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Performance assessment of trigeneration system based on a renewable fuelled HCCI engine for the efficient production of power, heating, and cooling

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Abstract: This study conducted a comparative thermodynamic assessment of an integrated system that utilises a wet-ethanol fuelled HCCI engine to meet energy needs in a sustainable way. Utilising the waste heat from HCCI engines in conjunction with the Kalina cycle and $\text{NH}_3\text{-H}_2\text{O}$ -based absorption refrigeration cycle (ARC), the system can simultaneously provide power, cooling, and heating. Various parameters were used to analyse the integrated system's performance. Results show that incorporating Kalina and ARC cycles effectively utilises the waste heat from the HCCI engine, increasing the thermal efficiency of combined cycle by 23.78% to 52.0% and exergy efficiency by 14.55% to 39.60%.

Keywords: wet-ethanol; HCCI engine; waste heat recovery; Kalina cycle; absorption refrigeration cycle; ARC; exergy.

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1 Introduction

The global greenhouse gas emissions are a significant issue, especially in the energy and transportation industries. By giving priority to clean fuel alternatives and advancements in combustion systems, we can establish a sustainable future that meets our energy requirements while reducing pollution levels (Sydney et al., 2019).

Wet ethanol and homogeneous charge compression ignition (HCCI) engines are vital options in biofuels and modern engine systems, offering potential for renewable fuel and advanced combustion technology within the transportation sector. Wet ethanol is a promising biofuel derived from renewable sources (Gainey et al., 2023), while HCCI combustion combines the advantages of diesel and gasoline engines by using a compressed, lean fuel-air mixture to reduce PM and NO_x emissions while enhancing efficiency (Almatrafi and Siddiqui, 2024). The utilisation of waste heat from internal combustion (IC) engines can improve overall energy efficiency by generating additional power and cooling, leading to economic benefits while reducing harmful emissions. The efficient exploitation of these heat sources using the dual NH₃-H₂O mixture-based Kalina cycle has the potential to significantly improve both the efficiency and electricity production compared to traditional Rankine cycles (Meftahpour et al., 2024). Several researchers have investigated that the Kalina cycle not only surpasses traditional steam power cycles in efficiency but also holds significant potential as a bottoming cycle (Amiri Rad and Mohammadi, 2018; Shokati et al., 2015).

Around 35% of the fuel utilised by the vehicle's engine is allocated for the purposes of air-cooling and refrigeration of food during the operation of vapour-compression refrigerators, which are commonly used in vehicles for cooling and cold storage (Huang et al., 2019). To tackle this issue, the exploration of waste heat alternatives such as thermally activated absorption refrigerators presents innovative solutions for cooling requirements without heavy reliance on engine power (Xu and Wang, 2018). The absorption refrigeration cycle (ARC) employs ammonia-water or lithium bromide-water without the use of compressors; ammonia is preferred due to its efficient evaporation range suitable for temperatures between 50–200°C, making it the primary refrigerant fluid over LiBr-H₂O alternatives (Wu et al., 2020).

To the author's knowledge, there has been no investigation into the utilisation of a Kalina cycle in conjunction with the NH₃-H₂O-based ARC for the goal of efficiently

converting a substantial amount of waste heat generated by a wet-ethanol fuelled HCCI engine into useful electricity, cooling, and heating outcomes that meet diverse consumer requirements in an environmentally sustainable manner. The comprehensive thermodynamic analysis conducted in this study employing engineering equation solver (EES) software to determine the optimal design variables and configuration for maximising the energetic and exergetic performance of the integrated system. A comprehensive parametric assessment is examined to analyse the influences of significant parameters as engine speed, turbo-compressor pressure ratio, equivalence ratio, inlet pressure of separator, and evaporator temperature on achieving maximum thermal and exergetic performance of the integrated system and the HCCI engine. By harnessing exhaust heat from HCCI engines, we can achieve significant energy savings while ensuring that food remains fresh during transportation and passengers enjoy a comfortable journey. This comprehensive approach not only enhances the overall energy efficiency of the system but also reduces fuel consumption and emissions, offering a more environmentally friendly solution in line with current sustainability initiatives.

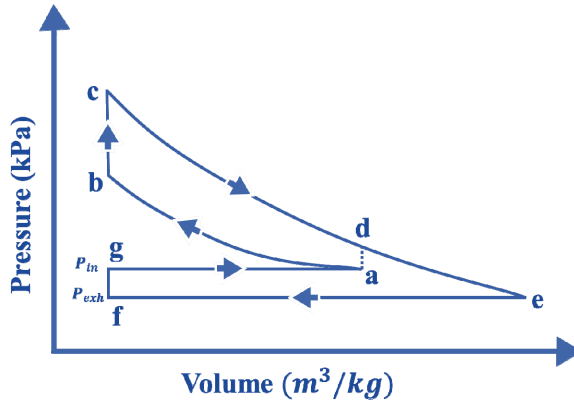
2 System description

Figure 1 displays the schematic layout of our innovative integrated system, which is powered by wet ethanol fuelled HCCI engine.

Ambient air (1) enters the compressor which increases its pressure and temperature. The compressed air is heated (2–3) in regenerator and mixed with water and ethanol mixture (4) in a vaporiser, where it vaporises and forms a homogeneous mixture (air-water-ethanol) (5) and flows into the HCCI engine. Subsequently, this mixture combined gases (residual) within the cylinder, and exhaust gases (6) are directed to the catalytic convertor. During the conversion process within the converter, the temperature of exhaust gases is raised as unburned harmful gases from the engine exhaust are transformed into harmless gases and then fed into the turbine at high temperatures (7), subsequently generating energy to run the compressor. The turbine's exhaust gases (8) transfers thermal energy to compressed air before leaving of regenerator (9). The exhaust gas through the regenerator (9) from the HCCI engine is recovered in the super heater (9–10), evaporator 1 (10–11), and economiser (11–12). Despite exiting the economiser (12) with a high temperature, the exhaust is further utilised through a process heater (12–13) to generate heating ($\dot{Q}_h$) for high temperature users, thereby enhancing system efficiency. Within the separator, the working fluid, which is a mixture of $\text{NH}_3\text{-H}_2\text{O}$, undergoes separation into two components: rich $\text{NH}_3\text{-H}_2\text{O}$ saturated vapour (22) and weak $\text{NH}_3\text{-H}_2\text{O}$ saturated liquid (23). The rich $\text{NH}_3\text{-H}_2\text{O}$ saturated vapour obtained from the separator (22) is then directed to the super heater for further heating, resulting in the production of superheated steam (14). This superheated steam is then utilised to generate mechanical power by expanding through turbine 2. The exhaust from the Kalina turbine, which is a part of the system, is then directed into the mixer (15). Conversely, in the $\text{NH}_3\text{-H}_2\text{O}$ operated ARC cycle, the generator receives the $\text{NH}_3\text{-H}_2\text{O}$ saturated liquid from the separator and supplies heat to produce refrigeration for the system. The output from the generator goes to the recuperator (24) to discharge some thermal power before passing via expansion valve 1 (25). The observed two-phase flow (26) combines with the Kalina turbine exhaust and then enters condenser 1 (16). The pressurised liquid (17) goes

