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Experimental Investigation of CO_2/R32 Mixture Combined Cooling and Power Cycle under Variable Heat Source Conditions

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ABSTRACT

CO_2 owing to its excellent heat transfer properties and environmental sustainability, has been extensively explored for integration into future energy systems. CO_2-based combined cooling and power cycles exhibit great potential for diverse multi-energy complementarity scenarios, offering high source-sink matching and adjustability. However, they face challenges related to stringent condensation requirements and suboptimal energy conversion. In response, CO_2/R32 mixture systems have emerged as a promising solution to enhance efficiency and condensation performance. This paper presents preliminary experimental research results on the CO_2/R32 mixture combined cooling and power cycle. It showcases the performance of CO_2/R32 system under varying heat source conditions and its impact on power and cooling subcycles. CO_2/R32 demonstrates excellent thermal matching performance and outstanding potential for utilizing medium and low-temperature thermal energy. The CCPC system can simultaneously produce 536W of predicted net power and 1550W of cooling capacity with an energy efficiency of 13.91% and a COP of 4.71. Additionally, the reduction in critical pressure and the increase in critical temperature have effectively improved the operating pressure and condensing conditions of the system. Under the same condensing conditions, the condensing pressure has decreased from 6.7MPa to 3.0MPa, and the operable condensing temperature range has significantly expanded.

Keywords: clean energy conversion technology, CO_2-based mixture, cogeneration and polygeneration, combined cooling and power cycle, experimental investigation

NOMENCLATURE

Abbreviations	
CCP	combined cooling and power cycle
COP	coefficient of performance
CSC	cooling sub-cycle
ODP	ozone depletion potential
PSC	power sub-cycle
Symbols	
h	specific enthalpy (kJ/kg)
m	mass flow rate (kg/s)
p	pressure (MPa)
P	power (kW)
Q	heat flow rate (kW)
T	temperature (°C)
η	efficiency (%)

1. INTRODUCTION

The CO_2 direct utilization technology is becoming a new trend in future energy systems [1]. For CO_2, its environmental friendliness (ODP=0, GWP=1) ensures the potential for emission reduction [2], and its thermal stability opens up prospects for high-parameter utilization [3]. Supercritical CO_2 exhibits excellent heat and mass transfer performance, achieving good thermal source matching while significantly reducing the size of components [4].

Currently, CO_2 is widely studied as a working fluid in fields such as power generation, refrigeration, heat pumps, and energy storage [5, 6]. However, the promotion of CO_2 is constrained by insufficient capability of power generation, high operating pressures, and stringent condensation conditions [7]. The blend of refrigerants into CO_2 to form mixtures has become an effective solution to address the aforementioned challenges [8], which can increase the critical temperature, reduce the critical pressure, and improve compressibility.

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Currently, CO_2 mixtures have been widely studied in theoretical analyses, such as additive selection, parameter optimization, configuration matching, and component regulation [9]. CO_2/R32 has emerged as a recommended choice due to its excellent thermodynamic performance and system efficiency [10].

However, there is currently a scarcity of experimental research on CO_2-based mixtures, mainly due to the specialized requirements of experimental setups for mixtures [11]. For combined cooling and power cycle (CCPC), whose sub-cycles are coupled through a shared condenser, the multi-energy outputs and the shared condensation will further challenge the capability improvement of mixtures. In this paper, the CCPC experimental platform using a CO_2-based mixture was constructed. Based on this experimental platform, we have preliminarily investigated the impact of heat sources on the CO_2/R32 system, explored the coupling relationship between sub-cycles, and paid attention to the unique properties of CO_2/R32. Recent work will be dedicated to exploring variations in condensation conditions, mixture compositions, dynamic component adjustments, and other aspects. This represents a completely new exploration of CO_2-based mixture systems.

2. SYSTEM INTRODUCTION

2.1 Working fluid selection

CO_2 and R32 are both non-toxic and safe common working fluids used in the fields of industry and transportation. Previous research has demonstrated that CO_2/R32 mixtures exhibit good performance in combined cooling and power generation, along with moderate temperature glide. Simultaneously, as the composition of the refrigerant increases, there is an enhancement in the thermodynamic performance of the mixture. Considering both thermodynamic performance and safety requirements, a mass fraction of 0.25/0.75 for CO2/R32 is chosen.

