0.10	0.69	0.230	23.3%	6.4
4.0	89.7	1.12	0.32	0.13	0.93	0.233	25.6%	7.6

Table A8. Statistics of system performance and engineering energy efficiency under 72-hour continuous operation

Metric	Mean	Standard Deviation	Minimum	Maximum
Q_{chip} (kW)	3.00	0.04	2.92	3.08
$T_{chip,max}$ (°C)	82.6	1.2	79.8	85.4
W_{comp} (kW)	0.800	0.030	0.740	0.870
$P_{ORC,net}$ (kW)	0.210	0.012	0.186	0.234

Metric	Mean	Standard Deviation	Minimum	Maximum
W_{aux} (kW)	0.100	0.008	0.084	0.118
$W_{net,cool}$ (kW)	0.690	0.035	0.623	0.763
Net cooling power consumption ratio	0.230	0.012	0.208	0.254
Cooling power compensation rate	23.3%	1.4%	20.2%	26.5%
coolingp PUE	1.230	0.012	1.208	1.254

A.9 Model validation and energy closure results

As shown in Table A9, under independent validation conditions, mean absolute percentage error values for key outputs are all below 5%, and average energy closure errors for the two key heat exchangers do not exceed 4.0%. The calibrated model is used to expand the parameter space of T_{cut} , $T_{cond,VCR}$, $T_{eva,VCR}$, and working-fluid combinations. In the model validation record, the number of validation points, the calibration-set/validation-set partitioning method, and the uncertainty propagation process are documented.

A.10 Summary of the result chain

Under the specified high ambient temperature condition, the net cooling power consumption of the vapor-compression refrigeration-only scheme was 0.90 kW, and it was reduced to 0.69 kW after selective organic Rankine cycle recovery was

introduced, with the net cooling power consumption ratio decreased from 0.300 to 0.230. As the recovery termination temperature was continuously varied, the organic Rankine cycle net electrical power was increased from 0.080 to 0.225 kW when T_{cut} was reduced from 85 °C to 68 °C. When T_{cut} was further reduced, although the recoverable heat was increased, the marginal net power benefit became negative and UA increased rapidly; consequently, a critical recovery temperature of approximately 68 °C was obtained. From further sweeping of the condensing temperature, an optimal balance between compressor temperature-lift cost and organic Rankine cycle power-generation benefit was identified at approximately 70 °C, whereas the evaporation temperature was constrained below approximately 40 °C by the maximum chip temperature limit. Taken together, these results form a unified optimization logic of heat quality–marginal net power–equipment scale–thermal safety.

Table A9. Prediction accuracy of the thermodynamic model and energy closure results for key heat exchangers

Validation Quantity	Mean Absolute Percentage Error	Root Mean Square Error	Assessment
W_{comp}	2.8%	0.024kW	Pass
$P_{ORC,net}$	4.2%	0.010kW	Pass
$T_{chip,max}$	1.1%	0.92 °C	Pass
Net cooling power consumption ratio	3.4%	0.009	Pass
Vapor-compression refrigeration evaporator energy closure error	—	3.1% (average)	Pass
Organic Rankine cycle coupling heat exchanger energy closure error	—	4.0% (average)	Pass

Appendix B: Definition boundaries of the model and metrics

Table B1. Definition boundaries and treatment approaches for the thermodynamic model and system performance metrics

Model or Metric Item	Treatment Adopted in This Study
Low-temperature condensation heat recovery	A small pinch temperature difference is not adopted as a direct criterion for high exergy loss; the recovery value of low-temperature heat is evaluated jointly by the organic Rankine cycle evaporation temperature, marginal net power benefit, and UA .
System exergy evaluation	Component exergy loss, recovery-subsystem exergy efficiency, and net cooling power consumption ratio are adopted as metrics so that cooling service and heat-source exergy are not double-counted within the system boundary.
Expander efficiency	Determined jointly by pressure ratio, volumetric flow rate, built-in volume ratio matching, rotational speed, leakage, and working-fluid thermophysical properties.
Energy coupling mode	Electrical coupling among the expander, generator, and power-electronics interface is adopted, and power and efficiency at each stage are measured separately.
Data center energy efficiency metrics	Net cooling power consumption ratio, cooling power compensation rate, and cooling partial power usage effectiveness are adopted at the laboratory boundary; full power usage effectiveness is calculated only when the facility-level boundary is complete.
Long-term stability	The 72-hour stability conclusion is supported only by actual continuous operation records; computational/simulation data are used only for model and metric closure verification.