- HCCI combustion demonstrates a high rate of combustion and, when timed correctly within the engine cycle, can exhibit similarities to the ideal Otto cycle. The optimal operation PV diagram of the HCCI process is illustrated in Figure 2.
- Pressure and heat loss are neglected.
- State of working substance is saturated liquid at outlet of condenser.
- It is assumed that the efficiencies of pumps and turbine operation isentropically are 85%.

Figure 2 PV diagram of the HCCI combustion process (see online version for colours)



3.2 Thermodynamic properties of an operating fluid

It is often assumed that air is the fluid utilised in the cylinder during an ideal engine cycle. However, real-life engine cycles involve a combination of various gases from different categories acting as the working fluid. It is crucial to emphasise that the specific heat of gases in actual engines varies with temperature, with its sensitivity to pressure being relatively insignificant. To adequately represent the correlation in our model for the variation of molar specific heat $\bar{C}_p(T)$, we have utilised five coefficients (a_1 – a_5) obtained from credible sources, covering the temperature range from 300 K to 3,000 K (Turns and Kraige, 2007; Siddiqui, 2023).

$$\bar{C}_p(T) = \bar{R}(a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^5) \quad (1)$$

3.3 HCCI engine

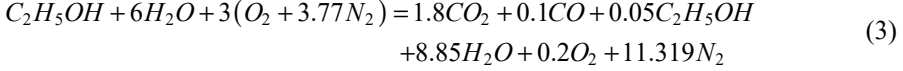
The operational aspects of HCCI engines can be analysed by employing the Otto cycle, as illustrated in Figure 2 and consider the efficiency of combustion 94% (Frias et al., 2007).

The chemical exergy pertaining to the wet-ethanol is computed by (Kotas, 1995)

$$e_{fuel}^{ch} = \left[1.0401 + 0.1728 \frac{a}{b} + 0.0432 \frac{c}{b} + 0.2169 \frac{d}{b} \left(1 - 2.0628 \frac{e}{b} \right) \right] * LHV \quad (2)$$

where a , b , c , d , and e are the mass fraction of hydrogen atom (H), carbon atom (C), oxygen atom (O), sulphur atom (S), and nitrogen atom (N), respectively.

In this study, the combustion gases are regarded as a combination of the following elements: carbon dioxide (CO_2), carbon monoxide (CO), ethanol ($\text{C}_2\text{H}_5\text{OH}$), water (H_2O), nitrogen (N_2), oxygen (O_2), and hydrogen (H_2). The equation below illustrates the combustion of wet-ethanol in the presence of compressed air as an oxidiser (Siddiqui et al., 2022).

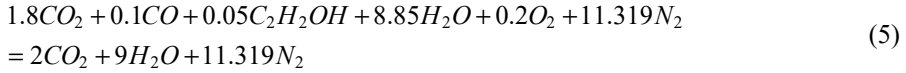


3.4 Catalytic converter

The implementation of an oxidising catalyst in HCCI engines is crucial for effectively managing escaped fuel and CO pollutants, as the HCCI engine generates very little NO_x . The Pd-loaded $\text{SiO}_2\text{-Al}_2\text{O}_3$ can burn the unburnt HC and CO. By employing the balances of enthalpy and mass of the species, we are able to accurately determine the constituents of engine exhaust and its temperature, adopting complete combustion.

$$\sum_{\text{Reactants}} n_i (\bar{h}_f^0 + \Delta \bar{h})_e = \sum_{\text{Products}} n_e (\bar{h}_f^0 + \Delta \bar{h})_e \quad (4)$$

where i denotes the reactants to CC and e represents products from the CC. The coefficients n_i and n_e represent the number of reactants and products per mole of fuel, correspondingly. The chemical equation that is considered for the reaction that occurs in the catalytic converter may be shown as (Siddiqui et al., 2022)



3.5 Energy and exergy analyses

When analysing a control volume in a state of equilibrium, it is crucial to take into account the conservation of mass and energy. These principles allow us to comprehend the movement of mass and energy within the system, while assuming negligible changes in potential and kinetic energy. Consequently, we can express these principles using the following equations (Dincer and Rosen, 2013)

$$\sum \dot{Q} + \sum \dot{m}_i h_i = \sum \dot{W} + \sum \dot{m}_o h_o \quad (6)$$

It can be stated that a thermodynamic system, during a reversible transition process from environmental state to ambient state, capable of generating the highest amount of work known as exergy. The exergy balance equation can be mathematically expressed as (Bejan and Tsatsaronis, 1996)

$$\dot{E}_{x,Q} + \sum \dot{m}_i e_{x,i} = \dot{E}_{x,W} + \sum \dot{m}_e e_{x,e} + \dot{E}_{x,D} \quad (7)$$

where $\dot{E}_{x,Q}$ and $\dot{E}_{x,W}$ are the exergy accompanied by rate of heat and work transfer, $\dot{m}$ represents the mass flow rate of the stream, $\dot{E}_{x,D}$ represents the exergy destruction rate in the process and e_x is the exergy flow, which can be determined as

$$e_x = (h - h_0) - T_0 (s - s_0) \quad (8)$$

The mathematical formulation of balancing equations of exergy and energy for the individual component of integrated system is developed after employing the above general equations and resulting expressions are shown in Table 1.

