PERFORMANCE ANALYSIS OF THE HIGH-TEMPERATURE HEAT PUMP SYSTEM WITH THE REVERSED BRAYTON CYCLE

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ABSTRACT

This study explores a high-temperature heat pump (HTHP) system based on the reversed Brayton cycle with air as the working fluid. The purpose of this system is to address the constraints encountered in traditional HTHP systems, which typically yield thermal energy below 250℃, and to fulfill the requirements of the industrial sectors that mostly demand the thermal energy over 300℃. This study investigates two different systems: the basic type reversed Brayton cycle (BRBC) and the reheat type reversed Brayton cycle (RRBC). The RRBC system is designed to augment the system efficiency by maximizing the power recovery from the expander. This can be achieved through iterative procedures involving isobaric heating via the heat exchanger and isentropic expansion through the expander. The default objective system aims to deliver the thermal energy of 300°C with a capacity of 300 kW. A well-organized simulation model has been developed to facilitate the comprehensive comparison of two proposed systems. The model is utilized to investigate the performance of the RRBC system which varies with the stage numbers accordingly. It is shown that the RRBC with 5 stages can improve system efficiency by about 9.6% compared to the BRBC system. Additionally, the system performance is evaluated under various operating conditions and component efficiencies. The COP maps with the variations of the component efficiency factors are suggested as a guideline to the system design.

1 INTRODUCTION

The increasing global consensus on the imperative of achieving carbon neutrality has underscored the need for concerted efforts towards reducing thermal energy consumption in industrial sectors, which currently accounts for approximately 20% of overall energy usage (Adamson *et al*., 2022). One promising approach to address this challenge is the utilization of industrial heat pump systems, which efficiently harness waste heat to provide thermal energy, thus playing a pivotal role in realizing carbon neutrality within industrial contexts. However, while the majority of heat demand in industrial processes is above 300℃ (Rehfeldt *et al*., 2018), the maximum temperature of commercially available heat pumps currently is less than 250℃ (Jiang *et al*., 2022). This limitation stems from the inherent challenges in developing HTHP systems beyond 300℃ in the conventional vapor compression system (Zühlsdorf *et al*., 2019), attributed to constraints posed by conventional synthetic refrigerants' maximum temperature tolerances and thermal management issues under harsh operational conditions characterized by high temperatures and pressures (Arpagaus *et al*., 2018).
Meanwhile, some researchers suggested that the HTHP system based on the reversed Brayton cycle (RBC) with natural refrigerants can make a breakthrough in this limitation (Shuailing *et al.*, 2023). In the RBC systems, the working fluid operates solely in the gas phase and remains chemically stable under high temperature conditions, thereby imposing fewer constraints on operational temperature ranges. Zhang *et al.* (2017) presented an air-cycle heat pump with a turbocharger driven by a blower. Bi *et al.* (2012) suggested a guideline for optimal design of the RBC systems in terms of the heating load density, COP, and system footprint. Yang *et al.* (2024) conducted the optimization of the RBC systems with intercooling and regeneration. Walen *et al*. (2023) delved into the nonlinear operational optimization of the industrial RBC HTHP systems.

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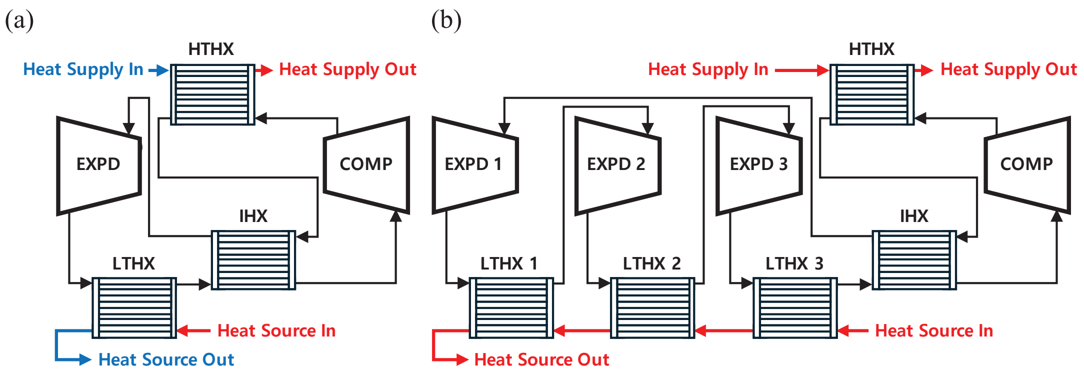