At this composition, R32 effectively increases the critical temperature of the mixture and reduces the condensation pressure, while CO2 lowers the GWP of R32 and to some extent, inhibits the thermal decomposition of R32 at high temperatures.

The main physical parameters of pure CO_2, R32, and CO_2/R32 mixtures are listed in *Table 1*. The thermodynamic properties of the working fluids were obtained using REFPROP 10.0. In actual cylinder charging processes, achieving precise filling is challenging due to the uneven pressure inside the steel cylinder. Through calibration using a gas chromatograph, the actual proportion of the CO_2/R32 mixture is determined to be 0.247:0.753.

Table 1 Physical properties of working fluids

Terms	CO_2	R32	CO_2/R32 (0.25/0.75)	CO_2/R32 (0.247/0.753)
Molecular mass (g/mol)	44.01	52.02	50.02	50.04
Critical temperature (°C)	31.1	78.1	68.8	69.0
Critical pressure (MPa)	7.38	5.78	6.78	6.77
GWP	1	675	506.5	508.8
ODP	0	0	0	0
ASHRAE 34 safety group	A1	A2L	-	-

The improvement in physical properties brought about by the mixed working fluid to some extent helps address the condensation conditions and operating pressure issues in the system.

2.2 Experimental facility

A small-scale combined cooling and power cycle (CCPC) test bench has been built, which can be used to perform CO_2 mixture tests under specific conditions. The test bench is equipped with measuring instruments such as pressure transmitters, flowmeters, and thermocouple temperature sensors with their main parameters listed in *Table 2*.

Table 2 Sensor parameter of CCPC test bench

Terms	Sensor types	Measuring range (Accuracy)
Air temperature	K-Thermocouple	0~500 °C (±0.5°C)
Other temperature	PT100	−50~150 °C (±0.15°C)
Air pressure difference	Differential pressure transmitter	0~0.5 MPa (±0.1%)
PSC pressure	Pressure transmitter	0~16 MPa (±0.25%)
Other pressure	Pressure transmitter	0~8 MPa (±0.25%)
Cooling water flow	Turbine flowmeter	0~1000 L/h (±1.0%)

Chilled water flow	Turbine flowmeter	0~270 L/h (±1.0%)
Air flow	Venturi flowmeter	0~200 kg/h (±1.0%)
Power subcycle flow	Coriolis flowmeters	0~200 kg/h (±0.15%)
Cooling subcycle flow	Coriolis flowmeters	0~50 kg/h (±0.15%)

The overall experimental platform includes an air compressor, an air heater, a chiller unit, and a water tank. In the experiment, heated compressed air serves as a simulated heat source, allowing for precise control of heat source flow and temperature. A water tank and a water chiller unit are used as simulated cooling sources. Since the water chiller unit employs a thermostat control logic, the outlet water temperature fluctuates within the set threshold. An electric heater is installed in the water tank. At this point, cold water from the water chiller unit is blended with hot water from the water tank based on flow allocation, maintaining the cooling water's temperature and flow rate at the set values in the condenser.

The system structure and the T-S diagram of CCPC test platform are shown in *Fig. 1*. CCPC consists of a transcritical power sub-cycle (PSC) and a subcritical compression cooling sub-cycle (CSC), coupled by sharing a condenser and liquid reservoir. It includes multiple heat exchangers, a diaphragm pump, an expansion valve, a throttle valve, and a compressor, with detailed parameters displayed in *Table 3*.

The gas heater is a shell-and-tube heat exchanger, while the remaining heat exchangers are plate heat exchangers. All heat exchangers and pipelines are insulated with thermal insulation to minimize heat loss to the environment. The flow of the working fluid in the PSC is controlled by a diaphragm pump. As the expansion valve for the CO_2 mixture is still in the development stage and considering the potential damage due to the instability of the mixture, a custom-made expansion valve is used to regulate system pressure. The throttle valve and compressor jointly control the flow and pressure of CSC. The isentropic efficiency of the simulated expander is assumed to be 0.6. For safety reasons, the maximum system pressure cannot exceed 13 MPa, and the maximum temperature cannot exceed 230°C.