Table 1 Balance equations of exergy and energy for the all components of HCCI engine-based integrated system

Component		Balance equations
Compressor	Energy balance	$\dot{W}_C = \dot{m}_a (h_2 - h_1)$
	Exergy balance	$\dot{E}_{x,D,C} = \dot{m}_1 (e_{x,1} - e_{x,2}) + \dot{W}_C$
Regenerator	Energy balance	$\dot{m}_a (h_3 - h_2) = \dot{m}_{exh} (h_8 - h_9)$
	Exergy balance	$\dot{E}_{x,D,Reg} = \dot{m}_2 (e_{x,2} - e_{x,3}) + \dot{m}_8 (e_{x,8} - e_{x,9})$
Fuel vaporiser	Energy balance	$\dot{m}_a h_3 + \dot{m}_f h_4 = (\dot{m}_a + \dot{m}_f) h_5$
	Exergy balance	$\dot{E}_{x,D,FV} = \dot{m}_3 e_{x,3} + \dot{m}_4 e_{x,4} - \dot{m}_5 e_{x,5}$
HCCI engine (compression) process, 1'-2'	Energy balance	$(\dot{m}_a + \dot{m}_f + \dot{m}_{rg})(h_{2'} - h_{1'}) = \dot{W}_{comp} - \dot{Q}_{surr}$
	Exergy balance	$\dot{E}_{x,D,comp} = \dot{m}_5 (e_{x,1'} - e_{x,2'}) + \dot{W}_{comp} - \dot{Q}_{surr} \left(1 - \frac{T_0}{T}\right)$
Heat addition process, 2'-3'	Energy balance	$(\dot{m}_a + \dot{m}_f + \dot{m}_{rg})(h_{3'} - h_{2'}) = \dot{Q}_H$
	Exergy balance	$\dot{E}_{x,D,Heat\ addition} = \dot{m}_5 (e_{x,2'} - e_{x,3'}) + \dot{Q}_H \left(1 - \frac{T_0}{T_H}\right)$
Expansion process, 3'-4'	Energy balance	$(\dot{m}_a + \dot{m}_f + \dot{m}_{rg})(h_{3'} - h_{4'}) = \dot{W}_{exp} + \dot{Q}_{surr}$
	Exergy balance	$\dot{E}_{x,D,exp} = \dot{m}_5 (e_{x,3'} - e_{x,4'}) - \dot{W}_{exp} - \dot{Q}_{surr} \left(1 - \frac{T_0}{T}\right)$
Blowdown process, 4'-5'	Energy balance	$\dot{m}_{exh} h_{4'} = \dot{m}_{exh} h_{5'} + \dot{Q}_{surr}$
	Exergy balance	$\dot{E}_{x,D,exh\ \&\ blowdown} = \dot{m}_5 (e_{x,4'} - e_{x,5'}) - \dot{Q}_{surr} \left(1 - \frac{T_0}{T}\right)$
Catalytic convertor	Energy balance	$\sum_{Product} n_e (\bar{h}_f^\circ + \Delta \bar{h})_e = \sum_{Reactant} n_i (\bar{h}_f^\circ + \Delta \bar{h})_i$
	Exergy balance	$\dot{E}_{x,D,CC} = \dot{m}_6 e_{x,6} - \dot{m}_7 e_{x,7} + \dot{m}_6 e_{ch}$

Table 1 Balance equations of exergy and energy for the all components of HCCI engine-based integrated system (continued)

<i>Component</i>		<i>Balance equations</i>
Turbine 1	Energy balance	$\dot{W}_{T1} = \dot{m}_{exh}(h_7 - h_8)$
	Exergy balance	$\dot{E}_{x,D,T1} = \dot{m}_7(e_{x,7} - e_{x,8}) - \dot{W}_{T1}$
Super heater	Energy balance	$\dot{m}_9(h_9 - h_{10}) = \dot{m}_{22}(h_{14} - h_{22})$
	Exergy balance	$\dot{E}_{x,D,SH} = \dot{m}_9(e_{x,9} - e_{x,10}) + \dot{m}_{22}(e_{x,22} - e_{x,14})$
Evaporator 1	Energy balance	$\dot{m}_{10}(h_{10} - h_{11}) = \dot{m}_{21}(h_{21} - h_{20})$
	Exergy balance	$\dot{E}_{x,D,E1} = \dot{m}_{10}(e_{x,10} - e_{x,11}) + \dot{m}_{20}(e_{x,20} - e_{x,21})$
Economiser	Energy balance	$\dot{m}_{11}(h_{11} - h_{12}) = \dot{m}_{19}(h_{20} - h_{19})$
	Exergy balance	$\dot{E}_{x,D,Eco} = \dot{m}_{11}(e_{x,11} - e_{x,12}) + \dot{m}_{19}(e_{x,19} - e_{x,20})$
Process heater	Energy balance	$\dot{Q}_{PH} = \dot{m}_{12}(h_{12} - h_{13})$
	Exergy balance	$\dot{E}_{x,D,PH} = \dot{m}_{12}e_{x,12} - \dot{m}_{13}e_{x,13} + \dot{m}_{45}e_{x,45} - \dot{m}_{46}e_{x,46}$
Separator	Energy balance	$\dot{m}_{21}h_{21} = \dot{m}_{22}h_{22} + \dot{m}_{23}h_{23}$
	Exergy balance	$\dot{E}_{x,D,S} = \dot{m}_{21}e_{x,21} - \dot{m}_{22}e_{x,22} - \dot{m}_{23}e_{x,23}$
Turbine 2	Energy balance	$\dot{W}_{T2} = \dot{m}_{14}(h_{14} - h_{15})$
	Exergy balance	$\dot{E}_{x,D,T2} = \dot{m}_{14}(e_{x,14} - e_{x,15}) - \dot{W}_{T2}$
Mixer	Energy balance	$\dot{m}_{16}h_{16} = \dot{m}_{15}h_{15} + \dot{m}_{26}h_{26}$
	Exergy balance	$\dot{E}_{x,D,M} = \dot{m}_{15}e_{x,15} + \dot{m}_{26}e_{x,26} - \dot{m}_{16}e_{x,16}$
Condenser 1	Energy balance	$\dot{Q}_{Cond1} = \dot{m}_{16}(h_{16} - h_{17})$
	Exergy balance	$\dot{E}_{x,D,Cond1} = \dot{m}_{16}(e_{x,16} - e_{x,17}) - \dot{Q}_{Cond1} \left(1 - \frac{T_0}{T_{Cond1}} \right)$
Pump 1	Energy balance	$\dot{W}_{P1} = \dot{m}_{17}(h_{18} - h_{17})$
	Exergy balance	$\dot{E}_{x,D,P1} = \dot{m}_{17}(e_{x,17} - e_{x,18}) + \dot{W}_{P1}$
Recuperator	Energy balance	$\dot{m}_{18}(h_{19} - h_{18}) = \dot{m}_{24}(h_{24} - h_{25})$
	Exergy balance	$\dot{E}_{x,D,rec} = \dot{m}_{18}(e_{x,18} - e_{x,19}) + \dot{m}_{24}(e_{x,24} - e_{x,25})$

Table 1 Balance equations of exergy and energy for the all components of HCCI engine-based integrated system (continued)