Figure 1: Schematics of the system configurations: (a) BRBC and (b) RRBC

However, there are few papers that presents the design guidelines that practically suggest the performance of the HTHP systems based on the RBC under various operational conditions and system specifications. Therefore, this research endeavors to develop a simulation model capable of predicting system efficiency across diverse design parameters, encompassing variations in heat source and sink temperatures, as well as component efficiencies such as isentropic efficiency of fluid machines and the effectiveness of heat exchangers. This modeling effort extends not only to the basic type reversed Brayton cycle (BRBC) but also to an advanced variant named as reheat type reversed Brayton cycle (RRBC). Leveraging this model, the study extrapolates expected efficiencies of the system under varying design parameters. Ultimately, the objective is to establish criteria and rational benchmarks for the development and evaluation of the RBC HTHP systems, specifically tailored for real-world industrial applications.

2 SYSTEM MODELING

2.1 System Configurations and Theoretical Cycle Analysis

Figure 1(a) depicts the schematic diagrams of the BRBC configuration. The BRBC system consists of 4 essential components: compressor, expander, high-temperature heat exchanger (HTHX), and low-temperature heat exchanger (LTHX). The internal heat exchanger (IHX) is also the optional components of the system (Kabat *et al*., 2022). While the HTHX and LTHX play roles corresponding to the condenser and evaporator at the conventional vapor compression cycle (VCS) respectively, it is better to be referred to as those names because there is no phase change involved. Air was chosen as the working fluid due to its cost-effectiveness, high stability, eco-friendliness, and the appropriate compression ratio (Shuailing *et al*., 2023). Air was also employed as the secondary fluid at both heat source and heat sink. The heat source and sink airs are designated as source air and supply air respectively in this study. The inlet temperature of the source air, referred to as the source temperature, is set as 100°C. Also, the inlet and outlet temperatures of the supply airs are denoted as supply temperature and target temperature, respectively, with values set at 100°C and 300°C.

The working fluid undergoes the following thermodynamic processes within the system cycle. Initially, through the isentropic compression process, pressure and inherently temperature of the working fluid rise through the compressor. Then, the air with high temperature and pressure transfers thermal energy to the secondary fluid (supply air) in the HTHX to generate the usable thermal energy. Afterwards, through the isentropic expansion process, the pressure and temperature of the working fluid decreases through the expander, simultaneously generating power that can compensate the power consumption at the compressor. Lastly, the air absorbs the thermal energy from the heat source at the LTHX and is then supplied to the compressor inlet again, completing the cycle. If the inlet temperature of the supply air is larger than that of the source heat, the internal heat exchanger can be employed to reduce the compression ratio and net power consumption of the system. If so, the working fluid at the HTHX outlet transfers the thermal energy to the working fluid at the LTHX outlet so that the temperature of the compressor inlet rises near to the supply air inlet temperature.