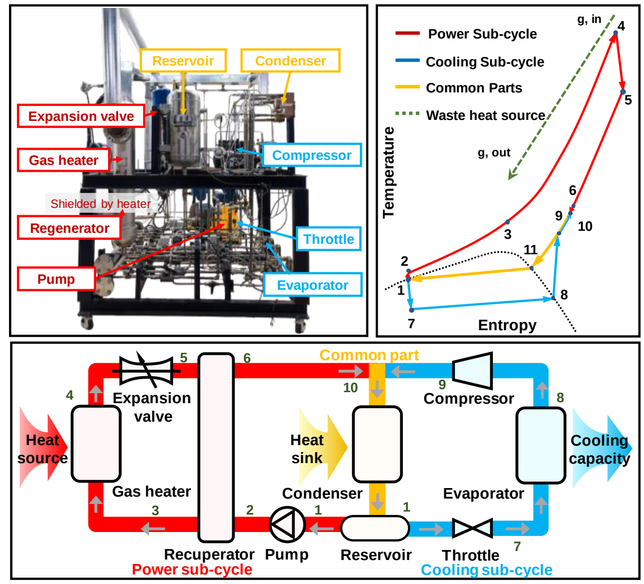

Fig. 1 Experimental facility of CO_2/R32 CCPC

Table 3 Component parameters of CCPC bench

Equipment	Main parameters	Model
Pump	4L/min, 15MPa	Diaphragm pump
Expansion valve	Normal flow 130~200kg/h, Design high pressure 12MPa, low pressure 5.73 MPa	V8030HPS/006
Compressor	50 Hz, 1450 rad/min, Discharge volume 1.12m^3/h	CD180H
Throttle	Normal flow 1.1±0.7m^3/h, Maximum pressure difference 10.0 MPa	UKV-J08D41
Heater	Heat transfer area 1.6 m^2, Design pressure 15 MPa	shell and tube
Recuperator	Heat transfer area 0.328 m^2, Design pressure 12 MPa	Plate
Condenser	Heat transfer area 0.696 m^2, Design pressure 5.4 MPa	Plate
Evaporator /Subcooler	Heat transfer area 0.216 m^2, Design pressure 5.4 MPa	Plate

The system's operating process is as follows: During combined cooling and power generation, a portion of the working fluid from the liquid reservoir is allocated to the power subcycle. It is pressurized by a pump and then sequentially heated by the hot working fluid in the recuperator and the air in the gas heater. Afterward, it passes through an expansion valve and undergoes expansion, simulating power generation. Meanwhile, another portion of the working fluid enters the cooling subcycle, goes through throttling, enters the evaporator for refrigeration, and is subsequently compressed by the compressor. The two streams of working fluid from the PSC and CSC mix and return to the condenser, completing the cycle.

3. EVALUATION MODEL

Based on the measured parameters, we estimated the thermodynamic performance of the system, including predicted net power, cooling capacity, energy efficiency, and COP. MATLAB software was used to establish the mathematical models. The mathematical equations for each component and system performance are described below.

Table 4 Evaluation model of CCPC components

Term	Energy equation
Pump	$W_{\text{pump}} = m_{\text{f,P}}(h_2 - h_1) = m_{\text{f,P}}(h_{2s} - h_1)\eta_{\text{pump}}$
Recuperator	$Q_{\text{re}} = m_{\text{f,P}}(h_3 - h_2) = m_{\text{f,P}}(h_5 - h_6)$
Gas heater	$Q_{\text{gh}} = m_{\text{f,P}}(h_4 - h_3)$
Expander	$W_{\text{exp}} = m_{\text{f,P}}(h_5 - h_4) = m_{\text{f,P}}(h_5 - h_{4s}) \cdot \eta_{\text{turb}}$
Mixing	$m_{\text{f,total}} h_{10} = m_{\text{f,R}} h_9 + m_{\text{f,P}} h_6$
Condenser	$Q_{\text{cond}} = m_{\text{f,total}}(h_{10} - h_1)$
Throttle valve	$m_{\text{f,R}} h_1 = m_{\text{f,R}} h_7$
Evaporator	$Q_{\text{ref}} = m_{\text{f,R}}(h_8 - h_7)$
Compressor	$W_{\text{comp}} = m_{\text{f,R}}(h_9 - h_8) = m_{\text{f,R}}(h_{9s} - h_8) / \eta_{\text{comp}}$
Net power	$W_{\text{net}} = W_{\text{exp}} - W_{\text{pump}} - W_{\text{comp}}$
Energy efficiency	$\eta_{\text{en}} = W_{\text{net}} / Q_{\text{gh}}$
COP	$COP = Q_{\text{ref}} / W_{\text{comp}}$