Component		Balance equations
Expansion valve (EV1)	Energy balance	$h_{25} = h_{26}$
	Exergy balance	$\dot{E}_{x,D,EV1} = \dot{m}_{25}(e_{x,25} - e_{x,26})$
Generator	Energy balance	$\dot{m}_{23}h_{23} - \dot{m}_{24}h_{24} = \dot{m}_{27}h_{27} + \dot{m}_{34}h_{34} - \dot{m}_{33}h_{33}$
	Exergy balance	$\dot{E}_{x,D,gen} = \dot{m}_{23}(e_{x,23} - e_{x,24}) + \dot{m}_{33}e_{x,33} - \dot{m}_{27}e_{x,27} - \dot{m}_{34}e_{x,34}$
Condenser-2	Energy balance	$\dot{Q}_{Cond2} = \dot{m}_{27}(h_{27} - h_{28})$
	Exergy balance	$\dot{E}_{x,D,Cond2} = \dot{m}_{27}(e_{x,27} - e_{x,28}) - \dot{Q}_{Cond1}\left(1 - \frac{T_0}{T_{Cond2}}\right)$
Expansion valve (EV2)	Energy balance	$h_{28} = h_{29}$
	Exergy balance	$\dot{E}_{x,D,EV2} = \dot{m}_{28}(e_{x,28} - e_{x,29})$
Evaporator 2	Energy balance	$\dot{Q}_{E2} = \dot{m}_{29}(h_{30} - h_{29})$
	Exergy balance	$\dot{E}_{x,D,E2} = \dot{m}_{29}(e_{x,29} - e_{x,30}) - \dot{m}_{39}(e_{x,39} - e_{x,40})$
Absorber	Energy balance	$\dot{Q}_{abs} = \dot{m}_{30}h_{30} + \dot{m}_{36}h_{36} - \dot{m}_{31}h_{31} = \dot{m}_{37}(h_{38} - h_{37})$
	Exergy balance	$\dot{E}_{x,D,abs} = \dot{m}_{30}e_{x,30} + \dot{m}_{36}e_{x,36} - \dot{m}_{31}e_{x,31} + \dot{m}_{37}e_{x,37} - \dot{m}_{38}e_{x,38}$
Solution pump-2	Energy balance	$\dot{W}_{P2} = \dot{m}_{31}(h_{32} - h_{31})$
	Exergy balance	$\dot{E}_{x,D,P2} = \dot{m}_{31}(e_{x,31} - e_{x,32}) + \dot{W}_{P2}$
Heat exchanger	Energy balance	$\dot{m}_{32}(h_{33} - h_{32}) = \dot{m}_{34}(h_{34} - h_{35})$
	Exergy balance	$\dot{E}_{x,D,HE} = \dot{m}_{32}(e_{x,32} - e_{x,33}) + \dot{m}_{34}(e_{x,34} - e_{x,35})$
Expansion valve (EV3)	Energy balance	$h_{35} = h_{36}$
	Exergy balance	$\dot{E}_{x,D,EV3} = \dot{m}_{35}(e_{x,35} - e_{x,36})$

3.6 Performance parameters

The thermal efficiency of integrated system can be determined as

$$\eta_{th,comb} = \frac{\dot{W}_{HCCI} + \dot{W}_{KC} + \dot{Q}_h + \dot{Q}_{ref}}{\dot{m}_f LHV \eta_C} \quad (9)$$

where $\dot{W}_{HCCI}$ represents the HCCI engine power output, $\dot{W}_{KC}$ represents the net power output of Kalina cycle, $\dot{Q}_h$ is the heating capacity and the cooling capacity, denoted as $\dot{Q}_{ref}$. The product of the LHV of the fuel, the flowing rate of mass of fuel ($\dot{m}_f$) and the combustion efficiency (η_c) is the total energy supplied to the system.

Exergetic efficiency of the combined cycle can be determined as

$$\eta_{ex,comb} = \frac{\dot{W}_{HCCI} + \dot{W}_{KC} + \dot{E}_{x,h} + \dot{E}_{x,ref}}{\dot{m}_f LHV \eta_c \varphi} \quad (10)$$

where φ represents the ratio of fuel exergy to energy and $\dot{E}_{x,ref}$ is the exergy attributed to the refrigeration capacity ($\dot{Q}_{ref}$) and $\dot{E}_{x,h}$ is the exergy attributed to the heating capacity ($\dot{Q}_h$).

Table 2 Operational parameters employed for the computational analysis of an integrated system driven by HCCI engine

<i>Parameter</i>	<i>Value</i>
Swept volume	2,400 cm ³
Stroke	16.5 cm
Bore	13.7 cm
Connecting rod length	26.2 cm
Volumetric efficiency	100%
Regenerator effectiveness	0.78
Compression ratio	16:1
Turbocharger efficiency	80%
Ethanol (LHV)	26,900 kJ/kg
Engine speed	1,200–2,400 rpm
Equivalence ratio	0.3–0.7
Turbo-compressor pressure ratio	2.4–3.6
Inlet pressure of separator	5,700–89,00 kPa
Evaporator temperature	270–282 K
ARC Condenser temperature	308 K
Heat exchanger effectiveness	0.8
Isentropic efficiency of pump and turbine	85%

4 Model validation

The study presents a model powered by a wet-ethanol-fuelled HCCI engine-based combined system, which has been validated using data from previous studies (Khaliq et al., 2011; Júnior et al., 2019; Sun, 1998). This validation offers valuable insights into the performance of the model for both the HCCI engine and the combined system. The thermal efficiency of the wet-ethanol fuelled HCCI engine showed significant improvements, deviating by 6.80% from the findings of Khaliq et al. (2011). Moreover,

the thermal and exergy efficiencies of the Kalina cycle also exhibited notable enhancements, with deviations of 2.79% and 1.50% from reference study reported by Júnior et al. (2019). Additionally, the calculated energetic COP for the $\text{NH}_3\text{-H}_2\text{O}$ -based ARC cycle closely aligns with the values reported by Sun (1998), with only a 0.39% deviation. These results emphasise the potential benefits of adopting the proposed model for HCCI engine and combined system, as it can lead to more energy-efficient operations and better utilisation of resources.

5 Results and discussion

This section presents a thermodynamic analysis of an HCCI engine fuelled by a wet ethanol-driven integrated system for power, heating, and cooling. The analysis examines the impact of different design parameters such as engine speed, turbo-compressor pressure ratio, equivalence ratio, inlet pressure of separator, and evaporator temperature on achieving maximum thermal and exergetic efficiencies of the combined system and the efficiencies of the wet-ethanol fuelled HCCI engine. Table 2 displays the main operational parameters of the system proposed in Figure 1 for conducting thermodynamic analysis.