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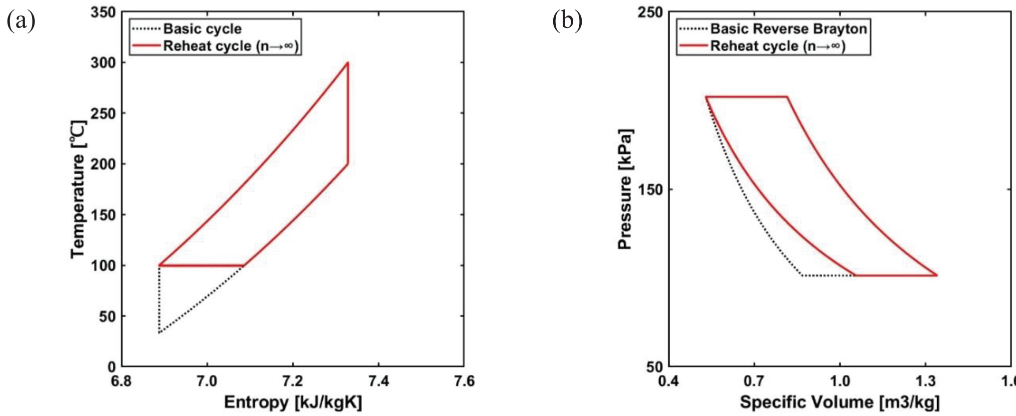

Figure 2: Comparison of the theoretical BRBC and RRBC cycles: (a) T-s and (b) P-v diagram

Despite the theoretical limitations on the BRBC efficiency, it is possible to achieve a modest improvement in efficiency through the integration with the concept of the Ericsson cycle. The reversed Ericsson cycle follows an isothermal process instead of an isentropic process in the fluid machines. Given that our system does not utilize the cold energy with low temperature from the expander outlet, it is more beneficial for the working fluid to follow the isothermal line than the isentropic line. However, there is no practical fluid machine that can realize the isothermal expansion. In this case, the RRBC can be a good practical alternative that can closely follow the isothermal process of the reversed Ericsson cycle. Figure 1(b) shows the schematic diagram of the reheat type reversed Brayton cycle, especially three steps expansion in the figure. Instead of the one expansion and heat recovery procedure, the procedure is repeated as the number of the expander and LTHX pairs (3 stages in the figure). The alternating repetition of the isentropic expansion and isobaric expansion enables the working fluid to closely approximate the isothermal process.

The theoretical efficiency of the RBRC has been derived in this study. Firstly, the theoretical efficiency of the BRBC system can be determined only with the temperature parameters (Borgnakke and Sonntag, 2020): heat source temperature (T_L), supply temperature (T_H), target temperature (T_P). The equation is suggested as follows.

$$\mathrm{COP}_{BRBC} = \frac{1}{1-\frac{T_L}{T_P}} = \frac{1}{1-r_p^{-(k-1)/k}} < \mathrm{COP}_{Carnot} = \frac{1}{1-\frac{T_L}{T_H}} \tag{1}$$

Since T_P is larger than T_H, the COP (Coefficient of Performance) of the BRBC is always lower than that of the Carnot cycle as presented in the Equation (1). Meanwhile, the theoretical efficiency of the RRBC can be presented based on the derivation conducted this study as follows.

$$\mathrm{COP}_{RRBC} = \frac{1}{1-\frac{T_L}{T_P}\frac{n(1-(T_H/T_P)^{1/n})}{1-T_H/T_P}} \;\rightarrow\; \frac{1}{1-\frac{T_L}{T_P}\frac{-log\,(T_H/T_P)}{(1-T_H/T_P)}}\;(n \rightarrow \infty) \tag{2}$$

where n denotes the number of the reheat repetition (referred stage numbers). When the number of the expander and LTHX pairs become larger, the expansion process approaches to the isothermal expansion. Figure 2 compares the T-s and P-v diagrams of the theoretical BRBC and RRBC (when $n \rightarrow \infty$). The dotted black and red solid line denote the BRBC and RRBC cycle respectively. The difference between the isentropic and isobaric expansion processes is shown in T-s diagram. The integrated area beneath the expansion process line in the P-v diagram denotes the amount of the power recovery from the expander. Thus, the benefit of the RRBC is depicted clearly in the P-v diagram. Due to the larger area beneath the expansion process line, the RRBC can reduce power consumption for the same amount of the heating capacity compared to the BRBC. For the RRBC system with a finite number of the expansion stage, the system efficiency is defined between the BRBC and RRBC with infinite stages.