4. EXPERIMENTAL METHOD

This study is a preliminary exploratory experiment with CO_2/R32 working fluid. During the experimental process, due to limitations in mechanical conditions and control accuracy, there will inevitably be some fluctuations in the boundary conditions. When the fluctuations are less than 3%, they are considered acceptable.

The entire experiment used three strategies, including open-loop control, constant flow control, and flow-following control. This paper presents the results of open-loop control, where the pump load was set to 47.5%, the expansion valve opening was 35.15%, the throttle valve opening was 20.3%, and the compressor load was 70.0%.

The conditions of the heat sources and sinks in the system are as follows: the cooling water in the condenser is set at 0.5 m^3/h with an inlet temperature of 20.0°C, while the chilled water in the evaporator is set at 0.16 m^3/h with an inlet temperature of 20.0°C. The air, simulating the heat source, has a flow rate set at 200 kg/h, and the temperature varies in increments of 30°C, ranging from 180°C to 300°C.

The research objective is to explore the variation patterns of the output and the coupling effect of the sub-cycles under changing heat source conditions.

5. RESULT AND DISCUSSION

5.1 Change in key parameters

As the temperature of the heat source increases, the key parameters change in *Fig. 2*. The trends in the changes of these key parameters are as follows: The temperature and pressure before the expansion valve in the system increase significantly. The outlet pressure of the liquid reservoir remains relatively stable. The inlet pressure of the compressor slightly rises.

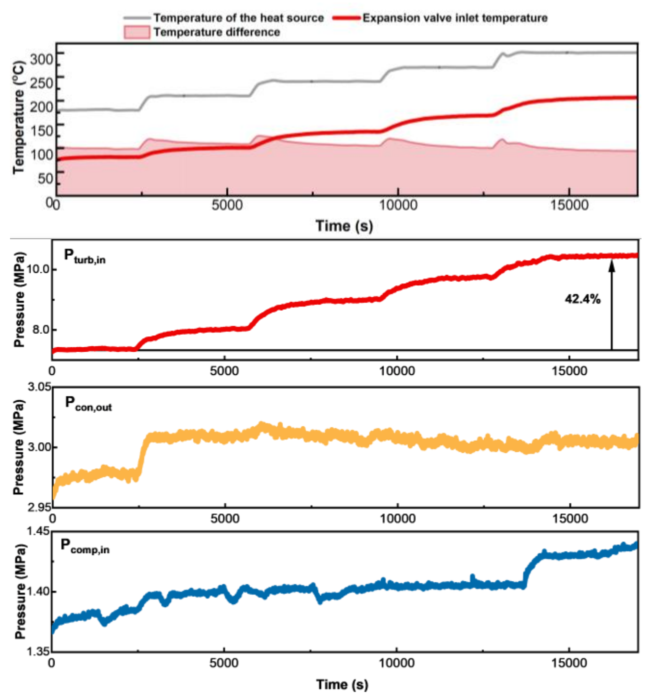

Fig. 2 Key parameters change with the increase of heat source temperature

These changes have a certain impact on the system's performance, which will be demonstrated in the following sections. Pressure ratios for the PSC and CSC are introduced for a more intuitive analysis in *Fig. 3*. The pressure ratio for PSC continuously increases, while the pressure ratio for CSC shows a decreasing trend.

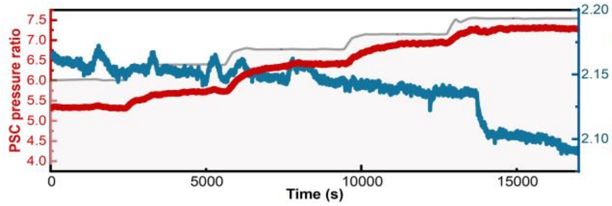

Fig. 3 Pressure ratio change with the increase of heat source temperature

5.2 Change in working fluid flow rate

The working fluid flow variations in PSC and CSC are shown in *Fig. 4*. As the temperature of the heat source rises, the power subcycle exhibits a higher pressure ratio, which results in increased flow resistance, and the volumetric efficiency of the pump decreases. Consequently, the working fluid flow rate within the power subcycle decreases from 94.9 kg/h to 89.7 kg/h.