Figure 3(a) shows the effects of varying the turbo-compressor pressure ratio (r_p) on different output of the HCCI engine incorporated with the Kalina and ARC sub-system. The results clearly indicate that increasing the turbo-compressor pressure ratio leads to higher power output in the HCCI system, as well as increased Kalina turbine output, heating output, and refrigeration output. It demonstrates that increasing the pressure ratio enhances fuel-air mixing and charge density, resulting in increased engine power output. This also raises the temperature of the engine exhaust, thereby enhancing the heat transfer to the working medium of the bottoming cycle. As a result, the heating output, Kalina turbine output, and refrigeration output are all increased. The net work rate of the HCCI engine rises from 115.8 kW to 164.8 kW, while the power output of the Kalina turbine rises from 9.24 kW to 11.33 kW. Additionally, the heating output increases from 14.18 kW to 19.1 kW, and the refrigeration output increases from 5.57 kW to 7.31 kW, all achieved through a rise in the pressure ratio of the turbo-compressor from 2.4 to 3.6.

Figure 3(b) illustrates the influence of the turbo-compressor pressure ratio (r_p) on thermal and exergy efficiencies of HCCI engine driven integrated system for power, heating and refrigeration. It is clear from the figure that the efficiencies of HCCI and combined system rises with the rise in the pressure ratio of turbo-compressor. By raising the pressure ratio of the turbo-compressor from 2.4 to 3.6, the HCCI thermal efficiency rises from 38.44% to 44.38%, and the combined system thermal efficiency rises from 48.06% to 54.54%. On the other hand, the exergy efficiencies generally show lower values as a result of the greater input of fuel exergy in comparison to fuel energy. The exergy efficiency of HCCI increases from 31.04% to 37.11% when the turbo-compressor pressure ratio is increased from 2.4 to 3.6. Similarly, the exergy efficiency of the combined system increases from 35.87% to 42.25% under the same conditions.

Figure 4(a) depicts the changes of equivalence ratio on diverse outputs of wet-ethanol driven HCCI engine-based integrated system for power, heating and cooling. The figure reveals that as the equivalence ratio increase, the energetic outputs of the combined system improves. This phenomenon occurs because a higher equivalence ratio enables

the release of higher energy, thereby leading to an improve in in-cylinder temperature and pressure and subsequently resulting in enhanced HCCI engine output. The rise in in-cylinder temperature causes an increase in engine exhaust energy, which is transferred as heat to the working medium of the Kalina cycle, resulting in improved turbine power. Subsequently, the high-temperature ammonia-water mixture discharged from the separator is directed to the generator to serve as a heat source for the ARC, thereby boosting the cooling capacity. The net work rate of HCCI engine rises from 137.2 kW to 142.1 kW, and the Kalina turbine's power output improves from 10.02 kW to 10.45 kW. Moreover, the heating output goes up from 16.65 kW to 16.74 kW, and the refrigeration output improves from 6.40 kW to 6.49 kW, all as a result of increasing the equivalence ratio from 0.3 to 0.7.

Figure 3 Effect of turbo-compressor pressure ratio on, (a) diverse outputs (b) the thermal and exergy efficiencies of the proposed system (see online version for colours)

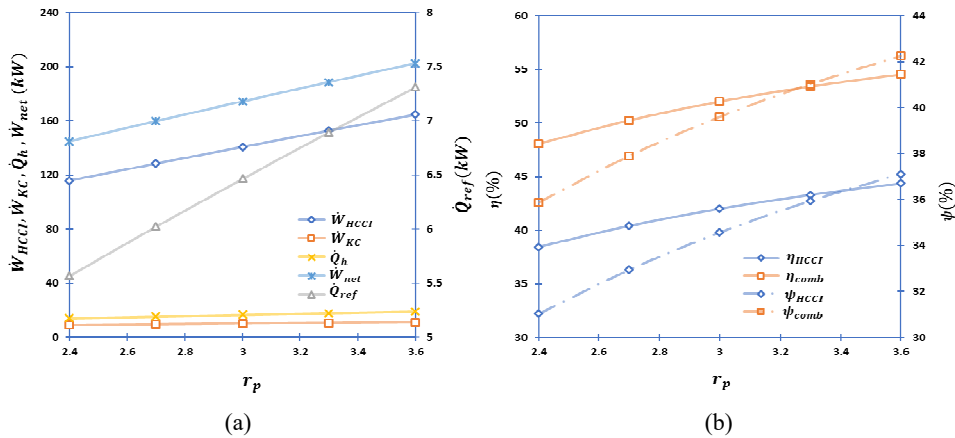


Figure 4 Impact of equivalence ratio on, (a) diverse outputs (b) the thermal and exergy efficiencies of the proposed system (see online version for colours)

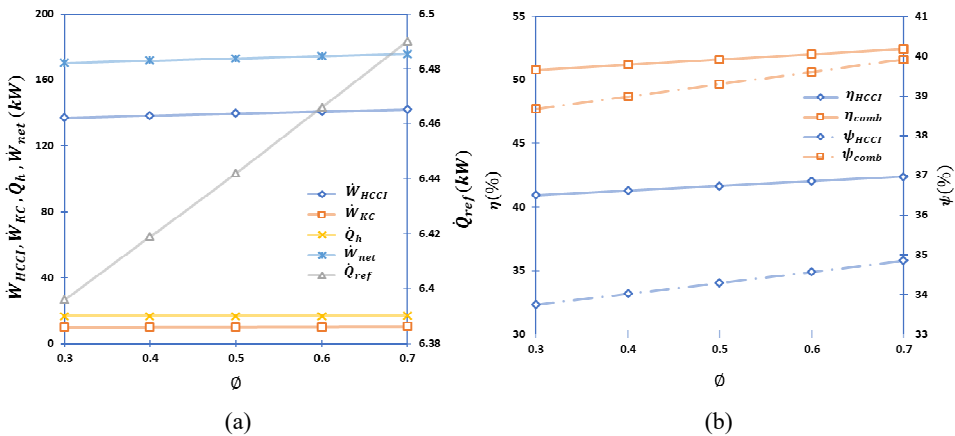


Figure 4(b) illustrates the efficiencies of the proposed system with change in equivalence ratio. The efficiencies of proposed system experience an improvement, as equivalence

ratio rises. By increasing the equivalence ratio from 0.3 to 0.7, there is an observed rise in the thermal efficiency of the HCCI engine and combined cycle, from 40.93% to 42.37% and from 50.79% to 52.41%, respectively. Additionally, the exergy efficiency of the HCCI engine and combined cycle demonstrates an improvement from 33.74% to 34.85% and from 38.67% to 39.91%, respectively.