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2.2 System Operating Conditions

The primary objective of this study is to provide guidelines for system design and efficiency prediction when introducing heat pumps into industrial sites. With this context in mind, target efficiency for each component was set at reasonable and practical levels. The default operating conditions are summarized in Table 1. Firstly, regarding the operating temperature conditions, the scenario assumes that the thermal energy from the heat source at 100°C is employed to elevate the supply air temperature from 100°C to 300°C. In the reversed Brayton cycle with air working fluid, the operating temperature and pressure are decoupled, so thus it is necessary to designate the one pressure condition as well. In this study, the compressor outlet pressure was set at 2000 kPa. The component efficiency factors considered in this study include the isentropic efficiency of the fluid machines, effectiveness of the heat exchangers, and the pressure loss of the working fluid in each heat exchanger and its adjacent pipelines. those were determined at the realistic and achievable levels based on the practical specifications of the product. The efficiency of the compressor and expander are 85% and 80% respectively (White, 2016). In addition, the effectiveness values are set as 93% for all the heat exchangers. Regarding pressure loss, the target value is defined as a relative ratio to the outlet pressure of the compressor, rather than being specified with a certain differential pressure value. This is because the performance drop due to the pressure loss can be underestimated for the systems with lower overall operational pressure.

Table 1: Default system operating conditions and component efficiencies

Classification	Parameter [Unit]	Values
System operating conditions	Heat source temperature [°C]	100
	Supply temperature [°C]	100
	Target temperature [°C]	300
	System capacity [kW]	300
	System maximum pressure [kPa]	2000
Component design efficiency	Compressor efficiency	85%
	Expander efficiency	80%
	Heat exchanger effectiveness	93%
	Heat exchanger pressure drop (HTHX)	5% of the compressor outlet pressure
	Heat exchanger pressure drop (LTHX)	7% of the compressor outlet pressure

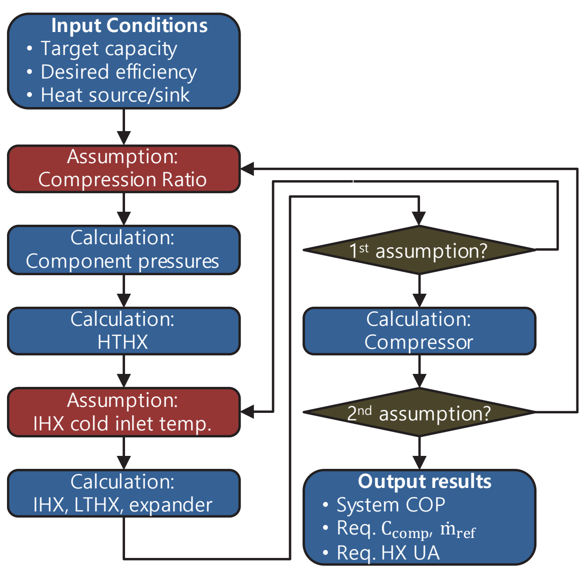

Figure 3: Flowchart of the simulation model

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Table 2: Simulations results with different stage numbers

Output parameters	1-stage (BRBC)	3-stage (RRBC)	5-stage (RRBC)
System efficiency (COP)	**1.58**	**1.70 (+7.6%)**	**1.73 (+9.6%)**
Compression ratio of the compressor [-]	3.32	3.33	3.33
Mass flow rate of the working fluid [kg/s]	1.55	1.55	1.55
Compressor power input [kW]	301	301	301
Expander power recovery [kW]	111	125	128
HTHX heat transfer rate [kW]	300	300	300
LTHX heat transfer rate [kW]	110	124	127
Required UA of HTHX [kW/K]	21.2	21.2	21.2
Required UA of LTHX [kW/K]	4.5	13.6	22.7
Required UA of IHX [kW/K]	21.2	21.2	21.2