Due to the diversion effect between PSC and CSC [12], as the working fluid flow rate into the power subcycle decreases, more working fluid enters the cooling subcycle. This leads to an increase in the working fluid flow rate into CSC, rising from 18.4 kg/h to 19.5 kg/h, primarily because of the lower resistance along the CSC.

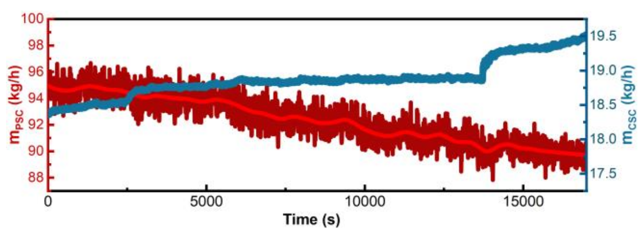

Fig. 4 Flow rate change with the increase of heat source temperature

5.3 Matching of heat source and working fluid

The heat matching of CCPC between the heat source and the working fluid is depicted in *Fig. 5*. As the temperature rises, the heat source can provide more heat. However, limited by the heat exchange capacity of the gas heater, the final heat supply stabilizes at 6.5 kW. Meanwhile, the working fluid can effectively absorb 6.2 kW of heat. The heat exchange efficiency of the CO_2/R32 mixture remains at around 95%, demonstrating good heat matching.

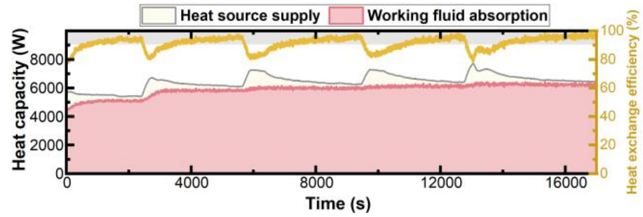

Fig. 5 Flow rate change with the increase of heat source temperature

5.4 Change in system output

The changes in energy conversion, predicted net power, and cooling capacity of CCPC as the heat source temperature rises are shown in *Fig. 6*. Under the open-loop control strategy, the fluid mechanical load and valve openings remain constant. With the increase in heat source temperature, the compressor consumption, pump consumption, and simulated expansion power all show an upward trend. Among these, the simulated expansion power exhibits the largest increase of 708W (200.6%), rising from 353W to 1061W, displaying a dominant trend. This increase is attributed to the higher inlet temperature and pressure of the simulated expander. The compressor's increase is 33W (11.2%) due to the increase in refrigeration cycle fluid flow, and the pump's consumption rises by 69W (47.4%) due to the higher compression ratio.

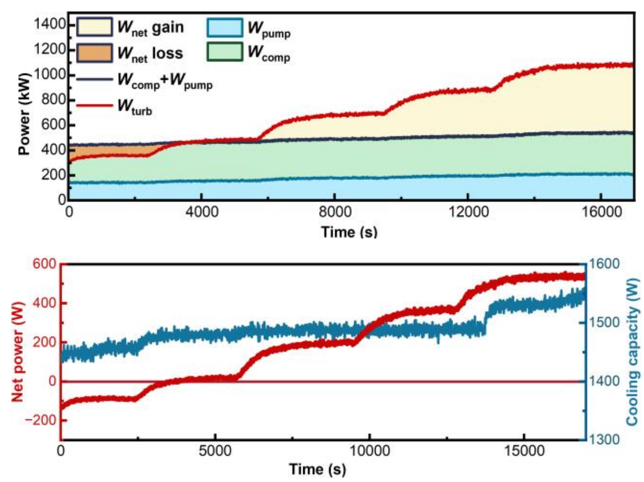

Fig. 6 Changes in energy conversion, net power and cooling capacity with the increase of heat source temperature

As the heat source temperature rises, the system's predicted net power consistently exhibits an upward trend, increasing from -90W to 536W. When the heat source reaches 210°C, the system becomes self-sustaining, with the power generated by the expansion turbine precisely covering the consumption of the compressor and pump. When the heat source temperature exceeds 210°C, the CCPC generates a surplus, while temperatures below 210°C require additional power consumption.