Figure 5(a) depicts the effects of varying the engine speed (N) on different output parameters of the wet ethanol powered HCCI engine integrated with Kalina and ARC cycles. It is evident that with the rise of engine speed, the HCCI power output, Kalina turbine output, heating output and refrigeration output increases. This is because a higher engine speed enhances the charge-flow intensity, resulting in a more effective combination of fuel and air mixture. This leads to a more efficient combustion process and ultimately improves the overall power output of the engine. Raising the engine speed results in a decrease in heat loss to the cylinder wall because of the shorter combustion duration leads to a higher amount of fuel energy exiting the combustion zone as exhaust energy, serving as a heat source for the Kalina/ARC cycles and meeting the required loads. These factors combine to improve the performance of both the HCCI engine and integrated system. The net work rate of the HCCI engine increases from 132.61 kW to 149.51 kW, while the power output of the Kalina turbine rises from 9.74 kW to 10.97 kW. Additionally, the heating output improves from 15.77 kW to 17.78 kW, and the refrigeration output increases from 6.04 kW to 6.86 kW, all achieved through an increase in the engine speed from 1,200 rpm to 2,400 rpm.

Figure 5(b) illustrates the variation of the engine speed (N) on efficiencies of wet-ethanol fuelled HCCI engine driven integrated system for power, heating and refrigeration. According to the figure the efficiencies of HCCI and combined system improves with the rise in the engine speed. By increasing the engine speed from 1,200 rpm to 2,400 rpm, there is an observed increase in the thermal efficiency of the HCCI engine and combined cycle, from 39.55% to 44.59% and from 48.96% to 55.21%, respectively. Furthermore, the exergy efficiency of the HCCI engine and combined cycle shows an increase from 32.89% to 36.30% and from 37.68% to 41.59%, respectively.

In a proposed wet-ethanol fuelled HCCI engine-based integrated system, altering the inlet pressure of separator ($P_{m,s}$) leads to varying effects on different output parameters, as displayed in Figure 6(a). The figure illustrates that the vapour at a higher pressure from the separator generates increased power output when it passes through the Kalina turbine at a constant back pressure, attributable to an elevation in the specific enthalpy drop. Nevertheless, the power produced by the Kalina turbine decreases as a result of a reduction in vapour mass flow rate. The power produced by the Kalina turbine rises with the rising inlet pressure of the separator. However, at a certain point, the influence of expansion ratio on power output turns less prominent compared to that of reduced flowing rate of vapour mass, leading to a subsequent decrease in power production. Similarly, as the inlet pressure of the separator rises, it has a direct impact on the flowing rate of liquid mass leaving the separator in a saturated condition. The flowing rate of mass rises as the inlet pressure of the separator rises. This rise in mass flow rate is significant because it plays a crucial role in enhancing the heat input to the ARC, resulting in a higher cooling output. The refrigeration output improves from 5.77 kW to 6.98 kW as a result of increasing the inlet pressure of separator from 5,700 kPa to 8,900 kPa. The net work rate of the HCCI engine and heating output remain unaffected

by the increase in expander entry pressure, as the operative conditions of the engine remain consistent.

Figure 5 (a) Effect of engine speed on the different outputs (b) The thermal and exergy efficiencies of the wet-ethanol fuelled-based proposed integrated system (see online version for colours)

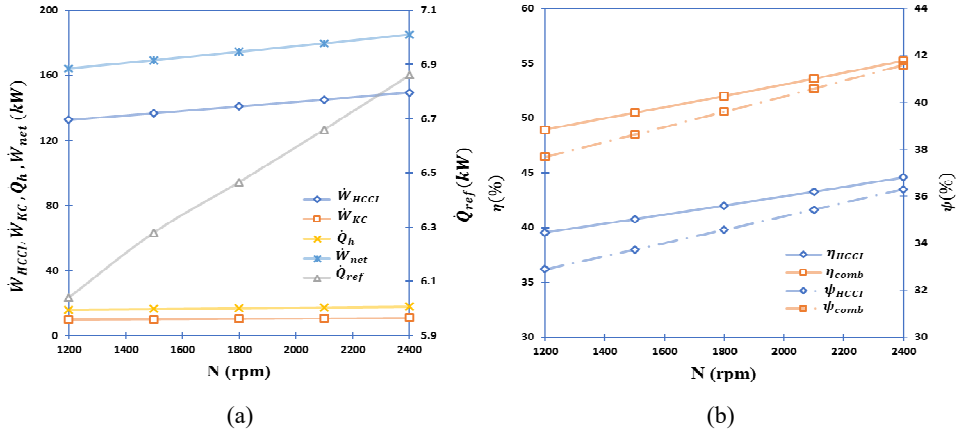
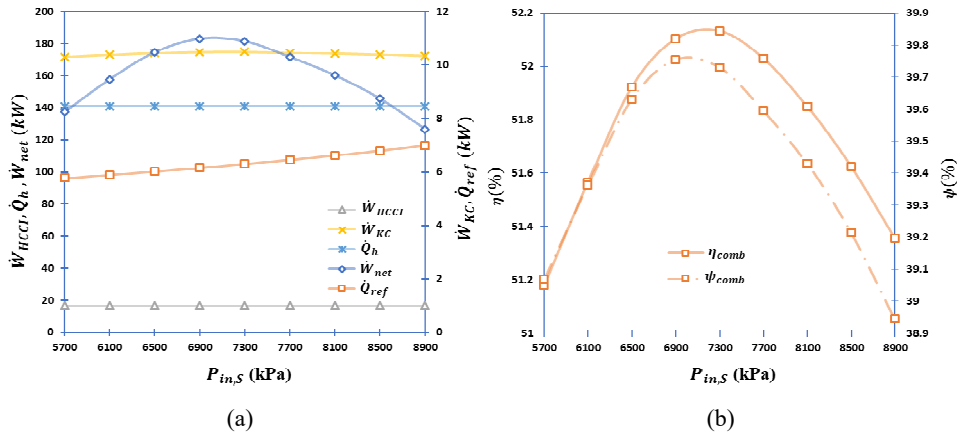