2.3 Simulation Model

The computational procedure of the model proceeds as follows. Initially, the compression ratio of the compressor is assumed, which is the first assumption of the model. Subsequently, the inlet and outlet pressures of all the components are computed based on specified target pressure drop conditions, Then, the model starts computations to obtain the inlet and outlet temperature of the HTHX. As a next step, the inlet conditions of the IHX cold side are assumed to obtain the IHX heat transfer rate, which is the second assumption of the model. After that, subsequent calculations encompass the IHX hot side, expanders, and LTHXs. The model subsequently checks whether the IHX cold side inlet temperature aligns with the initial assumption. Then, the model adjusts the second assumption and conducts iterative computations. Subsequently, the model determines the compressor outlet conditions and verifies their congruence with the HTHX inlet conditions. If inconsistencies arise, the initial assumption is modified, prompting iterative recalculations. The described process is summarized as a flowchart in Figure 3.

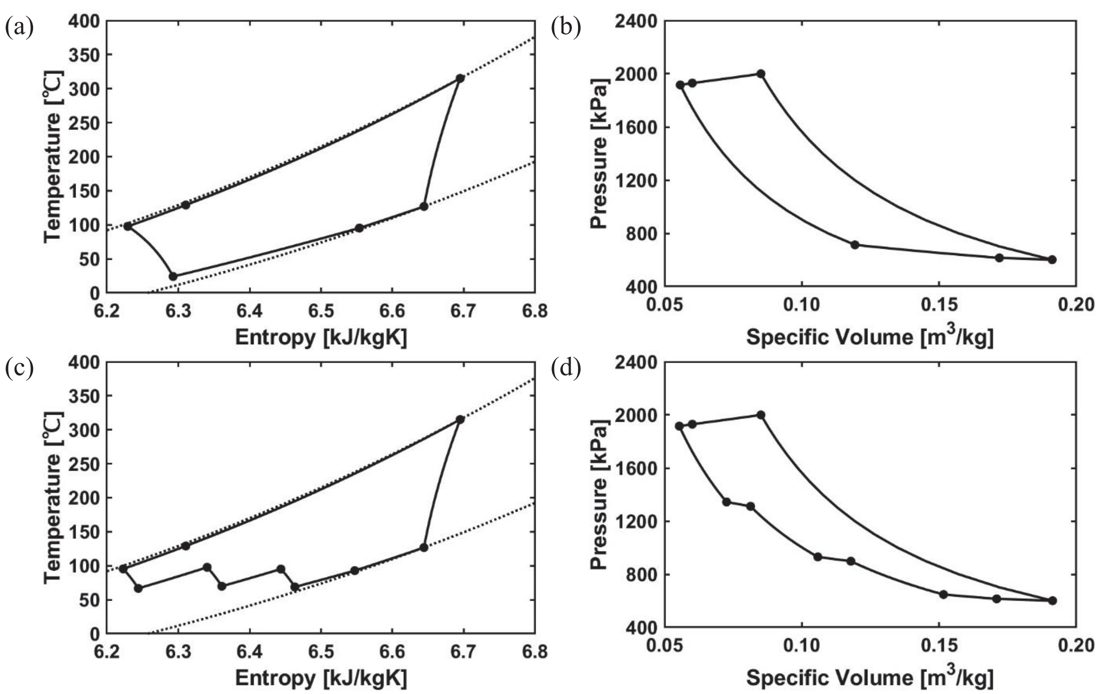

Figure 4: Comparison of the 1-stage (=BRBC) and 3-stage (=RRBC) cycle simulation results: (a) 1-stage T-s, (b) 1-stage P-v, (c) 3-stage T-s, and (d) 3-stage P-v diagram