The cooling capacity increased from 1445W to 1550W, primarily due to the combined effect of the rising evaporator pressure and increased refrigerant flow rate.

5.5 Change in system efficiency and COP

The assessment of CCPC's efficiency and COP is presented in *Fig. 7*. As the heat source temperature increased, the system's energy efficiency showed an upward trend, rising from 4.15% to 13.91%, demonstrating a substantial potential for utilizing medium to low-grade thermal energy. On the other hand, the COP (Coefficient of Performance) of the refrigeration cycle initially remained at 4.82, then decreased and eventually stabilized at 4.71.

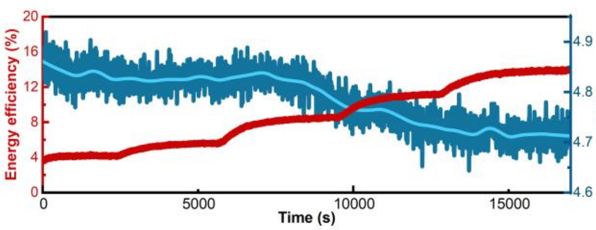

Fig. 7 Change in system efficiency and COP with the increase of heat source temperature

5.6 Compressor Blocked Discharge

In the aforementioned parameter changes, there were some peculiar variations in the refrigeration sub-cycle. For instance, between 13700s and 14000s, there were significant step changes in pressure, compression ratio, flow rate, and refrigeration capacity. As the CSC flow rate increased, the volumetric flow rate of gas inside the compressor gradually approached its maximum value, leading to the formation of a blocked condition around 13700s. This resulted in a rapid increase in pressure ahead of the compressor, a rapid decrease in the compression ratio of the CSC, and, to maintain system safety, the compressor autonomously increased its rotational speed and enhanced the volumetric efficiency. This caused a step change in the working fluid flow rate, leading to an increase in refrigeration capacity.

On the other hand, as the system approached the blocked condition (between 7000s and 14000s), there was a noticeable decrease in system COP due to reduced compressor efficiency. However, after the system autonomously adjusted, the COP stabilized, further confirming this phenomenon.

6. CONCLUSIONS

This study established a CO_2/R32 combined cooling and power cycle experimental bench and conducted preliminary experimental tests. The main exploration focused on the variations of key parameters and interactions between the power subcycle and the cooling subcycle under changing heat source conditions. The main conclusions are as follows:

(1) The CO_2/R32 combined cooling and power cycle is an effective solution for utilizing medium and low grade thermal energy. In an open-loop configuration, the system can achieve a maximum energy efficiency of 13.91%, and the maximum cooling COP can reach 4.82.

(2) The interference between the power subcycle and the cooling subcycle is mainly manifested in the effects at the condenser end and the distribution of flow. In this experiment, as the heat source temperature rises, the decrease in the working fluid flow in the power subcycle results in more flow being allocated to the cooling subcycle. The increase in the predicted net power comes from the rise in parameters before the expansion valve, while the increase in cooling capacity results from the growth of the flow rate.

(3) CO_2/R32 mixture did indeed improve the system's condensation requirements and operating pressure. The condenser outlet temperature for the CO_2/R32 system is approximately 27°C, and the condensation pressure is about 3.0MPa. Under the same condensation conditions, the CO_2 system approaches critical conditions, with a condensation pressure of up to 6.7MPa. The potential for a significant improvement in system performance has been demonstrated.

Due to the current progress of the experimental work, this paper has only presented preliminary research on CO_2/R32. In future work, comparative experiments between CO_2/R32 and pure CO_2 and pure R32 will be conducted, and variable composition experiments of CO_2/R32 will be further explored. These experiments will provide a deeper understanding of the working mechanisms of CO_2 mixtures and guide the practical applications of CO_2 mixtures.

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DECLARATION OF INTEREST STATEMENT

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper. All authors read and approved the final manuscript.

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