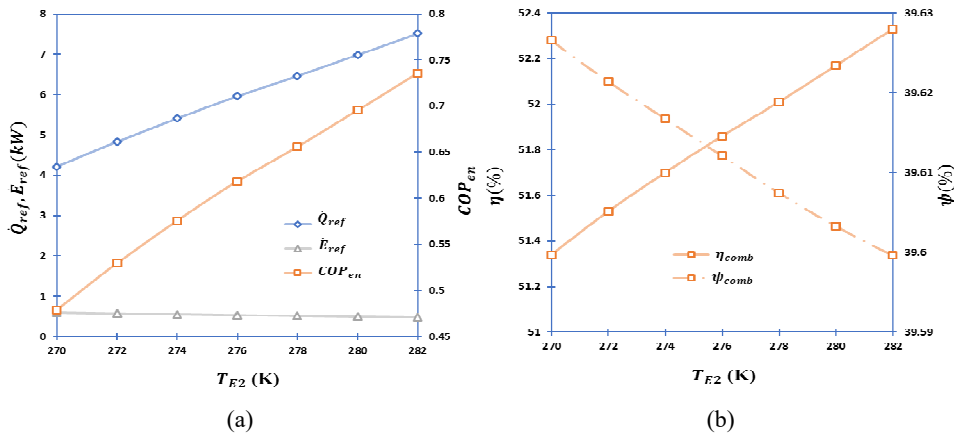
Figure 6 Variation of inlet pressure of separator on, (a) diverse outputs (b) the thermal and exergy efficiencies of the proposed system (see online version for colours)



Results presented in Figure 6(b) clearly demonstrate the impact of rising the inlet pressure of separator ($P_{in,S}$) on the thermal and exergy efficiencies of combined system. The figure demonstrates a clear pattern in the thermal and exergy efficiencies of the combined cycle, showing an initial rise followed by a decline as the inlet pressure of the separator increases from 5,700 kPa to 8,900 kPa. This can be attributed to the more substantial reduction in power output of the Kalina turbine compared to the increase in refrigeration output of ARC as mentioned in Figure 6(b). As a result, there is a corresponding relationship between the thermal and exergy efficiency of the integrated systems and the power output of Kalina turbine.

Figure 7(a) illustrates the impact of varying the evaporator temperature (T_{E2}) on refrigeration output, exergy of refrigeration output and energetic COP of $\text{NH}_3\text{-H}_2\text{O}$ -based ARC cycle. The refrigeration output and COP of the ARC cycle demonstrate a rise in response to a rise in evaporator temperature. This phenomenon occurs because the rise in evaporator temperature contributes to a rise in heat transfer to the ARC, resulting a higher flowing rate of mass from the ARC generator. The refrigeration output increases from 4.21 kW to 7.52 kW, and the energetic COP rises from 0.4792 to 0.7359 as the evaporator temperature is raised from 270 K to 282 K. Conversely, there is a declining pattern in the exergy of refrigeration output with the rise in evaporator temperature. As per the definition of exergy of refrigeration output, a rise in evaporator temperature has an adverse impact on the exergy of refrigeration output and also results in lower exergy associated with refrigeration output in comparison to the refrigeration output itself. The exergy of refrigeration output declines from 0.59 kW to 0.48 kW as a result of rising the evaporator temperature from 270 K to 282 K.

Figure 7 Impact of evaporator temperature on, (a) refrigeration output, exergy of refrigeration output and energetic COP of ARC cycle (b) the thermal and exergy efficiencies of proposed system (see online version for colours)

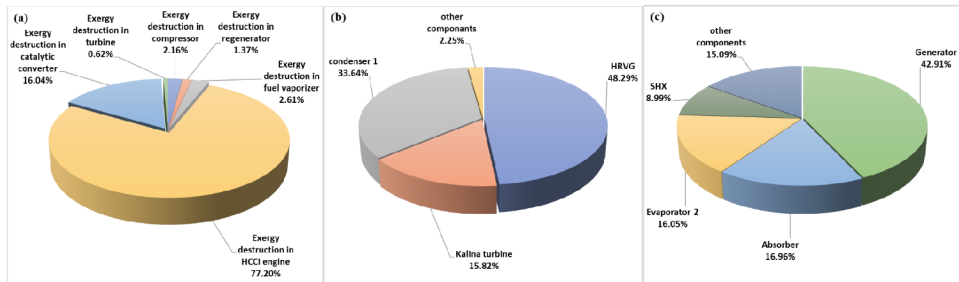


Results presented in Figure 7(b) clearly demonstrate the efficiencies of system with change in evaporator temperature (T_{E2}). It is evident that rising the temperature of evaporator leads to a rise in the thermal efficiency of the combined system, but it leads to a decrease in the exergy efficiency of the system. It is interesting to note that the ARC generator is capable of absorbing more heat as the evaporator temperature rises. This has a direct effect on the refrigeration output, as it also increases with the elevation of the evaporator temperature. The variations in thermal and exergy efficiency are closely connected to these changes in refrigeration output and the exergy of refrigeration output. It is important to consider that while refrigeration output shows a positive trend with evaporator temperature, the impact on the exergy of refrigeration output is negative. As a result, as evaporator temperature rises from 270 K to 282 K, the thermal efficiency of the combined cycle rises from 51.34% to 52.33%, while the exergy efficiency of combined cycle declines slightly from 39.63% to 39.59%.

Figure 8 depicts the shares of exergy destruction rate in the individual components of the HCCI engine system, Kalina cycle, and ARC cycle. The analysis of exergy shows

that the HCCI engine in the wet-ethanol fuelled HCCI engine system has the highest exergy destruction rate accounting for 77.20% due to significant irreversibilities in the combustion process, as illustrated in Figure 8(a). Following this, the catalytic converter ranks second in terms of exergy destruction rate, accounting for 16.04% due to irreversibilities arising from burning unburnt fuel exiting the engine. Additionally, in the Kalina cycle, HRVG (comprising super heater, evaporator, and economiser) exhibits the maximum exergy destruction rate at 48.29% followed by condenser 1 at 33.64% and Kalina turbine at 15.82% as shown in Figure 8(b). Finally, within the ARC cycle, the generator shows the highest amount of exergy destruction rate at 42.91%, followed by absorber at 16.96% and evaporator 2 at 16.05% as depicted in Figure 8(c).