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3 SIMULATION RESULTS

3.1 Simulation Results for the Default Conditions

Figure 4 presents the T-s and P-v diagrams of the reversed Brayton cycle with basic and reheat types considering the component efficiency. The dotted line in the T-s diagram denotes the isobaric line for the pressures at the compressor inlet and outlet. As seen in the figure, the polytropic expansion of the working fluid occurs at higher temperature and pressure conditions, which are favorable for the power recovery from the expander. This efficiency improvement varies according to the stage number of the RRBC. Therefore, Table 2 presents simulation results for different stage numbers: 1-stage (=BRBC), 3 stages, and 5 stages. The expected COP of the basic system can achieve 1.58. The COP is enhanced for the reheating system about 7.6% with 3 stages and 9.6% with 5 stages respectively. Given that the demanding heating capacity and its relevant parameters of the secondary fluid (supply air) are constantly defined as the input conditions, the mass flow rate and the compression ratio of the working fluid remain nearly identical regardless of the stage number. However, it can be also seen that the number of the expander and the total required LTHX UA increase with the larger stage number. Therefore, it is important to note that the RRBC also has the disadvantage, particularly concerning the initial equipment cost. Furthermore, increasing the stages cannot improve system efficiency infinitely, as there is a theoretical limitation as seen in the Equation (2). Figure 5 shows the relationship between the stage number (<11) and system COP.

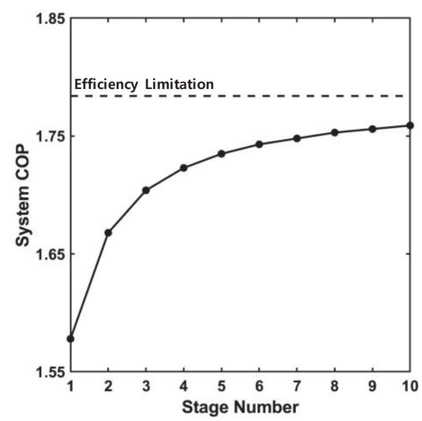

Figure 5: The relationship between the stage number and COP increase

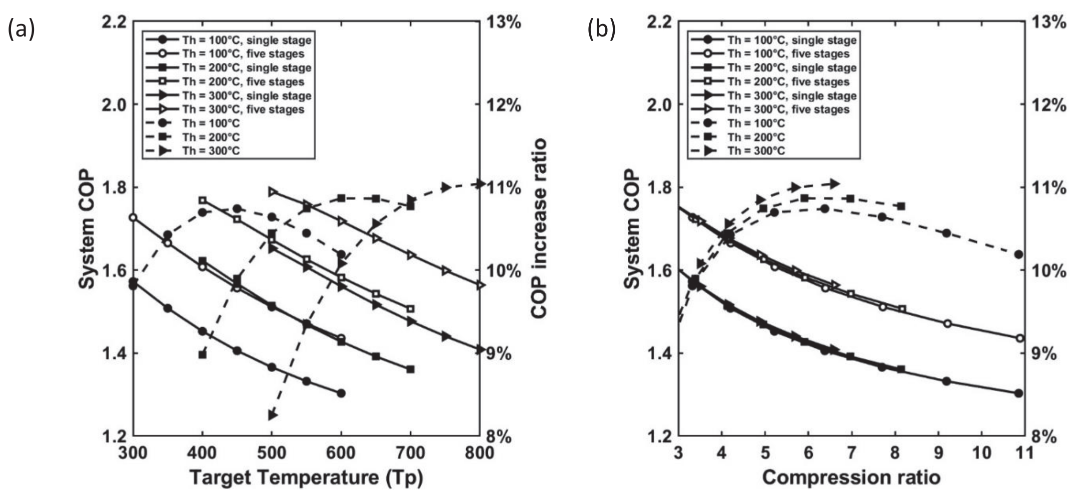

Figure 6: The relationship between the stage number and COP increase

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3.2 Simulation Results for the Variant Conditions

The operating parameters and component specifications vary depending on the characteristics of the industrial application sites. Consequently, it is imperative to assess the system's suitability under diverse operating conditions and component specifications as the same manner in the default conditions. Firstly, the cycle simulation presented in the preceding section was performed under varying heat sink inlet and outlet temperature conditions (i.e. supply temperature (T_H), and target temperature (T_P)) as seen in the Figure 6(a). While the heat source temperature (T_L) is also the crucial parameter, it has been concluded that what influences the system performance trend is the relative difference between those three temperature parameters, and it has been observed that if this temperature difference is the same, the absolute temperature level has relatively little effect to the trends. Accordingly, while keeping the heat source temperature constant, only the other two parameters were varied in this study. The target temperature varies from 300 to 800°C and also the supply temperature differs with 3 cases: 100, 200, and 300°C. The results are for the BRBC and RRBC with 5 stages systems. It is obvious that the system COP drops according to the increase of the target temperature because the pressure lift through the compressor becomes larger. While the system COP becomes larger for the RRBC with 5 stages, the aforementioned trend is the same. In additions, it can be seen that the COP increase ratio of the RRBC varies with the target temperature and has the optimum point which maximizes the merit of the RRBC system. The same result was re-demonstrated by substituting the x-axis with compression ratio in Figure 6(b). The results became much clear, and this implies that the system COP is mainly related to the compression ratio rather than the single temperature parameters.

(b)

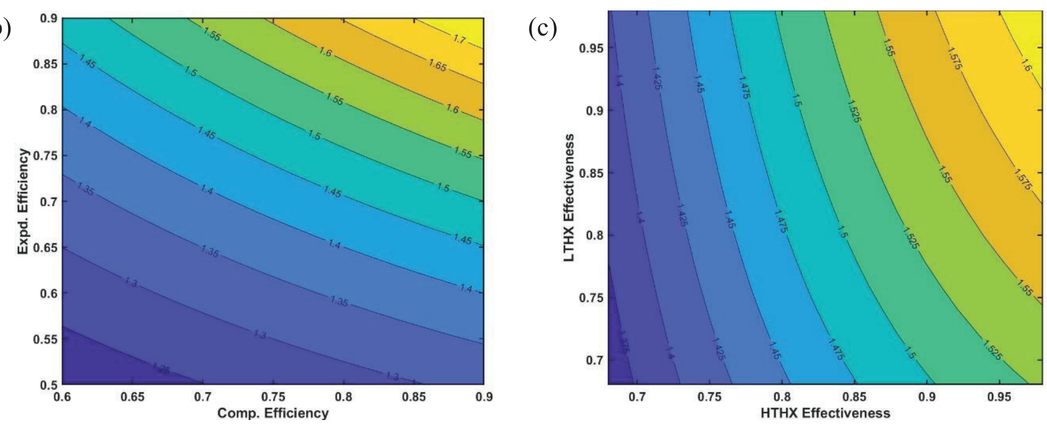

Figure 7: COP maps of the BRBC system with the variations of the (a) efficiency of the fluid machines, and (b) effectiveness of the heat exchangers

(a)