Figure 8 Shares of exergy destruction rate within the main components of the proposed system for, (a) HCCI engine system (b) Kalina cycle (c) ARC cycle (see online version for colours)



6 Conclusions

This study conducted a thermodynamic evaluation of an integrated system powered by wet-ethanol fuelled HCCI engine, aiming to fulfil the energy requirements (electricity, heating, and cooling) for diverse consumer needs in an environmentally sustainable manner. A comprehensive parametric analysis is examined to analyse the influences of significant parameters on achieving maximum thermal and exergetic efficiencies of the integrated system and the HCCI engine. The study found that the HCCI engine, without employing a Kalina and ARC cycles, exhibited thermal and exergy efficiencies of 42.01% and 34.57%, respectively. Although, the integration of Kalina and ARC cycles led to notably increased system efficiencies of 52.00% and 39.60%, respectively. The power generated by the HCCI engine and the Kalina turbine, as well as their heating and cooling capacities, have been observed to rise from 142.35 kW to 160.48 kW, 15.77 kW to 17.78 kW, and 6.04 kW to 6.86 kW, respectively, as the engine speed is varied from 1,200 rpm to 2,400 rpm. Furthermore, The HCCI engine, HRVG, and generator are identified as the primary exergy destructive components, exhibiting exergy dissipation of 77.20%, 48.29%, and 42.91% within the HCCI engine system, Kalina cycle, and ARC cycle, respectively.

HCCI engines face significant obstacles before widespread implementation, including limited operational range and challenges in regulating combustion. These issues can lead to knocking under high loads or rich fuel mixtures and misfiring under low loads or lean mixtures. Addressing these technical challenges through the development of advanced

control models is crucial for the successful integration and widespread adoption of HCCI engines in the future.

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References

- Almatrafi, E. and Siddiqui, M.A. (2024) 'Thermodynamic investigation of a hydrogen enriched natural gas fueled HCCI engine for the efficient production of power, heating, and cooling', *Int. J. Hydrogen Energy*, Vol. 82, pp.111–122.
- Amiri Rad, E. and Mohammadi, S. (2018) 'Energetic and exergetic optimized Rankine cycle for waste heat recovery in a cement factory', *Appl. Therm. Engg.*, Vol. 132, pp.410–422.
- Bejan, A. and Tsatsaronis, G. (1996) *Thermal Design and Optimization*, John Wiley & Sons, Hoboken, New Jersey, USA.
- Dincer, I. and Rosen, M.A. (2013) *Exergy: Energy, Environment and Sustainable Development*, 2nd ed., Elsevier Science, <https://doi.org/10.1016/C2010-0-68369-6>.
- Frias, J.M., Aceves, S.M. and Flowers, D.L. (2007) 'Improving ethanol life cycle energy efficiency by direct utilization of wet-ethanol in HCCI engines', *ASME, J. Energy Resour. Technol.*, Vol. 129, No. 4, pp.332–337.
- Gaíne, B., Yan, Z., Gandolfi, J. and Lawler, B. (2023) 'Methanol and wet ethanol as interchangeable fuels for internal combustion engines: LCA, TEA, and experimental comparison', *Fuel*, Vol. 333, No. Part 1, p.126257, <https://doi.org/10.1016/j.fuel.2022.126257>.
- Huang, W., Wang, J., Xia, J., Zhao, P. and Dai, Y. (2019) 'Performance analysis and optimization of a combined cooling and power system using low boiling point working fluid driven by engine waste heat', *Energy Convers Manage*, Vol. 180, pp.962–976.
- Júnior, E.P.B., Arrieta, M.D.P., Arrieta, F.R.P., Horizonte, I.B. and Silva, C.H.F. (2019) 'Assessment of a kalina cycle for waste heat recovery in the cement industry', *Appl. Therm. Eng.*, Vol. 147, pp.421–437, <https://doi.org/10.1016/j.applthermaleng.2018.10.088>.
- Khaliq, A., Trivedi, S.K. and Dincer, I. (2011) 'Investigation of a wet ethanol fuelled HCCI engine based on first and second law analysis', *Appl. Therm. Eng.*, Vol. 31, No. 10, pp.1621–1629.
- Kotas, T.J. (1995) *The Exergy Method of Thermal Plant Analysis*, Anchor Brendon Ltd., Great Britain.
- Meftahpour, H., Saray, R.K., Aghaei, A.T. and Bahlouli, K. (2024) 'Comprehensive analysis of energy, exergy, economic, and environmental aspects in implementing the Kalina cycle for waste heat recovery from a gas turbine cycle coupled with a steam generator', *Energy*, Vol. 290, p.130094, <https://doi.org/10.1016/j.energy.2023.130094>.
- Shokati, N., Ranjbar, F. and Yari, M. (2015) 'Exergoeconomic analysis and optimization of basic, dual-pressure and dual-fluid ORCs and Kalina geothermal power plants: a comparative study', *Renew Energy*, Vol. 83, pp.527–542, <https://doi.org/10.1016/j.renene.2015.04.069>.
- Siddiqui, M.A. (2023) 'Development and assessment of a novel natural gas fuelled HCCI engine based combined power, heating, and refrigeration system', *Energy*, Vol. 283, p.128994, <https://doi.org/10.1016/j.energy.2023.128994>.

- Siddiqui, M.A., Khaliq, A. and Kumar, R. (2022) 'Thermodynamic and comparative analysis of ejector refrigeration cycle and absorption refrigeration cycle integrated wet ethanol-fueled HCCI engine for cogeneration of power and cooling', *ASME. J. Thermal Sci. Eng. Appl.*, Vol. 14, No. 4, p.41003.
- Sun, D.W. (1998) 'Comparison of the performances of $\text{NH}_3\text{-H}_2\text{O}$, $\text{NH}_3\text{-LiNO}_3$ and $\text{NH}_3\text{-NaSCN}$ absorption refrigeration systems', *Energy Convers Mgmt.*, Vol. 39, Nos. 5–6, pp.357–368, [https://doi.org/10.1016/S0196-8904\(97\)00027-7](https://doi.org/10.1016/S0196-8904(97)00027-7).
- Sydney, E.B. et al. (2019) 'Current analysis and future perspective of reduction in worldwide greenhouse gases emissions by using first and second generation bioethanol in the transportation sector', *Bioresour. Technol. Rep.*, Vol. 7, p.100234.
- Turns, S.R. and Kraige, D.R. (2007) *Properties Tables Booklet for Thermal Fluids Engineering*, Cambridge University Press, New York.
- Wu, C., Xu, X., Li, Q., Li, J., Wang, S. and Liu, C. (2020) 'Proposal and assessment of a combined cooling and power system based on the regenerative supercritical carbon dioxide Brayton cycle integrated with an absorption refrigeration cycle for engine waste heat recovery', *Energy Convers Manage.*, Vol. 207, p.112527.
- Xu, Z.Y. and Wang, R.Z. (2018) 'Comparison of absorption refrigeration cycles for efficient air cooled solar cooling', *Solar Energy*, Vol. 172, Part 1, pp.14–23.