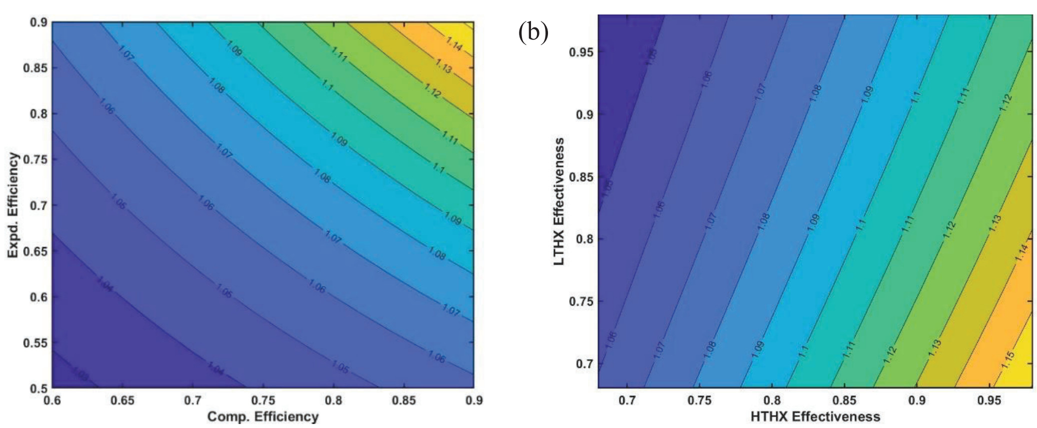

Figure 8: COP increase ratio maps with the variations of the (a) efficiency of the fluid machines and (b) effectiveness of the heat exchangers

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Despite the appropriate assumption for the component efficiency, the efficiency may not be achieved in the practical applications. Thus, there also exists a need to examine the impact of the variant system efficiency to the system COP to establish the design margin. Figure 7(a) and (b) respectively present the COP maps according to variations of the fluid machine isentropic efficiency and heat exchanger effectiveness for the BRBC system. While the COP is highly relevant to both expander and compressor efficiencies, the effectiveness of the HTHX has the greater impact on the system COP compared to that of the LTHX. This can be attributed to the following rationale: A decease in LTHX effectiveness primarily affect the expander power recovery and the IHX can compensates the low temperature of the working fluid toward the compressor inlet. Conversely, a decrease in HTHX effectiveness directly makes the outlet temperature of the compressor larger, consequently influencing the compressor work as well.

The same work was conducted for the RRBC systems with 5 stages. The results were depicted in Figure 8 as the COP increase ratio relative to the BRBC system rather than the COP value itself. Instead of the COP result. The results show that the reheat system is more sensitive to the component efficiency. Thus, it is possible to expect the better performance improvement if we utilize the better components. Especially, the component efficiency dependency is more severe for the fluid machines. Nevertheless, it appears that there is a somewhat reversed trend observed for the LTHX effectiveness parameters. This arises due to the continuous reheating of the working fluid in the stepwise expansion process, resulting in the expander operating under better conditions in spite of the poor LTHX effectiveness value. Thus, it can be asserted that the reheat system demonstrates a greater resilience in comparison to the fundamental system.

4 CONCLUSIONS

This research evaluated the operational efficiency of a reversed Brayton cycle heat pump system utilizing air as the working fluid. This study also investigated the impact of implementing multiple pairs of expanders and LTHX, referred to as the reheat cycle. The results suggested that under the default operating conditions, the basic system (BRBC system) can attain a Coefficient of Performance (COP) of 1.6, whereas employing the reheat system (RRBC system) with 5 stages can enhance the COP by 11.3%. As follow-up research, there arises a necessity for a study to evaluate the economic feasibility of the reheat system. Due to the trade-off relationship between the efficiency improvement and the initial cost increase in the reheat system, it is expected that there exists an optimal stage number that can minimize the lifetime cost (LC). It is possible to find the optimal stage numbers through quantitative analysis of capital expenditure (CAPEX) and operational expenditure (OPEX). The CAPEX varies with the footprint of the system and its industrial application and OPEX will depend on the electricity tariff policy of each country. Therefore, in-depth research reflecting the regional characteristics of each country and the industrial characteristics of the applications will be meaningful. This pursuit promises to offer a deeply engaging subject of inquiry within academic discourse.

NOMENCLATURE

COP	Coefficient of Performance	(–)
k	heat capacity ratio	(–)
$\dot{m}$	mass flow rate	(kg/s)
n	stage number	(–)
T	temperature	(K)
UA	heat conductance	(W/K)

Subscript

H	heat sink inlet
P	heat sink outlet
L	heat source inlet

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